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TWELVE VOLUMES

CHICAGO

AMERICAN SCHOOL OF CORRESPONDENCE

1908

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Grateful acknowledgment is here made also for the invaluable co-operation of the foremost engineering firms, in making these volumes thoroughly representative of the best and latest practice in the design and construction of steam and electrical machines and machine tools; also for the valuable drawings and data, suggestions, criticisms, and other courtesies.

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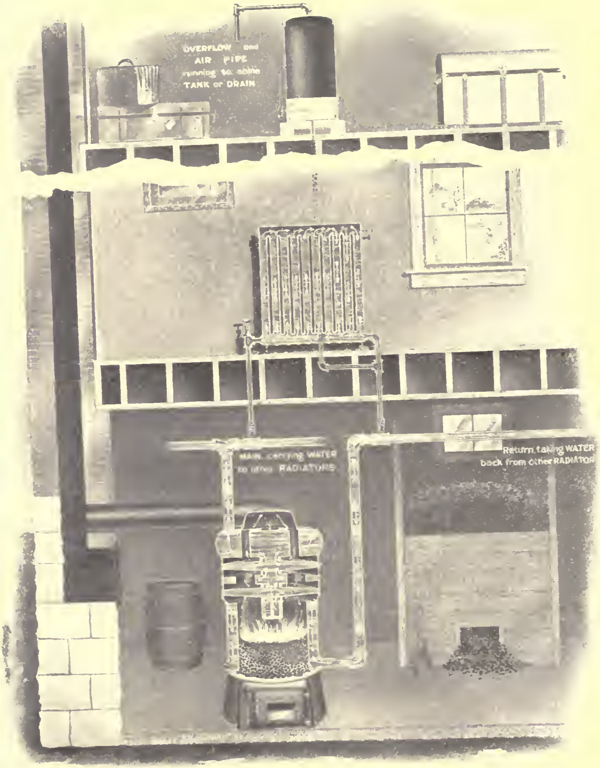

PRINCIPLE OF HOT WATER HEATING ILLUSTRATED BY TRANSVERSE SECTIONAL VIEW SHOWING BOILER, RADIATOR AND EXPANSION TANK.

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MACHINE DESIGN.

PART I.

Definition. Machine Design is the art of mechanical thought development, and specification.

It is an art, in that its routine processes can be analyzed and systematically applied. Proficiency in the art positively cannot be attained by any "short cut" method. There is nothing of a spectacular nature in the methods of Machine Design. Large results cannot be accomplished at a single bound, and success is possible only by a patient, step-by-step advance in accordance with well-established principles.

"Mechanical thought" means the thinking of things strictly from their mechanical side; a study of their mechanical theory, structure, production, and use; a consideration of their mechanical fitness as parts of a machine.

"Mechanical development" signifies the taking of an idea in the rough, in the crude form, for example, in which it comes from the inventor, working it out in detail, and refining and fixing it in shape by the designing process. Ideas in this way may become commercially practicable designs.

"Mechanical specification" implies the detailed description of designs, in such exact form that the shop workmen are enabled to construct completely and put in operation the machines represented in the designs.

The object of Machine Design is the creation of machinery for specific purposes. Every department of a manufacturing plant is a controlling factor in the design and production of the machines built there. A successful design cannot be out of harmony with the organized methods of production. Hence in the high development of the art of Machine Design is involved a knowledge of the operations in all the departments of a manufacturing plant. The student is therefore urged not only to familiarize himself with the direct production of machinery, but to study the relation thereto of the allied commercial departments

He should get into the spirit of business at the start, get into the shop atmosphere, execute his work just as though the resulting design were to be built and sold in competition. He should visit shops, work in them if possible, and observe details of design and methods of finishing machine parts. In this way he will begin to store up bits of information, practical and commercial, which will have valuable bearing on his engineering study.

The labor involved in the design of a complicated automatic machine is evidenced by the designer's wonderful familiarity with its every detail as he stands before the completed machine in operation and explains its movements to an observer. The intricate mass of levers, shafts, pulleys, gears, cams, clutches, etc., etc., packed into a small space, and confusing even to a mechanical mind, seems like a printed book to the designer of them.

This is so because it is a familiar journey for the designer's mind to run over a path which it has already traversed so many times that he can see every inch of it with his eyes shut. Every detail of that machine has been picked from a score or more of possible ideas. One by one, ideas have been worked out, laid aside, and others taken up. Little by little, the special fitness of certain devices has become established, but only by patient, careful consideration of others, which at first seemed equally good.

Every line, and corner, and surface of each piece, however small that piece may be, has been through the refining process of theoretical, practical, and commercial design. Every piece has been followed in the mind's eye of its designer from the crude material of which it is made, through the various processes of finishing, to its final location in the completed machine; thus its bodily existence there is but the realization of an old and familiar picture.

What wonder that the machine seems simple to the designer of it! As he looks back to the multitude of ideas invented, worked out, considered and discarded, the machine in its final form is but a trifle. It merely represents a survival of the fittest.

No successful machine, however simple, was ever designed that did not go through this slow process of evolution. No machine ever just simply happened by accident to do the work for which it is valued. No other principle upon which the successful design of machinery depends is so important as this careful, patient consideration of detail. A machine is seldom unsuccessful because some main point of construction is wrong. The principal features of a machine are usually the easiest to determine. It is a failure because some little detail was overlooked, or hastily considered, or allowed to be neglected, because of the irksome labor necessary to work it out properly.

There is no task so tedious, for example, as the devising of the method of lubricating the parts of a complicated machine. Yet there is no point of design so vital to its life and operation as an absolute assurance of an adequate supply of oil for the moving parts at all times and under all circumstances. Suitable means often cannot be found, after the parts are together, hence the machine goes into service on a risky basis, with the result, perhaps, of early failure, due to "running dry." Good designers will not permit a design to leave their hands which does not provide practically automatic oiling, or at least such means of lubrication that the operator can offer no excuse for neglecting to oil his machine. This is but a single illustration of many which might be presented to impress the definite and detail character necessary in work in Machine Design.

Relation. The relation which Machine Design should correctly bear to the problems that it seeks to solve, is twofold; and there are, likewise, two points of view corresponding to this twofold relation, from which a study of the subject should be traced. Neither of these can be discarded and an efficient mastery of the art attained. These points are—

I. Theory.

II. Production.

I. *Theory.* From this point of view, Machine Design is merely a skeleton or framework process, resulting in a representation of ideas of pure motion, fundamental shape, and ideal proportion. It implies a working knowledge of physical and mathematical laws. It is a strictly scientific solution of the problem at hand, and may be based purely on theory which has been reasoned out by calculation or deduced from experiment. This is the only sure foundation for intelligent design of any sort.

But it is not enough to view the subject from the standpoint of theory alone. If we stopped here we should have nothing but mechanisms, mere laboratory machines, simply structures of ingenuity and examples of fine mechanical skill. A machine may be correct in the theory of its motions; it may be correct in the theoretical proportions of its parts; it may even be correct in its operation for the time being; and yet its complication, its misdirected and wasteful effort, its lack of adjustment, its expensive and irregular construction, its lack of compactness, its difficulty of ready repair, its inability to hold its own in competition—any of these may throw the balance to the side of failure. Such a machine, commercially considered, is of little value. No shop will build it, no machinery house will sell it, nobody will buy it if it is put on the market.

Thus we see that, aside from the theoretical correctness of principle, the design of a machine must satisfy certain other exacting requirements of a distinctly business nature.

II. *Production.* From this point of view, Machine Design is the practical, marketable development of mechanical ideas. Viewed thus, the theoretical, skeleton design must be so clothed and shaped that its production may be cheap, involving simple and efficient processes of manufacture. It must be judged by the latest shop methods for exact and maximum output. It must possess all the good points of its competitor, and, withal, some novel and valuable ones of its own. In these days of keen competition it is only by carefully studied, well-directed effort toward rapid, efficient, and, therefore, cheap production that any machine can be brought to a commercial basis, no matter what its other merits may be. All this must be thought of and planned for in the design, and the final shapes arrived at are quite as much a result of this second point of view as of the first.

As a good illustration of this, may be cited the effect of the present somewhat remarkable development of the so-called "high speed" steels. The speeds and feeds possible with tools made of these steels are such that the driving power, gearing, and feed mechanism of the ordinary lathe are wholly inadequate to the demands made upon them when working the tool to its limit. This means that the basis of design as used for the ordinary tool steel will not do, if the machine is expected to stand up to the cuts possible with the new steels. Hence, while the old designs were right for the old standard, a new one has been set, and a thorough revision on a high-speed basis is imminent, else the market for them as machines of maximum output will be lost.

From these definitions it is evident that the designer must not only use all the theory at his command, but must continually inform himself on all processes and conditions of manufacture, and keep an eye on the tendency of the sales markets, both of raw material and the finished machinery product. This is what in the broadest sense is meant by the term "Mechanical Thought," thought which is directed and controlled, not only by theoretical principle but by closely observed practice. From the feeblest pretenders of design to those engineers who consummate the boldest feats and control the largest enterprises, the process which produces results is always the same. Although experience is necessary for the best mechanical judgment, yet the student must at least begin to cultivate good mechanical sense very early in his study of design.

Invention. Invention is closely related to Machine Design, but is not design itself. Whatever is invented has yet to be designed. An invention is of little value until it has been refined by the process of design.

Original design is of an inventive nature, but is not strictly invention. Invention is usually considered as the result of genius, and is announced in a flash of brilliancy. We see only the flash, but behind the flash is a long course of the most concentrated brain effort. Inventions are not spontaneous, are not thrown off like sparks from the blacksmith's anvil, but are the result of hard and applied thinking. This is worth noting carefully, for the same effort which produces original design may develop a valuable invention. But there is little possibility of inventing anything except through exhaustive analysis and a clear interpretation of such analysis.

Handbooks and Empirical Data. The subject matter in these is often contradictory in its nature, but valuable nevertheless. Empirical data are data for certain fixed conditions and are not general. Hence, when handbook data are applied to some specific case of design, while the information should be used in the freest manner, yet it must not be forgotten that the case at hand is probably different, in some degree, from that upon which the data were based, and unlike any other case which ever existed or will ever again exist. Therefore the data should be applied with the greatest discretion, and when so applied will contribute to the success of the design at least as a check, if not as a positive factor.

The student should at the outset purchase one good handbook, and acquire the habit of consulting it on all occasions, checking and comparing his own calculations and designs therefrom. Care must be taken not to become tied to a handbook to such an extent that one's own results are wholly subordinated to it. Independence in design must be cultivated, and the student should not sacrifice his calculated results until they can be shown to be false or based on false assumption. Originality and confidence in design will be the result if this course be honestly pursued.

Calculations, Notes, and Records. Accurate calculations are the basis of correct proportions of machine parts. There is a right way to make calculations and a wrong way, and the student will usually take the wrong way unless he is cautioned at the start.

The wrong way of making calculations is the loose and shiftless fashion of scratching upon a scrap of detached paper marks and figures, arranged in haphazard form, and disconnected and incomplete. These calculations are in a few moments' time totally meaningless, even to the author of them himself, and are so easily lost or mislaid that when wanted they usually cannot be found.

Engineering calculations should always be made systematically, neatly, and in perfectly legible form, in some permanently bound blank book, so that reference may always be had to them at any future time for the purpose of checking or reviewing. Put all the data down. Do not leave in doubt the exact conditions under which the calculations were made. Note the date of calculation.

If a mistake in figures is made, or a change is found necessary, never rub out the figures or tear out the leaf, or in any way obliterate the figures. Simply draw a bold cross through the wrong part and begin again. Often a calculation which is supposed to be wrong is later shown to be right, or the facts which caused the error may be needed for investigation and comparison. Time which is spent in making figures is always valuable time, time too precious to be thrown away by destroying the record.

The recording of calculations in a permanent form, as just described, is the general practice in all modern engineering offices. This plan has been established purely as a business policy. In case of error it locates responsibility and settles dispute. Consistent designing is made possible through the records of past designs. Proposals, estimates, and bids may often be made instantly, on the basis of what these record books show of sizes and weights. This bookkeeping of calculations is as important a factor of systematic engineering as bookkeeping of business accounts is of financial success.

The student should procure for this purpose a good blank book with a firm binding, size of page not smaller than 6 by 8 inches (perhaps 8 by 11 inches may be better), and every calculation, however small and apparently unimportant, should be made in it.

Sample pages of engineering calculations are reproduced in Figs. 3 to 9. Note the sketch showing the forces. Note the clear statement of data. Note the systematic writing of the equations, and the definite substitutions therein. Note the heavy double underscoring of the result, when obtained. There is nothing in the whole process of the calculation that cannot be reviewed at any moment by anybody, and in the briefest time.

The development of a personal note-book is of great value to the designer of machinery. The facts of observation and experience recorded in proper form, bearing the imprint of intimate personal contact with the points recorded, cannot be equalled in value by those of any hand or reference book made by another. There is always a flavor about a personal note-book, a sort of guarantee, which makes the use of it by its author definite and sure.

The habit of taking and recording notes, or even knowing what notes to take, is an art in itself, and the student should begin early to make his note-book. Aside from the value of the notes themselves as a part of his personal equipment, the facility with which his eye will be trained to see and record mechanical things will be of great value in all of his study and work. How many men go through a shop and really see nothing of the operations going on therein, or, seeing them, remember nothing! An engineer, trained in this respect, will to a surprising degree be able to retain and sketch little details which fall under his eye for a brief moment only, while he is passing through a crowded shop.

Some draftsmen have the habit of copying all the standard tables of the various offices in which they work. While these are of some value in a few cases, yet this is not what is meant by a good note-book in the best sense. Ideas make a good note-book, not a mere tabulation of figures. If the basis upon which standards are founded can be transferred to permanent personal record, or novel methods of calculation, or simple features of construction, or data of mechanical tests, or efficient arrangement of machinery—if *these* can be preserved for reference, the note-book will be of greatest value.

Whatever is noted down, make clear and intelligible, illustrating by a sketch if possible. Make the note so clear that reference to it after a long space of years would bring the whole subject before the mind in an instant. If this is not done the author of the note himself will not have patience to dig out the meaning when it is needed; and the note will be of no value.

METHOD OF DESIGN.

The fundamental lines of thought and action which every designer follows in the solution of any problem in any class of work whatsoever, are four in number. The expert may carry all these in mind at the same time, without definite separation into a a step-by-step process; but the student must master them in their proper sequence, and thoroughly understand their application. In these four are concentrated the entire art of Machine Design. When they have become so familiar as to be instinctively applied on any and all occasions, good design is the result. The only other quality which will facilitate still further the design of good machinery is experience; and that cannot be taught, it must be acquired by actual work.

1. Analysis of Conditions and Forces. First, take a good square look at the problem to be solved. Study it from all sides, view it in all lights, note the worst conditions which can possibly exist, note the average conditions of service, note any special or irregular service likely to be called for.

With these conditions well in mind, make a careful analysis of all the forces, maximum as well as average, which may be brought into play. Make a rough sketch of the piece under consideration, and put in these forces. Be sure that these forces are at least approximately right. Go over the analysis carefully again and again. Remember that time saved at the beginning by hasty and poor analysis will actually be time lost at the end; and if the machine actually fails from this reason, heavy financial loss in material and labor will occur. Any haste toward completion of the structure beyond the roughest outline, without this careful study of forces, is a blind leap in the dark, entirely unscientific, and almost certain to result in ultimate failure.

On the other hand this principle may be carried too far. In trying to make the analysis thorough and the forces accurate, it is quite possible to consume more than a reasonable amount of time. Again, it is not always easy, and frequently impossible, to determine exactly the forces acting on a given piece. But their *nature*, whether sudden or slowly applied, rapid in action or only occurring at intervals, and their *approximate* direction and magnitude at least, are always capable of analysis. There are few, if any, cases where close assumptions cannot be made on the above basis and the design proceeded with accordingly. Hence the danger of too great refinement of analysis is simply to be avoided by the designer's plain business sense.

The first tendency of the student is to pass over the study of the forces as dull and dry, and attempt the design at once. He soon finds himself facing problems of which he sees no possible solution, and he bases his design on pure guess-work. This is the only solution possible from such a point of view, and is really no solution at all. A guess which has some rational backing is often successful; but in that case some analysis is required, and it is not a pure guess, but falls under the very principle we are considering.

There is no short cut to the design of machine parts which avoids this full understanding of the forces that they must sustain. The size of a belt depends upon the maximum pull upon it, and the designing of belts is nothing but providing sufficient cross-section of leather to prevent the belt tearing under the pull. Again, if pulley arms are not to break, or shafts twist off, or bolts be torn apart, or the teeth of gears fail, or keys and pins shear off, we must first, of course, find out what forces exist which are likely to produce stress that may lead to such breakage. We should not guess at the sizes, and then run the machine to see if breakage results, and then guess again. Machines are sometimes built in this way, but it is an unreasonable and uncertain method. We must use every effort to foresee the stress which a piece is liable to receive, before we decide its size. We must know all the forces approximately, if not positively. The analysis must be thorough enough to permit of reasonable assumption, if not positive assertion. It is manifestly impossible to solve any problem until we know exactly what the problem is; and a full analysis *is* the statement of the problem.

2. **Theoretical Design.** After we know by careful analysis what stress the machine part has to sustain, the next step is so to design it that it will theoretically resist the applied forces with the least expenditure of material.

We often see machinery with the metal of which it is made distributed in the worst possible manner. In places where the stress is heavy and a rigid member is needed, we find a weak, springy part; while in other parts, where there are no forces to be resisted, or vibration to be absorbed, there seems to be a waste of good material. Whether in such case the analysis of the forces was poor, or perhaps not made at all, or whether a knowledge of how to design so as to resist the given forces was wholly absent, cannot be told. At any rate, lack of either or both is clearly shown in the result.

Any member of a machine may vary in form from a solid block or chunk of material to an open ribbed structure. The solid chunk fills the requirement as far as strength is concerned, unless it is so heavy as to fail from its own weight. But such construction is poor design, except in cases where the concentration of heavy mass is necessary to absorb repeated blows like those of a hammer. The possibility of these blows should, however, have been determined in the analysis; and the solid, anvil construction then becomes theoretical design for that analysis.

For steadily applied loads an open, ribbed, or hollow box structure can be made which will distribute the metal where it is theoretically needed, and each fiber will then sustain its proper share of the load. In this way weight, cost, and appearance are heeded; and the service of the piece is as good as, and probably better than, it would be with the clumsy, solid form.

There is no such thing as putting too much theory into the design of machinery. The strongest trait which an engineer can have is absolute faith in his analysis and calculations, and their reproduction in his theoretical design. Theoretical design is an indication of scientific advance in the art, and some of the greatest steps of progress which have been made in recent years have been accomplished through a purely theoretical study of machine structure.

It will never do, however, to be satisfied with theoretical design when it is not in accord with modern commercial and manufacturing considerations. Hence the next step after the determination of the theoretical design is the study of it from the producing standpoint.

3. Practical Modification. All theoretical design viewed from the business standpoint is worthless, unless it has been subjected to the test of cheap and efficient production. Each machine detail, though correct in theory, may yet be improperly shaped and unfit for the part it is to play in the general scheme of manufacture.

The conditions here involved are changeable. What is good design in this decade may be bad in the next. In this light the designer must be a close student of the signs of the times; he must follow the march of progress, closely applying existing resources, conditions, and facilities, otherwise he cannot produce up-to-date designs. The introduction of new raw materials, the cheapening of production of others, the changing of shop methods, the use of special machinery, the opening of new markets, the development of new motive agents,—all these and many others are constantly demanding some modification in design to meet competition.

Illustrative of this, note the change which has been wrought by the development of electric power, the rise and decline of the bicycle business, the present manufacture of automobiles, the last named especially with reference to the development of the small motive unit, the gasolene engine, the steam engine, etc. The design of much machinery has been materially changed to meet the exacting demands of these new enterprises.

Practical modifications of design necessary to meet the limitations of construction in the pattern shop, foundry, and machine shop are of daily application in the designer's work. He must keep in his mind's eye at all times the workmen and the processes they use to create his designs in metal in the shop.

"How can this be made?" "Can it be made at all?" "Can it be made cheaply?" "Will it be simple in operation after it is made?" "Can it be readily removed for repair?" "Can it be lubricated?" "How can it be put in place?" "How can it be gotten out?" "Will it be made in small quantities or large?" "Will it sell as a special or standard machine?" etc., etc.

The consideration of such questions as these is a practical necessity as a business matter. No other feature affects the design of machinery more, perhaps; for designs which cannot be built as business propositions are no designs at all.

The student, it is true, may not have the extended shop knowledge which is essential to this; but he can do much for himself by visiting shops whenever possible, getting hold of shop ways of doing things, and invariably treating his work as a business matter. Though a man may not be a pattern maker, molder, blacksmith, or machinist, yet he can soon gain ideas of the processes in each of these branches which will be of immense advantage to him in his designing work.

4. Delineation and Specification. This means the clear and concise representation of the design by mechanical drawings.

This is as much a part of the routine method of Machine Design as the other three points which have been discussed. The mere act of putting the results of mechanical thinking on paper is one of the greatest helps to force thinking machinery to systematic and definite action. A designer never thinks very long without drawing something, and the student must bring himself to feel that a drawing in its first sense is a means of helping his own thought, and must freely use it as such.

In its second and final sense, the drawing is an order and specification sheet from the designer to the workman. Design which stops short of exact, finished delineation in the form of working shop drawings is only half done. In fact the possibility of a piece being thus exactly drawn is often the crucial test of its feasibility as a part of a machine. It is easy to make general outlines, but it is not so easy to get down to finished detail. It is safe to say that there is no one thing productive of more trouble, delay and embarrassment, and waste of time and money in the shop, when there need be none from this cause, than a poor detail drawing. The efficiency of the process of design is not fully realized, and failures are often recorded where there should be success, merely because the indefiniteness permitted by the designer in the drawings naturally transmitted itself to the workman, and he in turn produced a part indefinite in form and operation.

The actual process of drawing in the development of a design may be outlined as follows:

Rough sketches merely representing ideas, not drawn to scale, are first made. These are of use only so far as the choice of mechanical ideas is concerned, and to carry preliminary dimensions.

Following these sketches, comes a layout to scale, of the favored sketch, a working out of the relative sizes and location of the parts. This drawing may be of a sketchy nature, carrying a principal dimension here and there to fix and control the detailed design. In this drawing the design is developed and general detail worked out. The minute detail of the individual parts is, however, left to the subsequent working drawing.

This layout drawing may now be turned over to an expert draftsman or detail designer, who picks out each part, makes an exact drawing of it, studying every little detail of its shape, and finally adds complete dimensions and specifications so that the workman is positively informed as to every point of its construction.

General drawings and cross sections constitute the last step in the process of complete delineation. These show the parts assembled in the complete machine. They also serve a valuable purpose to the draftsman in checking up the dimensions of the detail drawings. Errors which have escaped previous notice are often discovered in this way. The layout, mentioned above, is sometimes finished up into a general drawing; but it is safer to make an entirely new drawing, as changes in detail are often necessary after the layout is made.

The four fundamental lines of thought and action noted above may be summarized thus—"**analyze and theorize, modify and delineate.**" This is a maxim easy to remember, applicable to every problem in Machine Design, and always provides the answer to the question "What shall I do, how shall I proceed?" by pointing out the proper sequence in the course to be followed.

CONSTRUCTIVE MECHANICS.

Mechanics is a constructive science, its principles lying at the root of the design and operation of all machinery. It is usually taught, however, as an advanced mathematical subject; and the student gets his original conceptions of forces, moments, and beams in the abstract, before he realizes the constructive value of such conceptions. By "Constructive Mechanics" is meant the study of a machine purely from its constructive side, the viewing of the parts with respect to their "mechanics," and satisfying the requirements of the same in form and arrangement.

The student may cultivate this habit of clear, mechanical perception by constantly noting the "mechanics" of the simple structures which he sees in his daily routine of work. Aside from machinery, in which the "mechanics" is often obscure, the world is full of simple examples of natural strength and symmetry, explainable by application of the principles of pure "mechanics."

Posts and pillars are largest at their bases; overhanging brackets or arms are spread out at the fastening to the wall; heavy swinging gates are counter-balanced by a ponderous weight; the old-fashioned well sweep carries its tray of stones at the end, adjusting the balance to a nicety; these are examples of things depending for their form and operation upon the principles of "mechanics." The building of them involved "constructive mechanics," and yet their constructor perhaps never heard of the science, using merely his natural sense of mechanical fitness Such simple reasoning is, however, Constructive Mechanics.

Forces, Moments, and Beams. Machines are nothing but a collection of (1) parts taking direct stress, or (2) parts acting as loaded beams. Forces acting *without* leverage produce direct stress on the sustaining part. Forces acting *with* leverage produce a moment; the sustaining member is a beam, and the stress therein depends on the theory of beams, as explained in "Mechanics."

An example of the first is the load on a rope, the force acting without leverage, and the rope therefore having a direct stress put upon it.

An example of the second is a push of the hand on the crank of a grindstone. A moment is produced about the hub of the crank; the arm of the crank is a beam, and the stress at any point of it may be found by the method of theory of beams.

Tension, Compression, and Torsion. The stress induced in the sustaining part, whether tensile, compressive, or torsional, is caused by the application of forces, either acting directly without leverage, or with leverage in the production of moments.

The forces applied from external sources are at constant war with the resisting forces due to the strength of the fibres of the material composing the machine members. The moments of the *external* forces are constantly exerted against and balanced by the moments of the *internal* resistance of the material. Hence, design, from a strength standpoint, is merely a balancing of internal strength against external force. In other words, we may in all cases write a sign of equality, place the applied effort on one side, the effective resistance on the other, and we shall have an equation, which, if capable of solution, will give the proper proportions of the parts considered.

External Force = Internal Resistance.

External Moment = Internal Moment of Resistance.

Expressed in terms of the "Mechanics:"

$$P = AS \qquad (1)$$

$$B \text{ or } T = \frac{SI}{c} \qquad (2)$$

In these formulas, which are perfectly general,

P = direct load in pounds.

A = area of effective material, in square inches.

S = working fibre stress of the material (tensile, compressive, or shearing), in pounds per square inch.

B or T = external moment (bending or torsional), in inch-pounds.

I = moment of inertia (direct or polar), of the resisting section.

c = distance of the most remote fibre of the resisting section from the neutral axis.

P may produce direct tensile, compressive, or shearing stress.

B may produce tensile or compressive stress, and requires use of direct moment of inertia in either case.

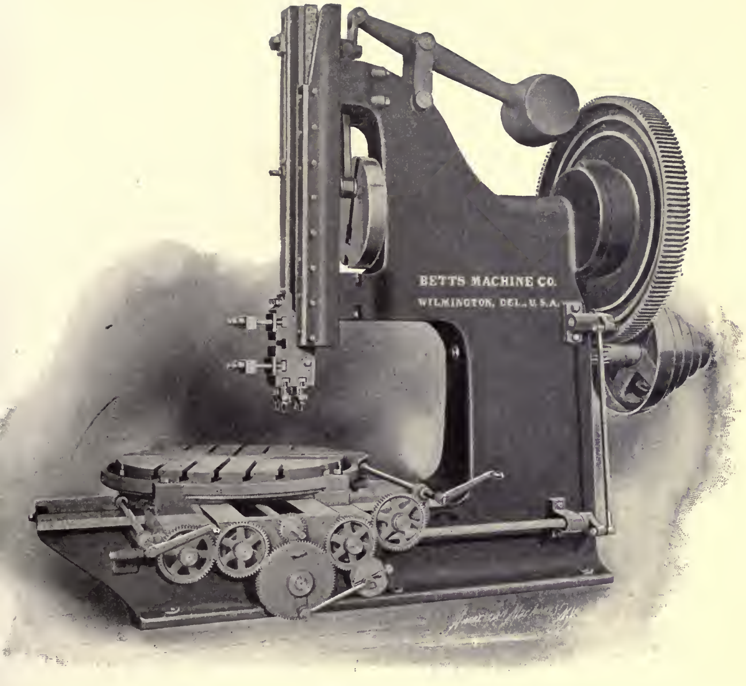
T produces shearing stress, and requires use of polar moment of inertia.

The origin of formula (1) is obvious, the assumption being that the fibre stress is equally distributed to every particle in the area "A."

The development of formula (2) is given in any text-book in Mechanics. It requires the aid of the Calculus, however. Any good handbook gives values for both the direct moment of inertia and the polar moment of inertia for quite a large variety of sections, so that further reference is an easy matter for the student. These values are also obtained through the methods of the Calculus.

The reason for introducing these formulas at this time is to call the attention of the student especially to the fact of their universal and fundamental use in all problems concerning the strength of machine parts. Nearly every computation may be reduced to or expanded from these two simple equations. Many complex combinations occur, of course, which will not permit simple and direct application of these formulas, but the student will do well to place himself in perfect command of these two. Assuming that he is able to analyze forces, and compute the simple moment at the point where he wishes to find the strength of section, the rest is the mere insertion of the assumed working fibre stress of the material in the formula (2) above, and solution for the quantity desired.

When the case is one of combined stress, the relation becomes more complicated and difficult of analysis and solution. The most common case is where bending is combined with torsion, as in the case of a shaft transmitting power, and at the same time loaded transversely between bearings. In fact there are very few cases of shafts in machines, which, at some part of their length, do not have this combined stress. In this case the method of procedure is to find the simple bending moment and the simple torsional moment separately, in the ordinary way. Then the theory of elasticity furnishes us with a formula for an equivalent bending or an equivalent torsional moment which is supposed to produce the same effect upon the fibres of the material as the combined



21-INCH SLOTTING MACHINE.

action of the two simple moments acting together. In other words, the separate moments combined in action, being impossible of solution in that form, are reduced to an equivalent simple moment and the solution then becomes the same as for the previous case.

These equivalent equations are given below, the subscript "e" being added to express separation from the simple moment:

$$B_e=\frac{B}{2}+\tfrac{1}{2}\sqrt{B^2+T^2} \qquad (3)$$

$$T_e=B+\sqrt{B^2+T^2} \qquad (4)$$

B_e and T_e, found from these equations, are the external moments, and are to be equated to the internal moments of resistance of the section precisely as if they were simple bending or torsional moments. Either may be used. For shafts (4) is generally used, being the simpler of the two in form.

FRICTION AND LUBRICATION.

The parts of a machine which have no relative motion with regard to each other are not dependent upon lubrication of their surfaces for the proper performance of their functions. In cases where relative motion does occur, as between a planer bed and its ways, a shaft and its bearing, or a driving screw and its nut, friction, and consequent resistance to motion, will inevitably occur. Heat will be generated, and cutting or scoring of the surfaces will take place if the surfaces are allowed to run together dry.

This difficulty, which exists with all materials, cannot be overcome, for it is a result of roughness of surface, characteristic of the material even when highly finished. The problem of the designer, then, is to take conditions as he finds them, and, as he cannot change the physical characteristics of materials, so choose those which are to rub together in the operation of the machine that friction will be reduced to the lowest possible limit. Now it fortunately happens that there are certain agents like oil and graphite, which seem to fill up the hollows in the surface of a solid material, and which themselves have very little friction on other substances. Hence, if a machine permits by its design an automatic supply of these lubricating agents to all surfaces having motion between them, friction may be reduced to the lowest limit.

If this full supply of lubricant be secured, and the parts still heat and cut, then the fault may be traced to other causes, such as springy surfaces, localization of pressure, or insufficient radiating surface to carry away the heat of friction as fast as it is generated.

Lubricating agents are of a nature running from the solid graphite form to a thick grease, then to a heavy dark oil, and finally to a thin, fluid oil flowing as freely as water. The solid and heavy lubricants are applicable to heavily loaded places where the pressure would squeeze out the lighter oils. Grease, forced between the surfaces by compression grease cups, is an admirable lubricator for heavy machinery under severe service. High-speed and accurate machinery, lightly loaded, requires a thin oil, as the fits would not allow room for the heavier lubricants to find their way to the desired spot. The ideal condition in any case is to have a film of lubricant always between the surfaces in contact, and it is this condition at which the designer is always aiming in his lubricating devices.

Oil ways and channels should be direct, ample in size, readily accessible for cleaning, and distributing the oil by natural flow over the full extent of the surface. Hidden and remote bearings must be reached by pipes, the mouths of which should be clearly indicated and accessible to the operator of the machine. Such pipes must be straight, if possible, and readily cleaned.

There is one practical principle affecting the design of methods of lubrication of a machine which should be borne in mind. This is, "Neglect and carelessness by the operator *must* be provided for." It is of no use to say that the ruination of a surface or hidden bearing is due to neglect by the operator, if the means for such lubrication are not perfectly obvious. This is "locking the door after the horse is stolen." The designer has not done his duty until he has made the scheme of lubrication so plain that every part *must* receive its proper supply of oil, except by gross and willful negligence, for which there can be no possible just excuse.

WORKING STRESSES AND STRAINS.

Some persons object to the use of these terms, as one is frequently used for the other, and misunderstanding results. This is doubtless true; but the student may as well learn the true relation of the terms once for all, because he will frequently run across them in his reading and reference work, and should interpret them rightly. The strict relation of the two is as follows:

Stress is the internal force in a piece resisting the external force applied to it. A weight of ten pounds hanging on a rope produces a *stress* of ten pounds in the rope.

Strain is the change of shape, or deformation, in a piece resisting an external force applied to it. If the above weight of ten pounds stretches the rope $\frac{1}{4}$ inch, the *strain* is $\frac{1}{4}$ inch.

Unit stress is stress per unit area, e. g., per square inch.

Unit strain is strain per unit length, e. g., per inch length.

In the above case, if the rope were $\frac{1}{2}$ square inch in area and 30 inches long, the unit stress, or intensity of stress, is $10 \div \frac{1}{2} = 20$ pounds per square inch; the unit strain is $\frac{1}{4} \div 30 = \frac{1}{120}$ inch per inch.

When stress is induced in a piece, the strain is practically proportional to the stress for all values of the stress below the elastic limit of the material; and when the external load is removed the strain will entirely disappear, or the recovering power of the material will restore the piece to the original length.

Illustrating by the case above, on the supposition that the elastic limit has not been reached by the stress of 20 pounds per square inch, if the load of 10 pounds were taken off, the $\frac{1}{4}$-inch strain would disappear and the rope return to its original length; if the load were changed to $\frac{1}{2}$ of 10 pounds, or 5 pounds, the strain would be $\frac{1}{2}$ of $\frac{1}{4}$ inch, or $\frac{1}{8}$ inch.

Now it is found that if we wish a piece to last in service for a long time without danger of breakage, we must not permit it to be stressed anywhere near the elastic limit value. If we do, although it will probably not break at once, it is in a dangerous condition, and not well suited to its requirements as a machine member. The technical name for this weakening effect is "fatigue." It is further found that the fatigue due to this repeated stress is reached at a lower limit when the stress is alternating in character than when it is not. In other words, if we first pull on a piece and then push on it, we shall first have the piece in tension and then in compression; this alternation of stress repeated to near the elastic limit of the material will fatigue it, or wear out the fibres, and it will finally fail. If, however, we first pull on the piece with the same force as before, and then let go, we shall first have the piece in tension and then entirely relieved; such repetition of stress will finally "fatigue" the material, but not so quickly as in the first case. Experiments indicate that it may take twice as many applications in the latter case as in the former.

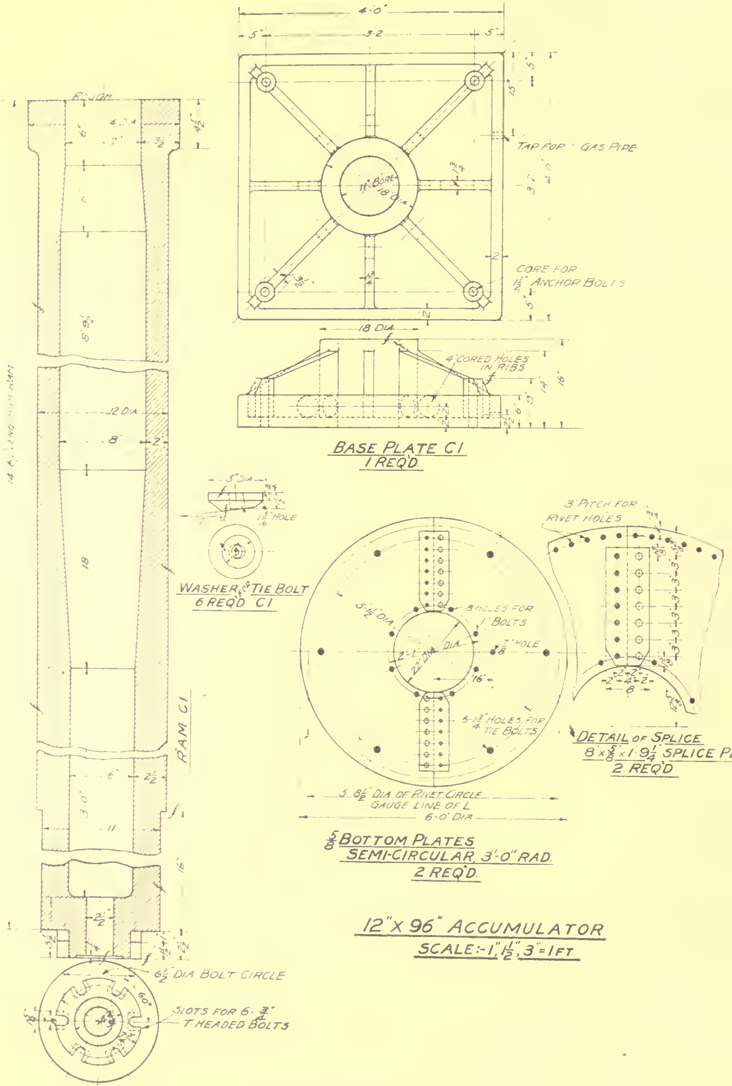
The working stress of materials permissible in machines is based on the above facts. The breaking strength divided by a liberal factor of safety will not necessarily give a desirable working stress. The question to be answered is, "Will the assumed working fibre stress permit an indefinite number of applications of the load without fatiguing the material?"

Hence we see that the same material may be safely used under different assumptions of working stress. For example, a rotating shaft, heavily loaded between bearings, acts as a beam which in each revolution is having its particles subjected, first to a maximum tensile stress, and then to a maximum compressive stress. This is obviously a very different stress from that which the same piece would receive if it were a pin in a bridge truss. In the former we have a case where the stress on each particle reverses at each revolution, while in the latter we have merely the same stress recurring at intervals, but never becoming of the opposite character. For ordinary steel, a value of 8,000 would be reasonable in the former case, while in the latter it may be much higher with safety, perhaps nearly double.

From the facts stated above, it is evident that exact values for working fibre stress cannot be assumed with certainty and applied broadly in all cases. If the elastic limit of the material is definitely known we can base our working value quite surely on that.

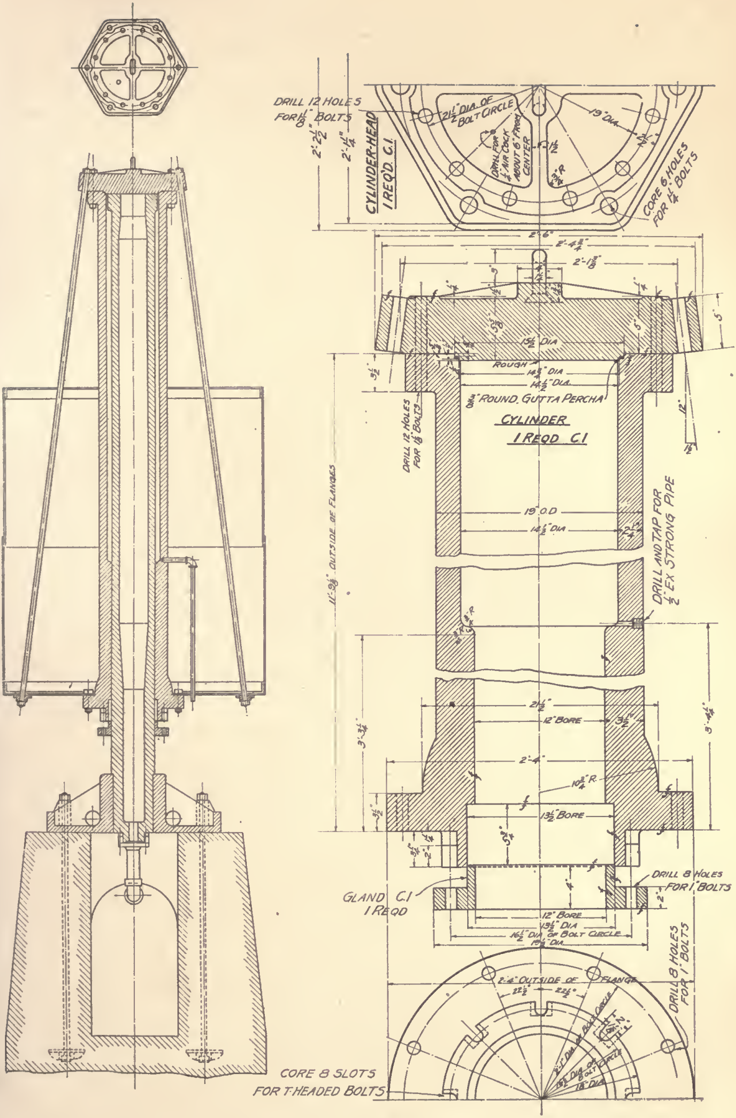
With but a general knowledge of the elastic limit, ordinary steel is good for from 12,000 to 15,000 pounds per square inch non-reversing stress, and 8,000 to 10,000 reversing stress. Cast iron is such an uncertain metal on account of its variable structure that stresses are always kept low, say from 3,000 to 4,000 for non-reversing stress, and 1,500 to 2,500 for reversing stress.

With these values as a guide, and the special conditions controlling each case carefully studied, reasonable limits may be



HYDRAULIC ACCUMULATOR

Detail Drawing, to be used with Figure on Opposite Page



HYDRAULIC ACCUMULATOR

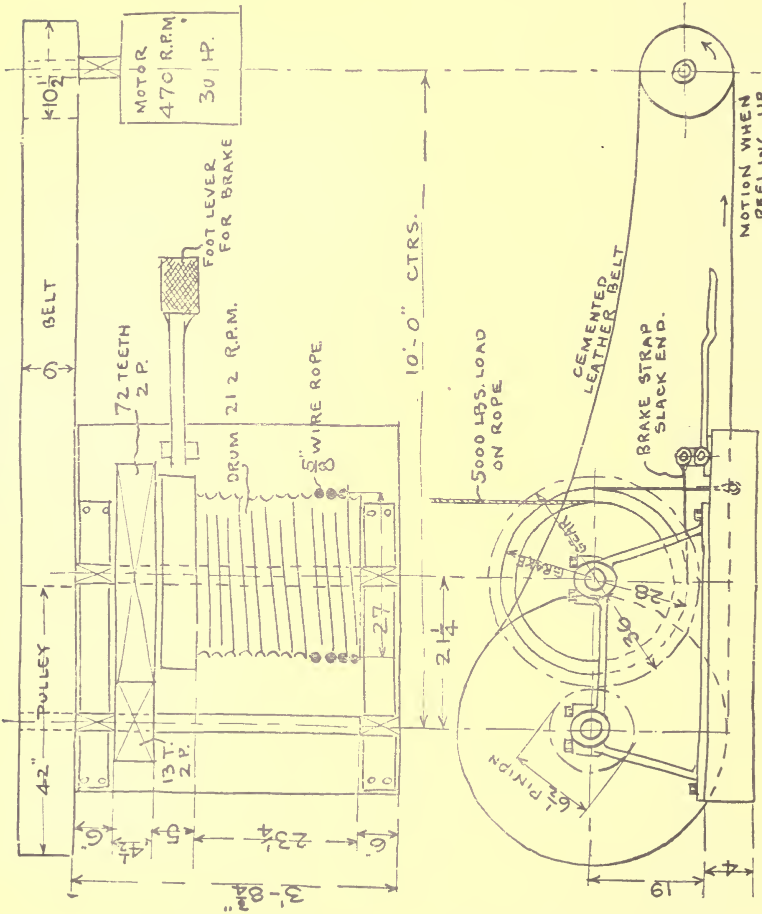
Assembled Drawing with Details on the Same Sheet

assigned for working stress, not only of steels, various grades of cast iron, and mixtures of the same, but of other alloys, brass, bronze, etc. Gun metal, semi-steel, and bronze are intermediate in strength between cast iron and steel. Data on the strength of materials are available in any of the handbooks, and should be consulted freely by the student. They will be found somewhat conflicting, but will assist the judgment in coming to a conclusion.

Application to Practical Case. In actual practice the only information which the designer has, upon which to base his design, is the object to be accomplished. He must choose or originate suitable devices, develop the arrangement of the parts, make his own assumptions regarding the operation of the machine, then *Analyze and Theorize, Modify and Delineate* each detail as he meets it.

This, it will be found, is a very different matter from taking some familiar piece of machinery, such as a pulley, or a shaft, or a gear, as an isolated case, the load being definitely given, and proceeding with the design. This is easily done, but is only half the problem, for machine parts, such as pulleys, gears, and shafts, do not confront the designer tagged or labeled with the conditions they are to meet. He is to provide parts to meet the specific conditions, and it is as much a part of his designing method to know how to attack the design of a machine as it is to know how to design the parts in detail after the attack has reduced the members to definitely loaded structures. The whole process must be gone through, the preliminary sketches, calculations, and layout, all of which precede the detail design and working drawings; and no step of the process can be omitted.

It is for this reason that the present case used for illustration is carried out quite thoroughly. The student should make himself familiar with every step of the designing method as applied to this simple case of design. More complex problems, handled in the same way, will simplify themselves; and when the point is reached where confidence exists to take hold of the design of any machine, however unfamiliar its object may be, or however involved its probable detail appears, the student has become the true designer. It is the knowing how to attack a problem, to start definite work on it, to go ahead boldly, confident that the method applied will produce results, that gives command of the design of machinery and wins engineering success.



The special case which has been chosen to illustrate the application of the principles stated in the foregoing pages is ideal, in that it does not represent any actual machine at present in operation. Probably builders of hoisting machinery have devices which would improve the machine as shown. In detail, as well as arrangement, they could doubtless make criticism as manufacturers. The arrangement as shown is merely intended to bring out in simplest form the common elements of transmission machinery as parts of some definite machine, instead of as isolated details. The design is one entirely possible, practical, and mechanical, but special attention has been paid to simplicity in order to enable the student to follow the method closely, for the *method* is the chief thing for him to acquire.

The student is expected to refer constantly to Part II for a more formal and general discussion of the simple machine elements involved in the case considered. Part II is intended to be a simplified and condensed reference book, carried out in accordance with the method of machine design as specified in Part I. The student should not wait until he has completed the study of this part before taking up Part II, for the latter is intended for use with the former in the solution of the problems.

In the case of power transmission about to be studied, the running, conversational method employed assumes that the student is in possession of the matter in Part II on the subject considered. Thus, in the design of the pulley, reference to the subject of "Pulleys" in Part II is necessary to follow the train of calculation; in designing the gear, consult "Gears;" in calculating size of shafts, see "Shafts," etc., etc.

Problem. A machine is to be designed to be set on the floor of a building to drive a wire rope falling from the overhead sheaves of an elevator or hoist. Without regard to details of this overhead arrangement, for its design would be a separate problem, suppose that the data for the rope are as follows:

Load on rope	5,000 pounds.
Speed of rope	150 feet per minute.
Length of rope to be reeled in	200 feet.

We shall further assume that the driving power is to be an electric motor belted to the machine, that the required speed reduction can be satisfactorily obtained by a single pair of pulleys and one pair of gears, and that a plain band brake is to be applied to the drum.

With this data we shall proceed to work out the detail design of the machine.

Preliminary Sketch. The first thing to do is to sketch roughly the proposed arrangement of the machine.

This might appear like Fig. 1 except that it would have no dimensions in addition to the data given above. If the scheme seems suitable, the next step is to make such preliminary calculations as will give further data, exact or closely approximate sizes, to be put at once on the sketch, to outline the future design.

Rope and Drum. Referring to tables of strength of wire rope (Kent's Pocket Book gives the manufacturers' list), we find that a $\frac{5}{8}$-inch cast-steel rope will carry 5,000 pounds safely, and that the proper size of drum to avoid excessive bending of the rope around it is 27 inches diameter.

Allowing $\frac{1}{8}$ inch between the coils as the rope winds on the drum, the pitch of coil will be $\frac{3}{4}$ inch as shown in sketch, Fig. 2. The length of one complete coil is, practically, $\frac{27\times 3.1416}{12}$ $=7.07$ feet. To provide for 200 feet will require $\frac{200}{7.07}=28+$ coils.

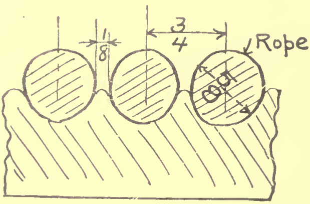


Fig. 2.

To be safe, let us provide for 30 coils, for which a length of drum $(30\times\frac{3}{4})+\frac{3}{4}=23\frac{1}{4}$ inches is required.

The space for brake strap may be assumed at 5 inches, and the thickness to provide necessary strength determined later in the design. The frictional surface of the strap may be of basswood blocks, say $1\frac{1}{4}$ inches thick, screwed to the metal band. The diameter of brake surface may be 28 inches.

Driving Gears. The size of drum gear evidently depends upon the method of fastening to the drum, and, other things being equal, should be kept as small as possible. One way would be to key the gear on the outside of the drum, another to bolt the gear to the end ot the drum. The latter has the advantage that a standard gear pattern can be used with the slight change of addition of bolt flange on the arms. This makes a simple, direct, and strong drive, the bolts being in shear.

Sketching this arrangement as the preferred one (Fig. 2A), it is evident that the diameter of the gear should be at least as large as the drum in order to keep the tooth load down to a reasonable figure. On the other hand, if made too large, it spreads out the machine and destroys its compactness. As a diameter of 36 inches is not excessive, let us assume this, and see if a desirable proportion of gear tooth can be found to carry the load.

For a pitch diameter of 36 inches there will be a theoretical load of $\frac{5{,}000\times 27}{36}=3{,}750$ pounds at the pitch line. But the load

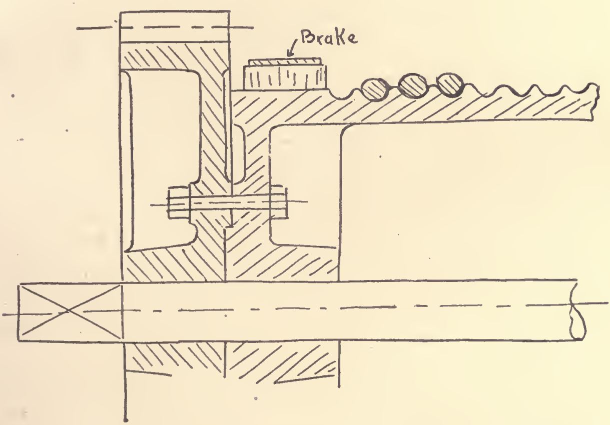


Fig. 2A.

on the tooth must not only impart a pull of 5,000 pounds to the rope, but must overcome friction between the gear teeth in action, also between the drum shaft and its bearings. Assuming the efficiency between the rope and tooth load to be 95 per cent, the net load, therefore, which the tooth must take is $\frac{3{,}750}{.95}=3{,}947$, say 4,000 pounds.

Assuming involute teeth, and applying the "Lewis" formula, (Part II, "Gears"):

$W = s \times p \times f \times y$	$W = 4{,}000$ $s = 6{,}000$
$4{,}000 = 6{,}000 \times p \times f \times .116$	$y = .116$ (number of teeth assumed at 75)
$p \times f = \dfrac{4{,}000}{6{,}000 \times .116} = 5.7$ inches	p=circular pitch f=face of gear

Let $f = 3\,p$ (a reasonable proportion for machine-cut teeth).

Then $3 \times p^2 = 5.7$

$p^2 = 1.9$

$p = \sqrt{1.9} = 1.378$ inches

The diametral pitch corresponding to this is

$$\frac{3.1416}{1.378} = 2.28$$

which is just between the regular standard pitches, 2 and $2\frac{1}{2}$, for which stock cutters are made. To be safe, let us take the coarser pitch, which is 2. The circular pitch corresponding to this is $\dfrac{3.1416}{2} = 1.57$, and making the face about three times the circular pitch gives

$$3 \times 1.57 = 4.71, \text{ say } 4\tfrac{1}{2} \text{ inches.}$$

The number of teeth in the gear is then $36 \times 2 = 72$. Referring to the value assumed for the tooth factor in calculation above, it is seen that y was based on 75 as the number of teeth, which is near enough to 72 to avoid the necessity of further checking the result.

The pinion to mesh with this gear should be as small as possible in order to get a high-speed ratio between pinion shaft and drum, otherwise an excessive ratio will be required in the pulleys, making the large one of inconvenient size. Small pinions have the teeth badly undercut and therefore weak, 13 teeth being the lowest limit usually considered desirable, for that reason. Choosing that number, we have a pitch diameter of $\dfrac{13}{2} = 6.5$ in., which is probably ample to take the shaft and key, and still leave sufficient stock under the tooth for strength. If made of cast iron, however, the pinion teeth, on account of the low number, will be narrower at the root than those of the gear of 72 teeth. Yet it was upon the basis of the latter that the pitch was chosen, for it will be remembered that the value of y in the formula was taken at .116. Hence the pinion will be weaker than the gear unless we make it of stronger material than cast iron, of which the large gear is supposed to be made. Steel lends itself very readily to this requirement; and in practice, pinions of less than 20 teeth are usually made of this material, hence we shall specify the pinion to be of steel.

Pulleys. The question now is whether or not we can get a suitable ratio in the pulleys without making the large one of inconvenient size, or giving the motor too slow speed for an economical proportion.

Suppose we limit ourselves to a diameter of 42 inches for the large pulley, and try a ratio of 4 to 1; this will give a diameter for the small pulley of $\frac{42}{4}=10\frac{1}{2}$ inches. We shall then have

Total ratio between drum and motor........	$\frac{72}{13}\times 4=\frac{288}{13}=22.2$
Rev. per min. of drum to give 150 f. p. m. of rope	$\frac{150}{7.07}=21.2$
Rev. per min. of motor......................	$22.2\times 21.2=470$
Horse-power of motor at 80 per cent efficiency	$\frac{150\ \times 5{,}000}{33{,}000\times\ .80}=30$

A 30 H. P. motor running 470 r. p. m. would be classed as a slow speed motor and would be a heavier machine and cost more than one of higher speed. It will be noticed, however, that the diameter of the small pulley is already quite reduced, and it is hardly desirable to decrease it still further. Neither can we increase the large pulley, as we have already set the limit at 42 inches. Hence, for our present problem we cannot improve matters much without increasing the size of the large gear, which is undesirable, or putting in another pair of gears, which is contrary to the conditions of the problem. As such a motor is perfectly reasonable, we shall assume it to be chosen for the purpose.

In commercial practice it would be well to pick out some standard make of motor of the required horse-power, note the speed as specified by the makers, and then, if possible, suit the ratio in the machine to this speed. It is always best to use standard machinery, if possible, both from the standpoint of first cost, as well

<u>Width of belt:</u> <u>May 20, 1903</u>

$$lg\,\frac{T_n}{T_0} = 2.729\,\mu(1-z)n \qquad \mu = .3$$

$$T_n - T_0 = P \qquad n = .5$$

$$z = \frac{1}{9660}\,\frac{wV^2}{t} \qquad t = 400 \text{ lbs.} \qquad w = .036 \text{ lbs.}$$

$$P = \frac{13541}{21} = 644.8 \text{ lbs.}$$

$$V = \frac{470 \times \overset{.2618}{\cancel{3.1416}} \times 10.5}{\cancel{12}} = 1292 \text{ (say 1300)}$$

$$\therefore z = \frac{1}{966\cancel{0}} \times \frac{\overset{.009}{\cancel{.036}} \times 1690\cancel{000}}{\cancel{4}\cancel{0}\cancel{0}} = \frac{15.210}{966} = \underline{\underline{.015}}$$

(.015 small, can be disregarded)

$$lg\,\frac{T_n}{T_0} = 2.729 \times 3 \times 5 = 0.409 \text{ for which the natural number is } 2.56$$

$$\therefore \frac{T_n}{T_0} = 2.56 \qquad T_0 = \frac{T_n}{2.56}$$

$$T_n - T_0 = 645$$

$$T_n - \frac{T_n}{2.56} = \frac{1.56\,T_n}{2.56} = 645$$

$$\therefore T_n = \frac{645 \times 2.56}{1.56} = \underline{\underline{1059}}$$

$$T_0 = 1059 - 645 = \underline{\underline{414}}$$

$$T_n = b \times h \times t \qquad b = \text{belt width in.}$$

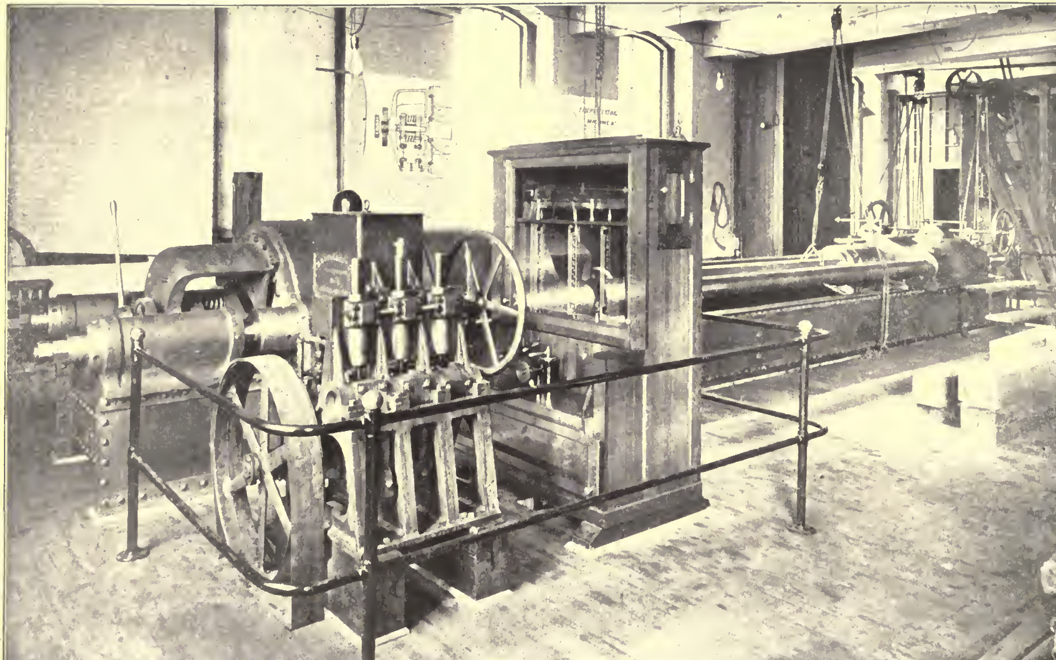
$$1059 = b \times .3 \times 400 \qquad h = \text{" thickness " (say .3)}$$

$$t = 400 \text{ (as above)}$$

$$b = \frac{1059}{.3 \times 400} = \underline{\underline{8.8}} \text{ (say 9" belt)}$$

(make pulley face $9\frac{1}{2}$")

Fig. 3



EMERY TESTING MACHINE.

Mechanical Engineering Department, Massachusetts Institute of Technology.

as ease of replacing worn parts. Machinery ordered special is expensive in first cost of designing, patterns, and tools, and extra spare parts for emergency orders are not often kept on hand.

Tabulation of Torsional Moments. For future reference, it is desirable at this point to tabulate the torsional moment, or torque, about each of the three shaft axes, assuming reasonable efficiencies for the various parts, as follows:

Efficiency between drum and gear tooth.............95 per cent
Efficiency between drum and pinion shaft...........90 per cent
Efficieney between drum and motor shaft ....80 per cent

TABLE OF TORSIONAL MOMENTS.

Axis.	Inch Lbs. Torque at 100 Per Cent Efficiency.	Inch Lbs. Torque, Efficiency as Above.
Drum	$5{,}000\times\frac{27}{2}$ ....$=67{,}500$	$\frac{67{,}500}{.95}=71{,}052$
Pinion......	$5{,}000\times\frac{27}{2}\times\frac{13}{72}$ ......$=12{,}187$	$\frac{12{,}187}{.90}=13{,}541$
Motor	$5{,}000\times\frac{27}{2}\times\frac{13}{72}\times\frac{10.5}{42}=3{,}047$	$\frac{3{,}047}{.80}=3{,}809$

This means that the motor develops a torque of 3,809 inch-pounds delivering to pinion shaft 13,541 inch-pounds, and to drum 71,052 inch-pounds.

Width of Belt. The page of calculation for belt width is reproduced in Fig. 3.

The calculation as given is strictly scientific, based on the working strength of a cemented joint ($t=400$ lbs. per square inch). This is a favorable situation for the use of a cemented joint, because it is easy to provide means of adjusting the belt tension by placing the motor on a sliding base. Otherwise a laced joint could be used, requiring relacing when the belt slackens through its stretch in service. Under the assumption that a double *laced* belt is used, the empirical formula below is one often applied:

$$\text{H. P.}=\frac{w\times V}{540}=\frac{w\times 1{,}300}{540}=30$$

This gives $w=\frac{540\times 30}{1{,}300}=12{\cdot}4$ inches (say 12 inches).

It should be remembered that this value is purely empirical; it applies to a *laced* joint, and could not be expected to check the value of 9 inches obtained by the first computation for a cemented joint. It is fairly in proportion. For the quite definite service required of the belt in the present case, the width of 9 inches is doubtless sufficient, considering the cemented joint.

Length of Bearings. Considerable latitude in choice of length of bearings is permissible, especially in such slow-speed machinery. There is probably little danger from heating, and the question then becomes one of wear. It is better in such cases as the one in question, to choose boldly a length which seems to be reasonable and proceed with the design on that basis, even if the length be later found out of proportion to the shaft diameter, than to waste too much time in the preliminary calculation over the exact determination of this question. Probably in most cases of commercial practice the existence of patterns, or some other practical consideration, will decide the limits of length.

In the present instance it seems reasonable that a length of 6 inches would fill the requirement for the worst case, that of the drum shaft, and it is obvious that the bearings for the pinion shaft would naturally be of the same length on account of being cast on the same bracket, and faced at the same setting of the planer tool.

Height of Centers. The large pulley should naturally swing clear of the floor. This will require, say, a total height of 23 inches, out which we may take 4 inches for the base, leaving 19 inches as the height, center of bearing to base of bracket.

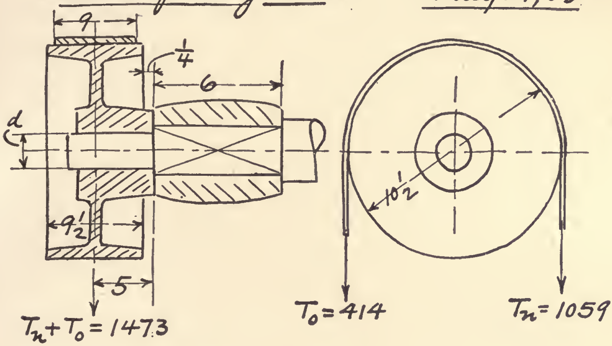
Data on Sketch. The data as found above should now be put on the sketch previously made; it will then have the appearance shown in Fig. 1.

This sketch is now in form to control all the subsequent detail design, and it is expected that the figured dimensions as shown can be maintained. It is impossible to predict this with positiveness, however, as in the working out of the minor details certain changes may be found desirable, when, of course, they should be made.

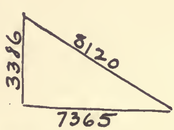
The shaft sizes do not appear on this sketch, hence before proceeding further the several shaft diameters must be calculated.

Sizes of Shafts. The calculations of the shaft diameters are good instances of systematic engineering computations, hence they are reproduced in the exact form in which they were made. The student should learn a valuable lesson in making and recording calculations by following these carefully. Note that each set of figures is independent, both in the statement of given data, as well as in the actual computation. Observe how easy it would be for the author of these figures or anyone else to check them even after

Small pulley bore. *May 21, 03*



$$B = 1473 \times 5 = 7365$$
$$T = 645 \times 5.25 = 3386$$
$$T_e = B + \sqrt{B^2 + T^2}$$
$$= 7365 + \sqrt{7365^2 + 3386^2}$$
$$= 7365 + 8120 = 15485$$



$$T_e = \frac{sI}{c} = \frac{sd^3}{5.1} \qquad \text{let } s = 8000$$
$$15485 = \frac{8000\,d^3}{5.1}$$
$$d^3 = \frac{15485 \times 5.1}{8000} = 9.872$$
$$d = \sqrt[3]{9.872} = 2.14 \quad \text{say } 2\tfrac{1}{8}$$

Fig. 4.

a long lapse of time. If the machine should unexpectedly fail in service the figures are always available to prove or disprove theoretical weakness. The right triangles merely indicate that the value of $\sqrt{B^2+T^2}$ was found by the graphical method suggested in

Part II, "Shafts," the figures being put on the triangle as a simple and direct way of recording both process and result.

Attention is especially called to the fact that in the pinion shaft the size is changed for each piece upon the shaft. This is

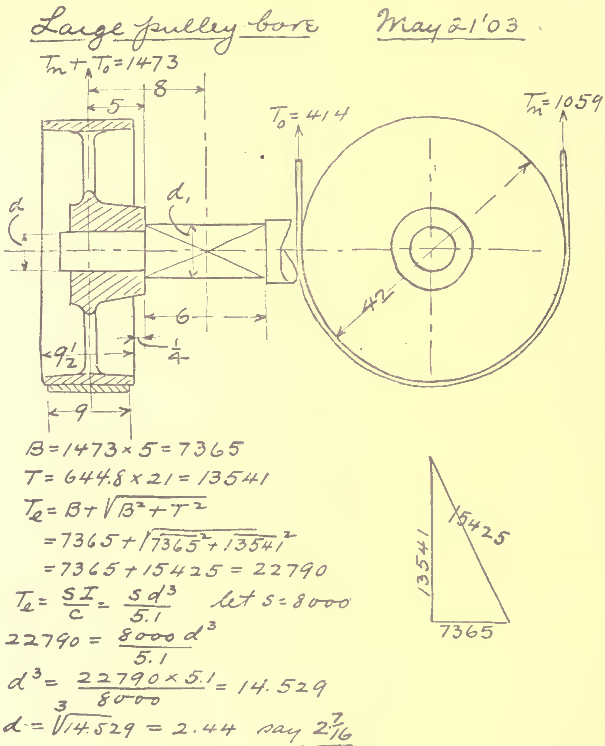


Fig. 5.

done partly because it is desired to show the student that the shaft at each of these points should be theoretically of different size. It is also done because as a practical feature of construction it is a good plan to change the size when the fit changes, partly for reasons of production in the shop, partly for ease in slipping pieces freely endwise on the shaft until they reach their proper fit and location in the assembling of the machine.

This should not be taken as an absolute requirement in any sense. A straight shaft would be satisfactory in the present case; but the shouldered shaft is a little better construction, in a mechanical sense, and does not cost much more. Hence it is used. For the drum the straight shaft seems to answer the requirement well enough.

Bearing next to large pulley. May 21-03

see Fig. 5 and Fig. 7

$B = 1473 \times 8 = 11784$ (due to belt pull).

$B = 3458 \times 3 = 10374$ (due to load on pinion tooth)

Taking the greater value

$B = 11784$

$T = 644.8 \times 21 = 13541$

$T_e = B + \sqrt{B^2 + T^2}$

$= 11784 + \sqrt{11784^2 + 13541^2}$

$= 11784 + 17950 = 29734$

$T_e = \frac{SI}{c} = \frac{S d_1^3}{5.1}$ let $S = 8000$

$29734 = \frac{8000 d_1^3}{5.1}$

$d_1^3 = \frac{29734 \times 5.1}{8000} = 18.955$

$d_1 = \sqrt[3]{18.955} = 2.66$ say $2\frac{11}{16}$

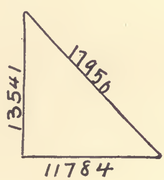


Fig. 6.

Small Pulley Bore. Fig 4.

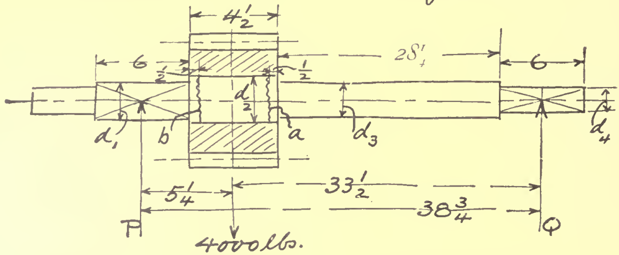
Large Pulley Bore. Fig. 5.

Bearing Next to Large Pulley. Fig. 6.

The diameter, $2\frac{11}{16}$, as calculated, is based on the supposition that the greatest bending moment is caused by the belt pull on the overhanging pulley, that is, by the forces existing at the left-hand side of the center of the bearing.

But the pinion tooth load produces a heavy bending on the shaft in the bearing, the shaft in this case acting as a beam sup-

Pinion Bore *May 21 '03*



$P \times 38.75 = 4000 \times 33.5$ (moments about Q)

$$P = \frac{4000 \times 33.5}{38.75} = 3458$$

$$Q = 4000 - 3458 = 542$$

$B = 542 \times 31.75 = 17208$ (due to Q)

$B = 3458 \times 3.5 = 12103$ (due to P)

Taking the greater value

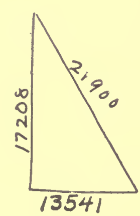
$B = 17208$

$T = 13541$ (as before)

$$T_e = B + \sqrt{B^2 + T^2}$$

$$= 17208 + \sqrt{17208^2 + 13541^2}$$

$$= 17208 + 21900 = 39108$$



$$T_e = \frac{SI}{c} = \frac{s d_2^3}{5.1} \quad \text{let } s = 8000$$

$$39108 = \frac{8000 d_2^3}{5.1}$$

$$d_2^3 = \frac{39108 \times 5.1}{8000} = 24.928$$

$$d_2 = \sqrt[3]{24.928} = 2.92 \quad \underline{\text{say } 2\tfrac{15}{16}}$$

Fig. 7.

ported at the two bearings and having the tooth load applied as shown. If this latter effect be greater than the former, that is, if the bending moment produced by the pinion tooth load be greater than the bending moment produced by the belt pull, then the diameter must be increased to satisfy the latter case. As is seen by the second calculation of Fig. 6, this is not the case, and the diameter stands at $2\frac{1}{16}$ as made.

Pinion Bore. Fig. 7. The pinion being a driving fit upon the shaft, reinforces the shaft to such an extent that it is hardly possible for the shaft to break off very far inside the face of the pinion; but it is quite possible that the metal of the pinion may give enough, or be a little free at the ends of the hole, so that the shaft may be broken off, say $\frac{1}{2}$ inch inside the face. In this case, it may fail from the moment of the force at the left-hand bearing or of that at the right. It may fail then at (a) or (b), depending on which section has the greater bending moment. Trying both, it is seen by the calculation that the right-hand moment is the controlling one, and it, therefore, is used.

Shaft Outside of Pinion. Fig. 8. As there is no power transmitted through this portion of the shaft, there is no torsional moment in it, and the bending moment remains practically the same as inside the pinion.

The size figures about $2\frac{13}{16}$, but since there is no use in turning off material just to reduce the size to this, it is well to make it $2\frac{7}{8}$, or just smaller than the fit in the pinion.

Pinion Shaft Outer Bearing. Fig. 8. This diameter, of course, figures small, as there is no torsion in it, and the bending moment is not heavy. The practical question comes in, however, whether it is advisable to make the outer bracket different from the inner one just on account of this bearing. The commercial answer to this would probably be "No," hence the size as figured next to the pinion will be maintained ($2\frac{11}{16}$).

Drum Shaft. Fig. 9. In this case, as previously inferred, the simplest thing to do is to use a piece of straight cold-rolled steel, and make both bearings alike, the size being determined according to the worst case of loading which can occur as the rope travels from end to end of the drum. This case is evidently when the rope is at the end of its travel close to the brake, for at that time both the load on the rope and the load on the pinion tooth which is driving it are exerted upward, and produce the greatest reaction at the bearing next to the gear. The analysis of the forces for this condition is shown in Fig. 9.

Other conditions of loading would be when the brake is on and the tooth load relieved, but then the resultant of the brake strap tensions would be diagonally downward and would reduce

Shaft outside of pinion. May 22-'03

See sketch of Fig. 7

$B = 542 \times 31.25 = 16937 \qquad T = 0$

$B = \frac{SI}{C} = \frac{Sd_3^3}{10.2} \qquad$ let $S = 8000$

$16937 = \frac{8000\, d_3^3}{10.2}$

$d_3^3 = \frac{16937 \times 10.2}{8000} = 21.594$

$d_3 = \sqrt[3]{21.594} = 2.78$ say $2\frac{13}{16}$

Pinion shaft outer bearing. May 22-'03

See sketch of Fig. 7

$B = 542 \times 3 = 1626 \qquad T = 0$

$B = \frac{SI}{C} = \frac{Sd_4^3}{10.2} \qquad$ let $S = 8000$

$1626 = \frac{8000\, d_4^3}{10.2}$

$d_4^3 = \frac{1626 \times 10.2}{8000} = 2.073$

$d_4 = \sqrt[3]{2.073} = 1.27$ say $1\frac{5}{16}$

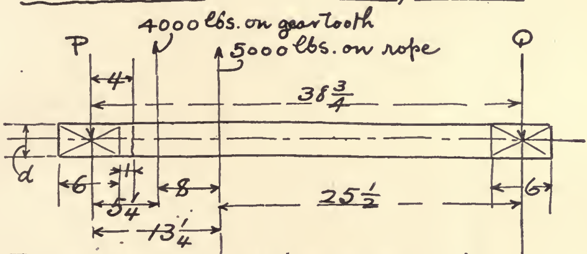
Fig. 8.

rather than add to the rope load. Again, when the rope is at the end of the drum farthest from the gear, the load on it and the load on the pinion tooth are both exerted upward as before, but the reaction cannot be as great as in the case of Fig. 9, because the tooth load is still concentrated at the other end of the shaft and produces a relatively small reaction at the rope end

Preliminary Layout. Fig. 10. Proceeding now with the layout to scale, the detail of the parts may be worked out as completely as the scale of the drawing will permit. The work on this drawing may be of an unfinished, sketchy nature, but the measurements must be exact as far as they go, for this drawing is to serve as the reference sheet, from which all future detail is to be worked up.

In this layout may be worked out the sizes of the arms and hubs of pulleys and gears, the proportions of the drum and brake

Drum shaft May 22 '03



$$P \times 38.75 = 5000 \times 25\tfrac{1}{2} + 4000 \times 33\tfrac{1}{2}$$

$$P = \frac{261500}{38.75} = 6748$$

$$B = 6748 \times 4 = 26992$$

$$B = \frac{SI}{c} = \frac{Sd^3}{10.2} \quad \text{let } S = 8000$$

$$26992 = \frac{8000\,d^3}{10.2}$$

$$d^3 = \frac{26992 \times 10.2}{8000} = 34.415$$

$$d = \sqrt[3]{34.415} = 3.25 \text{ say } 3\tfrac{1}{4}$$

Fig. 9.

strap, and the general dimensions of the side brackets and the base. When the detail becomes too fine to work out to advantage on this drawing it may be worked out full size by a separate sketch, or left to be finished when it is regularly detailed. The preliminary layout, it should be remembered, is a service sheet only, a means of carrying along the design, and not intended for

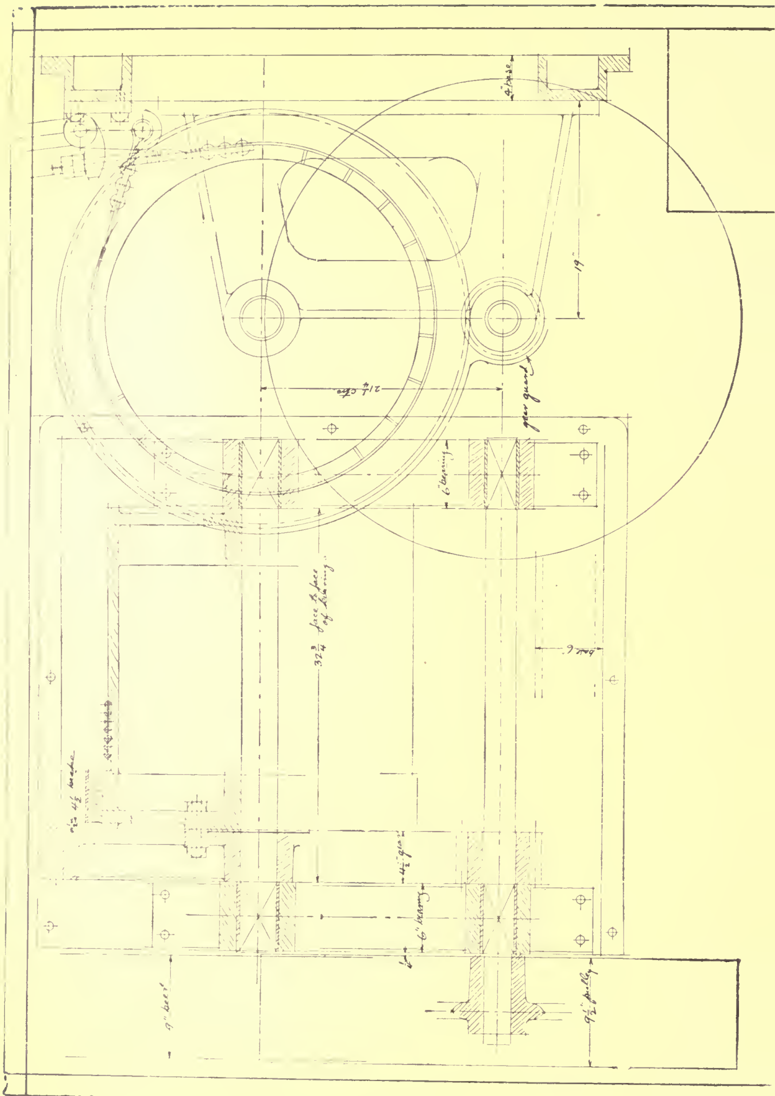


Fig. 10.

a finished drawing. The moment that the free use of the layout is impaired by trying to make too much of a drawing of it, its value is largely lost. A designer must have some place to try out his schemes and devices, and the layout drawing is the place to do it. This drawing may be recurred to at intervals in the progress of the design, details being filled in as they are worked out, as they may control the design of adjacent parts.

As the discussion of the design of each of the members involved in the present problem can be better taken up in connection with the detail drawing of each, it will be given there, rather than in connection with the layout, although many of the proportions thus discussed could be worked out directly from the latter.

Pulleys. Fig. 11. The analysis of the forces in the belt gives, according to the calculation of Fig. 3, a tension in the tight side of 1,059 pounds, and in the slack side 414 pounds. The difference of these, or 1,059—414=645 pounds, is transmitted to the pulley and produces the torque in the shaft. Of course in the small pulley the torque is transmitted from the motor through the pulley to the belt, but both cases are the same as far as the loading of the pulleys is concerned.

The only other force theoretically acting is the centrifugal force due to the speed of the pulley. This produces tension in the rim and arms, but for the low value of 1,300 feet per minute peripheral velocity in this case may be disregarded.

Considering the arms as beams loaded at the ends, and that one-half the whole number of arms take the load, and for convenience, figuring the size of the arms at the center of the pulley gives the following calculation for the large pulley:

$$\frac{645}{3}\times 21=\frac{S\times I}{c}=.0393\times 2{,}500\times h^3$$

Let $S=2{,}500$
" h = breadth of oval
" $.4h$ = thickness of oval

$$h^3=\frac{4{,}515}{98.25}=46$$

$$h=\sqrt[3]{46}=3.6 \text{ (say 3.5)}$$

$$.4h=.4\times 3.5=1.4 \text{ (say 1 7-16)}$$

This is about all the theoretical figuring necessary on this pulley. The rim is made as thin as experience judges it capable of being cast; the arms are tapered to suit the eye, thus giving ample fastening to the rim to provide against shearing off the rim

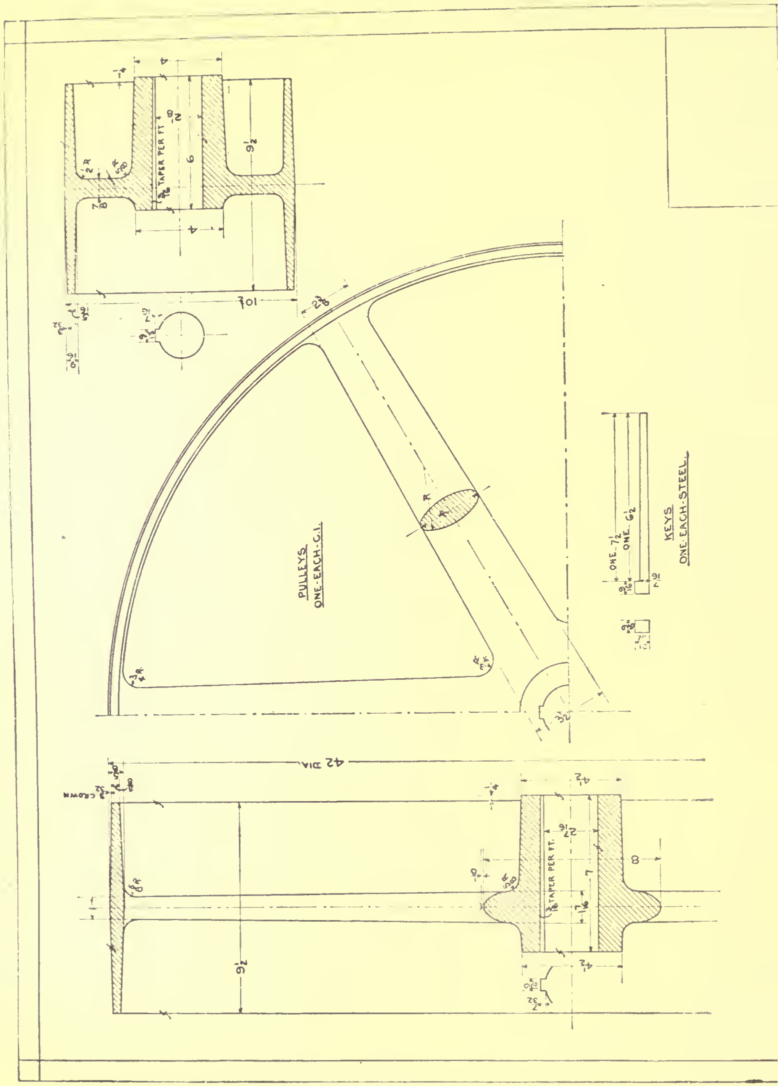

PULLEYS
ONE-EACH-C.I.
KEYS
ONE-EACH-STEEL.
ONE-$7\frac{1}{2}$
ONE-$6\frac{1}{2}$
42 DIA
CROWN
TAPER PER FT.

Fig. 11

from the arms; generous fillets join the arms to both rim and hub; and the hub is given thickness to carry the key, and length enough to prevent tendency to rock on the shaft. Uncertain strains due to unequal cooling in the foundry mold may be set up in the arms and rim, but with careful pouring of the metal they should not be serious, and the low value chosen for the fibre stress allows considerable margin for strength.

The small pulley has the same forces to withstand as the large pulley, but on account of its small diameter there is not room enough for arms between the rim and the hub, hence it is made with a web. The web cannot be given any bending by the belt pull, the only tendency which exists in this case being a shearing where the web joins the hub. This shearing also exists throughout the web as well, but at other points farther from the center it is of less magnitude, and moreover, there is more area of metal to take it. The natural way to proportion the thickness of the web is to give it an intermediate thickness between that of the hub and rim, thus securing uniform cooling, and then figure the stress as a check. Making this value $\frac{7}{8}$ inch gives a shearing area of $\frac{7}{8}$ multiplied by the circumference of the hub, which is $3.1416 \times 4 = 12.56$. The shearing force at the hub is $\frac{645 \times 5.25}{2} = 1,693$ pounds. Equating the external force to the internal resistance

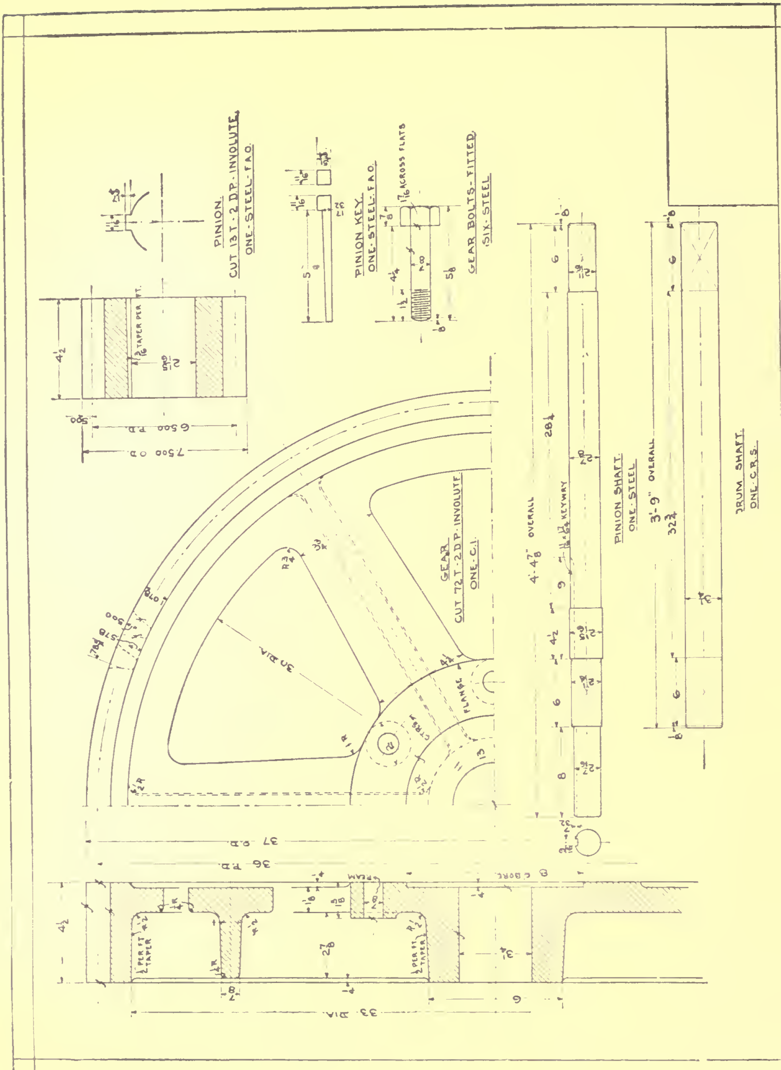
$$1,693 = \frac{7}{8} \times 12.56 \times S$$

$$S = \frac{1,693 \times 8}{7 \times 12.56} = 154 \text{ pounds per square inch (approx.).}$$

This is a very low figure, even for cast iron, hence the web is amply strong. The rim and hub are proportioned as for the large pulley.

The keys are taken from the standard list. They may be checked for shear, crushing in the hub, and crushing in the shaft, but the hubs are so long that it is at once evident without figuring that the stress would run very low in both cases.

Gears. Fig. 12. The analysis of the forces acting on the gears has been given on page 28, 4,000 pounds being taken at the pitch line. Using this same value, and choosing a T-shaped arm as a good form for a heavily loaded gear like the present one, let us consider that the rim is stiff enough to distribute the load



PINION.
CUT 13 T · 2 D.P. · INVOLUTE.
ONE · STEEL · F.A.O.
PINION KEY.
ONE · STEEL · F.A.O.
GEAR BOLTS · FITTED.
SIX · STEEL
GEAR.
CUT 72 T · 2 D.P. INVOLUTE.
ONE · C.I.
PINION SHAFT.
ONE · STEEL
3'-9" OVERALL
DRUM SHAFT.
ONE · C.R.S.

Fig. 12.

equally between all the arms, and that each acts as a beam loaded at the end with its proportion of the tooth load. Before we can determine the length of these arms, however, we must fix upon the size of the flange which is to carry the driving bolts. This is taken at 13 inches. It could be smaller if desired, but drawing the bolts in toward the center increases the load on them, and 13 inches seems reasonable until it is proved otherwise. This makes the maximum moment which can come on an arm $\frac{4{,}000 \times 11.5}{6} = 7{,}666$ inch-pounds.

Now it is evident that the base of the T arm section, which lies in the plane of rotation, is most effective for driving, and that the center leg of the T does not add much to the driving capacity of the arm, although it increases the lateral stiffness of the arm, as well as providing in casting a free flow of metal between the rim and the hub. Hence the simplest way of treating the section of the arm for strength is to consider the base of the T only, of rectangular section, breadth b, and depth h, for which the internal moment of resistance is $\frac{S \times b \times h^2}{6}$.

Also, it is simplest to assume one dimension, say the breadth, and the allowable fibre stress, and figure for the depth. Taking the breadth at $1\frac{1}{8}$ inches, which looks about right, and the fibre stress at 2,500, and equating the external moment to the internal, we have

$$7{,}666 = \frac{2{,}500 \times 1.125 \times h^2}{6}$$

$$h^2 = \frac{6 \times 7{,}666}{2{,}500 \times 1.125} = 16.4$$

$$h = \sqrt{16.4} = 4.05 \text{ (say } 4\tfrac{1}{4})$$

Drawing in this size, and tapering the arm to the rim as in the case of the pulleys, making the depth of the rim according to the suggested proportions given in Part II, "Gears," giving the center leg of the T a thickness of $\frac{7}{8}$ inch tapering to 1 inch, and heavily filleting the arms to the rim and center flange, we have a fairly well proportioned gear.

The next thing to determine is the size of the driving bolts. The circle upon which their centers lie may be 11 inches in diam-

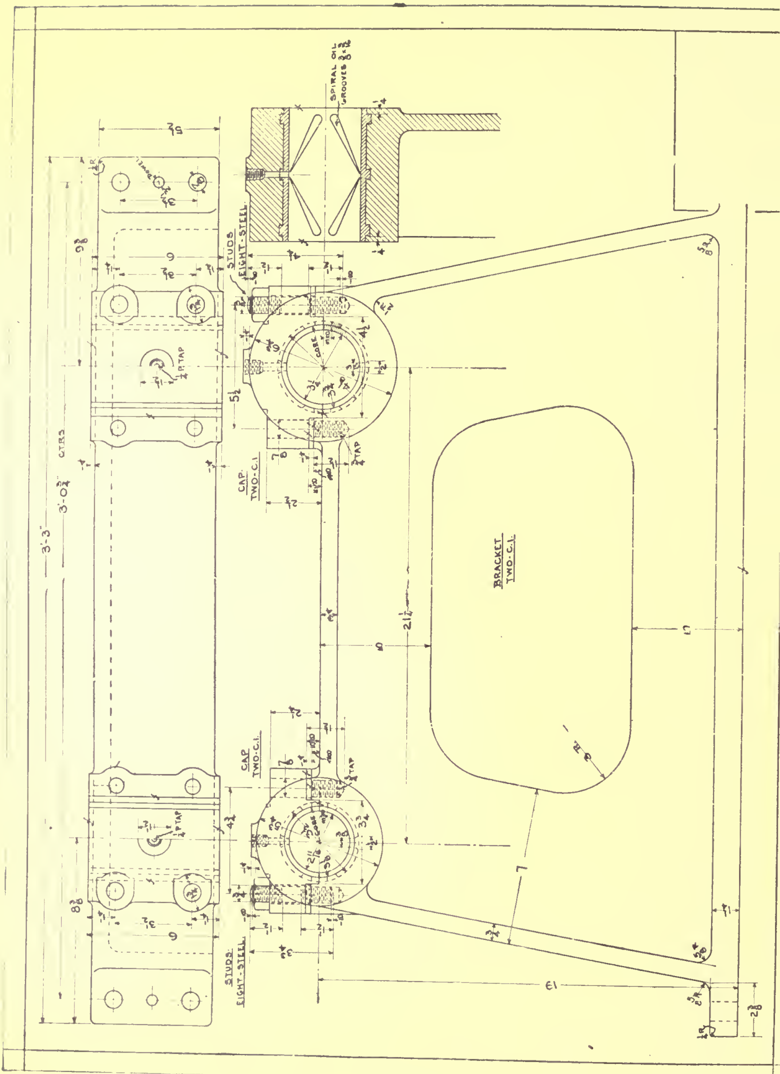

BRACKET
TWO-C.I.
CAP
TWO-C.I.
CAP
TWO-C.I.
STUDS
EIGHT-STEEL
STUDS
EIGHT-STEEL
SPIRAL OIL
GROOVES
CTRS
3'-3"
3'-0¾"

Fig. 13.

eter, and there will naturally be six bolts, one between each arm. These bolts are in pure shear, and the material of which they are to be made ought to be good for at least 8,000 pounds per square inch fibre stress. The force acting at the circumference of an 11-inch circle would be $\frac{4{,}000\times 18}{5.5}=13{,}091$ pounds.

Equating the load on each bolt to the resisting shear gives

$$\frac{13{,}091}{6}=8{,}000\times A=\frac{8{,}000\times 3.1416\times d^2}{4}$$

Let A=area resisting shear.
Let d=dia. of bolt.
Then $A=\frac{\pi\, d^2}{4}$

$$d^2=\frac{4\times 13{,}091}{6\times 8{,}000\times 3.1416}=.35$$

$d=\sqrt{.35}$ (say .6) $\frac{5}{8}$-inch bolts would do.

But $\frac{5}{8}$-inch bolts are pretty small to use in connection with such heavy machinery. They look out of proportion to the adjacent parts. Hence $\frac{7}{8}$-inch bolts have been substituted as being better suited to the place in spite of the fact that theoretically they are larger than necessary. The extra cost is a small matter. These bolts may crush in the flange as well as shear off, but as there is an area of $\frac{7}{8}\times 1\frac{5}{8}=1.422$ square inches to take $\frac{13{,}091}{6}=2{,}182$ pounds, the pressure per square inch of projected area is only $\frac{2{,}182}{1.422}=1{,}534$ pounds, which is very low.

This gear needs no key to the shaft because all the power comes down the arms and passes off to the drum through the bolts, thus putting no torsional stress in the shaft. The face of the flange is counterbored so as to center the gear upon the drum, without relying upon the fit of the gear upon the shaft to do this.

The pinion is solid and needs no discussion for its design.

Brackets and Caps. Fig. 13. As the size of the drum shaft was determined by considering the rope wound close up to the brake, thus giving in combination with the load on the gear tooth the maximum reaction at the bearing as 6,748 pounds, the cap and bolts should be designed to carry the same load.

For a bearing but 6 inches long, two bolts are sufficient under ordinary conditions and might perhaps do for this case. The load is pretty heavy, however, and it is deemed wise to provide four bolts, thus securing extra rigidity, and permitting the use of bolts of comparatively small size. If the load were distributed equally over all the bolts each would take one-fourth of the whole load, but it is not usually safe to figure them on this basis, because it is difficult to guarantee that each bolt will receive its exact share of stress. Assuming that the two bolts on one side take $\frac{2}{3}$ the whole load instead of $\frac{1}{2}$, which provides for this uncertain extra stress, each bolt must take care of $\frac{1}{3}$ of 6,748, or 2,249, pounds. Allowing 8,000 pounds per square inch fibre stress calls for an area at the root of the thread of $\frac{2,249}{8,000} = .281$ square inch. Consulting a table of bolts we find that the next standard size of bolt greater than this is $\frac{3}{4}$, which gives an area of .302 square inch.

Choosing this size as satisfactory, the bolts should be located as close to the shaft as will permit the hole to be drilled and tapped without breaking out. A center distance of $5\frac{1}{2}$ inches accomplishes this result. The distance between centers in the other direction is somewhat arbitrary, although the theoretical distance between the bolt and the end of the bearing to give equal bending moment at the center of the cap and at the line of the bolts is about $\frac{5}{24}$ of the length, or $\frac{5}{24}$ of $6 = 1\frac{1}{4}$ inches. This proportion answers well for the present case, although for long caps it brings the bolts too far in to look well.

The thickness of the cap may be determined by assuming it to be a beam supported at the bolts and loaded at the middle. This is not strictly true, for the load is distributed over at least a portion of the shaft diameter; moreover, the bolts to some extent make the beam fixed at the ends. It being impossible to determine the exact nature of the loading, we may take it as stated, supported at the ends and loaded in the middle, and allow a higher fibre stress than usual, say 3,500. The longitudinal section at the middle of the cap is rectangular, of breadth 6 inches, and depth unknown, say h. The equation of moments is

$$\frac{W \times l}{4} = \frac{S \times I}{c} = \frac{S \times b \times h^2}{6}$$

$$\frac{6,748 \times 5.5}{4} = \frac{3,500 \times 6 \times h^2}{6}$$

$$h^2 = \frac{6 \times 6,748 \times 5.5}{4 \times 3,500 \times 6} = 2.65$$

$h = \sqrt{2.65} = 1.62$ ($1\frac{1}{2}$ will probably answer)

For the other bearing next to the pinion, the load on the tooth acts downward, and the resultant pull of the belt is nearly horizontal, hence the cap and bolts must stand but little load, and calculation would give minute values. In a case like this it is well to make the size the same as for the larger bearing, unless the construction becomes very clumsy thereby. This saves changing drills and taps in making the holes, and preserves the symmetry of the bracket. The $\frac{3}{4}$-inch bolts are good proportion for the smaller bearing, hence that size will be maintained throughout.

The body of the bracket is conveniently made with the web at the side and horizontal ribs extending to the outside. The load due to the rope is carried directly down the side ribs and web into the bottom flanges and to the bolts. The analysis of the forces on these bolts is shown in Fig. 14. It is evident from the figure that the resultant belt pull tends to hold the bracket down, while the load on the rope tends to pull it up, the point about which it tends to rotate being the corner furthest from the drum. It is also evident that the bolts nearest this corner can have little effect on the holding down, because their leverage is so small about the corner, hence we shall assume that the pair of bolts at the right-hand end of the bracket takes all the load. The belt pull, being horizontal, tends to slide the bracket along the base, but this tendency is small, and at any rate is easily taken care of by the two dowel pins, which are thus put in shear.

The load on the bolts being 4,954 pounds, a heavy bending moment is thrown on the flange of the bracket, tending to break it off at the root of the fillet. The distance to the root of the fillet is $\frac{3}{4}$ inch; the section tending to break is rectangular, of breadth $5\frac{1}{2}$ inches, and unknown depth h. The equation of moments is

$$W \times l = \frac{S \times I}{c} = \frac{S \times b \times h^2}{6}$$

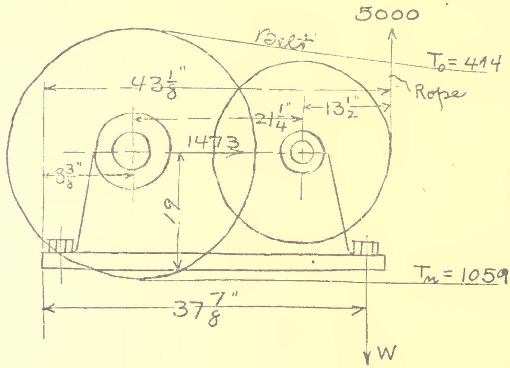
$$\frac{4{,}954 \times 3}{4} = \frac{2{,}500 \times 5.5 \times h^2}{6}$$

$$h^2 = \frac{6 \times 4{,}954 \times 3}{4 \times 2{,}500 \times 5.5} = 1.62$$

$$h = \sqrt{1.62} = 1.3 \text{ (say } 1\tfrac{1}{4}).$$

The thickness of the web and ribs of this bracket is hardly capable of calculation. The figure $\frac{3}{4}$ inch has been chosen in proportion to the size of the large drum bearing, giving ample stiffness and rigidity, and permitting uniform flow and cooling of the metal in the mold. The opening in the center is made merely to save material, as in that part little stress would exist, the two sides

Bracket base bolts May 25-1903



$$5000 \times 43\tfrac{1}{8} = 37\tfrac{7}{8} \times W + 1473 \times 19$$

$$W = \frac{5000 \times 43.125 - 1473 \times 19}{37.875}$$

W = 4954 lbs on two bolts

$\frac{4954}{2}$ = 2477 lbs per bolt.

$\frac{2477}{8000}$ = .31 sq. in. area at 8000 lb. stress.

Area at root of thread for a $\frac{3}{4}$ bolt is .28

Hence $\frac{3}{4}$ bolts will answer.

Fig. 14.

carrying the load down to the base bolts, and the top serving as a tie between the bearings.

This bracket might be made with the web in the center of the bearings instead of at the side, in which case the expense of the pattern would be slightly greater. It could also be made of closed box form, but would in that case probably weigh more than as shown.

Drum and Brake. Fig. 15. The analysis of the forces acting on the drum is simple, but its theoretical design is more complicated. It is evident that the drum acts as a beam of hollow circular cross section, and that its worst case of loading is when the rope is at or near the middle of the drum length. At the same time the metal of this circular cross section is in a state of torsion between the free end of the rope and the driving gear, due to the load on the gear tooth and the reaction of the rope. Also the wrapping of the rope around the drum tends to crush the metal of the section beneath it, the maximum effect of this action being near the free end of the rope where its tension has not been reduced by friction on the drum surface.

Now the "mechanics" to solve the problem of these three combined actions is rather complicated. It can be at least approximately solved, however, for it satisfies fairly well the case of combined compression and shear. But on a further study of this particular case, it is seen at once that the diameter of the drum is relatively large with respect to its length, which means that the thickness of the metal may be very small and yet give a large resisting area, or value of "I," both in direct bending as well as torsion; also it is so short that the external bending moment will be small. The practical condition now comes in, that the drum can be safely cast only when the thickness of the metal is at a minimum limit, for the core may be out of round, not set centrally, or by some other variation produce thin spots or even develop holes reaching out into the rope groove, discovered only when the latter is turned in the lathe.

Hence it seems reasonable and safe in this case to make the thickness of the drum depend simply upon the crushing caused by the wrapping of the rope around it, and we shall take the coil nearest the free end of the rope, assuming that it carries the full load of 5,000 pounds throughout one complete wrap around the drum.

The area resisting the crushing action may be considered to be that of the cross section of a ring, of width equal to the pitch

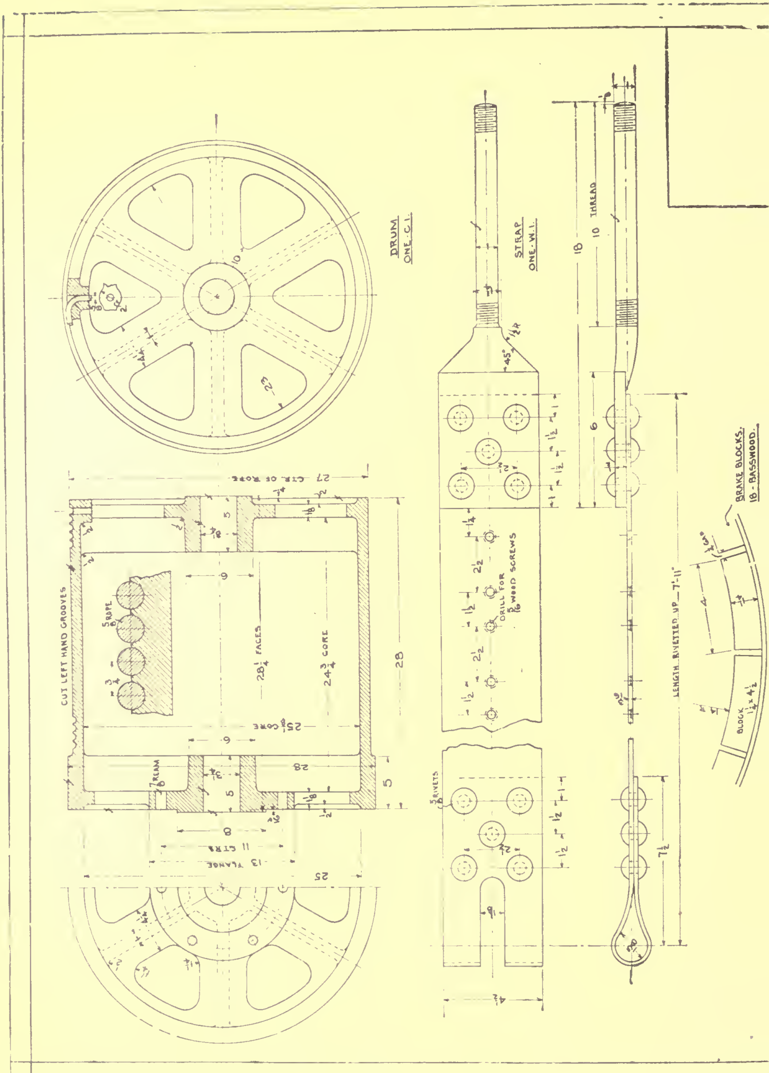

DRUM
ONE-C.I.
STRAP
ONE-W.I.
BRAKE BLOCKS.
18-BASSWOOD.
CUT LEFT HAND GROOVES
27 CTR. OF ROPE
THREAD
LENGTH RIVETED UP 7'-11"
DRILL FOR 3/16 WOOD SCREWS
28 1/4 FACES
24 3/4 CORE
25 1/8 CORE
13 FLANGE
11 CTRS
REAM
RIVETS
BLOCK

Fig. 15.

of the groove. Assuming that $\frac{5}{8}$ inch is the least thickness which can be safely allowed under the groove for casting purposes, let us figure the crushing fibre stress to see if this is sufficiently strong. Disregarding the small amount of metal existing above the bottom of the groove, this gives the area to resist the crushing $\frac{5}{8}\times\frac{3}{4}=\frac{15}{32}$, or .47 inch. Since there are two of these sections and the rope acts on both sides, the equation of forces is:

$$5{,}000\times 2 = S\times .47\times 2$$

$$S=\frac{5{,}000\times 2}{.47\times 2}=10620 \text{ pounds per square inch.}$$

This, for cast iron, in pure crushing, allows plenty of margin for the extra bending and torsional stress, which for such a considerable thickness would be slight.

The above case indicates a method of reasoning much used in designing machinery, which while following out the specified routine of thought as previously given in these pages, stops short of elaborate and minute theoretical calculation when such is obviously unnecessary. If a drum of great length were to be designed, and of small diameter, the same method of reasoning would deduce the fact that the design should be based on the bending and the torsional moments, the thickness in such a case being so great to withstand these that the intensity of the crushing due to wrap of the rope becomes of inappreciable value.

The remaining points of design of the drum are determined from practical considerations and judgment of appearance. The ribs behind the arms are put in to give lateral stiffness and guard against endwise collapse. The arms are subject to the same bending as those of the gear, but as they are equally heavy it is not necessary to calculate them. The flange at the driving end is of course matched to that already designed for the gear. The rope is intended to be brought through the right-hand end with an easy bend and the standard form of button wedged on to prevent its pulling through.

This drum would probably be cast with its axis vertical, and the driving flange down to secure sound metal at that point. Heavy risers would be left at the other end to secure soundness where the rope is fastened. Drums are often cast with the axis horizontal, but the vertical method is more certain to produce a sound casting. The grooves should be turned from the solid metal, partly because it is a difficult matter to cast them, but principally because the rope should run on as smooth surface as possible to avoid undue wear. On drums which carry chain instead of wire rope the grooves are sometimes cast with success, although even in this case the turned groove is generally preferable.

The brake consists of a wrought-iron band to which are fastened wooden blocks, the iron band giving the requisite strength while the blocks give frictional grip on the drum surface and can be easily replaced when worn. As in the designing of a belt the object in view is the grip on the pulley surface by the leather to enable power to be transmitted from the belt to the pulley, so in the case of the brake if we put the proper tension in the strap it can be made to grip the brake drum so tightly that motion between it and the drum cannot occur. The latter case is really the reverse of the first, if the driven pulley be considered, but is identical with the case of the driving pulley, in which the power is transmitted from the pulley to the belt. Of course in the case of the brake no power is transmitted, as when the brake holds no motion occurs, but the principle of the relative tensions in the strap is the same as for the belt.

Since the brake drum surface is 28 inches in diameter, the load at that surface which the brake must hold is

$$P = \frac{5{,}000 \times 27}{14 \times 2} = 4{,}821 \text{ pounds.}$$

We have then the following calculation corresponding exactly to that of the belt given in Fig. 3.

$\log. \frac{T_n}{T_o} = 2.729 \times \mu \times n$ Let $\mu = .25$
" $n = .75$

$T_n - T_o = P = 4{,}821$

$\log. \frac{T_n}{T_o} = 2.729 \times .25 \times .75 = 0.512$ (for which the natural number is 3.25).

Then $\frac{T_n}{T_o} = 3.25 \quad T_o = \frac{T_n}{3.25}$

$T_n - T_o = 4{,}821 \qquad T_n - \frac{T_n}{3.25} = \frac{2.25 \times T_n}{3.25} = 4.821$

$$T_n = \frac{4{,}821 \times 3.25}{2.25} = 6{,}963 \text{ pounds (say 7,000)}$$

$$T_o = 6{,}963 - 4{,}821 = 2{,}142 \text{ pounds (say 2,200)}$$

The tight end of the strap must then be capable of carrying a load of 7,000 pounds, and since the width has already been taken at $4\frac{1}{2}$ inches, the problem is to find the necessary thickness. Equating the external load to the internal resistance we have

$7{,}000 = A \times S$ Let $t =$ thickness

" $S =$ fibre stress $= 12{,}000$

$$7{,}000 = 4.5 \times t \times 12{,}000$$

$$t = \frac{7{,}000}{4.5 \times 12.000} = .13 \text{ inch}$$

This, however, can be but a preliminary figure, for the riveting of the strap will take out some of the effective area, and the thickness will have to be increased to allow for this. Suppose on the basis of this figure we assume the thickness at a slightly increased value, say $\frac{3}{16}$ inch, and proceed to calculate the rivets.

A group of five rivets will work in well for this case, which gives $\frac{7{,}000}{5} = 1{,}400$ pounds per rivet. A safe shearing fibre stress is 6,000, hence the area necessary per rivet is $\frac{1{,}400}{6{,}000} = .23$ square inch. This comes nearest to the area $\frac{9}{16}$ diameter, but for the sake of using the more general size of rivet ($\frac{5}{8}$ inch) the latter is chosen, for which the area is .30.

We must now try these rivets in a $\frac{3}{16}$-inch plate for their safe bearing value. The projected area of a $\frac{5}{8}$-inch hole in a $\frac{3}{16}$-inch plate is $\frac{5}{8} \times \frac{3}{16} = .117$ square inch. $\frac{1{,}400}{.117} = 11{,}965$ (15,000 would be safe)

Taking out two $\frac{5}{8}$-inch rivets from the full width of $4\frac{1}{2}$ inches leaves $4\frac{1}{2} - (2 \times \frac{5}{8}) = 3.25$, and makes the net area of strap to take stress $3.25 \times \frac{3}{16} = .61$ square inches. Re-calculating the fibre stress for this area gives

$$7{,}000 = .61 \times S$$

$$S = \frac{7{,}000}{.61} = 11{,}475$$ (which approximates the previous value of 12,000).

The slack end of the strap has to take but 2,200 pounds, hence a different calculation might be made for this end giving smaller rivets; but as it is impractical to change the thickness of the strap to meet this reduced load, it is well to maintain the same proportion of joint as at the tight end. The spacing of the rivets in both

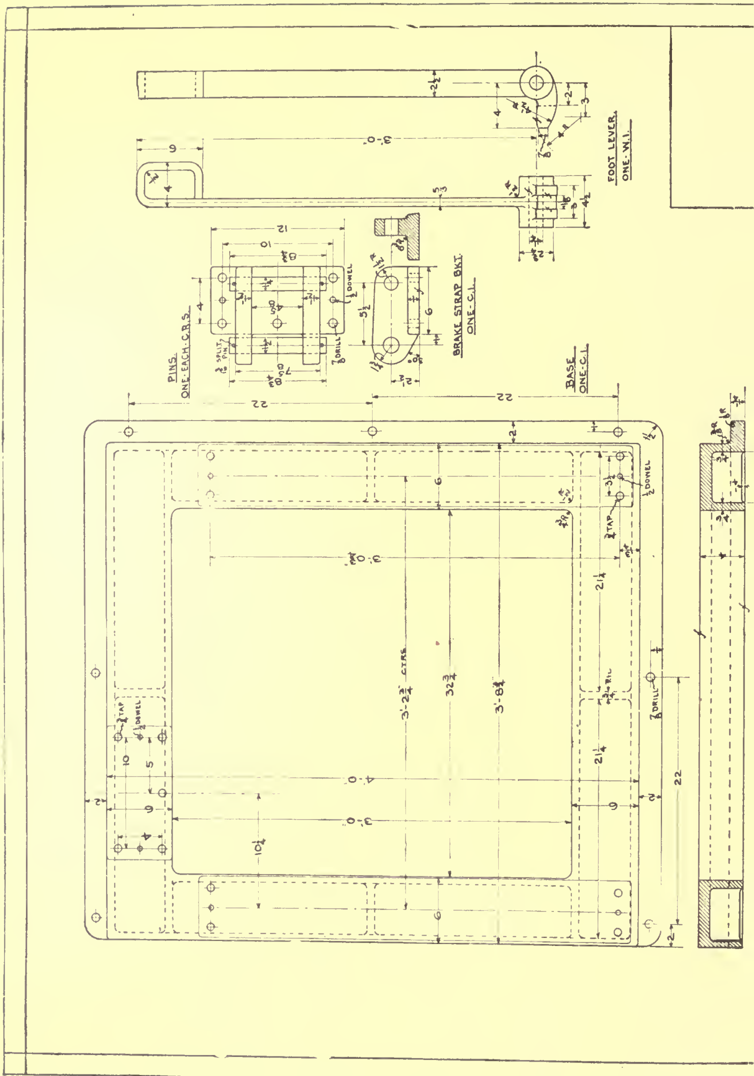

FOOT LEVER.
ONE·W.I.
PINS.
ONE·EACH·C.R.S.
BRAKE STRAP BKT.
ONE·C.I.
BASE
ONE·C.I.

Fig. 16.

cases follows the ordinary rule allowing at least three times the diameter of the rivet as center distance, and one-half this value to the edge of the plate.

The threaded end of the forging on the strap also has to carry the load of 2,200 pounds, for which a size smaller than 1 inch would suffice. It is natural, however, for the sake of general proportion to make the bolt as strong as the strap, and a 1-inch bolt gives an area of .52 square inch, nearly equalling the value of .61 net area of strap noted above.

Base, Brake-Strap Bracket and Foot Lever. Fig 16. The base cannot be definitely calculated, and can best be proportioned by judgment. It must not distort, twist, or spring in any way to throw the shafts out of line. The area in contact with the foundation upon which it rests must be ample to carry the weight of the whole machine with a low unit pressure. Although the form shown is perfectly practicable to cast and machine, and is simple and rigid, yet it is questionable if a bolted-up construction, say of four pieces, might not be equally rigid and yet involve greater facility of production in both the foundry and machine shop on account of the reduced sizes of parts to be handled. This is a question which depends on the equipment and methods of the individual shop, and is an illustration of the practical control of design by manufacturing conditions.

The brake-strap bracket and foot lever, also shown in this figure, are examples of machine parts which are quite definitely loaded, and the designing of which is a simple matter. Further discussion of their design is not made, the student being given opportunity for some original thought in determining the forces and moments that control their design.

Gear Guard and Brake-Relief Spring. In exposed machinery of this character it is desirable to cover over the gears with a guard to prevent anything accidentally dropping between the teeth and perhaps wrecking the whole machine. This guard is not shown, as it involves little of an engineering nature to interest the student. It could readily be made of sheet metal or light boiler plate, bent to follow the contour of the gears and fastened to the top flange of the main bracket.

If the brake be not automatically supported at its top it will lie with considerable pressure, due to its own weight, on the brake surface when it is supposed to be free from it, and by the friction thereby created will produce a heavy drag and waste of power. A spring connection fastened to an overhead beam is a simple way of accomplishing the desired result. A flat supporting strap carried out from the gear guard, having some degree of spring in it, is a neater method of solving the problem. The spring should be just strong enough to counterbalance the weight of the strap and yet not resist to an appreciable degree the force applied to throw the brake on.

GENERAL DRAWING.

The last step in the process of design of a machine is the making of the assembled or general drawing. This should be built up piece by piece from the detail drawings, thereby serving as a last check on the parts going together. This drawing may be a cross section or an outside view. In any case it is not wise to try to show too much of the inside construction by dotted lines, for if this be attempted, the drawing soon loses its character of clearness, and becomes practically useless. A general drawing should clearly **hint at, but not specify,** detailed design. It is just as valuable a part of the design as the detail drawing, but it cannot be made to answer for both with any degree of success. A good general drawing has plenty of views, and an abundance of cross sections, but few dotted lines.

The general drawing of the machine under consideration is left for the student to work up from the complete details shown. It would look something like the preliminary layout of Fig. 10, if the same were carefully carried out to finished form. A plain outside view would probably be more satisfactory in this case than a cross section, as the latter would show little more of value than the former. The functions which the general drawing may serve are many and varied. Its principal usefulness is, perhaps, in showing to the workman how the various parts go together, enabling him to sort out readily the finished detail parts and assemble them, finally producing the complete structure. Otherwise the making of a machine, even with the parts all at hand, would be like the putting together of the many parts of an intricate puzzle, and much time would be wasted in trying to make the several parts fit, with perhaps never complete success in giving each its absolutely correct location.

The general drawing also gives valuable information as to the total space occupied by the completed machine, enabling its location in a crowded manufacturing plant to be planned for, its connection to the main driving element arranged, and its convenience of operation studied.

In some classes of work it is a convenient practice to letter each part on the general drawing, and to note the same letters on the specification or order sheet, thus enabling the whole machine to be ordered from the general drawings. This is a very excellent service performed by the general drawing in certain lines of work, but for such a purpose the drawing is quite inapplicable in others.

Merely as a basis for judgment of design, the general drawing fulfils an important function in any class of work, for it approaches the nearest possible to the actual appearance that the machine will have when finished. A good general drawing is, for critical purposes, of as much value to the expert eye of the mechanical engineer as the elaborate and colored sketch of the architect is to the house builder or landscape designer.

From the above it is readily understood that the general drawing, although a mere putting together of parts in illustration, is yet of great assistance in producing finished and exact machine design.

GENERAL COMMENTS ON PRECEDING PROBLEM.

After following through the detail of work as given in the preceding pages, it is worth while to stop for a moment and take a brief survey or review of the subject as illustrated therein.

If the text be carefully studied it will be seen that in every part to be designed the same routine method has been followed, regardless of the final outcome. In some cases it may seem a roundabout procedure to follow a train of thought that finally ends in a design apparently based on purely practical judgment, the theory having had but very little if any influence. The question at once arises—Why not use the empirical rule or formula in the first place? Why not make a good guess at once? Why not save all the time and energy devoted to a careful analysis and theory, if we are finally to throw them away and not base our design on them?

The principle to be noted in this connection is, that it is just as fatal to good design to rely upon bare experience and upon judgment alone, as it is to construct solely according to what pure theory tells us. There are many things in the operation of machinery that are totally inexplicable from the purely practical point of view, and will forever remain so until we analyze them and theorize on them. Many good things in machinery have been the result of what might be called "reversed" machine design. When a new machine is started, it frequently, or we might almost say always, fails to do its work just as it is expected to do it. This is because some little point of design is bad, owing to the inability of drawings, however good they may be, to show all that the machine itself in bodily form and in motion shows.

Now, if our analysis and theory have been good in the designing process, it is almost sure that we can very readily analyze and theorize on the trouble that exists when the machine is finished, can detect the weakness, and can correct it with comparatively small change in the general design. This is "reversed" machine design.

If, on the contrary, we have based our design purely on guesswork, allowing our fancy full and free play to work out the details without further basis, we may consider ourselves lucky if the machine runs at all. This, however, is not the worst of the situation. If the machine does actually operate, even as well as it might reasonably be expected to, but still has the usual difficulty of some little kink or hitch that was not expected, then, as a result of the method upon which the whole thing has been constructed, we have no definite plan of action to proceed upon. We must try first this, then that scheme to obviate the trouble. We may be fortunate enough to "strike it" the first time; we may never strike it. It is doubtful if the machine ever can be made to work at highest efficiency; and if fairly good results be finally obtained we never know the reason why, and have nothing on which to base any future action or design.

This haphazard process is not machine design at all, either in name or in result.

As has previously been stated in these pages, there is no such thing as too much analysis or theory in the designing of machinery. Even if we carefully analyze, theorize with rigorous exactness, and then practically modify our construction to such a point that the original theoretical shape is almost or entirely lost, the apparently roundabout process is not in vain, for we are in perfect control of our design. We know exactly what it has to take in the way of forces, blows and vibrations. We know what its ideal shape should be. We know where we can practically modify its form without weakening it excessively or adding excess of material. In other words we know all about it, and therefore know exactly what we can do with it; and whether it follows in its shape the outline that pure theory gives it or some other outline, it is nevertheless well designed.

"Reversed" machine design, as described above, based on observation and experiment with regard to machines already in operation, is just as impossible without exact analysis and theory as is original design based merely on mechanical ideas in the abstract. The method once learned and made a habit of mind will produce results with equal facility in either case, and results are what the mechanical world is seeking.

Another point worth noting in the progress of the problem as given is the absolute necessity of possessing some knowlege of Mechanics. The more of this subject the designer can have at his finger ends, the more ready and successful will he be in all problems of Machine Design. However, the principles of forces and moments clearly understood, and the application of the same in the all-important subject, "Strength of Beams," constitute a fund of information that will give a splendid start and a good working basis for simple designs. It should always be remembered that a complicated design is little more than a combination of simple designs, and if one has the ability to dissect and analyze what seems at first like a bewildering maze of parts, complication is speedily changed to simplicity.

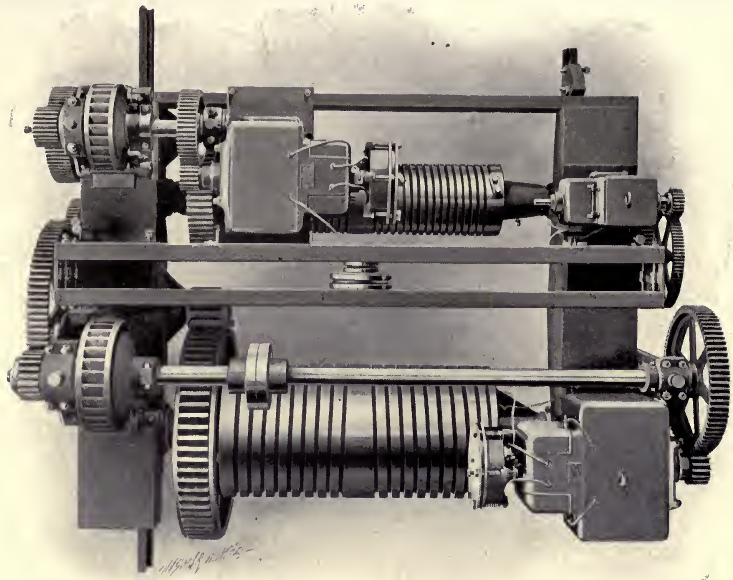
Common sense goes a long way in good designing. There is nothing mysterious about the process If the beginner will only avoid doing things that are foolish and ridiculous on their very face, if he will exercise the same judgment that he uses in the daily affairs of his life and will mix in something of mechanics and mechanical method, he will be on the direct road to success in the art.

Good drawing is an essential element of good design, and it is especially urged that the sketches and drawings as reproduced in the preceding text be studied with this in mind. By a good drawing is meant not a showy piece of work, finely shaded or artistically lettered, but an exact layout, definite and measurable, correctly dimensioned if in detail, and meaning exactly what it says. Machine design is an exact science, and the designer cannot shirk responsibility by permitting his work to be shiftless and loose. If he cannot delineate clearly and in definite form what he determines in his mind the structure should be, then it is purely good luck if he achieves success, and it may safely be asserted that the success is due to some subsequent care and finished design added to his feeble effort, rather than to any expertness of his own. Such success is of a very doubtful nature, and if not bordering on financial loss it is at least secured only at a low working efficiency.

As examples of good drawings the plates shown are not claimed to be anything extraordinary, but it will be noted that they are clean-cut and definite, and that even the sketches are unmistakable as to that which they are intended to illustrate. The information as to the design is all there; nothing is left to the imagination.

Classification of Machinery. It is intended to be made clear in all that has preceded, that the same method of attack and procedure may be applied to the designing of machinery, whatever may be the class or kind. This is a fundamental principle. When it is logically carried out, however, it produces very different results, as is evidenced by the characteristics of style peculiar to each of the classes of machinery to one or another of which all machines belong.

For example, an engine lathe has a style similar to a drill press, or a boring mill, or a screw machine, or a milling machine. It is very different, however, from the style of a steam engine, or a pump, or an air compressor, or a locomotive; it is still more dif-



TROLLEY OF 15 TONS CAPACITY.
The Case M'f'g Co.

ferent from the style of a rolling mill, or a link belt conveyor, or a coal crusher, or a stamp mill.

These classes of machinery are so distinctly marked that the novice is easily able to perceive that there is some controlling influence in each which marks its peculiar style. He should at the same time see that the very analysis that has been so strongly insisted upon in these pages is the direct cause of the marked characteristic in design. Each class of machinery must satisfy certain exacting conditions different from those of any other, and it is the careful study of these conditions, as fundamentally enforced, which leads to the strictly logical design.

A few of the most common classes are enumerated below, and their prominent features noted. It is hoped that a study of them will familiarize the student in a general way with the requirements of each, and serve as a guide to a more comprehensive study of their detail design than is possible in these pages.

Machine Tools. Examples:—lathe, planer, milling machine, drill press, screw machine, boring mill, grinding machine, etc., etc.

The machines of this class are all utilized for the finishing of metal surfaces. They are really at the root of the production of machinery of all other classes. Accuracy is their prime characteristic—accuracy of construction, accuracy of operation, accuracy of adjustment. Any inaccuracy that exists primarily in a machine tool is reproduced in every piece upon which it produces a finished surface; and since the mere act of finishing a surface upon anything implies that a rough and inaccurate surface will not answer, the tool then fails of its purpose if it cannot produce a true surface: it does not accomplish that for which it was designed.

The effect that this element of accuracy has upon the design of a machine tool is to require long bearings, convenient and exact methods of adjustment, stiffness, excess of material to absorb vibration, special shapes to facilitate application of jigs, fixtures, and exact manufacturing devices insuring interchangeability of parts, dust guards, and automatic lubrication.

Machine tools are essentially machines of maximum output, and depend for their success, not only upon their accuracy as noted, but also upon their ability to do the greatest amount of work per square foot of space occupied, with the least amount of manual labor and attention on the part of the operator. This is especially true of automatic machinery, which perhaps might be classed by itself in this respect, but which is nevertheless included under the broad term of a machine for producing finished surfaces, being merely the highest and most refined form of same. For machines of this class the designer has to study every detail with the most minute attention, packing away the operating parts into the smallest space and yet providing ready means for access, removal, and repair. Clearances that would be too little for other kinds of machinery are permitted and provided for; material of high grade, strength, and wearing quality, though expensive in first cost, and requiring the most expert skill to finish and to fit into place, must be used in order to keep the machine compact and yet of large capacity, to make it reasonably light in weight and yet amply strong.

Another point which has a great influence on the design of a machine tool is that we can never tell in advance just what it will have to stand in work, for the variation in the material that it finishes, the uncertain skill of the operator who runs it, the crowding to its limit of capacity and even beyond in times of press of business, and the many other stresses that may suddenly and without warning be thrown upon it, must all be thought of and provided for.

The points above mentioned are but a few of those which the designer of machine tools has to meet, and are presented merely as illustrations to show the special skill required in this class of machinery. It is readily seen that while the machine tool designer has great latitude in choice of material and in expenditure of money for refinement of structure—perhaps greater latitude than in any other class, yet he is held down as in no other to the final productive results, a small percentage of failure entirely throwing out the machine as a marketable product.

The style and external appearance of machine tools have a character of their own resulting from this extreme detailed care in design. Corners and fillets are carefully rounded; surfaces and intersections are definitely made; in short, the mechanical beauty of a machine tool is seen only from a near view and close inspection, and it is to this end that the design is constantly directed Appearance is a large factor in the sale of a fine tool, and the prestige of the American trade abroad in this respect is very noticeable.

Motive-Power Machinery. Examples:—Steam engine, gas engine, air compressor, steam pump, hydraulic machinery, etc., etc.

The element of heat enters into the design of all machinery in this class. The natural agents, air, gas, and water, in their various forms, are taken into the machine in the most efficient form in which it is possible to obtain them, are robbed of their energy to provide power, and are discharged in a form as weak and inert as the efficiency of the machine will determine.

In contrast to the class of machinery just studied, it should be noted that these machines do not produce any material thing; that is, they do not produce finished surfaces on metals, make screws or bolts, bore holes in castings, or turn line shafting. They merely take the energy of the natural agent, which is not in a form available for use, and transform it into motive power for general use.

Hence the element of accuracy as entering into the design of these machines is necessary only for their own efficient operation, and not for the quality of the thing which they produce, as in the case of machine tools. For example, the power furnished by one steam engine to drive a line shaft is as good as that of another as far as the rotating of the shaft is concerned, provided, of course, that both are equipped with the same quality of governing mechanism. The fact that one of the engines has a good adjusting device on the main bearing while the other has not is of no consequence from the standpoint of the line shaft, but it is, of course, of consequence respecting the efficient operation of the engines.

The design of steam engines and similar machines is of a rough nature compared with that of machine tools, as far as the detail of surface is concerned. General accuracy is nevertheless essential for the machine's own sake, but while in the machine tool we deal with thousandths of an inch, in the steam engine hundredths of an inch indicates fine work.

These machines are subject to extremes of temperature that have to be provided for in the design and arrangement of the parts. Being prime movers, controlling the operation of many machines, they must be certain to run during their period of work; hence design and adjustment must be positive, and when the latter cannot be made while running, it must be quickly and definitely accomplished when a stop is made. Simplicity of construction is essential, facilitating cheap and quick repairs. The design should be such that constant attention while running is avoided, the usual attention of the engineer being a safeguard rather than an implied factor of the original design. General rigidity and stiffness are important, also good balancing of the moving parts, and weight for absorption of vibration; otherwise under the constant daily run the machines will tear to pieces not only themselves but their foundations.

As far as external appearance goes in this and subsequent classes to be mentioned we are on a very different basis from that of machine tools. *General* mechanical symmetry of form is aimed at in the design, and the several smaller parts depend for their outline (aside from considerations of strength, which are, of course, always in order) upon the harmonious relation which they bear to the main and fundamental elements of the machine. Such machinery as air compressors, steam engines, pumps, and the like are viewed as a whole, and criticised, not detail by detail, as is the machine tool, but as to general effect of outline observed from some distance. To convey the desired effect to the eye the design must be bold and massive, connections simple and direct, and the smaller parts must not be so dwarfed in size as to appear like delicate ornaments instead of integral parts of the machine. The lines of connected parts must be continuous from one part to the other; and when interrupted by flanges, bosses, or lugs, the latter, which are merely incidental to the former must not be allowed to obscure wholly the main lines of the fundamental pieces.

It is attention to such points as these that marks the difference between well-designed motive-power machinery and that of the opposite character. Even though the little details of fillets and corners and surfaces may have their effect from a close point of view, the design will stand or fall in excellence on its bolder features, as noted above.

Structural Machinery. Examples:—Hoists, cranes, elevators, transfer tables, locomotives, cars, conveyors, cable-ways, etc., etc.

In the two preceding classes that have been noted, cast iron in the form of foundry castings enters as the principal material. Steel is utilized for shafts, studs, pins, and keys. Also special forgings, malleable iron and steel castings enter as factors in the production of the machinery discussed. Foundry castings, however, compose the great body of the material used, and the chief problems involved are those of the expert moulding of cast iron, and the handling and finishing of the same. For the operating parts, steel of fine grade is used in highly finished form, expensive because of its fineness, and yet a necessity to the extent it is used. Brass and bronze are used in the same way, generally in connection with the bearings for the shafts.

Structural machinery, on the contrary, uses steel as the basis of its construction. The fundamental structure is built up of plates, channels, beams, and angles; castings, though numerous, are relatively small, being riveted or bolted to the main structure and controlled in their design by its requirements.

Steel is used in this manner partly because the exclusive use of castings is prohibited on account of the excessive weight, and therefore expense, and partly because castings could not be made which would possess the necessary toughness and strength. In many cases the size of the machinery is such that castings, even if they could be made, would not support their own weight. Moreover, machinery of this class is subjected to rough service, and yet must be practically infallible under all conditions, neither being uncertain in operation at critical moments nor entirely failing under an extraordinary load.

The design of structural machinery is tied up to conditions existing largely outside of the locality in which the machinery is built. The steel plates and structural shapes required, being products of the rolling mill, have to conform to the latter's standards. The rivets, bolts and other fastenings have to be in accordance with the established practice of the structural iron worker, in order to permit punching, shearing and bending machinery of regular form to be utilized. Shipment on standard railway cars has to be considered, the design often requiring to be modified to permit this and nevertheless insure positive and accurate assembling in the field.

Steel castings, both large and small, find ready application in this class of work; also steel forgings, requiring to be worked under a heavy hammer and in many cases by specially devised processes.

In structural design less of the actual process of manufacture is under the eye of the designer than in the former classes of machinery which have been considered, and hence more allowance has to be made for things not coming exactly right to the fraction of an inch. It would be bad design to plan any structural piece of work with the same closeness of detail permitted, and in fact required, in the case of machine tools, or even in the case of motive-power machinery. In planning structural work the idea must be carried out, of certainty of operation in spite of roughness of detail and variations of construction. This does not necessarily imply inaccuracy, or shiftless, loosely constructed machinery; on the contrary, quite the reverse. The locomotive, for example, is one of the most refined pieces of mechanism that exists today; and yet the methods applied to the construction of machine tools would prove a failure on the locomotive. The design of a car axle box has to be just right else it will heat and destroy itself; the same is true of the spindle of a fine engine lathe; and yet how rough the former is compared with the latter, and how unsuited either would be for use on the service of the other.

As a general rule structural machinery can be more closely proportioned to theoretically calculated size than can the preceding types. The rolled material of which it is made is of a uniform and homogeneous nature owing to its process of manufacture, hence its every fibre may be counted on to sustain its share of the total load imposed upon it. This is in sharp contrast to the case of cast iron, which is of such a porous and irregular structure that we have to use a large factor of safety to cover this inherent defect.

Steel castings of both small and large size (which are quite apt to be utilized in this class of machinery for parts that can with difficulty be made out of rolled material), if properly designed of uniform thickness, with all corners well filleted and with the channels for the flow of the molten metal direct and ample, are nearly as reliable as rolled steel. In parts subject to excessive vibration, shocks, and sudden wrenchings, as, for example, the side frames or the connecting rod of a locomotive, the forged and hammered material is practically a necessity. This is especially the case when the possible breakage of the part would cause serious consequences involving heavy loss of life and property.

From the several points of view as above considered, it can be readily appreciated that, while structural work is in one sense rough and unpolished, yet it requires, from an engineering standpoint, quite as much breadth of experience and judgment as any of the other types. The fine-tool designer, least of all, perhaps, requires book theory, but does require an extended machine-shop experience. The designer of motive-power machinery needs pure physical theory and shop experience of a large and broad scope. The structural designer is least of all concerned with refined and minute finishing processes, but utilizes his theory absolutely, even though roughly.

Mill and Plant Machinery. Examples:— Rolling mills, mining machinery, crushers, stamps, rock drills, coal cutters, the machinery of blast furnaces and steel mills, tube mills, etc., etc.

This machinery constitutes a class which in the roughness of its operation exceeds all others. Moreover, it is machinery which for the most part is in continuous operation—24 hours per day and 365 days in the year. Hence refinement, even such as might be permitted in the preceding class of Structural Machinery, would be fatal here. The conditions that surround plant machinery are unfavorable in the extreme to the life of any material or metal, and it is not possible to change these conditions or give more than partial protection to the operating parts. Hence the design of such machinery must proceed primarily on the assumption that abuse and neglect, grinding away of surfaces, chemical eating away of metal, flooding of parts with water gritty and corrosive, subjection to sudden bursts of flame and intense heat, etc., will in a relatively short time totally destroy, perhaps, the entire structure.

In view of the continuous nature of the working process, which must be kept up in spite of these almost insurmountable conditions, the problem in each case becomes one of expediency; and the designs and arrangement of machinery must be so worked out that operation, repair, construction, and installation can all go on simultaneously without stopping the continuous process, and with but a small degree of inconvenience to the operation of the plant.

This problem, difficult though it may seem, can be worked out successfully, as is evidenced by the great number of plants of the continuous character operating at high efficiency throughout the world. The engineering and designing skill required to accomplish this, is perhaps of the highest degree met with in modern practice, for in it is involved a working knowledge of the possibilities, if not the detailed designs of machinery included in all classes. And yet, as in the most elementary case of simple design that can be conceived, the result is accomplished in the same way, namely, by studying the conditions (analysis), developing an ideal application to those conditions (theory), and then reducing the ideal design to a practical basis (modification).

A Few Pointed Suggestions on Original Design. Original design deals with the development of original mechanical ideas. The prime requisite for the development of an idea is to understand thoroughly the idea in the rough. See distinctly the mark aimed at, and never lose sight of it. If a method of reaching it is already outlined, understand that also thoroughly and the principles involved. It is impossible to go ahead blindly and hope to come out right. No good machine was ever built that does not stand for hours of concentrated thought on the part of its designer. Good machines never *happen*, they always *grow*.

Just as soon as the object to be accomplished is clearly understood, begin to produce some visible work on the problem. Sketch something. Get some ideas on paper. Ideas on paper suggest other ideas. If the problem, for example, is one of lathe design, sketch a rectangle, and call it the headstock; another rectangle, and call it the footstock; a couple of scratches for the centers; some steps for the cone pulley; three or four lines for the bed; and as many more for the supports. There is now something on paper to look at; the design is begun.

It is much better to stare at this sketch, than into blank space trying to imagine the finished design. No matter how rough the sketch may be, a short study of it will develop some limiting conditions that before were not apparent. Guess at a few rough dimensions; put them on the sketch; develop another view—a plan or a side elevation—all still in the roughest style, without any regard to finished detail. Information will be growing all the while, and the problem will be opening up. At this stage it is probable that the sketch can easily be seen to be wrong in many respects. Perhaps the arrangement will not do at all.

This is a good sign. It shows that the design is progressing. It is a valuable thing to know that certain plans *cannot* be followed. Do not rub out part of the sketch already made and try to remedy it. Begin again. Make another sketch. Sketch paper is cheap. By and by it may prove to be very desirable to have that first rough outline available for comparison; or it may be that some of its ideas can be applied on other sketches. The second sketch may "show up" little or no better than the first. Make another, and another, and another, until the subject is thoroughly digested. It is wonderful how helpful it is to have some marks on paper relative to a design, even though they be of the utmost crudeness. They save imaginative power tremendously; and, even with them, all available powers of imagination will be needed before the design is perfected.

A careful comparison of one's sketches, rejecting here, and approving there, will, little by little, bring about a definite opinion, and the scale drawing can be begun.

As in the case of the first sketch, so in the case of the first scale drawing, get some lines on paper as quickly as possible. Draw something, even if it is nothing more than a straight horizontal line. Do not stare at blank paper for an hour trying to imagine how the tenth or eleventh line is going to be drawn in relation to the first line. Do not worry about the later lines until it is time for them Draw the first line at once; and, when the second line is drawn, if the first line proves to be wrong, make it right. As in the rough sketch, that first horizontal line is an immense relief from the great waste of blank paper of a fresh sheet. It is something to look at. It is the beginning of a detailed design. If it happens not to be the absolutely correct foundation to build upon, it at least is something to tear down. The main purpose of these preliminary drawings is to keep the mind active on the problem; and advance toward the final accomplishment of the design is often made quite as rapidly by discovering what to tear down as by consistently building up.

When a detail draftsman who has been used to having all his work laid out for him by an expert designer attempts to take up original work for himself, he encounters the drawing of that first line in a way he never did before. He is apt to worry for some time over the possible or impossible results of drawing that first line. If he continue this, he will be sure to fail. The second line is much easier to draw than the first, and the third than the second; and the next hundred will follow on in comparatively smooth sequence, all because of bold action on the first few lines.

And yet, just as the design appears to be progressing smoothly, and the advanced progress of the drawing seems cause for congratulation, careful consideration may disclose a "snag" not previously known to exist in the problem. Further study pursued along the line of this new discovery may show that the whole layout thus far has been radically wrong, and that a fresh start will have to be made. At such a time the young designer is apt to feel that his labor has all been thrown away, and he becomes discouraged. There is, however, no cause for discouragement. Machine Design might almost be defined to be the "successful elimination of snags." It takes some ability to discover an obstacle of this sort; to know a "snag" when an opportunity to see it is given. It takes a good designer to eliminate such a difficulty after it has been found. If there were no "snags" it would not require great ability to design machines. Many machines fail because in them there are a lot of undiscovered "snags." Others fail because the "snags," although discovered, were not eliminated by careful design.

Do not be afraid to make a lot of "first" drawings. It is just as important to digest the design thoroughly by means of scale drawings, as it was to digest it originally by means of the rough sketches. An attempt to make the first drawing of an original design absolutely right would, it is safe to say, produce a poor design, one that could be much improved by further trial. Let the drawings multiply, one after another, until the final one is reached, in which the perfection of detail will eliminate all the bad points of the preceding drafts and incorporate good ones of its own based on the study of the others.

And yet it is often true that the first design laid out, even after many others have been developed, may be found to possess features that render a return to it desirable. This is why it is always better to produce a collection of designs than to attempt to rub out and work over the first one. The best designers usually have a great number of sketches showing how to accomplish a single result. Likewise, they also have a series of layouts to scale, showing in detailed form the development of their various ideas. This is because, without a careful consideration of many methods, they themselves feel incompetent to judge of the best design possible for accomplishing a given result.

Sketches and original designs should always be dated and signed. Different designers may be working on the same problem, and priority of design will never be allowed except upon signed and witnessed papers. It is embarrassing to find, after months and perhaps years have passed since an original drawing was made, that one's rights have been preëmpted merely because there was no date or signature to define them.

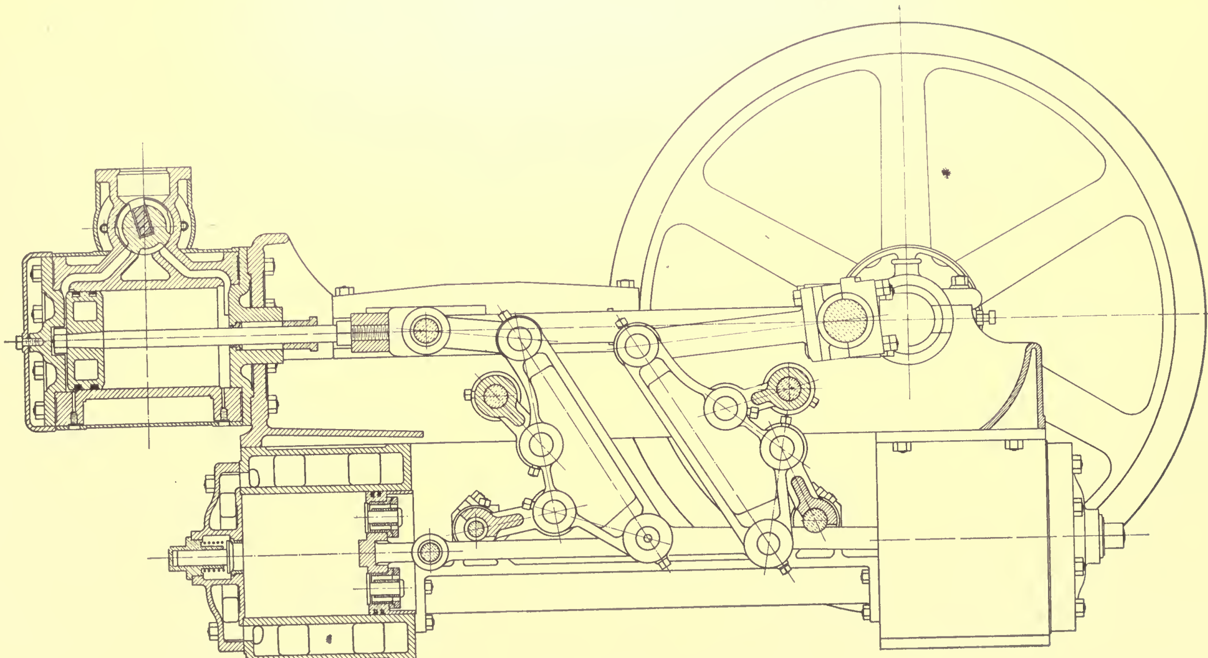
In redesigning or modifying an existing machine, never make a change merely for the sake of doing so. Give the good points of the machine a chance, and devote attention in the new design to correcting the bad points. It is in bad taste, if it be not actually childish, to "look wise and suggest a change" in details which happen to have been designed by another party, but which, nevertheless, are by common engineering judgment pronounced good for the special work intended. This element of unfair and selfish criticism has more than a moral bearing. When it is carried into the superintendence of designing work, it extinguishes the personality of the subordinate draftsman; his efficiency as an original thinker is lowered; and narrow designs are produced.

"The best way for a subordinate to dispose of what appears to be a poor suggestion from a superior, is to work it out to the best degree possible." If it turns out to be good the credit of working it out belongs to the man who did it. If it is actually bad, a careful working out will usually develop the fact beyond dispute, and save unprofitable argument. For the success or failure of a machine there is only one argument better than the detail drawings, and that is the machine itself in operation.

Detail drawings, however, are infinitely better prosecutors or defendants than a multitude of wordy counsel.

Summary. The above classification of machinery might be subdivided and extended indefinitely, and on the broad basis on which it is given it doubtless does not cover the entire field. As an illustration, however, not only of types of machinery, but of methods of design and study, it is hoped that it may be of assistance in giving a start to the student of machine design, in whatever class his interests may happen to lie.

It is the general principles of the art which it is important to master. It is not the designing of a locomotive, or a stationary steam engine, or a crane, or an engine lathe, or a rolling mill, which should be sought to be learned, but the designing of *anything* that may confront us. Specializing is sure to come to the designer in the course of his experience, and when it does he merely fits to the particular specialty the principles he knows for all, and practically develops them along that individual line.



AIR COMPRESSOR (N. Y. AIR BRAKE CO.)

MACHINE DESIGN.

PART II.

Introduction. In Part I is illustrated a definite and systematic method of attacking the design of a *machine as a whole.* In Part II the same plan is followed with regard to the detail of its *component parts*; the machine elements which are chosen as illustrations of the method, being the simplest and most familiar forms in common use.

As before, the student must strive to grasp and absorb the *method* of design rather than any specific and established form of a machine part. Part II is not a compendium of design, does not attempt to be complete or exhaustive in any of its chapters, but is condensed and simplified in order to lead the student into systematic mechanical thinking and logical and definite action. Each chapter is intended to stimulate to further and more exhaustive study along lines broader than, and under conditions different from those that can be specified in a general discussion. But no matter how deeply investigation may be carried, or how specialized the study may become, the student must realize that his path of action in any case whatsoever must lie along the lines of *Analysis*, *Theory*, and *Practical Modification* systematically applied.

BELTS.

NOTATION—The following notation is used throughout the chapter on Belts:

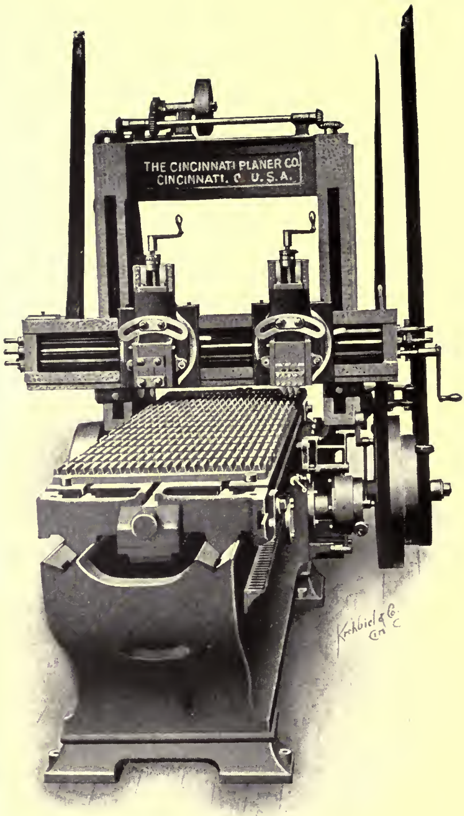
A = Sectional area of belt (square inches) $= bh$.
b = Width of belt (inches).
F = Force of friction at pulley rim (lbs.).
h = Thickness of belt (inches)
μ = Coefficient of friction.
N = Number of revolutions of pulley per minute.
n = Fraction of circumference of pulley embraced by belt.
P = Driving force at pulley rim (lbs.) $= F$.
R = Radius of pulley (feet).
r = Radius of pulley (inches).
T = Initial tension (lbs.).
T_n = Total tension on tight side (lbs.).
T_o = Total tension on slack side (lbs.).
t = Working tension of belt (lbs. per square inch).
V = Velocity of belt (feet per minute).
w = Weight of belt per cubic inch (lbs.).
z = Factor due to centrifugal force.

ANALYSIS. When a belt is stretched over a pair of pulleys, is cut off at the proper length, and is laced together into an endless band, it is evident that as long as the belt is at rest there is a nearly uniform tension in it throughout its length, due to the tightness with which the lacing is drawn up. If the distance between the pulleys is considerable, the weight of the belt itself as it hangs between the pulleys will produce a slightly greater tension next to the pulleys than exists in the middle of the span. This increase of tension due to the weight of the belt would make but little difference in the unit-stress in the material of which the belt is made; hence it may safely be assumed that the tension in the belt when at rest is uniform throughout its entire length.

When we start to transmit power through the belt by turning one of the pulleys, thereby driving the other pulley the condition of stress in the belt is at once materially changed. As the belt is a flexible member, we can transmit only a pull to the other pulley, thereby turning it around, the push which is at the same time given to the other side of the belt merely acting to make the belt sag or become slack. Hence the immediate effect of starting motion in a belt is to change the condition of equal tension throughout its length, to that of unequal tension in the two sides. The driving side is tight, while the other is loose, the former having gained as much tension as the latter has lost, and the sum of the two being practically equal to the sum of the tensions in the two sides of the belt when at rest. This is not strictly true, as will be shown later; but it is sufficiently accurate to form a good basis for the practical design, at least of slow-speed belts.

This condition of tight and slack sides is made possible by the fact that the belt, in being wrapped around the pulleys under tension, has friction on their surfaces. Thus, we can pull hard on one side without slipping the belt around the pulleys, but could not do this if the pulleys were perfectly smooth or frictionless, for in that case the slightest pull on one side would slip the belt around the pulleys. In fact, it would be impossible to produce any pull by means of the driving pulley, for the pulley would merely slip around inside the belt.

The amount of pull we can apply to the belt is therefore limited by the tension at which the belt slips around the pulley. Moreover, since the force of friction between the belt and pulley is dependent upon the normal force with which the belt is pressed against the pulley, and the coefficient of friction between the two, it is evident that the tighter the belt is laced up, and the rougher the surfaces of the pulley and belt, the greater is the force that can be transmitted through the belt. This leads to the conclusion that it would be possible to transmit any amount of power through



RACK CUTTING PLANER.

any belt however small, if the belt were only laced up tight enough.

This conclusion is literally true; but the important fact now comes in, that the strength of the material of which the belt is made is limited, and while theoretically we might be able to accomplish the above, it would be impossible to do so in practice, for at a certain point the belt would break under the strain. Other practical considerations also come in, which fix this limit of power transmission at a point far below the breaking strength of the material.

The complete analysis is not quite as simple as the above, especially for high-speed belts. When the driving side of the belt becomes tight, it stretches and grows longer; and at the same time the other side of the belt becomes slack and grows shorter. But it is not true that the increase in the one side is the same as the decrease in the other, and this fact produces the condition that the sum of the tensions in motion is not quite the same as the sum of the tensions at rest.

Again, when the belt, as it passes around the pulley, changes its straight-line direction to circular motion, each particle of the belt—like a body whirling at the end of a cord about a center of rotation—tends by centrifugal force to fly away from the surface of the pulley, thereby decreasing the normal pressure, and hence the friction. This centrifugal force also changes somewhat the tensions in the belt between the pulleys. As the centrifugal force increases in proportion to the square of the linear velocity, it is evident that the effect is greater at high speeds than at moderate or low speeds.

A further circumstance that affects the driving power of a belt is the stiffness of the leather or other material of which the belt is made. As it passes around the pulley, the belt is bent to conform to the circumference of the pulley, and is again straightened out as it leaves the pulley. Hence the theoretically perfect action is modified somewhat according to the sharpness of the bending and the thickness or flexibility of the belt; in other words, a small pulley carrying a thick belt would be the worst case for successful calculation on a theoretical basis.

THEORY. The condition of the tight and loose sides of a belt transmitting power, is similar to that of the weighted strap and fixed pulley shown in Fig. 17. If motion is desired of the strap around the pulley, it is necessary to make the weight W_2 of such a magnitude that it will overcome not only the weight W_1, but also the friction between the strap and the pulley. The strap tension T_n is, of course, equal to W_2, and T_o to W_1. The equation showing the balance of forces for the condition when motion is about to occur, is:

$$T_n - T_o = F = P \text{ (driving force).} \qquad \textbf{(5)}$$

If the pulley be free to turn on its axis, instead of being fixed as in Fig. 17, the strap by its friction on the pulley will turn the pulley, and the force of friction F becomes the driving force for the pulley as noted in equation 5 above.

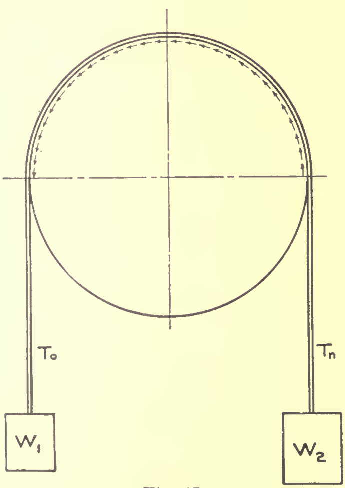


Fig. 17.

In Fig. 18, let us suppose that W is a weight representing the resistance to be overcome. The tensions T_n and T_o, equal at first owing to stretching the belt tightly over the pulleys at rest, change when an attempt is made to raise the weight by turning the larger pulley; and just as the weight leaves the floor, the equality of moments about the axis of the driven pulley gives the following equation:

$$(T_n - T_o) r = F \times r = P \times r = W \times r_1. \qquad \textbf{(6)}$$

This equality of moments remains as long as the motion of the weight is uniform, and represents closely the conditions under which belt pulleys work.

Although we know from the above what the *difference* of the belt tensions is, and what this difference will do when applied to the surface of a given pulley, we do not yet know what either T_n or T_o actually is; and until we do know, we cannot correctly proportion the belt. Hence we must find another relation between T_n and T_o which we can combine with equations 5 and 6. This relation is deduced by a process of higher mathematics, which results as follows:

$$\text{Common logarithm } \frac{T_n}{T_o} = 2.729\ \mu\ (1 - z)n. \qquad \textbf{(7)}$$

Treating equations 5 and 7 as simultaneous, values of both T_n and T_o can be found by the regular algebraic solution. As T_n is the larger, the actual area of belt to provide the necessary strength must be made to depend upon it.

The factor z in equation 7 depends upon the centrifugal force

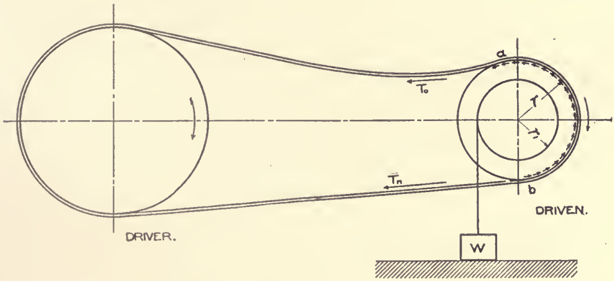


Fig. 18.

developed by the weight of the belt passing around the pulley. Its value, found from mechanics, is:

$$z = \frac{w \times V^2}{9{,}660 \times t}.$$

Having found the maximum pull on the belt, it now remains to write the equation:

External force = Internal resistance;

or, $$T_n = b \times h \times t. \qquad \textbf{(8)}$$

Usually the most convenient way to handle this equation is to assume h and t, and then solve for b.

Summing up the theoretical treatment of belt design, we simply combine equations 5, 6, 7, and 8, and solve for the quantity desired. Discussion of the constants involved in these equations, and of the practical factors controlling them, is given in the following:

PRACTICAL MODIFICATION. The force of friction F, which is the same as driving force P, depends on:

Coefficient of friction (μ) between belt and pulley;

Tightness of the belt;

Centrifugal force of the belt;

Angle of contact of belt with pulley.

The coefficient of friction (μ), according to experiments and observed operation of belts transmitting power, varies from .15 to .56 for leather on cast iron. An average value consistent with a reasonable amount of slip, the belt being in good running order, is .30. If the belt is oily, or likely to become so in use, a lower value should be taken.

The tighter the belt is drawn up, the greater is the pressure against the pulley, and hence the greater is the force of friction. But if we pull the belt up too tightly, when we begin to drive, T_n becomes too great, and the belt breaks or is under such stress that it wears out quickly. Moreover, the great side pressure on the bearings carrying the shaft produces excessive friction, and the drive is inefficient. This is why a narrow belt driven at high speed is more efficient than a wide belt at slow speed, for we cannot pull up the former as tightly as the latter without overstraining it, and yet it is possible to get the required power out of the narrow belt by running it at high speed.

The centrifugal force is of small importance for low speeds, say of 3,000 feet per minute and less; and it therefore may usually be neglected. The factor z then becomes zero in the expression $1 - z$ in equation 7, and the second member of the equation stands simply $2.729 \times \mu \times n$.

The angle of contact of belt with pulley is important, as a large value gives a great difference between T_n and T_o; and it is desirable to make this difference as great as possible, because thereby the driving force is increased. The loose side of a horizontal belt should always be above, as then the natural sag of the loose side due to its slackness tends to increase the angle of contact with the pulley, while the tightening up of the lower side acts against its sag to make the loss of wrap as little as possible. Vertical belts which have the driving pulley uppermost, utilize the weight of the belt to increase the pressure against the surface of the pulley, slightly increasing its capacity for driving. The angle of contact may be artificially increased by a tightening pulley which presses the belt further around the pulley than it would naturally lie. It adds however, the friction of its own bearing, and impairs the efficiency of the drive. For ordinary horizontal belts, the angle of contact is but little more than 180°, and the value of n in equation 7 may be safely assumed at $\frac{1}{2}$ unless the pulleys are of relatively great difference of diameter and very close together.

Strength of Leather Belting. The breaking tensile strength of leather belting varies from 3,000 to 5,000 pounds per square inch. Joints are made by lacing, by metal fasteners, or by cementing. The strength of a laced joint may be about $\frac{7}{10}$, of a metal-fastened joint, about $\frac{1}{2}$, and of a cemented joint, about equal to the full strength of the belt cross-sectional area. The proper working strength of belting depends on the use to which the belt is put. A continuously running belt should have a low tension in order to have long life and a minimum loss of time for repairs. For double leather belting it has been shown that a working tension of 240 pounds per square inch of sectional area gives an annual cost — for repairs, maintenance, and renewals — of 14 per cent of first cost. At 400 pounds working tension, the annual expense becomes 37 per cent of first cost. These results apply to belts running continuously; larger values may be used where the full load comes on but a short time, as in the case of dynamos.

Good average values for working tensions of leather belts are:

Cemented joints,	400	pounds	per	square	inch.
Laced joints,	300	"	"	"	"
Metal joints,	250	"	"	"	"

Horse-Power Transmitted by Belting. If P is the driving force in pounds at the rim of the pulley, and V is the velocity of the belt in feet per minute, the theoretical horse-power transmitted is evidently:

$$H.\ P. = \frac{P \times V}{33,000}. \qquad (9)$$

It is evident from the above that the horse-power of a belt depends upon two things, the driving force P and the velocity V. If either of these factors is increased, the horse-power is increased. Increasing P means a tight belt. Hence a tight belt and high speed together give maximum horse-power. But a tight belt means more side strain on shaft and journal. Therefore, from the standpoint of efficiency, *use a narrow belt under low tension at as high a speed as possible.*

Empirical rules for horse-power of belting, if used with judgment, give safe results when applied to very general cases. A common rule used by American engineers is:

$$H.\ P. = \frac{b \times V}{1,000}. \qquad (10)$$

For a double belt, assuming double strength, this becomes:

$$H.\ P. = \frac{b \times V}{500}. \qquad (11)$$

With large pulleys and moderate velocities, this may hold good. With small pulleys and high velocities, however, the uncertain stresses induced by the bending of the fibers of the belt around the pulley, and the relatively great loss due to centrifugal force, modify this relation and a safer value for a double belt of the ordinary kind is:

$$H.\ P. = \frac{b \times V}{540}; \qquad (12)$$

or, still safer,

$$H.\ P. = \frac{b \times V}{700}. \qquad (13)$$

If we compare the theoretical value of equation 9 with the empirical value of equation 10 by putting them equal to each other, thus:

$$H.\ P. = \frac{P \times V}{33,000} = \frac{b \times V}{1,000},$$

and solve for P, we get :

$$P = 33b. \qquad (14)$$

This develops the fact that the empirical rule of equation 10 assumes a driving force of 33 pounds per inch of width of single belt.

Another way of expressing equation 10 is: A single belt will transmit one horse-power for every inch of width at a belt speed of 1,000 feet per minute.

Speed of Belting. The most economical speed is somewhere between 4,000 and 5,000 feet per minute. Above these values the life of the belt is shortened; also "flapping," "chasing," and centrifugal force cause considerable loss of power. The limit of speed with cast-iron pulleys is fixed at the safe limit for bursting of the rim, which may be taken at one mile per minute.

Material of Belting. Oak-tanned leather, made from the part of the hide which covers the back of the ox, gives the best results for leather belting. The thickness of the leather varies from .18 to .25 inch. It weighs from .03 to .04 pound per cubic inch. The average thickness of double leather belts may be taken as .33 inch, although a variation in thickness from $\frac{1}{4}$ inch to $\frac{7}{16}$ inch is not uncommon. Double leather belts may be ordered light, medium, or heavy.

In a single-thickness belt the grain or hair side should be next to the pulley, for the flesh side is the stronger and is therefore better able to resist the tensile stress due to bending set up where the belt makes and leaves contact with the pulley face. Double leather belts are made by cementing the flesh sides of two thicknesses of belt together, leaving the grain side exposed to surface wear.

Raw hide and semi-raw hide belts have a slightly higher coefficient of friction than ordinary tanned belts. They are useful in damp places. The strength of these belts is about one and one-half times that of tanned leather.

Cotton, cotton-leather, rubber, and leather link belting are some of the forms on the market, each of which is especially adapted to certain uses. For their weights and their tensile and working strengths consult the manufacturers' catalogues.

A prominent manufacturer's practice in regard to the sizes of leather belting will be found useful for comparison, and is indicated in the table on page 12.

Initial Tension in Belt. On the assumption that the sum of the tensions is unchanged, whether the belt be at rest or driving, we should have the following relation :

$$T_n + T_o = 2T;$$

whence,

$$T = \frac{T_n + T_o}{2}. \qquad (15)$$

This is not strictly true, however, as is stated in the "Analysis" of "Belts." It has been found that in a horizontal belt working at about 400 lbs. tension per square inch on the tight side, and having 2 per cent slip on cast-iron pulleys (*i. e.*, the surface of the

Sizes of Leather Belting.

WIDTH.	THICKNESS.	
	Single.	Double.
1 inch.	$\frac{5}{32}$ inch.	$\frac{5}{16}$ inch.
2 "	$\frac{3}{16}$ "	$\frac{5}{16}$ "
3 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
4 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
5 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
6 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
10 "	$\frac{5}{16}$ "	$\frac{3}{8}$ "
12 "	....	$\frac{3}{8}$ "
14 "	....	$\frac{13}{32}$ "
20 "	....	$\frac{7}{16}$ "

driven pulley moving 2 per cent slower than that of the driver), the increase of the sum of the tensions when in motion over the sum of the tensions at rest, may be taken at about $\frac{1}{3}$ the value of the tensions at rest. Expressing this in the form of an equation

$$T_n + T_o = \frac{4}{3}(2T) = \frac{8 \times T}{3}.$$

$$T = \frac{3}{8}(T_n + T_o). \qquad (16)$$

The value of T thus found would be the pounds initial tension to which the belt should be pulled up when being laced, in order to produce T_n and T_o when driving.

This value is not of very great practical importance, as the proper tightness of belt is usually secured by trial, by tightening pulleys, by pulley adjustment (as in motor drives), or by shortening the belt from time to time as needed. It is worth noting, however, that for the most economical life of the belt it would be very desirable in every case to weigh the tension by a spring balance when giving the belt its initial tension. This, however, is not always easy or even feasible; hence it is a refinement with which good practice usually dispenses, except in the case of large and heavy belts.

PROBLEMS ON BELTS.

1. Determine the belt tensions in a laced belt transmitting 50 horse-power at a velocity of 3,500 feet per minute. Suppose that the arc of contact is 180°; weight of belt = .035 pound per cub. in.; and coefficient of friction 25 per cent.

2. What is the width of above belt if it is $\frac{3}{16}$ inch in thickness?

3. What initial tension must be placed on above belt?

4. The main drive pulley of a 120-horse-power water wheel is 6 feet in diameter. A cemented leather belt is to connect the main pulley to a 3-foot pulley on the line shafting in a mill. The horizontal distance between centers of shafting is 24 feet; coefficient of friction, 30 per cent; revolutions per minute of line shafting, 180. Design the belt for this drive.

5. An 8-inch double belt $\frac{3}{8}$ inch thick connects 2 pulleys of 30-inch and 20-inch diameter respectively. The horizontal distance between the centers is 12.5 feet. The coefficient of friction is 0.3, and the weight of belt per cubic inch is 0.035 pound. Working tension, 300 pounds per square inch. Speed of belt 5,000 feet per minute. Lower face of 30-inch pulley is the driving face. Required the H. P. which may be transmitted (theoretically).

6. Compare the theoretical horse-power in problem 5 with that obtained by the use of empirical formula.

PULLEYS.

NOTATION—The following notation is used throughout the chapter on Pulleys:

A = Area of rim (sq. in.).
a = " " arm (" ").
b = Center of pulley to center of belt (inches; practically equal to R).
C_1 = Total centrifugal force of rim (lbs.).
c = Distance from neutral axis to outer fiber (inches).
D = Diameter of pulley (inches).
D_1 = " " hub (").
d_1 = " " bolt at root of thread (inches).
d = Diameter of bolt holes (inches).
g = Acceleration due to gravity (ft. per sec.).
h = Width of arm at any section (inches).
I = Moment of inertia
L = Length of arm, center of belt to hub (inches).
L_1 = Length of rim flange of split pulley (inches).
l = Length of hub (inches).
N = Number of arms.
n = " " rim bolts, each side.
P = Driving force of belt (lbs.).
P_1 = Force at circumference of shaft (lbs.).
P_2 = Force at circumference of hub (lbs.).
p = Stress in rim due to centrifugal force (lbs. per sq. in.).
R = Radius of pulley (inches).
S = Fiber stress (lbs. per sq. in.).
s = Fiber stress in flange (lbs. per sq. in.).
T = Thickness of web (inches).
t = " " rim (").
t_2 = " " " bolt flange (inches).
T_n = Tension of belt on tight side (lbs.).
T_0 = " " " " loose " (").
v = Velocity of rim (ft. per sec.).
w = Weight of material (lbs. per cub. in.).

ANALYSIS. If a flexible band be wrapped *completely* about a pulley, and a heavy stress be put upon each end of the band, the rim of the pulley will tend to collapse just like a boiler tube with steam pressure on the outside of it. A compressive stress is induced which is very nearly evenly distributed over the cross-section of the rim, except at points where the arms are connected thereto. At these points the arms, acting like rigid posts, take this compressive stress. Now, a pulley never has a belt wrapped *completely* round it, the fraction of the circumference embraced by the belt being usually about $\frac{1}{2}$, and seldom, even with a tightener pulley, reaching $\frac{3}{4}$. Assuming the wrap to be $\frac{1}{2}$ the circumference, and that all the side pull of the belt comes on the rim, none being transmitted through the arms to the hub, we then have one-half of the rim pressed hard against the other half by a force equal to the resultant of the belt tensions, which, in this case, would be the sum of them. Dividing the pulley by a plane through its center and perpendicular to the belt, the cross-section of the rim cut by this plane has to take this compressive stress.

This analysis is satisfactory from an ideal standpoint only, for the intensity of stress due to the direct pull of the belt, with the usual practical proportions of rim, would be very small. Moreover, the element of speed has not been considered.

When the pulley is under speed, a set of conditions which complicates matters is introduced. The centrifugal force due to the weight of the rim and arms is no longer negligible, but has an important influence upon the design and material used. This centrifugal force acts against the effect of the belt wrap, tending to reduce the compressive stress, or, overcoming the latter entirely, sets up a tensional stress both in the rim and in the arms. It also tends to distort the rim from a true circle by bowing out the rim between the arms, thus producing a bending moment in the rim, maximum at the points where the rim joins each arm.

It can readily be imagined that the analysis in detail of these various stresses in the rim acting in conjunction with each other is quite complicated — far too much so in fact, to be introduced here. As in most cases of such design, however, one controlling influence can be separated out from the others, and the design based thereon with sufficient margin of strength to satisfy the more obscure conditions. This is rational treatment, and the "theory" will be studied accordingly.

The rim, being fastened to the ends of the arms, tends, when driving, to be sheared off, the resisting area being the areas of the cross-sections of the arms at their point of joining the rim. The force that produces this shearing tendency is the driving force of the belt, or the difference between the tensions of the tight and loose sides.

Again, at the point of connection of the arms to the hub, a shearing action takes place, so that, if this shearing tendency were carried to rupture, the hub would literally be torn out of the arms. Now, viewing the arms as beams loaded at the end with the driving force of the belt, and fixed at the hub, a heavy bending stress is set up, which is maximum at the point of connection to the hub. If the rim were stiff enough to distribute this driving force equally between the arms, each arm would take its proportional share of the load. The rim, however, is quite thin and flexible; and it is not safe to assume this perfect distribution. It is usual to consider that one-half the whole number of arms take the full driving force.

THEORY—Pulley Rim. Evidently it is practically impossible to make so thin a rim that it will collapse under the pull of a belt. As far as the *theory* of the rim is concerned, its proportion probably depends more upon the calculation for centrifugal force than upon anything else.

In order to separate this action from that of any other forces, let us suppose that the rim is entirely free from the arms and hub, and is rotating about its center. Every particle, by centrifugal force, tends to fly radially outward from the center. This condition is represented in Fig. 19. The tendency with which one-half of the rim tends to fly apart from the other is indicated by the force C_1; and the relation between C_1 and the small radial force c for each unit-length of rim can readily be found from the principles of mechanics. The case is exactly like that of a boiler or a thin pipe subjected to uniform internal pressure, which, if carried to rupture, would split the rim along a longitudinal seam.

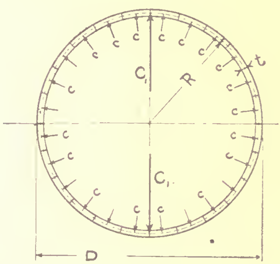


Fig. 19.

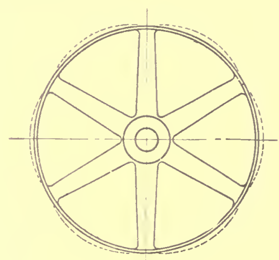


Fig. 20.

The tensile stress thus induced per square inch can be found by simple mechanics to be:

$$p = \frac{12wv^2}{g}; \qquad (17)$$

or, since $w = 0.26$ pound, and $g = 32.2$ feet per second,

$$p = 0.097\,v^2 \quad \left(\text{say } \frac{v^2}{10}\right); \qquad (18)$$

and, if p be taken equal to 1,000 pounds per square inch, which is as high as it is safe to work cast iron in this place,

$$v = 100 \text{ feet per second.} \qquad (19)$$

This shows the curious fact that the intensity of stress in the rim is directly proportional to the square of the linear velocity, and wholly independent of the area of cross-section. It is also to be noted that 100 feet per second is about the limit of speed for cast-iron pulleys to be safe against bursting.

If we wish to consider theoretically the rim together with the arms as actually connected to it, we get a much more complicated relation. This condition is shown in Fig. 20, where the rim, expanding more than the arms, bulges out between them. This makes the rim act something like a continuous beam uniformly loaded; but even then the resulting stress is not clearly defined on account of the variable stretch in the arms. Investigation on this basis is not needed further than to note that it is theoretically better, in the case of a split pulley, to make the joint close to the arms, rather than in the middle of a span.

Pulley Arms. The centrifugal force developed by the rim and arms tends to pull the arms from the hub. On the belt side, this is balanced to some extent by the belt wrap, which tends to compress the arm and relieve the tension. On the side away from the belt, the centrifugal action has full play, but the arm is usually of such cross-section that the intensity of this stress is very low. It may safely be neglected.

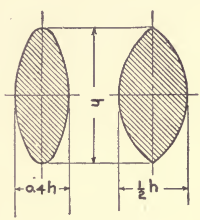


Fig. 21.

The rim being very thin in most cases, its distributing effect cannot be depended on, hence the driving force of the belt may be taken entirely by the arms immediately under the portion of the belt in contact with the pulley face. For a wrap of 180° this means that only one-half of the pulley arms can be considered as effective in transmitting the turning effort to the hub. Each of these arms is a lever fixed at one end to the hub and loaded at the other. A lever of this description is called a "cantilever" beam, its maximum moment existing at its fixed end. The load that each of these beams may be subjected to is $\frac{P}{\frac{N}{2}}$, and therefore the maximum external moment at the hub is $\frac{2PL}{N}$. From mechanics we know that the internal moment of resistance of any beam section is $\frac{SI}{c}$, and that equilibrium of the beam can be satisfied only when the external moment is equal to the internal moment of resistance of the beam section. Equating these two, we have:

$$\frac{2PL}{N} = \frac{SI}{c}. \qquad \textbf{(20)}$$

The arms of a pulley are usually of the elliptical or segmental cross-section, and may be of the proportions shown in Fig. 21. For either of these sections the fraction $\frac{I}{c}$ is approximately equal to $0.0393h^3$. For convenience (the error caused being on the safe side), L may be taken as equal to the full radius of the pulley R, whence

$$\frac{2PR}{N} = \frac{2(T_n - T_o)R}{N} = 0.0393Sh^3, \qquad \textbf{(21)}$$

in which S may be from 2,000 to 2,250 for cast iron

Taking moments about the center of the pulley, and solving for P_2, the force acting at the circumference of the hub, we have:

$$\frac{2PR}{N} = \frac{P_2D_1}{2};$$

or

$$P_2 = \frac{4PR}{ND_1} \qquad \textbf{(22)}$$

The area of an elliptical section is π times the product of the half axes. With the proportions of Fig. 21, this becomes:

$$a = \pi \times 0.2h \times 0.5h = \frac{\pi h^2}{10} \qquad \textbf{(23)}$$

Equating the external force to the internal shearing resistance, we have:

$$\frac{4PR}{ND_1} = \frac{\pi h^2 S_s}{10}$$

or,

$$S_s = \frac{40PR}{D_1 N \pi h^2}, \qquad \textbf{(24)}$$

in which the shearing stress S_s may run from 1,500 to 1,800 for cast iron.

Although both bending and shearing stresses as calculated above exist at the base of the arms, the bending is, in practically every case, the controlling factor in the design of the arms. An arm-section large enough to resist bending would have a very low intensity of shear.

If the number of arms be increased indefinitely, we come to a continuous arm or web, in which the bending action is eliminated. It may still shear off at the hub, where the area of metal is the least, at minimum circumference. In this case the area under shearing stress is $\pi D_1 T$; and the force at the circumference of the hub, is

$$\frac{PR}{\frac{D_1}{2}} = \frac{2PR}{D_1}$$

Equating external force to internal shearing resistance, we have:

$$\frac{2PR}{D_1} = \pi D_1 T S_s;$$

$$\text{or,} \quad S_s = \frac{2PR}{\pi D_1^2 T}. \quad \textbf{(25)}$$

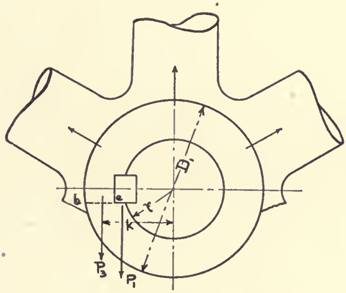


Fig. 22.

Pulley Hub. As in the case of the arms, centrifugal force does not play much part in the design of the hub of a pulley. The hub is designed principally to carry the key, and through it transmit the turning moment to the shaft. Considered thus, the hub may tear along the line of the key or crush in front of the key.

For example, in Fig. 22, if the connection with the lower arms be neglected, and the upper arms be held fast while a turning force P_1, at the surface of the shaft, is transmitted to the hub through the key, then the metal of the hub directly in front of the key is under crushing stress; and the metal along the line *eb*, from the corner to the outside, is under tensile stress. This condition is the worst that could possibly happen, because the bracing effect of the lower arms has been neglected, and the key is located between the arms.

Taking moments about the center of the shaft, the value of the force at the shaft circumference, or the "key pull," is:

$$P_1 = \frac{PR}{r}. \qquad (26)$$

Now $\frac{P_1}{P_3} = \frac{k}{r}$, k being the distance from the center of shaft to center of eb, and the area of metal which is subjected to the tearing action P_3 is $l \times eb$. Equating the external force to the internal resistance, and assuming that the stress is equally distributed over the area $l \times eb$, we have:

$$P_3 = \frac{r}{k} P_1 = \frac{r}{k} \times \frac{PR}{r} = S \times l \times eb;$$

or,

$$S = \frac{PR}{k \times l \times eb}. \qquad (27)$$

The intensity of crushing on the metal in front of the key, due to force P_1, depends upon the thickness of the key, and is properly discussed later under "Keys."

PRACTICAL MODIFICATION—Pulley Rim. The theoretical calculation for the thickness of the rim may give a thickness that could not be cast in the foundry, and the section in that case will have to be increased. As light a section as can be readily cast will usually be found abundantly strong for the forces it has to resist. A minimum thickness at the edge of the rim is about $\frac{3}{16}$ inch; and as the pulleys increase in size, the rim also must be made thicker; otherwise the rim will cool so much more quickly than the arms, that the latter, on cooling, will develop shrinkage cracks at the point of junction.

For a velocity of 6,000 feet per minute, we find from equation 18 that the tension in pounds per square inch, in the rim, due to centrifugal force, is 970. Though this in itself is a low value, yet the uncertain nature of cast iron, its condition of internal stress, due to casting, and the likely existence of hidden flaws and pockets, have established the usage of this figure as the highest safe limit for the peripheral speed of cast-iron pulleys. It is easily remembered that *cast-iron pulleys are safe for a linear velocity of about one mile per minute.*

To prevent the belt from running off the pulley, a "crown" or rounding surface is given the rim. A tapered face, which is more easily produced in the ordinary shop, may be used instead. This taper should be as little as possible, consistent with the belt staying on the pulley; $\frac{1}{2}$ inch per foot each way from the center is not too much for faces 4 inches wide and less; while above this width $\frac{1}{4}$ inch per foot is enough. As little as $\frac{1}{8}$ inch total crown has been found to be sufficient on a 24-inch face, but this is probably too little for general service.

Instead of being "crowned," the pulley may be flanged at the edges; but flanged pulley rims chafe and wear the edge of the belt.

The inside of the rim of a cast-iron pulley should have a taper of $\frac{1}{2}$ inch per foot to permit easy withdrawal from the foundry

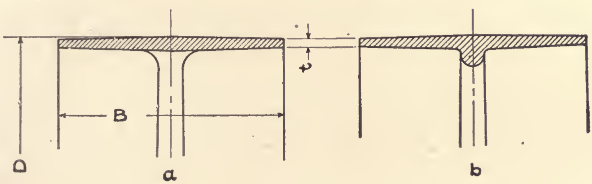


Fig. 23.

mould. This is known as "draft." If the pattern be of metal, or if the pulley be machine-moulded, the greater truth of the casting does not require that the inside of the rim be turned, as the pulley, at low speeds, will be in sufficiently good balance to run smoothly. For roughly moulded pulleys, and for use at high speeds, however, it is necessary that the rim be turned on the inside to give the pulley a running balance.

Fig. 23 shows a plain rim a also one stiffened by a rib b. Where heavy arms are used this rib is essential so that there will not be too sudden change of section at the junction of rim and arm. and consequent cracks or spongy metal.

Pulley Arms. The arms should be well filletted at both rim and hub, to render the flow of metal free and uniform in the mould. The general proportions of arms and connections to both hub and rim may perhaps be best developed by trial to scale on the drawing board. The base of the arm being determined, it may gradually taper to the rim, where it takes about the relation of $\frac{2}{3}$ to $\frac{3}{4}$ the dimensions chosen at the hub. The taper may be modified until it looks right, and then the sizes checked for strength.

Six arms are used in the great majority of pulleys. This number not only looks well, but is adapted to the standard three-jawed chucks and common clamping devices found in most shops. Elliptical arms look better than the segmental style. The flat, rectangular arm gives a very clumsy and heavy appearance, and is seldom found except on the very cheapest work.

A double set of arms may be used on an excessively wide face, but it complicates the casting to some extent.

Although a web pulley may be calculated for shear at the hub, yet it will usually be found that with a thickness of web intermediate between the thickness of the rim and that of the hub, which will satisfy the casting requirements, the requirements as to strength will be fully met.

Pulley Hub. The hub should have a taper of $\frac{1}{2}$ inch per foot draft, similar to that of the inside of the rim. The length of the hub is arbitrary, but should be ample to prevent rocking on the shaft. A common rule is to make it about $\frac{3}{4}$ the face width of the pulley.

The diameter of the hub, aside from the theoretical consideration given above, must be sufficient to take the wedging action of a taper key without splitting. This relation cannot well be calculated. Probably the best rule that exists is the familiar one that the hub should be twice the diameter of the shaft. This rule, however, cannot be literally adhered to, as it gives too small hubs for small shafts and too large ones for large shafts. It is always well to locate the key, if possible, underneath an arm instead of between the arms, thus gaining the additional strength due to the backing of the arm.

SPLIT PULLEYS.

ANALYSIS and THEORY. The split pulley is made in halves and provided with bolts through flanges and bosses on the hub for holding the two halves together. When the pulley is in place on the shaft, bolted up as one piece, it is subjected to the same forces as the simple pulley. Hence its general design follows the same principles, and we need only study the fastening of the two halves, and the effect of this fastening on the detail of rim and hub.

The simplest stress we have to consider on the rim bolts is one of pure tension, due to the centrifugal force of the halves of the pulley, A safe assumption to make is that the rim is free

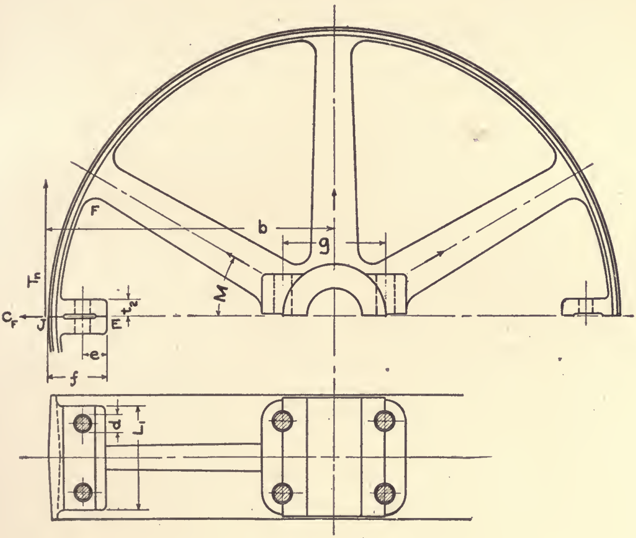


Fig. 24.

from the arms and hub, as in the simple pulley, and that the centrifugal force developed by it has to be taken by the rim bolts alone. In other words, consider the rim bolts as belonging entirely to the rim, and make them as strong as the rim, leaving the hub bolts to take the centrifugal force of the arms and hub, and the spreading tendency due to the key.

Another tensile stress is induced in the rim bolts by the fact, that, having made an open joint in the rim, and in addition placed the extra weight of lugs there, the centrifugal action at this point is increased, and at the same time a point of weakness in the rim introduced. Referring to Fig. 24, the rim flanges EJ tend to fly out due to the centrifugal force C_F. This tends to open the joint J at the outside of the rim ; to throw a bending stress on the rim, maximum at the point F ; and to "heel" the rim flanges about the point E. The rim bolts acting on the leverage e about the point E must resist these tendencies, and are thereby put in tension.

Referring to equation 18, we find the intensity of stress due to the centrifugal force of the rim in lbs. per square inch to be :

$$p = \frac{v^2}{10}.$$

If A is the sectional area of the rim in square inches, this means that the total strength of the rim is represented by $\frac{Av^2}{10}$. The strength of a bolt is represented by the expression $\frac{S\pi d_1^2}{4}$. If, now, there are n bolts in the flange, the total resisting force of the bolts is $\frac{nS\pi d_1^2}{4}$; and the equation representing equality of strength between rim and bolts is :

$$\frac{Av^2}{10} = \frac{nS\pi d_1^2}{4}, \quad (28)$$

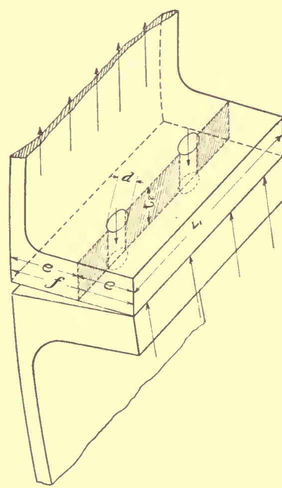


Fig. 25.

from which, by a proper assumption of the fiber stress S, which should be low, the opening-up tendency of the joint being neglected, the diameter at the root of the thread d_1 may be calculated, and the nominal bolt diameter chosen. Reference to the table for strength of bolts, given in the chapter on Bolts, Studs, etc., will be found convenient.

It is very doubtful if the tension on the flange bolts, due to the "heeling" about E can be calculated with sufficient accuracy to be of much value. It is probably better to assume S at a low value, say 4,000, and, in addition, for large and high-speed pulleys, stiffen the rim by running a rib between the flange and the adjacent arm. It is evident that if we make the rim so stiff that it cannot deflect, there will be no "heeling" about E; and the bolts will be well proportioned by the preceding calculation, giving them equal strength to that of the rim section.

For the bolt flange itself, any tendency to open at the joint J would cause it to act like a beam loaded at some point near its middle with the bolt load, and supported at J and E. This condition is shown in Fig. 25. Probably the weakest section would be along the line of the bolt centers. We have just noted that the carrying capacity of the bolts is $\frac{nS\pi d_1^2}{4}$. Hence, assuming that $e = \frac{1}{2}f$, which is about the worst case which could happen, we have a beam of length f loaded at the middle with $\frac{nS\pi d_1^2}{4}$ and supported at the ends. Equating the external moment to the internal moment, we have:

$$\frac{nS\pi d_1^2}{4} \times \frac{f}{4} = \frac{s(L_1 - nd)t_2^2}{6}, \qquad (29)$$

from which the fiber stress s in the flange may be calculated and judged for its allowable value.

L_1 may be assumed a little narrower than the pulley face; and t_2 from 1 inch to 2 inches or more, depending on the thickness of the rim.

The hub bolts doubtless assist the rim bolts in preventing the halves of the pulley from flying apart. They also clamp the hub tightly to the shaft, preventing any looseness on the key. Their function is a rather general one; and the specific stress which they receive is practically impossible to calculate. As a matter of fact, if the hub bolts were left out entirely, the pulley would still drive fairly well, but general rigidity and steadiness would be impaired. Hence the size of the hub bolts is more a practical question than one involving calculation. The rim bolts should be figured first, and their size determined on; then the hub bolts can be judged in proportion to the rim bolts, the diameter of shaft, the thickness and length of the hub, and the general form of the pulley. Often appearance is the deciding factor, it being manifestly inconsistent to associate small fastenings with large shafts or hubs, even though the load be actually small.

PRACTICAL MODIFICATION. Practical considerations are chiefly responsible for the location of the joint in a split pulley between the arms instead of directly at the end of an arm, where theoretically it would seem to be required. It is usually more convenient in the foundry and machine shop to have the joint between the arms; so we generally find it placed there, and strength provided to permit this. It is possible, however, to provide a double arm, or a single split arm, in which case the joint of the pulley comes at the arm, and the "heeling" action of the rim flanges is prevented.

The rim bolts should be crowded as close as possible to the rim in order to reduce the stress on them, and also the stress in the flange itself. The practical point must not be forgotten, however that the bolts must have sufficient clearance to be put into place beneath the rim.

While it is evident that the rim bolts are most effective in taking care of the centrifugal action of the halves, yet in small split pulleys it is quite common to omit the rim bolts and to use the hub bolts for the double purpose of clamping the shaft and holding the two halves together. The pulley is cast with its rim continuous throughout the full circle, and it is machined in this form. It is then cracked in two by a well-directed blow of a cold chisel, the casting being especially arranged for this along the division line by cores so set that but a narrow fin of metal holds the two parts together. This provides sufficient strength for casting and turning, but permits the cold chisel to break the connection easily.

SPECIAL FORMS OF PULLEYS.

The plain cast-iron pulley has been used in the foregoing discussion as a basis of design. A pulley is, however, such a common commercial article, and finds such universal use, that special forms, which can be bought in the open market, are not only cheaper but better than the plain cast-iron pulley, at least for regular line-shaft work.

Cast iron is a treacherous and uncertain material for rims of pulleys. It is not well suited to high fiber stresses; hence the range of speed permissible for pulley rims of cast iron is limited. Steel and wrought iron, having several times the tensional strength of cast iron, and being, moreover, much more nearly homogeneous in texture, are well suited for this work; one of the best pulleys on the market consists of a steel rim riveted to a cast-iron spider. Such an arrangement combines strength and lightness, without increasing complication or expense.

The all-steel pulley is a step further in this direction. Here the rim, arms, and hub are each pressed into shape by specially devised machinery, then riveted and bolted together. This pulley is strictly a manufactured article, which could not compete with the simpler forms unless built in large quantities, enabling automatic machinery to be used. Large numbers of pulleys are built in this way, and are put on the market at reasonable prices.

Wood-rim pulleys have been made for many years, and, except for their clumsy appearance, are excellent in many respects. The rim is built up of segments in much the same way as an ordinary pattern is made, the segments being so arranged that they will not shrink or twist out of shape from moisture. The hubs may be of cast iron, bolted to wooden webs, and carrying hardwood split bushings, which may be varied in bore within certain limits so as to fit different sizes of shafting. The wooden pulley is readily and most often used in the split form, thus enabling it to be put in position easily at any point of a crowded shaft. It is often merely clamped in place, thus avoiding the use of keys or set screws, and not burring or roughening the shaft in any way.

PROBLEMS ON PULLEYS.

1. Calculate the tensile stress due to centrifugal force in the rim of a cast-iron pulley 30 inches in diameter, at 500 revolutions per minute.

2. The driving force of a belt on a 36-inch pulley is 800 lbs., and the belt wrap about 180°. Calculate proportions of el-

liptical arms to resist bending, the allowable fiber stress being 2,000.

3. A pulley 12 inches in diameter, $\frac{5}{8}$-inch web, 4-inch diameter hub, transmits 25 horse-power at a belt speed of 3,000 ft. per minute. Calculate the maximum shearing stress in the web.

4. In Fig. 24 assume the following data: $L_1 = 7$ inches; $t_2 = 1$ inch; $e = 1\frac{1}{2}$ inches; $f = 3$ inches; area of rim $= 3$ sq. in.; allowable tensile stress in rim 1,000 lbs. per sq. in. Calculate the diameter of the rim bolts.

5. Calculate the fiber stress in the rim bolt flange along the line of the bolts.

SHAFTS.

NOTATION—The following notation is used throughout the chapter on Shafts:

A_d = Angular deflection (degrees).
B = Simple bending moment (inch-lbs.).
B_e = Equivalent bending moment (inch-lbs.).
c = Distance from neutral axis to outer fiber (inches).
d, d_0, d_2, d_3, d_4 = Diameters of shaft (inches).
d_1 = Internal diameter of shaft (inches).
E = Direct modulus of elasticity (a ratio).
e = Transverse deflection (inches).
G = Transverse modulus of elasticity (a ratio).
H = Horse-power (33,000 ft.-lbs. per minute).
I = Moment of inertia.
K = Distance between bearings (inches).
L = Length along shaft (inches).
L_1, L_2 = Length of bearings (inches.)
M = Distance between bearings (feet.)
N = Number of revolutions per minute.
P = Driving force of belt (lbs.).
P_1 = Load applied as stated (lbs.).
R = Radius at which load as stated acts (inches).
S = Fiber stress, tension, compression, or shearing (lbs. per sq. in.).
T = Simple twisting moment (inch-lbs.).
T_e = Equivalent twisting moment (inch-lbs.).
T_n = Tension in tight side of belt (lbs.).
T_o = Tension in loose side of belt (lbs.).
W = Load applied as stated (lbs.).

ANALYSIS. The simplest case of shaft loading is shown in Fig. 26. The equal forces W, similarly applied to the disc at the distance R from its center, tend to twist the shaft off, the tendency being equal at all points of the length L between the disc and the post, to which the shaft is rigidly fastened. The fastening to the post, of course, in this ideal case, takes the place of a resisting member of a machine. A state of pure torsion is induced in the shaft; and any element, such as *ca*, is distorted to the position *cb*, *aob* being the angular deflection for the distance L.

The case of Fig. 27 is illustrative of what occurs when a belt pulley is substituted for the simple disc. Here the twisting action is caused by the driving force of the belt, which is $T_n - T_o = P$, acting at the radius R. Torsion and angular deflection exist in the shaft, as in Fig. 26. In addition, however, another stress of a different kind has been introduced; for not only does the shaft tend to be twisted off, but the forces T_n and T_o, acting together, tend to bend the shaft, the bending moment varying with every section of the shaft, being nothing at the point *o*, and maximum at the point *c*. This combined action is the most common of any that we find in ordinary machinery, occurring in nearly every case with which we have to deal.

In Fig. 27, if the forces T_n and T_o be made equal, there will be no tendency at all to twist off the shaft, but the bending will remain, being maximum at the point *c*. This condition is illustrative of the case of all ordinary pins and studs in machines. In this sense, a pin or a stud is simply a shaft which is fixed to the frame of the machine, there being no tendency to turning of the pin or stud itself. The same condition would be realized if the disc in Fig. 27 were loose upon the shaft. In that case, the bending moment would be caused by $T_n + T_o$ acting with the leverage L. Of course there would have to be some resistance for $T_n - T_o$ to work against, in order that torsion should not be transmitted through the shaft. This condition might be introduced by having a similar disc lock with the first one by means of lugs on its face, thus receiving and transmitting the torsion.

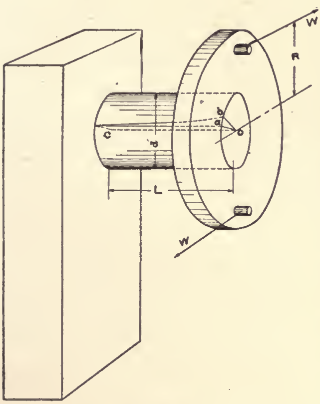


Fig. 26.

If the distance L becomes very great, both the angular deflection due to twisting, and the sidewise deflection due to bending, become excessive, and not permissible in good design. This trouble is remedied by placing a bearing at some point closer to the disc, which, as it decreases L, of course, decreases the bending moment and therefore the transverse deflection. The angular deflection can be decreased only by bringing the resistance and load nearer together.

The above implies, of course, that the diameter of the shaft is not changed, it being obvious that increase of diameter means increase of strength and corresponding decrease of both angular and transverse deflection.

If the speed of the shaft be very high, and the distance between bearings, represented by L, be very great, the shaft will take a shape like a bow string when it is vibrated, and smooth action cannot be maintained.

It is necessary to carry the cases of Figs. 26 and 27 but a

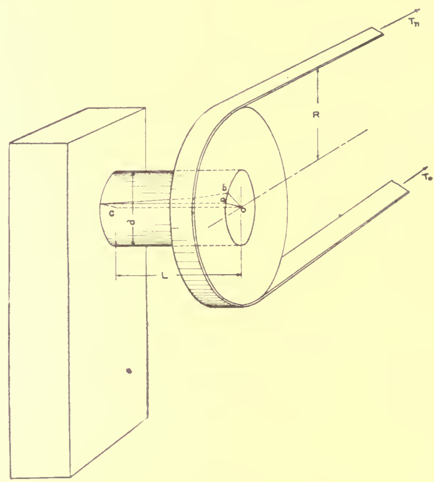


Fig. 27.

single step farther to illustrate the actual working conditions of shafting in machines. Suppose the rigid post to have the shaft passing clear through it, and to act as a bearing, so that the shaft can freely rotate in it, the resistance being exerted somewhere beyond. The twisting moment will be unchanged, also the bending moment; but the effect of the bending moment will be on each particle of the shaft in succession, now putting compression on a given particle, and then tension, then compression again, and so on, a complete cycle being performed for each revolution. This brings out a very important difference between the bending stress in pins and the bending stress in rotating shafts. In the one case the bending stress is non-reversing; in the other, reversing; and a much higher fiber stress is permissible in the former than in the latter.

THEORY—Simple Torsion. In the case of simple torsion the stress induced in the shaft is a shearing one. The external moment acts about the axis of the shaft, or is a polar moment; hence in the expression for the moment of the internal forces, the polar moment of inertia must be used. Now, from mechanics we have:

$$T = \frac{SI}{c};$$

and $$\frac{I}{c} = \frac{d^3}{5.1} \text{ (for circular section of diameter } d\text{);}$$

therefore, $$T = \frac{Sd^3}{5.1}, \qquad (30)$$

from which the diameter for any given twisting moment and fiber stress can readily be found.

For a hollow shaft this expression becomes:

$$T = \frac{S(d_o^4 - d_1^4)}{5.1 d_o}. \qquad (31)$$

Simple Bending. The stresses induced in a pin or shaft under simple bending are compression and tension. The external moment in this case is transverse, or about an axis across the shaft; hence the direct moment of inertia is applicable to the equation of forces.

$$B = \frac{SI}{c};$$

and $$\frac{I}{c} = \frac{d^3}{10.2} \text{ (for circular section of diameter } d\text{);}$$

therefore, $$B = \frac{Sd^3}{10.2}. \qquad (32)$$

For a hollow shaft or pin this expression becomes:

$$B = \frac{S(d_o^4 - d_1^4)}{10.2 d_o}. \qquad (33)$$

Combined Stresses. In the greater number of cases met with in practice, we find two or more simple stresses acting at the same time, and, although the shaft may be strong enough for any one of them alone, it may fail under their combined action. The most common cases are discussed below.

Tension or Pressure Combined with Bending. In Fig. 28, the load W produces a tension acting over the whole area of d, due to its direct pull. It also produces a bending action due to the leverage R, which puts the fibers at B in tension and those at the opposite side in compression. It is evident, therefore, that by taking the algebraic sum of the stresses at either side we shall obtain the net stress. It is also evident that the greatest and

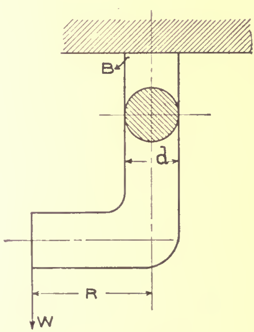


Fig. 28.

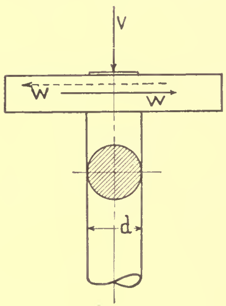


Fig. 29

controlling stress will occur on the side where the stresses add, or on the tension side. Hence, from mechanics,

$$W = \frac{\pi d^2 S}{4};$$

or,

$$S = \frac{4W}{\pi d^2} \text{ (due to direct tension).} \quad \textbf{(34)}$$

Also,

$$WR = \frac{Sd^3}{10.2};$$

or,

$$S = \frac{10.2\ WR}{d^3} \text{ (due to bending).} \quad \textbf{(35)}$$

Hence the combined tensional stress acting at the point B, or, in fact, at any point on the extreme outside of the vertical shaft toward the force W, is:

$$S = \frac{4W}{\pi d^2} + \frac{10.2\ WR}{d^3}. \qquad \textbf{(36)}$$

If W acted in the opposite direction, the greatest stress would still be at the side B, but would be a compression instead of a tension, of the same magnitude as before.

Tension or Compression Combined with Torsion. In Fig. 29, V might be the end load on a vertical shaft; and the two forces W might act in conjunction with it as in the case of Fig. 26, at the radius R. This case is not very often met with. It is usually possible to combine the moments, find an equivalent moment of a simple kind, and use the corresponding simple fiber stress. In the case in question we have a direct stress to be combined with a shearing stress, and mechanics gives us the following solution:

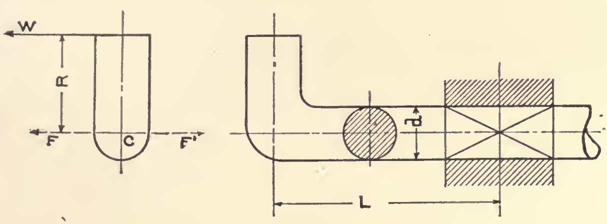


Fig. 30.

Let S_s = simple shearing stress (lbs. per sq in.).
Let S_c = simple compressive stress (lbs. per sq. in.).
Let S_{rs} = resultant shearing stress (lbs. per sq. in.).
Let S_{rc} = resultant compressive stress (lbs. per sq. in.).

We then have:

$$2WR = \frac{S_s d^3}{5.1};$$

or,

$$S_s = \frac{5.1(2WR)}{d^3}. \qquad \textbf{(37)}$$

Also,

$$V = \frac{\pi d^2 S_c}{4};$$

or,
$$S_c = \frac{4}{\pi d^2} V. \qquad (38)$$

Now, from a solution given in simplest form in "Merriman's Mechanics"—which the student may consult, if desired—values for the resultant stresses can be found. Whichever of these is the critical one for the material used, should form the basis for its diameter:

$$S_{rs} = \sqrt{S_s^2 + \frac{S_c^2}{4}}. \qquad (39)$$

Also,
$$S_{rc} = \frac{S_c}{2} + \sqrt{S_s^2 + \frac{S_c^2}{4}}. \qquad (40)$$

Bending Combined with Torsion. In Fig. 30, the load W acts not only to twist the shaft off, but also presses it sidewise against the bearing. As it is usually customary to figure the maximum moment as taking place at the center of the bearing, the length L, which determines the bending moment, is taken to that point. The theory of the stress induced in this case is complicated. In order to make the magnitude of the moments clearer, let us introduce the two equal and opposite forces F and F^1, each equal to W, at the point C. We can evidently do this without changing the equilibrium of the shaft in any way. We now see that W and F^1 act as a couple giving a twisting moment WR; and that F acts with a leverage L, producing a bending moment FL = WL, at the middle of the bearing.

If, now, we find an equivalent twisting moment, or an equivalent bending moment, which would produce the same effect on the fibers of the shaft as the two combined, we can treat the calculation of the diameter as a simple case, and proceed as in the cases of simple torsion and simple bending considered above. This relation is given us in mechanics:

$$B_e = \frac{B}{2} + \frac{1}{2}\sqrt{B^2 + T^2}. \qquad (41)$$

$$T_e = B + \sqrt{B^2 + T^2}. \qquad (42)$$

These expressions are true in relation to each other, on the assumption that the allowable fiber stress S is the same for tension, compression, and shearing. For the material of which shafts are usually made, this is near enough to the truth to give safe and practical results. Using the expressions for internal moments of resistance as previously noted for circular sections, we then have :

$$B_e = \frac{Sd^3}{10.2}. \qquad (43)$$

Also,

$$T_e = \frac{Sd^3}{5.1}. \qquad (44)$$

Either equation may be used ; the diameter d will result the same whichever equation is taken. For the sake of simplicity, equation 42 is generally preferred, equation 44 being taken in conjunction with it.

The expression $\sqrt{B^2 + T^2}$ is one that would be a long and tedious task to calculate. By inspection it is readily seen that this quantity can be graphically represented by means of a right-angled triangle having B and T as the sides. We may then lay down on a piece of paper, to some convenient scale, the moments B and T as the sides of a right-angled triangle, when, upon measuring the hypothenuse, we can easily read off to the same scale $\sqrt{B^2 + T^2}$. Even if the drawing is made to a small scale, the accuracy of the reading will be sufficient to enable the value for d to be solved very closely. This graphical method is illustrated in Part I.

Deflection. For a shaft subjected to pure torsion, as in Fig. 26, the angular deflection due to the load may be carried to a certain point before the limit of working fiber stress is exceeded. The equation worked out from mechanics for this condition, is:

$$A^\circ = \frac{584\ TL}{Gd^4}, \qquad (45)$$

which at once gives the number of degrees of angular deflection for a shaft whose modulus of elasticity, torsional moment, and length are known.

The shearing modulus of elasticity of ordinary shaft steel runs from 10,000,000 to 13,000,000, giving as an average about 12,000,000.

By the well-known relation of "Hooke's law" (stresses proportional to strains within the elastic limit of the material), we have:

$$\frac{A^\circ}{360^\circ} = \frac{SL}{\pi G d};$$

or

$$S = \frac{A\pi G d}{360\ L}. \qquad (46)$$

A twist of one degree in a length of twenty diameters is a usual allowance. Substituting $A = 1$, $L = 20d$, and $G = 12{,}000{,}000$, we have:

$$S = 5{,}240 \text{ (nearly)}. \qquad (47)$$

This is a safe value for shearing fiber stress in steel. In fact, in calculations for strength, even for reversing stresses, the usual figure is 8,000 (lbs. per square inch), thus indicating that the relation of one degree to twenty diameters is well within the limit of strength.

For a hollow shaft the above formula becomes :

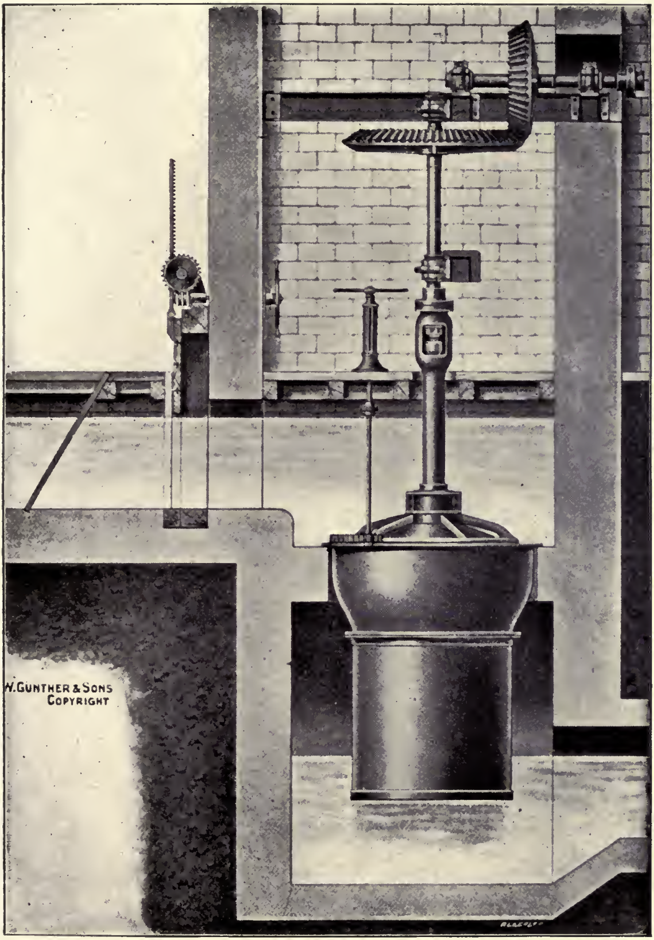
$$A^\circ = \frac{584\ TL}{G(d_0^4 - d_1^4)}. \qquad (48)$$

Transverse deflection occurs when the shaft is subjected to a bending moment. It may therefore exist alone or in conjunction with angular deflection. Transverse deflection of shafts, however, rarely exists up to the point of limiting fiber stress, because before that point is reached the alignment of the shaft is so disturbed that it is not practicable as a device for transmitting power. A transverse deflection of .01 inch per foot of length is a common allowance ; but it is impossible to fix any general limit, as in many cases this figure, if exceeded, would do no harm, while in others—such as heavily loaded or high-speed bearings—even the figure given might be fatal to good operation.

The formula for transverse deflection, deduced from mechanics, varies with the system of loading. The three most common conditions only are given below, reference to the handbook being necessary if other conditions must be satisfied:

Fixed at one end, loaded at the other,

$$e = \frac{WL^3}{3\ EI}. \qquad (49)$$



GENERAL ARRANGEMENT OF JONVAL TURBINE.
Central Engineering Works, Oldham, Eng.

Supported at ends, loaded in middle,

$$e = \frac{WL^3}{48EI}. \qquad (50)$$

Supported at ends, loaded uniformly,

$$e = \frac{5WL^3}{384EI}. \qquad (51)$$

For transverse deflection the direct modulus of elasticity must be used, for the fibers are stretched or compressed, instead of being subjected to a shearing action. The most usual value of the direct modulus of elasticity for ordinary steel is 30,000,000, and is denoted in most books by the symbol E. Both the shearing and direct moduli of elasticity are really nothing but the ratio of the stress to the strain produced by that stress, it being assumed that the given material is perfectly elastic. A material is supposed to be perfectly elastic up to a certain limit of stress, and it is within this limit that the relation as above holds good.

Expressed in the form of an equation this would be :

$$E = \frac{S}{\frac{e}{L}} = \frac{SL}{e}. \qquad (52)$$

Centrifugal Whirling. If a line shaft deflect but slightly, due to its own weight, or the weight or pressure of other bodies upon it, and then be run at a high speed, the centrifugal force set up increases the deflection, and the shaft whirls about the geometrical line through the centers of the bearings, causing vibration and wear in the adjoining members. It is evident that the practical remedy for this tendency in a shaft of given diameter and speed is to locate the bearings sufficiently close to render the action of small effect.

Many formulæ might be given for this relation, each being based on different assumptions. Perhaps as widely applied and as simple as any, is the "Rankine" formula, which sets the limit of length between bearings for shafts not greatly loaded by intermediate pulleys or side strains :

$$M = 175 \sqrt{\frac{d}{N}}. \qquad (53)$$

Horse-Power of Shafting. Horse-power is a certain specific rate of doing work, *viz.*, 33,000 foot-pounds per minute. Hence, to find the horse-power that a shaft will transmit, we must first find the work done, and then relate it to the speed. Take, for example, the case of a pulley, the symbols being the same as before —namely, P = driving force at rim of pulley (lbs.); R = radius of pulley (inches); N = number of revolutions per minute; and H = horse-power. Then,

$$\text{Work} = \text{force} \times \text{distance} = P \times (2\pi RN) = H \times 33{,}000 \times 12;$$

or,

$$PR = \frac{63{,}025H}{N}. \qquad \textbf{(54)}$$

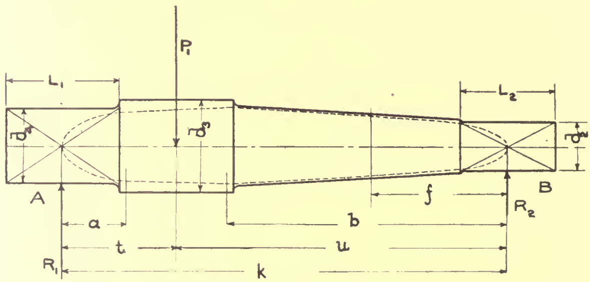


Fig. 31.

This is one of the most useful equations for calculations involving horse-power. By it the number of inch-pounds torsion for any horse-power can be at once ascertained.

It should be clearly noted, however, that in this equation the bending moment does not enter at all. Hence any shaft based in size on *horse-power alone*, is based on *torsional moment alone*, bending moment being entirely neglected. In many cases the bending moment is the controlling one as to limiting fiber stress. Hence empirical shafting formulæ depending upon the horse-power relation are unsafe, unless it is definitely known just what torsional and bending moments have been assumed.

The only safe way to figure the size of a shaft is to find accurately what torsional moment and bending moment it has to sustain, and then combine them according to equation 41 or 42 introducing the element of speed as basis for assumption of a high or low working fiber stress.

PRACTICAL MODIFICATION. The practical methods of handling the theoretical shaft equations have reference to the fit of the shaft within the several pieces upon it. The running fit of a shaft in a bearing is usually considered to be so loose that the shaft could freely deflect to the center of the bearing. This is doubtless an extreme view of the case, but it is the only safe assumption. Hence a shaft running in bearings (see Fig. 31) is supposed to be supported at the centers of those bearings, and its theoretical strength is based on this supposition.

For a tight or driving fit upon the shaft, a safe assumption to make is that there is looseness enough at the ends of the fit to permit the shaft to be stressed by the load a short distance within the faces of the hub, say from $\frac{1}{2}$ inch to 1 inch. For example, referring to Fig. 31, suppose P_1 to be the transverse load, exerted through a hub fast upon the part of the shaft d_3. Taking moments about the center of one bearing, and solving for the reaction at the center of the other, we have :

$$P_1\, u = R_1\, K;$$

or,

$$R_1 = \frac{P_1\, u}{K}. \tag{55}$$

Also,

$$P_1\, t = R_2\, K;$$

or,

$$R_2 = \frac{P_1\, t}{K}. \tag{56}$$

Now, as far as the part of shaft d_3 is concerned, it may depend for its size on the bending moment $R_2\, b$, or on $R_1\, a$. The reason the lever arm is not taken to the point directly under the load P_1, is because it is not practically possible to break the shaft at that point, on account of the reinforcement of the hub, which is tightly fitted upon it. Trying these moments to see which is the greater, we shall find that the greater moment always occurs in connection with the longer lever arm. Hence $R_2\, b$ will be greater than $R_1\, a$. We then write the equation of *external moment = internal moment:*

$$R_2\, b = \frac{S\, d_3^{\,3}}{10.2};$$

or,
$$d_3 = \sqrt[3]{\frac{10.2 \ R_2 b}{S}}. \tag{57}$$

For the size of bearing A we have the maximum bending moment:

$$R_1 \frac{L_1}{2} = \frac{S \, d_4^3}{10.2};$$

or,
$$d_4 = \sqrt[3]{\frac{10.2 \ R_1 L_1}{2 \ S}}. \tag{58}$$

For the size of bearing B we have the maximum moment:

$$R_2 \frac{L_2}{2} = \frac{S \, d_2^3}{10.2};$$

or,
$$d_2 = \sqrt[3]{\frac{10.2 \ R_2 L_2}{2 \ S}}. \tag{59}$$

The above calculations are, of course, on the assumption that no torsion is transmitted either way through this axle. We should in that case have combined torsion and bending. This has been made sufficiently clear in preceding paragraphs and in Part I, to require no further illustration.

The dotted line in Fig. 31 shows the theoretical shape the axle should take under the assumed conditions. The practical modification of this shape is obvious. At the shoulders of the shaft the corners should not be sharp, but carefully filleted, to avoid the possible starting of a crack at those points.

Often the diameter of certain parts of a shaft may be larger than strength actually calls for. For example, in Fig. 31, the part d_3 need only be as large as the dotted line; but it is obvious that unless the key is sunk in the body of the shaft, the hub could not be slipped into place over the part d_4. If, however, the diameter d_3 be made large enough so that the bottom of the key will clear d_4, the rotary cutter which forms the key way in d_3 will also clear d_4, and the key way can be more easily produced.

In cases where fits are not required to be snug, a straight shaft of cold-rolled steel is commonly used. Here any parts fastened on the middle of the shaft have to be driven over a considerable length of the shaft before they reach their final position. Moreover, there is no definite shoulder to stop against, and measurement has to be resorted to in locating them.

It does not pay to turn any portion of a cold-rolled shaft, unless it be the very ends, for relieving the "skin tension" in such material is sure to throw the shaft out of line and necessitate subsequent straightening.

Turned-steel shafts for machines may with advantage be slightly varied in diameter wherever the fit changes; and although the production of shoulders costs something, yet it assists greatly in bringing the parts to their exact location, and enables the workman to concentrate his best skill on the fine bearing fits, and to save time by rough-turning the parts that have no fits.

Hollow shafts are practicable only for large sizes. The advantages of removing the inner core of metal, aside from some specific requirement of the machine, are that it eliminates all possibility of cracks starting from the checks that may exist at the center, permits inspection of the material of a shaft, and, in case of hollow-forged shafts, gives an opening for the forging mandrel. In the last case, the material is improved by a rolling process.

The material most common for use in machine shafting is the ordinary "Machinery Steel," made by the Bessemer process. This steel is apt to be "seamy," and often contains checks and flaws that are detected only upon sudden and unexpected breakage of a part apparently sound. This characteristic is a result of the process employed in the manufacture of the steel, and thus far has never been wholly eliminated. **Bessemer steel** is, nevertheless, a very useful material, and the above weakness is not so serious but that this kind of steel can be used with success in the great majority of cases.

When a more homogeneous shaft is desired, **open-hearth steel** is available. This is a more reliable material to use than the Bessemer, and costs somewhat more. It makes a stiff, true, fine-surfaced shaft, high-grade in every respect. It is usually specified for armature shafts of dynamos and motors.

Steels of special strength, toughness, and elasticity are made under numerous processes. **Nickel steel** is perhaps the most conspicuous example. While for this steel a high price has to be paid, yet its great strength, in connection with other valuable qualities, makes it a material extremely valuable for service where light weight is essential, or where contracted space demands small size.

The range of strength of these various steels is so great that it is well-nigh useless to go into a discussion of it here. Reference should be had to the extended discussions of the handbooks, and to special trade pamphlets. A study of the possibilities of steel in its various forms for use in shafting, is very valuable as a basis for design, as it can almost be said that a machine consists chiefly of a "collection of shafts with a structure built round them." The shafts are like a core, and evidently the size of the core determines the shell about it.

PROBLEMS ON SHAFTS.

1. Required the twisting moment on a shaft that transmits 30 horse-power at 120 revolutions per minute.

2. Find the diameter of a steel shaft designed to transmit 50 horse-power at 150 revolutions per minute.

3. Assuming same data as in Problem 1, find the diameters of a hollow shaft for a value of $S = 8{,}000$.

4. A belt on an idler pulley embraces an angle of 120 degrees. Assuming tension of belt 1,000 pounds on each side, and pulley located midway between bearings, which are 30 inches from center to center, what is the diameter of shaft required?

5. Calculate the diameter of a steel shaft designed to transmit a twisting moment of 400,000 inch-pounds and also to take a bending moment of 300,000 inch-pounds.

6. Find the angular deflection in a 4-inch shaft 20 feet long when subjected to a load of 5,500 pounds applied to an arm of 30-inch radius. Assume transverse modulus of elasticity equal to 12,000,000.

7. The overhung crank of a steam engine has a force of 32,000 lbs. at the center of the crank pin, which is 12 inches from the center of the shaft bearing, measured parallel to the shaft. The radius of crank arm is 10 inches. Assume S equal to 10,000. Calculate the diameter of the crank shaft.

8. On a short, vertical steel shaft the load is 5,000 pounds. A gear, 36 teeth, $1\frac{1}{2}$ diametral pitch, at top of shaft, transmits a load of 4,000 pounds at the pitch line. Safe shear = 7,500. What is the diameter of the shaft?

SPUR GEARS.

NOTATION—The following notation is used throughout the chapter on Spur Gears:

b = Breadth of rectangular section of arm (inches).

M. M_1 = Revolutions per minute.

μ = Coefficient of friction between teeth.

C = Width of arm extended to pitch line (inches).
c = Distance from neutral axis to outer fiber (inches).
D = Pitch diameter of gear (inches).
F = Face of gear (inches).
f = Clearance of tooth at bottom (inches).
G = Thickness of arm extended to pitch line (inches).
H = Thickness of tooth at any section (inches).
h = Depth of rectangular section of arm (inches).
I = Moment of inertia.
K = Thickness of rim (inches).
L = Distance from top of tooth to any section (inches).
N = Number of teeth.
n = Number of arms.
P = Diametral pitch (teeth per inch of diameter).
P^1 = Circular pitch (inches).
Q, Q_1 = Normal pressure between teeth (lbs.).
R, R_1 = Resultant pressure between teeth (lbs.).
r, r_1 = Radius of pitch circles (inches).
S = Fiber stress of material (lbs. per sq. in.).
s = Addendum of tooth (inches) = Dedendum of tooth.
t = Thickness of tooth at pitch line (inches).
W = Load at pitch line (lbs.).
y = Coefficient for "Lewis" formula.

ANALYSIS. If a cylinder be placed on a plane surface, with its axis parallel to the plane, an attempt to rotate the cylinder about its axis would cause it to roll on the plane.

Again, if two cylinders be provided with axial bearings, and be slightly pressed together, motion of one about its axis will cause a similar motion of the other, the two surfaces rolling one on the other at their common tangent line. If moved with care, there will be no slipping in either of the above cases—which is explained by the fact that no matter how smooth the surfaces may appear to be, there is still sufficient roughness to make the little irregularities interlock and act like minute teeth.

The magnitude of the force possible to be transmitted depends not only on the roughness of the surfaces, but on the amount of pressure between them. Suppose that one cylinder is a part of a hoisting drum, on which is wound a rope with a weight attached. We can readily make the weight so great that, no matter how hard we press the two cylinders together, the driving cylinder will not turn the hoisting cylinder, but will slip past it. If now, instead of increasing the pressure, which is detrimental both to cylinders and bearings of same, we increase the coarseness of the surfaces, or, in other words, put teeth of appreciable size on these surfaces, we attain the desired result of positively driving without excessive side pressure.

These artificial projections, or teeth, must fit into one another; hence the surfaces of the original cylinders, having been broken up into alternate projections and hollows, have entirely disappeared to the eye; they nevertheless exist as ideal or imaginary surfaces, which roll together with the same surface velocities as if in bodily form, provided that the curves of the teeth are correctly formed. Several mathematical curves are available for use as tooth outlines, but in practice the **involute** and **cycloidal curves** are the only ones used for this purpose.

The ideal surfaces are known as **pitch cylinders** or **pitch circles.** In Fig. 32 is shown an end view of such a pair of cylinders in contact at their pitch point P. In gear calculations we assume that there is no slip between the pitch circles, acting as driving cylinders; hence the speeds of the two pitch circles at the

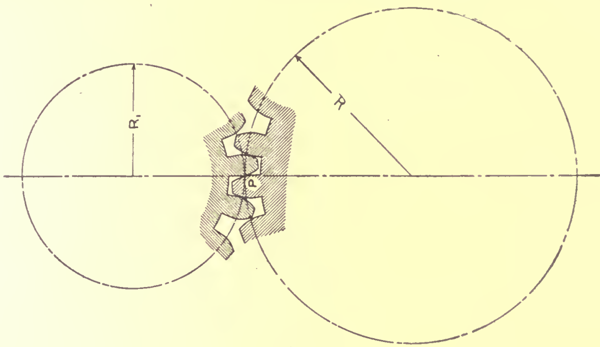


Fig. 32.

pitch point are equal. If M and M_1 be the revolutions per minute of the cylinders respectively, r and r_1 their radii, then

$$2\pi r M = 2\pi r_1 M_1;$$

or,

$$\frac{M}{M_1} = \frac{r_1}{r}. \qquad (60)$$

That is, the number of revolutions varies inversely as the radii.

The simple calculation as above is the key to all calculations involving gear trains in reference to their speed ratio.

Fig. 33 represents cycloidal teeth in the two extreme positions of beginning and ending contact. The normal pressure Q or Q_1 between the teeth in each position acts through the pitch point O, as it must always do in order to insure the condition of ideal rolling of the pitch circles, and the velocity ratio proportional to $\frac{r_1}{r}$ As the surfaces of the teeth slide together, frictional resistance is produced at their point of contact. This force is widely variable, depending on the material and condition of the tooth surfaces, whether smooth and well lubricated, or rough and gritty. As this resistance acts in conjunction with the normal force between the teeth, we may construct a parallelogram of forces on these two as a base, the resultant pressure between the teeth being slightly changed thereby, as shown in Fig 33.

Assuming a coefficient of friction μ, the force of friction is μ Q or μ Q_1 and the resultant pressure R or R_1.

Tooth B of the FOLLOWER is therefore under a heavy bending moment measured by the product RL, L being the perpendicular distance from the center of the tooth at its base to the line of the force. This tooth also has a relatively small compressive stress due to the resolved part of R along the radius, and a relatively small shearing stress due to the resolved part of R along a tangent to the pitch circle.

Tooth D of the driven wheel or FOLLOWER has a relatively large shearing stress, a small bending moment, and practically no direct compressive stress.

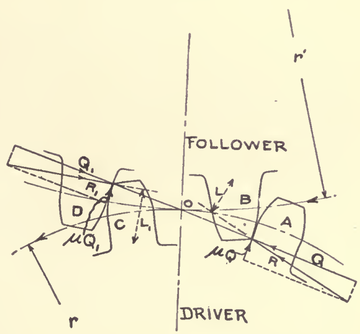


Fig. 33.

Tooth A of the driving wheel or DRIVER has a relatively large shearing stress, a small bending moment, and small compressive stress.

Tooth C of the DRIVER has a large bending moment, but small compressive and shearing stresses.

The conditions as noted above are not those of every pair of gears, in fact they vary with every difference of pitch circle, or of detail and position of tooth. It is true, however, that in nearly all cases in practice the bending stress is the controlling one from a theoretical standpoint. Moreover, the designer must consider the form and strength of the tooth when it is under the condition of maximum moment. This evidently, from the above, occurs at the beginning of contact, for the follower teeth; and at the end of contact, for the driver teeth. In the particular case illustrated in

Fig. 33, if the material in both gears were the same, tooth C, being the weaker at the root, would probably break before B; but if C were of steel, and B of cast iron, B might break first.

It will be noticed that R is nearly parallel to the top of the tooth; and it may easily happen that the friction may become of such a value that it will turn the direction of R until it lies along the top of the tooth exactly, which is the condition for maximum moment. For strength calculations it is usual to consider this condition as existing in all cases.

At the beginning of contact there is more or less shock when the teeth strike together, and this effect is much more evident at high speeds. There is also at the beginning of contact a sort of chattering action as the driving tooth rubs along the driven tooth.

Uniform distribution of pressure along the face of the tooth is often impaired by uneven wear of the bearings supporting the gear shafts, the pressure being localized on one corner of the tooth. The same effect is caused by the accidental presence of foreign material between the teeth. Again, in cast gearing, the spacing may be irregular, or, on account of draft on the pattern, the teeth may bear at the high points only. While it is usual to consider that the load is evenly distributed along the face of the tooth, yet the above considerations show that *an ample margin of strength must always be allowed* on account of these uncertainties.

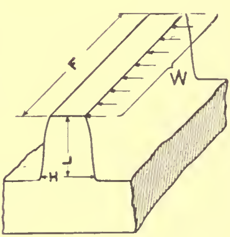


Fig. 34.

When the number of teeth in the mating gears is high, the load will be distributed between several teeth; but, as it is almost certain that at some time the proper distribution of load will not exist, and that one tooth will receive the full load, it is considered that practically the only safe method is so to design the teeth that a single tooth may be relied upon to withstand the full load without failure.

THEORY. Based on the Analysis as given, the theory of gear teeth assumes that one tooth takes the whole load, and that this load is evenly distributed along the top of the tooth and acts parallel with its base, thus reducing the condition of the tooth to that of a cantilever beam. The magnitude of this load at the top of the tooth is taken for convenience the same as the force transmitted at the pitch circle. This condition is shown in Fig. 34. Equating the external moment to the internal moment, we then have, from mechanics:

$$WL = \frac{SI}{c} = \frac{SFH^2}{6}. \qquad \textbf{(61)}$$

The thickness H is usually taken either at the pitch line or at the root of the tooth just before the fillet begins; and L, of course, is dependent on the tooth dimensions. The formula is most readily used when the outline of the tooth is either assumed or known, a trial calculation being made to see if it will stand the load, and a series of subsequent calculations followed out in the same way until a suitable tooth is found. This method is pursued because there are certain even pitches which it is desirable to use; and it is safe to say that any calculation figured the reverse way would result in fractional pitches. The latter course may be used, however, and the nearest even pitch chosen as the proper one.

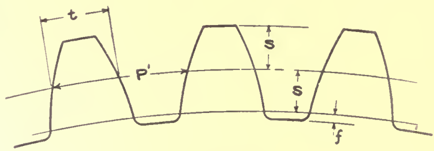
As stated under "Analysis," there are a great many circumstances attending the operation of gears which make impossible the purely theoretical application of the beam formulæ. For this reason there is no one element of machinery which depends so much on experience and judgment for correct proportion as the tooth of a gear. Hence it is true that a rational formula based on the theoretical one is really of the greater practical value in tooth design.

If we examine formula 61, we find that in a form solved for W, we have:

$$W = \frac{SFH^2}{6\ L}. \qquad \textbf{(62)}$$

Of these quantities, H and L are the only variables, for we can make the others what we choose. H and L depend upon the circular pitch P^1 and the curvature and outline of the tooth. If now we could settle on a standard system of teeth, we could establish a coefficient to be used to take the place of the variable part of H and L, which depends on the outline of tooth, and we should thus have an empirical formula which would be on a theoretical basis.

This, Mr. Wilfred Lewis has done; and it is safe to say that this formula is more universally used and with more satis-



$$t = \frac{P^1}{2} \qquad f = \frac{t}{10} \qquad s = \frac{P^1}{\pi}$$

Fig. 35.

factory practical results than any other formula, theoretical or practical, that has ever been devised. His coefficient is known as y, and was determined from many actual drawings of different forms of teeth showing the weakest section. This coefficient is worked out for the three most common systems as follows:

For 20° involute, $$y = 0.154 - \frac{0.912}{N}. \qquad (63)$$

For 15° involute and cycloidal, $$y = 0.124 - \frac{0.684}{N}. \qquad (64)$$

For radial flanks, $$y = 0.075 - \frac{0.276}{N}. \qquad (65)$$

The tooth upon which the above is based is the American standard or Brown & Sharpe tooth, for which the proportions are shown in Fig. 35.

The "Lewis" formula* is:

$$W = SP^1 Fy. \qquad (66)$$

A table indicating the value of S for different speeds follows:

Safe Working Stresses for Different Speeds.

Speed of teeth, ft. per min.	100	200	300	600	900	1200	1800	2400
Cast iron	8000	6000	4800	4000	3000	2400	2000	1706
Steel	20000	15000	12000	10000	7500	6000	5000	4300

*NOTE. A full and convenient statement of the Lewis formula will be found in "Kent's Pocket Book."

A usual relation of F to P^1 is:

$$\text{For cast teeth, } F = 2P^1 \text{ to } 3P^1. \qquad (66)$$

$$\text{For cut teeth, } F = 3P^1 \text{ to } 4P^1. \qquad (67)$$

The usual method of handling these formulæ is as follows:

The pitch circles of the proposed gears are known or can be assumed; hence W can readily be figured, also the speed of the teeth, whence S can be read from the table. The desired relation of F to P^1 can be arbitrarily chosen, when P^1 and y become the only unknown quantities in the equation. A shrewd guess can be made for the number of teeth, and y calculated therefrom. Then solve the equation for P^1 which will undoubtedly be fractional. Choose the nearest even pitch, or, if it is desired to keep an even diametral pitch, the fractional pitch that will bring an even diametral pitch. Now, from this final and corrected pitch, and the diameter of the pitch circle, calculate the number of teeth N in the gear. Check the assumed value of y by this positive value of N.

Another good way of using this formula is to start with the pitch and face desired, and the diameter of the pitch circle. In

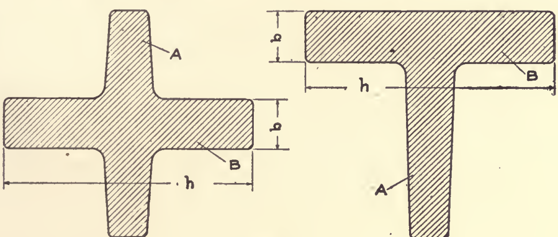


Fig. 37.

this case W is the only unknown quantity, and when found can be compared with the load required to be carried. If too small, make another and successive calculations until the result approximates the required load.

SPUR GEAR RIM, ARMS, AND HUB.

ANALYSIS and THEORY. The rim of a gear has to transmit the load on the teeth to the arms. It is thus in tension on one side of the teeth in action, and in compression on the other. The section of the rim, however, is so dependent on other practical considerations which call for an excess of strength in this respect, that it is not considered worth while to attempt a calculation on this basis.

Gears seldom run fast enough to make necessary a calculation for centrifugal force ; and in general it can be said that the design of the rim is entirely dependent on practical considerations. These will appear later under "Practical Modification."

The arms of a gear are stressed the same as pulley arms, the same theory answering for both, except that a gear rim always being much heavier than a pulley rim, the distribution of load amongst the arms is better in the case of a gear than of a pulley, and it is usually safe to assume that each arm of a gear takes its full proportion of load ; or, for an oval section, equating the external moment to the internal moment as in the case of pulleys, we have :

$$\frac{WD}{n2} = 0.0393\ Sh^3. \qquad (68)$$

Heavy spur gears have the arms of a cross or T section (Fig.

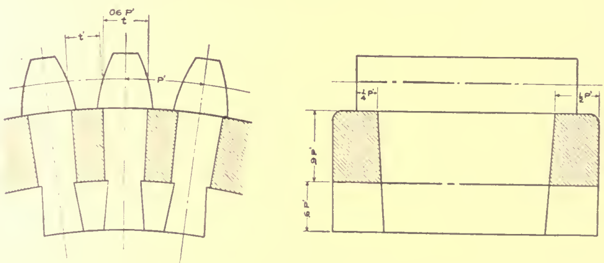


Fig. 38.

37), the latter being especially applicable to the case of bevel gears where there is considerable side thrust. The simplest way of treating such sections is to consider that the whole bending moment is taken by the rectangular section whose greater dimension is in the direction of the load. The rest of the section, being close to the neutral axis of the section, is of little value in resisting the direct load, its function being to give sidewise stiffness. The equation for the cross or T style of arm, then is :

$$\frac{W}{n} \times \frac{D}{2} = \frac{Sbh^2}{6}. \qquad (69)$$

Either b or h may be assumed, and the other determined. As a guide to the section, b may be taken at about the thickness of the tooth.

Gear hubs are in no wise different from the hubs of pulleys or other rotating pieces. The depth necessary for providing sufficient strength over the key to avoid splitting is the guiding element, and can usually be best determined by careful judgment.

PRACTICAL MODIFICATION. The practical requirements, which no theory will satisfy, are many and varied. Sudden and severe shock, excessive wear due to an atmosphere of grit and corrosive elements, abrupt reversal of the mechanism, the throwing-in of clutches and pawls, the action of brakes—these and many other influences have an important bearing on gear design, but not one that can be calculated. The only method of procedure in such cases is to base the design on analysis and theory as previously given, and then add to the face of gear, thickness of tooth, or pitch an amount which judgment and experience dictate as sufficient.

Excessive noise and vibration are difficult to prevent at high speeds. At 1,000 feet per minute, gears are apt to run with an unpleasant amount of noise. At speeds beyond this, it is often necessary to provide **mortise teeth,** or teeth of hard wood set into a cast-iron rim (see Fig. 38). Rawhide pinions are useful in this regard. Fine pitches with a long face of tooth run much more smoothly at high speeds than a coarse pitch and narrow-faced tooth of equal strength. Greater care in alignment of shafts, however, is necessary, also stiffer supports.

Should it be impracticable to use a standard tooth of sufficient strength, there are several ways in which we can increase the carrying capacity without increasing the pitch. These are:

1. Use a stronger material, such as steel.
2. Shroud the teeth.
3. Use a hook tooth.
4. Use a stub tooth.

Shrouding a tooth consists in connecting the ends of the teeth with a rim of metal. When this rim is extended to the top of the tooth, the process is called "full-shrouding" (Fig. 39); and when carried only to the pitch line, it is termed "half-shrouding" (Fig. 40). The theoretical effect of shrouding is to make the tooth act like a short beam built in at the sides; and the tooth will practically have to be sheared out in order to fail. This modification of gear design requires the teeth to be cast, as the cutter cannot pass through the shrouding. The strength of the shrouded gear is estimated to be from 25 to 50 per cent above that of the plain-tooth type.

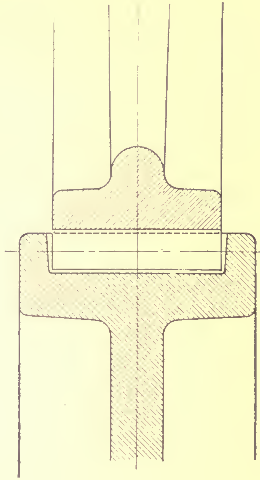


Fig. 39.

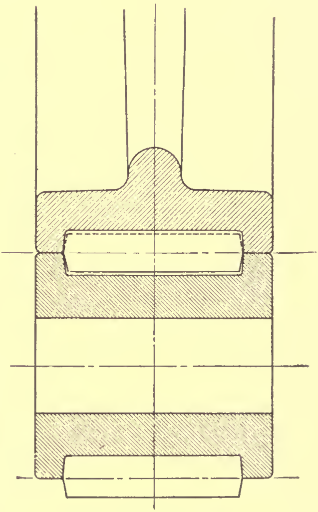
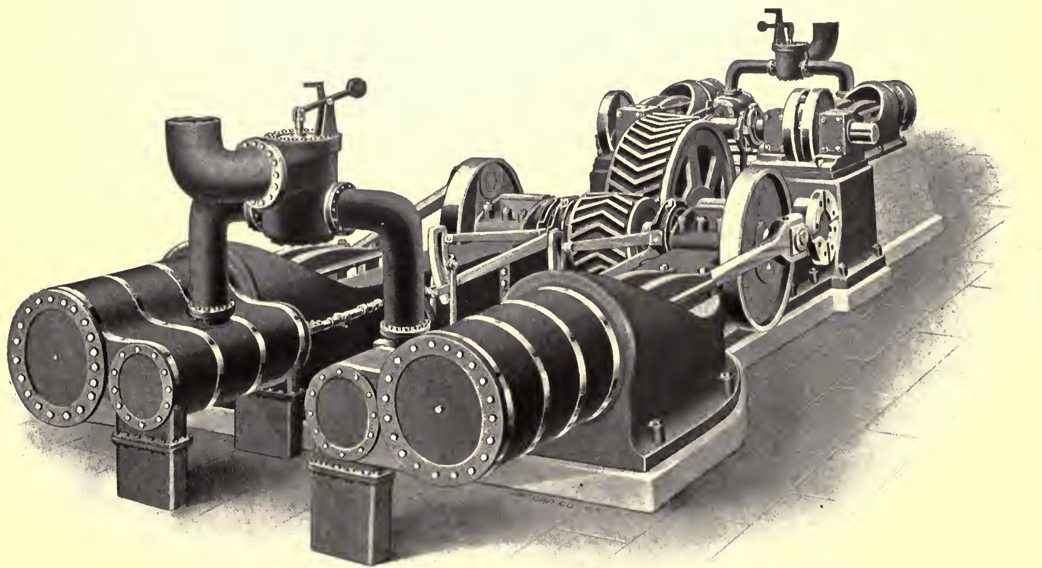


Fig. 40.

The **hook-tooth gear** (Fig. 41) is applicable only to cases where the load on the tooth does not reverse. The working side of the tooth is made of the usual standard curve, while the back is made of a curve of greater obliquity, resulting in a considerable increase of thickness at the root of the tooth. A comparison of strength between this form and the standard may be made by drawing the two teeth for a given pitch, measuring their thickness just at top of the fillet, and finding the relation of the squares of these dimensions. The truth of this relation is readily seen from an inspection of formula 61.

The **stub tooth** merely involves the shortening of the height



UNIVERSAL SLABBING MILL ENGINES.
46 x 60 inches and 26 x 30 inches
William Tod Co

of the tooth in order to reduce the lever arm on which the load acts, thus reducing the moment, and thereby permitting a greater load to be carried for the same stress.

The rim of a gear is dependent for its proportions chiefly on questions of practical moulding and machining. It must bear a certain relation to the teeth and arms, so that, when it is cooling in the mould, serious shrinkage stresses will not be set up, forming pockets and cracks. Moreover, when under pressure of the cutter in the producing of the teeth, it must not chatter or spring. This condition is quite well attained in ordinary gears when the thickness of the rim below the base of the tooth is made about the same as the thickness of the tooth.

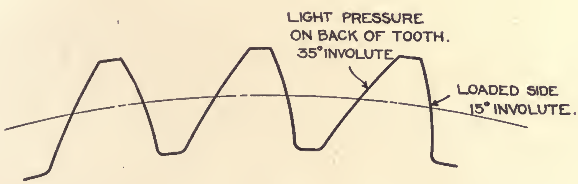


Fig. 41.

The stiffening ribs and arms must all be joined to the rim by ample fillets, and the cross-section must be as uniform as possible, to prevent unequal cooling and consequent pulling-away of the arms from the rim or hub. Often the calculated size of the arms at both rim and hub has to be modified considerably to meet this requirement.

The arms are usually tapered to suit the designer's eye, a small gear requiring more taper per foot than a large one. Both rim and hub should be tapered $\frac{1}{2}$ inch per foot to permit easy drawing-out from the mould.

The proportions given in the following table have been used with success as a basis of gear design in manufacturing practice. The table will serve as an excellent guide in laying out, and can be closely followed, in most cases with but slight modification. **Web gears** are introduced for small diameters where the arms begin to look awkward and clumsy.

Gear Design Data.

Measurements given in inches. Letters refer to Fig. 42.

Diametral pitch ..	P	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5	6	8
Face............	F	$6\frac{1}{4}$	$5\frac{1}{2}$	$4\frac{3}{4}$	$3\frac{3}{4}$	$3\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{7}{8}$	$1\frac{1}{2}$
Thickness of arm when extended to pitch line....	G	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{1}{2}$
Width of arm when extended to pitch line	C	4	$3\frac{1}{2}$	3	$2\frac{1}{2}$	$2\frac{1}{4}$	2	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{8}$
Thickness of rim..	K	$2\frac{3}{4}$	$2\frac{3}{8}$	$2\frac{1}{8}$	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	1	$\frac{7}{8}$	$\frac{3}{4}$
Depth of rib......	E	2	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	1	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{8}$
Thickness of web.	T	$1\frac{1}{8}$	1	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{5}{16}$

Number of arms, 6.

Give inside of rims and hub a draft of $\frac{1}{2}$ inch per foot.

BEVEL GEARS.

NOTATION—The following notation is used throughout the chapter on Bevel Gears:

A =Apex distance at pitch element of cone (inches).
A^1=Apex distance at bottom element of tooth (inches).
B =Angle of bottom of tooth (degrees).
C =Pitch angle (degrees).
D =Pitch diameter (inches).
E =Radius increment of gear (inches).
F =Face of gear (inches).
f =Clearance at bottom (inches).
G =Angle of face (degrees).
H =Cutting angle (degrees).
K =Radius increment of pinion (inches).
N =Number of teeth.
N^1=Formative number of teeth, or the number corresponding to the spur gear on which the outline of tooth is made.
O D=Outside diameter (inches).
P =Diametral pitch related to pitch diameter (teeth per inch).
P^1 =Circular pitch measured on the circumference of D (inches).
S =Working strength of material (lbs. per sq. in.).
s =Addendum, or height of tooth above pitch line (inches).
$s+f$=Depth of tooth below pitch line (inches).
T =Angle of top of tooth (degrees).
t =Thickness of tooth at pitch line (inches).
W =Working load at pitch line (lbs.).
y =Factor in "Lewis" formula.

ANALYSIS. It is possible to consider bevel gears as the general case of which spur gears are a special form. The pitch surfaces of spur gears described above as cylinders, mathematically considered, are cones whose vertices are infinitely distant, while bevel gears likewise are based on pitch cones, but with a vertex at some finite point, common to the mating pair. Hence, as we might expect, the laws of tooth action are similar in bevel gears to those in the case of spur gears. The profile of the tooth in the former case, however, is based, not on the real radius of the pitch cone, but on the radius of the normal cone ; and in the development of the outline the latter is treated just as though it were the radius of a spur gear. The tooth thus formed is wrapped back upon the normal cone face, and becomes the large end of the tapering bevel-gear tooth (see Fig. 44).

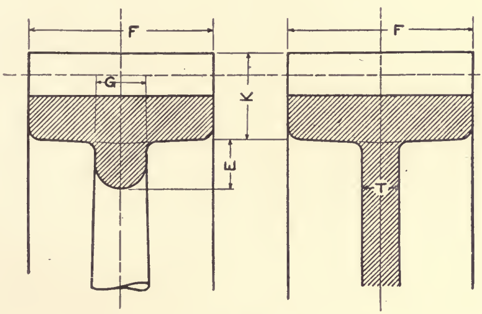


Fig. 42.

The teeth of bevel gears, being simply projections with bases on the pitch cones, have a varying cross-section decreasing toward the vertex ; also a trapezoidal section of root, the latter section acting as a beam section to resist the cantilever moment due to the tooth load.

The arms must, as in the case of spur gears, transmit the load from the tooth to the shaft; in addition, the arms of a bevel gear are subjected to a side thrust due to the wedging action of the cones. Hence sidewise stiffness of the arms is more essential in this type of gear than in the case of the spur gear.

THEORY. It is evident that the calculation of tooth strength based on a trapezoidal section of root would be somewhat compli-

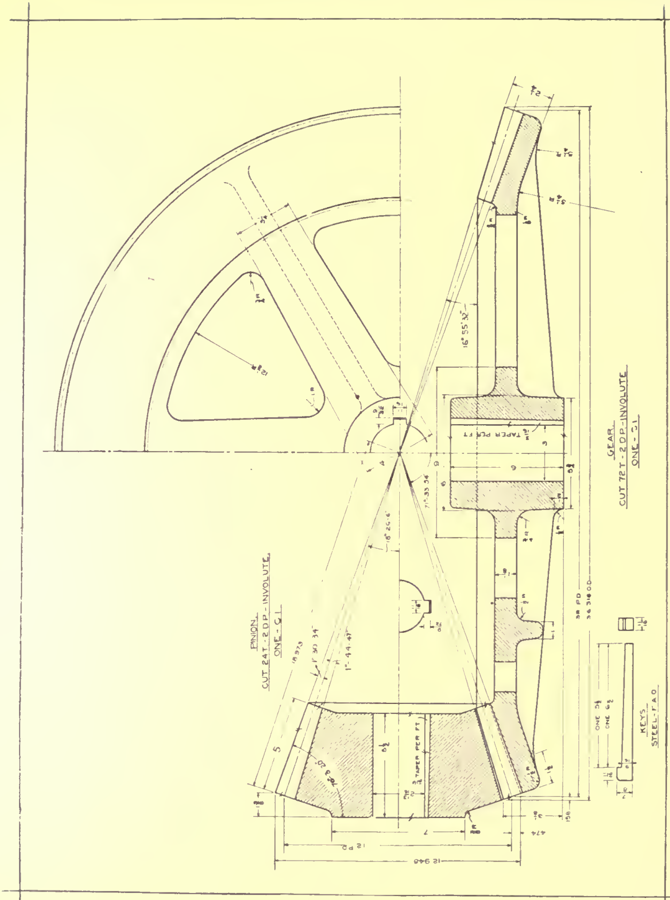


Fig. 43.

cated ; also that the trapezoid in most cases would be but little different from a true rectangle. Hence the error will be but slight if the average cross-section of the tooth be taken to represent its strength, and the calculation made accordingly.

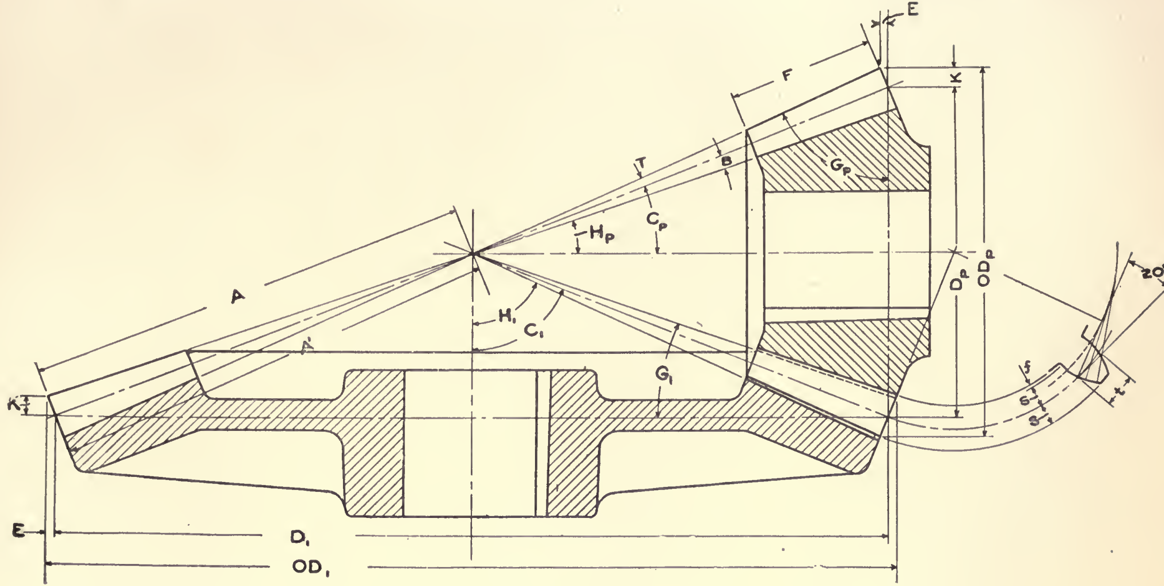


Fig. 44.

Fig. 45 shows a bevel-gear tooth with the average cross-section in dotted lines. For the purpose of calculation, the assumption is made that the section A is carried the full length of the face of the gear, and that the load which this average tooth must carry is the calculated load at the pitch line of section A. This is equivalent to saying that the strength of a bevel-gear tooth is equal to that of a spur-gear tooth which has the same face, and a section identical with that cut out by a plane at the middle of the bevel tooth. The load, as in the case of the spur gear, should be taken at the top of the tooth; and its magnitude can be conveniently calculated at the mean pitch radius of the bevel face, without appreciable error.

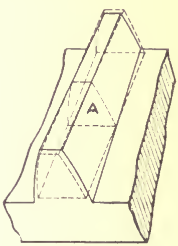


Fig. 45.

This similarity to spur gears being borne in mind, the calculation for strength needs no further treatment. Once the average tooth is assumed or found by layout, a strict following-out of the methods pursued for spur-gear teeth will bring consistent results.

The detail design of a pair of bevel gears involves some trigonometrical computations in order properly to dimension the drawing for use in finishing the blanks and subsequently in cutting the teeth, or, in the case of cast gears, in making the pattern. These calculations, although simple, are yet apt to be tedious; and inaccuracies are likely to creep in if a definite system of relations be not maintained. Hence the results of these calculations are given below in condensed and reduced form. The deduction of these formulæ is a simple and interesting exercise in trigonometry; and it is urged that they be worked out by the student from the figure, in which case he will feel greater confidence in their use.

Axes of Gears at 90 Degrees.

Use subscript 1 for gear; P for pinion. Letters refer to Fig. 44.

$$P = \frac{N}{D} = \frac{\pi}{P^1}. \qquad (70)$$

$$s = \frac{1}{P} = \frac{P^1}{\pi}. \qquad (71)$$

$$t = \frac{P^1}{2} = \frac{\pi}{2P}. \qquad (72)$$

$$f = \frac{t}{10} = \frac{P^1}{20} = \frac{\pi}{20P}. \qquad (73)$$

$$\tan C_p = \frac{N_p}{N_1};\ \tan C_1 = \frac{N_1}{N_p}. \qquad (74)$$

$$\tan T = \frac{s}{A} = \frac{2 \sin C}{N}. \qquad (75)$$

$$\tan B = \frac{s+f}{A} = \frac{2.314 \sin C}{N}. \qquad (76)$$

$$s + f = A \tan B = \frac{1.157}{P} = 0.368P^1. \qquad (77)$$

$$A = \frac{N}{2P \sin C} = \frac{1}{2P}\sqrt{N_1^2 + N_p^2} = \frac{1}{2}\sqrt{D_1^2 + D_p^2}. \qquad (78)$$

$$A^1 = \frac{A}{\cos B} = \frac{N}{2P \cos B \sin C}. \qquad (79)$$

$$G_1 = 90^\circ - (C_1 + T);\ G_p = 90^\circ - (C_p + T). \qquad (80)$$

$$E = S \cos C_1 = S \sin C_p. \qquad (81)$$

$$K = S \cos C_p = S \sin C_1. \qquad (82)$$

PRACTICAL MODIFICATION. The practical requirements to be met in transmission of power by bevel gears are the same as for spur gears; but in the case of bevel gears even greater care is necessary to provide stiffness, strength, true alignment, and rigid supports. As far as the gears themselves are concerned, a long face is desirable; but it is much more difficult to gain the advantage of its strength than in the case of spur gears, because full bearing along the length of the tooth is hard to guarantee.

The rim usually requires a series of ribs running to the hub to give required stiffness and strength against the side thrust which is always present in a pair of bevel gears. Instead of arms, the tendency of bevel-gear design, except for very large gears, is toward a web on account of the better and more uniform connection thereby secured between rim and hub. This web may be lightened by a number of holes, so that the resultant effect is that of a number of wide and flat arms.

The hubs naturally have to be fully as long as those of spur gears, because there is greater tendency to rock on the shaft, due to the side thrust from the teeth, mentioned above.

The teeth on small gears are cut with rotary cutters, at least two finishing cuts being necessary, one for each side of the tapering tooth. The more accurate method is to plane the teeth on a special gear planer, and this method is followed on all gears of any considerable size. The practical requirement here is that no portion of the hub shall project so as to interfere with the stroke of the planer tool. The requirements of gear planers vary somewhat in this regard.

Finally, after all that is possible has been done in the design of the gear itself to render it suitable to withstand the varied stresses, especial attention must be paid to the rigidity of the supporting shafts and bearings. Bearings should always be close up to the hubs of the gears, and, if possible the bearing for both pinion and gear should be cast in the same piece. If this is not done, the tendency of the separate bearings to get out of line and destroy the full bearing of the teeth is difficult to control. Thrust washers are desirable against the hubs of both pinion and gear; also proper means of well lubricating the same.

With these considerations carefully met, bevel gears are not the bugbear of machine design that they are sometimes claimed to be. The common reason why bevel gears cut and fail to work smoothly, is that the gears and supports are not designed carefully enough in relation to each other. This is also true of spur gears, but the bevel gear will reveal imperfections in its design far the more quickly of the two.

WORM AND WORM GEAR.

NOTATION—The following notation is used throughout the chapter on Worm and Worm Gear:

D = Pitch diameter of gear (inches).
E = Efficiency between worm shaft and gear shaft (per cent).
f = Clearance of tooth at bottom (inches).
i = Index of worm thread (1 for single; 2 for double, etc.).
L = Lead of worm thread (inches).
M = Revolutions of gear shaft per minute.
M_w = Revolutions of worm shaft per minute.
N = Number of teeth in gear.
P1 = Circular pitch = Pitch of worm thread (inches).
R = Radius of pitch circle of worm gear (inches).
s = Addendum of tooth (inches).
T = Twisting moment on gear shaft (inch-lbs.).
T_w = Twisting moment on worm shaft (inch-lbs.).
t = Thickness of tooth at pitch line (inches).
W = Load at pitch line (lbs.).

ANALYSIS. The simplest way of analyzing the case of the worm and worm gear is to base it upon an ordinary screw and nut. Take, for example, the lead screw of a common lathe. The carriage carries a nut, through which the lead screw passes. By the rotation of the screw, the carriage, being constrained by the guides to travel lengthwise of the ways, is moved. This motion is, for a single-threaded screw, a distance per revolution equal to the lead of the screw.

Now, suppose that the carriage, instead of sliding along the ways, is compelled to turn about an axis at some point below the ways. Also, suppose the top of the nut to be cut off, and its length made endless by wrapping it around a circle struck from the center about which the carriage rotates. This reduces the nut to a peculiar kind of spur gear, the partial threads of the nut now having the appearance of twisted teeth.

This special form of spur gear, based on the idea of a threaded nut, is known as a **worm gear**, and the screw is termed a **worm.** The teeth are loaded similarly to those of a spur gear, but with the additional feature of a large amount of sliding along the tooth surfaces. This, of course, means considerable friction; and it is in fact possible to utilize the worm and worm gear as an efficient device, only by running the teeth constantly in a bath of oil. Even then the pressures have to be kept well down to insure the required term of life of the tooth surfaces.

It is evident that for one revolution of a single-threaded worm, one tooth of the gear will be passed. The speed ratio between the worm gear and worm shaft will then be equal to the number of teeth in the gear, which is relatively great. Hence the worm and worm gear are principally useful in giving large speed reduction in a small amount of space.

THEORY. The theory of worm-wheel teeth is complicated and obscure. The production of the teeth is simple, a dummy worm with cutting edges, called a "hob," being allowed to carve its way into the worm-gear blank, thus producing the teeth and at the same time driving the worm gear about its axis.

It is clear that if we know the torsional moment on the worm-gear shaft, and the pitch radius of the worm gear, we can find the load on the teeth at the pitch line by dividing the former by the latter. Expressed as an equation:

$$\mathrm{WR} = \mathrm{T}; \text{ or } \mathrm{W} = \frac{\mathrm{T}}{\mathrm{R}}. \qquad (83)$$

How we shall consider this value of W as distributed on the teeth, is a question difficult to answer. The teeth not only are curved to embrace the worm, but are twisted across the face of the gear, so that it would be practically impossible to devise a purely theoretical method of exact calculation. The most reasonable thing to do is to assume the teeth as being equally as strong as spur-gear teeth of the same circular pitch, and to figure them accordingly. It is probably true, however, that the load is carried by more than one tooth, especially in a hobbed wheel; so we shall be safe in assuming that two—and, in case of large wheels, three—teeth divide the load between them. With these considerations borne in mind, the case reduces itself to that of a simple spur-gear tooth calculation, which has already been explained under the heading "Spur Gears."

The worm teeth, or threads, are probably always stronger than the worm-gear teeth; so no calculation for their strength need be made.

The twisting moment on the worm shaft is not determined so directly as in the case of spur gears. The relative number of revolutions of the two shafts depends upon the "lead" of the worm thread and the number of teeth in the gear.

Lead (L) is the distance parallel to the axis of the worm which any point in the thread advances in one revolution of the worm. **Pitch** (P^1) is the distance parallel to the axis of the worm between corresponding points on adjacent threads. The distinction between lead and pitch should be carefully observed, as the two are often confounded, one with the other.

The thread may be single, double, triple, etc., the index of the thread i, being 1, 2, 3, etc., in accordance therewith. The relation between lead and pitch may then be expressed by an equation, thus:

$$L = i\,P^1. \qquad (84)$$

When the index of the thread is changed the speed ratio is changed, the relation being shown by the equation:

$$\frac{M}{M_w} = \frac{i}{N} \qquad (85)$$

If the efficiency were 100 per cent between the two shafts, the twisting moments would be inversely as the ratio of the speeds thus:

$$\frac{T_w}{T} = \frac{M}{M_w} = \frac{i}{N};$$

or,
$$T_w = \frac{Ti}{N}; \tag{86}$$

but for an efficiency E the equation would be:

$$\frac{T_w}{T} = \frac{i}{EN};$$

or,
$$T_w = \frac{Ti}{EN}. \tag{87}$$

The diameter of the worm is arbitrary. Change of this diameter has no effect on the speed ratio. It has a slight effect on the efficiency, the smaller worm giving a little higher efficiency. The diameter of the worm runs ordinarily from 3 to 10 times the circular pitch, an average value being $4P^1$ or $5P^1$.

A longitudinal cross-section through the axis of the worm cuts out a rack tooth, and this tooth section is usually made of the standard $14\frac{1}{2}°$ involute form shown in Fig. 46 for a rack.

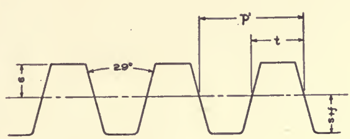


Fig. 46.

The end thrust, of a magnitude practically equal to the pressure between the teeth, has to be taken by the hub of the worm against the face of the shaft bearing. A serious loss of efficiency from friction is likely to occur here. This is often reduced, however, by roller or ball bearings. With two worms on the same shaft, each driving into a separate worm gear, it is possible to make one of the worms right-hand thread, and the other left-hand, in which case the thrust is self-contained in the shaft itself, and there is absolutely no end thrust against the face of the bearing. This involves a double outfit throughout, and is not always practicable.

There are few mathematical equations necessary for the dimensioning of a worm and worm gear. The formulæ for the tooth parts as given on page 120 apply equally well in this case.

PRACTICAL MODIFICATION. The discussion of the efficiency E of the worm and worm gear is more of a practical than of a theoretical nature. It seems to be true from actual operation, as well as theory, that the steeper the threads the higher the efficiency. In actual practice we seldom have opportunity to change the slope of the thread to get increased efficiency. The slope is usually settled from considerations of speed ratio, or available space, or some other condition. The usual practical problem is to take a given worm and worm gear, and to make out of it as efficient a device as possible. With hobbed gears running in oil baths, and with moderate pressures and speeds, the efficiency will range between 40 per cent and 70 per cent. The latter figure is higher than is usually attained.

To avoid cutting and to secure high efficiency, it seems essential to make the worm and the gear of different materials. The worm-thread surfaces being in contact a greater number of times than the gear teeth, should evidently be of the harder material. Hence we usually find the worm of steel, and the gear of cast iron, brass, or bronze. To save the expense of a large and heavy bronze gear, it is common to make a cast-iron center and bolt a bronze rim to it.

The worm, being the most liable to replacement from wear, it is desirable so to arrange its shaft fastening and general accessibility that it may be readily removed without disturbing the worm gear.

The circular pitch of the gear and the pitch of the worm thread must be the same, and the practical question comes in as to the threads per inch possible to be cut in the lathe in the production of the worm thread. The pitch must satisfy this requirement; hence the pitch will usually be fractional, and the diameter of the worm gear, to give the necessary number of teeth, must be brought to it. While it would perhaps be desirable to keep an even diametral pitch for the worm gear, yet it would be poor design to specify a worm thread which could not be cut in a lathe.

The standard involute of $14\frac{1}{2}^\circ$, and the standard proportions of teeth as given on page 120 are usually used for worm threads. This system requires the gear to have at least 30 teeth, for if fewer teeth are used the thread of the worm will interfere with the flanks of the gear teeth. This is a mathematical relation, and there are methods of preventing it by change of tooth proportions or of angle of worm thread; but there are few instances in which less than 30 teeth are required, and it is not deemed worth while to go into a lengthy discussion of this point.

The angle of the worm embraced by the worm-gear teeth varies from 60° to 90°, and the general dimensions of rim are made about the same as for spur gears. The arms, or the web, have the same reasons for their size and shape. Probably web gears and cross-shaped arms are more common than oval or elliptical sections.

Worm gears sometimes have cast teeth, but they are for the roughest service only, and give but a point bearing at the middle of the tooth. An accurately hobbed worm gear will give a bearing clear across the face of the tooth, and, if properly set up and cared for, makes a good mechanical device although admittedly of somewhat low efficiency.

Fig. 47 shows a detail drawing of a standard worm and worm gear. It should serve as a suggestion in design, and an illustration of the shop dimensions required for its production.

PROBLEMS ON SPUR, BEVEL, AND WORM GEARS.

1. Calculate proportions of a standard Brown & Sharpe gear tooth of $1\frac{1}{2}$ diametral pitch, making a rough sketch and putting the dimensions on it.

2. Suppose the above tooth to be loaded at the top with 5,000 lbs. If the face be 6 inches, calculate the fiber stress at the pitch line, due to bending.

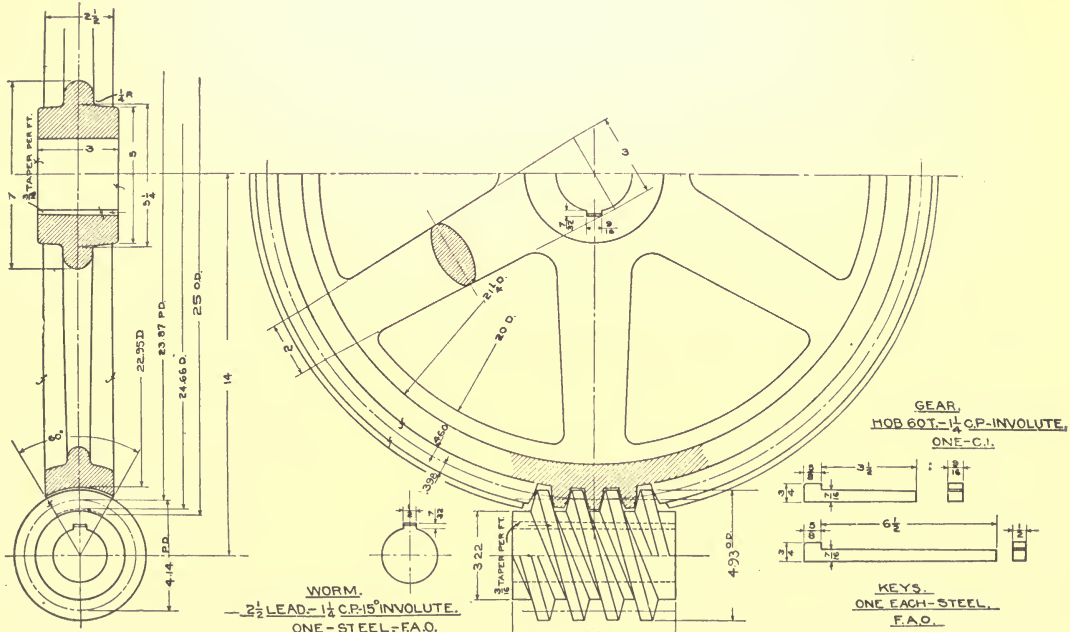
3. A tooth load of 1,200 lbs. is transmitted between two spur gears of 12-inch and 30-inch diameter, the latter gear making 100 revolutions per minute. Calculate a suitable pitch and face of tooth by the "Lewis" formula.

4. Assuming a $\frac{1}{2}$-inch web on the 12-inch gear, calculate the shearing fiber stress at the outside of a hub 4 inches in diameter

5. Design elliptical arms for the 30-inch gear, allowing $S = 2{,}200$.

6. Design cross-shaped arms for 30-inch gear.

7. Calculate the dimensions shown in formulæ 70 to 82 inclusive for a pair of bevel gears of 20 and 60 teeth respectively, 2 diametral pitch, and 4-inch face. (The use of logarithmic tables makes the calculation much easier than with the natural functions.)



SECTION AND ELEVATION OF WORM GEAR.
Fig 47

8. A worm wheel has 40 teeth, 3 diametral pitch, and double thread. Calculate (*a*) its lead; (*b*) its pitch diameter.

FRICTION CLUTCHES.

NOTATION—The following notation is used throughout the chapter on Friction Clutches:.

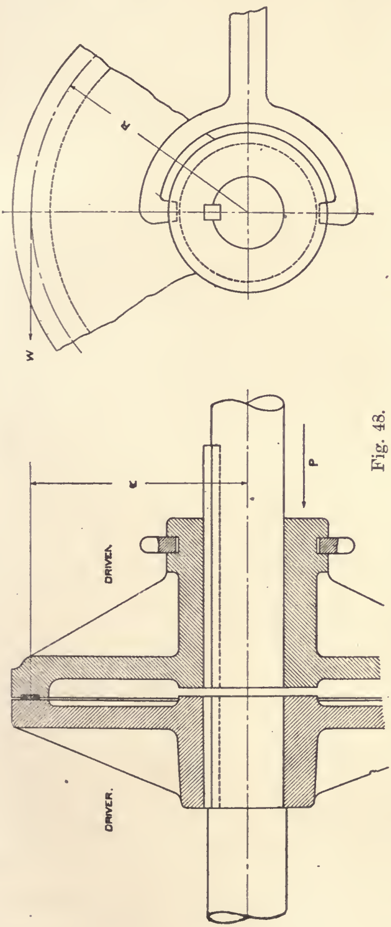


Fig. 43.

- a =Angle between clutch face and axis of shaft (degrees)
- H =Horse-power (33,000 ft.-lbs. per minute).
- μ =Coefficient of friction (per cent).
- N =Number of revolutions per minute.
- P =Force to hold clutch in gear to produce W (lbs.).
- R =Mean radius of friction surface (inches).
- T =Twisting moment about shaft axis (inch-lbs.).
- V =Force normal to clutch face (lbs.).
- W =Load at mean radius of friction surface (lbs.).

ANALYSIS. The friction clutch is a device for connecting at will two separate pieces of shaft, transmitting an amount of power between them to the capacity of the clutch. The connection is usually accomplished while the driving shaft is under full speed, the slipping between the surfaces which occurs during the throwing-in of the clutch, permitting the driven shaft to pick up the speed of the other gradually, without appreciable shock. The disconnection is made in the same manner, the amount of slipping which occurs depending on the suddenness with which the clutch is thrown out.

The force of friction is the sole driving element, hence the problem is to secure as large a force of friction as possible. But friction cannot be secured without a heavy normal pressure between surfaces having a high coefficient of friction between them. The many varieties of friction clutches which are on the market or designed for some special purpose, are all devices for accomplishing one and the same effect, *viz.*, the production of a heavy normal force or pressure between surfaces at such a radius from the driven axis, that the product of the force of friction thereby created and the radius shall equal the desired twisting moment about that axis.

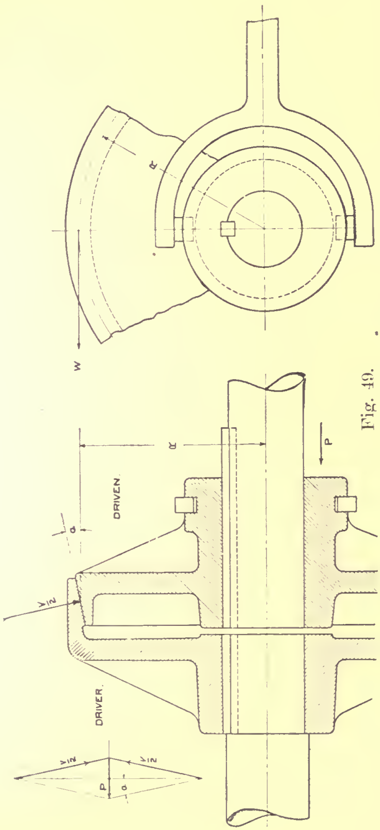


Fig. 49.

Three typical methods of accomplishing this are shown in Figs. 48, 49, and 50. None of these drawings is worked out in operative detail. They are merely illustrations of principle, and are drawn in the simplest form for that purpose.

In Fig. 48 the normal pressure is created in the simplest possible way, an absolutely direct push being exerted between the discs, due to the thrust P of the clutch fork.

In Fig. 49 advantage is taken of the wedge action of the inclined faces, the result being that it takes less thrust P to produce the required normal pressure at the radius R.

In Fig. 50 the inclination of the faces is carried so far that the angle a of Fig. 49 has become zero; and by the toggle-joint action of the link pivoted to the clutch collar, the normal force produced may be very great for a slight thrust P. By careful adjustment of the length of the link so that the jaw takes hold of the clutch surface, when the link stands nearly vertical, a very easy operating device is secured, and the thrust P is made a minimum.

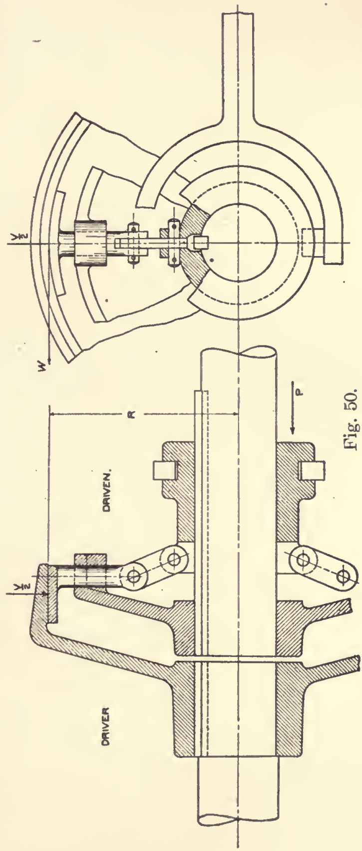


Fig. 50.

THEORY. Referring to Fig. 48 in order to calculate the twisting moment, we must remember that the force of friction between two surfaces is equal to the normal pressure times the coefficient of friction. This, in the form of an equation, using the symbols of the figure, is :

$$W = \mu P. \qquad (88)$$

Hence we may consider that we have a force of magnitude μP acting at the mean radius R of the clutch surface. The twisting moment will then be:

$$T = WR = \mu PR. \qquad (89)$$

Referring to equation 54, which gives twisting moment in terms of horse-power, and putting the two expressions equal to each other, we have:

$$T = \frac{63{,}025H}{N} = \mu PR;$$

or,

$$H = \frac{\mu NPR}{63{,}025}. \qquad (90)$$

This expression gives at once the horse-power that the clutch will transmit with a given end thrust P.

In Fig. 49 the equilibrium of the forces is shown in the little sketch at the left of the figure. The clutch faces are supposed to be in gear, and the extra force necessary to slide the two together is not considered, as it is of small importance. The static equations then are:

$$P = 2\frac{V}{2}\sin\alpha;$$

or,

$$V = P\operatorname{cosec}\alpha. \qquad (91)$$

$$W = \mu V = \mu P\operatorname{cosec}\alpha. \qquad (92)$$

$$T = WR = \mu PR\operatorname{cosec}\alpha. \qquad (93)$$

$$T = \frac{63{,}025H}{N} = \mu PR\operatorname{cosec}\alpha;$$

or,

$$H = \frac{\mu NPR\operatorname{cosec}\alpha}{63{,}025}. \qquad (94)$$

In Fig. 50, P would of course be variable, depending on the inclination of the little link. The amount of horse-power which this clutch would transmit would be the same as in the case of the device illustrated in Fig. 49, for an equal normal force V produced.

The further theoretical design of such clutches should be in accordance with the same principles as for arms and webs of pulleys, gears, etc. The length of the hubs must be liberal in order to prevent tipping on the shaft as a result of uneven wear. The end thrust is apt to be considerable; and extra side stiffness must be provided, as well as a rim that will not spring under the radial pressure.

PRACTICAL MODIFICATION. It is desirable to make the most complicated part of a friction clutch the driven part, for then the mechanism requiring the closest attention and adjustment may be brought to and kept at rest when no transmission of power is desired.

Simplicity is an important practical requirement in clutches. The wearing surfaces are subjected to severe usage; and it is essential that they be made not only strong in the first place, but also capable of being readily replaced when worn out, as they are sure to be after some service.

The form of clutch shown in Fig. 50 is the most efficient form of the three shown, although its commercial design is considerably different from that indicated. Usually the jaws grip both sides of the rim, pinching it between them. This relieves the clutch rim of the radial unbalanced thrust.

Adjusting screws must be provided for taking up the wear, and lock nuts for maintaining their position.

Theoretically, the rubbing surfaces should be of those materials whose coefficient of friction is the highest; but the practical question of wear comes in, and hence we usually find both surfaces of metal, cast iron being most common. For metal on metal the coefficient of friction μ cannot be safely assumed at more than 15 per cent, because the surfaces are sure to get oily.

A leather facing on one of the surfaces gives good results as to coefficient of friction, μ having a value, even for oily leather, of 20 per cent. Much slipping, however, is apt to burn the leather; and this is most likely to occur at high speeds.

Wood on cast iron gives a little higher coefficient of friction for an oily surface than metal on metal. Wood blocks can be so set into the face of the jaws as to be readily replaced when worn, and in such case make an excellent facing.

The angle a of a cone friction clutch of the type shown in Fig. 49, may evidently be made so small that the two parts will wedge together tightly with a very slight pressure P; or it may

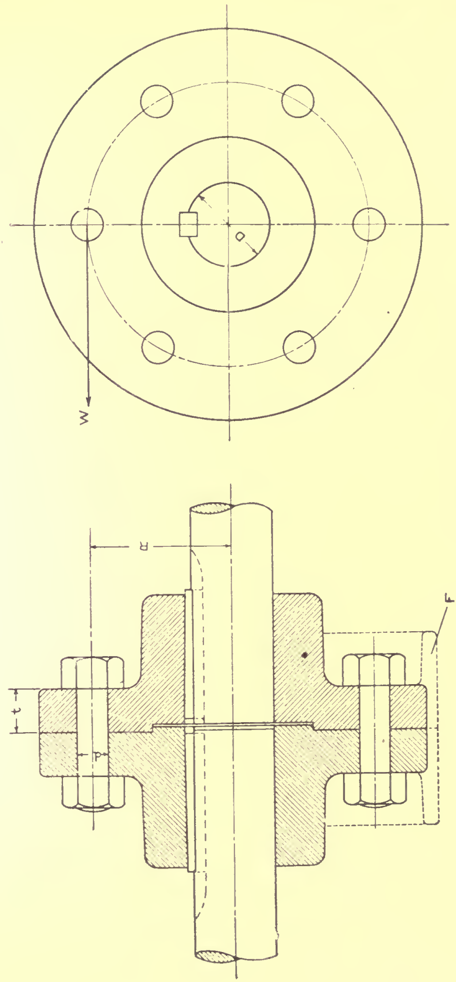


Fig. 51.

be so large as to have little wedging action, and approach the condition illustrated in Fig. 48. Between these limits there is a practical value which neither gives a wedging action so great as to make the surfaces difficult to pull apart, nor, on the other hand, requires an objectionable end thrust along the shaft in order to make the clutch drive properly.

For $a =$ about 15°, the surfaces will free themselves when P is relieved.
" $a =$ " 12°, " " require slight pull to be freed.
" $a =$ " 10°, " " cannot be freed by direct pull of the hand, but require some leverage to produce the necessary force P.

PROBLEMS ON FRICTION CLUTCHES.

1. With what force must we hold a friction clutch in to transmit 30 horse-power at 200 revolutions per minute, assuming working radius of clutch to be 12 inches; coefficient of friction 15 per cent; angle $a = 10°$?

2. How much horse-power could be transmitted, other conditions remaining the same, if the working radius were increased to 18 inches ?

3. What force would be necessary in problem 1, if the angle a were 15°, other conditions remaining the same ?

COUPLINGS.

NOTATION.—The following notation is used throughout the chapter on Couplings:

D =Diameter of shaft (inches).
d =Diameter of bolt body (inches).
n =Number of bolts.
R =Radius of bolt circle (inches).
S =Safe shearing fiber stress (lbs. per sq. in.).
S_c =Safe crushing fiber stress (lbs. per sq. in.).
T =Twisting moment (inch-lbs.).
t=Thickness of flange (inches).
W =Load on bolts (lbs.).

ANALYSIS. Rigid couplings are intended to make the shafts which they connect act as a solid, continuous shaft. In order that the shaft may be worked up to its full strength capacity, the coupling must be as strong in all respects as the shaft, or, in other words, it must transmit the same torsional moment. In the analysis of the forces which come upon these couplings, it is not considered that they are to take any side load, but that they are to act purely as torsional elements. It is doubtless true that in many cases they do have to provide some side strength and stiffness, but this is not their natural function, nor the one upon which their design is based.

Referring to Fig. 51, which is the type most convenient for analysis, we have an example of the simplest form of **flange coupling.** It consists merely of hubs keyed to the two portions, with flanges driving through shear on a series of bolts arranged concentrically about the shaft. The hubs, keys, and flanges are subject to the same conditions of design as the hubs, keys, and web of a gear or pulley, the key tending to shear and be crushed in the hub and shaft, and the hub tending to be torn or sheared from the flange. The driving bolts, which must be carefully fitted in reamed holes, are subject to a purely shearing stress over their full area at the joint, and at the same time tend to crush the metal in the flange, against which they bear, over their projected area. This latter stress is seldom of importance, the thickness of the flange, for practical reasons, being sufficient to make the crushing stress very low.

THEORY. The theory of hubs, keys, and flanges, being like that already given for pulleys and gears, need not be repeated for couplings. The shearing stress on the bolts is the only new point to be studied.

In Fig. 51, for a twisting moment on the shaft of T, the load at the bolt circle is $W = \frac{T}{R}$. If the number of bolts be n, equating the external force to the internal strength, we have:

$$W = \frac{T}{R} = \frac{S\pi d^2}{4} n. \qquad (95)$$

Although the crushing will seldom be of importance, yet for the sake of completeness its equation is given, thus:

$$W = \frac{T}{R} = S_c dtn. \qquad (96)$$

The internal moment of resistance of the shaft is $\frac{SD^3}{5.1}$; hence the equation representing full equality of strength between the shaft and the coupling, depending upon the shearing strength of the bolts, is:

$$\frac{SD^3}{5.1R} = \frac{S\pi d^2}{4} n. \qquad (97)$$

The theory of the other types of couplings is obscure, except as regards the proportions of the key, which are the same in all cases. The shell of the clamp coupling, Fig. 52, should be thick enough to give equal torsional strength with the shaft; but the exact function which the bolts perform is difficult to determine. In general the bolts clamp the coupling tightly on the shaft and provide rigidity, but the key does the principal amount of the driving. The bolt sizes, in these couplings, are based on judgment and relation to surrounding parts, rather than on theory.

PRACTICAL MODIFICATION. All couplings must be made with care and nicely fitted, for their tendency, otherwise, is to

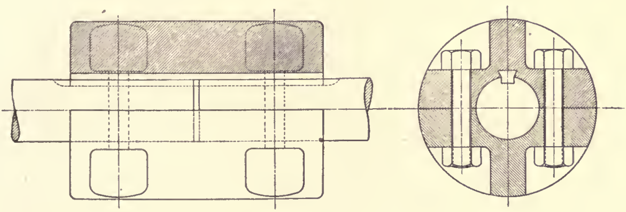


Fig. 52.

spring the shafts out of line. In the case of the flange coupling, the two halves may be keyed in place on the shafts, the latter then swung on centers in the lathe, and the joint faced off. Thus the joint will be true to the axis of the shaft; and, when it is clamped in position by the bolts, no springing out of line can take place.

A flange F (see Fig. 51) is sometimes made on this form of coupling, in order to guard the bolts. It may be used, also, to take a light belt for driving machinery; but a side load is thereby thrown on the shaft at the joint, which is at the very point where it is desirable to avoid it.

The simplest form of rigid coupling consists of a plain sleeve slipped over from one shaft to the other, when the second is butted up against the first. This is known as a **muff coupling.** When once in place, this is a very excellent coupling, as it is perfectly smooth on the outside, and consists of the fewest possible parts, merely a sleeve and a key. It is, however, expensive to fit, difficult to remove, and requires an extra space of half its length on the shaft over which to be slipped back.

The **clamp coupling** is a good coupling for moderate-sized shafts, where the flange type of Fig. 51 would be unnecessarily expensive. The clamp coupling, Fig. 52, is simply a muff coupling split in halves, and recessed for bolts. It is cheap and is easily applied and removed, even with a crowded shaft. If bored with a piece of paper in the joint, when it is clamped in position it will pinch the shaft tightly and make a rigid connection. It is desirable to have the bolt-heads protected as much as possible, and this may be accomplished by making the outside diameter large enough so that the bolts will not project. Often an additional shell is provided to encase the coupling completely after it is located.

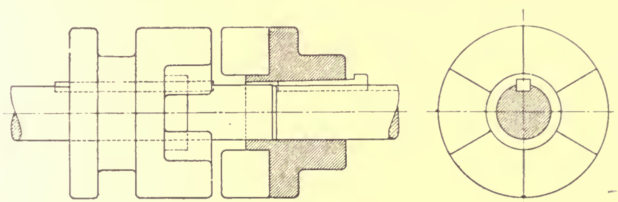


Fig. 53.

There are many other special forms of couplings, some of them adjustable. Most of them depend upon a wedging action exerted by taper cones, screws, or keys. Trade catalogues are to be sought for their description.

The **claw coupling**, Fig. 53, is nothing but a heavy flange coupling with interlocking claws or jaws on the faces of the flanges, to take the place of the driving bolts. This coupling can be thrown in or out as desired, although it usually performs the service of a rigid coupling, as it is not suited to clutching-in during rapid motion, like a friction clutch.

Flexible couplings, which allow slight lack of alignment, are made by introducing between the flanges of a coupling a flexible disc, the one flange being fastened to the inner circle of the disc, the other to the outer circle. This is also accomplished by providing the faces of the flange coupling with pins that drive by pressure together or through leather straps wrapped round the pins. These devices are mostly of a special and often uncertain nature, lacking the positiveness which is one essential feature of a good coupling.

PROBLEMS ON COUPLINGS.

1. A flange coupling of the type of Fig. 51 is usea on a shaft 2 inches is diameter. The hub is 3 inches long, and carries a standard key, of proportions indicated below in the table of "Proportions for Gib Keys" (page 166). The bolt circle is 7 inches in diameter, and it is desired to use $\frac{5}{8}$-inch bolts. How many bolts are needed to transmit 60,000 inch-lbs., for a fiber stress in the bolt of 6,000?

2. Using 6 bolts, what diameter of bolt would be required?

3. If four $\frac{3}{4}$-inch bolts were used on a circle of 8 inches diameter, what diameter of shaft would be used in the coupling to give equal strength with the bolts?

BOLTS, STUDS, NUTS, AND SCREWS.

NOTATION—The following notation is used throughout the chapter on Bolts, Studs, Nuts, and Screws:

d = Diameter of bolt (inches).
d_1 = Diameter at root of thread (inches).
H = Height of nut (inches).
I = Initial axial tension (lbs.).
k = Allowable bearing pressure on surface of thread (lbs. per sq. in.).
L = Lead, or distance nut advances along axis in one revolution (inches).
l = Length of wrench handle (inches)
n = Number of threads in nut= $\frac{H}{p}$.
P = Axial load (lbs.).
p = Pitch of thread, or distance between similar points on adjacent threads, measured parallel to axis (inches).
S = Fiber stress (lbs. per sq. in.).
W = Load on bolt (lbs.).

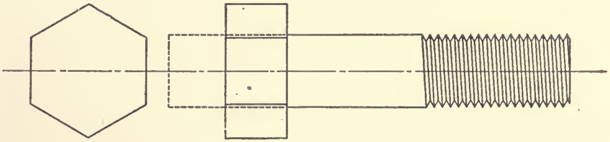


Fig. 54.

ANALYSIS. A **bolt** is simply a cylindrical bar of metal upset at one end to form a head, and having a thread at the other end, Fig. 54. A **stud** is a bolt in which the head is replaced by a thread; or it is a cylindrical bar threaded at both ends, usually having a small plain portion in the middle, Fig. 55. The object of bolts and studs is to clamp machine parts together, and yet permit these same parts to be readily disconnected. The bolt passes through the pieces to be connected, and, when tightened, causes surface compression between the parts, while the reactions on the head and nut produce tension in the bolt. Studs and tap bolts pass through one of the connected parts and are screwed into the other, the stud remaining in position when the parts are disconnected.

As all materials are elastic within certain limits, the action of

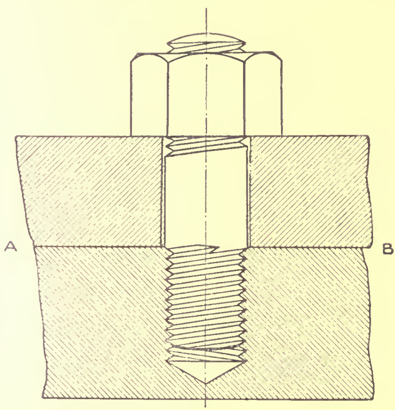


Fig. 55.

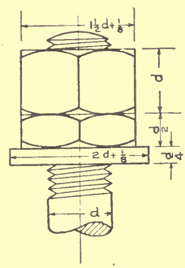


Fig. 55*a*.

a bolt in clamping two machine parts together, more especially if there is an elastic packing between them, may be represented diagrammatically by Fig. 56, in which a spring has been introduced to take the compression due to screwing up the nut. Evidently the tension in the bolt is equal to the force necessary to compress the spring. Now, suppose that two weights, each equal to $\frac{1}{2}$ W, are placed symmetrically on either side of the bolt, then the tension in the bolt will be increased by the added weights if the bolt is perfectly rigid. The bolt, however, stretches; hence some of the compression on the spring is relieved and the total tension in the bolt is less than W + I, by an amount depending on the relative elasticity of the bolt and spring.

Suppose that the stud in Fig. 55 is one of the studs connecting the cover to the cylinder of a steam engine, and that the studs have a small initial tension; then the pressure of the steam loads each stud, and, if the studs stretch enough to relieve the initial pressure between the two surfaces, then their stress is due to the steam pressure only; or, from Fig. 56, when I = W; the initial pressure due to the elasticity of the joint is entirely relieved by the assumed stretch of the studs. Except to prevent leakage, it is seldom necessary to consider the initial tension, for the stretch of the bolt may be counted on to relieve this force, and the working tension on the bolt is simply the load applied.

For shocks or blows, as in the case of the bolts found on the marine type of connecting-rod end, the stretch of the bolts acts like a spring to reduce the resulting tensions. So important is this feature that the body of the bolt is frequently turned down to the diameter of the bottom of the thread, thus uniformly distributing the stretch through the full length of the bolt, instead of localizing it at the threaded parts.

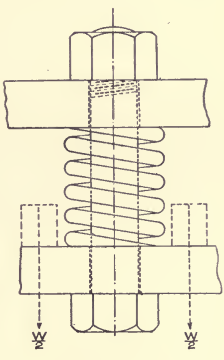


Fig. 56.

In tightening up a bolt, the friction at the surface of the thread produces a twisting moment, which increases the stress in the bolts, just as in the case of shafting under combined tension and torsion; but the increase is small in amount, and may readily be taken care of by permitting low values only for the fiber stress.

In a flange coupling, bolts are acted upon by forces perpendicular to the axis, and hence are under pure shearing stress. If the torque on the shaft becomes too great, failure will occur by the bolts shearing off at the joint of the coupling.

A bolt under tension communicates its load to the nut through the locking of the threads together. If the nut is thin, and the number of threads to take the load few, the threads may break or shear off at the root. With a V thread there is produced a component force, perpendicular to the axis of the bolt, which tends to split the nut.

In screws for continuous transmission of motion and power, the thread may be compared to a rough inclined plane, on which a small block, the nut, is being pushed upward by a force parallel to the base of the plane. The angle at the bottom of the plane is the angle of the helix, or an angle whose tangent is the lead divided by the circumference of the screw. The horizontal force corresponds to the tangential force on the screw. The friction at the surface of the thread produces a twisting moment about the axis of the screw, which, combined with the axial load, subjects the screw to combined tension and torsion. Screws with square threads are generally used for this service, the sides of the thread exerting no bursting pressure on the nut. The proportions of screw thread for transmission of power depend more on the bearing pressure than on strength. If the bearing surface be too small and lubrication poor, the screw will cut and wear rapidly.

THEORY. A direct tensile stress is induced in a bolt when it carries a load exerted along its axis. This load must be taken by the section of the bolt at the bottom of the thread. If the area at the root of the thread is $\frac{\pi d_1^2}{4}$, and if S is the allowable stress per square inch, then the internal resistance of the bolt is $\frac{S\pi d_1^2}{4}$. Equating the external load to the internal strength we have:

$$W = \frac{S\pi d_1^2}{4}. \qquad (98)$$

For bolts which are used to clamp two machine parts together so that they will not separate under the action of an applied load, the initial tension of the bolt must be at least equal to the applied load. If the applied load is W, then the parts are just about to separate when $I = W$. Therefore the above relation for strength is applicable. As the initial tension to prevent separation should be a little greater than W, a value of S should be chosen so that there will be a margin of safety. For ordinary wrought iron and steel, S may be taken at 6,000 to 8,000.

If, however, the joints must be such that there is no leakage between the surfaces, as in the case of a steam cylinder head, and supposing that elastic packings are placed in the joints, then a much larger margin should be made, for the maximum load which may come on the bolt is I + W, where W is the proportional share of the internal pressure carried by the bolt. In such cases S = 3,000 to 5,000, using the lower value for bolts of less than $\frac{3}{4}$-inch diameter.

The table given on page 154 will be found very useful in proportioning bolts with U. S. standard thread for any desired fiber stress.

To find the initial tension due to screwing up the nut, we

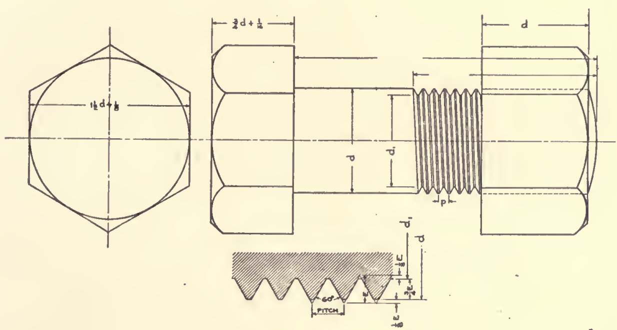


Fig. 56*a*.

may assume the length of the handle of an ordinary wrench, measured from the center of the bolt, as about 16 times the diameter of the bolt. For one turn of the wrench a force F at the handle would pass over a distance $2\pi l$, and the work done is equal to the product of the force and space, or $F \times 2\pi l$. At the same time the axial load P would be moved a distance p along the axis. Assuming that there is no friction, the equation for the equality of the work at the handle and at the screw is:

$$F 2\pi l = Pp. \qquad \textbf{(99)}$$

Friction, however, is always present; hence the ratio of the useful work (Pp) to the work applied ($F2\pi l$) is not unity as above re-

TABLE FOR STRENGTH OF BOLTS. U. S. Standard Thread.

Bolt.		Diameters.		Areas.		Approximate Tensile Strength at Bottom of Thread. (In Hundreds of Pounds.)					Approximate Shearing Strength. Full Bolt Diameter. (In Hundreds of Pounds.)			
Diameter. Inches.	Threads per inch.	Bottom of Thread.	Tap Drill.	Bolt Body.	Bottom of Thread.	At 4,000 lbs. per sq. in.	At 5,000 lbs. per sq. in.	At 6,000 lbs. per sq. in.	At 7,000 lbs. per sq. in.	At 10,000 lbs. per sq. in.	At 4,000 lbs. per sq. in.	At 5,000 lbs. per sq. in.	At 6,000 lbs. per sq. in.	At 7,000 lbs. per sq. in.
1/4	20	.18	3/16	.05	.03	1.08	1.35	1.60	1.88	2.69	1.96	2.15	2.95	3.43
5/16	18	.24	1/4	.08	.04	1.82	2.27	2.72	3.18	4.51	3.04	3.83	4.60	5.37
3/8	16	.29	5/16	.11	.07	2.71	3.39	4.07	4.75	6.78	4.40	5.52	6.62	7.73
7/16	14	.34	23/64	.15	.09	3.73	4.67	5.60	6.53	9.33	6.00	7.51	9.02	11.00
1/2	13	.40	13/32	.20	.13	5.00	6.25	7.50	8.75	12.00	7.84	9.81	12.00	14
9/16	12	.45	15/32	.25	.16	6.18	8.10	9.62	11.00	16	9.92	12.00	15	17
5/8	11	.51	17/32	.31	.20	8.04	10.00	12.00	14	20	12.00	15	18	21
3/4	10	.62	5/8	.44	.30	12.00	15	18	21	30	18	22	26	31
7/8	9	.73	3/4	.60	.42	17	21	25	29	42	24	30	36	42
1	8	.84	27/32	.78	.55	22	27	33	38	55	31	39	47	55
1 1/8	7	.94	31/32	.99	.69	28	34	41	48	69	40	50	60	70
1 1/4	7	1.06	1 3/32	1.23	.89	31	39	47	55	78	49	61	74	86
1 3/8	6	1.16	1 3/16	1.48	1.05	42	53	64	72	106	59	74	89	104
1 1/2	6	1.28	1 9/32	1.77	1.29	51	64	77	90	128	71	88	106	124
1 5/8	5 1/2	1.39	1 13/32	2.07	1.51	61	78	92	109	153	83	104	124	145
1 3/4	5	1.49	1 1/2	2.40	1.71	70	88	106	123	176	96	120	144	168
1 7/8	5	1.61	1 5/8	2.76	2.05	81	101	122	142	203	110	138	166	193
2	4 1/2	1.74	1 3/4	3.14	2.30	92	115	138	161	230	126	157	186	220
2 1/4	4 1/2	1.96	1 31/32	3.98	3.02	125	156	187	218	312	159	199	238	278
2 1/2	4	2.17	2 3/16	4.91	3.72	148	185	222	259	370	196	245	294	344
2 3/4	4	2.42	2 7/16	5.94	4.62	184	230	276	322	460	237	297	356	416
3	3 1/2	2.63	2 5/8	7.07	5.43	218	272	326	381	544	283	353	424	495
3 1/4	3 1/2	2.88	2 29/32	8.29	6.51	264	330	396	462	660	332	415	498	581
3 1/2	3 1/4	3.10	3 1/32	9.62	7.55	302	376	452	528	754	385	481	577	673
3 3/4	3	3.32	3 11/32	11.04	8.64	344	430	516	602	860	412	552	663	773
4	3	3.57	3 19/32	12.57	9.99	396	495	594	693	990	503	628	754	880
4 1/4	2 7/8	3.80	3 13/16	14.19	11.33	452	565	678	791	1,130	567	709	851	993
4 1/2	2 3/4	4.03	4 1/32	15.90	12.71	507	631	760	888	1,268	636	795	954	1,113
4 3/4	2 5/8	4.25	4 9/32	17.72	14.22	567	709	851	1,993	1,420	709	886	1,063	1,240
5	2 1/2	4.48	4 1/2	19.63	15.76	630	788	946	1,103	1,676	785	982	1,178	1,374
5 1/2	2 3/8	4.95	4 31/32	23.76	19.27	770	962	1,154	1,347	1,924	950	1,188	1,425	1,663
6	2 1/4	5.42	5 7/16	28.27	23.09	923	1,159	1,384	1,617	2,307	1,128	1,414	1,696	1,979

lations assume. From numerous experiments on the friction of screws and nuts, it has been found that the efficiency may be as low as 10 per cent. Introducing the efficiency in above equation, it may be written:

$$\frac{Pp}{F2\pi l} = \frac{1}{10}. \qquad (100)$$

Assuming that 50 pounds is exerted by a workman in

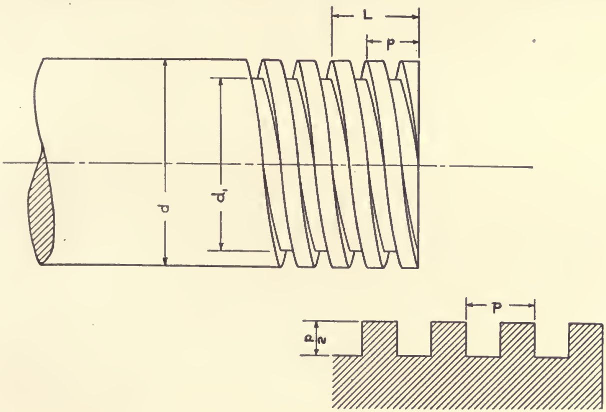


Fig. 58.

tightening up the nut on a 1-inch bolt, the equation above shows that P = 4,021 pounds; or the initial tension is somewhat less than the tabular safe load shown for a 1-inch bolt, with S assumed at 10,000 pounds per sq. inch.

For shearing stresses the bolt should be fitted so that the body of the bolt, not the threads, resists the force tending to shear off the bolt perpendicular to its axis. The internal strength of the bolt to resist shear is the allowable stress S times the area of the bolt in shear, or $\frac{S\pi d^2}{4}$. If W represents the external force tending to shear the bolt the equality of the external force to the internal strength is:

$$W = \frac{S\pi d^2}{4}. \qquad (101)$$

Reference to the table on page 154 for the shearing strength of bolts, may be made to save the labor of calculations.

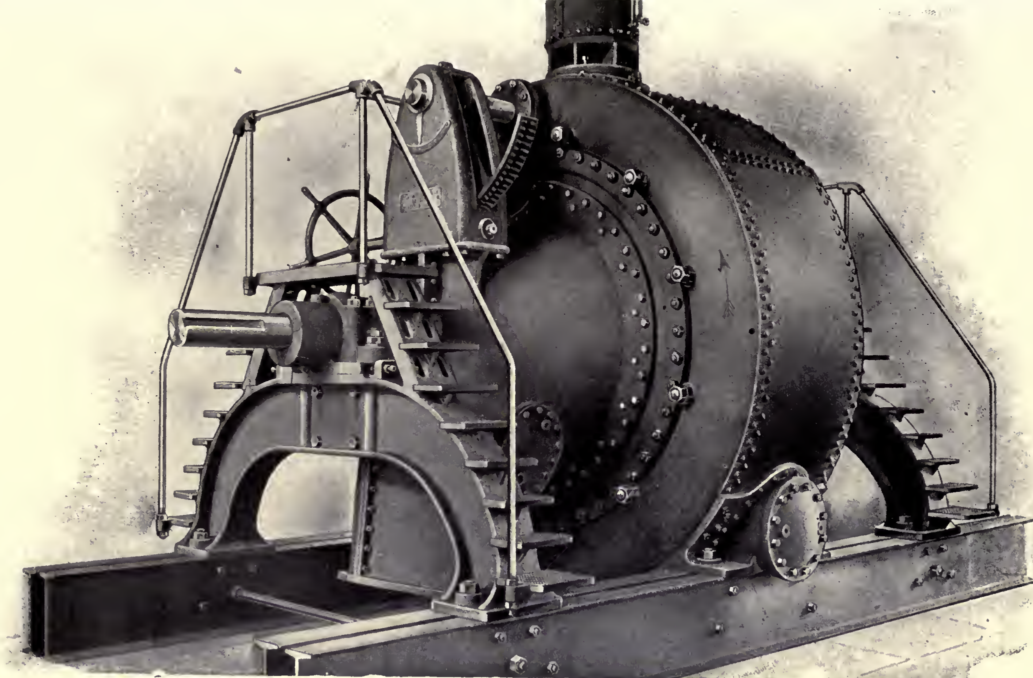
Let Fig. 58 represent a square thread screw for the transmission of motion. The surface on which the axial pressure bears, if n is the number of threads in a nut, is $\frac{\pi}{4}(d^2 - d_1^2)\ n$. Suppose that a pressure of k pounds per square inch is allowed on the surface of the thread. Then the greatest permissible axial load P must not exceed the allowable pressure; or, equating,

$$P = k\frac{\pi}{4}(d^2 - d_1^2)\ n \qquad (102)$$

The value of k varies with the service required. If the motion be slow and the lubrication very good, k may be as high as 900. For rapid motion and doubtful lubrication, k may not be over 200. Between these extremes the designer must use his judgment, remembering that the higher the speed the lower is the allowable bearing pressure.

PRACTICAL MODIFICATION. It will be noted in the formulæ for bolt strengths that different values for S are assumed. This is necessary on account of the uncertain initial stresses which are produced is setting up the nuts. For cases of mere fastening, the safe tension is high, as just before the joint opens the tension is about equal to the load and yet the fastening is secure. On the other hand, bolts or studs fastening joints subjected to internal fluid pressure must be stressed initially to a greater amount than the working pressure which is to come on the bolt. As this initial stress is a matter of judgment on the part of the workman, the designer, in order to be on the safe side, should specify not less than $\frac{5}{8}$-inch or $\frac{3}{4}$-inch bolts for ordinary work, so that the bolts may not be broken off by a careless workman accidentally putting a greater force than necessary on the wrench handle. In making a steam-tight joint, the spacing of the bolts will generally determine their number; hence we often find an excess of bolt strength in joints of this character.

Through bolts are preferred to studs, and studs to tap bolts or cap screws. If possible, the design should be such that through bolts may be used. They are cheapest, are always in standard



HIGH DUTY SAMPSON TURBINE. NIAGARA DESIGN.
James Leffel and Co.

stock, and well resist rough usage in connecting and disconnecting. The threads in cast iron are weak and have a tendency to crumble; and if a through bolt cannot be used in such a case, a stud, which can be placed in position once for all, should be employed—not a tap bolt, which injures the thread in the casting every time it is removed.

The plain portion of a stud should be screwed up tight against the shoulder, and the tapped hole should be deep enough to prevent bottoming. To avoid breaking off the stud at the shoulder, a neck, or groove, may be made at the lower end of the thread entering the nut.

To withstand shearing forces the bolts must be fitted so that no lost motion may occur, otherwise pure shearing will not be secured.

Nuts are generally made hexagonal, but for rough work are often made square. The hexagonal nut allows the wrench to turn through a smaller angle in tightening up, and is preferred to the square nut. Experiments and calculations show that the height of the nut with standard threads may be about $\frac{1}{2}$ the diameter of the bolt and still have the shearing strength of the thread equal to the tensile strength of the bolt at the root of the thread. Practically, however, it is difficult to apply such a thin wrench as this proportion would call for on ordinary bolts. More commonly the height of the nut is made equal to the diameter of the bolt so that the length of thread will guide the nut on the bolt, give a low bearing pressure on the threads, and enable a suitable wrench to be easily applied. The standard proportions for bolts and nuts may be found in any handbook. Not all manufacturers conform to the United States standard; nor do manufacturers in all cases conform to one another in practice.

If the bolt is subject to vibration, the nuts have a tendency to loosen. A common method of preventing this is to use double nuts, or **lock nuts**, as they are called (see Fig. 55 A). The under nut is screwed tightly against the surface, and held by a wrench while the second nut is screwed down tightly against the first. The effect is to cause the threads of the upper nut to bear against the under sides of the threads of the bolt. The load on the bolt is sustained therefore by the upper nut, which should be the thicker of the two ; but for convenience in applying wrenches the position of the nuts is often reversed.

The form of thread adapted to transmitting power is the square thread, which, although giving less bursting pressure on the nut, is not as strong as the V thread for a given length, since the total section of thread at the bottom is only $\frac{1}{2}$ as great. If the pressure is to be transmitted in but one direction, the two

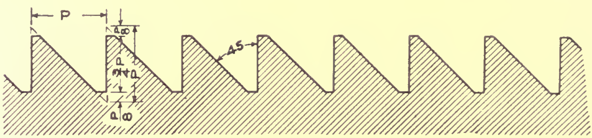


Fig. 59.

types may be combined advantageously to form the buttress thread of the proportions shown in Fig. 59. Often, as in the carriage of a lathe, to allow the split nut to be opened and closed over the lead screw, the sides of the thread are placed at a small angle, say 15°, to each other, as illustrated in Fig. 60.

The practical commercial forms in which we find screwed fastenings are included in five classes, as follows:

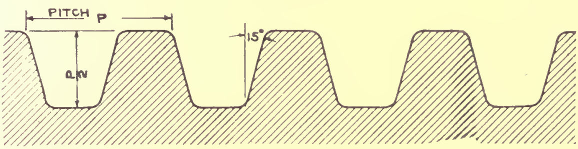


Fig. 60.

1. **Through bolts** (Fig. 61), usually rough stock, with square upset heads, and square or hexagonal nuts.

2. **Tap bolts** (Fig. 62), also called **cap screws**. These usually have hexagonal heads, and are found both in the rough form, and finished from the rolled hexagonal bar in the screw machine.

3. **Studs** (Fig. 63), rough or finished stock, threaded in the screw machine.

4. **Set screws** (Fig. 64), usually with square heads, and case-hardened points. Many varieties of set screws are made, the

principal distinguishing feature of each being in the shape of the point. Thus, in addition to the plain beveled point, we find the "cupped," rounded, conical, and "teat" points.

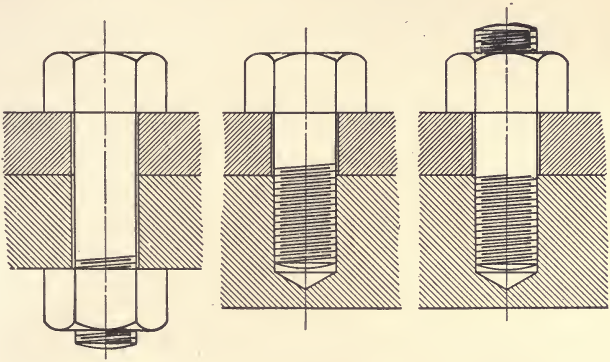


Fig. 61. Fig. 62. Fig. 63.

5. **Machine screws** (Fig. 64*a*), usually round, "button," or countersunk head. Common proportions are indicated relative to diameter of body of screw.

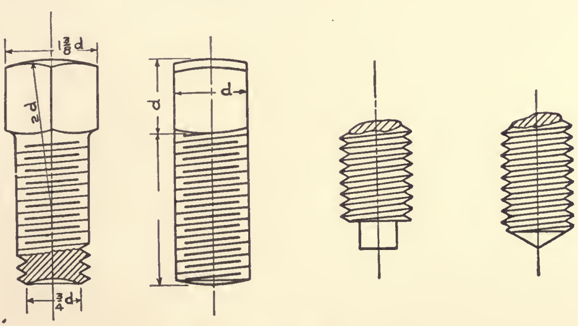


Fig. 64.

PROBLEMS ON BOLTS, STUDS, NUTS, AND SCREWS.

1. Calculate the diameter of a bolt to sustain a load of 5,000 lbs.

2. The shearing force to be resisted by each of the bolts of a flange coupling is 1,200 lbs. What commercial size of bolt is required?

3. With a wrench 16 times the diameter of the bolt, and an efficiency of 10 per cent, what axial load can a man exert on a standard $\frac{3}{4}$-inch bolt, if he pulls 40 lbs. at the end of the wrench handle?

4. A single, square-threaded screw of diameter 2 inches, lead $\frac{1}{4}$ inch, depth of thread $\frac{1}{8}$ inch, length of nut 3 inches, is to be allowed a bearing pressure of 300 lbs. per square inch. What axial load can be carried?

5. Calculate the shearing stress at the root of the thread in problem 4.

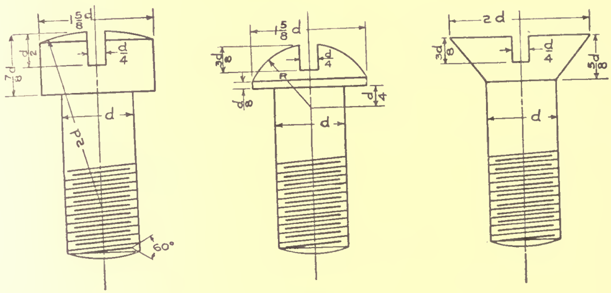


Fig. 64*a*.

KEYS, PINS, AND COTTERS.

NOTATION—The following notation is used throughout the chapter on Keys, Pins, and Cotters:

D = Average diameter of rod (inches).
D_1 = Outside diameter of socket (inches).
d = Diameter of shaft (inches).
L = Length of key (inches).
P = Driving force (lbs.).
P_1 = Axial load on rod (lbs.).
R = Radius at which P acts (inches).
S_c = Safe crushing fiber stress (lbs. per sq. in.).
S_s = Safe shearing fiber stress (lbs. per sq. in.).
S_t = Safe tensile fiber stress (lbs. per sq. in.).
T = Thickness of key (inches).
W = Width of key (inches).
w = Average width of cotter (inches).
w_1 = End of slot to end of rod (inches).
w_2 = End of slot to end of socket (inches).

KEYS AND PINS.

ANALYSIS. Keys and pins are used to prevent relative rotary motion between machine parts intended to act together as one piece. If we drill completely through a hub and across the shaft, and insert a tightly fitted pin, any rotary motion of the one will be transmitted to the other, provided the pin does not fail by shearing off at the joint between the shaft and the hub. The shearing area is the sum of the cross-sections of the pin at the joint.

We may drill a hole in the joint, the axis of the hole being parallel to the axis of the shaft, and drive in a pin, in which case we introduce a shearing area as before, but the area is now equal to the diameter of the pin multiplied by its length, and the pin is stressed sidewise, instead of across. It is evident in the sidewise case that we may increase the shearing area to anything we please, without changing the diameter of the pin, merely by increasing the length of the pin.

As there are some manufacturing reasons why a round pin placed lengthwise in the joint is not always applicable, we may make the pin a rectangular one, in which case it is called a **key.**

When pins are driven across the shaft as in the first instance, they are usually made taper. This is because it is easier to ream a taper hole to size than a straight hole, and a taper pin will drive more easily than a straight pin, it not being necessary to match the hole in hub and shaft so exactly in order that the pin may enter. The taper pin will draw the holes into line as it is driven, and can be backed out readily in removal.

Keys of the rectangular form are either straight or tapered, but for different reasons from those just stated for pins. Straight keys have working bearing only at the sides, driving purely by shear, crushing being exerted by the side of the key in both shaft and hub, over the area against the key. The key itself does not prevent end motion along the shaft; and if end motion is not desired, auxiliary means of some sort must be resorted to, as, for example, set screws through the hub jamming hard against the top of the key.

If end motion along the shaft is desired, the key is called a **spline,** and, while not jammed against the shaft, is yet prevented from changing its relation to the hub by some means such as illustrated in Fig. 65.

Taper keys not only drive through sidewise shearing strength, but prevent endwise motion by the wedging action exerted between the shaft and hub. These keys drive more like a strut from corner to corner; but this action is incidental rather than intentional, and the proportions of a taper key should be such that it will give its full resisting area in shearing and crushing, the same as a straight key.

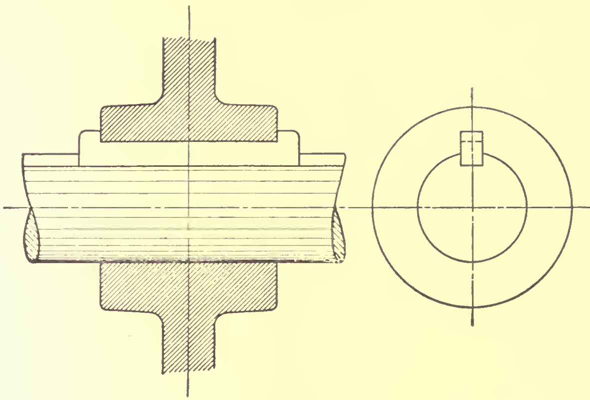


Fig. 65.

THEORY. Suppose that the pin illustrated in Fig. 66 passes through hub and shaft, and the driving force P acts at the radius R; then the force which is exerted at the surface of the shaft to shear off the pin at the points A and B is $\frac{2\,PR}{d}$. If D_1 is the average diameter of the pin, its shearing strength is $\frac{2\pi D_1^2 S_s}{4}$. Equating the external force to the internal strength, we have:

$$\frac{2PR}{d} = \frac{2\pi D_1^2\, S_s}{4};$$

or,

$$D_1 = \sqrt{\frac{4PR}{\pi d S_s}}. \qquad (103)$$

In Fig. 67 a rectangular key is sunk half way in hub and shaft according to usual practice. Here the force at the surface of the shaft, calculated the same as before, not only tends to shear off the key along the line AB, but tends to crush both the portion in the shaft and in the hub. The shearing strength along the

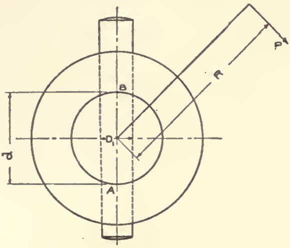


Fig. 66.

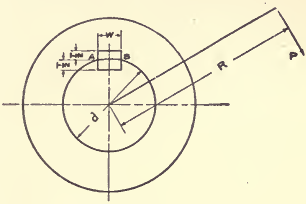


Fig. 67.

line AB is LWS_s. Equating external force to internal strength, we have:

$$\frac{2PR}{d} = LWS_s;$$

or,

$$W = \frac{2PR}{dLS_s}. \qquad (104)$$

The crushing strength is, of course, that due to the weaker metal, whether in shaft or hub. Let S_c be this least safe crushing fiber stress. The crushing strength then is $\frac{LT}{2}$ S_c, and, equating external force to internal strength, we have:

$$\frac{2PR}{d} = \frac{LT}{2} S_c;$$

or,

$$T = \frac{4PR}{dLS_c}. \qquad (105)$$

The proportions of the key must be such that the equations as above, both for shearing and for crushing, shall be satisfied.

PRACTICAL MODIFICATION. Pins across the shaft can be used to drive light work only, for the shearing area cannot be very large. A large pin cuts away too much area of the shaft, decreasing the latter's strength, Pins are useful in preventing end motion, but in this case are expected to take no shear, and may be of small diameter. The common split pin is especially adapted to this service, and is a standard commercial article.

Taper pins are usually listed according to the Morse standard taper, proportions of which may be found in any handbook. It is desirable to use *standard* taper pins in machine construction, as the reamers are a commercial article of accepted value, and readily obtainable in the machine-tool market.

With properly fitted keys, the shearing strength is usually the controlling element. For shafts of ordinary size, the standard proportions as given in tables like that below are safe enough without calculation, up to the limit of torsional strength of the shaft. For special cases of short hubs or heavy loads, a calculation is needed to check the size, and perhaps modify it.

Splines, also known as "feather keys," require thickness greater than regular keys, on account of the sliding at the sides. A table suggesting proportions for splines is given on page 166.

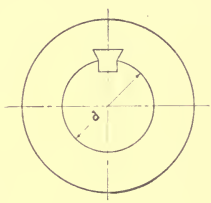


Fig. 67*a*.

Though the spline may be either in the shaft or hub, it is the more usual thing to find the spline dovetailed (Fig. 67*a*), "gibbed," or otherwise fastened in the hub; and a long spline way made in the shaft, in which it slides.

The straight key, accurately fitted, is the most desirable fastening device for accurate machines, such as machine tools, on account of the fact that there is absolutely no radial force exerted to throw the parts out of true. It, however, requires a tight fit of hub to shaft, as the key cannot be relied upon to take up any looseness.

The taper key (Fig. 68), by its wedging action, will take up some looseness, but in so doing throws the parts out slightly. Or, even if the bored fit be good, if the taper key be not driven home with care, it will spring the hub, and make the parts run untrue. The great advantage, however, that the taper key has of holding the hub from endwise motion, renders it a very useful and practical article. It is usually provided with a head, or "gib," which permits a draw hook to be used to wedge between the face of the hub and the key to facilitate starting the key from its seat.

Two keys at 90° from each other may be used in cases where one key will not suffice. The fine workmanship involved in spacing these keys so that they will drive equally makes this plan inadvisable except in case of positive and unavoidable necessity.

The "Woodruff" key (Fig. 69) is a useful patented article for certain locations. This key is a half-disc, sunk in the shaft

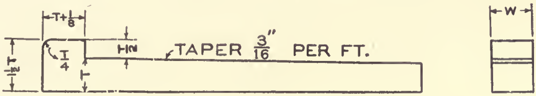


Fig. 68.

and the hub is slipped over it. A simple rotary cutter is dropped into the shaft to produce the key seat; and on account of the depth in the shaft, the tendency to rock sidewise is eliminated, and the drive is purely by shear.

Keys may be milled out of solid stock, or drop-forged to within a small fraction of finished size. The drop-forged key is an excellent modern production and requires but a minimum

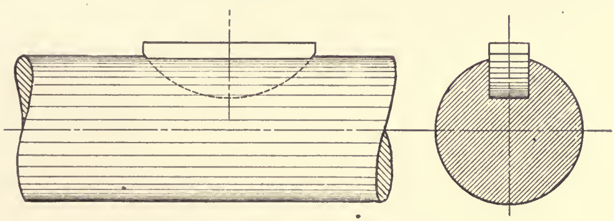


Fig. 69.

amount of fitting. Any key, no matter how produced, requires some hand fitting and draw filing to bring it properly to its seat and give it full bearing.

It is good mechanical policy to avoid keyed fastenings whenever possible. This does not mean that keys may never be used, but that a key is not an ideal way to produce an absolutely positive drive, partly because it is an expensive device, and partly because the tendency of any key is to work itself loose, even if carefully fitted.

The following tables are suggested as a guide to proportions of gib keys and feather keys, and will be found useful in the absence of any manufacturer's standard list:

Fig. 70. **PROPORTIONS FOR GIB KEYS.**

Diameter of shaft (d), inches.	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{5}{8}$	2	$2\frac{1}{2}$	$3\frac{1}{4}$	4	5	$6\frac{1}{2}$
Width (W), inches.	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{5}{16}$	$1\frac{5}{8}$
Thickness (T), inches.	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{17}{32}$	$\frac{21}{32}$	$\frac{13}{16}$	1	$1\frac{1}{4}$

Fig. 71. **PROPORTIONS FOR FEATHER KEYS.**

Diameter of Shaft (d), inches,	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$
Width (W), inches.	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$
Thickness (T), inches.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$

COTTERS.

ANALYSIS. Cotters are used to fasten hubs to rods rather than shafts, the distinction between a rod and a shaft being that a rod takes its load in the direction of its length, and does not drive by rotation. A cotter, therefore, is nothing but a cross-pin of modified form, to take shearing and crushing stress in the direction of the axis of the rod, instead of perpendicular to it.

Referring to Fig. 72, one will see that the cotter is made long and thin—long, in order to get sufficient shearing area to resist shearing along lines A and B; thin, in order to cut as little cross-sectional area out of the body of the shaft as possible. The cotter itself tends to shear along the lines A and B, and crush along the surfaces K, G, and J. The socket tends to crush along the surfaces K and G. The rod end D tends to be sheared out along the lines O H and Q E, and also to be crushed along the surface J. The socket tends to be sheared along the lines V U and X Y.

The cotter is made taper on one side, thus enabling it to draw up the flange of the rod tightly against the head of the socket. This taper must not be great enough to permit easy "backing out" and loosening of the cotter under load or vibration in the rod. In responsible situations this cannot be safely guarded against except through some auxiliary locking device, such as lock nuts on the end of the cotter (Fig. 73).

THEORY. Referring to Fig. 72, assume an axial load of P_1, as shown. The successive equations of external force to internal strength are enumerated below, for the different actions that take place:

For shearing along lines A and B, w being the average width of cotter, and S_s safe shearing stress of cotter,

$$P_1 = 2TwS_s. \qquad (106)$$

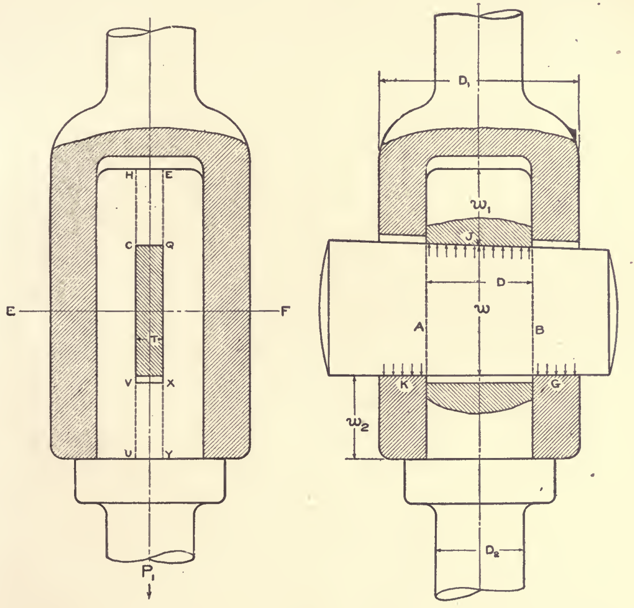


Fig. 72.

For crushing along surfaces K and G, S_c being least safe crushing stress, whether of cotter or socket,

$$P_1 = T(D_1 - D)S_c. \qquad (107)$$

For crushing along surface J, S_c being least safe crushing stress, whether of cotter or socket,

$$P_1 = DTS_c. \qquad (108)$$

For shearing along surfaces CH and QE, S_s being safe shearing stress of rod end, and w_1 end of slot to end of rod,

$$P_1 = 2w_1DS_s. \qquad \textbf{(109)}$$

For tension in rod end at section across slot, S_t being safe tensile stress in rod end,

$$P_1 = \left(\frac{\pi D^2}{4} - TD\right)S_t. \qquad \textbf{(110)}$$

For tension in socket at section across slot, S_t being safe tensile stress in socket,

$$P_1 = \left[\frac{\pi D_1^2}{4} - \frac{\pi D^2}{4} - T(D_1 - D)\right]S_t. \qquad \textbf{(111)}$$

For shearing in socket along the lines VU and XY, S_s being safe shearing stress in the socket, and w_2 end of slot to end of socket,

$$P_1 = 2w_2(D_1 - D)S_s. \qquad \textbf{(112)}$$

The proportions of cotter and socket may be fixed to some extent by practical or assumed conditions. The dimensions may then be tested by the above equations, that the safe working stresses may not be exceeded, the dimensions being then modified accordingly.

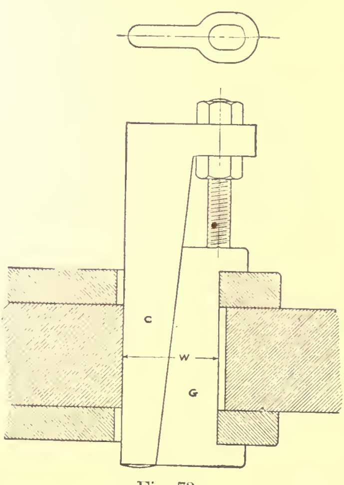


Fig. 73.

The steel of which both cotter and rod would ordinarily be made has range of working fiber stress as follows :

Tension, 8,000 to 12,000 (lbs. per sq. in.)
Compression, 10,000 to 16,000 (lbs per sq. in.)
Shear, 6,000 to 10,000 (lbs. per sq. in.)

The socket, if made of cast iron, will be weak as regards tension, tendency to shear out at the end, and tendency to split. The uncertainty of cast iron to resist these is so great that the hub or socket must be very clumsy in order to have enough surplus strength. This is always a noticeable feature of the cotter type of fastening, and cannot well be avoided.

PRACTICAL MODIFICATION. The driving faces of the cotter are often made semicircular. This not only gives more shearing area at the sides of the slots, but makes the production of the slots easier in the shop. It also avoids the general objection to sharp corners—namely, a tendency to start cracks.

A practicable taper for cotters is $\frac{1}{2}$ inch per foot. This will under ordinary circumstances prevent the cotter from backing out under the action of the load. When set screws against the side of the cotter, or lock nuts are used, as in Fig. 73, the taper may be greater than this, perhaps as much as $1\frac{1}{2}$ inches per foot.

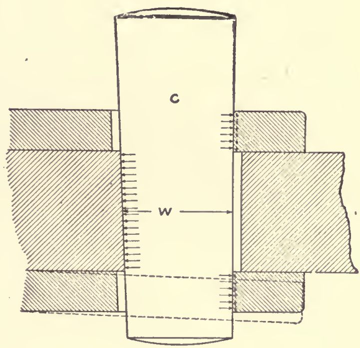


Fig. 74.

In the common use of the cotter for holding the strap at the ends of connecting rods, the strap acts like a modified form of socket. This is shown in Figs. 73 and 74. Here, in addition to holding the strap and rod together lengthwise, it may be necessary to prevent their spreading, and for this purpose an auxiliary piece G with gib ends is used. The tendency without this extra piece is shown by the dotted lines in Fig. 74.

The general mechanical fault with cottered joints is that the action of the load, especially when it constantly reverses, as in pump piston rods, always tends to work the cotter loose. Vibration also tends to produce the same effect. Once this looseness is started in the joint, the cotter loses its pure crushing and shearing action, and begins to partake of the nature of a hammer, and pounds itself and its bearing surfaces out of their true shape. Instead of a collar on the rod, we often find a taper fit of the rod in the socket; and any looseness in this case is still worse, for the rod then has end play in the socket, and by its "shucking" back and forth tends to split open the socket.

The only answer to these objections is to provide a positive locking device, and take up any looseness the instant it appears.

PROBLEMS ON KEYS, PINS, AND COTTERS.

1. Calculate the safe load in shear which can be carried on a key $\frac{1}{2}$ inch wide, $\frac{3}{8}$ inch thick, and 5 inches long. Assume $S_s = 6{,}000$.

2. Assuming the above key to be $\frac{3}{16}$ inch in hub and $\frac{3}{16}$ inch in shaft, test its proportions for crushing, at $S_c = 16{,}000$.

3. A gear 60 inches in diameter has a load of 3,000 lbs. at the pitch line. The shaft is 4 inches in diameter, in a hub, 5 inches long; and the key is a standard gib key as given in the table. Test its proportions for shearing.

4. A piston rod 2 inches in diameter carries a cotter $\frac{3}{8}$ inch thick, and has an axial load of 20,000 lbs. Calculate the average width of the cotter. $S_s = 9{,}000$.

5. Calculate fiber stress in rod in preceding problem at section through slot.

6. How far from the end of rod must the end of slot be?

7. Calculate the crushing fiber stresses on cotter, rod, and socket.

8. How far from the end of socket must the end of slot be, assuming the socket to be of steel?

BEARINGS, BRACKETS, AND STANDS.

NOTATION—The following notation is used throughout the chapters on Bearings Brackets, and Stands.

A = Area (square inches).
a = Distance between bolt centers (inches).
b = Width of bracket base (inches).
c = Distance of neutral axis from outer fiber (inches).
D = Diameter of shaft (inches).
d = Diameter of bolt body (inches).
d_1 = Diameter at root of thread (inches).
H = Horse-power.
h = Thickness of cap at center (inches).
I = Moment of inertia.
L = Length of bearing (inches).
μ = Coefficient of friction (per cent).
N = Number of revolutions per minute.
n = Number of bolts in cap.
n_1 = Number of bolts in bracket base.
P = Total pressure on bearing (lbs.).
p = Pressure per square inch of projected area (lbs).
S = Safe tensile fiber stress (lbs.).
S_s = " shearing " (lbs.).
T = Total load on bolts at top of bracket (lbs.).
t = Thickness of bracket base (inches).
x = Distance from line of action of load to any section of bracket (inches).

ANALYSIS. Machine surfaces taking weight and pressure of other parts in motion upon them are, in general, known as **bearings.** If the motion is rectilinear, the bearing is termed a **slide, guide,** or **way,** such as the cross slide of a lathe, the cross-head guide of a steam engine, or the ways of a lathe bed.

If the motion is a rotary one, like that of the spindle of a lathe, the simple word "bearing" is generally used.

In any bearing, sliding or rotary, there must be strength to carry the load, stiffness to distribute the pressure evenly over the full bearing surface, low intensity of such pressure to prevent the lubricant from being squeezed out and to minimize the wear, and sufficient radiating surface to carry away the heat generated by friction of the surfaces as fast as it is generated. Sliding bearings are of such varied nature, and exist under conditions so peculiar to each case, that a general analysis is practically impossible beyond that given in the sentence above.

Rotary bearings can be more definitely studied, as there are but two variable dimensions, diameter and length, and it is the proper relation between these two that determines a good bearing. The size of the shaft, as noted under "Shafts," is calculated by taking the bending moment at the center of the bearing, combining it with the twisting moment, and solving for the diameter consistent with the assumed fiber stress. But this size must then be tried for deflection due to the bending load, in order that the requirement for stiffness may be fulfilled. When this is accomplished, the friction at the bearing surface may still generate so much heat that the exposed surface of the bearing will not radiate it as fast as generated, in which case the bearing gets hotter and hotter, until it finally burns out the lubricant and melts the lining of the bearing, and ruin results.

The heat condition is usually the critical one, as it is very easy to make a short bearing which is strong enough and amply stiff for the load it carries, but which nevertheless is a failure as a bearing, because it has so small a radiating surface that it cannot run cool.

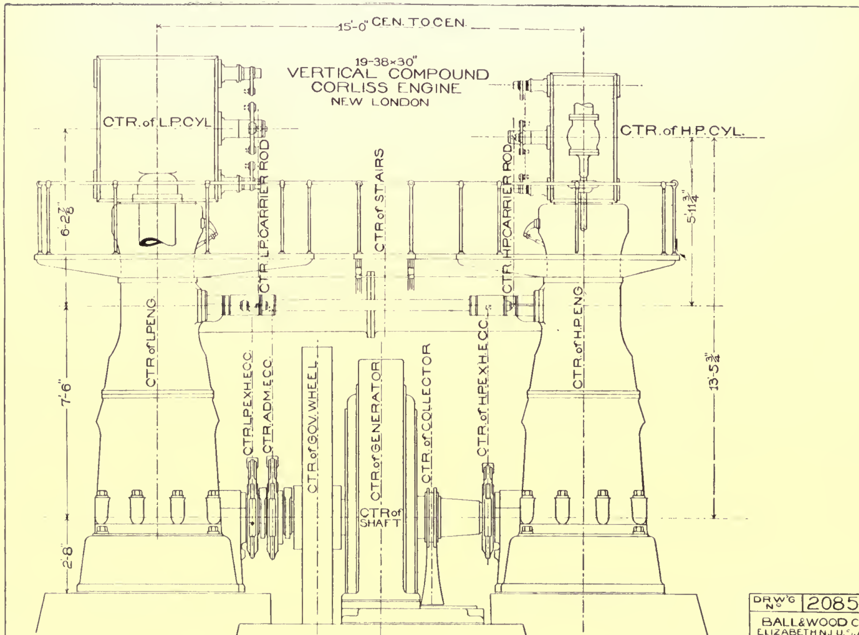
The side load which causes the friction and the consequent development of heat, is due to the pull of the belt in the case of pulleys, the load on the teeth of gears, the pull on cranks and levers, the weight of parts, etc. If we could exert pure torsion on shafts without any side pressure, and counteract all the weight that comes on the shaft, we should not have any trouble with the development of heat in bearings; in fact, there would theoretically be no need of bearings, as the shafts would naturally spin about their axes, and would not need support.

It can be shown, theoretically, that the radiating surface of a bearing increases relatively to the heat generated by a given side load, *only when the length of the bearing is increased*. In other words, increasing the diameter and not the length, theoretically increases the heat generated per unit of time just as much as it increases the radiating surface; hence nothing is gained, and heat accumulates in the bearing as before. This important fact is verified by the design of high-speed bearings, which, it is always noted, are very long in proportion to their diameter, thus giving relatively high radiating power.

Bearings must be rigidly fastened to the body of the machine in some way, and the immediate support is termed a **bracket, frame,** or **housing.** "Bracket" is a very general term, and applies to the supports of other machine parts besides "bearings." It is especially applicable to the more familiar types of bearing supports, and is here introduced to make the analysis complete.

The bracket must be strong enough as a beam to take the side load, the bending moment being figured at such points as are necessary to determine its outline. It may be of solid, box, or ribbed form, the latter being the most economical of material, and usually permitting the simplest pattern. The fastening of the bracket to the main body of the machine must be broad to give stability; the bolts act partly in shear to keep the bracket from sliding along its base, and partly in tension to resist its tendency to rotate about some one of its edges, due to the side pull of the belt, gear tooth, or lever load, as the case may be. The weight of the bracket itself and of the parts it sustains through the bearing, has likewise to be considered; and this acts, in conjunction with the working load on the bearing, to modify the direction and magnitude of the resultant load on the bracket and its fastening.

Stands are forms of brackets, and are subject to the same analysis. The distinction is by no means well defined, although



PLAN SHOWING OVERALL DIMENSIONS OF VERTICAL COMPOUND CORLISS ENGINE DIRECT CONNECTED TO GENERATOR.

we usually think more readily of a stand as having an upright or inverted position with reference to the ground. The ordinary "hanger" is a good example of an inverted stand; and the regular "floor stand," found on jack shafts in some power houses, is an example of the general class.

THEORY. As the method of calculation of the diameter of the shaft, as well as its deflection, has been considered under "Shafts," we may assume that the theoretical study of bearings starts on a given basis of shaft diameter D. The main problem then being one of heat control, let us first calculate the amount of heat developed in a bearing by a given side load. The force of friction acts at the circumference of the shaft, and is equal to the coefficient of friction times the normal force; or, for a given side load P, Fig. 75, the force of friction would be μP. The peripheral speed of the shaft for N revolutions per minute is $\frac{\pi DN}{12}$ feet per minute. As work is "force times distance," the work wasted in friction is then $\frac{\mu P \pi DN}{12}$ foot-pounds per minute. One horse-power being equal to 33,000 foot-pounds per minute, we have the equation,

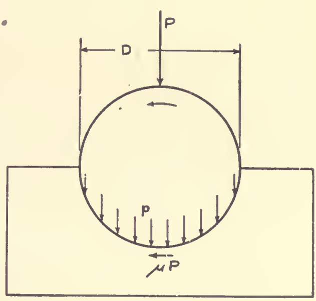


Fig. 75.

$$H = \frac{\mu P \pi DN}{12 \times 33{,}000}. \qquad (113)$$

The value of μ for ordinary, well-lubricated bearings, may run as low as 5 per cent; but as the lubrication is often impaired, it quite commonly rises to 10 or 12 per cent. A value of 8 per cent is a fair average. This amount of horse-power is dissipated through the bearing in the form of heat. If we could exactly determine the ability that each particle of the metal around the shaft had to transmit the heat, or to pass it along to the outside of the casting, and if we could then determine the ability of the particles of air surrounding the casting to receive and carry away this heat, we could calculate just such proportions of the bearing and its casing as would never choke or retard this free transfer of heat away from the running surface.

Such refined theory is not practical, owing to the complicated shapes and conditions surrounding the bearing. The best that we can do is to say that for the usual proportions of bearings the side load may exist up to a certain intensity of "pressure per square inch of projected area" of bearing, or, in form of an equation,

$$P = pLD. \qquad (114)$$

The constant p is of a variable nature, depending on lubrication, speed, air contact, and other special conditions. For ordinary bearings having continuous pressure in one direction, and only fair lubrication, 400 to 500 is an average value. When the pressure changes direction at every half-revolution, the lubricant has a better chance to work fully over the bearing surface, and a higher value is permissible, say, 500 to 800. In locations where mere oscillation takes place, not continuous rotation, and reversal of pressure occurs, as on the cross-head pin of a steam engine, p may run as high as 900 to 1,200. On the crank pins of locomotives, which have the reversal of pressure, and the benefit of high velocity through the air to facilitate cooling, the pressures may run equally high. On the eccentric crank pins of punching and shearing machines, where the pressure acts only for a brief instant and at intervals, the pressure ranges still higher without any dangerous heating action.

When a bearing, for practical reasons, is provided with a cap held in place by bolts or studs, the *theory* of the cap and bolts is of little importance, unless the load comes directly against the cap and bolts. Except in the latter case, the proportions of the cap and the size of the bolts are dependent upon general appearance and utility, it being manifestly desirable to provide a substantial design, even though some excess of strength is thereby introduced.

For the worst case of loading, however, which is when the cap is acted upon by the direct load, such as P in Fig. 76, we have the condition of a centrally loaded beam supported at the bolts. It is probable that the beam is partially fixed at the ends by the clamping of the nut; also that the load P, instead of being concentrated at the center, is to some extent distributed. It is hardly fair to assume the external moment equal to $\frac{Pa}{8}$ or $\frac{Pa}{4}$, the one being too small, perhaps, and the other too large. It will be reasonable to take the external moment at $\frac{Pa}{6}$, in which case, equat-

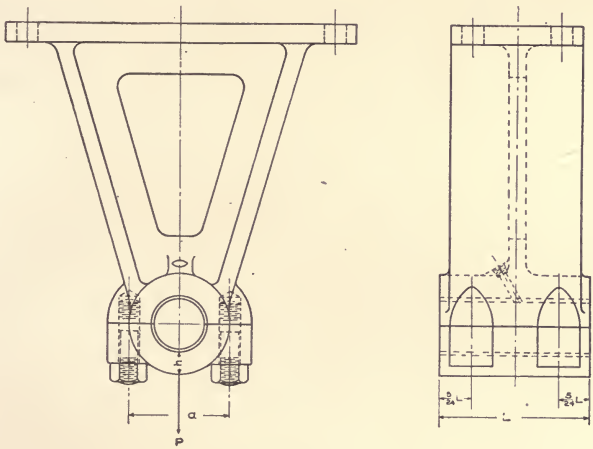


Fig. 76.

ing the external moment to the internal moment of resistance,

$$\frac{Pa}{6} = \frac{SI}{c} = \frac{SLh^2}{6}, \qquad (115)$$

from which, the length of bearing being known, we may calculate the thickness h.

One bolt on each side is sufficient for bearings not more than 6 inches long, but for longer bearings we usually find two bolts on a side. The theoretical location for two bolts on a side, in order that the bearing may be equally strong at the bolts and at the center of the length, may be shown by the principles of mechanics to be $\frac{5}{24}$ L from each end, as indicated in Fig. 76.

The bolts are evidently in direct tension, and if equally loaded would each take their fractional share of the whole load P. This is difficult to guarantee, and it is safer to consider that $\frac{2}{3}$ P may be taken by the bolts on one side. On this basis, for total number of bolts n, equating the external force to the internal resistance of the bolts, we have:

$$\frac{2}{3}P = \frac{S\pi d_1^2}{4} \times \frac{n}{2}, \qquad (116)$$

from which the proper commercial diameter may be readily found.

The bracket may have the shape shown in Fig. 77. The portion at B is under direct shearing stress; and if A be the area at this point, and S_s the safe shearing stress, then, equating the external force to the internal shearing resistance,

$$P = AS_s. \qquad (117)$$

The same shear comes on all parts of the bracket to the left of the load, but there is an excess of shearing strength at these points.

At the point of fastening, the bolts are in shear, due to the same load, for which the equation is

$$P = \frac{\pi d^2}{4} n_1 S_s. \qquad (118)$$

For the upper bolts, the case is that of direct tension, assuming that the whole bracket tends to rotate about the lower edge E. To find the load T on these bolts, we should take moments about the point E, as follows:

$$PL_1 = Tl; \text{ or } T = \frac{PL_1}{l}. \qquad (119)$$

Then, equating the external force to the internal resistance,

$$T = \frac{PL_1}{l} = \frac{\pi d_1^2}{4} \times \frac{n_1}{2} S. \qquad (120)$$

The upper flange is loaded with the bolt load T, and tends to break off at the point of connection to the main body of the bracket, the external moment, therefore, being Tr. The section of the flange is rectangular; hence the equation of external and internal moments is:

$$\mathrm{T}r = \frac{\mathrm{PL}_1}{l} r = \frac{\mathrm{S}bt^2}{6}. \qquad (121)$$

It may be noted that the lower bolts act on such a small leverage about E, that they would stretch and thus permit all the load to be thrown on the upper bolts; this is the reason why they are not subject to calculation for tension.

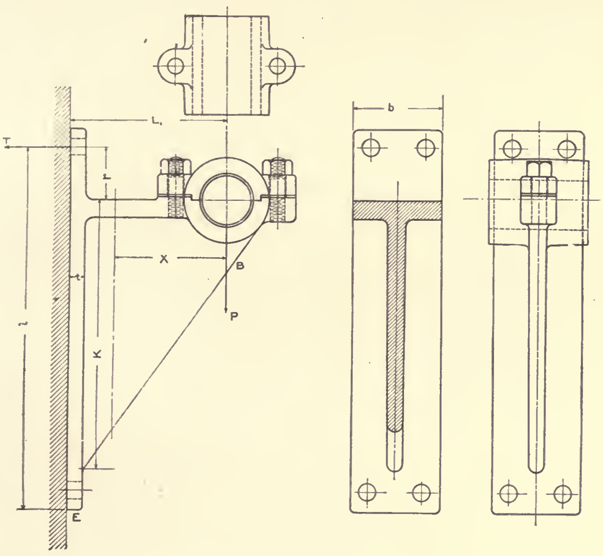


Fig. 77.

The section of the bracket to the left of the load P is dependent upon the bending moment, for, if this section is large enough to take the bending moment properly, the shear may be disregarded. It should be calculated at several points, to make sure that the fiber stress is within allowable limits. The general expression for the equation of moments is, for any section at leverage x,

$$\mathrm{P}x = \frac{\mathrm{SI}}{c}, \qquad (122)$$

from which, by the proper substitution of the moment of inertia of the section, the fiber stress can be calculated. The moment of inertia for simple ribbed sections can be found in most handbooks. The process of solution of the above equation, though simple, is apt to be tedious, and is not considered necessary to illustrate here.

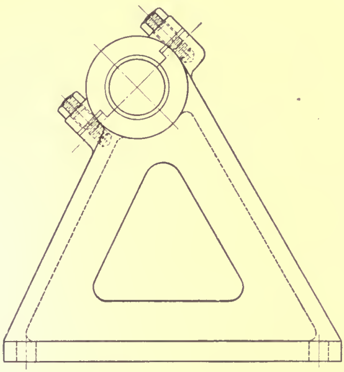


Fig. 78.

PRACTICAL MODIFICATION. Adjustment is an important practical feature of bearings. Unless the proportions are so ample that wear is inappreciable, simple and ready adjustment must be provided. The taper bushing, Fig. 79, is neat and satisfactory for machinery in which expense and refinement are permissible. This is true of some machine tools, but is not true of the general "run" of bearings. The most common form of adjustment is secured by the plain cap (which may or may

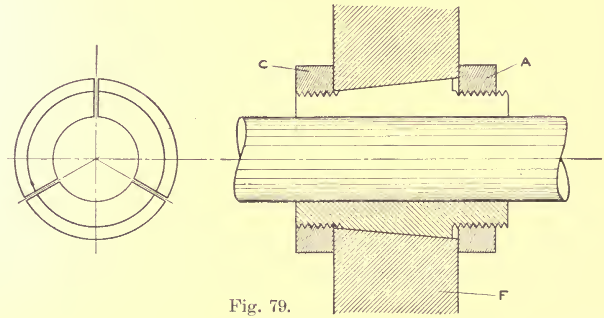


Fig. 79.

not be tongued into the bracket), with liners placed in the joint when new, which may subsequently be removed or reduced so as to allow the cap to close down upon the shaft. Several forms of cap bearings are illustrated in Figs. 80, 81, and 82.

Large engine shaft bearings have special forms of adjustment by means of wedges and screws, which take up the wear in all directions, at the same time accurately preserving the alignment of the shafts; but this refinement is seldom required for shafts of ordinary machinery.

In cases where the cap bearing is not applicable, a simple bushing may be used. This may be removed when worn, and a

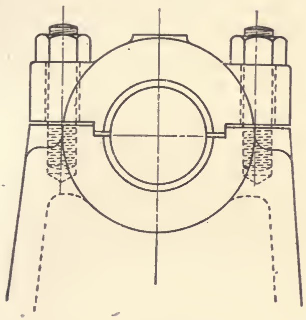


Fig. 80.

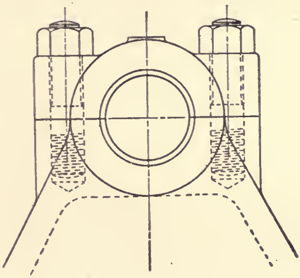


Fig. 81.

new one inserted, the exact alignment being maintained, as the outside will be concentric with the original axis of shaft, regardless of the wear which has taken place in the bore.

The lubrication of bearings is a part of the design, in that the lubricant should be introduced at the proper point, and pains taken to guarantee its distribution to all points of the running surface. The method of lubrication should be so certain that no excuse for its failure would be possible. Grease is a successful lubricator for heavy loads and slow speeds, oil for light loads and high speeds.

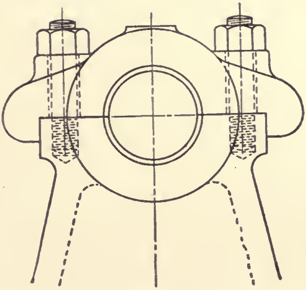


Fig. 82.

In order to insure the lubricant reaching the sliding surfaces and entering between them, it must be introduced at a point where the pressure is moderate, and where the motion of the parts will naturally lead it to all points of the bearing. Grooves and channels of ample size assist in this regard. A special form of bearing uses a ring riding on the shaft to carry the oil constantly from a small reservoir beneath the shaft up to the top, where it is distributed along the bearing and finally flows back to the reservoir and is used again.

The materials of which bearings are made vary with the service required and with the refinement of the bearing. Cast iron makes an excellent bearing for light loads and slow speeds, but it is very apt to "seize" the shaft in case the lubrication is in the least degree impaired. Bronze, in its many forms of density and hardness, is extensively used for high-grade bearings, but it also has little natural lubricating power, and requires careful attention to keep it in good condition.

Babbitt, a composition metal, of varying degrees of hardness, is the most universal and satisfactory material for ordinary bearings. It affords a cheap method of production, being poured in molten form around a mandrel, and firmly retained in its casing or shell through dovetailed pockets into which the metal flows and hardens. It requires no boring or extensive fitting. Some scraping to uniform bearing is necessary in most cases, but this is easily and cheaply done. Babbitt is a durable material, and has some natural lubricating power, so that it has less tendency to heat with scanty lubrication than any of the materials previously mentioned. Almost any grade of bearing may be produced with babbitt. In its finest form the babbitt is hammered, or pened, into the shell of the bearing, and then bored out nearly to size, a slightly tapered mandrel being subsequently drawn through, compressing the babbitt and giving a polished surface.

A combination bearing of babbitt and bronze is sometimes used. In this the bronze lies in strips from end to end of the bearing, and the babbitt fills in between the strips. The shell, being of bronze, gives the required stiffness, and the babbitt the favorable running quality.

PROBLEMS ON BEARINGS, BRACKETS, AND STANDS.

1. The allowable pressure on a bearing is 300 pounds per square inch of projected area. What is the required length of the bearing if the total load is 4,500 pounds and the diameter is 3 inches?

2. The cross-head pin of a steam engine must be 2.5 inches in diameter to withstand the shearing strain. If the maximum pressure is 10,000 pounds, what length should be given to the pin?

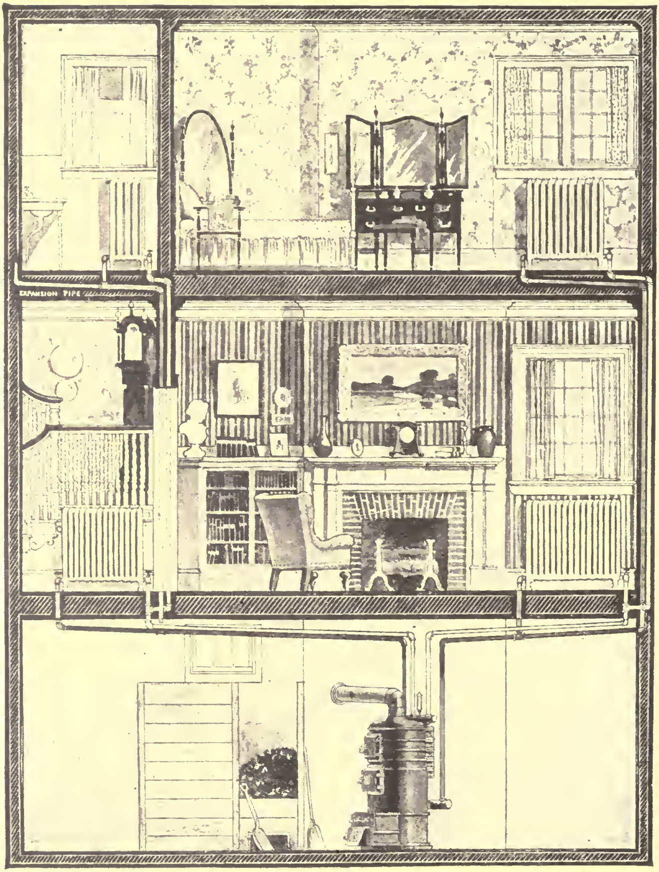
3. The journals on the tender of a locomotive are $3\frac{1}{2} \times 7$ inches. The total weight of the tender and load is 60,000 pounds. If there are 8 journals, what is the pressure per square inch of projected area?

4. What horse-power is lost in friction at the circumference of a 3-inch bearing carrying a load of 6,000 pounds, if the number of revolutions per minute is 150 and the coefficient of friction is assumed to be 5 per cent?

5. The cast-iron bracket in Fig. 77 has a load P of 1,000 pounds. Determine the fiber stress in the web section at the base of the bracket if the thickness is taken at $\frac{1}{2}$ inch, and $L_1 = 12$ inches; $l = 20$ inches; $k = 11$ inches; $t = 1$ inch.

6. Calculate the diameter of the bolts at the top of the bracket.

7. Assuming r equal to 6 inches, what is the fiber stress at the root of flange?



HOT WATER HEATER AND CONNECTIONS.

HEATING AND VENTILATION

PART I

SYSTEMS OF WARMING

Any system of warming must include, *first*, the combustion of fuel, which may take place in a fireplace, stove, or furnace, or a steam, or hot-water boiler; *second*, a system of transmission, by means of which the heat may be carried, with as little loss as possible, to the place where it is to be used for warming; and *third*, a system of diffusion, which will convey the heat to the air in a room, and to its walls, floors, etc., in the most economical way.

Stoves. The simplest and cheapest form of heating is the stove. The heat is diffused by radiation and convection directly to the objects and air in the room, and no special system of transmission is required. The stove is used largely in the country, and is especially adapted to the warming of small dwelling-houses and isolated rooms.

Furnaces. Next in cost of installation and in simplicity of operation, is the hot-air furnace. In this method, the air is drawn over heated surfaces and then transmitted through pipes, while at a high temperature, to the rooms where heat is required. Furnaces are used largely for warming dwelling-houses, also churches, halls, and schoolhouses of small size. They are more costly than stoves, but have certain advantages over that form of heating. They require less care, as several rooms may be warmed from a single furnace; and, being placed in the basement, more space is available in the rooms above, and the dirt and litter connected with the care of a stove are largely done away with. They require less care, as only one fire is necessary to warm all the rooms in a house of ordinary size. One great advantage in the furnace method of warming comes from the constant supply of fresh air which is required to bring the heat into the rooms. While this is greatly to be desired from a sanitary standpoint, it calls for the consumption of a larger amount of fuel than would otherwise be necessary. This is true because heat is required to warm the fresh air from out of doors up to the temperature of the rooms, in addition to replacing the heat lost by leakage and conduction through walls and windows.

A more even temperature may be maintained with a furnace than by the use of stoves, owing to the greater depth and size of the fire, which allows it to be more easily controlled.

When a building is placed in an exposed location, there is often difficulty in warming rooms on the north and west sides, or on that side toward the prevailing winds. This may be overcome to some extent by a proper location of the furnace and by the use of extra large pipes for conveying the hot air to those rooms requiring special attention.

Direct Steam. Direct steam, so called, is widely used in all classes of buildings, both by itself and in combination with other systems. The first cost of installation is greater than for a furnace; but the amount of fuel required is less, as no outside air supply is necessary. If used for warming hospitals, schoolhouses, or other buildings where a generous supply of fresh air is desired, this method must be supplemented by some form of ventilating system.

One of the principal advantages of direct steam is the ability to heat all rooms alike, regardless of their location or of the action of winds.

When compared with hot-water heating, it has still another desirable feature—which is its freedom from damage by the freezing of water in the radiators when closed, which is likely to happen in unused rooms during very cold weather in the case of the former system.

On the other hand, the sizes of the radiators must be proportioned for warming the rooms in the coldest weather, and unfortunately there is no satisfactory method of regulating the amount of heat in mild weather, except by shutting off or turning on steam in the radiators at more or less frequent intervals as may be required, unless one of the expensive systems of automatic control is employed. In large rooms, a certain amount of regulation can be secured by dividing the radiation into two or more parts, so that different combinations may be used under varying conditions of outside temperature. If two radiators are used, their surface should be proportioned, when convenient, in the ratio of 1 to 2, in which case one-third, two-thirds, or the whole power of the radiation can be used as desired.

Indirect Steam. This system of heating combines some of the advantages of both the furnace and direct steam, but is more costly to install than either of these. The amount of fuel required is about the same as for furnace heating, because in each case the cool fresh air must be warmed up to the temperature of the room, before it can become a medium for conveying heat to offset that lost by leakage and conduction through walls and windows.

A system for indirect steam may be so designed that it will supply a greater quantity of fresh air than the ordinary form of furnace, in which case the cost of fuel will of course be increased in proportion to the volume of air supplied. Instead of placing the radiators in the rooms, a special form of heater is supported near the basement ceiling and encased in either galvanized iron or brick. A cold-air supply duct is connected with the space below the heater, and warm air pipes are taken from the top and connected with registers in the rooms to be heated the same as in the case of furnace heating.

A separate stack or heater may be provided for each register if the rooms are large; but, if small and so located that they may be reached by short runs of horizontal pipe, a single heater may serve for two or more rooms.

The advantage of indirect steam over furnace heating comes from the fact that the stacks may be placed at or near the bases of the flues leading to the different rooms, thus doing away with long, horizontal runs of pipe, and counteracting to a considerable extent the effect of wind pressure upon exposed rooms. Indirect and direct heating are often combined to advantage by using the former for the more important rooms, where ventilation is desired, and the latter for rooms more remote or where heat only is required.

Another advantage is the large ratio between the radiating surface and grate-area, as compared with a furnace; this results in a large volume of air being warmed to a moderate temperature instead of a smaller quantity being heated to a much higher temperature, thus giving a more agreeable quality to the air and rendering it less dry.

Indirect steam is adapted to all the buildings mentioned in connection with furnace heating, and may be used to much better advantage in those of large size. This applies especially to cases where more than one furnace is necessary; for, with steam heat, a single boiler, or a battery of boilers, may be made to supply heat for a building of any size, or for a group of several buildings, if desired, and is much easier to care for than several furnaces widely scattered.

Direct-Indirect Radiators. These radiators are placed in the room the same as the ordinary direct type. The construction is such that when the sections are in place, small flues are formed between them; and air, being admitted through an opening in the outside wall, passes upward through them and becomes heated before entering the room. A switch damper is placed in the casing at the base of the radiator, so that air may be taken from the room itself instead of from out of doors, if so desired. Radiators of this kind are not used to any great extent, as there is likely to be more or less leakage of cold air into the room around the base. If ventilation is required, it is better to use the regular form of indirect heater with flue and register, if possible. It is sometimes desirable to partially ventilate an isolated room where it would be impossible to run a flue, and in cases of this kind the direct-indirect form is often useful.

Direct Hot Water. Hot water is especially adapted to the warming of dwellings and greenhouses, owing to the ease with which the temperature can be regulated. When steam is used, the radiators are always at practically the same temperature, while with hot water the temperature can be varied at will. A system for hot-water heating costs more to install than one for steam, as the radiators must be larger and the pipes more carefully run. On the other hand, the cost of operating is somewhat less, because the water need be carried only at a temperature sufficiently high to warm the rooms properly in mild weather, while with steam the building is likely to become overheated, and more or less heat wasted through open doors and windows.

A comparison of the relative costs of installing and operating hot-air, steam, and hot-water systems, is given in Table I.

TABLE I

Relative Cost of Heating Systems

	Hot Air	Steam	Hot Water
Relative cost of apparatus	9	13	15
Relative cost, adding repairs and fuel for five years	29½	29¾	27
Relative cost, adding repairs and fuel for fifteen years	81	63	52½

One disadvantage in the use of hot water is the danger from freezing when radiators are shut off in unused rooms. This makes it necessary in very cold weather to have all parts of the system turned on sufficiently to produce a circulation, even if very slow. This is sometimes accomplished by drilling a very small hole (about $\frac{1}{8}$ inch) in the valve-seat, to that when closed there will still be a very slow circulation through the radiator, thus preventing the temperature of the water from reaching the freezing point.

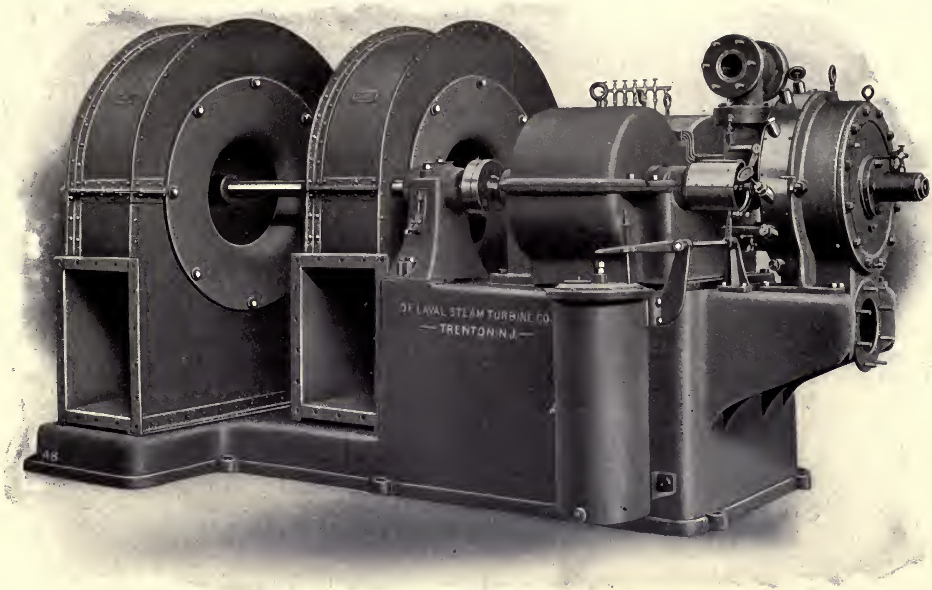
Indirect Hot Water. This is used under the same conditions as indirect steam, but more especially in the case of dwellings and hospitals. When applied to other and larger buildings, it is customary to force the water through the mains by means of a pump. Larger heating stacks and supply pipes are required than for steam; but the arrangement and size of air-flues and registers are practically the same, although they are sometimes made slightly larger in special cases.

Exhaust Steam. Exhaust steam is used for heating in connection with power plants, as in shops and factories, or in office buildings which have their own lighting plants. There are two methods of using exhaust steam for heating purposes. One is to carry a back pressure of 2 to 5 pounds on the engines, depending upon the length and size of the pipe mains; and the other is to use some form of *vacuum system* attached to the returns or air-valves, which tends to reduce the back pressure rather than to increase it.

Where the first method is used and a back pressure carried, either the boiler pressure or the cut-off of the engines must be increased, to keep the mean effective pressure the same and not reduce the horse-power delivered. In general it is more economical to utilize the exhaust steam for heating. There are instances, however, where the relation between the quantities of steam required for heating and for power are such—especially if the engines are run condensing—that it is better to throw the exhaust away and heat with live steam. Where the vacuum method is used, these difficulties are avoided; and for this reason that method is coming into quite common use. If the condensation from the exhaust steam is returned to the boilers, the oil must first be removed; this is usually accomplished by passing the steam through some form of grease extractor as it leaves the engine. The water of condensation is often passed through a separating tank in addition to this, before it is delivered to the return pumps. It is better, however, to remove a portion of the oil before the steam enters the heating system; otherwise a coating will be formed upon the inner surfaces of the radiators, which will reduce their efficiency to some extent.

Forced Blast. This method of heating, in different forms, is used for the warming of factories, schools, churches, theaters, halls—in fact, any large building where good ventilation is desired. The air for warming is drawn or forced through a heater of special design, and discharged by a fan or blower into ducts which lead to registers placed in the rooms to be warmed. The heater is usually made up in sections, so that steam may be admitted to or shut off from any section independently of the others, and the temperature of the air regulated in this manner. Sometimes a *by-pass damper* is attached, so that part of the air will pass through the heater and part around or over it; in this way the proportions of cold and heated air may be so adjusted as to give the desired temperature to the air entering the rooms. These forms of regulation are common where a blower is used for warming a single room, as in the case of a church or hall; but where several rooms are warmed, as in a schoolhouse. it is customary to use the main or primary heater at the blower for warming the air to a given temperature (somewhat below that which is actually required), and to supplement this by placing secondary coils or heaters at the bottoms of the flues leading to the different rooms. By means of this arrangement, the temperature of each room can be regulated independently of the others. The so-called *double-duct* system is sometimes employed. In this case, two ducts are carried to each register, one supplying hot air and the other cold or tempered air; and a damper for mixing these in the right proportions is placed in the flue, below the register.

Electric Heating. Unless electricity can be produced at a very low cost, it is not practicable for heating residences or large buildings. The electric heater, however, has quite a wide field of application in heating small offices, bathrooms, electric cars, etc. It is a convenient method of warming isolated rooms on cold mornings, in late spring and early fall, when the regular heating apparatus of the building is not in operation. It has the advantage of being instantly available, and the amount of heat can be regulated at will. Electric heaters are clean, do not vitiate the air, and are easily moved from place to place.



150 H.P. DE LAVAL TURBINE BLOWER.

PRINCIPLES OF VENTILATION

Closely connected with the subject of heating is the problem of maintaining air of a certain standard of purity in the various buildings occupied.

The introduction of pure air can be done properly only in connection with some system of heating; and no system of heating is complete without a supply of pure air, depending in amount upon the kind of building and the purpose for which it is used.

Composition of the Atmosphere. Atmospheric air is not a simple substance but a mechanical mixture. Oxygen and nitrogen, the principal constituents, are present in very nearly the proportion of one part of oxygen to four parts of nitrogen by weight. Carbonic acid gas, the product of all combustion, exists in the proportion of 3 to 5 parts in 10,000 in the open country. Water in the form of vapor, varies greatly with the temperature and with the exposure of the air to open bodies of water. In addition to the above, there are generally present, in variable but exceedingly small quantities, ammonia, sulphuretted hydrogen, sulphuric, sulphurous, nitric, and nitrous acids, floating organic and inorganic matter, and local impurities. Air also contains ozone, which is a peculiarly active form of oxygen; and lately another constituent called *argon* has been discovered.

Oxygen is the most important element of the air, so far as both heating and ventilation are concerned. It is the active element in the chemical process of combustion and also in the somewhat similar process which takes place in the respiration of human beings. Taken into the lungs, it acts upon the excess of carbon in the blood, and possibly upon other ingredients, forming chemical compounds which are thrown off in the act of respiration or breathing.

Nitrogen. The principal bulk of the atmosphere is nitrogen, which exists uniformly diffused with oxygen and carbonic acid gas. This element is practically inert in all processes of combustion or respiration. It is not affected in composition, either by passing through a furnace during combustion or through the lungs in the process of respiration. Its action is to render the oxygen less active, and to absorb some part of the heat produced by the process of oxidation.

Carbonic acid gas is of itself only a neutral constituent of the atmosphere, like nitrogen; and—contrary to the general impression—its presence in moderately large quantities (if uncombined with other substances) is neither disagreeable nor especially harmful. Its presence, however, in air provided for respiration, decreases the readiness with which the carbon of the blood unites with the oxygen of the air; and therefore, when present in sufficient quantity, it may cause indirectly, not only serious, but fatal results. The real harm of a vitiated atmosphere, however, is caused by the other constituent gases and by the minute organisms which are produced in the process of respiration. It is known that these other impurities exist in fixed proportion to the amount of carbonic acid present in an atmosphere vitiated by respiration. Therefore, as the relative proportion of carbonic acid can easily be determined by experiment, the fixing of a standard limit of the amount in which it may be allowed, also limits the amounts of other impurities which are found in combination with it.

When carbonic acid is present in excess of 10 parts in 10,000 parts of air, a feeling of weariness and stuffiness, generally accompanied by a headache, will be experienced; while with even 8 parts in 10,000 parts a room would be considered close. For general considerations of ventilation, the limit should be placed at 6 to 7 parts in 10,000, thus allowing an increase of 2 to 3 parts over that present in outdoor air, which may be considered to contain four parts in 10,000 under ordinary conditions.

Analysis of Air. An accurate qualitative and quantitative analysis of air samples can be made only by an experienced chemist. There are, however, several approximate methods for determining the amount of carbonic acid present, which are sufficiently exact for practical purposes. Among these the following is one of the simplest:

The necessary apparatus consists of six clean, dry, and tightly corked bottles, containing respectively 100, 200, 250, 300, 350, and 400 cubic centimeters, a glass tube containing exactly 15 cubic centimeters to a given mark, and a bottle of perfectly clear, fresh limewater. The bottles should be filled with the air to be examined by means of a hand-ball syringe. Add to the smallest bottle 15 cubic centimeters of the limewater, put in the cork, and shake well. If the limewater has a milky appearance, the amount of carbonic acid will be at least 16 parts in 10,000. If the contents of the bottle remain clear, treat the bottle of 200 cubic centimeters in the same manner; a milky appearance or turbidity in this would indicate 12 parts in 10,000. In a similar manner, turbidity in the 250 cubic centimeter bottle indicates

10 parts in 10,000; in the 300, 8 parts; in the 350, 7 parts; and in the 400, less than 6 parts. The ability to conduct more accurate analyses can be attained only by special study and a knowledge of chemical properties and of methods of investigation.

Another method similar to the above, makes use of a glass cylinder containing a given quantity of limewater and provided with a piston. A sample of the air to be tested is drawn into the cylinder by an upward movement of the piston. The cylinder is then thoroughly shaken, and if the limewater shows a milky appearance, it indicates a certain proportion of carbonic acid in the air. If the limewater remains clear, the air is forced out, and another cylinder full drawn in, the operation being repeated until the limewater becomes milky. The size of the cylinder and the quantity of limewater are so proportioned that a change in color at the first, second, third, etc., cylinder full of air indicates different proportions of carbonic acid. This test is really the same in principle as the one previously described; but the apparatus used is in more convenient form.

Air Required for Ventilation. The amount of air required to maintain any given standard of purity can very easily be determined, provided we know the amount of carbonic acid given off in the process of respiration. It has been found by experiment that the average production of carbonic acid by an adult at rest is about .6 cubic foot per hour. If we assume the proportion of this gas as 4 parts in 10,000 in the external air, and are to allow 6 parts in 10,000 in an occupied room, the gain will be 2 parts in 10,000; or, in other words, there will be $\frac{2}{10,000} = .0002$ cubic foot of carbonic acid mixed with each cubic foot of fresh air entering the room. Therefore, if one person gives off .6 cubic foot of carbonic acid per hour, it will require $.6 \div .0002 = 3,000$ cubic feet of air per hour per person to keep the air in the room at the standard of purity assumed—that is, 6 parts of carbonic acid in 10,000 of air.

Table II has been computed in this manner, and shows the amount of air which must be introduced for each person in order to maintain various standards of purity.

While this table gives the theoretical quantities of air required for different standards of purity, and may be used as a guide, it will be better in actual practice to use quantities which experience has shown to give good results in different types of buildings. In auditoriums where the cubic space per individual is large, and in which the atmosphere is thoroughly fresh before the rooms are occupied, and the occupancy is of only two or three hours' duration, the air-supply may be reduced somewhat from the figures given below.

TABLE II

Quantity of Air Required per Person

Standard Parts of Carbonic Acid in 10,000 of Air in Room	Cubic Feet of Air Required per Person	
	Per Minute	Per Hour
5	100	6,000
6	50	3,000
7	33	2,000
8	25	1,500
9	20	1,200
10	16	1,000

Table III represents good modern practice and may be used with satisfactory results:

TABLE III

Air Required for Ventilation of Various Classes of Buildings

Air-Supply per Occupant for	Cubic Feet per Minute	Cubic Feet per Hour
Hospitals	80 to 100	4, 800 to 6, 000
High Schools	50	3, 000
Grammar Schools	40	2, 400
Theaters and Assembly Halls	25	1, 500
Churches	20	1, 200

When possible, the air-supply to any given room should be based upon the number of occupants. It sometimes happens, however, that this information is not available, or the character of the room is such that the number of persons occupying it may vary, as in the case of public waiting rooms, toilet rooms, etc. In instances of this kind, the required air-volume may be based upon the number of changes per hour. In using this method, various considerations must be taken into account, such as the use of the room and its condition as to crowding, character of occupants, etc. In general, the following will be found satisfactory for average conditions:

TABLE IV

Number of Changes of Air Required in Various Rooms

Use of Room	Changes of Air per Hour
Public Waiting Room	4 to 5
Public Toilets	5 " 6
Coat and Locker Rooms	4 " 5
Museums	3 " 4
Offices, Public	4 " 5
Offices, Private	3 " 4
Public Dining Rooms	4 " 5
Living Rooms	3 " 4
Libraries, Public	4 " 5
Libraries, Private	3 " 4

Force for Moving Air. Air is moved for ventilating purposes in two ways: (1) by expansion due to heating; (2) by mechanical means. The effect of heat on the air is to increase its volume and therefore lessen its density or weight, so that it tends to rise and is replaced by the colder air below. The available force for moving air obtained in this way is very small, and is quite likely to be overcome by wind or external causes. It will be found in general that the heat used for producing velocity in this manner, when transformed into work in the steam engine, is greatly in excess of that required to produce the same effect by the use of a fan.

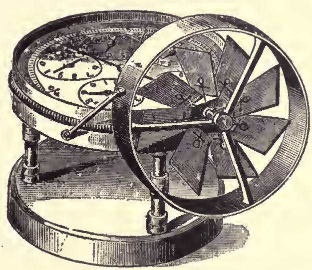


Fig. 1. Common Form of Anemometer, for Measuring Velocity of Air-Currents.

Ventilation by mechanical means is performed either by pressure or by suction. The former is used for delivering fresh air into a building, and the latter for removing the foul air from it. By both processes the air is moved without change in temperature, and the force for moving must be sufficient to overcome the effects of wind or changes in outside temperature. Some form of fan is used for this purpose.

Measurements of Velocity. The velocity of air in ventilating ducts and flues is measured directly by an instrument called an anemometer. A common form of this instrument is shown in Fig. 1. It consists of a series of flat vanes attached to an axis, and a series of dials.

The revolution of the axis causes motion of the hands in proportion to the velocity of the air, and the result can be read directly from the dials for any given period.

For approximate results the anemometer may be slowly moved across the opening in either vertical or horizontal parallel lines, so that the readings will be made up of velocities taken from all parts of the opening. For more accurate work, the opening should be divided into a number of squares by means of small twine, and readings taken at the center of each. The mean of these readings will give the average velocity of the air through the entire opening.

AIR DISTRIBUTION

The location of the air inlet to a room depends upon the size of the room and the purpose for which it is used. In the case of living rooms in dwelling-houses, the registers are placed either in the floor or in the wall near the floor; this brings the warm air in at the coldest part of the room and gives an opportunity for warming or drying the feet if desired. In the case of schoolrooms, where large volumes of warm air at moderate temperatures are required, it is best to discharge it through openings in the wall at a height of 7 or 8 feet from the floor; this gives a more even distribution, as the warmer air tends to rise and hence spreads uniformly under the ceiling; it then gradually displaces other air, and the room becomes filled with pure air without sensible currents or drafts. The cooler air sinks to the bottom of the room, and can be taken off through ventilating registers placed near the floor. The relative positions of the inlet and outlet are often governed to some extent by the building construction; but, if possible, they should both be located in the same side of the room. Figs. 2, 3, and 4 show common arrangements.

The vent outlet should always, if possible, be placed in an inside wall; otherwise it will become chilled and the air-flow through it will become sluggish. In theaters and churches which are closely packed, the air should enter at or near the floor, in finely-divided streams; and the discharge ventilation should be through openings in the ceiling. The reason for this is the large amount of animal heat given off from the bodies of the audience; this causes the air to become still further heated after entering the room, and the tendency is to rise continuously from floor to ceiling, thus carrying away all impurities from respiration as fast as they are given off.

All audience halls in which the occupants are closely seated should be treated in the same manner, when possible. This, however, cannot always be done, as the seats are often made removable so that the

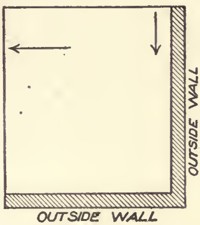


Fig. 2.

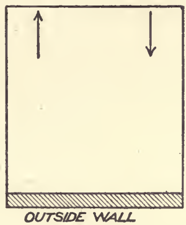


Fig. 3.

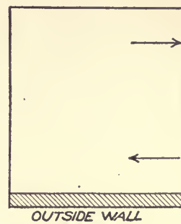


Fig. 4.

Diagrams Showing Relative Positions of Air Inlets and Outlets as Commonly Arranged.

floor can be used for other purposes. In cases of this kind, part of the air may be introduced through floor registers placed along the outer aisles, and the remainder by means of wall inlets the same as for schoolrooms. The discharge ventilation should be partly through registers near the floor, supplemented by ample ceiling vents for use when the hall is crowded or the outside temperature high.

The matter of air-velocities, size of flues, etc., will be taken up under the head of "Indirect Heating."

HEAT LOSS FROM BUILDINGS

A *British Thermal Unit*, or B. T. U., has been defined as the amount of heat required to raise the temperature of one pound of water one degree F. This measure of heat enters into many of the calculations involved in the solving of problems in heating and ventilation, and one should familiarize himself with the exact meaning of the term.

Causes of Heat Loss. The heat loss from a building is due to the following causes: (1) radiation and conduction of heat through walls and windows; (2) leakage of warm air around doors and windows and through the walls themselves; and (3) heat required to warm the air for ventilation.

Loss through Walls and Windows. The loss of heat through the walls of a building depends upon the material used in construction

TABLE V

Heat Losses in B. T. U. per Square Foot of Surface per Hour—Southern Exposure

Material	Difference between Inside and Outside Temperatures									
	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
8-in. Brick Wall	5	9	13	18	22	27	31	36	40	45
12-in. Brick Wall	4	7	10	13	16	20	23	26	30	33
16-in. Brick Wall	3	5	8	10	13	16	19	22	24	27
20-in. Brick Wall	2.8	4.5	7	9	11	14	16	18	20	23
24-in. Brick Wall	2.5	4	6	8	10	12	14	16	18	20
28-in. Brick Wall	2	3.5	5	7	9	11	13	14	16	18
32-in. Brick Wall	1.5	3	4.5	6	8	10	11	13	15	16
Single Window	12	24	36	49	60	73	85	93	110	122
Double Window	8	16	24	32	40	48	56	62	70	78
Single Skylight	11	21	31	42	52	63	73	84	94	104
Double Skylight	7	14	20	28	35	42	48	56	62	70
1-in. Wooden Door	4	8	12	16	20	24	28	32	36	40
2-in. Wooden Door	3	5	8	11	14	17	20	23	25	28
2-in. Solid Plaster Partition	6	12	18	24	30	36	42	48	54	60
3-in. Solid Plaster Partition	5	10	15	20	25	30	35	40	45	50
Concrete Floor on Brick Arch	2	4	6.5	9	11	13	15	18	20	22
Wood Floor on Brick Arch	1.5	3	4.5	6	7	9	10	12	13	15
Double Wood Floor	1	2	3	4	5	6	7	8	9	10
Walls of Ordinary Wooden Dwellings	3	5	8	10	13	16	19	22	24	27

For *solid stone walls,* multiply the figures for brick of the same thickness by 1.7. Where rooms have a *cold attic above or cellar beneath,* multiply the heat loss through walls and windows by 1.1.

Correction for Leakage. The figures given in the above table apply only to the most thorough construction. For the average well-built house, the results should be increased about 10 per cent; for fairly good construction, 20 per cent; and for poor construction, 30 per cent.

Table V applies only to a southern exposure; for other exposures multiply the heat loss given in Table V by the factors given in Table VI.

of the wall, the thickness, the number of layers, and the difference between the inside and outside temperatures. The exact amount of heat lost in this way is very difficult to determine theoretically, hence we depend principally on the results of experiments.

Loss by Air-Leakage. The leakage of air from a room varies from one to two or more changes of the entire contents per hour, depending upon the construction, opening of doors, etc. It is common practice to allow for one change per hour in well-constructed buildings where two walls of the room have an outside exposure. As the amount of leakage depends upon the extent of exposed wall and window surface, the simplest way of providing for this is to increase

TABLE VI
Factors for Calculating Heat Loss for Other than Southern Exposures

Exposure	Factor
N.	1.32
E.	1.12
S.	1.0
W.	1.20
N. E.	1.22
N. W.	1.26
S. E.	1.06
S. W.	1.10
N., E., S., and W., or total exposure	1.16

the total loss through walls and windows by a factor depending upon the tightness of the building construction. Authorities differ considerably in the factors given for heat losses, and there are various methods for computing the same. The figures given in Table V have been used extensively in actual practice, and have been found to give good results when used with judgment. The table gives the heat losses through different thicknesses of walls, doors, windows, etc., in B. T. U., per square foot of surface per hour, for varying differences in inside and outside temperatures.

In computing the heat loss through walls, only those exposed to the outside air are considered.

In order to make the use of the table clear, we shall give a number of examples illustrating its use:

Example 1. Assuming an inside temperature of 70°, what will be the heat loss from a room having an exposed wall surface of 200 square feet and a glass surface of 50 square feet, when the outside temperature is zero? The wall is of brick, 16 inches in thickness, and has a southern exposure; the windows are single; and the construction is of the best, so that no account need be taken of leakage

We find from Table V, that the factor for a 16-inch brick wall with a difference in temperature of 70° is 19, and that for glass (single window) under the same condition is 85; therefore,

Loss through walls = 200 × 19 = 3,800
Loss through windows = 50 × 85 = 4,250

Total loss per hour = 8,050 B. T. U.

Example 2. A room 15 ft. square and 10 ft. high has two exposed walls, one toward the north, and the other toward the west. There are 4 windows, each 3 feet by 6 feet in size. The two in the north wall are double, while the other two are single. The walls are of brick, 20 inches in thickness. With an inside temperature of 70°, what will be the heat loss per hour when it is 10° below zero?

Total exposed surface	= 15 × 10 × 2 =	300
Glass surface	= 3 × 6 × 4 =	72
Net wall surface	=	228

Difference between inside and outside temperature 80°.
Factor for 20-inch brick wall is 18.
Factor for single window is 93.
Factor for double window is 62.
The heat losses are as follows:

Wall,	228 × 18 =	4,104
Single windows,	36 × 93 =	3,348
Double windows,	36 × 62 =	2,232
		9,684 B. T. U.

As one side is toward the north, and the other toward the west, the actual exposure is N. W. Looking in Table VI, we find the correction factor for this exposure to be 1.26; therefore the total heat loss is

9,684 × 1.26 = 12,201.84 B. T. U.

Example 3. A dwelling-house of fair wooden construction measures 160 ft. around the outside; it has 2 stories, each 8 ft. in height; the windows are single, and the glass surface amounts to one-fifth the total exposure; the attic and cellar are unwarmed. If 8,000 B. T. U. are utilized from each pound of coal burned in the furnace, how many pounds will be required per hour to maintain a temperature of 70° when it is 20° above zero outside?

Total exposure	= 160 × 16 =	2,560
Glass surface	= 2,560 ÷ 5 =	512
Net wall		= 2,048

Temperature difference = 70 − 20 = 50°

Wall	2,048 × 13	= 26,624
Glass	512 × 60	= 30,720
		57,344 B. T. U.

As the building is exposed on all sides, the factor for exposure will be the average of those for N., E., S., and W., or

(1.32 + 1.12 + 1.0 + 1.20) ÷ 4 = 1 16

The house has a cold cellar and attic, so we must increase the heat loss

10 per cent for each of the first two conditions, and 20 per cent for the last. Making these corrections we have:

$$57{,}344 \times 1.16 \times 1.10 \times 1.10 \times 1.20 = 96{,}338 \text{ B. T. U.}$$

If one pound of coal furnishes 8,000 B. T. U., then $96{,}338 \div 8{,}000 =$ 12 pounds of coal per hour required to warm the building to 70° under the conditions stated.

Approximate Method. For dwelling-houses of the average construction, the following simple method for calculating the heat loss may be used. Multiply the total exposed surface by 45, which will give the heat loss in B. T. U. per hour for an inside temperature of 70° in zero weather.

This factor is obtained in the following manner: Assume the glass surface to be one-sixth the total exposure, which is an average proportion. Then each square foot of exposed surface consists one-sixth of glass and five-sixths of wall, and the heat loss for 70° difference in temperature would be as follows:

$$\text{Wall } \frac{5}{6} \times 19 = 15.8$$

$$\text{Glass } \frac{1}{6} \times 85 = \underline{14.1}$$

$$29.9$$

Increasing this 20 per cent for leakage, 16 per cent for exposure, and 10 per cent for cold ceilings, we have:

$$29.9 \times 1.20 \times 1.16 \times 1.10 = 45.$$

The loss through floors is considered as being offset by including the kitchen walls of a dwelling-house, which are warmed by the range, and which would not otherwise be included if computing the size of a furnace or boiler for heating.

If the heat loss is required for outside temperatures other than zero, multiply by 50 for 10 degrees below, and by 40 for 10 degrees above zero.

This method is convenient for approximations in the case of dwelling-houses; but the more exact method should be used for other types of buildings, and in all cases for computing the heating surface for separate rooms. *When calculating the heat loss from isolated rooms, the cold inside walls as well as the outside must be considered.*

The loss through a wall next to a cold attic or other unwarmed space may in general be taken as about two-thirds that of an outside wall.

Heat Loss by Ventilation. One B. T. U. will raise the temperature of 1 cubic foot of air 55 degrees at average temperatures and pressures, or will raise 55 cubic feet 1 degree, so that the heat required for the ventilation of any room can be found by the following formula:

$$\frac{\text{Cu. ft. of air per hour} \times \text{Number of degrees rise}}{55} = \text{B. T. U. required.}$$

To compute the heat loss for any given room which is to be ventilated, first find the loss through walls and windows, and correct for exposure and leakage; then compute the amount required for ventilation as above, and take the sum of the two. An inside temperature of 70° is always assumed unless otherwise stated.

Examples. What quantity of heat will be required to warm 100,000 cubic feet of air to 70° for ventilating purposes when the outside temperature is 10 below zero?

$$100{,}000 \times 80 \div 55 = 145{,}454 \text{ B. T. U.}$$

How many B. T. U. will be required per hour for the ventilation of a church seating 500 people, in zero weather?

Referring to Table III, we find that the total air required per hour is $1{,}200 \times 500 = 600{,}000$ cu. ft.; therefore $600{,}000 \times 70 \div 55 = 763{,}636$ B. T. U.

The factor $\frac{\text{Rise in Temperature}}{55}$ is approximately 1.1 for 60°, 1.3 for 70°, and 1.5 for 80°. Assuming a temperature of 70° for the entering air, we may multiply the air-volume supplied for ventilation by 1.1 for an outside temperature of 10° above 0, by 1.3 for zero, and by 1.5 for 10° below zero—which covers the conditions most commonly met with in practice.

EXAMPLES FOR PRACTICE

1. A room in a grammar school 28 ft. by 32 ft. and 12 feet high is to accommodate 50 pupils. The walls are of brick 16 inches in thickness; and there are 6 single windows in the room, each 3 ft. by 6 ft.; there are warm rooms above and below; the exposure is S. E. How many B. T. U. will be required per hour for warming the room, and how many for ventilation, in zero weather, assuming the building to be of average construction?

Ans. 24,261 + for warming; 152,727 + for ventilation.

2. A stone church seating 400 people has walls 20 inches in thickness. It has a wall exposure of 5,000 square feet, a glass expos-

ure (single windows) of 600 square feet, and a roof exposure of 7,000 square feet; the roof is of 2-inch pine plank, and the factor for heat loss may be taken the same as for a 2-inch wooden door. The floor is of wood on brick arches, and has an area of 4,000 square feet. The building is exposed on all sides, and is of first-class construction. What will be the heat required per hour for both warming and ventilation when the outside temperature is 20° above zero?

ANS. 296,380 for warming; 436,363 + for ventilation.

3. A dwelling-house of average wooden construction measures 200 feet around the outside, and has 3 stories, each 9 feet high. Compute the heat loss by the approximate method when the temperature is 10° below zero.

ANS. 270,000 B. T. U. per hour.

FURNACE HEATING

In construction, a furnace is a large stove with a combustion chamber of ample size over the fire, the whole being inclosed in a casing of sheet iron or brick. The bottom of the casing is provided with a cold-air inlet, and at the top are pipes which connect with registers placed in the various rooms to be heated. Cold, fresh air is brought from out of doors through a pipe or duct called the *cold-air box;* this air enters the space between the casing and the furnace near the bottom, and, in passing over the hot surfaces of the fire-pot and combustion chamber, becomes heated. It then rises through the warm-air pipes at the top of the casing, and is discharged through the registers into the rooms above.

As the warm air is taken from the top of the furnace, cold air flows in through the cold-air box to take its place. The air for heating the rooms does not enter the combustion chamber.

Fig. 5 shows the general arrangement of a furnace with its connecting pipes. The cold-air inlet is seen at the bottom, and the hot-air pipes at the top; these are all provided with dampers for shutting off or regulating the amount of air flowing through them. The feed or fire door is shown at the front, and the ash door beneath it; a *water-pan* is placed inside the casing, and furnishes moisture to the warm air before passing into the rooms; water is either poured into the pan through an opening in the front, provided for this purpose, or is supplied automatically through a pipe.

The fire is regulated by means of a draft slide in the ash door, and a cold-air or regulating damper placed in the smoke-pipe. Clean-out doors are placed at different points in the casing for the removal of

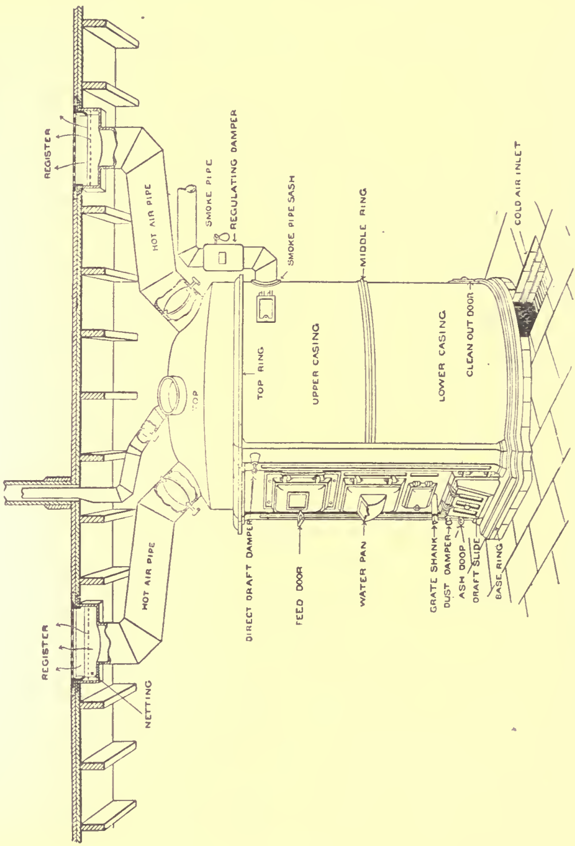


Fig. 5. General Arrangement of Details of a Hot-Air Furnace, with Connecting Pipes.

ashes and soot. Furnaces are made either of cast iron, or of wrought-iron plates riveted together and provided with brick-lined firepots.

Types of Furnaces. Furnaces may be divided into two general types known as *direct-draft* and *indirect-draft.* Fig. 6 shows a common form of *direct-draft* furnace with a brick setting; the better class have a radiator, generally placed at the top, through which the gases pass before reaching the smoke-pipe. They have but one damper, usually combined with a cold-air check. Many of the cheaper direct-

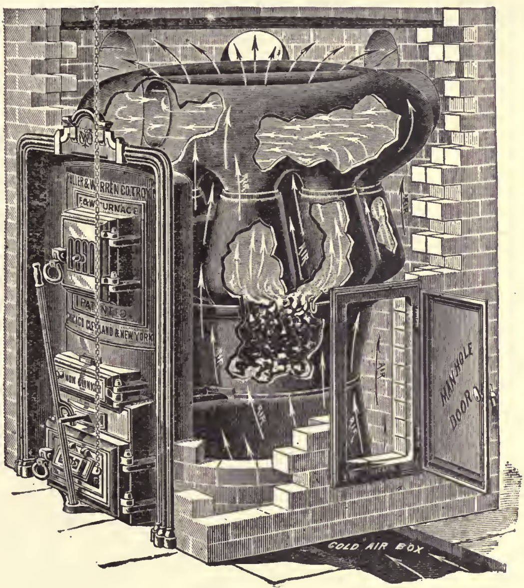


Fig. 6. A Common Type of Direct-Draft Furnace in Brick Setting. Cast-Iron Radiator at Top.

draft furnaces have no radiator at all, the gases passing directly into the smoke-pipe and carrying away much heat that should be utilized.

The furnace shown in Fig. 6 is made of cast iron and has a large radiator at the top; the smoke connection is shown at the rear.

Fig. 7 represents another form of direct-draft furnace. In this case the radiator is made of sheet-steel plates riveted together, and the outer casing is of heavy galvanized iron instead of brick.

In the ordinary *indirect-draft* type of furnace (see Fig. 8), the gases pass downward through flues to a radiator located near the base, thence upward through another flue to the smoke-pipe. In addition to the damper in the smoke-pipe, a direct-draft damper is required to give direct connection with the funnel when coal is first put on, to facilitate the escape of gas to the chimney. When the chimney draft

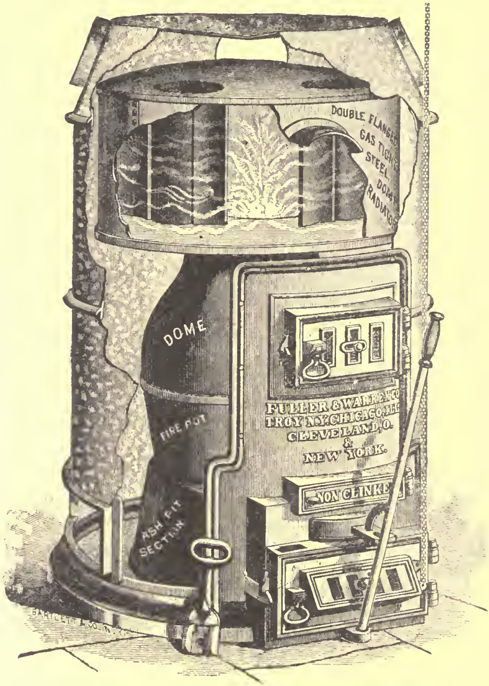
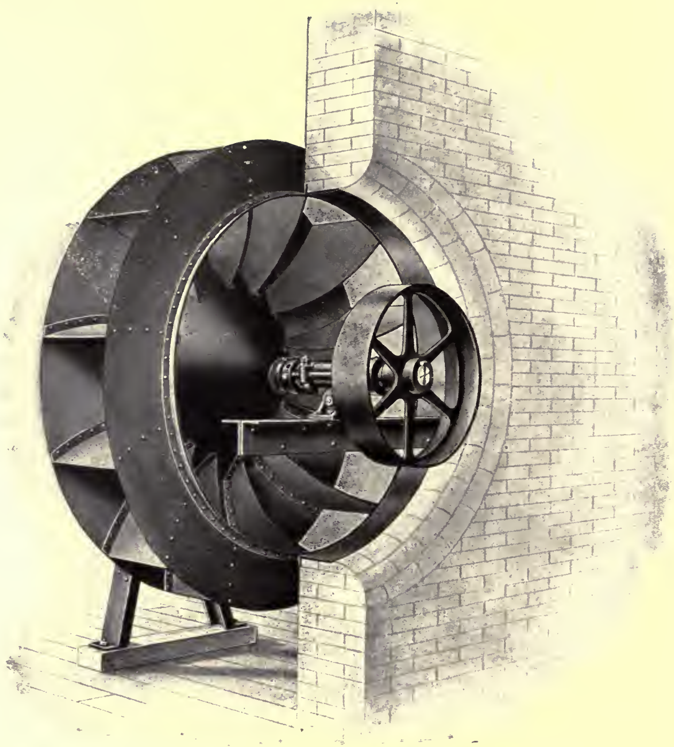


Fig. 7. Direct-Draft Furnace with Galvanized-Iron Casing. Radiator (at top) Made of Riveted Steel Plates.

is weak, trouble from gas is more likely to be experienced with furnaces of this type than with those having a direct draft.

Grates. No part of a furnace is of more importance than the grates. The plain grate rotating about a center pin was for a long time the one most commonly used. These grates were usually provided with a clinker door for removing any refuse too large to pass between the grate bars. The action of such grates tends to leave a



CONE EXHAUST FAN, INLET SIDE.
American Blower Co.

cone of ashes in the center of the fire causing it to burn more freely around the edges. A better form of grate is the revolving triangular pattern, which is now used in many of the leading furnaces. It consists of a series of triangular bars having teeth. The bars are connected by gears, and are turned by means of a detachable lever. If

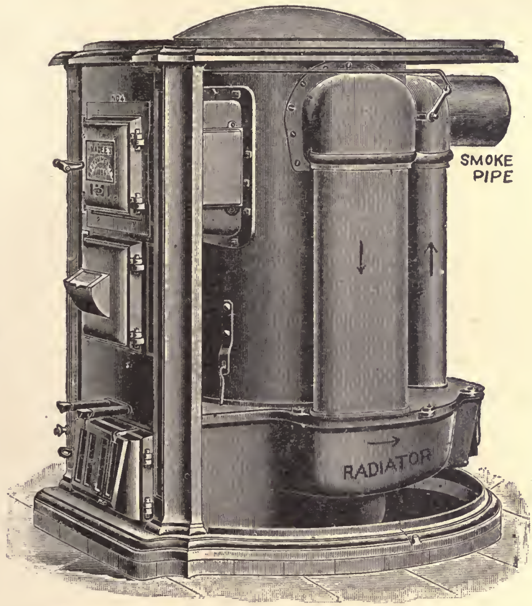


Fig. 8. Indirect-Draft Type of Furnace. Gases Pass Downward to Radiator at Bottom, Thence Upward to Smoke-Pipe.

properly used, this grate will cut a slice of ashes and clinkers from under the entire fire with little, if any loss of unconsumed coal.

The Firepot. Firepots are generally made of cast iron or of steel plate lined with firebrick. The depth ranges from about 12 to 18 inches. In cast-iron furnaces of the better class, the firepot is made very heavy, to insure durability and to render it less likely to become red-hot. The firepot is sometimes made in two pieces, to reduce the liability to cracking. The heating surface is sometimes increased by corrugations, pins, or ribs.

A firebrick lining is necessary in a wrought-iron or steel furnace to protect the thin shell from the intense heat of the fire. Since brick-lined firepots are much less effective than cast-iron in transmitting heat, such furnaces depend to a great extent for their efficiency on the heating surface in the dome and radiator; and this, as a rule, is much greater than in those of cast iron.

Cast-iron furnaces have the advantage when coal is first put on and the drop flues and radiator are cut out by the direct damper, of still giving off heat from the firepot, while in the case of brick linings very little heat is given off in this way, and the rooms are likely to become somewhat cooled before the fresh coal becomes thoroughly ignited.

Combustion Chamber. The body of the furnace above the firepot, commonly called the *dome* or *feed section*, provides a combustion chamber. This chamber should be of sufficient size to permit the gases to become thoroughly mixed with the air passing up through the fire or entering through openings provided for the purpose in the feed door. In a well-designed furnace, this space should be somewhat larger than the firepot.

Radiator. The radiator, so called, with which all furnaces of the better class are provided, acts as a sort of reservoir in which the gases are kept in contact with the air passing over the furnace until they have parted with a considerable portion of their heat. Radiators are built of cast iron, of steel plate, or of a combination of the two. The former is more durable and can be made with fewer joints, but owing to the difficulty of casting radiators of large size, steel plate is commonly used for the sides.

The effectiveness of a radiator depends on its form, its heating surface, and the difference between the temperature of the gases and the surrounding air. Owing to the accumulation of soot, the bottom surface becomes practically worthless after the furnace has been in use a short time; surfaces, to be effective, must therefore be self-cleaning.

If the radiator is placed near the bottom of the furnace the gases are surrounded by air at the lowest temperature, which renders the radiator more effective for a given size than if placed near the top and surrounded by warm air. On the other hand, the cold air has a tendency to condense the gases, and the acids thus formed are likely to corrode the iron.

Heating Surface. The different heating surfaces may be described as follows: Firepot surface; surfaces acted upon by direct rays of heat from the fire, such as the dome or combustion chamber; gas- or smoke-heated surfaces, such as flues or radiators; and extended surfaces, such as pins or ribs. Surfaces unlike in character and location, vary greatly in heating power, so that, in making comparisons of different furnaces, we must know the kind, form, and location of the heating surfaces, as well as the area.

In some furnaces having an unusually large amount of surface, it will be found on inspection that a large part would soon become practically useless from the accumulation of soot. In others a large portion of the surface is lined with firebrick, or is so situated that the air-currents are not likely to strike it.

The ratio of grate to heating surface varies somewhat according to the size of furnace. It may be taken as 1 to 25 in the smaller sizes, and 1 to 15 in the larger.

Efficiency. One of the first items to be determined in estimating the heating capacity of a furnace, is its efficiency—that is, the proportion of the heat in the coal that may be utilized for warming. The efficiency depends chiefly on the area of the heating surface as compared with the grate, on its character and arrangement, and on the rate of combustion. The usual proportions between grate and heating surface have been stated. The rate of combustion required to maintain a temperature of 70° in the house, depends, of course, on the outside temperature. In very cold weather a rate of 4 to 5 pounds of coal per square foot of grate per hour must be maintained.

One pound of good anthracite coal will give off about 13,000 B. T. U., and a good furnace should utilize 70 per cent of this heat. The efficiency of an ordinary furnace is often much less, sometimes as low as 50 per cent.

In estimating the required size of a first-class furnace with good chimney draft, we may safely count upon a maximum combustion of 5 pounds of coal per square foot of grate per hour, and may assume that 8,000 B. T. U. will be utilized for warming purposes from each pound burned. This quantity corresponds to an efficiency of 60 per cent.

Heating Capacity. Having determined the heat loss from a building by the methods previously given, it is a simple matter to compute the size of grate necessary to burn a sufficient quantity of coal to furnish the amount of heat required for warming.

In computing the size of furnace, it is customary to consider the whole house as a single room, with four outside walls and a cold attic. The heat losses by conduction and leakage are computed, and increased 10 per cent for the cold attic, and 16 per cent for exposure. The heat delivered to the various rooms may be considered as being made up of two parts—*first*, that required to warm the outside air up to 70° (the temperature of the rooms); and *second*, the quantity which must be added to this to offset the loss by conduction and leakage. Air is usually delivered through the registers at a temperature of 120°, with zero conditions outside, in the best class of residence work; so that $\frac{70}{120}$ of the heat given to the entering air may be considered as making up the first part, mentioned above, leaving $\frac{50}{120}$ available for purely heating purposes. From this it is evident that the heat supplied to the entering air must be equal to $1 \div \frac{50}{120} = 2.4$ times that required to offset the loss by conduction and leakage.

Example. The loss through the walls and windows of a building is found to be 80,000 B. T. U. per hour in zero weather. What will be the size of furnace required to maintain an inside temperature of 70 degrees?

From the above, we have the total heat required, equal to 80,000 $\times$ 2.4 = 192,000 B. T. U. per hour. If we assume that 8,000 B. T. U. are utilized per pound of coal, then 192,000 $\div$ 8,000 = 24 pounds of coal required per hour; and if 5 pounds can be burned on each square foot of grate per hour, then $\frac{24}{5} = 4.8$ square feet required. A grate 30 inches in diameter has an area of 4.9 square feet, and is the size we should use.

When the outside temperature is taken as 10° below zero, multiply by 2.6 instead of 2.4; and multiply by 2.8 for 20° below.

Table VII will be found useful in determining the diameter of firepot required.

TABLE VII

Firepot Dimensions

Average Diameter of Grate, in Inches	Area in Square Feet
18	1.77
20	2.18
22	2.64
24	3.14
26	3.69
28	4.27
30	4.91
32	5.58

EXAMPLES FOR PRACTICE

1. A brick apartment house is 20 feet wide, and has 4 stories, each being 10 feet in height. The house is one of a block, and is exposed only at the front and rear. The walls are 16 inches thick, and the block is so sheltered that no correction need be made for exposure. Single windows make up $\frac{1}{8}$ the total exposed surface. Figure for cold attic but warm basement. What area of grate surface will be required for a furnace to keep the house at a temperature of 70° when it is 10° below zero outside? Ans. 3.5 square feet.

2. A house having a furnace with a firepot 30 inches in diameter, is not sufficiently warmed, and it is decided to add a second furnace to be used in connection with the one already in. The heat loss from the building is found by computation to be 133,600 B. T. U. per hour, in zero weather. What diameter of firepot will be required for the extra furnace? Ans. 24 inches.

Location of Furnace. A furnace should be so placed that the warm-air pipes will be of nearly the same length. The air travels most readily through pipes leading toward the sheltered side of the house and to the upper rooms. Therefore pipes leading toward the north or west, or to rooms on the first floor, should be favored in regard to length and size. The furnace should be placed somewhat to the north or west of the center of the house, or toward the points of compass from which the prevailing winds blow.

Smoke-Pipes. Furnace smoke-pipes range in size from about 6 inches in the smaller sizes to 8 or 9 inches in the larger ones. They are generally made of galvanized iron of No. 24 gauge or heavier. The pipe should be carried to the chimney as directly as possible, avoiding bends which increase the resistance and diminish the draft. Where a smoke-pipe passes through a partition, it should be protected by a soapstone or double-perforated metal collar having a diameter at least 8 inches greater than that of the pipe. The top of the smoke-pipe should not be placed within 8 inches of unprotected beams, nor less than 6 inches under beams protected by asbestos or plaster with a metal shield beneath. A collar to make tight connection with the chimney should be riveted to the pipe about 5 inches from the end, to prevent the pipe being pushed too far into the flue. Where the pipe is of unusual length, it is well to cover it to prevent loss of heat and the condensation of smoke.

Chimney Flues. Chimney flues, if built of brick, should have walls 8 inches in thickness, unless terra-cotta linings are used, when only 4 inches of brickwork is required. Except in small houses where an 8 by 8-inch flue may be used, the nominal size of the smoke flue should be at least 8 by 12-inches, to allow for contractions or offsets. A clean-out door should be placed at the bottom of the flue, for removing ashes and soot. A square flue cannot be reckoned at its full area, as the corners are of little value. To avoid down drafts, the top of the chimney must be carried above the highest point of the roof unless provided with a suitable hood or top.

Cold-Air Box. The cold-air box should be large enough to supply a volume of air sufficient to fill all the hot-air pipes at the same time. If the supply is too small, the distribution is sure to be unequal, and the cellar will become overheated from lack of air to carry away the heat generated.

If a box is made too small, or is throttled down so that the volume of air entering the furnace is not large enough to fill all the pipes, it will be found that those leading to the less exposed side of the house or to the upper rooms will take the entire supply, and that additional air to supply the deficiency will be drawn down through registers in rooms less favorably situated. It is common practice to make the area of the cold-air box three-fourths the combined area of the hot-air pipes. The inlet should be placed where the prevailing cold winds will blow into it; this is commonly on the north or west side of the house. If it is placed on the side away from the wind, warm air from the furnace is likely to be drawn out through the cold-air box.

Whatever may be the location of the entrance to the cold-air box, changes in the direction of the wind may take place which will bring the inlet on the wrong side of the house. To prevent the possibility of such changes affecting the action of the furnace, the cold-air box is sometimes extended through the house and left open at both ends, with check-dampers arranged to prevent back-drafts. These checks should be placed some distance from the entrance, to prevent their becoming clogged with snow or sleet.

The cold-air box is generally made of matched boards; but galvanized iron is much better; it costs more than wood, but is well worth the extra expense on account of tightness, which keeps the dust and ashes from being drawn into the furnace casing to be discharged through the registers into the rooms above.

The cold-air inlet should be covered with galvanized wire netting with a mesh of at least three-eighths of an inch. The frame to which it is attached should not be smaller than the inside dimensions of the cold-air box. A door to admit air from the cellar to the cold-air box is generally provided. As a rule, air should be taken from this source, only when the house is temporarily unoccupied or during high winds.

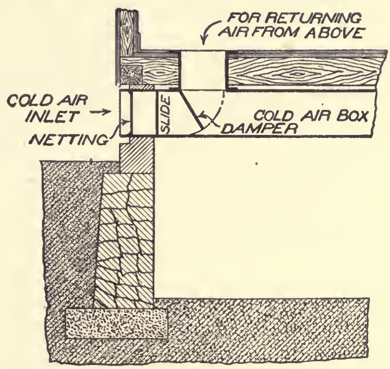


Fig 9. Common Method of Connecting Return Duct to Cold-Air Box.

Return Duct. In some cases it is desirable to return air to the furnace from the rooms above, to be reheated. Ducts for this purpose are common in places where the winter temperature is frequently below zero. Return ducts when used, should be in addition to the regular cold-air box. Fig. 9 shows a common method of making the connection between the two. By proper adjustment of the swinging damper, the air can be taken either from out of doors or through the register from the room above. The return register is often placed in the hallway of a house, so that it will take the cold air which rushes in when the door is opened and also that which may leak in around it while closed. Check-valves or flaps of light gossamer or woolen cloth should be placed between the cold-air box and the registers to prevent back-drafts during winds.

The return duct should not be used too freely at the expense of outdoor air, and its use is not recommended except during the night when air is admitted to the sleeping rooms through open windows.

Warm-Air Pipes. The required size of the warm-air pipe to any given room, depends on the heat loss from the room and on the volume of warm air required to offset this loss. Each cubic foot of air warmed from zero to 120 degrees brings into a room 2.2 B. T. U. We have already seen that in zero weather, with the air entering the registers at 120 degrees, only $\frac{50}{120}$ of the heat contained in the air is available for offsetting the losses by radiation and conduction, so that only $2.2 \times \frac{50}{120} = .9$ B. T. U. in each cubic foot of entering air can be utilized for warming purposes. Therefore, if we divide the computed heat loss in B. T. U. from a room, by .9, it will give the number of cubic feet of air at 120 degrees necessary to warm the room in zero weather.

As the outside temperature becomes colder, the quantity of heat brought in per cubic foot of air increases; but the proportion available for warming purposes becomes less at nearly the same rate, so

TABLE VIII

Warm-Air Pipe Dimensions

Diameter of Pipe, in Inches	Area in Square Inches	Area in Square Feet
6	28	.196
7	38	.267
8	50	.349
9	64	.442
10	79	.545
11	95	.660
12	113	.785
13	133	.922
14	154	1.07
15	177	1.23
16	201	1.40

that for all practical purposes we may use the figure .9 for all usual conditions. In calculating the size of pipe required, we may assume maximum velocities of 260 and 380 feet per minute for rooms on the first and second floors respectively. Knowing the number of cubic feet of air per minute to be delivered, we can divide it by the velocity, which will give us the required area of the pipe in square feet.

Round pipes of tin or galvanized iron are used for this purpose. Table VIII will be found useful in determining the required diameters of pipe in inches.

Example. The heat loss from a room on the second floor is 18,000 B. T. U. per hour. What diameter of warm-air pipe will be required?

18,000 ÷ .9 = 20,000 = cubic feet of air required per hour. 20,000 ÷ 60 = 333 per minute. Assuming a velocity of 380 feet per minute, we have 333 ÷ 380 = .87 square foot, which is the area of pipe required. Referring to Table VIII, we find this comes between a 12-inch and a 13-inch pipe, and the larger size would probably be chosen.

EXAMPLES FOR PRACTICE

1. A first-floor room has a computed loss of 27,000 B. T. U. per hour when it is 10° below zero. The air for warming is to enter through two pipes of equal size, and at a temperature of 120 degrees. What will be the required diameter of the pipes?

Ans. 14 inches.

2. If in the above example the room had been on the second floor, and the air was to be delivered through a single pipe, what diameter would be required?

Ans. 16 inches.

Since long horizontal runs of pipe increase the resistance and loss of heat, they should not in general be over 12 or 14 feet in length. This applies especially to pipes leading to rooms on the first floor, or to those on the cold side of the house. Pipes of excessive length should be increased in size because of the added resistance.

Figs. 10 and 11 show common methods of running the pipes in the basement. The first gives the best results, and should be used where the basement is of sufficient height to allow it. A damper should be placed in each pipe near the furnace, for regulating the flow of air to the different rooms, or for shutting it off entirely when desired.

While round pipe risers give the best results, it is not always possible to provide a sufficient space for them, and flat or oval pipes are substituted. When vertical pipes must be placed in single partitions, much better results will be obtained if the studding can be

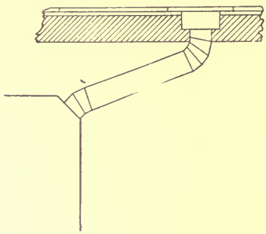


Fig. 10.

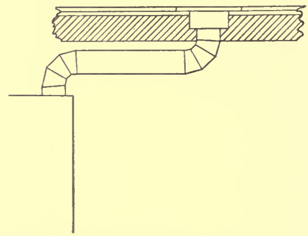


Fig. 11.

Common Methods of Running Hot-Air Pipes in Basement. Method Shown in Fig. 10 is Preferable where Feasible.

made 5 or 6 inches deep instead of 4 as is usually done. Flues should never in any case be made less than 3½ inches in depth. Each room should be heated by a separate pipe. In some cases, however, it is allowable to run a single riser to heat two unimportant rooms on an upper floor. A clear space of at least ½ inch should be left between the risers and studs, and the latter should be carefully tinned, and the

TABLE IX

Dimensions of Oval Pipes

Dimension of Pipe	Area in Square Inches
6 ovaled to 5 in.	27
7 " " 4 "	31
7 " " 3½ "	29
7 " " 6 "	38
8 " " 5 "	43
9 " " 4 "	45
9 " " 6 "	57
9 " " 5 "	51
10 " " 3½ "	46
11 " " 4 "	58
12 " " 3½ "	55
10 " " 6 "	67
11 " " 5 "	67
14 " " 4 "	76
15 " " 3½ "	73
12 " " 6 "	85
12 " " 5 "	75
19 " " 4 "	96
20 " " 3½ "	100

space between them on both sides covered with tin, asbestos, or wire lath.

Table IX gives the capacity of oval pipes. A 6-inch pipe ovaled to 5 means that a 6-inch pipe has been flattened out to a thickness of 5 inches, and column 2 gives the resulting area.

Having determined the size of round pipe required, an equivalent oval pipe can be selected from the table to suit the space available.

Registers. The registers which control the supply of warm air to the rooms, generally have a net area equal to two-thirds of their gross area. The net area should be from 10 to 20 per cent greater than the area of the pipe connected with it. It is common practice to use registers having the short dimensions equal to, and the long dimensions about one-half greater than, the diameter of the pipe. This would give standard sizes for different diameters of pipe, as listed in Table X.

TABLE X
Sizes of Registers for Different Sizes of Pipes

Diameter of Pipe	Size of Register
6 in.	6 × 10 in.
7 "	7 × 10 "
8 "	8 × 12 "
9 "	9 × 14 "
10 "	10 × 15 "
11 "	11 × 16 "
12 "	12 × 17 "
13 "	14 × 20 "
14 "	14 × 22 "
15 "	15 × 22 "
16 "	16 × 24 "

Combination Systems. A combination system for heating by hot air and hot water consists of an ordinary furnace with some form of surface for heating water, placed either in contact with the fire or suspended above it. Fig. 12 shows a common arrangement where part of the heating surface forms a portion of the lining to the firepot and the remainder is above the fire.

Care must be taken to proportion properly the work to be done by the air and the water; else one will operate at the expense of the other. One square foot of heating surface in contact with the fire is capable of supplying from 40 to 50 square feet of radiating surface, and one square foot suspended over the fire will supply from 15 to 25 square feet of radiation.

The value or efficiency of the heating surface varies so widely in different makes that it is best to state the required conditions to the

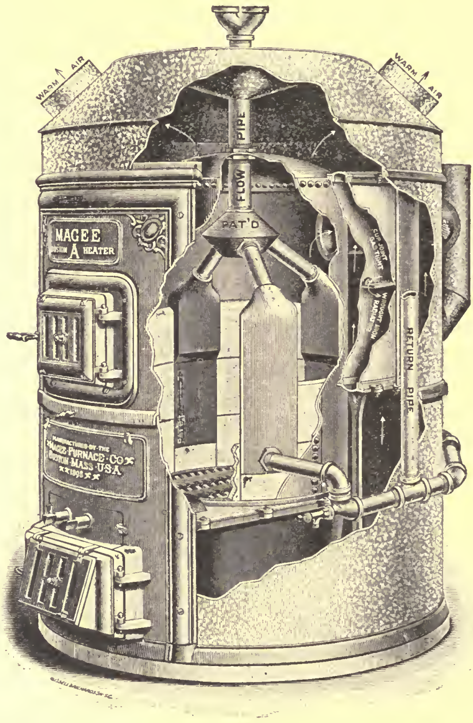


Fig. 12. Combination Furnace, for Heating by Both Hot Air and Hot Water.

manufacturers and have them proportion the surfaces as their experience has found best for their particular type of furnace.

Care and Management of Furnaces. The following general rules apply to the management of all hard coal furnaces.

The fire should be thoroughly shaken once or twice daily in cold weather. It is well to keep the firepot heaping full at all times. In this way a more even temperature may be maintained, less attention is required, and no more coal is burned than when the pot is only partly filled. In mild weather the mistake is frequently made of carrying a thin fire, which requires frequent attention and is likely to die out. Instead, to diminish the temperature in the house, keep the firepot full and allow ashes to accumulate on the grate (not under it) by shaking less frequently or less vigorously. The ashes will hold the heat and render it an easy matter to maintain and control the fire. When feeding coal on a low fire, open the drafts and neither rake nor shake the fire till the fresh coal becomes ignited. The air supply to the fire is of the greatest importance. An insufficient amount results in incomplete combustion and a great loss of heat. To secure proper combustion, the fire should be controlled principally by means of the ash-pit through the ash-pit door or slide.

The smoke-pipe damper should be opened only enough to carry off the gas or smoke and to give the necessary draft. The openings in the feed door act as a check on the fire, and should be kept closed during cold weather, except just after firing, when with a good draft they may be partly opened to increase the air-supply and promote the proper combustion of the gases.

Keep the ash-pit clear to avoid warping or melting the grate. The cold-air box should be kept wide open except during winds or when the fire is low. At such times it may be partly, but never completely closed. Too much stress cannot be laid on the importance of a sufficient air-supply to the furnace. It costs little if any more to maintain a comfortable temperature in the house night and day than to allow the rooms to become so cold during the night that the fire must be forced in the morning to warm them up to a comfortable temperature.

In case the warm air fails at times to reach certain rooms, it may be forced into them by temporarily closing the registers in other rooms. The current once established will generally continue after the other registers have been opened.

It is best to burn as hard coal as the draft will warrant. Egg size is better than larger coal, since for a given weight small lumps expose more surface and ignite more quickly than larger ones. The furnace and smoke-pipe should be thoroughly cleaned once a year.

This should be done just after the fire has been allowed to go out in the spring.

STEAM BOILERS

Types. The boilers used for heating are the same as have already been described for power work. In addition there is the cast-iron sectional boiler, used almost exclusively for dwelling-houses.

Tubular Boilers. Tubular boilers are largely used for heating purposes, and are adapted to all classes of buildings except dwelling-houses and the special cases mentioned later, for which sectional boilers are preferable. A *boiler horse-power* has been defined as the evaporation of $34\frac{1}{2}$ pounds of water from and at a temperature of 212 degrees, and in doing this 33,317 B. T. U. are absorbed, which are again given out when the steam is condensed in the radiators. Hence to find the boiler H. P. required for warming any given building, we have only to compute the heat loss per hour by the methods already given, and divide the result by 33,330. It is more common to divide by the number 33,000, which gives a slightly larger boiler and is on the side of safety.

The commercial horse-power of a well-designed boiler is based upon its heating surface; and for the best economy in heating work, it should be so proportioned as to have about 1 square foot heating of surface for each 2 pounds of water to be evaporated from and at 212 degrees F. This gives $34.5 \div 2 = 17.2$ square feet of heating surface per horse-power, which is generally taken as 15 in practice. Makers of tubular boilers commonly rate them on a basis of 12 square feet of heating surface per horse-power. This is a safe figure under the conditions of power work, where skilled firemen are employed and where more care is taken to keep the heating surfaces free from soot and ashes. For heating plants, however, it is better to rate the boilers upon 15 square feet per horse-power as stated above.

There is some difference of opinion as to the proper method of computing the heating surface of tubular boilers. In general, all surface is taken which is exposed to the hot gases on one side and to the water on the other. A safe rule, and the one by which Table XII is computed, is to take $\frac{1}{2}$ the area of the shell, $\frac{2}{3}$ of the rear head, less the tube area, and the interior surface of all the tubes.

The required amount of grate area, and the proper ratio of heating surface to grate area, vary a good deal, depending on the character of the fuel and on the chimney draft. By assuming the probable rates of combustion and evaporation, we may compute the required grate area for any boiler from the formula:

$$S = \frac{H.P. \times 34.5}{E \times C},$$

in which

S = Total grate area, in square feet;

E = Pounds of water evaporated per pound of coal;

C = Pounds of coal burned per square foot of grate per hour.

Table XI gives the approximate grate area per H. P. for different rates of evaporation and combustion as computed by the above equation.

TABLE XI

Grate Area per Horse-Power for Different Rates of Evaporation and Combustion

Pounds of Steam per Pound of Coal	Pounds of Coal Burned per Square Foot of Grate per Hour		
	8 lbs.	10 lbs.	12 lbs.
	Square Feet of Grate Surface per Horse-Power		
10	.43	.35	.28
9	.48	.38	.32
8	.54	.43	.36
7	.62	.49	.41
6	.72	.58	.48

For example, with an evaporation of 8 pounds of steam per pound of coal, and a combustion of 10 pounds of coal per square foot of grate, .43 of a square foot of grate surface per H. P. would be called for.

The ratio of heating to grate surface in this type of boiler ranges from 30 to 40, and therefore allows under ordinary conditions a combustion of from 8 to 10 pounds of coal per square foot of grate. This is easily obtained with a good chimney draft and careful firing. The larger the boiler, the more important the plant usually, and the greater the care bestowed upon it, so that we may generally count on a higher rate of combustion and a greater efficiency as the size of the boiler increases. Table XII will be found very useful in determining the size of boiler required under different conditions. The grate area is computed for an evaporation of 8 pounds of water per pound

TABLE XII

Diameter of Shell in Inches	Number of Tubes	Diameter of Tubes in Inches	Length of Tubes in Feet	Horse-Power	Size of Grate in Inches	Size of Uptake in Inches	Size of Smoke-pipe in Sq. In
30	28	2½	6	8.5	24 x 36	10 x 14	140
			7	9.9	24 x 36	10 x 14	140
			8	11.2	24 x 36	10 x 14	140
			9	12.6	24 x 42	10 x 14	140
			10	14.0	24 x 42	10 x 14	140
36	34	2½	8	13.6	30 x 36	10 x 16	160
			9	15.3	30 x 42	10 x 18	180
			10	16.9	30 x 42	10 x 18	180
			11	18.6	30 x 48	10 x 20	200
			12	20.9	30 x 48	10 x 20	200
42	34	3	9	18.5	36 x 42	10 x 20	200
			10	20.5	36 x 42	10 x 20	200
			11	22.5	36 x 48	10 x 25	250
			12	24.5	36 x 48	10 x 25	250
			13	26.5	36 x 48	10 x 28	280
			14	28.5	36 x 54	10 x 28	280
48	44	3	10	30.4	42 x 48	10 x 28	280
			11	33.2	42 x 48	10 x 28	280
			12	35.7	42 x 54	10 x 32	320
			13	38.3	42 x 54	10 x 32	320
			14	40.8	42 x 60	10 x 36	360
			15	43.4	42 x 60	10 x 36	360
			16	45.9	42 x 60	10 x 36	360
54	54	3	11	34.6	48 x 54	10 x 38	380
			12	37.7	48 x 54	10 x 38	380
			13	40.8	48 x 54	10 x 38	380
			14	43.9	48 x 54	10 x 38	380
			15	47.0	48 x 60	10 x 40	400
			16	50.1	48 x 60	10 x 40	400
	46	3½	17	53.0	48 x 60	10 x 40	400
60	72	3	12	48.4	54 x 60	12 x 40	460
			13	52.4	54 x 60	12 x 40	460
			14	56.4	54 x 60	12 x 40	460
			15	60.4	54 x 66	12 x 42	500
			16	64.4	54 x 66	12 x 42	500
	64	3½	17	71.4	54 x 72	12 x 48	550
			18	75.6	54 x 72	12 x 48	550
66	90	3	14	70.1	60 x 66	12 x 48	500
			15	75.0	60 x 72	12 x 52	620
			16	80.0	60 x 72	12 x 52	620
	78	3½	17	86.0	60 x 78	12 x 56	670
			18	91.1	60 x 78	12 x 56	670
			19	96.2	60 x 78	12 x 56	670
	62	4	20	93.1	60 x 78	12 x 56	670
72	114	3	14	87.4	66 x 72	12 x 56	670
			15	93.6	66 x 72	12 x 56	670
			16	99.7	66 x 78	12 x 62	740
	98	3½	17	106.4	66 x 78	12 x 62	740
			18	112.6	66 x 84	12 x 66	790
			19	118.8	66 x 84	12 x 66	790
	72	4	20	107.3	66 x 84	12 x 66	790

DIRECT-INDIRECT SYSTEM OF WARMING, SHOWING ADJUSTABLE DAMPER.
American Radiator Company.

of coal, which corresponds to an efficiency of about 60 per cent, and is about the average obtained in practice for heating boilers.

The areas of uptake and smoke-pipe are figured on a basis of 1 square foot to 7 square feet of grate surface, and the results given in round numbers. In the smaller sizes the relative size of smoke-pipe is greater. The rate of combustion runs from 6 pounds in the smaller sizes to $11\frac{1}{2}$ in the larger. Boilers of the proportions given in the table, correspond well with those used in actual practice, and may be relied upon to give good results under all ordinary conditions.

Water-tube boilers are often used for heating purposes, but more especially in connection with power plants. The method of computing the required H. P. is the same as for tubular boilers.

Sectional Boilers. Fig. 13 shows a common form of cast-iron boiler. It is made up of slabs or sections, each one of which is connected by nipples with headers at the sides and top. The top header acts as a steam drum, and the lower ones act as mud drums; they also receive the water of condensation from the radiators. The gases from the fire pass backward and forward through flues and are finally taken off at the rear of the boiler.

Another common form of sectional boiler is shown in Fig. 14. It is made up of sections which increase the length like the one just described. These boilers have no drum connecting with the sections; but instead, each section connects with the adjacent one through openings at the top and bottom, as shown.

The ratio of heating to grate surface in boilers of this type ranges from 15 to 25 in the best makes. They are provided with the usual attachments, such as pressure-gauge, water-glass, gauge-cocks, and safety-valve; a low-pressure damper regulator is furnished for operating the draft doors, thus keeping the steam pressure practically constant. A pressure of from 1 to 5 pounds is usually carried on these boilers, depending upon the outside temperature. The usual setting is simply a covering of some kind of non-conducting material like plastic magnesia or asbestos, although some forms are enclosed in light brickwork.

In computing the required size, we may proceed in the same manner as in the case of a furnace. For the best types of house-heating boilers, we may assume a combustion of 5 pounds of coal per square foot of grate per hour, and an average efficiency of 60 per cent, which corresponds to 8,000 B. T. U. per pound of coal, available for useful work.

In the case of direct-steam heating, we have only to supply heat to offset that lost by radiation and conduction; so that the grate area may be found by dividing the computed heat loss per hour by 8,000, which gives the number of pounds of coal; and this in turn, divided by 5, will give the area of grate required. The most efficient rate of

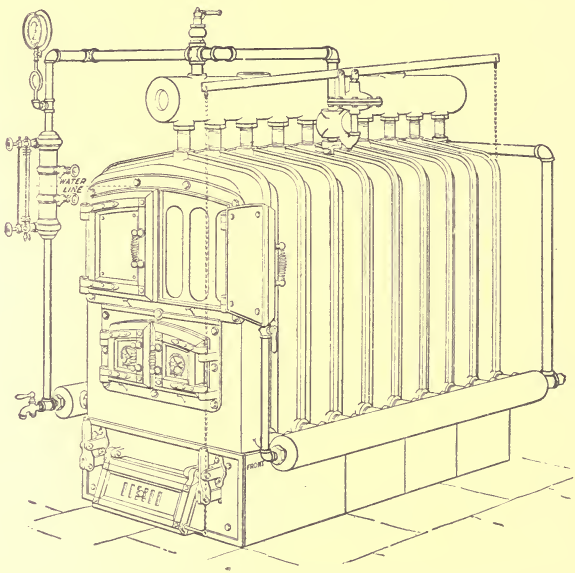


Fig. 13. Common Type of Cast-Iron Sectional Boiler. Note Headers at Sides and Top Acting as Drums.

combustion will depend somewhat upon the ratio between the grate and heating surface. It has been found by experience that about $\frac{1}{4}$ of a pound of coal per hour for each square foot of heating surface gives the best results; so that, by knowing the ratio of heating surface to grate area for any make of heater, we can easily compute the most efficient rate of combustion, and from it determine the necessary grate area.

For example, suppose the heat loss from a building to be 480,000 B. T. U. per hour, and that we wish to use a heater in which the ratio of heating surface to grate area is 24. What will be the most efficient rate of combustion and the required grate area? 480,000 ÷ 8,000 = 60 pounds of coal per hour, and 24 ÷ 4 = 6, which is the best rate of combustion to employ; therefore 60 ÷ 6 = 10, the grate area required.

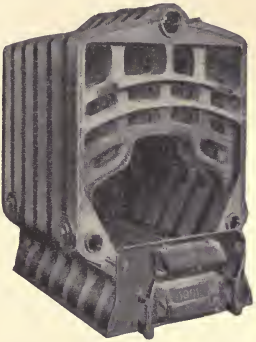


Fig. 14. Another Type of Sectional Boiler. Here there are no drums, the sections being directly connected through openings at top and bottom.
Courtesy of American Radiator Co.

There are many different designs of cast-iron boilers for low-pressure steam and hot-water heating. In general, boilers having a drum connected by nipples with each section give dryer steam and hold a steadier water-line than the second form, especially when forced above their normal capacity. The steam, in passing through the openings between successive sections in order to reach the outlet, is apt to carry with it more or less water, and to choke the openings, thus producing an uneven pressure in different parts of the boiler.

In the case of hot-water boilers this objection disappears.

In order to adapt this type of boiler to steam work, the opening between the sections should be of good size, with an ample steam space above the water-line; and the nozzles for the discharge of steam should be located at frequent intervals.

EXAMPLES FOR PRACTICE

1. The heat loss from a building is 240,000 B. T. U. per hour, and the ratio of heating to grate area in the heater to be used is 20. What will be the required grate area? Ans. 6 sq. ft.

2. The heat loss from a building is 168,000 B. T. U. per hour, and the chimney draft is such that not over 3 pounds of coal per hour can be burned per square foot of grate. What ratio of heating to grate area will be necessary, and what will be the required grate area?
Ans. Ratio, 12. Grate area, 7 sq. ft.

Cast-iron sectional boilers are used for dwelling-houses, small schoolhouses, churches, etc., where low pressures are carried. They are increased in size by adding more slabs or sections. After a certain length is reached, the rear sections become less and less efficient, thus limiting the size and power.

Horse=Power for Ventilation. We already know that one B. T. U. will raise the temperature of 1 cubic foot of air 55 degrees, or it will raise 100 cubic feet $\frac{1}{100}$ of 55 degrees, or $\frac{55}{100}$ of 1 degree; therefore, to raise 100 cubic feet 1 degree, it will take $1 \div \frac{55}{100}$, or $\frac{100}{55}$ B. T. U.; and to raise 100 cubic feet through 100 degrees, it will take $\frac{100}{55} \times 100$ B. T. U. In other words, the B. T. U. required to raise any given volume of air through any number of degrees in temperature, is equal to

$$\frac{\text{Volume of air in cubic ft.} \times \text{Degrees raised}}{55}$$

Example. How many B. T. U. are required to raise 100,000 cubic feet of air 70 degrees?

$$\frac{100{,}000 \times 70}{55} = 127{,}272 +$$

To compute the H. P. required for the ventilation of a building, we multiply the total air-supply, in cubic feet per hour, by the number of degrees through which it is to be raised, and divide the result by 55. This gives the B. T. U. per hour, which, divided by 33,000, will give the H. P. required. In using this rule, always take the air-supply in cubic feet per *hour*.

EXAMPLES FOR PRACTICE

1. The heat loss from a building is 1,650,000 B. T. U. per hour. There is to be an air-supply of 1,500,000 cubic feet per hour, raised through 70 degrees. What is the total boiler H. P. required?

Ans. 108.

2. A high school has 10 classrooms, each occupied by 50 pupils. Air is to be delivered to the rooms at a temperature of 70 degrees. What will be the total H. P. required to heat and ventilate the building when it is 10 degrees below zero, if the heat loss through walls and windows is 1,320,000 B. T. U. per hour? Ans. 106+.

DIRECT=STEAM HEATING

A system of direct-steam heating consists (1) of a furnace and boiler for the combustion of fuel and the generation of steam; (2) a system of pipes for conveying the steam to the radiators and for returning the water of condensation to the boiler; and (3) radiators or coils placed in the rooms for diffusing the heat.

Various types of boilers are used, depending upon the size and kind of building to be warmed. Some form of cast-iron sectional boiler is commonly used for dwelling-houses, while the tubular or water-tube boiler is more usually employed in larger buildings. Where the boiler is used for heating purposes only, a low steam-pressure of from 2 to 10 pounds is carried, and the condensation flows back by gravity to the boiler, which is placed below the lowest radiator. When, for any reason, a higher pressure is required, the steam for the heating system is made to pass through a reducing valve, and the condensation is returned to the boiler by means of a pump or return trap.

Fig. 15. Common Type of Cast-Iron Sectional Radiator.

Types of Radiating Surface. The radiation used in direct-steam heating is made up of cast-iron radiators of various forms, pipe radiators, and circulation coils.

Cast-Iron Radiators. The general form of a cast-iron sectional radiator is shown in Fig. 15. Radiators of this type are made up of sections, the number depending upon the amount of heating surface required. Fig. 16 shows an intermediate section of a radiator of this type. It is simply a loop with inlet and outlet at the bottom. The end sections are the same, except that they have legs, as shown in Fig. 17. These sections are connected at the bottom by special nipples, so that steam entering at the end fills the bottom of the radiator, and, being lighter than the air, rises through the loops and forces the air downward and toward the farther end, where it is discharged through an air-valve placed about midway of the last section. There are many different designs varying in height and width, to suit all conditions. The wall pattern shown in Fig. 18 is very convenient when it is desired to place the radiator above the floor, as in bathrooms, etc.; it is also a convenient form to place under the windows of halls and churches to counteract the effect of cold down drafts. It is adapted to nearly every place where the ordinary direct radiator can be used, and may be connected up in different ways to meet the various requirements.

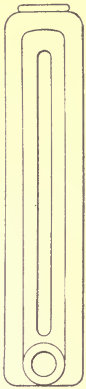


Fig. 16.

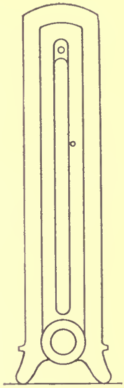


Fig. 17.

Intermediate and End Sections of Radiator Shown in Fig. 15. The end sections (at right) have legs.

A low and moderately shallow radiator, with ample space for the circulation of air between the sections, is more efficient than a deep radiator with the sections closely packed together. One- and two-column radiators, so called, are preferable to three- and four-column, when there is sufficient space to use them.

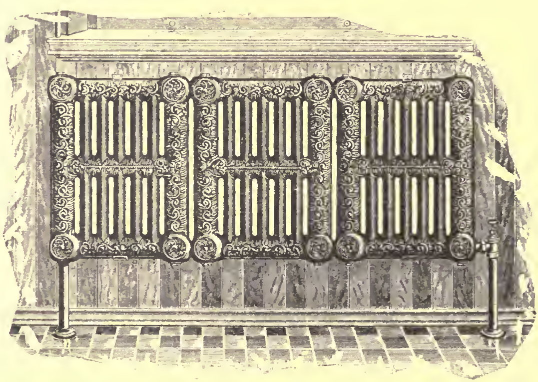


Fig. 18. Cast-Iron Sectional Radiator of Wall Pattern.

The standard height of a radiator is 36 or 38 inches, and, if possible, it is better not to exceed this.

For small radiators, it is better practice to use lower sections and increase the length; this makes the radiator slightly more efficient and gives a much better appearance.

To get the best results from wall radiators, they should be set out at least $1\frac{1}{2}$ inches from the wall to allow a free circulation of air back of them. Patterns having cross-bars should be placed, if possible, with the bars in a vertical position, as their efficiency is impaired somewhat when placed horizontally.

Pipe Radiators. This type of radiator (see Fig. 19) is made up of wrought-iron pipes screwed into a cast-iron base. The pipes are either connected in pairs at the top by return bends, or each separate tube has a thin metal diaphragm passing up the center nearly to the top. It is necessary that a loop be formed, else a "dead end" would occur. This would become filled with air and prevent steam from entering, thus causing portions of the radiator to remain cold.

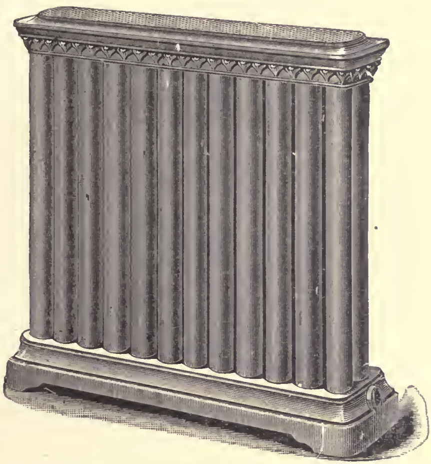


Fig. 19. Wrought-Iron Pipe Radiator.

Circulation Coils. These are usually made up of 1 or $1\frac{1}{4}$-inch wrought-iron pipe, and may be hung on the walls of a room by means of hook plates, or suspended overhead on hangers and rolls.

Fig. 20 shows a common form for schoolhouse and similar work; this coil is usually made of $1\frac{1}{4}$-inch pipe screwed into *headers* or *branch tees* at the ends, and is hung on the wall just below the windows. This is known as a *branch coil.* Fig. 21 shows a *trombone coil,* which is commonly used when the pipes cannot turn a corner, and where the entire coil must be placed upon one side of the room. Fig. 22

is called a *miter coil*, and is used under the same conditions as a trombone coil if there is room for the vertical portion. This form is not so pleasing in appearance as either of the other two, and is found only in factories or shops, where looks are of minor importance.

Fig. 20. Common Form of "Branch" Coil for Circulation of Direct Steam.

Overhead coils are usually of the miter form, laid on the side and suspended about a foot from the ceiling; they are less efficient than when placed nearer the floor, as the warm air stays at the ceiling and the lower part of the room is likely to remain cold. They are used

Fig. 21. "Trombone" Coil. Used where Entire Coil must be Placed on One Side of Room

only when wall coils or radiators would be in the way of fixtures, or when they would come below the water-line of the boiler if placed near the floor.

When steam is first turned on a coil, it usually passes through a

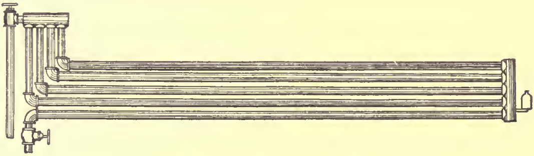


Fig. 22. "Miter" Coil. Adapted, like the "Trombone," Only to a Single Wall. Frequently Used in Factories and Shops.

portion of the pipes first and heats them while the others remain cold and full of air. Therefore the coil must always be made up in such a way that each pipe shall have a certain amount of spring and may expand independently without bringing undue strains upon the others. Circulation coils should incline about 1 inch in 20 feet toward the return end in order to secure proper drainage and quietness of operation.

Efficiency of Radiators. The efficiency of a radiator—that is, the B. T. U. which it gives off per square foot of surface per hour—depends upon the difference in temperature between the steam in the radiator and the surrounding air, the velocity of the air over the radiator, and the quality of the surface, whether smooth or rough. In ordinary low-pressure heating, the first condition is practically constant; but the second varies somewhat with the pattern of the radiator. An open design which allows the air to circulate freely over the radiating surfaces, is more efficient than a closed pattern, and for this reason a pipe coil is more efficient than a radiator.

In a large number of tests of cast-iron and pipe radiators, working under usual conditions, the heat given off per square foot of surface per hour for each degree difference in temperature between the steam and surrounding air was found to average about 1.7 B. T. U. The temperature of steam at 3 pounds' pressure is 220 degrees, and $220-70=150$, which may be taken as the average difference between the temperature of the steam and the air of the room, in ordinary low-pressure work. Taking the above results, we have $150 \times 1.7 = 255$ B. T. U. as the efficiency of an average cast-iron or pipe radiator. This, for convenient use, may be taken as 250. A circulation coil made up of pipes from 1 to 2 inches in diameter, will easily give off 300 B. T. U. under the same conditions; and a cast-iron wall radiator with ample space back of it should have an efficiency equal to that of a wall coil. While overhead coils have a higher efficiency than cast-iron radiators, their position near the ceiling reduces their effectiveness, so that in practice the efficiency should not be taken over 250 B. T. U. per hour at the most. Tabulating the above we have:

TABLE XIII

Efficiency of Radiators, Coils, etc.

Type of Radiating Surface	Radiation per Square Foot of Surface per Hour	
Cast-Iron Sectional and Pipe Radiators	250	B. T. U.
Wall Radiators	300	"
Ceiling Coils	200 to 250	"
Wall Coils	300	"

If the radiator is for warming a room which is to be kept at a temperature above or below 70 degrees, or if the steam pressure is greater than 3 pounds, the radiating surface may be changed in the same proportion as the difference in temperature between the steam and the air.

For example, if a room is to be kept at a temperature of 60°, the efficiency of the radiator becomes $\frac{150}{140} \times 250 = 268$; that is, the efficiency varies directly as the difference in temperature between the steam and the air of the room. It is not customary to consider this unless the steam pressure should be raised to 10 or 15 pounds or the temperature of the rooms changed 15 or 20 degrees from the normal.

From the above it is easy to compute the size of radiator for any given room. First compute the heat loss per hour by conduction and leakage in the coldest weather; then divide the result by the efficiency of the type of radiator to be used. It is customary to make the radiators of such size that they will warm the rooms to 70 degrees in the coldest weather. As the low-temperature limit varies a good deal in different localities, even in the same State, the lowest temperature for which we wish to provide must be settled upon before any calculations are made. In New England and through the Middle and Western States, it is usual to figure on warming a building to 70 degrees when the outside temperature is from zero to 10 degrees below.

The different makers of radiators publish in their catalogues, tables giving the square feet of heating surface for different styles and heights, and these can be used in determining the number of sections required for all special cases.

If pipe coils are to be used, it becomes necessary to reduce square feet of heating surface to linear feet of pipe; this can be done by means of the factors given below.

$$\text{Square feet of heating surface} \times \begin{cases} 3 = \text{linear ft. of } 1\text{-in. pipe} \\ 2.3 = \text{“ “ } 1\tfrac{1}{4}\text{-in. “} \\ 2 = \text{“ “ } 1\tfrac{1}{2}\text{-in. “} \\ 1.6 = \text{“ “ } 2\text{-in. “} \end{cases}$$

The size of radiator is made only sufficient to keep the room warm after it is once heated; and no allowance is made for *warming up*; that is, the heat given off by the radiator is just equal to that lost through walls and. windows. This condition is offset in two ways—

first, when the room is cold, the difference in temperature between the steam and the air of the room is greater, and the radiator is more efficient; and *second,* the radiator is proportioned for the coldest weather, so that for a greater part of the time it is larger than necessary.

EXAMPLES FOR PRACTICE

1. The heat loss from a room is 25,000 B. T. U. per hour in the coldest weather. What size of direct radiator will be required?
Ans. 100 square feet.

2. A schoolroom is to be warmed with circulation coils of $1\frac{1}{4}$-inch pipe. The heat loss is 30,000 B. T. U. per hour. What length of pipe will be required? Ans. 230 linear feet.

Location of Radiators. Radiators should, if possible, be placed in the coldest part of the room, as under windows or near outside doors. In living rooms it is often desirable to keep the windows free, in which case the radiators may be placed at one side. Circulation coils are run along the outside walls of a room under the windows. Sometimes the position of the radiators is decided by the necessary location of the pipe risers, so that a certain amount of judgment must be used in each special case as to the best arrangement to suit all requirements.

Systems of Piping. There are three distinct systems of piping, known as the *two-pipe system,* the *one-pipe relief system,* and the *one-pipe circuit system,* with various modifications of each and combinations of the different systems.

Fig. 23 shows the arrangement of piping and radiators in the two-pipe system. The steam main leads from the top of the boiler, and the branches are carried along near the basement ceiling. Risers are taken from the supply branches, and carried up to the radiators on the different floors; and return pipes are brought down to the return mains, which should be placed near the basement floor below the water-line of the boiler. Where the building is more than two stories high, radiators in similar positions on different floors are connected with the same riser, which may run to the highest floor; and a corresponding return drop connecting with each radiator is carried down beside the riser to the basement. A system in which the main horizontal returns are below the water-line of the boiler is said to have a *wet* or *sealed* return. If the returns are overhead and above the water-line, it is called a *dry* return. Where the steam is exposed to extended surfaces of water, as in overhead returns, where the condensation partially fills the pipes, there is likely to be cracking or *water-hammer,* due to the sudden condensation of the steam as it comes in contact with the cooler water. This is especially noticeable when steam is first turned into cold pipes and radiators, and the condensation is excessive. When dry returns are used, the pipes should be large and have a good pitch toward the boiler.

In the case of sealed returns, the only contact between the steam

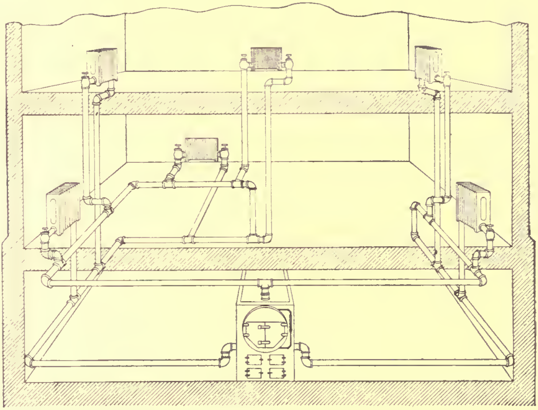


Fig. 23. Arrangement of Piping and Radiators in "Two-Pipe" System.

and standing water is in the vertical returns, where the exposed surfaces are very small (being equal to the sectional area of the pipes), and trouble from water-hammer is practically done away with. Dry returns should be given an incline of at least 1 inch in 10 feet, while for wet returns 1 inch in 20 or even 40 feet is ample. The ends of all steam mains and branches should be dripped into the returns. If the return is sealed, the drip may be directly connected as shown in Fig. 24; but if it is dry, the connection should be provided with a siphon loop as indicated in Fig. 25. The loop becomes filled with water, and prevents steam from flowing directly into the return. As the condensation collects in the loop, it overflows into the return pipe and is carried away. The return pipes in this case are of course filled with steam above the water; but it is steam which has passed through the radiators and their return connections, and is therefore at a slightly lower pressure; so that, if steam were admitted directly from the main, it would tend to hold back the water in more distant returns and cause surging and cracking in the pipes. Sometimes the boiler is at a lower level than the basement in which the returns are run, and it then becomes necessary to establish a *false* water-line. This is done by making connections as shown in Fig. 26.

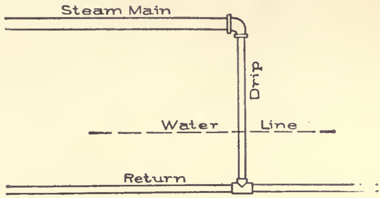


Fig. 24. Drip from Steam Main Connected Directly to Sealed Return.

It is readily seen that the return water, in order to reach the boiler, must flow through the trap, which raises the water-line or seal to the level shown by the dotted line. The balance pipe is to equalize the pressure above and below the water in the trap, and prevent siphonic action, which would tend to drain the water out of the return mains after a flow was once started.

The balance pipe, when possible, should be 15 or 20 feet in length, with a throttle-valve placed near its connection with the main. This valve should be opened just enough to allow the steam-pressure to act upon the air which occupies the space above the water in the trap; but it should not be opened sufficiently to allow the steam to enter in large volume and drive the air out. The success of this arrangement depends upon keeping a layer or cushion of cool air next to the surface of the water in the trap, and this is easily done by following the method here described.

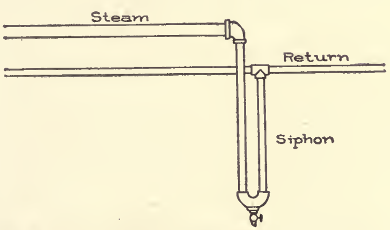


Fig. 25. Use of Siphon in Connecting Drip from Steam Main to a "Dry" Return.

One-Pipe Relief System. In this system of piping, the radiators have but a single connection, the steam flowing in and the condensation draining out through the same pipe. Fig. 27 shows the method of running the pipes for this system. The steam main, as before, leads from the top of the boiler, and is carried to as high a point as the basement ceiling will allow; it then slopes downward with a grade of about 1 inch in 10 feet, and makes a circuit of the building or a portion of it.

Risers are taken from the top and carried to the radiators above, as in the two-pipe system; but in this case, the condensation flows back through the same pipe, and drains into the return main near the floor through drip connections which are made at frequent intervals. In a two-story building, the bottom of each riser to the second floor is dripped; and in larger buildings, it is customary to drip each riser that has more than one radiator connected with it. If the radiators are large and at a considerable distance from the next riser, it is better to make a drip connection for each radiator. When the return-main is overhead, the risers should be dripped through siphon loops; but the ends of the branches should make direct connection with the returns. This is the reverse of the two-pipe system. In this case the lowest pressure is at the ends of the mains, so that steam introduced into the returns at these points will cause no trouble in the pipes connecting between these and the boiler.

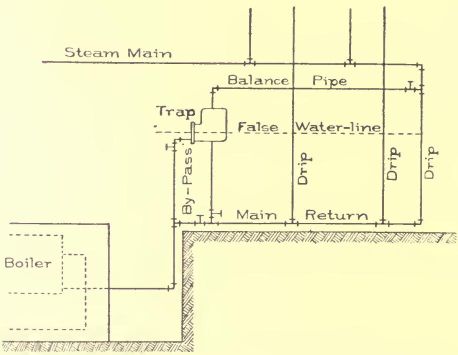


Fig. 26. Connections Made to Establish "False" Water-Line when Boiler is below Basement Level.

If no steam is allowed to enter the returns, a vacuum will be formed, and there will be no pressure to force the water back to the boiler. A check-valve should always be placed in the main return

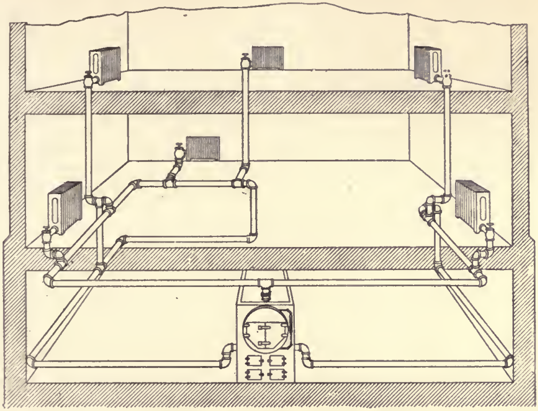


Fig. 27. Arrangement of Piping and Radiators in "One-Pipe Relief" System.

near the boiler, to prevent the water from flowing out in case of a vacuum being formed suddenly in the pipes.

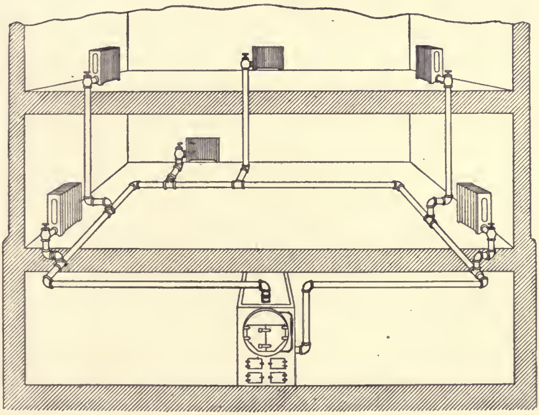


Fig. 28. Arrangement of Piping and Radiators in "One-Pipe Circuit" System.

There is but little difference in the cost of the two systems, as larger pipes and valves are required for the single-pipe method.

With radiators of medium size and properly proportioned connections, the single-pipe system in preferable, there being but one valve to operate and only one-half the number of risers passing through the lower rooms.

One-Pipe Circuit System. In this case, illustrated in Fig. 28, the steam main rises to the highest point of the basement, as before; and then, with a considerable pitch, makes an entire circuit of the building, and again connects with the boiler below the water-line. Single risers are taken from the top; and the condensation drains back through the same pipes, and is carried along with the flow of steam to the extreme end of the main, where it is returned to the boiler. The main is made large, and of the same size throughout its entire length. It must be given a good pitch to insure satisfactory results.

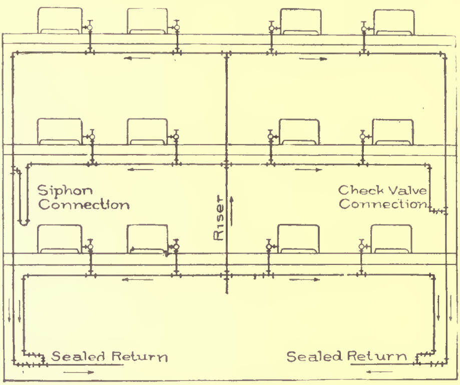


Fig. 29. "One-Pipe Circuit" System. Adapted to a Large Building.

One objection to a single-pipe system is that the steam and return water are flowing in opposite directions, and the risers must be made of extra large size to prevent any interference. This is overcome in large buildings by carrying a single riser to the attic, large enough to supply the entire building; then branching and running "drops" to the basement. In this system the flow of steam is downward, as well as that of water. This method of piping may be used with good results in two-pipe systems as well. Care must always be taken that no pockets or low points occur in any of the lines of pipe; but if for any reason they cannot be avoided, they should be carefully drained.

A modification of this system, adapting it to large buildings, is shown in diagram in Fig. 29. The riser shown in this case is one of

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American Radiator Company.

several, the number depending upon the size of the building; and may be supplied at either bottom or top as most desirable. If steam is supplied at the bottom of the riser, as shown in the cut, all of the drip connections with the return drop, except the upper one, should

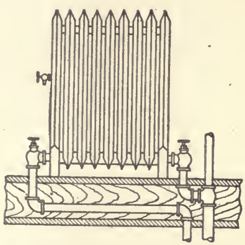


Fig. 30. "Two-Pipe" Connection of Radiator to Riser and Return.

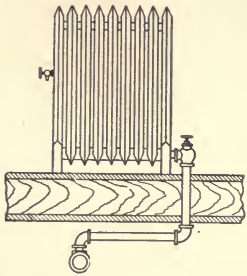


Fig. 31. "One-Pipe" Connection of Radiator to Basement Main.

be sealed with either a siphon loop or a check-valve, to prevent the steam from short-circuiting and holding back the condensation in the returns above. If an overhead supply is used, the arrangement should be the reverse; that is, all return connections should be sealed except the lowest.

Sometimes a separate drip is carried down from each set of radiators, as shown on the lower story, being connected with the main return below the water-line of the boiler. In case this is done, it is well to provide a check-valve in each drip below the water-line.

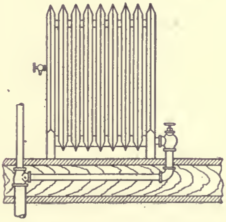


Fig. 32. "One-Pipe" Connection of Radiator to Riser.

In buildings of any considerable size, it is well to divide the piping system into sections by means of valves placed in the corresponding supply and return branches. These are for use in case of a break in any part of the system, so that it will be necessary to shut off only a small part of the heating system during repairs. In tall buildings, it is customary to place valves at the top and bottom of each riser, for the same purpose.

Radiator Connections. Figs. 30, 31, and 32 show the common methods of making connections between supply pipes and radiators. Fig. 30 shows a two-pipe connection with a riser; the return is carried down to the main below. Fig. 31 shows a single-pipe connection with a basement main; and Fig. 32, a single connection with a riser.

Care must always be taken to make the horizontal part of the piping between the radiator and riser as short as possible, and to give it a good pitch toward the riser. There are various ways of making these connections, especially suited to different conditions; but the examples given serve to show the general principle to be followed.

Figs. 20, 21, and 22 show the common methods of making steam and return connections with circulation coils. The position of the air-valve is shown in each case.

Expansion of Pipes. Cold steam pipes expand approximately

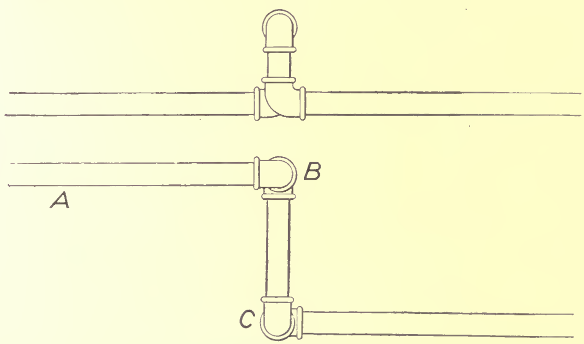


Fig. 33. Elevation and Plan of Swivel-Joint to Counteract Effects of Expansion and Contraction in Pipes.

1 inch in each 100 feet in length when low-pressure steam is turned into them; so that, in laying out a system of piping, we must arrange it in such a manner that there will be sufficient "spring" or "give" to the pipes to prevent injurious strains. This is done by means of offsets and bends. In the case of larger pipes this simple method will not be sufficient, and swivel or slip joints must be used to take up the expansion.

The method of making up a swivel-joint is shown in Fig. 33. Any lengthening of the pipe *A* will be taken up by slight turning or swivel movements at the points *B* and *C*. A slip-joint is shown in

Fig. 34. The part *c* slides inside the shell *d*, and is made steam-tight by a stuffing-box, as shown. The pipes are connected at the flanges *A* and *B*.

When pipes pass through floors or partitions, the woodwork should be protected by galvanized-iron sleeves having a diameter from $\frac{3}{4}$ to 1 inch greater than the pipe. Fig. 35 shows a form of adjustable floor-sleeve which may be lengthened or shortened to conform to the thickness of floor or partition. If plain sleeves are used, a plate should be placed around the pipe where it passes through the floor or partition. These are made in two parts so that they may be put in place after the pipe is hung. A plate of this kind is shown in Fig. 36.

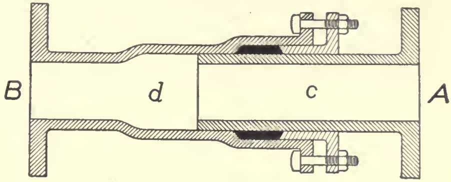


Fig. 34. "Slip-Joint" Connection to Take Care of Expansion and Contraction of Pipes.

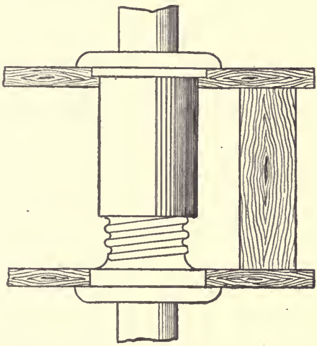


Fig. 35. Adjustable Metal Sleeve for Carrying Pipe through Floor or Partition.

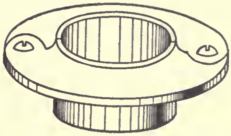


Fig. 36. Floor-Plate Adjusted to Plain Sleeve for Carrying Pipe through Floor or Partition.


Fig. 37. Angle Valve.


Fig. 38. Offset Valve.


Fig. 39. Corner Valve.

Valves for Radiator Connections.

Valves. The different styles commonly used for radiator connections are shown in Figs. 37, 38, and 39, and are known as *angle*, *offset*, and *corner* valves, respectively. The first is used when the radiator is at the top of a riser or when the connections are like those shown in Figs. 30, 31, and 32; the second is used when the connection

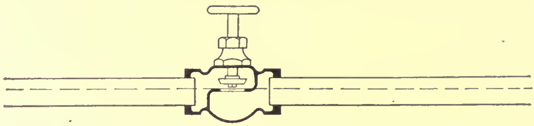


Fig. 40. Indicating Effect of Using Globe Valve on Horizontal Steam Supply Pipe or Dry Return.

between the riser and radiator is above the floor; and the third, when the radiator has to be set close in the corner of a room and there is not space for the usual connection.

A *globe* valve should never be used in a horizontal steam supply or dry return. The reason for this is plainly shown in Fig. 40. In order for water to flow through the valve, it must rise to a height shown by the dotted line, which would half fill the pipes, and cause serious trouble from water-hammer. The gate valve shown in Fig. 41 does not have this undesirable feature, as the opening is on a level with the bottom of the pipe.

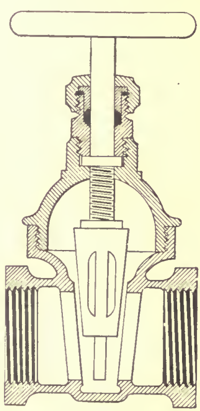


Fig. 41. Gate Valve.

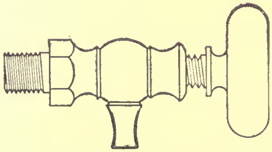


Fig. 42. Simplest Form of Air-Valve. Operated by Hand.

Air-Valves. Valves of various kinds are used for freeing the radiators from air when steam is turned on. Fig. 42 shows the simplest form, which is operated by hand. Fig. 43 is a type of automatic valve, consisting of a shell, which is attached to the radiator. *E* is a small opening which may be closed by the spindle *C*, which is provided with a conical end. D is a strip composed of a layer of iron or steel and one of brass soldered or brazed together. The action of the valve is as follows: when the radiator is cold and filled with air the valve stands as shown in the cut. When steam is turned on, the air is driven out through the opening B. As soon as this is expelled and steam strikes the strip D, the two prongs spring apart owing to the unequal expansion of the two metals due to the heat of the steam. This raises the spindle C, and closes the opening so that no steam can escape. If air should collect in the valve, and the metal strip become cool, it would contract, and the spindle would drop and allow the air to escape through B as before. E is an adjusting nut. F is a float attached to the spindle, and is supposed, in case of a sudden rush of water with the air, to rise and close the opening; this action, however, is somewhat uncertain, especially if the pressure of water continues for some time.

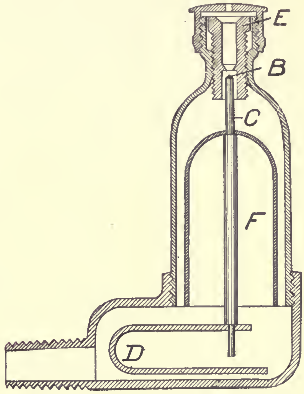


Fig. 43. Radiator Automatic Air-Valve. Operated by Metal Strip D, Consisting of Two Pieces of Metal of Unequal Expansive Power.

There are other types of valves acting on the same principle. The valve shown in Fig. 44 is closed by the expansion of a piece of vulcanite instead of a metal strip, and has no water float.

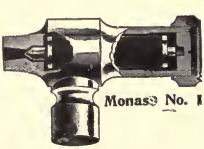


Fig. 44. Automatic Air-Valve. Closed by Expansion of a Piece of Vulcanite.

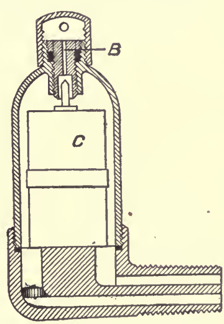


Fig. 45. Automatic Air-Valve. Operated by Expansion of Drum C Due to Vaporization of Alcohol with which it is Partly Filled.

The valve shown in Fig. 45 acts on a somewhat different principle. The float *C* is made of thin brass, closed at top and bottom, and is partially filled with wood alcohol. When steam strikes the float, the alcohol is vaporized, and creates a pressure sufficient to bulge out the ends slightly, which raises the spindle and closes the opening *B*.

Fig. 46 shows a form of so-called *vacuum valve*. It acts in a similar manner to those already described, but has in addition a ball check which prevents the air from being drawn into the radiator, should the steam go down and a vacuum be formed. If a partial vacuum exists in the boiler and radiators, the boiling point, and consequently the temperature of the steam, are lowered, and less heat is given off by the radiators. This method of operating a heating plant is sometimes advocated for spring and fall, when little heat is required, and when steam under pressure would overheat the rooms.

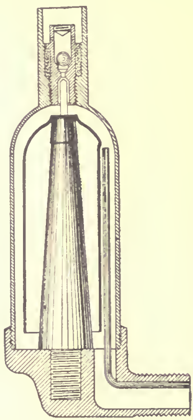


Fig. 46. Vacuum Valve.

Pipe Sizes. The proportioning of the steam pipes in a heating plant is of the greatest importance, and should be carefully worked out by methods which experience has proved to be correct. There are several ways of doing this; but for ordinary conditions, Tables XIV, XV, and XVI have given excellent results in actual practice. They have been computed from what is known as D'Arcy's formula, with suitable corrections made for actual working conditions. As the computations are somewhat complicated, only the results will be given here, with full directions for their proper use.

Table XIV gives the flow of steam in pounds per minute for pipes of different diameters and with varying drops in pressure between the supply and discharge ends of the pipe. These quantities are for pipes 100 feet in length; for other lengths the results must be corrected by the factors given in Table XVI. As the length of pipe increases, friction becomes greater, and the quantity of steam discharged in a given time is diminished.

Table XIV is computed on the assumption that the drop in

TABLE XIV

Flow of Steam in Pipes of Various Sizes, with Various Drops in Pressure between Supply and Discharge Ends

Calculated for 100-Foot Lengths of Pipe

Diam. of Pipe	Drop in Pressure (Pounds)								
	¼	½	¾	1	1½	2	3	4	5
1	.44	.63	.78	91	1.13	1.31	1.66	1.97	2.26
1¼	.81	1.16	1.43	1.66	2.05	2.39	3.02	3.59	4.12
1½	1.06	1.89	2.34	2.71	3.36	3.92	4.94	5.88	6.75
2	2.93	4.17	5.16	5.99	7.43	8.65	10.9	13.0	14.9
2½	5.29	7.52	9.32	10.8	13.4	15.6	19.7	23.4	26.9
3	8.61	12.3	15.2	17.6	21.8	25.4	32	31.8	43.7
3½	12.9	18.3	22.6	26.3	32.5	37.9	47.8	56.9	65.3
4	18 1	25.7	31.8	36.9	45.8	53.3	67.2	80.1	91.9
5	32.2	45.7	56.6	65.7	81.3	94.7	120	142	163
6	51.7	73.3	90.9	106	131	152	192	229	262
7	76.7	109	135	157	194	226	285	339	390
8	108	154	190	222	274	319	402	478	549
9	147	209	258	299	371	432	545	649	745
10	192	273	339	393	487	567	715	852	977
12	305	434	537	623	771	899	1,130	1,350	1,550
15	535	761	942	1,090	1,350	1,580	1,990	2,370	2,720

pressure between the two ends of the pipe equals the initial pressure. If the drop in pressure is less than the initial pressure, the actual discharge will be slightly greater than the quantities given in the table;

TABLE XV

Factors for Calculating Flow of Steam in Pipes under Initial Pressures above Five Pounds

To be used in connection with Table XIV

Drop in Pressure in Pounds	Initial Pressure (Pounds)					
	10	20	30	40	60	80
¼	1.27	1.49	1.68	1.84	2.13	2.38
½	1.26	1.48	1.66	1.83	2.11	2.36
1	1.24	1.46	1.64	1.80	2.08	2.32
2	1.21	1.41	1.59	1.75	2.02	2.26
3	1.17	1.37	1.55	1.70	1.97	2.20
4	1.14	1.34	1.51	1.66	1.92	2.14
5	1.12	1.31	1.47	1.62	1.87	2.09

but this difference will be small for pressures up to 5 pounds, and may be neglected, as it is on the side of safety. For higher initial pressures, Table XV has been prepared. This is to be used in connection with Table XIV as follows: First find from Table XIV the quantity of steam which will be discharged through the given diameter of pipe

TABLE XVI

Factors for Calculating Flow of Steam in Pipes of Other Lengths than 100 Feet

Feet	Factor	Feet	Factor	Feet	Factor	Feet	Factor
10	3.16	120	.91	275	.60	600	.40
20	2.24	130	.87	300	.57	650	.39
30	1.82	140	.84	325	.55	700	.37
40	1.58	150	.81	350	.53	750	.36
50	1.41	160	.79	375	.51	800	.35
60	1.29	170	.76	400	.50	850	.34
70	1.20	180	.74	425	.48	900	.33
80	1.12	190	.72	450	.47	950	.32
90	1.05	200	.70	475	.46	1,000	.31
100	1.00	225	.66	500	.45		
110	.95	250	.63	550	.42		

with the assumed drop in pressure; then look in Table XV for the factor corresponding with the assumed drop and the higher initial pressure to be used. The quantity given in Table XIV, *multiplied by* this factor, will give the actual capacity of the pipe under the given conditions.

Example—What weight of steam will be discharged through a 3-inch pipe 100 feet long, with an initial pressure of 60 pounds and a drop of 2 pounds?

Looking in Table XIV, we find that a 3-inch pipe will discharge 25.4 pounds of steam per minute with a 2-pound drop. Then looking in Table XV, we find the factor corresponding to 60 pounds initial pressure and a drop of 2 pounds to be 2.02. Then according to the rule given, $25.4 \times 2.02 = 51.3$ pounds, which is the capacity of a 3-inch pipe under the assumed conditions.

Sometimes the problem will be presented in the following way: What size of pipe will be required to deliver 80 pounds of steam a distance of 100 feet with an initial pressure of 40 pounds and a drop of 3 pounds?

We have seen that the higher the initial pressure with a given drop, the greater will be the quantity of steam discharged; therefore a smaller pipe will be required to deliver 80 pounds of steam at 40 pounds than at 3 pounds initial pressure From Table XV, we find that a given pipe will discharge 1.7 times as much steam per minute with a pressure of 40 pounds and a drop of 3 pounds, as it would with a pressure of 3 pounds, dropping to zero. From this it is evident that if we divide 80 by 1.7 and look in Table XIV under "3 pounds drop" for the result thus obtained, the size of pipe corresponding will be that required. Now, $80 \div 1.7 = 47$. The nearest number in the table marked "3 pounds drop" is 47.8, which corresponds to a $3\frac{1}{2}$-inch pipe, which is the size required.

These conditions will seldom be met with in low-pressure heating, but apply more particularly to combination power and heating plants, and will be taken up more fully under that head. For lengths of pipe other than 100 feet, multiply the quantities given in Table XIV by the factors found in Table XVI.

Example—What weight of steam will be discharged per minute through a $3\frac{1}{2}$-inch pipe 450 feet long, with a pressure of 5 pounds and a drop of $\frac{1}{2}$ pound?

Table XIV, which may be used for all pressures below 10 pounds, gives for a $3\frac{1}{2}$-inch pipe 100 feet long, a capacity of 18.3 pounds for the above conditions. Looking in Table XVI, we find the correction factor for 450 feet to be .47. Then $18.3 \times .47 = 8.6$ pounds, the quantity of steam which will be discharged if the pipe is 450 feet long.

Examples involving the use of Tables XIV, XV, and XVI in combination, are quite common in practice. The following example will show the method of calculation:

What size of pipe will be required to deliver 90 pounds of steam per minute a distance of 800 feet, with an initial pressure of 80 pounds and a drop of 5 pounds?

Table XVI gives the factor for 800 feet as .35, and Table XV, that for 80 pounds pressure and 5 pounds drop, as 2.09. Then $\frac{90}{.35 \times 2.09} = 123$, which is the equivalent quantity we must look for in Table XIV. We find that a 4-inch pipe will discharge 91.9 pounds, and a 5-inch pipe 163 pounds. A $4\frac{1}{2}$-inch pipe is not commonly carried in stock, and we should probably use a 5-inch in this case, unless it was decided to use a 4-inch and allow a slightly greater drop in pressure. In ordinary heating work, with pressures varying from 2 to 5 pounds, a drop of $\frac{1}{4}$ pound in 100 feet has been found to give satisfactory results.

In computing the pipe sizes for a heating system by the above methods, it would be a long process to work out the size of each branch separately. Accordingly Table XVII has been prepared for ready use in low-pressure work.

As most direct heating systems, and especially those in schoolhouses, are made up of both radiators and circulation coils, an efficiency of 300 B. T. U. has been taken for direct radiation of whatever variety, no distinction being made between the different kinds. This gives a slightly larger pipe than is necessary for cast-iron radiators; but it is probably offset by bends in the pipes, and in any case gives a slight factor of safety. We find from a steam table that the *latent heat* of steam at 20 pounds above a vacuum (which corresponds to 5 pounds' gauge-pressure) is 954 + B. T. U.—which means that, for every pound of steam condensed in a radiator, 954 B. T. U. are given off for warming the air of the room. If a radiator has an efficiency of 300 B. T. U., then each square foot of surface will condense 300 ÷ 954 = .314 pound of steam per hour; so that we may assume in round numbers a condensation of $\frac{1}{3}$ of a pound of steam per hour for each square foot of direct radiation, when computing the sizes of steam pipes in low-pressure heating. Table XVII has been calculated on this assumption, and gives the square feet of heating surface

TABLE XVII

Heating Surface Supplied by Pipes of Various Sizes

Length of Pipe, 100 Feet

Size of Pipe	Square Feet of Heating Surface	
	$\frac{1}{4}$ Pound Drop	$\frac{1}{2}$ Pound Drop
1	80	114
1¼	145	210
1½	190	340
2	525	750
2½	950	1,350
3	1,550	2,210
3½	2,320	3,290
4	3,250	4,620
5	5,800	8,220
6	9,320	13,200
7	13,800	19,620
8	19,440	27,720

which different sizes of pipe will supply, with drops in pressure of $\frac{1}{4}$ and $\frac{1}{2}$ pounds in each 100 feet of pipe. The former should be used for pressures from 1 to 5 pounds, and the latter may be used for pressures over 5 pounds, under ordinary conditions. The sizes of long mains and special pipes of large size should be proportioned directly from Tables XIV, XV, and XVI.

Where the two-pipe system is used and the radiators have separate supply and return pipes, the risers or vertical pipes may be taken from Table XVII; but if the single-pipe system is used, the risers must be increased in size, as the steam and water are flowing in opposite directions and must have plenty of room to pass each other. It is customary in this case to base the computation on the velocity of the steam in the pipes, rather than on the drop in pressure. Assuming, as before, a condensation of one-third of a pound of steam per hour per square foot of radiation, Tables XVIII and XIX have been prepared for velocities of 10 and 15 feet per second. The sizes given in Table XIX have been found sufficient in most cases; but the larger sizes, based on a flow of 10 feet per second, give greater safety and should be more generally used. The size of the largest riser should usually be limited to $2\frac{1}{2}$ inches in school and dwelling-house work, unless it is a special pipe carried up in a concealed position. If the length of riser is short between the lowest radiator and the main, a higher velocity of 20 feet or more may be allowed through this portion, rather than make the pipe excessively large.

TABLE XVIII **TABLE XIX**

Radiating Surface Supplied by Steam Risers

10 Feet per Second Velocity		15 Feet per Second Velocity	
Size of Pipe	Sq. Feet of Radiation	Size of Pipe	Sq. Feet of Radiation
1 in.	30	1 in.	50
$1\frac{1}{4}$ "	60	$1\frac{1}{4}$ "	90
$1\frac{1}{2}$ "	80	$1\frac{1}{2}$ "	120
2 "	130	2 "	200
$2\frac{1}{2}$ "	190	$2\frac{1}{2}$ "	290
3 "	290	3 "	340
$3\frac{1}{2}$ "	390	$3\frac{1}{2}$ "	590

EXAMPLES FOR PRACTICE

1. How many pounds of steam will be delivered per minute, through a $3\frac{1}{2}$-inch pipe 600 feet long, with an initial pressure of 5 pounds and a drop of $\frac{1}{2}$ pound? Ans. 7.32 pounds.

2. What size pipe will be required to deliver 25.52 pounds of steam per minute with an initial pressure of 3 pounds and a drop of $\frac{1}{4}$ pound, the length of the pipe being 50 feet? Ans. 4-inch.

3. Compute the size of pipe required to supply 10,000 square feet of direct radiation (assume $\frac{1}{3}$ of a pound of steam per square

foot per hour) where the distance to the boiler house is 300 feet, and the pressure carried is 10 pounds, allowing a drop in pressure of 4 pounds. Ans. 5-inch (this is slightly larger than is required, while a 4-inch is much too small).

TABLE XX

Sizes of Returns for Steam Pipes (in Inches)

Diameter of Steam Pipe	Diameter of Dry Return	Diameter of Sealed Return
1	1	$\frac{3}{4}$
$1\frac{1}{4}$	1	1
$1\frac{1}{2}$	$1\frac{1}{4}$	1
2	$1\frac{1}{2}$	$1\frac{1}{4}$
$2\frac{1}{2}$	2	$1\frac{1}{2}$
3	$2\frac{1}{2}$	2
$3\frac{1}{2}$	$2\frac{1}{2}$	2
4	3	$2\frac{1}{2}$
5	3	$2\frac{1}{2}$
6	$3\frac{1}{2}$	3
7	$3\frac{1}{2}$	3
8	4	$3\frac{1}{2}$
9	5	$3\frac{1}{2}$
10	5	4
12	6	5

Returns. The size of return pipes is usually a matter of custom and judgment rather than computation. It is a common rule among steamfitters to make the returns one size smaller than the corresponding steam pipes. This is a good rule for the smaller sizes, but gives a larger return than is necessary for the larger sizes of pipe. Table XX gives different sizes of steam pipes with the corresponding diameters for dry and sealed returns.

TABLE XXI

Pipe Sizes for Radiator Connections

Square Feet of Radiation		Steam	Return
Two-Pipe	10 to 30	$\frac{3}{4}$ inch	$\frac{3}{4}$ inch
	30 to 48	1 "	$\frac{3}{4}$ "
	48 to 96	$1\frac{1}{4}$ "	1 "
	96 to 150	$1\frac{1}{2}$ "	$1\frac{1}{4}$ "
Single-Pipe	10 to 24	1 inch	
	24 to 60	$1\frac{1}{4}$ "	
	60 to 80	$1\frac{1}{2}$ "	
	80 to 130	2 "	

The length of run and number of turns in a return pipe should be noted, and any unusual conditions provided for. Where the condensation is discharged through a trap into a lower pressure, the sizes given may be slightly reduced, especially among the larger sizes, depending upon the differences in pressure.

Radiators are usually tapped for pipe connections as shown in Table XXI, and these sizes may be used for the connections with the mains or risers.

Boiler Connections. The steam main should be connected to the rear nozzle, if a tubular boiler is used, as the boiling of the water is less violent at this point and dryer steam will be obtained. The shut-off valve should be placed in such a position that pockets for the accumulation of condensation will be avoided. Fig. 47 shows a good position for the valve.

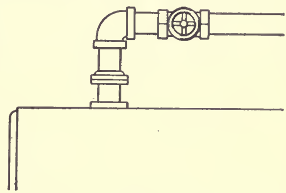


Fig 47. Good Position for Shut-Off Valve.

The size of steam connection may be computed by means of the methods already given, if desired. But for convenience the sizes given in Table XXII may be used with satisfactory results for the short runs between the boilers and main header.

TABLE XXII

Pipe Sizes from Boiler to Main Header

Diameter of Boiler	Size of Steam Pipe
36 inches	3 inches
42 "	4 "
48 "	4 "
54 "	5 "
60 "	5 "
66 "	6 "
72 "	6 "

The return connection is made through the blow-off pipe, and should be arranged so that the boiler can be blown off without draining the returns. A check-valve should be placed in the main return, and a plug-cock in the blow-off pipe. Fig. 48 shows in plan a good arrangement for these connections.

The feed connections, with the exception of that part exposed in the smoke-bonnet, are always made of brass in the best class of work. The small section referred to should be of extra heavy wrought

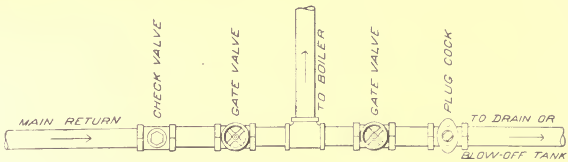


Fig. 48. A Good Arrangement of Return and Blow-Off Connections.

iron. The branch to each boiler should be provided with a gate or globe valve and a check-valve, the former being placed next to the boiler.

Table XXIII gives suitable sizes for return, blow-off, and feed pipes for boilers of different diameters.

TABLE XXIII
Sizes for Return, Blow-Off, and Feed Pipes

Diameter of Boiler	Size of Pipe for Gravity Return	Size of Blow-Off Pipe	Size of Feed Pipe
36 inches	1½ inches	1¼ inches	1 inch
42 "	2 "	1½ "	1 "
48 "	2 "	1½ "	1 "
54 "	2½ "	2 "	1¼ "
60 "	2½ "	2 "	1¼ "
66 "	3 "	2½ "	1½ "
72 "	3 "	2½ "	1½ "

Blow-Off Tank. Where the blow-off pipe connects with a sewer, some means must be provided for cooling the water, or the expansion and contraction caused by the hot water flowing through the drain-pipes will start the joints and cause leaks. For this reason it is customary to pass the water through a blow-off tank. A form of wrought-iron tank is shown in Fig. 49. It consists of a receiver supported on cast-iron cradles. The tank ordinarily stands nearly full of cold water.

The pipe from the boiler enters above the water-line, and the sewer connection leads from near the bottom, as shown. A vapor pipe is carried from the top of the tank above the roof of the building. When water from the boiler is blown into the tank, cold water from the bottom flows into the sewer, and the steam is carried off through the vapor pipe. The equalizing pipe is to prevent any siphon action which might draw the water out of the tank after a flow is once started. As only a part of the water is blown out of a boiler at one time, the blow-off tank can be of a comparatively small size. A tank 24 by 48 inches should be large enough for boilers up to 48 inches in diameter;

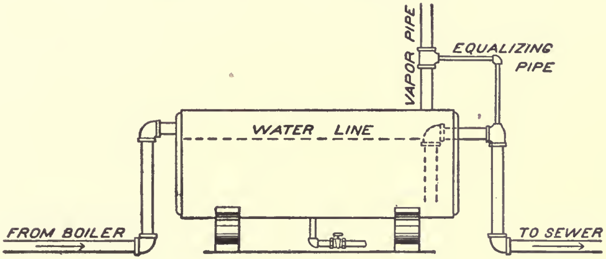
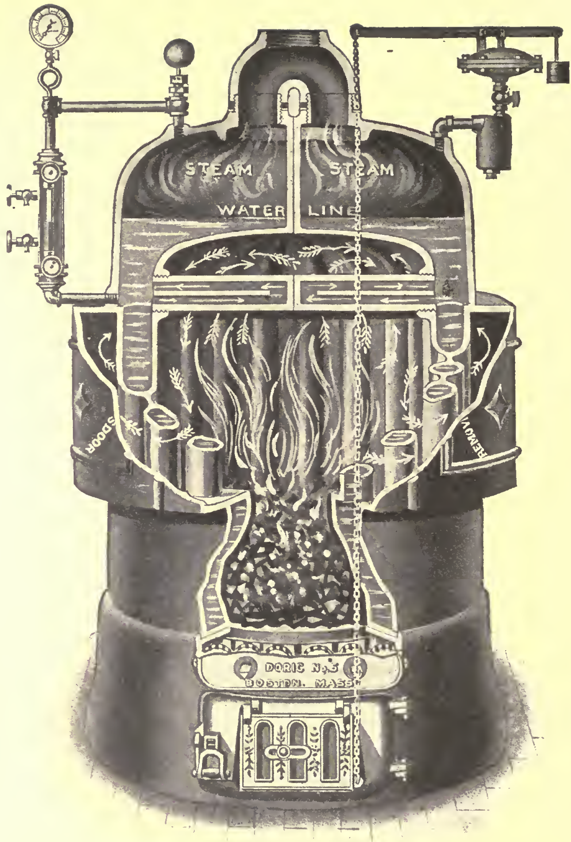


Fig. 49. Connections of Blow-Off Tank.

and one 36 by 72 inches should care for a boiler 72 inches in diameter. If smaller quantities of water are blown off at one time, smaller tanks can be used. The sizes given above are sufficient for batteries of 2 or more boilers, as one boiler can be blown off and the water allowed to cool before a second one is blown off. Cast-iron tanks are often used in place of wrought-iron, and these may be sunk in the ground if desired.



Cast Iron Seamless Tubular Steam Heater.

HEATING AND VENTILATION

PART II

INDIRECT STEAM HEATING

As already stated, in the indirect method of steam heating, a special form of heater is placed beneath the floor, and encased in galvanized iron or in brickwork. A cold-air box is connected with the space beneath the heater; and warm-air pipes at the top are connected with registers in the floors or walls as already described for furnaces. A separate heater may be provided for each register if the rooms are large, or two or more registers may be connected with the same heater if the horizontal runs of pipe are short. Fig. 50 shows a section through a heater arranged for introducing hot air into a room through a floor register; and Fig. 51 shows the same type of heater connected with a wall register. The cold-air box is seen at the bottom of the casing; and the air, in passing through the spaces between the sections of the heater, becomes warmed, and rises to the rooms above.

Different forms of indirect heaters are shown in Figs. 52 and 53. Several sections connected in a single group are called a *stack*. Sometimes the stacks are encased in brickwork built up from the basement floor, instead of in galvanized iron as shown in the cuts. This method of heating provides fresh air for ventilation, and for this reason is especially adapted for schoolhouses, hospitals, churches, etc. As compared with furnace heating, it has the advantage of being less affected by outside wind-pressure, as long runs of horizontal pipe

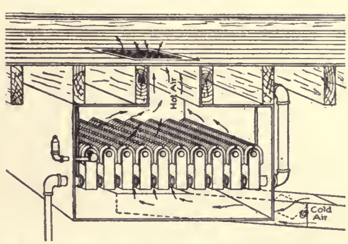


Fig. 50. Steam Heater Placed under Floor Register —Indirect System.

are avoided and the heaters can be placed near the registers. In a large building where several furnaces would be required, a single boiler can be used, and the number of stacks increased to suit the existing conditions, thus making it necessary to run but a single fire. Another advantage is the large ratio between the heating and grate surface as compared with a furnace; and as a result, a large quantity of air is warmed to a moderate temperature, in place of a smaller quantity heated to a much higher temperature. This gives a more agreeable quality to the air, and renders it less dry. Direct and indirect systems are often combined, thus providing the living rooms with ventilation, while the hallways, corridors, etc., have only direct radiators for warming.

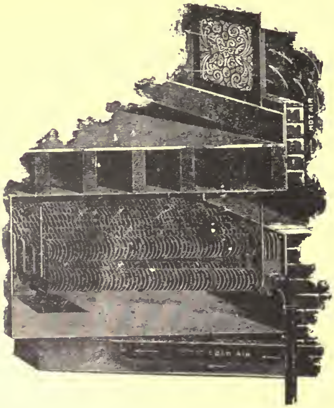


Fig. 51. Steam Heater Connected to Wall Register.—Indirect System.

Types of Heaters. Various forms of indirect radiators are shown in Figs. 52, 53, 54, and 56. A hot-water radiator may be used for steam; but a steam radiator cannot always be used for hot water, as

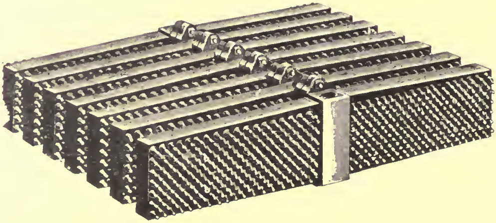


Fig. 52. One Form of Indirect Steam or Hot-Water Heater.

it must be especially designed to produce a continuous flow of water through it from top to bottom. Figs. 54 and 55 show the outside and the interior construction of a common pattern of indirect radiator designed especially for steam. The arrows in Fig. 55 indicate the path of the steam through the radiator, which is supplied at the right, while the return connection is at the left. The air-valve in this case should be connected in the end of the last section near the return.

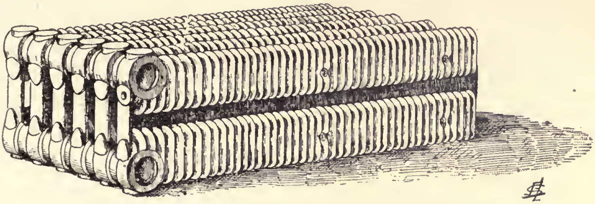


Fig. 53. Another Form of Indirect Steam or Hot-Water Heater.

A very efficient form of radiator, and one that is especially adapted to the warming of large volumes of air, as in schoolhouse work, is shown in Fig. 56, and is known as the *School pin* radiator. This can

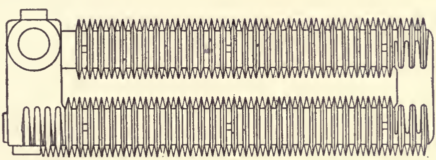


Fig. 54. Exterior View of a Common Type of Radiator for Indirect-Steam Heating.

be used for either steam or hot water, as there is a continuous passage downward from the supply connection at the top to the return at the bottom. These sections or slabs are made up in stacks after the

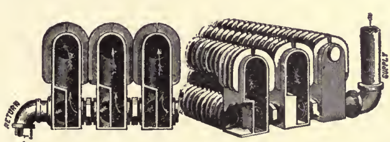


Fig. 55. Interior Mechanism of Radiator Shown in Fig. 54.

manner shown in Fig. 57, which represents an end view of several sections connected together with special nipples.

A very efficient form of indirect heater may be made up of wrought-iron pipe joined together with branch tees and return bends.

A heater like that shown in Fig. 58 is known as a *box coil.* Its efficiency is increased if the pipes are *staggered*—that is, if the pipes in alternate rows are placed over the spaces between those in the row below.

Efficiency of Heaters. The efficiency of an indirect heater

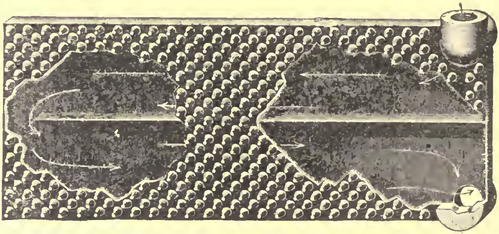


Fig. 56. "School Pin" Radiator, Especially Adapted for Warming Large Volumes of Air by Either Steam or Hot Water.

depends upon its form, the difference in temperature between the steam and the surrounding air, and the velocity with which the air passes over the heater. Under ordinary conditions in dwelling-house work, a good form of indirect radiator will give off about 2 B. T. U. per square foot per hour for each degree difference in temperature between the steam and the entering air. Assuming a steam pressure of 2 pounds and an outside temperature of zero, we should have a difference in temperature of about 220 degrees, which, under the conditions stated, would give an efficiency of $220 \times 2 = 440$ B. T. U. per hour for each square foot of radiation. By making a similar computation for 10 degrees below zero, we find the efficiency to be 460. In the same manner we may calculate the efficiency for varying conditions of steam pressure and outside temperature. In the case of schoolhouses and similar buildings where large volumes of air are warmed to a moderate tem-

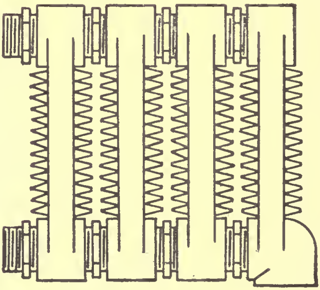


Fig. 57. End View of Several "School Pin" Radiator Sections Connected Together.

perature, a somewhat higher efficiency is obtained, owing to the increased velocity of the air over the heaters. Where efficiencies of 440 and 460 are used for dwellings, we may substitute 600 and 620 for schoolhouses. This corresponds approximately to 2.7 B. T. U. per square foot per hour for a difference of 1 degree between the air and steam.

The principles involved in indirect steam heating are similar to those already described in furnace heating. Part of the heat given off by the radiator must be used in warming up the air-supply to the temperature of the room, and part for offsetting the loss by conduction through walls and windows. The method of computing the heating surface required, depends upon the volume of air to be supplied to the room. In the case of a schoolroom or hall, where the air quantity

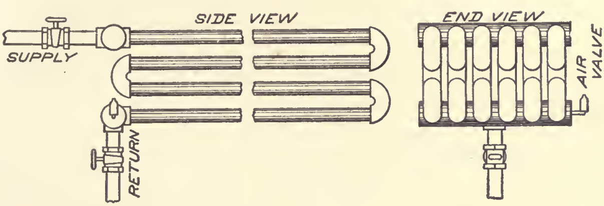


Fig. 58. "Box Coil," Built Up of Wrought-Iron Pipe, for Indirect-Steam Heating.

is large as compared with the exposed wall and window surface, we should proceed as follows:

First compute the B. T. U. required for loss by conduction through walls and windows; and to this, add the B. T. U. required for the necessary ventilation; and divide the sum by the efficiency of the radiators. An example will make this clear.

Example. How many square feet of indirect radiation will be required to warm and ventilate a schoolroom in zero weather, where the heat loss by conduction through walls and windows is 36,000 B. T. U., and the air-supplv is 100,000 cubic feet per hour?

By the methods given under "Heat for Ventilation," we have

$$\frac{100{,}000 \times 70}{55} \times 127{,}272 = \text{B. T. U. required for ventilation.}$$

$$36{,}000 + 127{,}272 = 163{,}272 \text{ B. T. U.} = \text{Total heat required.}$$

This in turn divided by 600 (the efficiency of indirect radiators under these conditions) gives 272 square feet of surface required.

In the case of a dwelling-house the conditions are somewhat changed, for a room having a comparatively large exposure will have perhaps only 2 or 3 occupants, so that, if the small air-quantity necessary in this case were used to convey the required amount of heat to the room, it would have to be raised to an excessively high temperature. It has been found by experience that the radiating surface necessary for indirect heating is about 50 per cent greater than that required for direct heating. So for this work we may compute the surface required for direct radiation, and multiply the result by 1.5.

Buildings like hospitals are in a class between dwellings and schoolhouses. The air-supply is based on the number of occupants, as in schools, but other conditions conform more nearly to dwelling-houses.

To obtain the radiating surface for buildings of this class, we compute the total heat required for warming and ventilation as in the case of schoolhouses, and divide the sum by the efficiencies given for dwellings—that is, 440 for zero weather, and 460 for 10 degrees below.

Example. A hospital ward requires 50,000 cubic feet of air per hour for ventilation; and the heat loss by conduction through walls, etc., is 100,000 B. T. U. per hour. How many square feet of indirect radiation will be required to warm the ward in zero weather?

$50{,}000 \times 70 \div 55 = 63{,}636$ B. T. U. for ventilation; then,

$$\frac{63{,}636 + 100{,}000}{440} = 372 + \text{square feet.}$$

EXAMPLES FOR PRACTICE

1. A schoolroom having 40 pupils is to be warmed and ventilated when it is 10 degrees below zero. If the heat loss by conduction is 30,000 B. T. U. per hour, and the air supply is to be 40 cubic feet per minute per pupil, how many square feet of indirect radiation will be required? Ans. 273.

2. A contagious ward in a hospital has 10 beds, requiring 6,000 cubic feet of air each, per hour. The heat loss by conduction in zero weather is 80,000 B. T. U. How many square feet of indirect radiation will be required? Ans. 355.

3. The heat loss from a sitting room is 11,250 B. T. U. per hour in zero weather. How many square feet of indirect radiation will be required to warm it? Ans. 75.

Stacks and Casings. It has already been stated that a group of sections connected together is called a stack, and examples of these with their casings are shown in Figs. 50 and 51. The casings are usually made of galvanized iron, and are made up in sections by means of small bolts so that they may be taken apart in case it is necessary to make repairs. Large stacks are often enclosed in brickwork, the sides consisting of 8-inch walls, and the top being covered over with a layer of brick and mortar supported on light wrought-iron tee-bars. Blocks of asbestos are sometimes used for covering, instead of brick, the whole being covered over with plastic material of the same kind.

Where a single stack supplies several flues or registers, the connections between these and the warm-air chamber are made in the same manner as already described for furnace heating. When galvanized-iron casings are used, the heater is supported by hangers from the floor above. Fig. 59 shows the method of hanging a heater from a wooden floor. If the floor is of fireproof construction, the hangers may pass up through the brickwork, and the ends be provided with nuts and large washers or plates; or they may be clamped to the iron beams which carry the floor. Where brick casings are used, the heaters are supported upon pieces of pipe or light I-beams built into the walls.

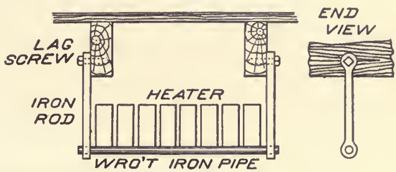


Fig. 59. Method of Hanging a Heater below a Wooden Floor.

The warm-air space above the heater should never be less than 8 inches, while 12 inches is preferable for heaters of large size. The cold-air space may be an inch or two less; but if there is plenty of room, it is good practice to make it the same as the space above.

Dampers. The general arrangement of a galvanized-iron casing and mixing damper is shown in Fig. 60. The cold-air duct is brought along the basement ceiling from the inlet window, and connects with the cold-air chamber beneath the heater. The entering air passes up between the sections, and rises through the register above, as shown by the arrows. When the mixing damper is in its lowest position, all air reaching the register must pass through the heater; but if the damper is raised to the position shown, part of the air will pass by without going through the heater, and the mixture entering through the register will be at a lower temperature than before. By changing

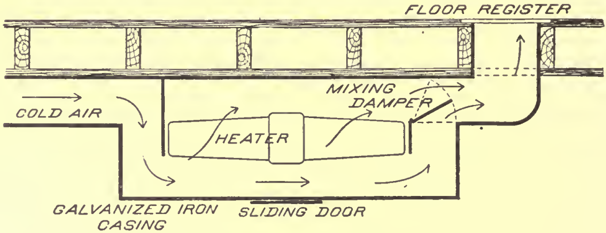


Fig. 60. General Arrangement of a Galvanized-Iron Casing and Mixing Damper. Damper between Heater and Register.

the position of the damper, the proportions of warm and cold air delivered to the room can be varied, thus regulating the temperature without diminishing to any great extent the quantity of air delivered.

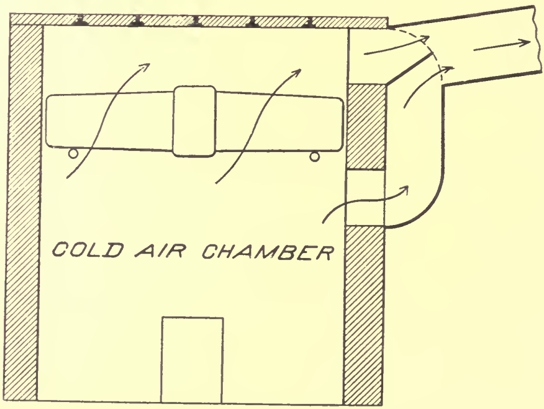


Fig. 61. Heater and Mixing Damper with Brick Casing. Damper between Heater and Register.

The objection to this form of damper is that there is a tendency for the air to enter the room before it is thoroughly mixed; that is, a stream of warm air will rise through one half of the register while cold air enters through the other. This is especially true if the connection between the damper and register is short. Fig. 61 shows a similar heater and mixing damper, with brick casing. Cold air is admitted to the large chamber below the heater, and rises through the sections to the register as before. The action of the mixing damper is the same as already described. Several flues or registers may be connected with a stack of this form, each connection having, in addition to its mixing damper, an adjusting damper for regulating the flow of air to the different rooms.

Another way of proportioning the air-flow in cases of this kind is to divide the hot-air chamber above the heater into sections, by means of galvanized-iron partitions, giving to each room its proper share of heating surface. If the cold-air supply is made sufficiently large, this arrangement is preferable to using adjusting dampers as

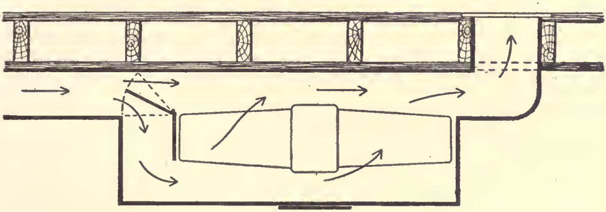


Fig. 62. Another Arrangement of Mixing Damper and Heater in Galvanized-Iron Casing. Heater between Damper and Register.

described above. The partitions should be carried down the full depth of the heater between the sections, to secure the best results.

The arrangement shown in Fig. 62 is somewhat different, and overcomes the objection noted in connection with Fig. 60, by substituting another. The mixing damper in this case is placed at the other end of the heater. When it is in its highest position, all of the air must pass through the heater before reaching the register; but when partially lowered, a part of the air passes over the heater, and the result is a mixture of cold and warm air, in proportions depending upon the position of the damper. As the layer of warm air in this case is below the cold air, it tends to rise through it, and a more thorough mixture is obtained than is possible with the damper shown in Fig. 60. One quite serious objection, however, to this form of damper, is illustrated in Fig. 63. When the damper is nearly closed so that the greater part of the air enters above the heater, it has a tendency to fall between the sections, as shown by the arrows, and, becoming heated, rises again, so that it is impossible to deliver air to a room below a certain temperature. This peculiar action increases as the quantity of air admitted below the heater is diminished. When the inlet register is placed in the wall at some distance above the floor, as in schoolhouse work, a thorough mixture of air can be obtained by placing the heater so that the current of warm air will pass up the front of the flue and be discharged into the room through the lower part of the register. This is shown quite clearly in Fig. 64, where the current of warm air is represented by crooked arrows, and the cold air by straight arrows. The two currents pass up the flue separately; but as soon as they are discharged through the register the warm air tends to rise, and the cold air to fall, with the result of a more or less complete mixture, as shown.

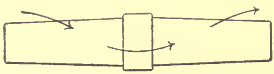


Fig. 63. Showing Difficulty of Regulating Temperature with Arrangement in Fig. 62.

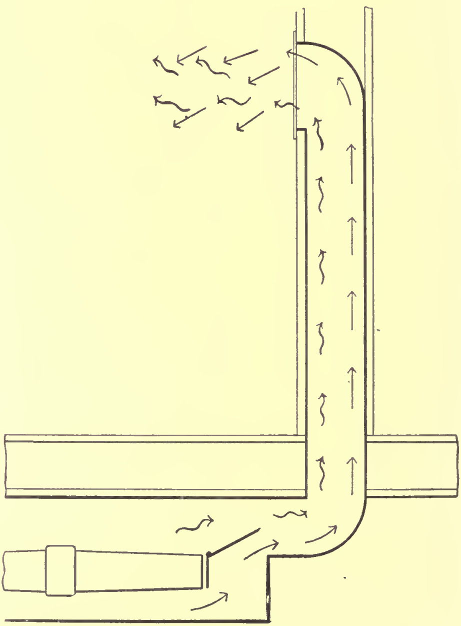


Fig. 64. Arrangement of Heater and Damper Causing Warm Air to Enter Room through Lower Part of Register, thus Securing Thorough Mixing

It is often desirable to warm a room at times when ventilation is not necessary, as in the case of living rooms during the night, or for quick warming in the morning. A register and damper for air rotation should be provided in this case. Fig. 65 shows an arrangement for this purpose. When the damper is in the position shown, air will be taken from the room above and be warmed over and over; but, by raising the damper, the supply will be taken from outside. Special care should be taken to make all mixing dampers tight against air-leakage, else their advantages will be lost. They should work easily and close tightly against flanges covered with felt. They may be operated from the rooms above by means of chains passing over

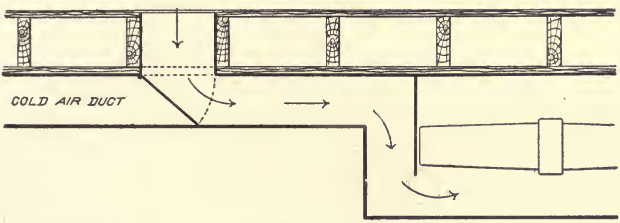


Fig. 65. Arrangement for Quick Heating without Ventilation. Damper Shuts off Fresh Air, and Air of Room Heated by Rotating Forth and Back through Register and Heater.

guide-pulleys; special attachments should be provided for holding in any desired position.

Warm-Air Flues. The required size of the warm-air flue between the heater and the register, depends first upon the difference in temperature between the air in the flue and that of the room, and second, upon the height of the flue. In dwelling-houses, where the conditions are practically constant, it is customary to allow 2 square inches area for each square foot of radiation when the room is on the first floor, and $1\frac{1}{2}$ square inches for the second and third floors. In the case of hospitals, where a greater volume of air is required, these figures may be increased to 3 square inches for the first floor wards, and 2 square inches for those on the upper floors.

In schoolhouse work, it is more usual to calculate the size of flue from an assumed velocity of air-flow through it. This will vary greatly according to the outside temperature and the prevailing wind conditions. The following figures may be taken as average velocities obtained in practice, and may be used as a basis for calculating the required flue areas for the different stories of a school building:

1st floor, 280 feet per minute.
2nd " , 340 " " "
3rd " , 400 " " "

These velocities will be increased somewhat in cold and windy weather and will be reduced when the atmosphere is mild and damp.

Having assumed these velocities, and knowing the number of cubic feet of air to be delivered to the room per minute, we have only to divide this quanity by the assumed velocity, to obtain the required flue area in square feet.

Example. A schoolroom on the second floor is to have an air-supply of 2,000 cubic feet per minute. What will be the required flue area?

ANS. $2000 \div 340 = 5.8 +$ sq. feet.

The velocities would be higher in the coldest weather, and dampers should be placed in the flues for throttling the air-supply when necessary.

Cold-Air Ducts. The cold-air ducts supplying heaters should be planned in a manner similar to that described for furnace heating. The air-inlet should be on the north or west side of the building; but this of course is not always possible. The method of having a large trunk line or duct with inlets on two or more sides of the building, should be carried out when possible. A cold-air room with large inlet windows, and ducts connecting with the heaters, makes a good arrangement for schoolhouse work. The inlet windows in this case should be provided with check-valves to prevent any outward flow of air. A detail of this arrangement is shown in Fig. 66.

This consists of a boxing around the window, extending from the floor to the ceiling. The front is sloped as shown, and is closed from the ceiling to a point below the bottom of the window. The remainder is open, and covered with a wire netting of about ½-inch mesh; to this are fastened flaps or checks of gossamer cloth about 6 inches in width. These are hemmed on both edges and a stout wire is run through the upper hem which is fastened to the netting by means of small copper or soft iron wire. The checks allow the air to flow inward but close when there is any tendency for the current to reverse.

The area of the cold-air duct for any heater should be about three-fourths the total area of the warm-air ducts leading from it.

If the duct is of any considerable length or contains sharp bends, it should be made the full size of all the warm-air ducts. Adjusting dampers should be placed in the supply duct to each separate stack. If a trunk with two inlets is used, each inlet should be of sufficient size to furnish the full amount of air required, and should be provided with cloth checks for preventing an outward flow of air, as already described. The inlet windows should be provided with some form of damper or slide, outside of which should be placed a wire grating, backed by a netting of about $\frac{3}{8}$-inch mesh.

Vent Flues. In dwelling-houses, vent flues are often omitted, and the frequent opening of doors and leakage are depended upon to carry away the impure air. A well-designed system of warming should provide some means for discharge ventilation, especially for bathrooms and toilet-rooms, and also for living rooms where lights are burned in the evening. Fireplaces are usually provided in the more important rooms of a well-built house, and these are made to serve as vent flues. In rooms having no fireplaces, special flues of tin or galvanized iron may be carried up in the partitions in the same manner as the warm-air flues. These should be gathered together in the attic, and connected with a brick flue running up beside the boiler or range chimney.

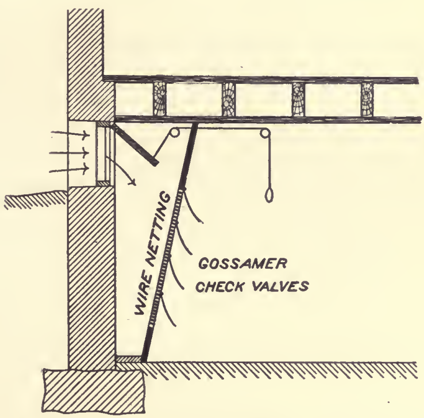


Fig. 66. Air-Inlet Provided with Check-Valves to Prevent Outward Flow of Air.

Very fair results may be obtained by simply letting the flues open into an unfinished attic, and depending upon leakage through the roof to carry away the foul air.

The sizes of flues may be made the reverse of the warm-air flues —that is, $1\frac{1}{2}$ square inches area per square foot of indirect radiation for rooms on the first floor, and 2 square inches for those on the second. This is because the velocity of flow will depend upon the height of flue, and will therefore be greater from the first floor. The flow of air through the vents will be slow at best, unless some means is provided for warming the air in the flue to a temperature above that of the room with which it connects.

The method of carrying up the outboard discharge beside a warm chimney is usually sufficient in dwelling-houses; but when it is desired to move larger quantities of air, a loop of steam pipe should be run inside the flue. This should be connected for drainage and air-venting as shown in Fig. 67. When vents are carried through the roof independently, some form of protecting hood should be provided for keeping out the snow and rain. A simple form is shown in Fig. 68. Flues carried outboard in this way should always be extended well above the ridges of adjacent roofs to prevent down drafts in windy weather.

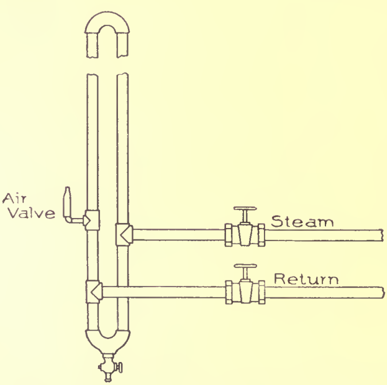


Fig. 67. Loop of Steam Pipe to be Run Inside Flue. Connected for Drainage and Air-Venting.

For schoolhouse work we may assume average velocities through the vent flues, as follows:

1st floor,	340 feet	per	minute.
2nd " ,	280 "	"	"
3rd " ,	220 "	"	"

Where flue sizes are based on these velocities, it is well to guard against down drafts by placing an aspirating coil in the flue. A single row of pipes across the flue as shown in Fig. 69, is usually sufficient for this purpose when the flues are large and straight;

otherwise, two rows should be provided. The slant height of the heater should be about twice the depth of the flue, so that the area between the pipes shall equal the free area of the flue.

Large vent flues of this kind should always be provided with dampers for closing at night, and for regulation during strong winds.

Sometimes it is desired to move a given quantity of air through a flue which is already in place. Table XXIV shows what velocities may be obtained through flues of different heights, for varying differences in temperature between the outside air and that in the flue.

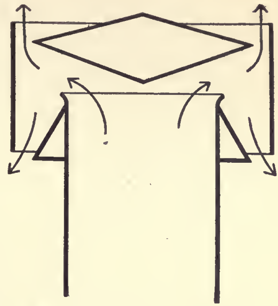


Fig. 68. Section Showing Simple Form of Protecting Hood for Vent Carried through Roof.

Example.—It is desired to discharge 1,300 cubic feet of air per minute through a flue having an area of 4 square feet and a height of 30 feet. If the efficiency of an aspirating coil is 400 B. T. U., how many square feet of surface will be required to move this amount of air when the temperature of the room is 70° and the outside temperature is 60°?

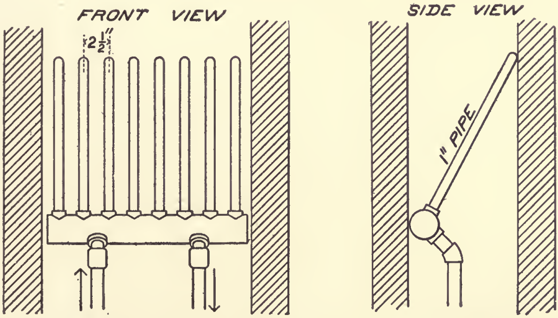


Fig. 69. Aspirating Coil Placed in Flue to Prevent Down Drafts.

1,300 ÷ 4 = 325 feet per minute = Velocity through the flue. Looking in Table XXIV, and following along the line opposite a 30-foot flue, we find that to obtain this velocity there must be a difference of 30 degrees between the air in the flue and the external air.

If the outside temperature is 60 degrees, then the air in the flue must be raised to 60 + 30 = 90 degrees. The air of the room being at 70 degrees, a rise of 20 degrees is necessary. So the problem resolves itself into the following: What amount of heating surface having an

TABLE XXIV

Air-Flow through Flues of Various Heights under Varying Conditions of Temperature

(Volumes given in cubic feet per square foot of sectional area of flue)

Height of Flue in Feet	Excess of Temperature of Air in Flue Above that of External Air					
	5°	10°	15°	20°	30°	50°
5	55	76	94	109	134	167
10	77	108	133	153	188	242
15	94	133	162	188	230	297
20	108	153	188	217	265	342
25	121	171	210	242	297	383
30	133	188	230	265	325	419
35	143	203	248	286	351	453
40	153	217	265	306	375	484
45	162	230	282	325	398	514
50	171	242	297	342	419	541
60	188	264	325	373	461	594

efficiency of 400 B. T. U. is necessary to raise 1,300 cubic feet of air per minute through 20 degrees?

1,300 cubic feet per minute = 1,300 × 60 = 78,000 per hour; and making use of our formula for "heat for ventilation," we have

$$\frac{78{,}000 \times 20}{55} = 28{,}363 \text{ B. T. U.};$$

and this divided by 400 = 71 square feet of heating surface required.

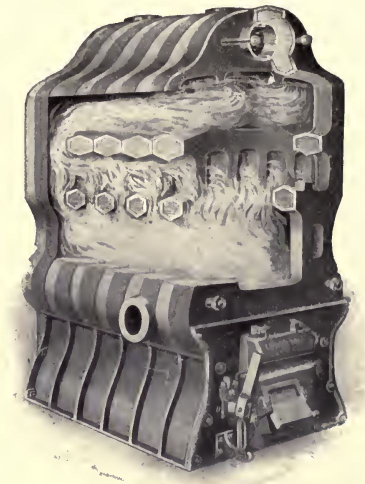
EXAMPLES FOR PRACTICE

1. A schoolroom on the third floor has 50 pupils, who are to be furnished with 30 cubic feet of air per minute each. What will be the required areas in square feet of the supply and vent flues?

Ans. Supply, 3.7 +. Vent, 6.8 +.

2. What size of heater will be required in a vent flue 40 feet high and with an area of 5 square feet, to enable it to discharge 1,530 cubic feet per minute, when the outside temperature is 60°? (Assume an efficiency of 400 B. T. U. for the heater.) Ans. 41.7 square feet.

SECTIONAL STEAM BOILER.



SECTIONAL WATER BOILER.

Made by American Radiator Company.

Registers. Registers are made of cast iron and bronze, in a great variety of sizes and patterns. The almost universal finish for cast-iron registers is black "Japan;" but they are also finished in colors and electroplated with copper and nickel. Fig. 70 shows a section through a floor register, in which *A* represents the valves, which may be turned in a vertical or horizontal position, thus opening or closing the register; *B* is the iron border; *C*, the register box of tin or galvanized iron; and *D*, the warm-air pipe. Floor registers are usually set in cast-iron borders, one of which is shown in Fig. 71; while wall registers may be screwed directly to wooden borders or frames to correspond with the finish of the room. Wall registers should be provided with pull-cords for opening and closing from the floor; these are shown in Fig. 72. The plain lattice pattern shown in Fig. 73 is the best for schoolhouse work, as it has a comparatively free opening for air-flow and is pleasing and simple in design. More elaborate patterns are used for fine dwelling-house work. Registers with shut-off valves are used for air-inlets, while the plain register faces without the valves are placed in the vent openings. The vent flues are usually gathered together in the attic, and a single damper may be used to shut off the whole number at once. Flat or round wire gratings of open pattern are often used in place of

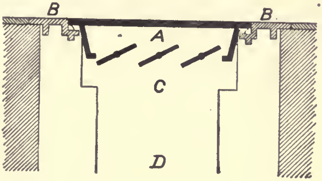


Fig. 70. Section through a Floor Register.

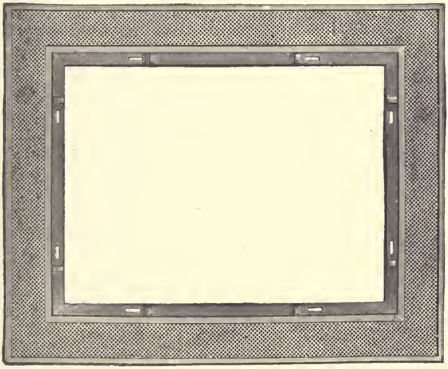


Fig. 71. Cast-Iron Border for a Floor Register.

register faces. The grill or solid part of a register face usually takes up about $\frac{1}{3}$ of the area; hence in computing the size, we must allow for this by multiplying the required "net area" by 1.5, to obtain the "total" or "over-all" area.

Example. Suppose we have a flue 10 inches in width and wish to use a register having a free area of 200 square inches. What will be the required height of the register?

$200 \times 1.5 = 300$ square inches, which is the total area required; then $300 \div 10 = 30$, which is the required height, and we should use a 10 by 30-inch register. When a register is spoken of as a 10 by

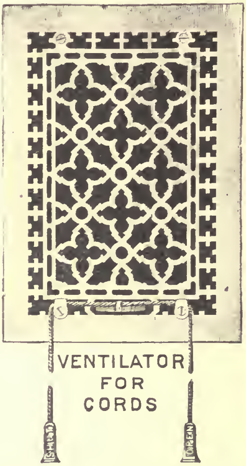


Fig. 72. Wall Register with Pull Cords for Opening and Closing.

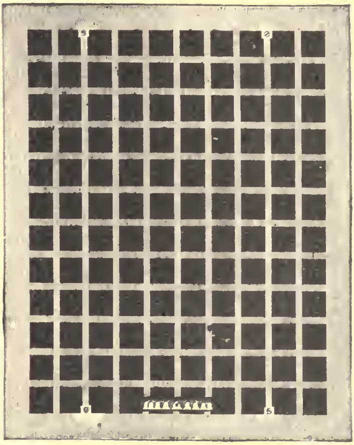


Fig. 73. Plain Lattice Pattern Register. Best for Schoolhouse Work.

30-inch or a 10 by 20-inch, etc., the dimensions of the latticed opening are meant, and not the outside dimensions of the whole register. The free opening should have the same area as the flue with which it connects. In designing new work, one should provide himself with a trade catalogue, and use only standard sizes, as special patterns and sizes are costly. Fig. 74 shows the method of placing gossamer check-valves back of the vent register faces to prevent down drafts, the same as described for fresh-air inlets.

Inlet registers in dwelling-house and similar work are placed either in the floor or in the baseboard; sometimes they are located under the windows, just above the baseboard. The object in view is to place them where the currents of air entering the room will not be objectionable to persons sitting near windows. A long, narrow floor-register placed close to the wall in front of a window, sends up a shallow current of warm air, which is not especially noticeable

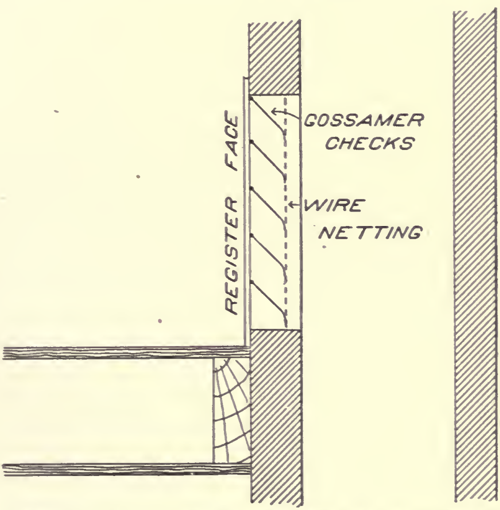


Fig. 74. Method of Placing Gossamer Check-Valves back of Vent Register Face to Prevent Down Drafts.

to one sitting near it. Inlet registers are preferably placed near outside walls, especially in large rooms. Vent registers should be placed in inside walls, near the floor.

Pipe Connections. The two-pipe system with dry or sealed returns is used in indirect heating. The conditions to be met are practically the same as in direct heating, the only difference being that the radiators are at the basement ceiling instead of on the floors above. The exact method of making the pipe connections will depend somewhat upon existing conditions; but the general method shown in Fig. 75 may be used as a guide, with modifications to suit any special case. The ends of all supply mains should be dripped, and the horizontal returns should be sealed if possible.

Pipe Sizes. The tables already given for the proportioning of pipe sizes can be used for indirect systems. The following table has been computed for an efficiency of 640 B. T. U. per square foot of surface per hour, which corresponds to a condensation of $\frac{2}{3}$ of a pound of steam. This is twice that allowed for direct radiation in Table

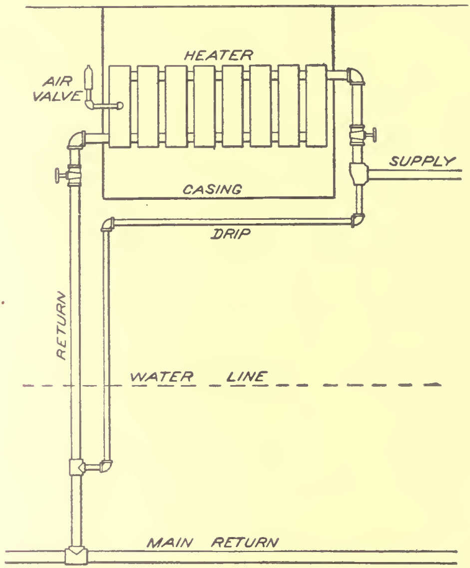


Fig. 75. General Method of Making Pipe and Radiator Connections, in Basement, in Indirect Heating.

XVII; so that we can consider 1 square foot of indirect surface as equal to 2 of direct in computing pipe sizes.

As the indirect heaters are placed in the basement, care must be taken that the bottom of the radiator does not come too near the water-line of the boiler, or the condensation will not flow back properly; this distance, under ordinary conditions, should not be less than 2 feet. If much less than this, the pipes should be made extra large, so that there may be little or no drop in pressure between the boiler

TABLE XXV

Indirect Radiating Surface Supplied by Pipes of Various Sizes

Size of Pipe	Square Feet of Indirect Radiation which will be Supplied with		
	¼ Pound Drop in 200 Feet	¼ Pound Drop in 100 Feet	½ Pound Drop in 100 Feet
1 in.	28	40	57
1¼ "	51	72	105
1½ "	67	95	170
2 "	185	262	375
2½ "	335	475	675
3 "	540	775	1, 105
3½ "	812	1, 160	1, 645
4 "	1, 140	1, 625	2, 310
5 "	2, 030	2, 900	4, 110
6 "	3, 260	4, 660	6, 600
7 "	4, 830	6, 900	9, 810
8 "	6, 800	9, 720	13, 860

and the heater. A drop in pressure of 1 pound would raise the water-line at the heater 2.4 feet.

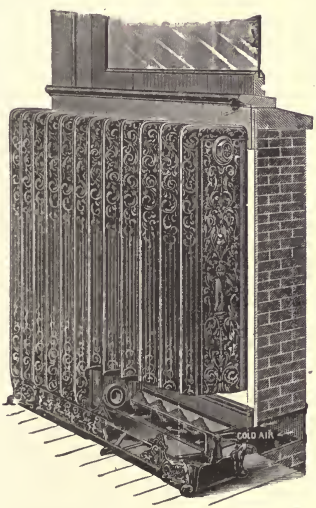


Fig. 76. General Form of Direct-Indirect Radiator.

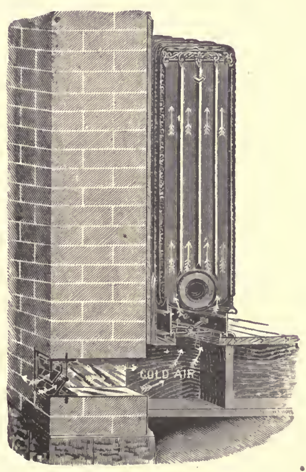


Fig. 77. Section through Radiator Shown in Fig. 76.

Direct-Indirect Radiators. A direct-indirect radiator is similar in form to a direct radiator, and is placed in a room in the same manner. Fig. 76 shows the general form of this type of radiator; and Fig. 77 shows a section through the same. The shape of the sections is such, that when in place, small flues are formed between them. Air is admitted through an opening in the outside wall; and, in passing upward through these flues, becomes heated before entering the room. A switch-damper is placed in the duct at the base of the radiator, so that the air may be taken from the room itself instead of from out of doors, if so desired. This is shown more particularly in Fig. 76.

Fig. 78 shows the wall box provided with louvre slats and netting, through which the air is drawn. A damper door is placed at either end of the radiator base; and, if desired, when the cold-air supply is shut off by means of the register in the air-duct, the radiator can be converted into the ordinary type by opening both damper doors, thus taking the air from the room instead of from the outside. It is customary to increase the size of a direct-indirect radiator 30 per cent above that called for in the case of direct heating.

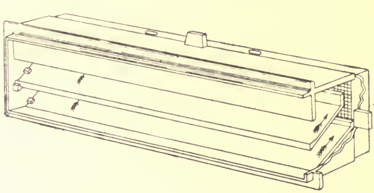


Fig. 78. Wall Box with Louvre Slats and Netting. Direct-Indirect System.

CARE AND MANAGEMENT OF STEAM-HEATING BOILERS

Special directions are usually supplied by the maker for each kind of boiler, or for those which are to be managed in any peculiar way. The following general directions apply to all makes, and may be used regardless of the type of boiler employed:

Before starting the fire, see that the boiler contains sufficient water. The water-line should be at about the center of the gauge-glass.

The smoke-pipe and chimney flue should be clean, and the draft good.

Build the fire in the usual way, using a quality of coal which is best adapted to the heater. In operating the fire, keep the firepot full of coal, and shake down and remove all ashes and cinders as often as the state of the fire requires it.

Hot ashes or cinders must not be allowed to remain in the ashpit under the grate-bars, but must be removed at regular intervals to prevent burning out the grate.

To control the fire, see that the damper regulator is properly attached to the draft doors and the damper; then regulate the draft by weighting the automatic lever as may be required to obtain the necessary steam pressure for warming. Should the water in the boiler escape by means of a broken gauge-glass, or from any other cause, the fire should be dumped, and the boiler allowed to cool before adding cold water.

An empty boiler should never be filled when hot. If the water gets low at any time, but still shows in the gauge-glass, more water should be added by the means provided for this purpose.

The safety-valve should be lifted occasionally to see that it is in working order.

If the boiler is used in connection with a gravity system, it should be cleaned each year by filling with pure water and emptying through the blow-off. If it should become foul or dirty, it can be thoroughly cleansed by adding a few pounds of caustic soda, and allowing it to stand for a day, and then emptying and thoroughly rinsing.

During the summer months, it is recommended that the water be drawn off from the system, and that air-valves and safety-valves be opened to permit the heater to dry out and to remain so. Good results, however, are obtained by filling the heater full of water, driving off the air by boiling slowly, and allowing it to remain in this condition until needed in the fall. The water should then be drawn off and fresh water added.

The heating surface of the boiler should be kept clean and free from ashes and soot by means of a brush made especially for this purpose.

Should any of the rooms fail to heat, examine the steam valves in the radiators. If a two-pipe system, both valves at each radiator must be opened or closed at the same time, as required. See that the air-valves are in working condition.

If the building is to be unoccupied in cold weather, draw all the water out of the system by opening the blow-off pipe at the boiler and all steam valves and air-valves at the radiators.

HOT-WATER HEATERS

Types. Hot-water heaters differ from steam boilers principally in the omission of the reservoir or space for steam above the heating surface. The steam boiler might answer as a heater for hot water; but the large capacity left for the steam would tend to make its operation slow and rather unsatisfactory, although the same type of boiler is sometimes used for both steam and hot water. The passages in a hot-water heater need not extend so directly from bottom to top as in a steam boiler, since the problem of providing for the free liberation of the steam bubbles does not have to be considered. In general, the heat from the furnace should strike the surfaces in such a manner as to increase the natural circulation; this may be accomplished to a certain extent by arranging the heating surface so that a large proportion of the direct heat will be absorbed near the top of the heater.

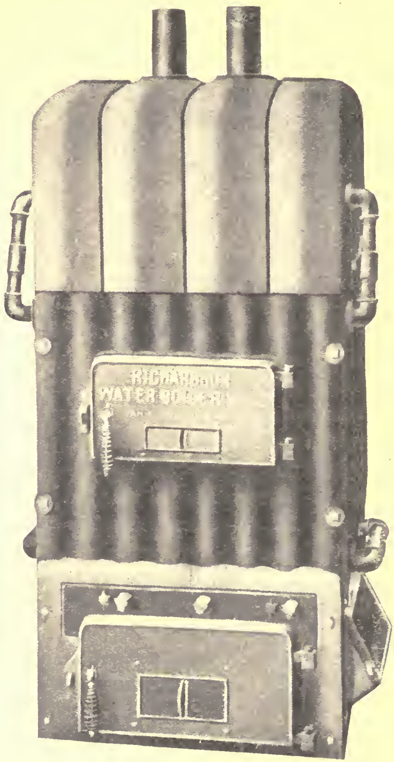


Fig. 79. Richardson Sectional Hot-Water Heater.

Practically the boilers for low-pressure steam and for hot water differ from each other very little as to the character of the heating surface, so that the methods already given for computing the size of grate surface, horse-power, etc., under the head of "Steam Boilers," can be used with satisfactory results in the case of hot-water heaters.

It is sometimes stated that, owing to the greater difference in temperature between the furnace gases and the water in a hot-water heater, as compared with steam, the heating surface will be more efficient and a smaller heater can be used. While this is true to a certain extent, different authorities agree that this advantage is so small that no account should be taken of it, and the general proportions of the heater should be calculated in the same manner as for steam. Fig. 79 shows a form of hot-water heater made up of slabs or sections similar to the sectional steam boiler shown in Part I. The size can be increased in a similar manner, by adding more sections. In this case, however, the boiler is increased in width instead of in length. This has an advantage in the larger sizes, as a second fire door can be added, and all parts of the grate can be reached as well in the large sizes as in the small.

Fig. 80 shows a different form of sectional boiler, in which the sections are placed one above another. These boilers are circular in form and well adapted to dwelling-houses and similar work.

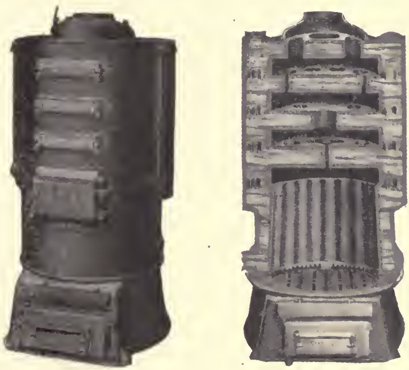


Fig. 80. "Invincible" Boiler, with Sections Superposed.
Courtesy of American Radiator Co.

Fig. 81 shows another type of cast-iron heater which is not made in sections. The space between the outer and inner shells surrounding the furnace is filled with water, and also the cross-pipes directly over the fire and the drum at the top. The supply to the radiators is taken off from the top of the heater, and the return connects at the lowest point.

The ordinary horizontal and vertical tubular boilers, with various modifications, are used to a considerable extent for hot-water heating, and are well adapted to this class of work, especially in the case of large buildings.

Automatic regulators are often used for the purpose of maintaining a constant temperature of the water. They are constructed in different ways—some depend upon the expansion of a metal pipe or rod at different temperatures, and others upon the vaporization and consequent pressure of certain volatile liquids. These means are usually employed to open small valves which admit water-pressure under rubber diaphragms; and these in turn are connected by means of chains with the draft doors of the furnace, and so regulate the draft as required to maintain an even temperature of the water in the heater. Fig. 82 shows one of the first kind. *A* is a metal rod placed in the flow pipe from the heater, and is so connected with the valve *B* that when the water reaches a certain temperature the expansion of the rod opens the valve and admits water from the street pressure through the pipes *C* and *D* into the chamber *E*. The bottom of *E* consists of a rubber diaphragm, which is forced down by the water-pressure and carries with it the lever which operates the dampers as shown, and checks the fire. When the temperature of the water drops, the rod contracts and valve *B* closes, shutting off the pressure from the chamber *E*. A spring is provided to throw the lever back to its original position,

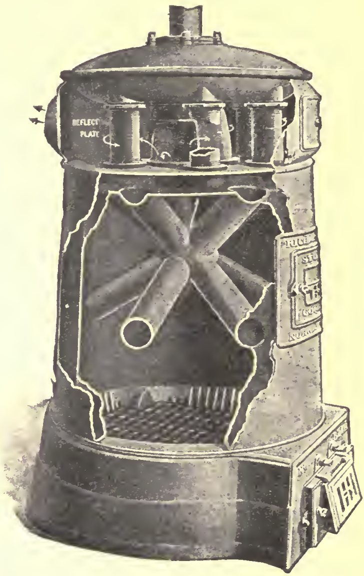


Fig. 81 Cast-Iron Heater Not Made in Sections. Water Fills Cross-Pipes and Space between Outer and Inner Shells.

and the water above the diaphragm is forced out through the pet-cock *G*, which is kept slightly open all the time.

DIRECT HOT-WATER HEATING

A hot-water system is similar in construction and operation to one designed for steam, except that *hot water* flows through the pipes and radiators instead.

The circulation through the pipes is produced solely by the difference in weight of the water in the supply and return, due to the difference in temperature. When water is heated it expands, and thus a given volume becomes lighter and tends to rise, and the cooler water flows in to take its place; if the application of heat is kept up, the circulation thus produced is continuous. The velocity of flow depends upon the difference in temperature between the supply and return, and the height of the radiator above the boiler. The horizontal distance of the radiator from the boiler is also an important factor affecting the velocity of flow.

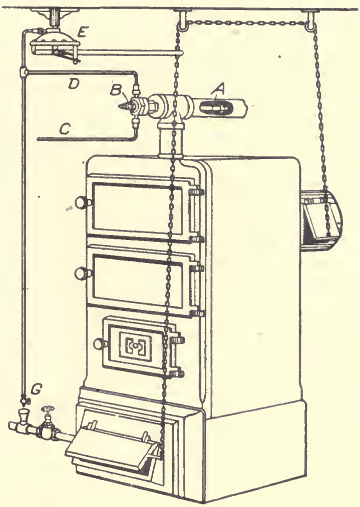


Fig. 82. Hot-Water Heater with Automatic Regulator Operated through Expansion and Contraction of Metal Rod in Flow Pipe.

This action is best shown by means of a diagram, as in Fig. 83. If a glass tube of the form shown in the figure is filled with water and held in a vertical position, no movement of the water will be noticed, because the two columns *A* and *B* are of the same weight, and therefore in equilibrium. Now, if a lamp flame be held near the tube *A*, the small bubbles of steam which are formed will show the water to be in motion, with a current flowing in the direction indicated by the arrows. The reason for this is, that, as the water in *A* is heated, it expands and becomes lighter for a given volume, and is forced upward by the heavier water in B falling to the bottom of the tube. The heated water flows from A through the connecting tube at the top, into B, where it takes the place of the cooler water which is settling to the bottom. If, now, the lamp be replaced by a furnace, and the columns A and B be connected at the top by inserting a radiator, the illustration will assume the practical form as utilized in hot-water heating (see Fig. 84).

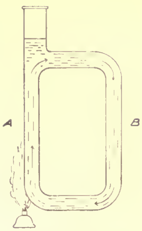


Fig. 83. Illustrating How the Heating of Water Causes Circulation.

The heat given off by the radiator always insures a difference in temperature between the columns of water in the supply and return pipes, so that as long as heat is supplied by the furnace the flow of water will continue. The greater the difference in temperature of the water in the two pipes, the greater the difference in weight, and consequently the faster the flow. The greater the height of the radiator above the heater, the more rapid will be the circulation, because the total difference in weight between the water in the supply and return risers will vary directly with their height. From the above it is evident that the rapidity of flow depends chiefly upon the *temperature* difference between the supply and return, and upon the *height of the radiator* above the heater. Another factor which must be considered in long runs of horizontal pipe is the *frictional resistance.*

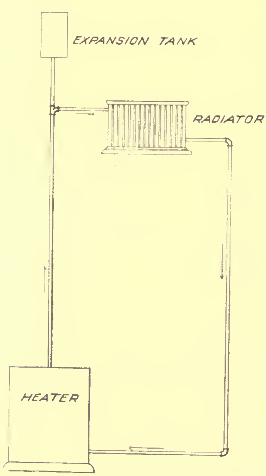


Fig. 84. Illustrating Simple Circulation in a Heating System.

Systems of Circulation. There are two distinct systems of circulation employed—one depending on the difference in temperature of the water in the supply and return pipes, called *gravity circulation*;

and another where a pump is used to force the water through the mains, called *forced circulation*. The former is used for dwellings and other buildings of ordinary size, and the latter for large buildings, and especially where there are long horizontal runs of pipe.

For gravity circulation some form of sectional cast-iron boiler is commonly used, although wrought-iron tubular boilers may be employed if desired. In the case of forced circulation, a heater designed to warm the water by means of live or exhaust steam is often used. A centrifugal or rotary pump is best adapted to this purpose, and may be driven by an electric motor or a steam engine, as most convenient.

Types of Radiating Surface. Cast-iron radiators and circulation coils are used for hot water as well as for steam. Hot-water radiators differ from steam radiators principally in having a horizontal passage at the top as well as at the bottom. This construction is necessary in order to draw off the air which gathers at the top of each loop or section. Otherwise they are the same as steam radiators, and are well adapted for the circulation of steam, and in some respects are superior to the ordinary pattern of steam radiator.

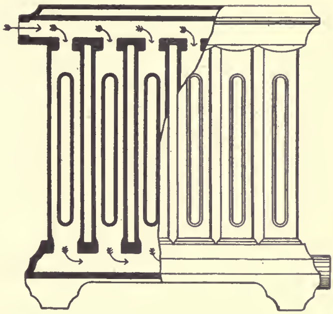


Fig. 85. Showing Construction of Radiator for Hot Water or Steam. Note Horizontal Passage along Top.

The form shown in Fig. 85 is made with an opening at the top for the entrance of water, and at the bottom for its discharge, thus insuring a supply of hot water at the top and of colder water at the bottom.

Some hot-water radiators are made with a cross-partition so arranged that all water entering passes at once to the top, from which it may take any passage toward the outlet. Fig. 86 is the more common form of radiator, and is made with continuous passages at top and bottom, the hot water being supplied at one side and drawn off at the other. The action of gravity is depended upon for making the hot and lighter water pass to the top, and the colder water sink to the bottom and flow off through the return. Hot-water radiators are usually tapped and plugged so that the pipe connections can be made either at the top or at the bottom. This is shown in Fig. 87.

Wall radiators are adapted to hot-water as well as steam heating.

Efficiency of Radiators. The efficiency of a hot-water radiator depends entirely upon the temperature at which the water is circulated. The best practical results are obtained with the water leaving the boiler at a maximum temperature of about 180 degrees in zero weather and returning at about 160 degrees; this gives an average

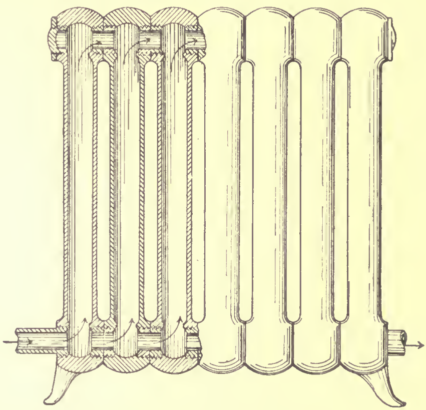


Fig. 86. Common Form of Hot-Water Radiator. Circulation Produced Wholly through Action of Gravity, Hot Water Rising to Top.

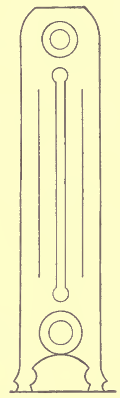


Fig. 87. End Elevation of Radiator Showing Taps at Top and Bottom for Pipe Connections.

temperature of 170 degrees in the radiators. Variations may be made, however, to suit the existing conditions of outside temperature. We have seen that an average cast-iron radiator gives off about 1.7 B.T.U. per hour per square foot of surface per degree difference in temperature between the radiator and the surrounding air, when working under ordinary conditions; and this holds true whether it is filled with steam or water.

If we assume an average temperature of 170 degrees for the water, then the difference in temperature between the radiator and the air will be 170 — 70 = 100 degrees; and this multiplied by 1.7 =

170, which may be taken as the efficiency of a hot-water radiator under the above average conditions.

This calls for a water radiator about 1.5 times as large as a steam radiator to heat a given room under the same conditions. This is common practice although some engineers multiply by the factor 1.6, which allows for a lower temperature of the water. Water leaving the boiler at 170 degrees should return at about 150; the drop in temperature should not ordinarily exceed 20 degrees.

Systems of Piping. A system of hot-water heating should produce a perfect circulation of water from the heater to the radiating

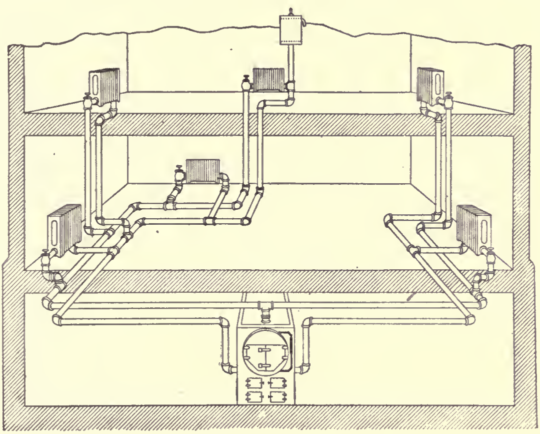


Fig. 88. System of Piping Usually Employed for Hot-Water Heating.

surface, and thence back to the heater through the returns. The system of piping usually employed for hot-water heating is shown in Fig. 88. In this arrangement the main and branches have an inclination upward from the heater; the returns are parallel to the mains, and have an inclination downward toward the heater, connecting with it at the lowest point. The flow pipes or risers are taken from the tops of the mains, and may supply one or more radiators as required. The return risers or drops are connected with the return mains in a similar manner. In this system great care must be taken to produce a nearly equal resistance to flow in all of the branches, so that each radiator may receive its full supply of water. It will always be found that the principal current of heated water will take the path of least resistance, and that a small obstruction or irregularity in the piping is sufficient to interfere greatly with the amount of heat received in the different parts of the same system.

Some engineers prefer to carry a single supply main around the building, of sufficient size to supply all the radiators, bringing back a single return of the same size. Practice has shown that in general it is not well to use pipes over 8 or 10 inches in diameter; if larger pipes are required, it is better to run two or more branches.

The boiler, if possible, should be centrally located, and branches carried to different parts of the building. This insures a more even circulation than if all the radiators are supplied from a single long main, in which case the circulation is liable to be sluggish at the farther end.

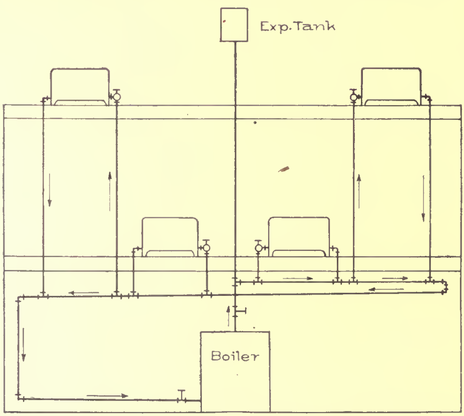
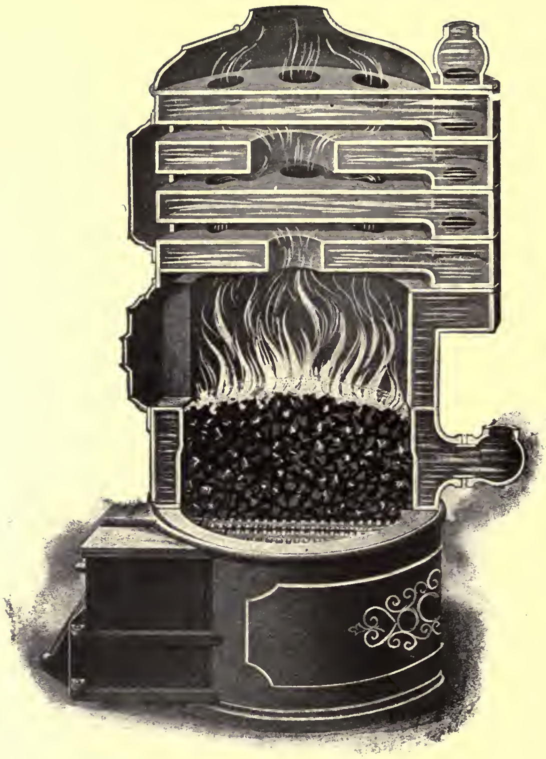


Fig. 89. System of Hot-Water Piping Especially Adapted to Apartment Buildings where Each Flat Has a Separate Heater.

The arrangement shown in Fig. 89 is similar to the circuit system for steam, except that the radiators have two connections instead of one. This method is especially adapted to apartment houses, where each flat has its separate heater, as it eliminates a separate return main, and thus reduces, by practically one-half, the amount of piping in the basement. The supply risers are taken from the top of the main; while the returns should connect into the side a short distance beyond, and in a direction *away* from the boiler. When this system is used, it is necessary to enlarge the radiators slightly as the distance from the boiler increases.

In flats of eight or ten rooms, the size of the last radiator may be increased from 10 to 15 per cent, and the intermediate ones propor-



SECTIONAL VIEW OF CAST IRON HOT WATER HEATER.

tionally, at the same time keeping the main of a large and uniform size for the entire circuit.

Overhead Distribution. This system of piping is shown in Fig. 90. A single riser is carried directly to the expansion tank, from which branches are taken to supply the various drops to which the radiators are connected. An important advantage in connection with this system is that the air rises at once to the expansion tank, and escapes through the vent, so that air-valves are not required on the radiators.

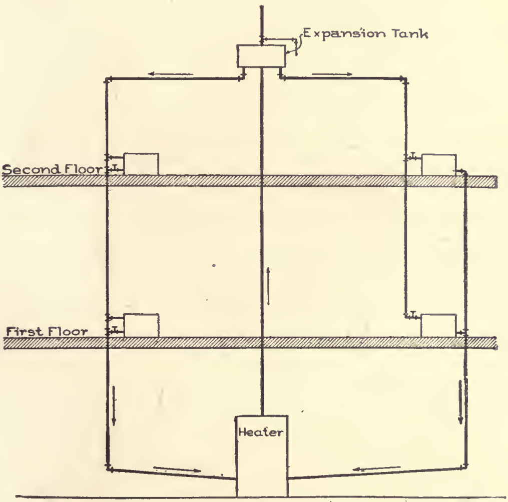


Fig. 90. "Overhead" Distribution System of Hot-Water Piping.

At the same time, it has the disadvantage that the water in the tank is under less pressure than in the heater; hence it will boil at a lower temperature. No trouble will be experienced from this, however, unless the temperature of the water is raised above 212 degrees.

Expansion Tank. Every system for hot-water heating should be connected with an expansion tank placed at a point somewhat above the highest radiator. The tank must in every case be connected to a line of piping which cannot by any possible means be shut off from the boiler. When water is heated, it expands a certain amount, depending upon the temperature to which it is raised; and a tank or reservoir should always be provided to care for this increase in volume.

Expansion tanks are usually made of heavy galvanized iron of one of the forms shown in Figs. 91 and 92, the latter form being used where the headroom is limited. The connection from the heating system enters the bottom of the tank, and an open vent pipe is taken from the top. An overflow connected with a sink or drain-pipe should be provided. Connections should be made with the water supply both at the boiler and at the expansion tank, the former to be used when first filling the system, as by this means all air is driven from the bottom upward and is discharged through the vent at the expansion tank. Water that is added afterward may be supplied directly to the expansion tank, where the water-line can be noted in the gauge-glass. A ball-cock is sometimes arranged to keep the water-line in the tank at a constant level.

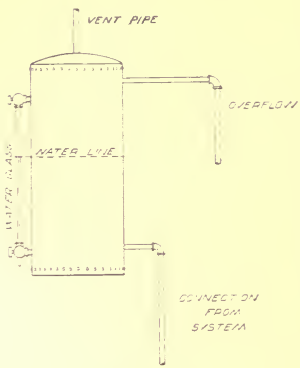


Fig 91. A Common Form of Galvanized-Iron Expansion Tank.

An *altitude gauge* is often placed in the basement with the colored hand or pointer set to indicate the normal water-line in the expansion tank. When the movable hand falls below the fixed one, more water may be added, as required, through the supply pipe at the boiler. When the tank is placed in an attic or roof space where there is danger of freezing, the expansion pipe may be connected into the side of the

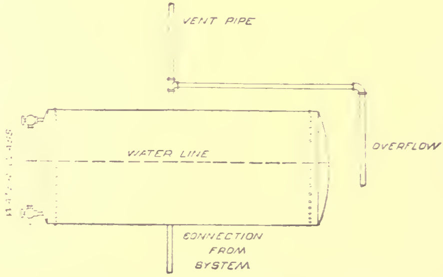


Fig 92. Form of Expansion Tank Used where Headroom is Limited.

tank, 6 or 8 inches from the bottom, and a circulation pipe taken from the lower part and connected with the return from an upper-floor radiator. This produces a slow circulation through the tank, and keeps the water warm.

The size of the expansion tank depends upon the volume of water contained in the system, and on the temperature to which it is heated. The following rule for computing the capacity of the tank may be used with satisfactory results:

Square feet of radiation, *divided by* 40, equals required capacity of tank in gallons.

Air-Venting. One very important point to be kept in mind in the design of a hot-water system, is the removal of air from the pipes and radiators. When the water in the boiler is heated, the air it contains forms into small bubbles which rise to the highest points of the system.

In the arrangement shown in Fig. 88, the main and branches grade upward from the boiler, so that the air finds its way into the radiators, from which it may be drawn off by means of the air-valves.

A better plan is that shown in Fig. 89. In this case the expansion pipe is taken directly off the top of the main over the boiler, so that the larger part of the air rises directly to the expansion tank and escapes through the vent pipe. The same action takes place in the overhead system shown in Fig. 90, where the top of the main riser is connected with the tank. Every high point in the system and every radiator, except in the downward system with top supply connection, should be provided with an air-valve.

Pipe Connections. There are various methods of connecting the radiators with the mains and risers. Fig. 93 shows a radiator connected with the horizontal flow and return mains, which are located below the floor. The manner of connecting with a vertical riser and return drop is shown in Fig. 94. As the water tends to flow to the highest point, the radiators on the lower floors should be favored by making the connection at the top of the riser and taking the pipe for the upper floors from the side as shown. Fig. 95 illustrates the manner of connecting with a radiator on an upper floor where the supply is connected at the top of the radiator.

The connections shown in Figs. 96 and 97 are used with the overhead system shown in Fig. 90.

Where the connection is of the form shown at the left in Fig. 90, the cooler water from the radiators is discharged into the supply pipe again, so that the water furnished to the radiators on the lower floors is at a lower temperature, and the amount of heating surface must be correspondingly increased to make up for this loss, as already described for the circuit system.

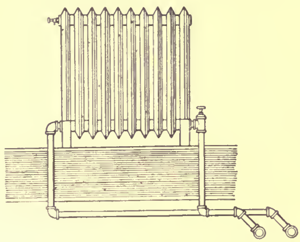


Fig 93 Radiator Connected with Horizontal Flow and Return Mains Located below Floor.

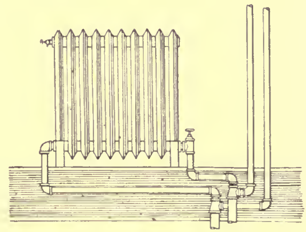


Fig. 94. Radiator Connected to Vertical Riser and Return Drop.

For example, if in the case of Fig. 90 we assume the water to leave at 180 degrees and return at 160, we shall have a drop in temperature of 10 degrees on each floor; that is, the water will enter the radiator on the second floor at 180 degrees and leave it at 170, and will enter the radiator on the first floor at 170 and leave it at 160.

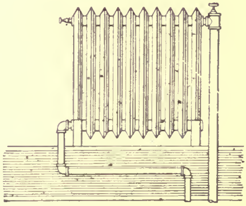


Fig. 95. Upper-Floor Radiator with Supply Connected at Top.

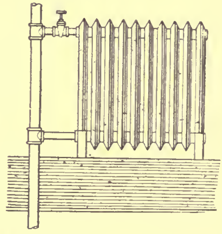


Fig 96. Radiator Connections, Overhead Distribution System.

The average temperatures will be 175 and 165, respectively. The efficiency in the first case will be 175 — 70 = 105; and 105 × 1.5 = 157. In the second case, 165 — 70 = 95; and 95 × 1.5 = 142; so that the radiator on the first floor will have to be larger than that on the second floor in the ratio of 157 to 142, in order to do the same work.

This is approximately an increase of 10 per cent for each story downward to offset the cooling effect; but in practice the supply drops are made of such size that only a part of the water is by-passed through the radiators. For this reason an increase of 5 per cent for each story downward is probably sufficient in ordinary cases.

Where the radiators discharge into a separate return as in the case of Fig. 88, or those at the right in Fig. 90, we may assume the temperature of the water to be the same on all floors, and give the radiators an equal efficiency.

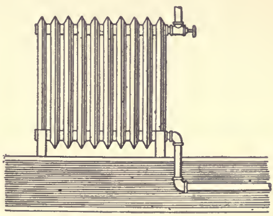


Fig. 97. Another Form of Radiator Connection, Overhead Distribution System.

In a dwelling-house of two stories, no difference would be made in the sizes of radiators on the two floors; but in the case of a tall office building, corrections would necessarily be made as above described.

Where circulation coils are used, they should be of a form which will tend to produce a flow of water through them. Figs. 98, 99, and 100 show different ways of making up and connecting these coils. In Figs. 98 and 100, supply pipes may be either drops or risers; and

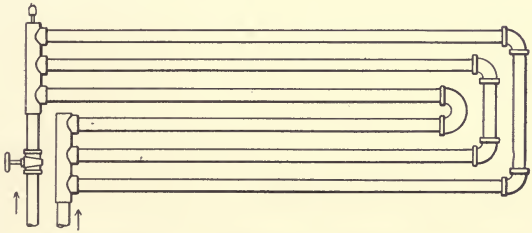


Fig. 98. Circulation Coil, One Method of Construction. Supply Pipes may be Either Drops or Risers.

in the former case the return in Fig. 100 may be carried back, if desired, into the supply drop, as shown by the dotted lines.

Combination Systems. Sometimes the boiler and piping are arranged for either steam or hot water, since the demand for a higher or lower temperature of the radiators might change.

The object of this arrangement is to secure the advantages of a hot-water system for moderate temperatures, and of steam heating for extremely cold weather.

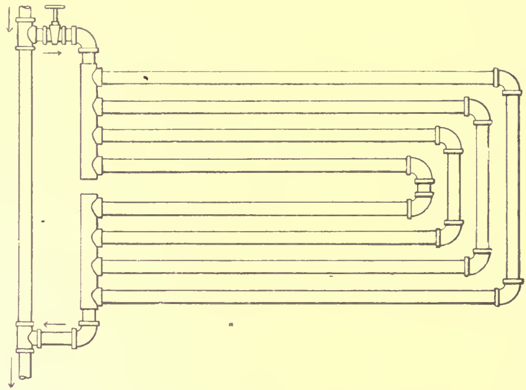


Fig. 99. Another Method of Building Up a Circulation Coil.

As less radiating surface is required for steam heating, there is an advantage due to the reduction in first cost. This is of considerable importance, as a heating system must be designed of such dimensions as to be capable of warming a building in the coldest weather;

Fig. 100. Circulation Coil with Either Drop or Riser Supply. In former case, return may be carried into Supply Drop as shown by Dotted Lines.

and this involves the expenditure of a considerable amount for radiating surfaces, which are needed only at rare intervals. A combination system of hot-water and steam heating requires, *first*, a heater or boiler which will answer for either purpose; *second,* a system of piping which will permit the circulation of either steam or hot water; and *third,* the use of radiators which are adapted to both kinds of heating. These requirements will be met by using a steam boiler provided with all the fittings required for steam heating, but so arranged that the damper regulator may be closed by means of valves when the system is to be used for hot-water heating. The addition of an expansion tank is required, which must be so arranged that it can be shut off when the system is used for steam heating. The system of piping shown in Fig. 88 is best adapted for a combination system, although an overhead distribution as shown in Fig. 90 may be used by shutting off the vent and overflow pipes, and placing air-valves on the radiators.

While this system has many advantages in the way of cost over the complete hot-water system, the labor of changing from steam to hot water will in some cases be troublesome; and should the connections to the expansion tank not be opened, serious results would follow.

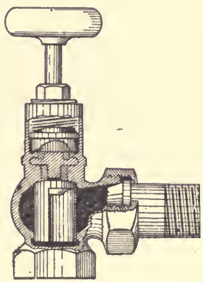


Fig. 101. Radiator Valve for Hot-Water Work.

Valves and Fittings. Gate-valves should always be used in connection with hot-water piping, although angle-valves may be used at the radiators. There are several patterns of radiator valves made especially for hot-water work; their chief advantage lies in a device for quick closing, usually a quarter-turn or half-turn being sufficient to open or close the valve. Two different designs are shown in Figs. 101 and 102.

It is customary to place a valve in only one connection, as that is sufficient to stop the flow of water through the radiator; a fitting known as a *union elbow* is often employed in place of the second valve. (See Fig. 103.)

Air-Valves. The ordinary pet-cock air-valve is the most reliable for hot-water radiators, although there are several forms of automatic valves which are claimed to give satisfaction. One of these is shown in Fig. 104. This is similar in construction to a steam trap. As air collects in the chamber, and the water-line is lowered, the float drops, and in so doing opens a small valve at the top of the chamber, which allows the air to escape. As the water flows in to take its place, the float is forced upward and the valve is closed.

All radiators which are supplied by risers from below, should be provided with air-valves placed in the top of the last section at the return end. If they are supplied by drops from an over-

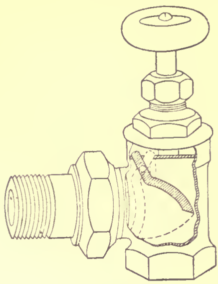


Fig. 102. Another Type of Hot-Water Radiator Valve.

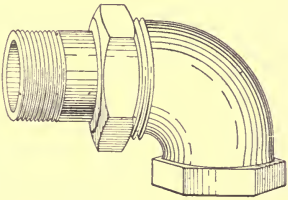


Fig. 103. Union Elbow.

head system, the air will be discharged at the expansion tank, and air-valves will not be necessary at the radiators.

Fittings. All fittings, such as elbows, tees, etc., should be of the *long-turn* pattern. If the common form is used, they should be a size larger than the pipe, bushed down to the proper size. The long-turn fittings, however, are preferable, and give a much better appearance. Connections between the radiators and risers may be made with the ordinary short-pattern fittings, as those of the other form are not well adapted to the close connections necessary for this work.

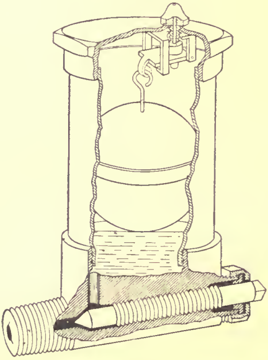


Fig. 104. Automatic Air-Valve for Hot-Water Radiator. Operated by a Float.

Pipe Sizes. The size of pipe required to supply any given radiator depends upon four conditions; *first*, the size of the radiator; *second*, its elevation above the boiler; *third*, the length of pipe required to connect it with the boiler; and *fourth*, the difference in temperature between the supply and the return.

As it would be a long and rather complicated process to work out the required size of each pipe for a heating system, Tables XXVI and XXVII have been prepared, covering the usual conditions to be met with in practice.

TABLE XXVI

Direct Radiating Surface Supplied by Mains of Different Sizes and Lengths of Run

Size of Pipe	Square Feet of Radiating Surface								
	100 ft. Run	200 ft. Run	300 ft. Run	400 ft. Run	500 ft. Run	600 ft. Run	700 ft. Run	800 ft. Run	1,000 ft. Run
1 in.	30								
1¼ "	60	50							
1½ "	100	75	50						
2 "	200	150	125	100	75				
2½ "	350	250	200	175	150	125			
3 "	550	400	300	275	250	225	200	175	150
3½ "	850	600	450	400	350	325	300	250	225
4 "	1,200	850	700	600	525	475	450	400	350
5 "		1,400	1,150	1,000	700	850	775	725	650
6 "				1,600	1,400	1,300	1,200	1,150	1,000
7 "							1,706	1,600	1,500

These quantities have been calculated on a basis of 10 feet difference in elevation between the center of the heater and the radiators, and a difference in temperature of 17 degrees between the supply and the return.

TABLE XXVII

Radiating Surface on Different Floors Supplied by Pipes of Different Sizes

Size of Riser	Square Feet of Radiating Surface					
	1st Story	2d Story	3d Story	4th Story	5th Story	6th Story
1 in.	30	55	65	75	85	95
1¼ "	60	90	110	125	140	160
1½ "	100	140	165	185	210	240
2 "	200	275	375	425	500	
2½ "	350	475				
3 "	550					
3½ "	850					

Table XXVI gives the number of square feet of direct radiation which different sizes of mains and branches will supply for varying lengths of run.

Table XXVI may be used for all horizontal mains. For vertical risers or drops, Table XXVII may be used. This has been computed for the same difference in temperature as in the case of Table XXVI (17 degrees), and gives the square feet of surface which different sizes of pipe will supply on the different floors of a building, assuming the height of the stories to be 10 feet. Where a single riser is carried to the top of a building to supply the radiators on the floors below, by drop pipes, we must first get what is called the *average elevation of the system* before taking its size from the table. This may be illustrated by means of a diagram (see Fig. 105).

In *A* we have a riser carried to the third story, and from there a drop brought down to supply a radiator on the first floor. The elevation available for producing a flow in the riser is only 10 feet, the same as though it extended only to the radiator. The water in the two pipes above the radiator is practically at the same temperature, and therefore in equilibrium, and has no effect on the flow of the water in the riser. (Actually there would be some radiation from the pipes, and the return, above the radiator, would be slightly cooler, but for purposes of illustration this may be neglected). If the radiator was on the second floor the elevation of the system would be 20 feet (see *B*); and on the third floor, 30 feet; and so on. The distance which the pipe is carried above the first radiator which it supplies has but little effect in producing a flow, especially if covered, as it should be in practice. Having seen that the flow in the main riser depends upon the elevation of the radiators, it is easy to see that the way in which it is distributed on the different floors must be considered. For example, in *B*, Fig. 105, there will be a more rapid flow through the riser with the radiators as shown, than there would be if they were reversed and the largest one were placed upon the first floor.

We get the average elevation of the system by multiplying the square feet of radiation on each floor by the elevation above the heater, then adding these products together and dividing the same by the total radiation in the whole system. In the case shown in *B*, the average elevation of the system would be

$$\frac{(100 \times 30) + (50 \times 20) + (25 \times 10)}{100 + 50 + 25} = 24 \text{ feet};$$

and we must proportion the main riser the same as though the whole radiation were on the second floor. Looking in Table XXVII, we find, for the second story, that a $1\frac{1}{2}$-inch pipe will supply 140 square feet; and a 2-inch pipe, 275 feet. Probably a $1\frac{1}{2}$-inch pipe would be sufficient.

Although the height of stories varies in different buildings, 10 feet will be found sufficiently accurate for ordinary practice.

INDIRECT HOT-WATER HEATING

This is used under the same conditions as indirect steam, and the heaters used are similar to those already described. Special

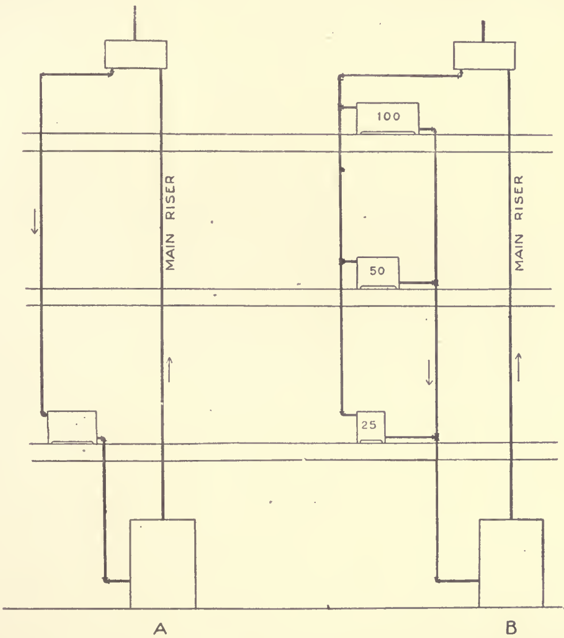


Fig. 105. Diagram to Illustrate Finding of Average Elevation of Heating System.

attention is given to the form of the sections, in order that there may be an even distribution of water through all parts of them. As the stacks are placed in the basement of a building, and only a short distance above the boiler, extra large pipes must be used to secure a proper circulation, for the *head* producing flow is small. The stack casings, cold-air and warm-air pipes, and registers are the same as in steam heating.

Types of Radiators. The radiators for indirect hot-water heating are of the same general form as those used for steam. Those shown in Figs. 52, 53, 56, 106, and 107 are common patterns. The *drum pin*, Fig. 106, is an excellent form, as the method of making the connections insures a uniform distribution of water through the stack.

Fig. 107 shows a radiator of good form for water circulation, and also of good depth, which is a necessary point in the design of hot-water radiators. They should be not less than 12 or 15 inches deep for good results. Box coils of the form given for steam may also be

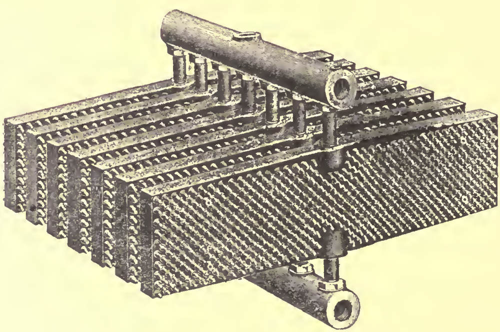


Fig. 106. "Drum Pin" Indirect Hot-Water Radiator.

used, provided the connections for supply and return are made of good size.

Size of Stacks. As indirect hot-water heaters are used principally in the warming of dwelling-houses, and in combination with direct radiation, the easiest method is to compute the surfaces required for direct radiation, and multiply these results by 1.5 for pin radiators of good depth. For other forms the factor should vary from 1.5 to 2, depending upon the depth and proportion of free area for air-flow between the sections.

If it is desired to calculate the required surface directly by the thermal unit method, we may allow an efficiency of from 360 to 400 for good types in zero weather.

In schoolhouse and hospital work, where larger volumes of air are warmed to lower temperatures, an efficiency as high as 500 B. T. U. may be allowed for radiators of good form.

Flues and Casings. For cleanliness, as well as for obtaining the best results, indirect stacks should be hung at one side of the register or flue receiving the warm air, and the cold-air duct should enter beneath the heater at the other side. A space of at least 10 inches, and preferably 12, should be allowed for the warm air above the stack. The top of the casing should pitch upward toward the warm-air outlet at least an inch in its length. A space of from 8 to 10 inches should be allowed for cold air below the stack.

As the amount of air warmed per square foot of heating surface is less than in the case of steam, we may make the flues somewhat smaller as compared with the size of heater. The following proportions may be used under usual conditions for dwelling-houses: $1\frac{1}{2}$ square inches per square foot of radiation for the first floor, $1\frac{1}{4}$ square inches for the second floor, and $1\frac{1}{4}$ square inches for the cold-air duct.

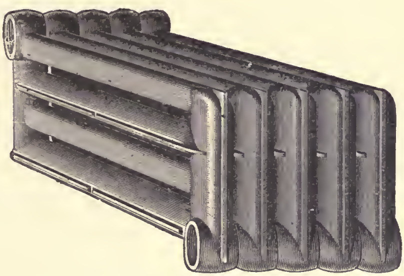


Fig. 107. Indirect Hot-Water Radiator.

Pipe Connections. In indirect hot-water work, it is not desirable to supply more than 80 to 100 square feet of radiation from a single connection. When the requirements call for larger stacks, they should be divided into two or more groups according to size.

It is customary to carry up the main from the boiler to a point near the basement ceiling, where it is air-vented through a small pipe leading to the expansion tank. The various branches should grade downward and connect with the tops of the stacks. In this way, all air, both from the boiler and from the stacks, will find its way to the highest point in the main, and be carried off automatically.

As an additional precaution, a pet-cock air-valve should be placed in the last section of each stack, and brought out through the casing by means of a short pipe.

TABLE XXVIII

Radiating Surface Supplied by Pipes of Various Sizes—Indirect Hot-Water System

Diameter of Pipe	Square Feet of Radiating Surface			
	100 Ft. Run	200 Ft. Run	300 Ft. Run	400 Ft. Run
1 in.	15			
1¼ "	30	25		
1½ "	50	40	25	
2 "	100	75	60	50
2½ "	175	125	100	90
3 "	275	200	150	140
3½ "	425	300	225	200
4 "	600	425	350	300
5 "		700	575	500
6 "				800
7 "				1,200

Some engineers make a practice of carrying the main to the ceiling of the first story, and then dropping to the basement before branching to the stacks, the idea being to accelerate the flow of water through the main, which is liable to be sluggish on account of the small difference in elevation between the boiler and stacks. If the return leg of the loop is left uncovered, there will be a slight drop in temperature, tending to produce this result; but in any case it will be exceedingly small. With supply and return mains of suitable size and properly graded, there should be no difficulty in securing a good circulation in basements of average height.

Pipe Sizes. As the difference in elevation between the stacks and the heater is necessarily small, the pipes should be of ample size to offset the slow velocity of flow through them. The sizes mentioned in Table XXVIII, for runs up to 400 feet, will be found to supply ample radiating surface for ordinary conditions. Some engineers make a practice of using somewhat smaller pipes, but the larger sizes will in general be found more satisfactory.

CARE AND MANAGEMENT OF HOT-WATER HEATERS

The directions given for the care of steam-heating boilers apply in a general way to hot-water heaters, as to the methods of caring for the fires and for cleaning and filling the heater. Only the special points of difference need be considered. Before building the fire, all the pipes and radiators must be full of water, and the expansion tank should be partially filled as indicated by the gauge-glass. Should the water in any of the radiators fail to circulate, see that the valves are wide open and that the radiator is free from air. Water must always be added at the expansion tank when for any reason it is drawn from the system.

The required temperature of the water will depend upon the outside conditions, and only enough fire should be carried to keep the rooms comfortably warm. Thermometers should be placed in the flow and return pipes near the heater, as a guide. Special forms are made for this purpose, in which the bulb is immersed in a bath of oil or mercury (see, Fig. 108).

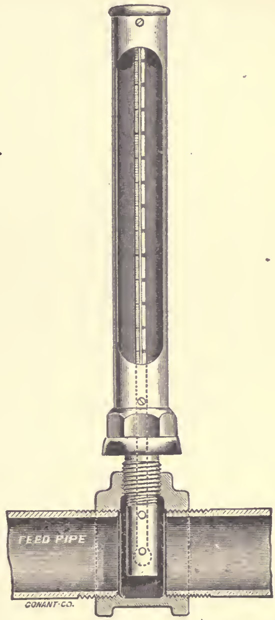


Fig. 108. Thermometer Attached to Feed-Pipe near Heater, to Determine Temperature of Water.

FORCED HOT-WATER CIRCULATION

While the gravity system of hot-water heating is well adapted to buildings of small and medium size, there is a limit to which it can be carried economically. This is due to the slow movement of the water, which calls for pipes of excessive size. To overcome this difficulty, pumps are used to force the water through the mains at a comparatively high velocity.

The water may be heated in a boiler in the same manner as for gravity circulation, or exhaust steam may be utilized in a feed-water heater of large size. Sometimes part of the heat is derived from an economizer placed in the smoke passage from the boilers.

Systems of Piping. The mains for forced circulation are usually run in one of two ways. In the *two-pipe system*, shown in Fig. 109, the supply and return are carried side by side, the former reducing in size, and the latter increasing as the branches are taken off.

The flow through the risers is produced by the difference in pressure in the supply and return mains; and as this is greatest nearest the pump, it is necessary to place throttle-valves in the risers to prevent short-circuiting and to secure an even distribution through all parts of the system.

Fig. 110 shows the *single-pipe* or *circuit system.* This is similar to the one already described for gravity circulation, except that it can be used on a much larger scale.

A single main is carried entirely around the building in this case, the ends being connected with the suction and discharge of the pump as shown.

As the pressure or head in the main drops constantly throughout the circuit, from the discharge of the pump back to the suction, it is

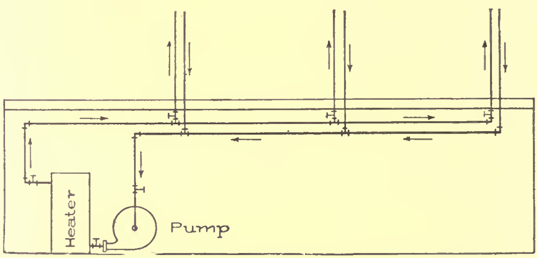


Fig. 109. "Two-Pipe" System for Forced Hot-Water Circulation.

evident that if a supply riser be taken off at any point, and the return be connected into the main a short distance along the line, there will be a sufficient difference in pressure between the two points to produce a circulation through the two risers and the connecting radiators. A distance of 8 or 10 feet between the connections is usually ample to produce the necessary circulation, and even less if the supply is taken from the top of the main and the return connected into the side.

Sizes of Mains and Branches. As the velocity of flow is independent of the temperature and elevation when a pump is used, it is necessary to consider only the volume of water to be moved and the length of run.

TYPICAL HEATING INSTALLATION SHOWING SECTIONAL BOILER AND RADIATOR.

American Radiator Company.

The volume is found by the equation

$$Q = \frac{R\ E}{500\ T},$$

in which

Q = Gallons of water required per minute;
R = Square feet of radiating surface to be supplied;
E = Efficiency of radiating surface in B. T. U. per sq. foot per hour;
T = Drop in temperature of the water in passing through the heating system.

In systems of this kind, where the circulation is comparatively rapid, it is customary to assume a drop in temperature of 30° to 40°, between the supply and return.

Having determined the gallons of water to be moved, the required size of main can be found by assuming the velocity of flow, which for pipes from 5 to 8 inches in diameter may be taken at 400 to 500

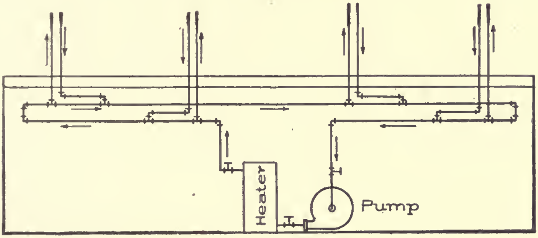


Fig. 110. "Single-Pipe" or "Circuit" System for Forced Hot-Water Circulation.

feet per minute. A velocity as high as 600 feet is sometimes allowed for pipes of large size, while the velocity in those of smaller diameter should be proportionally reduced to 250 or 300 feet for a 3-inch pipe. The next step is to find the pressure or head necessary to force the water through the main at the given velocity. This in general should not exceed 50 or 60 feet, and much better pump efficiencies will be obtained with heads not exceeding 35 or 40 feet.

As the water in a heating system is in a state of equilibrium, the only power necessary to produce a circulation is that required to overcome the friction in the pipes and radiators; and, as the area of the passageways through the latter is usually large in comparison with the former, it is customary to consider only the head necessary to force the water through the mains, taking into consideration the additional friction produced by valves and fittings.

Each long-turn elbow may be taken as adding about 4 feet to the length of pipe; a short-turn fitting, about 9 feet; 6-inch and 4-inch swing check-valves, 50 feet and 25 feet, respectively; and 6-inch and 4-inch globe check-valves, 200 feet and 130 feet, respectively.

Table XXIX is prepared especially for determining the size of mains for different conditions, and is used as follows:

Example. Suppose that a heating system requires the circulation of 480 gallons of water per minute through a circuit main 600 feet in length. The pipe contains 12 long-turn elbows and 1 swing check-valve. What diameter of main should be used?

Assuming a velocity of 480 feet per minute as a trial velocity, we follow along the line corresponding to that velocity, and find that a 5-inch pipe will deliver the required volume of water under a head of 4.9 feet for each 100 feet length of run.

The actual length of the main, including the equivalent of the fittings as additional length, is

$$600 + (12 \times 9) + 50 = 758 \text{ feet};$$

hence the total head required is $4.9 \times 7.58 = 37$ feet. As both the assumed velocity and the necessary head come within practicable limits, this is the size of pipe which would probably be used. If it were desired to reduce the power for running the pump, the size of main could be increased. That is, Table XXIX shows that a 6-inch pipe would deliver the same volume of water with a friction head of only about 2 feet per 100 feet in length, or a total head of $2 \times 7.58 = 15$ feet.

The risers in the circuit system are usually made the same size as for gravity work. With double mains, as shown in Fig. 109, they may be somewhat smaller, a reduction of one size for diameters over $1\frac{1}{4}$ inches being common.

The branches connecting the risers with the mains may be proportioned from the combined areas of the risers. When the branches are of considerable size, the diameter may be computed from the available head and volume of water to be moved.

Pumps. Centrifugal pumps are usually employed in connection with forced hot-water circulation, in preference to pumps of the piston or plunger type. They are simple in construction, having no valves, produce a continuous flow of water, and, for the low heads

TABLE XXIX

Capacity in Gallons per Minute Discharged at Velocities of 300 to 540 Feet per Minute—Also Friction Head in Feet, per 100 Feet Length of Pipe

	DIAMETER OF PIPE											
	3-Inch		4-Inch		5-Inch		6-Inch		7-Inch		8-Inch	
Velocity	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction
300	110	3.41	195	2.56	306	2.05	440	1.70	600	1.46	783	1.28
480	176	8.16	314	6.12	490	4.9	705	4.08	959	3.49	1,253	3.06
540	198	10.1	352	7.64	550	6.11	794	5.09	1,079	4.36	1,410	3.82

against which they are operated, have a good efficiency. A pump of this type, with a direct-connected engine, is shown in Fig. 111.

Under ordinary conditions the efficiency of a centrifugal pump falls off considerably for heads above 30 or 35 feet; but special high-speed pumps are constructed which work with a good efficiency against 500 feet or more.

Under favorable conditions an efficiency of 60 to 70 per cent is often obtained; but for hot-water circulation it is more common to assume an efficiency of about 50 per cent for the average case.

The horse-power required for driving a pump is given by the following formula:

$$\text{H. P.} = \frac{H \times V \times 8.3}{33{,}000 \times E},$$

in which

H = Friction head in feet;
V = Gallons of water delivered per minute;
E = Efficiency of pump.

Centrifugal pumps are made in many sizes and with varying proportions, to meet the different requirements of capacity and head.

Heaters. If the water is heated in a boiler, any good form may be used, the same as for gravity work. In case tubular boilers are used, the entire shell may be filled with tubes, as no steam space is required.

In order to prevent the water from passing in a direct line from the inlet to the outlet, a series of baffle-plates should be used to bring it in contact with all parts of the heating surface.

When steam is used for heating the water, it is customary to employ a closed feed-water heater with the steam on the inside of the tubes and the water on the outside.

Any good form of heater can be used for this purpose by providing it with steam connections of sufficient size. In the ordinary form of heater, the feed-water flows through the tubes, and the connections are therefore small, making it necessary to substitute special nozzles of large size when used in the manner here described.

When computing the required amount of heating surface in the tubes of a heater, it is customary to assume an efficiency of about 200 B. T. U. per square foot of surface per hour, per degree difference in temperature between the water and steam.

It is usual to circulate the water at a somewhat higher temperature in systems of this kind, and a maximum initial temperature of 200 degrees, with a drop of 40 degrees in the heating system, may be used in computing the size of heater. If exhaust steam is used at atmospheric pressure, there will be a difference of $212 - 180 = 32$ degrees, between the *average* temperature of the water and the steam, giving an efficiency of $200 \times 32 = 6{,}400$ B. T. U. per square foot of heating surface.

From this it is evident that $6{,}400 \div 170 = 38$ square feet of direct radiating surface, or $6{,}400 \div 400 = 16$ square feet of indirect, may be supplied from each square foot of tube surface in the heater.

Example. A building having 6,000 square feet of direct, and 2,000 square feet of indirect radiation, is to be warmed by hot water under forced circulation. Steam at atmospheric pressure is to be used for heating the water. How many square feet of heating surface should the heater contain?

$6{,}000 \div 38 = 158$; and $2{,}000 \div 16 = 125$; therefore, $158 + 125 = 283$ square feet, the area of heating surface called for.

When the exhaust steam is not sufficient for the requirements, an auxiliary live steam heater is used in connection with it.

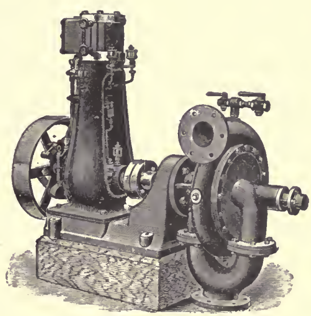


Fig. 111. Centrifugal Pump Direct-Connected to Engine, for Forced Hot-Water Circulation.

EXAMPLES FOR PRACTICE

1. A building contains 10,000 square feet of direct radiation and 4,000 square feet of indirect radiation. How many gallons of water must be circulated through the mains per minute, allowing a drop in temperature of 40 degrees? Ans. 165 gal.

2. In the above example, what size of main should be used, assuming the circuit to be 300 feet in length and to contain ten long-turn elbows? The friction head is not to exceed 10 ft., and the velocity of flow not to exceed 300 feet per minute. Ans. 4-inch.

3. What horse-power will be required to drive a centrifugal pump delivering 400 gallons per minute against a friction head of 40 feet, assuming an efficiency of 50 per cent for the pump?

Ans. 8 H. P.

4. A building contains 10,000 square feet of direct radiation and 5,000 square feet of indirect radiation. Steam at atmospheric pressure is to be used. The initial temperature of the water is to be 200°; and the final 160°. How many square feet of heating surface should the heater contain? Ans. 575 sq. ft.

5. How many square feet would be required in the above heater (Example 4) if the initial temperature of the water were 180° and the final temperature 150°? Ans. 399 sq. ft.

EXHAUST-STEAM HEATING

Steam, after being used in an engine, contains the greater part of its heat; and if not condensed or used for other purposes, it can usually be employed for heating without affecting to any great extent the power of the engine. In general, we may say that it is a matter of economy to use the exhaust for heating, although various factors must be considered in each case to determine to what extent this is true. The more important considerations bearing upon the matter are: the relative quantities of steam required for power and for heating; the length of the heating season; the type of engine used; the pressure carried; and, finally, whether the plant under consideration is entirely new, or whether, on the other hand, it involves the adapting of an old heating system to a new plant.

The first use to be made of the exhaust steam is the heating of the feed-water, as this effects a constant saving both summer and winter, and can be done without materially increasing the back-pressure on the engine. Under ordinary conditions, about one-sixth of the steam supplied to the engine can be used in this way, or more nearly one-fifth of the exhaust *discharged* from the engine.

We may assume in average practice that about 80 per cent of the steam supplied to an engine is discharged in the form of steam at a lower pressure, the remaining 20 per cent being partly converted into work and partly lost through cylinder condensation. Taking this into account, there remains, after deducting the steam used for feed-water heating, $.8 \times \frac{4}{5} = .64$ of the entire quantity of steam supplied to the engine, available for heating purposes.

When the quantity of steam required for heating is small compared with the total amount supplied to the engine, or where the heating season is short, it is often more economical to run the engine condensing and use the live steam for heating. This can be determined in any particular case by computing the saving in fuel by the use of a condenser, taking into account the interest and depreciation on the first cost of the condensing apparatus, and the cost of water, if it must be purchased, and comparing it with the cost of heating with live steam.

Usually, however, in the case of office buildings and institutions, and commonly in the case of shops and factories, especially in northerly latitudes, it is advantageous to use the exhaust for heating, even if a condenser is installed for summer use only. The principal objection raised to the use of exhaust steam has been the higher back-pressure required on the engines, resulting in a loss of power nearly proportional to the ratio of the back-pressure to the mean effective pressure. There are two ways of offsetting this loss—one, by raising the initial or boiler pressure; and the other, by increasing the cut-off of the engine. Engines are usually designed to work most economically at a given cut-off, so that in most cases it is undesirable to change it to any extent. Raising the boiler pressure, on the other hand, is not so objectionable if the increase amounts to only a few pounds.

Under ordinary conditions in the case of a simple engine, a rise of 3 pounds in the back-pressure calls for an increase of about 5 pounds in the boiler pressure, to maintain the same power at the engine.

The indicator card shows a back-pressure of about 2 pounds when an engine is exhausting into the atmosphere, so that an increase of 3 pounds would bring the pressure up to a total of 5 pounds which should be more than sufficient to circulate the steam through any well-designed heating system.

If it is desired to reduce rather than increase the back-pressure, one of the so-called *vacuum systems*, described later, can be used.

The systems of steam heating which have been described are those in which the water of condensation flows back into the boiler by gravity. Where exhaust steam is used, the pressure is much below that of the boiler, and it must be returned either by a pump or by a return trap. The exhaust steam is often insufficient to supply the entire heating system, and must be supplemented by live steam taken directly from the boiler. This must first pass through a reducing valve in order to reduce the pressure to correspond with that carried in the heating system.

An engine does not deliver steam continuously, but at regular intervals, at the end of each stroke; and the amount is likely to vary with the work done, since the governor is adjusted to admit steam in such a quantity as is required to maintain a uniform speed. If the work is light, very little steam will be admitted to the engine; and for this reason the supply available for heating may vary somewhat, depending upon the use made of the power delivered by the engine. In mills the amount of exhaust steam is practically constant; in office buildings where power is used for lighting, the variation is greater, especially if power is also required for the running of elevators.

The general requirements for a successful system of exhaust steam heating include a system of piping of such proportions that only a slight increase in back-pressure will be thrown upon the engine; a connection which shall automatically supply live steam at a reduced pressure as needed; provision for removing the oil from the exhaust steam; a relief or back-pressure valve arranged to prevent any sudden increase in back pressure on the engine; and a return system of some kind for returning the water of condensation to the boiler against a higher pressure. These requirements may be met in various ways, depending upon actual conditions found in different cases.

To prevent sudden changes in the back-pressure, due to irregular supply of steam, the exhaust pipe from the engine is often carried to a closed tank having a capacity from 30 to 40 times that of the engine cylinder. This tank may be provided with baffle-plates or other arrangements and may serve as a separator for removing the oil from the steam as it passes through.

Any system of piping may be used; but great care should be taken that as little resistance as possible is introduced at bends and fittings; and the mains and branches should be of ample size. Usually the best results are obtained from the system in which the main steam pipe is carried directly to the top of the building, the distributing pipes being run from that point, and the radiating surfaces supplied by a down-flowing current of steam.

Before taking up the matter of piping in detail a few of the more important pieces of apparatus will be described in a brief way.

Reducing Valves. The action of pressure-reducing valves has been taken up quite fully in "Boiler Accessories," and need not be repeated here. When the reduction in pressure is large, as in the case of a combined power and heating plant, the valve may be one or two sizes smaller than the low-pressure main into which it discharges. For example, a 5-inch valve will supply an 8-inch main, a 4-inch a 6-inch main, a 3-inch a 5-inch main, a 2½-inch a 4-inch main, etc.

For the smaller sizes, the difference should not be more than one size. All reducing valves should be provided with a valved by-pass for cutting out the valve in case of repairs. This connection is usually made as shown in plan by Fig. 112.

Grease Extractor. When exhaust steam is used for heating purposes, it must first be passed through some form of separator for removing the oil; and as an additional precaution it is well to pass the

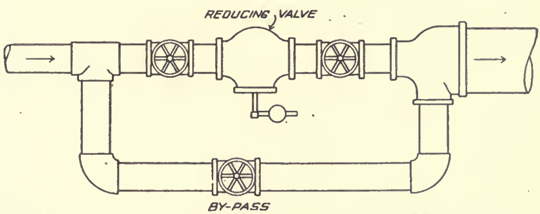


Fig. 112. Connections of Reducing Valve in Exhaust-Steam Heating System.

water of condensation through a separating tank before returning it to the boilers.

Such an arrangement is shown in Fig. 113. As the oil collects on the surface of the water in the tank, it can be made to overflow into the sewer by closing the valve in the connection with the receiving tank, for a short time.

As much of the oil as possible should be removed before the steam enters the pipes and radiators, else a coating will be formed on their inner surfaces, which will reduce their heating efficiency. The separation of the oil is usually effected by introducing a series of baffling plates in the path of the steam; the particles of oil striking these are stopped, and thus separated from the steam. The oil drops into a receiver provided for this purpose and is discharged through a trap to the sewer.

In the separator, or extractor, shown in Fig. 114, the separation is accomplished by a series of plates placed in a vertical position in the body of the separator, through which the steam must pass. These plates consist of upright hollow columns, with openings at regular intervals for the admission of water and oil, which drain downward to the receiver below. The steam takes a zigzag course, and all of it comes in contact with the intercepting plates, which insures a thorough separation of the oil and other solid matter from the steam. Another form, shown in Fig. 115, gives excellent results, and has the advantage of providing an equalizing chamber for overcoming, to some extent, the unequal pressure due to the varying load on the engine. It consists of a tank or receiver about 4 feet in diameter, with heavy boiler-iron heads slightly crowned to give stiffness.

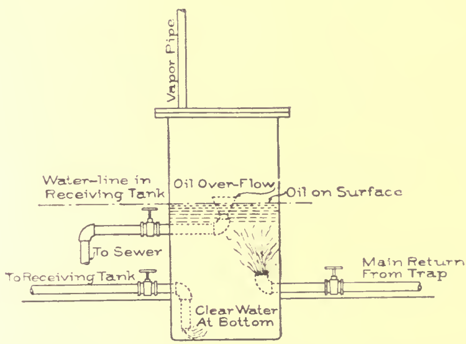


Fig. 113. Separator for Removing Oil from Exhaust Steam and Water Condensation.

Through the center is a layer of excelsior (wooden shavings of long fibre) about 12 inches in thickness, supported on an iron grating, with a similar grating laid over the top to hold it in place. The steam enters the space below the excelsior and passes upward, as shown by the arrows. The oil is caught by the excelsior, which can be renewed from time to time as it becomes saturated. The oil and water which fall to the bottom of the receiver are carried off through a trap. Live steam may be admitted through a reducing valve, for supplementing the exhaust when necessary.

Back-Pressure Valve. This is a form of relief valve which is placed in the outboard exhaust pipe to prevent the pressure in the heating system from rising above a given point. Its office is the reverse of the reducing valve, which supplies more steam when the pressure becomes too low. The form shown in Fig. 116 is designed for a vertical pipe. The valve proper consists of two discs of unequal area, the combined area of which equals that of the pipe. The force tending to open the valve is that due to the steam pressure acting on an area equal to the difference in area between the two discs; it is clear from the cut that the pressure acting on the larger disc tends to open the valve while the pressure on the smaller acts in the opposite direction. The valve-stem is connected by a link and crank arm with a spindle upon which is a lever and weight outside. As the valve opens, the weight is raised, so that, by placing it in different positions on the lever arm, the valve will open at any desired pressure.

Fig. 117 shows a different type, in which a spring is used instead of a weight. This valve has a single disc moving in a vertical direction. The valve stem is in the form of a piston or dash-pot which prevents a too sudden movement and makes it more quiet in its action. The disc is held on its seat against the steam pressure by a lever attached to the spring as shown. When the pressure of the steam on the underside becomes greater than the tension of the spring, the valve lifts and allows the steam to escape. The tension of the spring can be varied by means of the adjusting screw at its upper end.

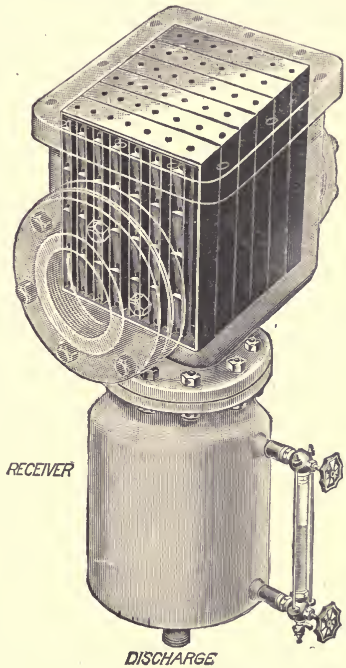


Fig. 114. Oil Separator Consisting of Vertical Plates with Openings Giving Steam a Zigzag Course.

A back-pressure valve is simply a low-pressure safety-valve designed with a specially large opening for the passage of steam through it. These valves are made for horizontal as well as for vertical pipes.

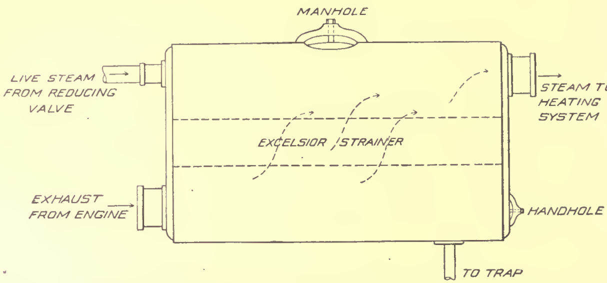


Fig. 115. Oil Separator Consisting of a Tank in which Steam is Filtered by Passing Upward through a Layer of Excelsior.

Exhaust Head. This is a form of separator placed at the top of an outboard exhaust pipe to prevent the water carried up in the steam from falling upon the roofs of buildings or in the street below. Fig. 118 is known as a centrifugal exhaust head. The steam, on entering at the bottom, is given a whirling or rotary motion by the spiral deflectors; and the water is thrown outward by centrifugal force against the sides of the chamber, from which it flows into the shallow trough at the base, and is carried away through the drip-pipe, which is brought down and connected with a drain-pipe inside the building. The passage of the steam outboard is shown by the arrows. Other forms are used in which the water is separated from the steam by deflectors which change the direction of the currents.

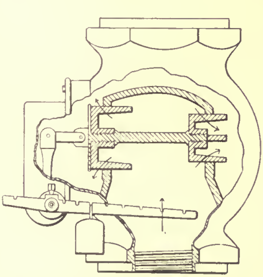


Fig. 116. Automatically Acting Back-Pressure Valve Attached to Vertical Pipe. For Preventing Rise of Pressure in System above any Desired Point.

Automatic Return-Pumps. In exhaust heating plants, the condensation is returned to the boilers by means of some form of return-pump. A combined pump and receiver of the form illustrated in Fig. 119 is generally used. This consists of a cast-iron or wrought-iron tank mounted on a base in connection with a boiler feed-pump. Inside the tank is a ball-float connected by means of levers with a valve in the steam pipe which is connected with the pump. When the water-line in the tank rises above a certain level, the float is raised and opens the steam valve, which starts the pump. When the water is lowered to its normal level, the valve closes and the pump stops. By this arrangement, a constant water-line is maintained in the receiver, and the pump runs only as needed to care for the condensation as it returns from the heating system. If dry returns are used, they may be brought together and connected with the top of the receiver. If it is desired to seal the horizontal runs, as

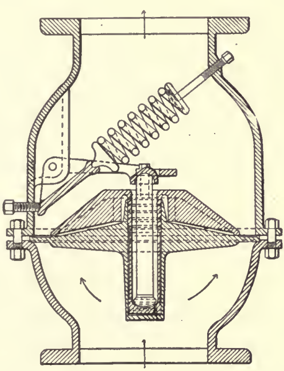


Fig. 117. Back-Pressure Valve Automatically Operated by a Spring.

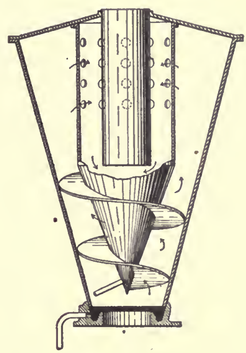


Fig. 118. Centrifugal Exhaust Head.

is usually the case, the receiver may be raised to a height sufficient to give the required elevation and the returns connected near the bottom below the water-line.

A *balance-pipe,* so called, should connect the heating main with the top of the tank, for equalizing the pressure; otherwise the steam above the water would condense, and the vacuum thus formed would draw all the water into the tank, leaving the returns practically empty and thus destroying the condition sought. Sometimes an independent regulator or pump governor is used in place of a receiver. One type is shown in Fig. 120. The return main is connected at the upper opening, and the pump suction at the lower. A float inside the chamber operates the steam valve shown at the top, and the pump works automatically as in the case just described.

If it is desired to raise the water-line, the regulator may be elevated to the desired height and connections made as shown in Fig. 121.

Return Traps. The principle of the return trap has been described in "Boiler Accessories," but its practical form and application

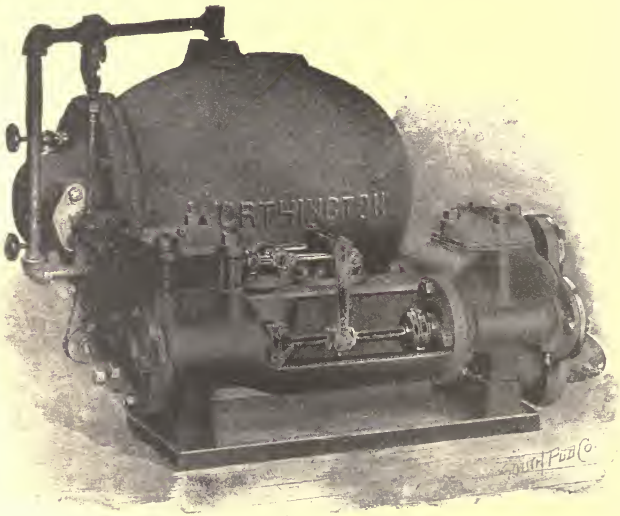


Fig. 119. Combined Receiver and Automatic Pump for Returning Water of Condensation to Boiler.

will be taken up here. The type shown in Fig. 122 has all its working parts outside the trap. It consists of a cast-iron bowl pivoted at *G* and *H*. There is an opening through *G* connecting with the inside of the bowl. The pipe *K* connects through *C* with an interior pipe opening near the top (see Fig. 123). The pipe *D* connects with a receiver, into which all the returns are brought. *A* is a check-valve allowing water to pass through in the direction shown by the arrow. *E* is a pipe connecting with the boiler below the water-line. *B* is a check opening toward the boiler, and K, a pipe connected with the steam main or drum.

The action of the trap is as follows: As the bowl fills with water from the receiver, it overbalances the weighted lever and drops to the bottom of the ring. This opens the valve C, and admits steam at boiler pressure to the top of the trap. Being at a higher level the water flows by gravity into the boiler, through the pipe E. Water and steam are kept from passing out through D by the check A.

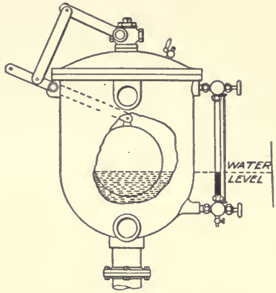


Fig. 120. Automatic Float-Operated Pump Governor Used instead of a Receiver.

When the trap has emptied itself, the weight of the ball raises it to the original position, which movement closes the valve C and opens the small vent F. The pressure in the bowl being relieved, water flows in from the receiver through D, until the trap is filled, when the

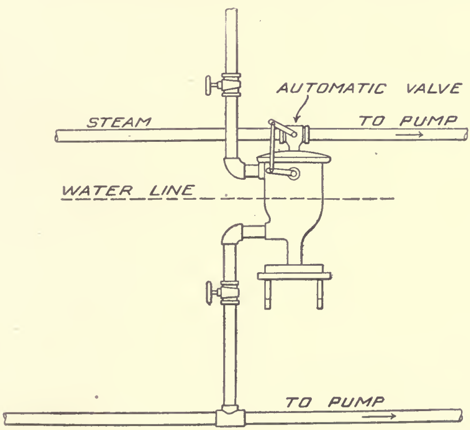


Fig. 121. Pump Regulator Placed at Sufficient Height to Raise Water-Line to Point Desired.

process is repeated. In order to work satisfactorily, the trap should be placed at least 3 feet above the water-level in the boiler, and the pressure in the returns must always be sufficient to raise the water from the receiver to the trap against atmospheric pressure, which is theoretically about 1 pound for every 2 feet in height. In practice there will be more or less friction to overcome, and suitable adjustments must be made for each particular case.

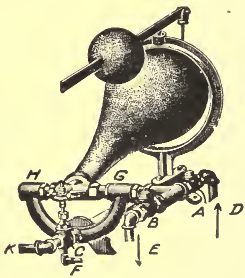


Fig. 122. Return Trap with Working Parts External.

Fig. 124 shows another form of trap acting upon the same principle, except that in this case the steam valve is operated by a bucket or float inside the trap. The pipe connections are practically the same as with the trap just described.

Return traps are more commonly used in smaller plants where it is desired to avoid the expense and care of a pump.

Damper-Regulators. Every heating and every power plant should be provided with automatic means for closing the dampers when the steam pressure reaches a certain point, and for opening them again when the pressure drops. There are various regulators designed for this purpose, a simple form of which is shown in Fig. 125.

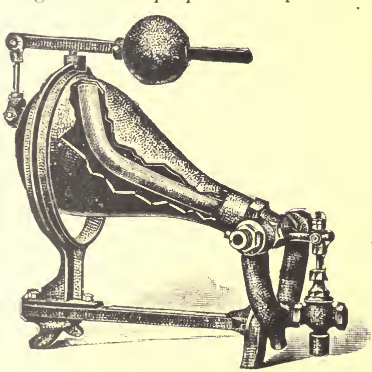
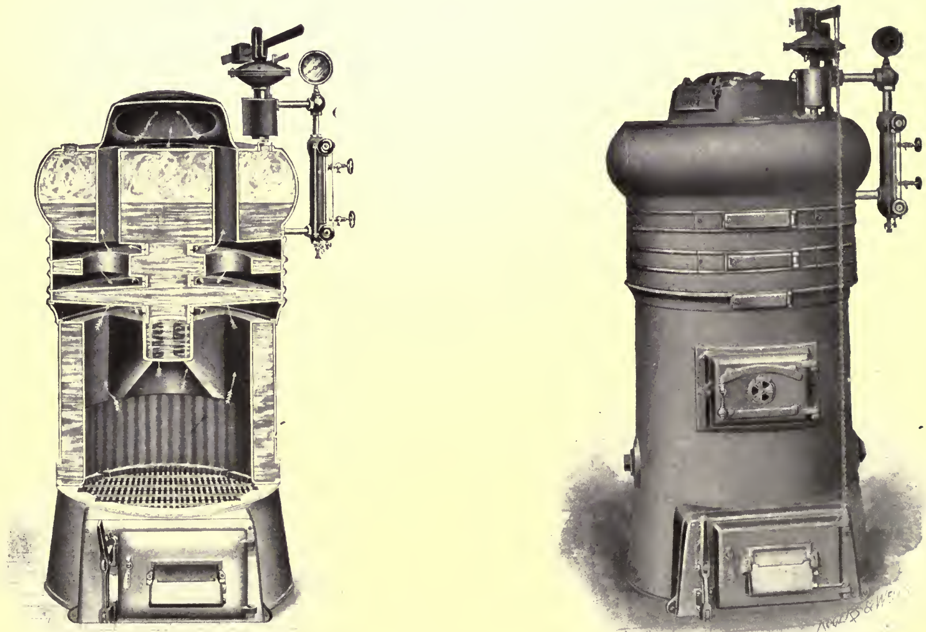


Fig. 123. Showing Interior Detail of Return Trap of Fig. 122.

Steam at boiler pressure is admitted beneath a diaphragm which is balanced by a weighted lever. When the pressure rises to a certain point, it raises the lever slightly and opens a valve which admits water under pressure above a diaphragm located near the smoke-pipe. This action forces down a lever connected by chains with the damper, and closes it. When the steam pressure drops, the water-valve is closed, and the different parts of the apparatus take their original positions.

Another form similar in principle is shown in Fig. 126. In this



VIEWS OF THE AMERICAN RADIATOR COMPANY'S STEAM BOILER FOR HARD COAL.

case a piston is operated by the water-pressure, instead of a diaphragm. In both types the pressures at which the damper shall open and close are regulated by suitable adjustments of the weights upon the levers.

Pipe Connections. The method of making the pipe connections in any particular case will depend upon the general arrangement of the apparatus and the various conditions. Fig. 127 illustrates

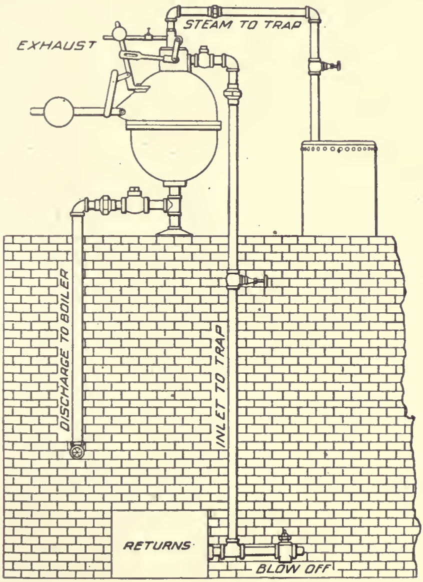


Fig. 124. Return Trap with Steam Valve Operated by Bucket or Float Inside.

the general principles to be followed, and by suitable changes may be used as a guide in the design of new systems.

Steam first passes from the boilers into a large drum or header. From this, a main, provided with a shut-off valve, is taken as shown; one branch is carried to the engines, while another is connected with the heating system through a reducing valve having a by-pass and cut-out valves. The exhaust from the engines connects with the large main over the boilers at a point just above the steam drum. The branch at the right is carried outboard through a back-pressure valve which may be set to carry any desired pressure on the system. The other branch at the left passes through an oil separator into the heating system. The connections between the mains and radiators are made in the usual way, and the main return is carried back to the return pump near the floor. A false water-line or seal is obtained by elevating the pump regulator as already described. An equalizing

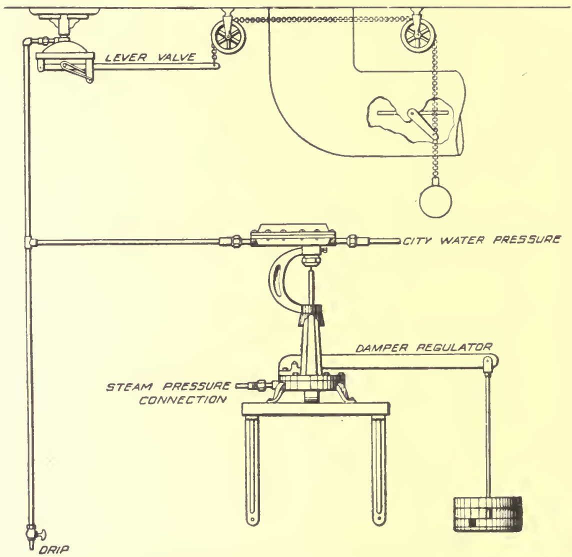


Fig. 125. Simple Form of Automatic Damper-Regulator. Operated by Lever Attached to Diaphragm, for Closing Dampers when Steam Pressure Reaches a Certain Point.

or balance pipe connects the top of the regulator with the low-pressure heating main, and high pressure is supplied to the pump as shown.

A sight-feed lubricator should be placed in this pipe above the automatic valve; and a valved by-pass should be placed around the regulator, for running the pump in case of accident or repairs. The oil separator should be drained through a special oil trap to a catch-basin or to the sewer; and the steam drum or any other low points or pockets in the high-pressure piping should be dripped to the return tank through suitable traps.

Means should be provided for draining all parts of the system to the sewer, and all traps and special apparatus should be by-passed. The return-pump should always be duplicated in a plant of any size, as a safeguard against accident; and the two pumps should be run alternately, to make sure that one is always in working order.

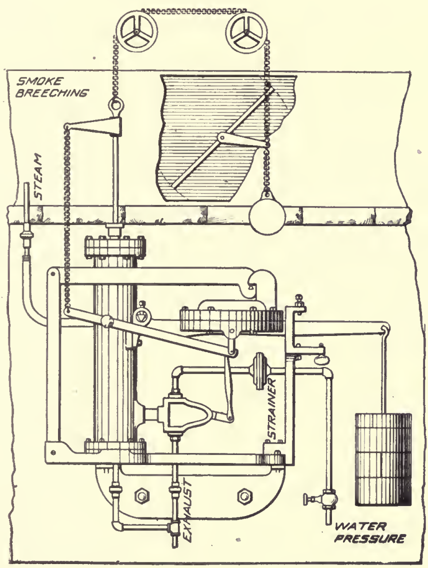


Fig. 126. Automatic Damper-Regulator Operated by Piston Actuated by Water-Pressure.

One piece of apparatus not shown in Fig. 127 is the feed-water heater. If all of the exhaust steam can be utilized for heating purposes, this is not necessary, as the cold water for feeding the boilers may be discharged into the return pipe and be pumped in with the condensation. In summertime, however, when the heating plant is not in use, a feed-water heater is necessary, as a large amount of heat

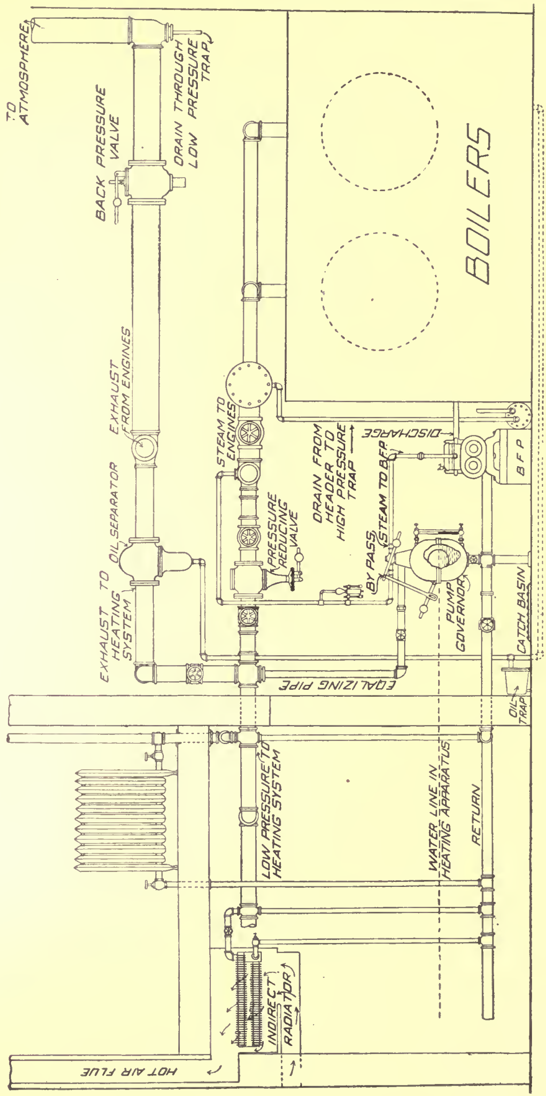
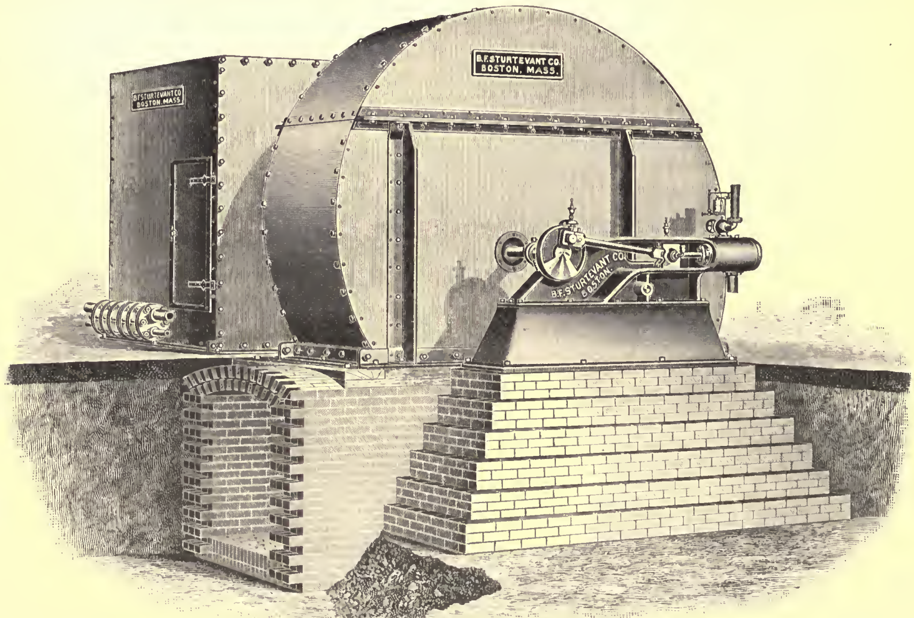


Fig. 127. General Method of Making Pipe Connections for Exhaust-Steam Heating.

which would otherwise be wasted may be saved in this way. The connections will depend somewhat upon the form of heater used; but in general a single connection with the heating main inside the back-pressure valve is all that is necessary. The condensation from the heater should be trapped to the sewer.



THE STURTEVANT STEAM HOT BLAST APPARATUS.
Three-Quarter Housing Steam Fan and Two-Group Corrugated Sectional Base Heater.

HEATING AND VENTILATION

PART III

VACUUM SYSTEMS

Low-Pressure or Vacuum Systems. In the systems of steam heating which have been described up to this point, the pressure carried has always been above that of the atmosphere, and the action of gravity has been depended upon to carry the water of condensation back to the boiler or receiver; the air in the radiators has been forced out through air-valves by the pressure of steam back of it. Methods will now be taken up in which the pressure in the heating system is less than the atmosphere, and where the circulation through the radiators is produced by suction rather than by pressure. Systems of this kind have several advantages over the ordinary methods of circulation under pressure. *First*—no back-pressure is produced at the engines when used in connection with exhaust steam; but rather there will be a reduction of pressure due to the partial vacuum existing in the radiators. *Second*—there is a complete removal of air from the coils and radiators, so that all portions are steam-filled and available for heating purposes. *Third*—there is complete drainage through the returns, especially those having long horizontal runs; and there is absence of water-hammer. *Fourth*—smaller return pipes may be used. The two older systems of this kind in common use are known as the Webster and Paul systems; other systems of recent introduction are described in the Instruction Paper on Steam and Hot-Water Fitting.

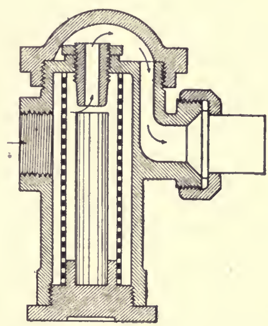


Fig. 128, Air Outlet-Valve for Radiator, Automatically Operated by Expansion and Contraction of Vulcanite Stem.

Webster System. This consists primarily of an automatic outlet-valve on each coil and radiator, connected with some form of suction apparatus such as a pump or ejector. One type of valve used is shown in section in Fig. 128, which replaces the usual hand-valve at the return end of the radiator. It is similar in construction to some of the air-valves already described, consisting of a rubber or vulcanite stem closing against a valve opening when made to expand by the presence of steam. When water or air fills the valve, the stem contracts and allows it to be sucked out as shown by the arrows. A perforated metal strainer surrounds the stem or expansion piece, to prevent dirt and sediment from clogging the valve.

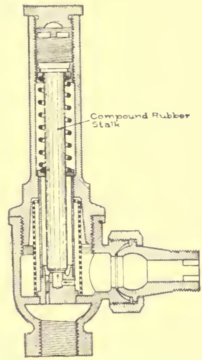


Fig. 129. Thermostat Attached to Angle-Valve with Top Removed.

Fig. 129 shows the valve—or *thermostat,* as it is called—attached to an ordinary angle-valve with the top removed; and Fig. 130 indicates the method of draining the bottoms of risers or the ends of mains.

Fig. 131 shows another form of this valve, called a *water-seal motor.* This is used under practically the same conditions as the one described above. Its action is as follows:

Ordinarily, the seal A is down, and the central tube-valve is resting upon the seat, closing the port K and preventing direct communication between the interior of the motor-body E and the outlet L. The outlet is attached to a pipe leading to a vacuum-pump, or other draining apparatus, which exhausts the space F above the seal through the annular space between the spindle B and the inside of the central tube G. The water of condensation, accumulating in the radiator or coil, passes into the chamber E, through the inlet C, rises in the chamber, and seals the space between the seal-shell A and the sleeve of the bonnet D. The differential pressure thus created causes the seal A to rise, lifting the end of the central tube off the seat, thus opening a clear passageway for the ejection of the water of condensation.

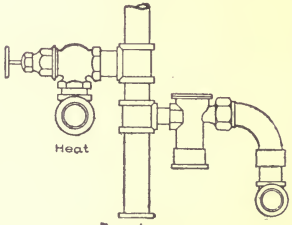


Fig. 130. Showing Method of Draining Bottoms of Risers or Ends of Mains.

When all the water of condensation has been drawn out of the radiator, the seal and tube are reseated by gravity, thus closing the port *K*, preventing waste or loss of steam; and the pressure is equalized above and below the seal because of the absence of water. This action is practically instantaneous. When the condensation is small in quantity, the discharge is intermittent and rapid.

The space between the seal *A* and the sleeve of the bonnet *D*, and the annular space between the central tube *G* and the spindle *B*,

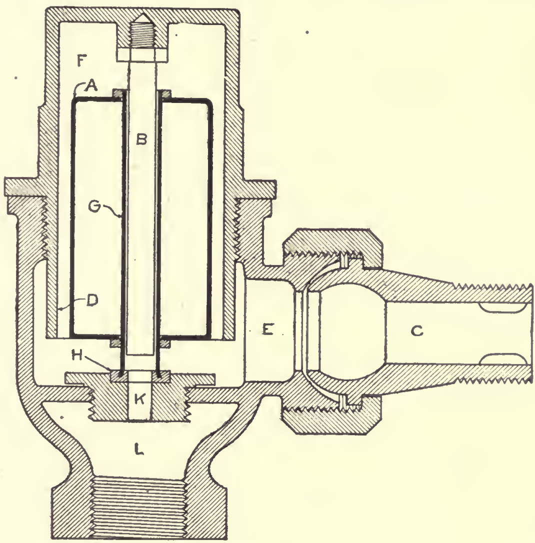


Fig. 131. Water-Seal Motor.

form a passageway through which the air is continually withdrawn by the vacuum pump or other draining apparatus.

The action outlined continues as long as water is present.

No adjustment whatever is necessary; the motor is entirely automatic.

One special advantage claimed for this system is that the amount of steam admitted to the radiators may be regulated to suit the requirements of outside temperature; and is possible without water-

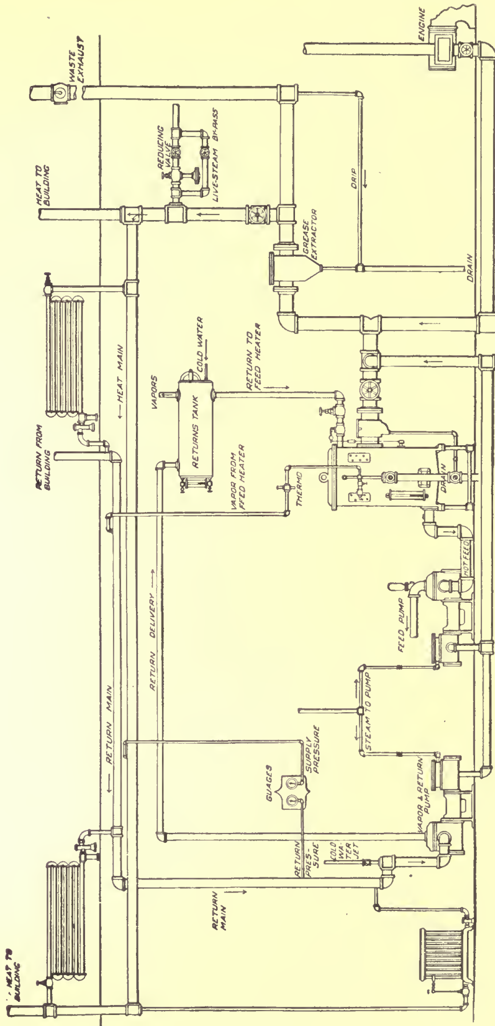


Fig. 132. Showing General Application of Webster System to Exhaust-Steam Heating.

logging or hammering. This may be done at will by closing down on the inlet supply to the desired degree. The result is the admission of a smaller amount of steam to the radiator than it is calculated to condense normally. The condensation is removed as fast as formed, by the opening of the thermostatic valve.

The general application of this system to exhaust heating is shown in Fig. 132. Exhaust steam is brought from the engine as shown; one branch is connected with a feed-water heater, while the other is carried upward and through a grease extractor, where it branches again, one line leading outboard through a back-pressure valve and the other connecting with the heating main. A live steam connection is made through a reducing valve, as in the ordinary system. Valved connections are made with the coils and radiators in the usual manner; but the return valves are replaced by the special thermostatic valves described above.

The main return is brought down to a vacuum pump which discharges into a *return tank,* where the air is separated from the water and passes off through the vapor pipe at the top. The condensation then flows into the feed-water heater, from which it is automatically pumped back into the boilers. The cold-water feed supply is connected with the return tank, and a small cold-water jet is connected into the suction at the vacuum pump for increasing the vacuum in the heating system by the condensation of steam at this point.

Paul System. In this system the suction is connected with the air-valves instead of the returns, and the vacuum is produced by means of a steam ejector instead of a pump. The returns are carried back to a receiving tank, and pumped back to the boiler in the usual manner. The ejector in this case is called the *exhauster.*

Fig. 133 shows the general method of making the pipe connections with the radiators in this system; and Fig. 134, the details of connection at the exhauster.

A A are the returns from the air-valves, and connect with the exhausters as shown. Live steam is admitted in small quantities through the valves *B B*; and the mixture of air and steam is discharged outboard through the pipe *C*. *D D* are gauges showing the pressure in the system; and *E E* are check-valves. The advantage of this system depends principally upon the quick removal of air from the various radiators and pipes, which constitutes the principal obstruction to circulation; the inductive action in many cases is sufficient to cause the system to operate somewhat below atmospheric pressure.

Where exhaust steam is used for heating, the radiators should

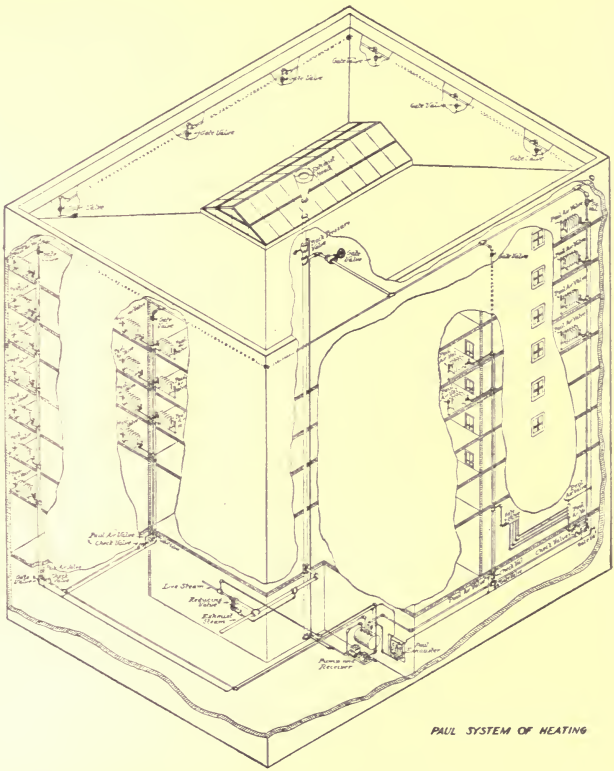


Fig. 133. Showing General Method of Making Pipe and Radiator Connections in Paul System.

be somewhat increased in size, owing to the lower temperature of the steam. It is common practice to add from 20 to 30 per cent to the sizes required for low-pressure live steam.

FORCED BLAST

In a system of forced circulation by means of a fan or blower the action is positive and practically constant under all usual conditions of outside temperature and wind action. This gives it a decided advantage over natural or gravity methods, which are af-

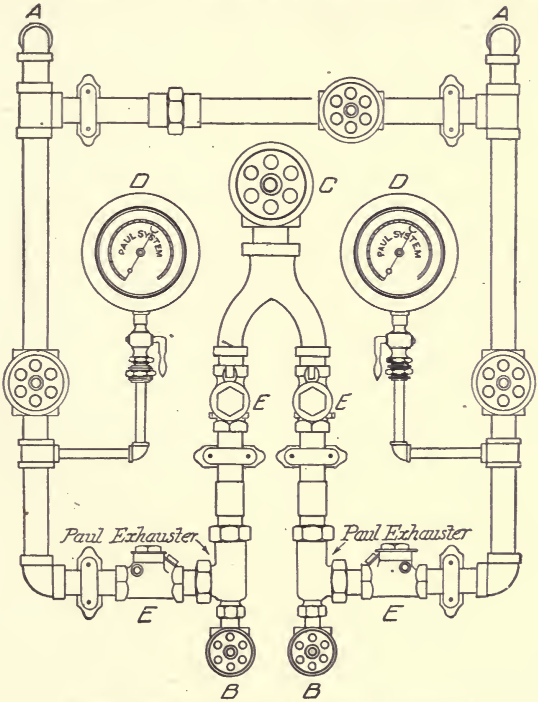


Fig. 134. Details of Connections at Exhauster, Paul System.

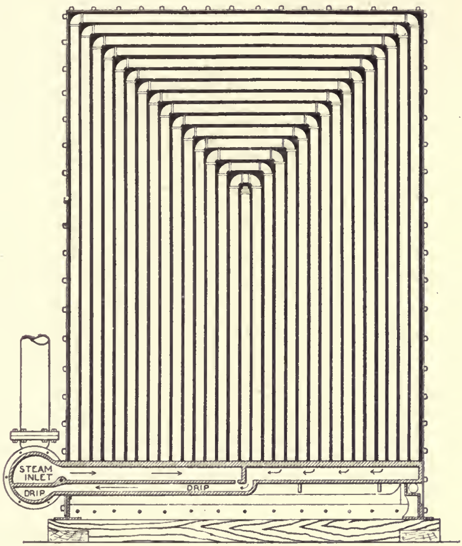
fected to a greater or less degree by changes in wind-pressure, and makes it especially adapted to the ventilation and warming of large buildings such as shops, factories, schools, churches, halls, theaters, etc., where large and definite air-quantities are required.

Exhaust Method. This consists in drawing the air out of a building, and providing for the heat thus carried away by placing steam coils under windows or in other positions where the inward leakage is supposed to be the greatest. When this method is used, a partial vacuum is created within the building or room, and all currents and leaks are inward; there is nothing to govern definitely the quality and place of introduction of the air, and it is difficult to provide suitable means for warming it.

Plenum Method. In this case the air is forced into the building, and its quality, temperature, and point of admission are completely under control. All spaces are filled with air under a slight pressure, and the leakage is outward, thus preventing the drawing of foul air into the room from any outside source. But above all, ample opportunity is given for properly warming the air by means of heaters, either in direct connection with the fan or in separate passages leading to the various rooms.

Form of Heating Surface. The best type of heater for any particular case will depend upon the volume and final temperature of the air, the steam pressure, and the available space. When the air is to be heated to a high temperature for both warming and ventilating a building, as in the case of a shop or mill, heaters of the general form shown in Figs. 135, 136, and 137 are used. These may also be adapted to all classes of work by varying the proportions as required. They can be made shallow and of large superficial area, for the comparatively low temperatures used in purely ventilating work; or deeper, with less height and breadth, as higher temperatures are required.

Fig. 135 shows in section a heater of this type, and illustrates the circulation of steam through it. It consists of sectional cast-iron bases with loops of wrought-iron pipe connected as shown. The steam enters the upper part of the bases or headers, and passes up one side of the loops, then across the top and down on the other side, where the condensation is taken off through the return drip, which is separated from the inlet by a partition. These heaters are made up in sections of 2 and 4 rows of pipes each. The height varies from $3\frac{1}{2}$ to 9 feet, and the width from 3 feet to 7 feet in the standard sizes. They are usually made up of 1-inch pipe, although $1\frac{1}{4}$-inch is commonly used in the larger sizes. Fig. 136 shows another form; in this case all the loops are made of practically the same length by the special form of construction shown. This is claimed to prevent the short-circuiting of steam through the shorter loops, which causes the outer pipes to remain cold. This form of heater is usually encased in a



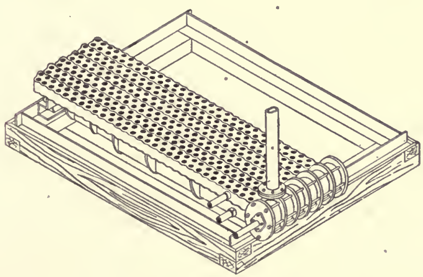


Fig. 135. Showing Circulation of Steam in Large Coil-Pipe Radiator for Heating Mills, Shops, etc.

sheet-steel housing as shown in Fig. 137, but may be supported on a foundation between brick walls if desired.

Fig. 138 shows a special form of heater particularly adapted to ventilating work where the air does not have to be raised above 70 or 80 degrees. It is made up of 1-inch wrought-iron pipe connected

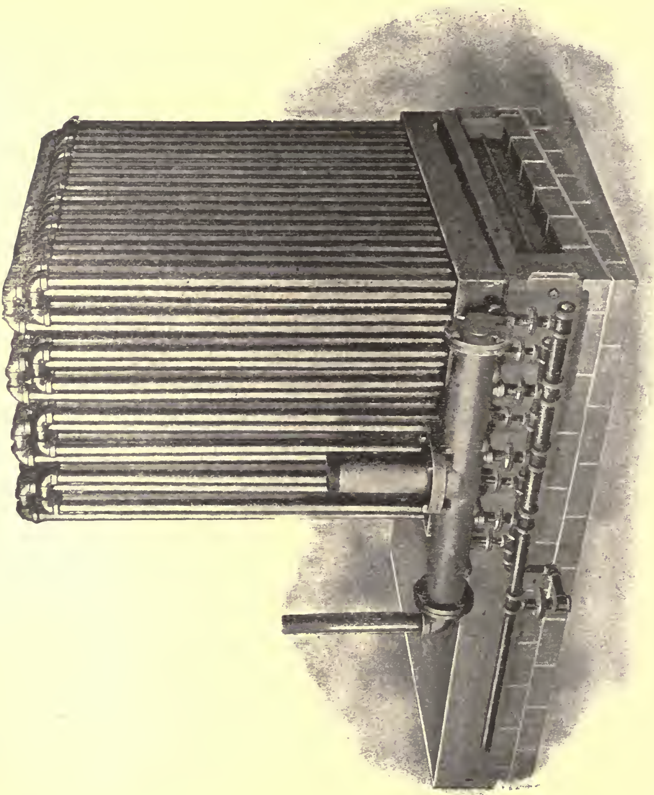
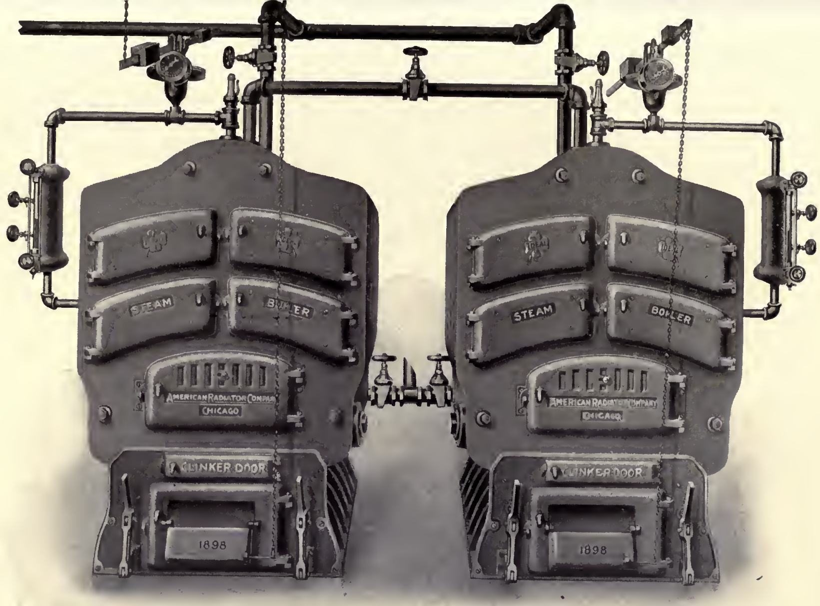


Fig. 136. Another Type of Large Coil Radiator for Mills, Factories, etc.

with supply and return headers; each section contains 14 pipes, and they are usually made up in groups of 5 sections each. These coils are supported upon tee-irons resting upon a brick foundation. Heat-



A BATTERY OF IDEAL STEAM BOILERS SHOWING METHOD OF YOKING THE MAIN SUPPLY AND RETURN PIPE.
American Radiator Company.

ers of this form are usually made to extend across the side of a room with brick walls at the sides, instead of being encased in steel housings.

Fig. 139 shows a front view of a cast-iron sectional heater for use under the same conditions as the pipe heaters already described. This heater is made up of several banks of sections, like the one shown in the cut, and enclosed in a steel-plate casing.

Cast-iron indirect radiators of the pin type are well adapted for use in connection with mechanical ventilation, and also for heating

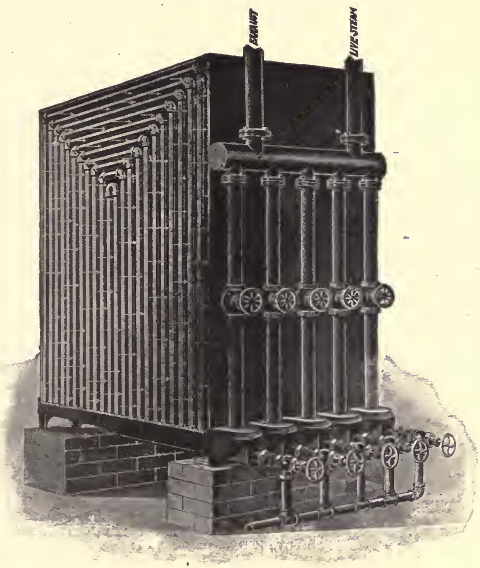


Fig. 137. Large Coil-Pipe Radiator Encased in Sheet-Steel Housing.

where the air-volume is large and the temperature not too high, as in churches and halls. They make a convenient form of heater for schoolhouse and similar work, for, being shallow, they can be supported upon I-beams at such an elevation that the condensation will be returned to the boilers by gravity.

In the case of vertical pipe heaters, the bases are below the water-line of the boilers, and the condensation must be returned by the use of pumps and traps.

Efficiency of Pipe Heaters. The efficiency of the heaters used in connection with forced blast varies greatly, depending upon the temperature of the entering air, its velocity between the pipes, the temperature to which it is raised, and the steam pressure carried in the heater. The general method in which the heater is made up is also an important factor.

In designing a heater of this kind, care must be taken that the free area between the pipes is not contracted to such an extent that an excessive velocity will be required to pass the given quantity of

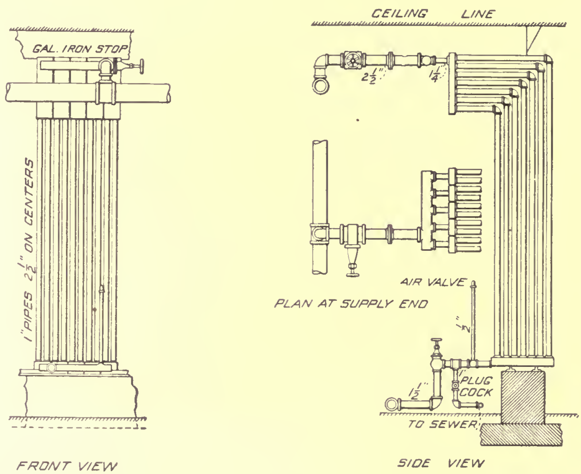


Fig. 138. Heater Especially Adapted to Ventilation where Air does not Have to be Heated above 70 to 80 degrees F.

air through it. In ordinary work it is customary to assume a velocity of 800 to 1,000 feet per minute; higher velocities call for a greater pressure on the fan, which is not desirable in ventilating work.

In the heaters shown, about .4 of the total area is free for the passage of air; that is, a heater 5 feet wide and 6 feet high would have a total area of $5 \times 6 = 30$ square feet, and a free area between the pipes of $30 \times .4 = 12$ square feet. The depth or number of rows of pipe does not affect the free area, although the friction is increased and additional work is thrown upon the fan. The efficiency in any given heater will be increased by increasing the velocity of the air through it; but the final temperature will be diminished; that is, a larger quantity of air will be heated to a lower temperature in the second case, and, while the total heat given off is greater, the air-quantity increases more rapidly than the heat-quantity, which causes a drop in temperature.

Increasing the number of rows of pipe in a heater, with a constant air-quantity, increases the final temperature of the air, but diminishes the efficiency of the heater, because the average difference in temperature between the air and the steam is less. Increasing the steam pressure in the heater (and consequently its temperature) increases both the final temperature of the air and the efficiency of the heater. Table XXX has been prepared from different tests, and may be used as a guide in computing probable results under ordinary working conditions. In this table it is assumed that the air enters the heater at a temperature of zero and passes between the pipes with a velocity of 800 feet per minute. Column 1 gives the number of rows of pipe in the heater, ranging from 4 to 20 rows; and columns 2, 3, and 4, show the final temperature to which the entering air will be raised from zero under various pressures. Under 5 pounds pressure, for example, the rise in temperature ranges from 30 to 140 degrees; under 20 pounds, 35 to 150 degrees; and under 60 pounds, 45 to 170 degrees. Columns 5, 6, and 7 give approximately the corresponding efficiency of the heater. For example, air passing through a heater 10 pipes deep and carrying 20 pounds pressure, will be raised to a temperature of 90 degrees, and the heater will have an efficiency of 1,650 B. T. U. per square foot of surface per hour.

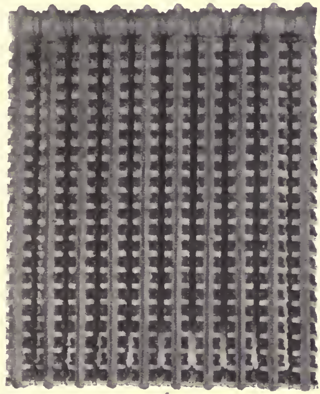


Fig. 139. Front View of Cast-Iron Sectional Heater. The Banks of Sections are Enclosed in a Steel-Plate Casing.

TABLE XXX

Data Concerning Pipe Heaters

Temperature of entering air, zero.—Velocity of air between the pipes, 800 feet per minute.

Rows of Pipe Deep	Temperature to which Air will be Raised from Zero			Efficiency of Heating Surface in B. T. U., per Square Foot per Hour		
	Steam Pressure in Heater			Steam Pressure in Heater		
	5 lbs.	20 lbs.	60 lbs.	5 lbs.	20 lbs.	60 lbs.
4	30	35	45	1,600	1,800	2,000
6	50	55	65	1,600	1,800	2,000
8	65	70	85	1,500	1,650	1,850
10	80	90	105	1,500	1,650	1,850
12	95	105	125	1,500	1,650	1,850
14	105	120	140	1,400	1,500	1,700
16	120	130	150	1,400	1,500	1,700
18	130	140	160	1,300	1,400	1,600
20	140	150	170	1,300	1,400	1,600

For a velocity of 1,000 feet, multiply the *temperatures* given in the table by .9, and the *efficiencies* by 1.1.

Example. How many square feet of radiation will be required to raise 600,000 cubic feet of air per hour from zero to 80 degrees, with a velocity through the heater of 800 feet per minute and a steam pressure of 5 pounds? What must be the total area of the heater front, and how many rows of pipes must it have?

Referring back to the formula for heat required for ventilation, we have

$$\frac{600{,}000 \times 80}{55} = 872{,}727 \text{ B. T. U. required.}$$

Referring to Table XXX, we find that for the above conditions a heater 10 pipes deep is required, and that an efficiency of 1,500 B. T. U. will be obtained. Then $\frac{872{,}727}{1{,}500} = 582$ square feet of surface required, which may be taken as 600 in round numbers. $\frac{600{,}000}{60} = 10{,}000$ cubic feet of air per minute; and $\frac{10{,}000}{800} = 12.5$ square feet of free area required through the heater. If we assume .4 of the total heater front to be free for the passage of air, then $\frac{12.5}{.4} = 31$ square feet, the total area required.

For convenience in estimating the approximate dimensions of a heater, Table XXXI is given. The standard heaters made by different manufacturers vary somewhat, but the dimensions given in the table represent average practice. Column 3 gives the square feet of heating surface in a single row of pipes of the dimensions given in columns 1 and 2; and column 4 gives the free area between the pipes.

TABLE XXXI

Dimensions of Heaters

Width of Section	Height of Pipes	Square Feet of Surface	Free Area through Heater in Sq. Ft.
3 feet	3 feet 6 inches	20	4.2
3 "	4 " 0 "	22	4.8
3 "	4 " 6 "	25	5.4
3 "	5 " 0 "	28	6.0
4 "	4 " 6 "	34	7.2
4 "	5 " 0 "	38	8.0
4 "	5 " 6 "	42	8.8
4 "	6 " 0 "	45	9.6
5 "	5 " 6 "	52	11.0
5 "	6 " 0 "	57	12.0
5 "	6 " 6 "	62	13.0
5 "	7 " 0 "	67	14.0
6 "	6 " 6 "	75	15.6
6 "	7 " 0 "	81	16.8
6 "	7 " 6 "	87	18.0
6 "	8 " 0 "	92	19.2
7 "	7 " 6 "	98	21.0
7 "	8 " 0 "	108	22.4
7 "	8 " 6 "	109	23.8
7 "	9 " 0 "	116	25.2

In calculating the total height of the heater, add 1 foot for the base.

These sections are made up of 1-inch pipe, except the last or 7-foot sections, which are made of $1\frac{1}{4}$-inch pipe.

Using this table in connection with the example just given, we should look in the last column for a section having a free area of 12.5 square feet; here we find that a 5 feet by 6 feet 6 inches section has a free opening of 13 square feet and a radiating surface of 62 square feet. The conditions call for 10 rows of pipes and $10 \times 62 = 620$ square feet of radiating surface, which is slightly more than called for, but which would be near enough for all practical purposes.

EXAMPLE FOR PRACTICE

Compute the dimensions of a heater to warm 20,000 cubic feet of air per minute from 10 below zero to 70 degrees above, with 5 pounds steam pressure.

Ans. 1,164 sq. ft. of rad. surface 10 pipes deep.
25 sq. ft. free area through heater.

Use twenty 5 ft. by 6 ft. sections, side by side, which gives 24 square feet area and 1,140 square feet of surface.

The general method of computing the size of heater for any given building is the same as in the case of indirect heating. First obtain the B. T. U. required for ventilation, and to that add the heat loss through walls, etc.; and divide the result by the efficiency of the heater under the given conditions.

Example. An audience hall is to be provided with 400,000 cubic feet of air per hour. The heat loss through walls, etc., is 250,000 B.T.U. per hour in zero weather. What will be the size of heater, and how many rows of pipe deep must it be, with 20 pounds steam pressure?

$$\frac{400{,}000 \times 70}{55} = 509{,}090 \text{ B. T. U. for ventilation.}$$

Therefore $250{,}000 + 509{,}090 = 759{,}090$ B. T. U., total to be supplied.

We must next find to what temperature the entering air must be raised in order to bring in the required amount of heat, so that the number of rows of pipe in the heater may be obtained and its corresponding efficiency determined. We have entering the room for purposes of ventilation, 400,000 cubic feet of air every hour, at a temperature of 70 degrees; and the problem now becomes: To what temperature must this air be raised to carry in 250,000 B. T. U. additional for warming?

We have learned that 1 B. T. U. will raise 55 cubic feet of air 1 degree. Then 250,000 B. T. U. will raise $250{,}000 \times 55$ cubic feet of air 1 degree.

$$\frac{250{,}000 \times 55}{400{,}000} = 34 +$$

The air in this case must be raised to $70 + 34 = 104$ degrees, to provide for both ventilation and warming. Referring to Table XXX, we find that a heater 12 pipes deep will be required, and that the corresponding efficiency of the heater will be 1,650 B. T. U. Then $\frac{759{,}090}{1{,}650}$ = 460 square feet of surface required.

Efficiency of Cast-Iron Heaters. Heaters made up of indirect pin radiators of the usual depth, have an efficiency of at least 1,500 B. T. U., with steam at 10 pounds pressure, and are easily capable of warming air from zero to 80 degrees or over when computed on this basis. The free space between the sections bears such a relation to the heating surface that ample area is provided for the flow of air through the heater, without producing an excessive velocity.

The heater shown in Fig. 139 may be counted on for an efficiency at least equal to that of a pipe heater; and in computing the depth, one row of sections may be taken as representing 4 rows of pipe.

Pipe Connections. In the heater shown in Fig. 135, all the sections take their supply from a common header, the supply pipe connecting with the top, and the return being taken from the lower division at the end, as shown.

In Fig. 137 the base is divided into two parts, one for live steam, and the other for exhaust. The supply pipes connect with the upper compartments, and the drips are taken off as shown. Separate traps should be provided for the two pressures.

The connections in Fig. 136 are similar to those just described, except that the supply and return headers, or bases, are drained through separate pipes and traps, there being a slight difference in pressure between the two, which is likely to interfere with the proper drainage if brought into the same one. This heater is arranged to take exhaust steam, but has a connection for feeding in live steam through a reducing valve if desired, the whole heater being under one pressure.

In heating and ventilating work where a close regulation of temperature is required, it is usual to divide the heater into several sections, depending upon its size, and to provide each with a valve in the supply and return. In making the divisions, special care should be taken to arrange for as many combinations as possible. For example, a heater 10 pipes deep may be made up of three sections—one of

2 rows, and two of 4 rows each. By means of this division, 2, 4, 6, 8, or 10 rows of pipe can be used at one time, as the outside weather conditions may require.

When possible, the return from each section should be provided with a water-seal two or three feet in depth. In the case of overhead heaters, the returns may be sealed by the water-line of the boiler or by the use of a special water-line trap; but vertical pipe heaters resting on foundations near the floor are usually provided with siphon loops extending into a pit. If this arrangement is not convenient, a separate trap should be placed on the return from each section. The main return, in addition to its connection with the boiler or

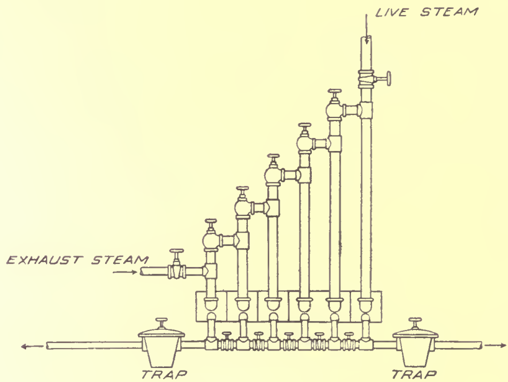


Fig. 140. Heater Made Up of Interchangeable Sections.

pump receiver, should have a connection with the sewer for blowing out when steam is first turned on. Sometimes each section is provided with a connection of this kind.

Large automatic air-valves should be connected with each section; and it is well to supplement these with a hand pet-cock, unless individual blow-off valves are provided as described above.

If the fan is driven by a steam engine, provision should be made for using the exhaust in the heater; and part of the sections should be so valved that they may be supplied with either exhaust or live steam.

Fig. 140 shows an arrangement in which all of the sections are interchangeable.

From 50 to 60 square feet of radiating surface should be provided in the exhaust portion of the heater for each engine horse-power, and should be divided into at least three sections, so that it can be proportioned to the requirements of different outside temperatures.

Pipe Sizes. The sizes of the mains and branches may be computed from the tables already given in Part II, taking into account the higher efficiency of the heater and the short runs of piping.

Table XXXII, based on experience, has been found to give satisfactory results when the apparatus is near the boilers. If the main supply pipe is of considerable length, its diameter should be checked by the method previously given.

TABLE XXXII

Pipe Sizes

Square Feet of Surface	Diameter of Steam Pipe	Diameter of Return
150	2 inches	1¼ inches
300	2½ "	1½ "
500	3 "	2 "
700	3½ "	2 "
1,000	4 "	2½ "
2,000	5 "	2½ "
3,000	6 "	3 "

Heaters of the patterns shown in Figs. 135, 136, and 137 are usually tapped at the factory for high or low pressure as desired, and these sizes may be followed in making the pipe connections.

The sizes marked on Fig. 136 may be used for all ordinary work where the pressure runs from 5 to 20 pounds; for pressures above that, the supply connections may be reduced one size.

FANS

There are two types of fans in common use, known as the *centrifugal fan* or *blower*, and the *disc fan* or *propeller*. The former consists of a number of straight or slightly curved blades extending radially from an axis, as shown in Fig. 141. When the fan is in motion, the air in contact with the blades is thrown outward by the action of centrifugal force, and delivered at the circumference or periphery of the wheel. A partial vacuum is thus produced at the center of the wheel, and air from the outside flows in to take the place of that which has been discharged.

Fig. 142 illustrates the action of a centrifugal fan, the arrows showing the path of the air. This type of fan is usually enclosed in a steel-plate casing of such form as to provide for the free movement of the air as it escapes from the periphery of the wheel. An opening in the circumference of the casing serves as an outlet into the distributing ducts which carry the air to the various rooms to be ventilated.

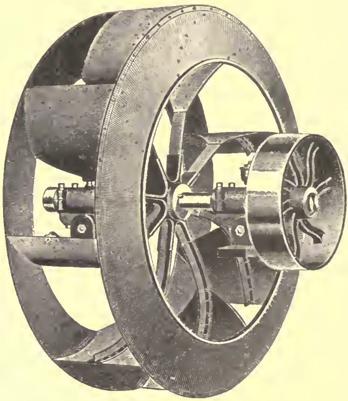


Fig. 141. Centrifugal Fan or Blower.

A fan with casing, is shown in Fig. 143; and a combined heater and fan, with direct-connected engine, is shown in Fig. 144.

The discharge opening can be located in any position desired, either up, down, top horizontal, bottom horizontal, or at any angle.

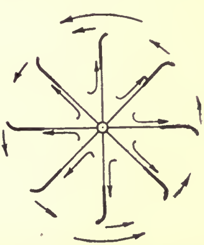


Fig. 142. Illustrating Action of Centrifugal Fan. The Arrows Show the Path of the Air.

Where the height of the fan room is limited, a form called the *three-quarter housing* may be used, in which the lower part of the casing is replaced by a brick or cemented pit extending below the floor-level as shown in Fig. 145.

Another form of centrifugal fan is shown in Fig. 146. This is known as the *cone fan*, and is commonly placed in an opening in a brick wall, and discharges air from its entire periphery into a room called a *plenum chamber*, with which the various distributing ducts connect.

This fan is often made double by placing two wheels back to back and surrounding them with a steel casing in a similar manner to the one shown in Fig. 143.

Cone fans are particularly adapted to church and schoolhouse work, as they are capable of moving large volumes of air at moderate speeds.

Fig. 147 shows a form of small direct-connected exhauster commonly used for ventilating toilet-rooms, chemical hoods, etc.

Centrifugal fans are used almost exclusively for supplying air for the ventilation of buildings, and for forced-blast heating. They are also used as exhausters for removing the air from buildings in cases where there is considerable resistance due to the small size or excessive length of the discharge ducts.

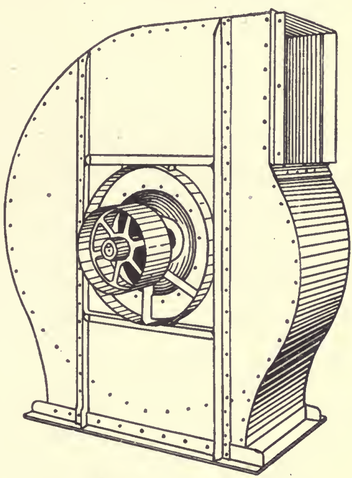


Fig. 143. Centrifugal Fan with Casing.

General Proportions. The general form of a fan wheel is shown in Fig. 141, which represents a single spider wheel with curved blades. Those over 4 feet in diameter usually have two spiders, while fans of large size are often provided with three or more. The number of floats or blades commonly varies from six to twelve, depending upon the diameter of the fan. They are made both curved and straight; the former, it is claimed, run more quietly, but, if curved too much, will not work so well against a high pressure as the latter form.

The relative proportions of a fan wheel vary somewhat in the case of different makes. The following are averages taken from fans of different sizes as made by several well-known manufacturers for general ventilating and similar work:

Width of fan at center = Diameter $\times$ .52
Width of fan at perimeter = Width at center $\times$ .8
Diameter of inlet = Diameter of wheel $\times$ .68

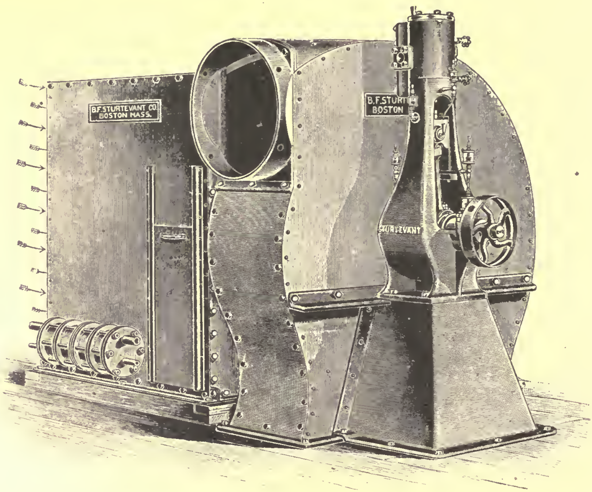


Fig. 144. Combined Heater and Centrifugal Fan with Direct-Connected Engine.

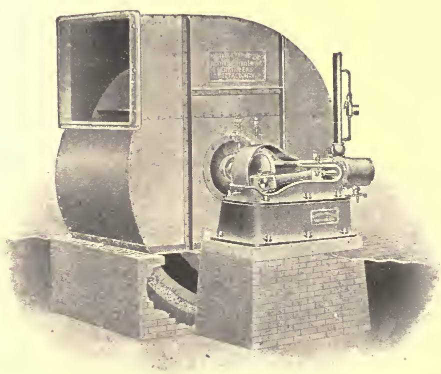


Fig. 145. Centrifugal Fan in "Three-Quarter Housing." Used where Headroom is Limited; Extra Space Provided by Pit under Floor-Level.

Fans are made both with double and with single inlets, the former being called *blowers* and the latter *exhausters.* The size of a fan is commonly expressed in inches, which means the approximate height of the casing of a full-housed fan. The diameter of the wheel is usually expressed in feet, and can be found in any given case by dividing the size in inches by 20. For example, a 120-inch fan has a wheel 120 ÷ 20 = 6 feet in diameter.

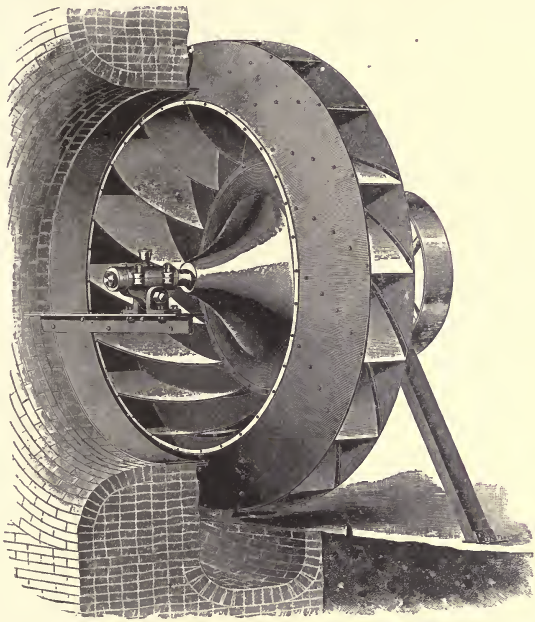


Fig. 146. "Cone" Fan. Discharges through Opening in Wall into a "Plenum Chamber" Connecting with Distributing Ducts.

Theory of Centrifugal Fans. The action of a fan is affected to such an extent by the various conditions under which it operates, that it is impossible to give fixed rules for determining the exact results to be expected in any particular instance. This being the case, it seems best to take up the matter briefly from a theoretical standpoint, and then show what corrections are necessary in the case of a given fan under actual working conditions.

There are various methods for determining the capacity of a fan at different speeds, and the power necessary to drive it; each manufacturer has his own formulæ for this purpose, based upon tests of his own particular fans. The methods given here apply in a general way to fans having proportions which represent the *average* of several standard makes; and the results obtained will be

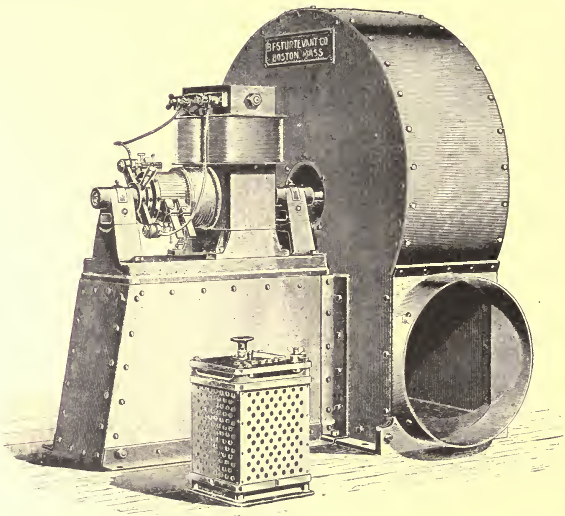


Fig. 147. Small, Direct-Connected Exhauster for Ventilating Toilet-Rooms, Chemical Hoods, etc.

found to correspond well with those obtained in practice under ordinary conditions.

As already stated, the rotation of a fan of this type sets in motion the air between the blades, which, by the action of centrifugal force, is delivered at the periphery of the wheel into the casing surrounding it. As the velocity of flow through the discharge outlet depends upon the pressure or head within the casing, and this in turn upon the velocity of the blades, it becomes necessary to examine briefly into the relations existing between these quantities.

Pressure. The pressure referred to in connection with a fan, is that in the discharge outlet, and represents the force which drives the air through the ducts and flues. The greater the pressure with a given resistance in the pipes, the greater will be the volume of air delivered; and the greater the resistance, the greater the pressure required to deliver a given quantity.

The pressure within a fan casing is caused by the air being thrown from the tips of the blades, and varies with the velocity of rotation; that is, the higher the speed of the fan, the greater will be the pressure produced. Where the dimensions of a fan and casing are properly proportioned, the velocity of air-flow through the outlet will be the same as that of the tips of the blades, and the pressure within the casing will be that corresponding to this velocity.

Table XXXIII gives the necessary speed for fans of different diameters to produce different pressures, and also the velocity of air-flow due to these pressures.

TABLE XXXIII

Fan Speeds, Pressures, and Velocities of Air-Flow

Pressure, in Ounces per Sq. Inch	Diameter of Fan Wheel, in Feet								Velocity of Flow, in Feet per Minute
	3	4	5	6	7	8	9	10	
	Revolutions per Minute								
$\frac{1}{4}$	274	206	164	137	117	103	92	82	2,585
$\frac{3}{8}$	336	252	202	168	144	126	112	101	3,165
$\frac{1}{2}$	338	291	232	194	166	146	129	116	3,653
$\frac{5}{8}$	433	325	260	217	186	163	144	130	4,084

The application of this table will be made plain by a brief discussion of *blast area.*

Blast Area. When the outlet from a fan casing is small, the air will pass out with a velocity equal to that of the tips of the blades; and the pressure within the casing will be that corresponding to the tip velocity. That is, a 3-foot fan wheel revolving at a speed of 274 revolutions per minute will produce a pressure within the fan casing of $\frac{1}{4}$ ounce per square inch, and will cause a velocity of flow through the discharge outlet of 2,585 feet per minute (see Table XXXIII).

Now, if the opening be slowly increased, while the speed of the fan remains constant, the air will continue to flow with the same velocity until a certain area of outlet is reached. If the outlet be still further increased, the pressure in the casing will begin to drop, and the velocity of outflow become less than the tip velocity. The effective area of outlet at the point when this change begins to take place, is called the *blast area* or *capacity area* of the fan. This varies somewhat with different types and makes of fans; but for the common form of blower, it is approximately $\frac{1}{3}$ of the projected area of the fan opening at the periphery—that is, $\frac{Dw}{3}$, in which D is the diameter of the fan wheel, and w its width at the periphery. It has already been stated under "General Proportions" that $W = .52\,D$, and $w = .8\,W$; so that we may write $A = \frac{D \times .8\,W}{3} = \frac{D \times .8 \times .52\,D}{3} = .14\,D^2$, in which A = the blast area, and D the diameter of the fan.

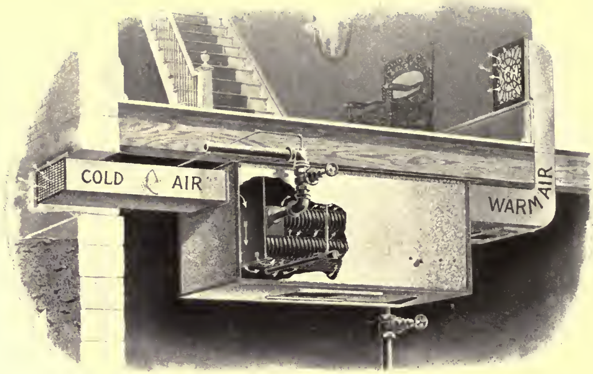
As a matter of fact, the outlet of a fan casing is always made larger than the blast area; and the result is that the pressure drops below that due to the tip velocity, and the velocity of flow through the outlet becomes less than that given in the last column of Table XXXIII for any given speed of fan.

Effective Area of Outlet. The size of discharge outlet varies somewhat for different makes; but for a large number of fans examined it was found to average about 2.22 times the blast area as computed by the above method. When air or a liquid flows through an orifice, the stream is more or less contracted, depending upon the form of the orifice.

In the case of a fan outlet, the *effective area* may be taken as about .8 of the actual area. This makes the effective area of a fan outlet equal to $.8 \times 2.22 = 1.78$ times the blast area.

Table XXXIV gives the effective areas of fans of different diameter as computed by the above method. That is, Effective area $= .14D^2 \times 1.78 = .25D^2$.

Speed. We have seen that when the discharge outlet is made larger than the blast area, the pressure within the fan casing drops below that due to the tip velocity; so that, in order to bring the pressure up to its original point, the speed of the fan must be increased above that given in Table XXXIII.



EXCELSIOR PATTERN RADIATOR USED IN THE INDIRECT METHOD OF WARMING
American Radiator Company

METHOD OF INDIRECT WARMING AND VENTILATION, SHOWING ROTARY CIRCULATION OF AIR
American Radiator Company

TABLE XXXIV

Effective Areas of Fans

Diameter of Fan, in Feet	Effective Area of Outlet, in Square Feet
3	2.3
4	4.0
5	6.3
6	9.1
7	12.3
8	16.0
9	20.4
10	25.2

Tests upon a fan of practically the same proportions as those previously given, show that, when the effective outlet area is made 1.78 the blast area, the speed must be increased 1.2 times in order to keep the pressure at the same point as when the outlet is equal to or less than the blast area.

Capacity. The capacity of a fan is the volume of air discharged in a given time, and is usually expressed in cubic feet per minute. It is equal to the effective area of discharge *multiplied by* the velocity of flow through it.

Example. At what speed must a 6-foot fan be run to maintain a pressure of ½ ounce, and what volume of air will be delivered per minute?

From Table XXXIII we find that a 6-foot fan must run at a speed of 194 revolutions per minute to maintain the given pressure when the outlet is equal to the blast area, or 194 × 1.2 = 233 revolutions per minute under actual conditions. The velocity of flow through the outlet at ½ ounce pressure, is 3,653 feet per minute (Table XXXIII); and the effective area of outlet of a 6-foot fan is 9.1 square feet (Table XXXIV). Therefore the volume of air delivered per minute is equal to 9.1 × 3,653 = 33,242 cubic feet.

Example. It is desired to move 52,000 cubic feet of air per minute at a pressure of ¼ ounce. What size and speed of fan will be required? Looking in Table XXXIII, we find that the velocity through the fan outlet for ¼-ounce pressure is 2,585, which calls for an outlet area of 52,000 ÷ 2,585 = 20.1 square feet. Looking in Table XXXIV, we find this corresponds very nearly to a 9-foot fan, which is the size called for. Referring again to Table XXXIII, the speed necessary to maintain the required pressure under the given conditions is found to be 92 × 1.2 = 110 revolutions per minute.

Effect of Resistance. Thus far it has been assumed that the fan was discharging into the open air against atmospheric pressure. The effect of adding a resistance by connecting it with a series of ventilating ducts, is the same as partially closing the discharge outlet. Carefully conducted tests upon this type of fan have shown that the reduction of air-flow is very nearly in proportion to the reduction of the discharge area. That is, if the outlet of the fan is closed to one-half its original area, the quantity of air discharged will be practically one-half that delivered by the fan with a free opening. The effect of attaching a fan to the ventilating flues of a building like a schoolhouse, church, or hall, where the ducts have easy bends and where the velocity of air-flow through them is not over 1,000 to 1,200 feet per minute, is about the same as reducing the outlet 20 per cent. For factories with deep heaters and smaller ducts, where the velocity runs up to 1,500 or 1,800 feet per minute, the effect is equivalent to closing the outlet at least 30 per cent, and even more in very large buildings.

For schoolhouses and similar work a fan should not be run much above the speed necessary to maintain a pressure of $\frac{3}{8}$ ounce at the outlet. Higher speeds are accompanied with greater expenditure of power, and are likely to produce a roaring noise or to cause vibration. A much lower speed does not provide sufficient pressure to give proper control of the air-distribution during strong winds. For factories, a higher pressure of $\frac{5}{8}$ to $\frac{3}{4}$ ounce is more generally employed.

Actually the pressure is increased slightly by restricting the outlet at constant speed; but this is seldom taken into account in ventilating work, as volume, speed, and power are the quantities sought.

Example. A school building requires 32,000 cubic feet of air per minute. What size and speed of fan will be required?

If the resistance of the ducts and flues is equivalent to cutting down the discharge outlet 20 per cent, we must make the computations for a fan which will discharge 32,000 ÷ .8 = 40,000 cubic feet in free air.

Looking in Table XXXIII, we find the velocity for $\frac{3}{8}$-ounce pressure to be 3,165 feet per minute; therefore the size of fan outlet must be 40,000 ÷ 3,165 = 12.6 square feet, which, from Table XXXIV, we find corresponds very nearly to a 7-foot fan.

Referring again to Table XXXIII, the required speed is found to be 144 × 1.2 = 173 revolutions per minute.

Example. A factory requires 21,000 cubic feet of air per minute for warming and ventilating. What size and speed of fan will be required?

21,000 ÷ .7 = 30,000, the volume to provide for with a fan discharging into free air. Assuming a pressure of $\frac{5}{8}$ ounce, the velocity will be 4,084 feet per minute, from which the area of outlet is found to be 30,000 ÷ 4,084 = 7.3 square feet. This, we find, does not correspond to any of the sizes given in Table XXXIV. As standard fans are not usually made in half-sizes above 5 feet, we shall use a 5-foot fan and run it at a higher speed.

A 5-foot fan has an outlet area of 6.3 square feet, and at $\frac{5}{8}$-ounce pressure it would deliver 6.3 × 4,084 = 25,729 cubic feet of air per minute, at a speed of 260 × 1.2 = 312 revolutions per minute. The volume of air delivered by a fan varies approximately as the speed; so, in order to bring the volume up to the required 30,000, the speed must be increased by the ratio 30,000 ÷ 25,729 = 1.16, making the final speed 312 × 1.16 = 362 revolutions per minute. In the same way, a 6-foot fan could have been used and run at a proportionally lower speed.

Power Required. The work done by a fan in moving air is represented by the pressure exerted, *multiplied by* the distance through which it acts.

Table XXXV gives the horse-power required for moving the air which will flow through each square foot of the effective outlet area, under different pressures.

This table gives only the power necessary for *moving* the air, and does not take into consideration the friction of the air in passing through the fan, nor that of the fan itself.

The efficiency of a fan varies with the speed, the size of outlet, and the pressure against which the fan is working. Under favorable conditions, with properly proportioned fans, we may count on an efficiency of about .35.

Example. What horse-power will be required to drive an 8-foot fan at such a speed as to maintain a pressure of $\frac{1}{2}$ ounce?

An 8-foot fan has an outlet area of 16 square feet (Table XXXIV); and from Table XXXV we find that .5 horse-power is required to move the air which will flow through each square foot of outlet under

TABLE XXXV

Power Required for Moving Air under Different Pressures

Pressure in Ounces per Square Inch	Horse-Power for Moving Air which will Flow through Each Square Foot of Effective Outlet Area
$\frac{1}{4}$	.18
$\frac{3}{8}$	.33
$\frac{1}{2}$	.50
$\frac{5}{8}$	.70

$\frac{1}{2}$-ounce pressure. Therefore the power required to move the air alone is $16 \times .5 = 8$, and the total horse-power is $8 \div .35 = 23$.

Effect of Resistance. In the above case, it is assumed that the fan is discharging into free air. If a resistance is added, the effect is the same as partially closing the outlet, and the volume of air moved and the horse-power required are both reduced in very nearly the same proportion. This reduction, as already stated, may be taken as 20 per cent for schoolhouse and similar work, and 30 per cent for factories.

For example, if the fan just considered was to be used for ventilating a schoolhouse, delivering air under a pressure of $\frac{1}{2}$ ounce, the necessary horse-power would be only $23 \times .8 = 18.4$. If used for a factory, delivering air under a pressure of $\frac{5}{8}$ ounce, the required horse-power would be $\frac{16 \times .7}{.35} \times .7 = 22.3$.

General Rules. The methods above described may be briefly expressed as follows:

Capacity—$Q = A \times v \times F$, in which

Q = Cubic feet of air per minute;

A = Effective area of fan outlet (Table XXXIV);

v = Velocity of flow through outlet;

$\begin{cases} 3{,}165 \ (\frac{3}{8}\text{-ounce pressure}) \text{ for schoolhouses, etc.;} \\ 4{,}084 \ (\frac{5}{8}\text{-ounce pressure}) \text{ for factories;} \end{cases}$

$F = \begin{cases} .8 \text{ for schoolhouses, etc.;} \\ .7 \text{ for factories.} \end{cases}$

Speed—Take the speed from Table XXXIII, corresponding to the given pressure and size of fan, and multiply by 1.2.

Horse-Power—$\text{H.P.} = \frac{A \times p \times F}{.35}$, in which

H.P. = Horse-power;

A = Effective area of fan outlet;

p = Horse-power to move air which will flow through 1 square foot of fan outlet under given pressure (Table XXXV);

$= \begin{cases} .33 \text{ for schoolhouses, etc.;} \\ .7 \text{ for factories.} \end{cases}$

$F = \begin{cases} .8 \text{ for schoolhouses, etc.;} \\ .7 \text{ for factories.} \end{cases}$

EXAMPLES

1. A schoolhouse requires an air-supply of 30,000 cubic feet per minute. What will be the required size of fan, its speed, and the H. P. of engine to drive it?

Ans. $\begin{cases} \text{7 ft. in diameter.} \\ \text{173 r. p. m.} \\ \text{9 H. P.} \end{cases}$

2. What will be the size and speed of fan, and horse-power of engine, to heat and ventilate a factory requiring 1,080,000 cubic feet of air per hour?

Ans. $\begin{cases} \text{6 ft. in diameter.} \\ \text{260 r. p. m.} \\ \text{8.8 H. P.} \end{cases}$

General Relations. The following general relations between the volume, pressure, and power will often be found useful in deciding upon the size of a fan:

(1) The volume of air delivered varies *directly* as the speed of the fan; that is, doubling the number of revolutions doubles the volume of air delivered.

(2) The pressure varies as the *square* of the speed. For example, if the speed is doubled, the pressure is increased $2 \times 2 = 4$ times; etc.

(3) The power required to run the fan varies as the *cube* of the speed. Thus, if the speed is doubled, the power required is increased $2 \times 2 \times 2 = 8$ times; etc.

The value of a knowledge of these relations may be illustrated by the following example:

Suppose for any reason it were desired to double the volume of air delivered by a certain fan. At first thought we might decide to use the same fan and run it twice as fast; but when we come to consider the power required, we should find that this would have to be increased 8 times, and it would probably be much cheaper in the long run to put in a larger fan and run it at lower speed.

Disc or Propeller Fans. When air is to be moved against a very slight resistance, as in the case of exhaust ventilation, the disc or propeller type of wheel may be used. This is shown in different forms in Figs. 149 and 150. This type of fan is light in construction, requires but little power at low speeds, and is easily erected. It may be conveniently placed in the attic or upper story of a building, where it may be driven either by a direct- or belt-connected electric motor. Fig. 148 shows a fan equipped with a direct-connected motor, and Fig. 151 the general arrangement when a belted motor is used. These fans are largely used for the ventilation of toilet and smoking rooms, restaurants, etc., and are usually mounted in a wall opening, as shown in Fig. 151. A damper should always be provided for shutting off the opening when the fan is not in use. The fans shown in Figs. 149 and 150 are provided with pulleys for belt connection.

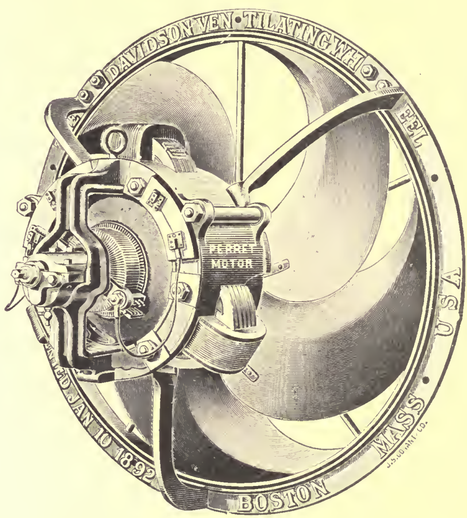


Fig. 148. Propeller Fan Direct-Connected to Motor.

Fans of this kind are often connected with the main vent flues of large buildings, such as schools, halls, churches, theaters, etc., and are especially adapted for use in connection with gravity heating systems. They are usually run by electric motors, and as a rule are placed in positions where an engine could not be connected and also in buildings where steam pressure is not available.

Capacity of Disc Fans. The capacity of a disc fan varies greatly with the type and the conditions under which it operates. The rated capacities usually given in catalogues are for fans revolving in free air—that is, mounted in an opening without being connected with ducts or subjected to other frictional resistance.

As the capacity and necessary power are so dependent upon the resistance to be overcome, it is difficult to give definite rules for determining them. The following data, based upon actual tests,

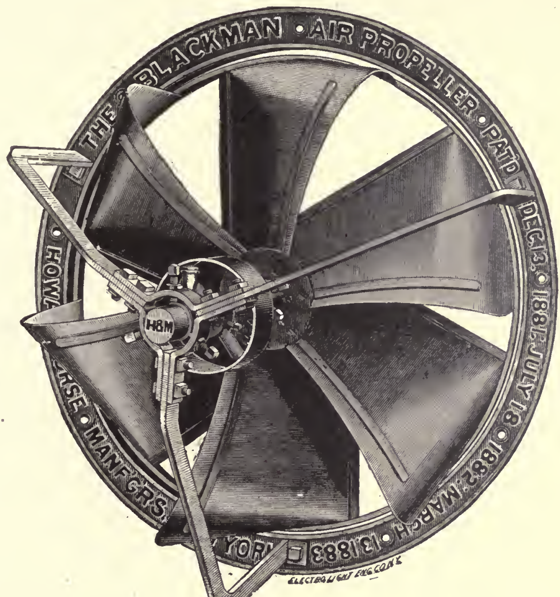


Fig. 149. Another Form of Propeller Fan, with Special Type of Blade.

apply to fans working against a resistance such as would be produced by connecting with a system of ducts of medium length through which the air was drawn at a velocity not greater than 600 or 800 feet per minute. Under these conditions, a good type of fan will propel the air in a direction parallel to the shaft a distance equal to about .7 of its diameter at each revolution; and from this we have the equation:

$$Q = .7\,D \times R \times A,$$

in which

Q = Cubic feet of air discharged per minute;
D = Diameter of fan, in feet;
R = Revolutions per minute;
A = Area of fan, in square feet.

In order to obtain the best results, the linear velocity of air-flow through the fan should range from 800 to 1,200 feet per minute.

Table XXXVI gives the revolutions per minute for fans of different diameter to produce a linear velocity of 1,000 feet, the volume delivered at this speed, and the horse-power required.

The horse-power is computed by allowing .14 H. P. for each 1,000 cubic feet of air moved, when the velocity through the fan is 800 feet per minute; .16 H. P. for 1,000 feet velocity; and .18 H. P. for 1,200 feet velocity. These factors are empirical, and based on tests.

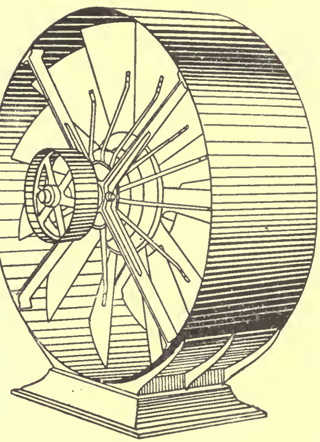


Fig. 150. Propeller Fan with Wheel on Shaft for Belt Connection.

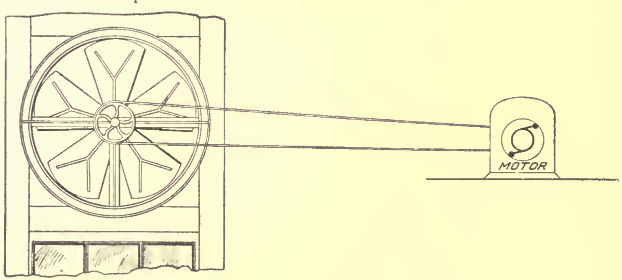


Fig. 151. Fan Belt-Connected to Motor.

Example. Assuming a velocity of 800 feet per minute through a 4-foot fan, what volume will be delivered per minute, and what speed and horse-power will be required?

TABLE XXXVI

Disc Fans, their Capacity, Speed, etc.

Dia. of Fan, in Inches	Rev. per Min.	Cubic Feet of Air Moved	Horse-Power Required
18	952	1,700	.27
24	716	3,100	.50
30	572	4,900	.78
36	476	7,100	1.2
42	408	9,400	1.5
48	343	12,000	1.9
54	317	15,800	2.5
60	286	19,400	3.1
72	238	28,300	4.5

The area of a 4-foot fan is 12.5 square feet; and at 800 velocity the volume would be $12.5 \times 800 = 10{,}000$ cubic feet. Next solve for the speed by the equation $Q = .7D \times R \times A$, which, when transposed, takes the form

$$R = \frac{Q}{.7\,D \times A}.$$

Substituting the known quantities, we have:

$$R = \frac{10{,}000}{.7 \times 4 \times 12.5} = 286.$$

The horse-power is $10 \times .14 = 1.4$.

Fan Engines. A simple, quiet-running engine is desirable for use in connection with a fan or blower. The engine may be either horizontal or vertical; and for schoolhouse and similar work, should be provided with a large cylinder, so that the required power may be developed without carrying a boiler pressure much above 30 pounds. In some cases, cylinders of such size are used that a boiler pressure of 12 or 15 pounds is sufficient. The quantity of steam which an engine consumes is of minor importance, as the exhaust can be turned into the coils and used for heating purposes. If space allows, the engine should always be belted to the fan. Where it is direct-connected, as in Fig. 144, there is likely to be trouble from noise, as any slight looseness or pounding in the engine will be communicated to the air-ducts, and the sound will be carried to the rooms above. Figs. 152 and 153 show common forms of fan engines. The latter is especially adapted to this purpose, as all bearings are enclosed

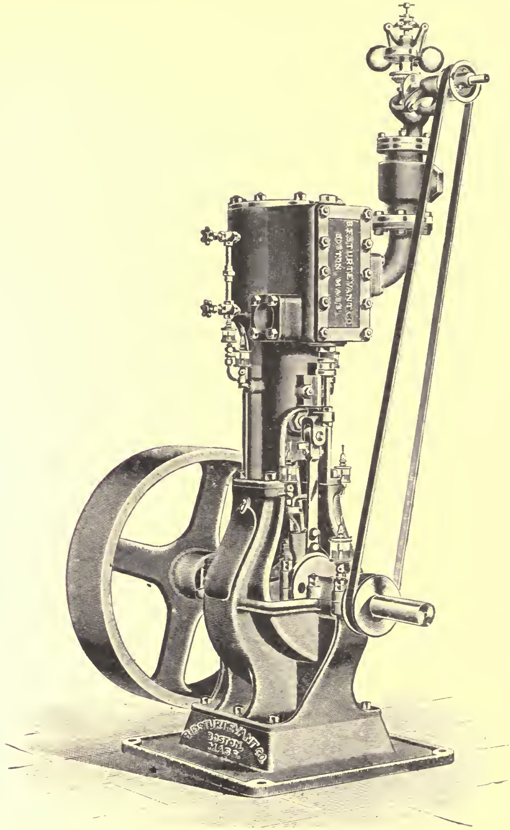


Fig. 152 A Common Form of Fan Engine.

and protected from dust and grit. A horizontal engine for fan use is shown in Fig. 154.

In case an engine is belted, the distance between the shafts of the fan and engine should not in general be much less than 10 feet for fans up to 7 or 8 feet in diameter, and 12 feet for those of larger size. When possible, the tight or driving side of the belt should be at the bottom, so that the loose side, coming on top, will tend to wrap around the pulleys and so increase the arc of contact.

Motors. Electric motors are especially adapted for use in connection with fans. This method of driving is more expensive

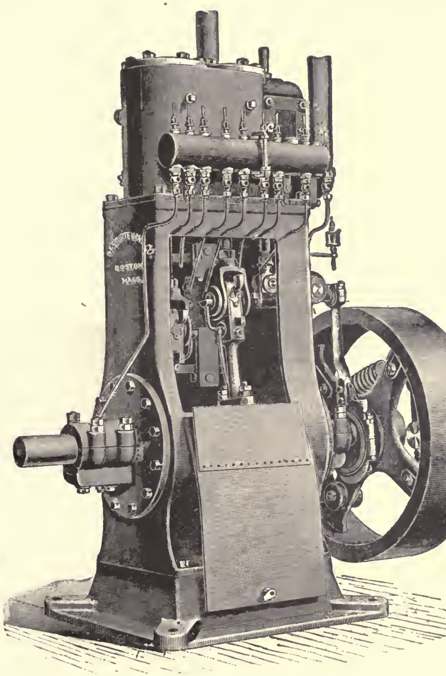


Fig. 153. Another Form of Fan Engine, with Bearings Enclosed to Protect Them from Dust and Grit.

than by the use of an engine, especially if electricity must be purchased from outside parties; but if the building contains its own power plant, so that the exhaust steam can be utilized for heating, the convenience and simplicity of motor-driven fans often more than offset the additional cost of operation.

Direct-connected motors are always preferable to belted, if a direct current is available, on account of greater quietness of action. This is due both to the slower speed of the motor and to the absence of belts.

Sufficient speed regulation can be obtained with direct-connected machines, without excessive waste of energy, by the use of a rheostat.

If a direct current is not available, and an alternating current must be used, the advantages of electric driving are greatly reduced, as high-speed motors with belts must be employed, and, furthermore, satisfactory speed regulation is not easily attainable.

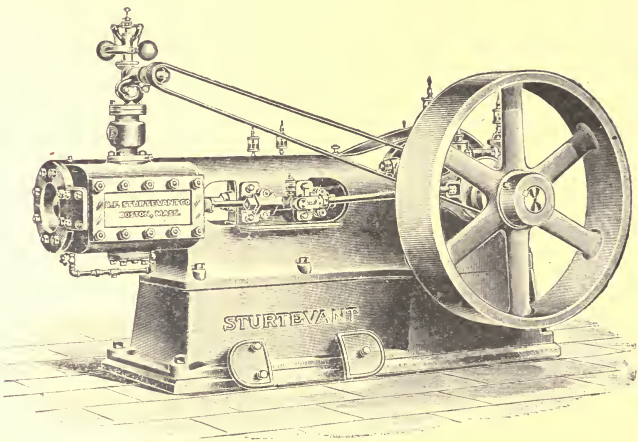


Fig. 154. Horizontal Engine for Fan Use.

Area of Ducts and Flues. With the blower type of fan, the size of the main ducts may be based on a velocity of 1,200 to 1,500 feet per minute; the branches, on a velocity of 1,000 to 1,200 feet per minute, and as low as 600 to 800 feet when the pipes are small. Flue velocities of 500 to 700 feet per minute may be used, although the lower velocity is preferable. The size of the inlet register should be such that the velocity of the entering air will not exceed about 300 feet per minute. The velocity between the inlet windows and the fan or heater should not exceed about 800 feet.

The air-ducts and flues are usually made of galvanized iron, the ducts being run at the basement ceiling. No. 20 and No. 22 iron is used for the larger sizes, and No. 24 to No. 28 for the smaller.

Regulating dampers should be placed in the branches leading to each flue, for increasing or reducing the air-supply to the different rooms. Adjustable deflectors are often placed at the fork of a pipe for the same purpose. One of these is shown in Fig. 155.

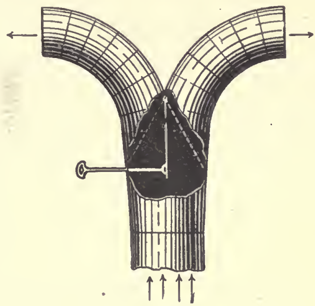


Fig. 155. Adjustable Deflector Placed at Fork of Pipe to Regulate Air-Supply.

Fig. 156 illustrates a common arrangement of fan and heater where the type of heater shown in Fig. 138 is used; and Fig. 157 is a self-contained apparatus in which the heater is inclosed in a steel casing.

Factory Heating. The application of forced blast for the warming of factories and shops, is shown in Figs. 158 and 159. The proportional heating surface in this case is generally expressed in the number of cubic feet in the building for each linear foot of 1-inch steam pipe in the heater. On this basis, in factory practice, with all of the air taken from out of doors, there are generally allowed from 100 to 150 cubic feet of space per foot of pipe, according as exhaust or live steam is used, live steam in this case indicating steam of about 80 pounds pressure. If practically all the air is returned from the

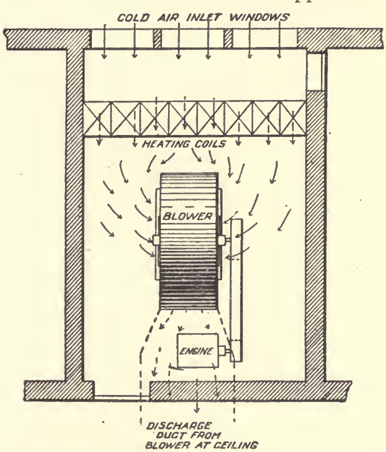


Fig. 156. Common Arrangement of Fan with Heater of Type Shown in Fig. 138.

buildings to the heater, these figures may be raised to about 140 as a minimum, and possibly 200 as a maximum, per foot of pipe. The

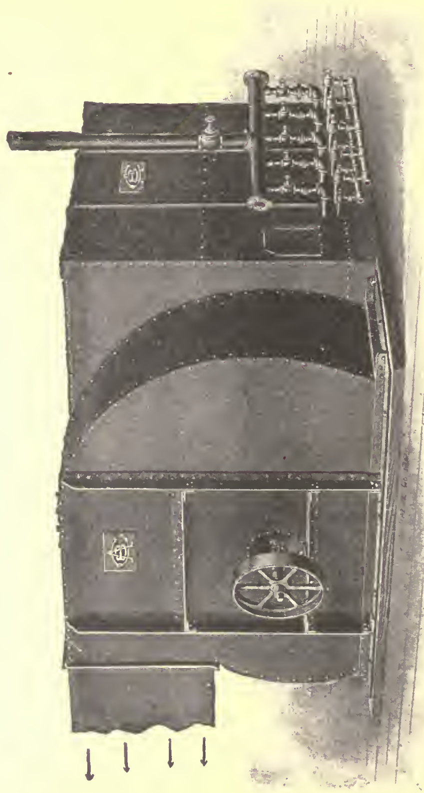


Fig. 157. Self Contained Heater and Blower in Steel Casing.

heaters in Table XXXI may be changed to linear feet of 1 inch pipe by multiplying the numbers in column three (square feet of surface) by three.

EXAMPLES FOR PRACTICE

1. A machine shop 100 feet long by 50 feet wide and having 3 stories, each 10 feet high, is to be warmed by forced blast, using

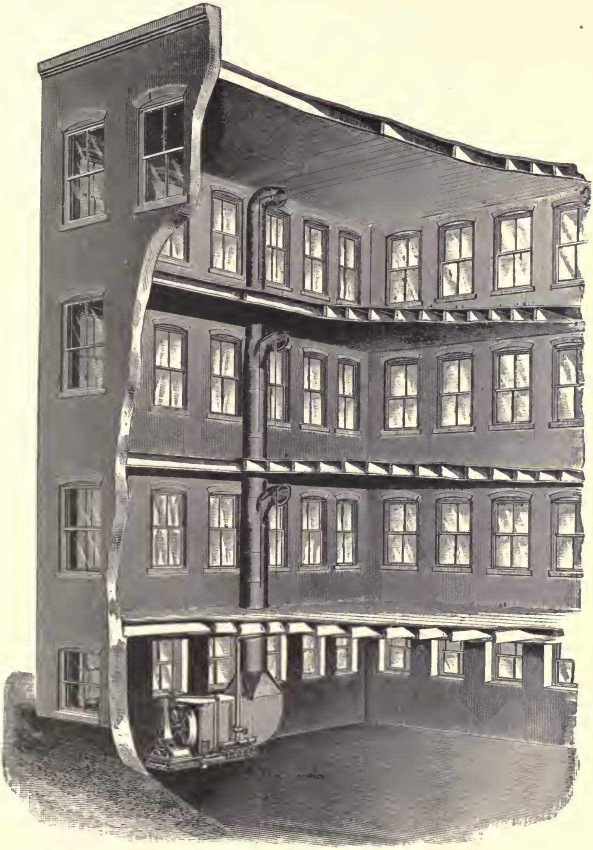


Fig. 158. Illustrating Application of Forced Blast for Warming a Factory.

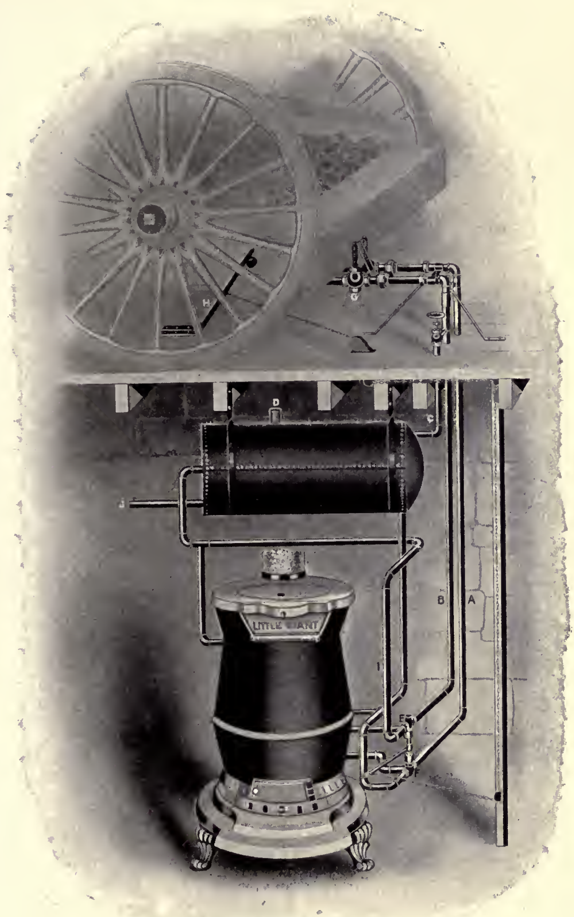
exhaust steam in the heater. The air is to be returned to the heater from the building, and the whole amount contained in the building is to pass through the heater every 15 minutes. What size of blower will be required, and what will be the H. P. of the engine required to run it? How many linear feet of 1-inch pipe should the heater contain?

Ans. $\begin{cases}\text{4-foot blower.}\\ \text{6 H. P. engine.}\\ \text{1,071 feet of pipe.}\end{cases}$

Fig. 159. Centrifugal Blower Producing Forced Blast for Heating a Shop.

2. Find the size of blower, engine, and heater for a factory 200 feet long, 60 feet wide, and having 4 stories, each 10 feet high, using live steam at 80 pounds pressure in the heater, and changing the air every 20 minutes by taking in cold air from out of doors.

Ans. $\begin{cases}\text{6-foot blower.}\\ \text{13 H. P. engine.}\\ \text{3,200 feet of pipe.}\end{cases}$



LITTLE GIANT BOILER AS INSTALLED FOR FIRE DEPARTMENT SERVICE
Pierce, Butler & Pierce Mfg. Co.

In using this method of computation, judgment must be employed, which can come only from experience. The figures given are for average conditions of construction and exposure.

Double-Duct System. The varying exposures of the rooms of a school or other building similarly occupied, require that more heat shall be supplied to some than to others. Rooms that are on the south side of the building and exposed to the sun, may perhaps be kept perfectly comfortable with a supply of heat that will maintain a temperature of only 50 or 60 degrees in rooms on the opposite side of the building which are exposed to high winds and shut off from the warmth of the sun.

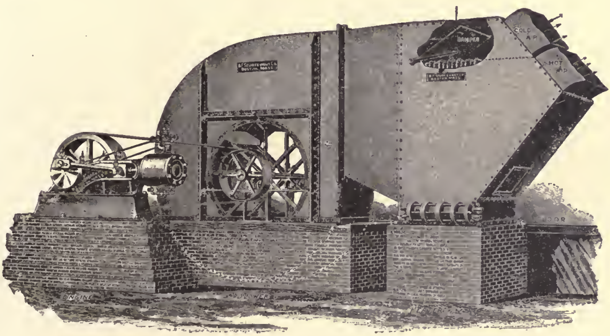


Fig. 160. Hot-Blast Apparatus with Double Duct for Supplying Air at Different Temperatures to Different Parts of a Building.

With a constant and equal air-supply to each room, it is evident that the temperature must be directly proportional to the cooling surfaces and exposure, and that no building of this character can be properly heated and ventilated if the temperature cannot be varied without affecting the air-supply.

There are two methods of overcoming this difficulty:

The older arrangement consists in heating the air by means of a primary coil at or near the fan, to about 60 degrees, or to the minimum temperature required within the building. From the coil it passes to the bases of the various flues, and is there still further heated as required, by secondary or supplementary heaters placed at the base of each flue.

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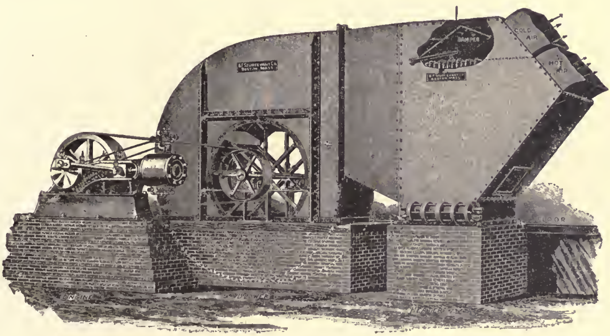

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With the second and more recent method, a single heater is employed, and all the air is heated to the maximum required to maintain the desired temperature in the most exposed rooms, while the temperature of the other rooms is regulated by mixing with the hot air a sufficient volume of cold air at the bases of the different flues. This result is best accomplished by designing a hot-blast apparatus so that the air shall be forced, rather than drawn through the heater, and by providing a by-pass through which it may be discharged without passing across the heated pipes.

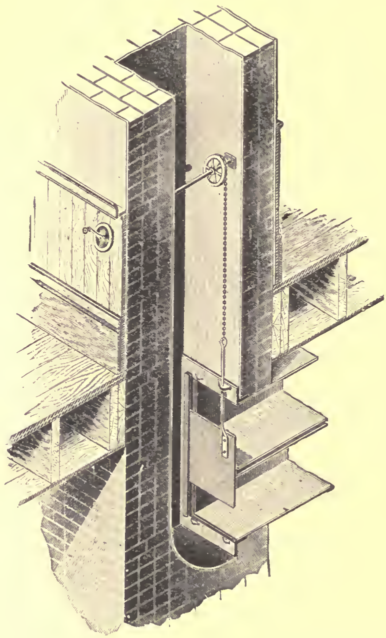


Fig. 161. Mixing Damper for Regulating Temperature of Air Supplied by Double-Duct System.

The passage for the cool air is usually above and separate from the heater pipes, as shown in Fig. 160. Extending from the apparatus is a double system of ducts, usually of galvanized iron, suspended from the ceiling. At the base of each flue is placed a mixing damper, which is controlled by a chain from the room above, and so designed as to admit either a full volume of hot air, a full volume of cool or tempered air, or to mix them in any desired proportion without affecting the resulting total volume delivered to the room. A damper o this form is shown in Fig. 161.

Fig. 162 shows an arrangement of disc fan and heater where the air is first drawn through a tempering coil, then a portion of it forced through a second heater and into the warm-air pipes, while the remainder is by-passed under the heater into the cold-air pipes. Mixing

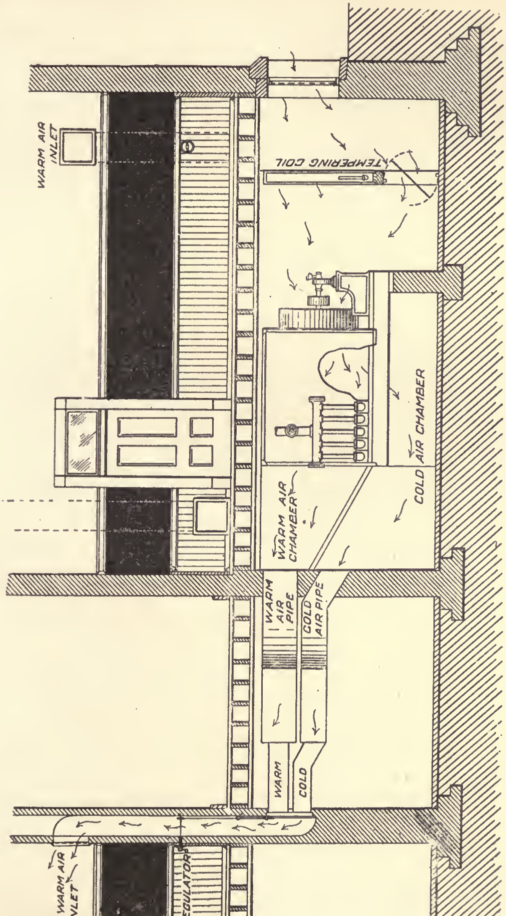


Fig. 162. Warming System Combining Tempering Coil, Disc Fan, Second Heater, and Double Ducts.

dampers are placed at the bases of the flues as already described, to regulate the temperature in different rooms.

ELECTRIC HEATING

Unless electricity is produced at a very low cost, it is not commercially practicable for heating residences or large buildings. The electric heater, however, has quite a wide field of application in heating small offices, bathrooms, electric cars, etc. It is a convenient method of warming rooms on cold mornings in late spring and early fall, when furnace or steam heat is not at hand. It has the special advantage of being instantly available, and the amount of heat can be regulated at will. The heaters are perfectly clean, do not vitiate the air, and are portable.

Electric Heat and Energy. The commercial unit for electricity is one watt for one hour, and is equal to 3.41 B. T. U. Electricity is usually sold on the basis of 1,000 watt-hours (called *Kilowatt-hours*),

Fig. 163. Electric Car-Heater.

which is equivalent to 3,410 B. T. U. A watt is the product obtained by multiplying a current of 1 ampere by an electromotive force of 1 volt.

From the above we see that the B. T. U. required per hour for warming, divided by 3,410, will give the kilowatt-hours necessary for supplying the required amount of heat.

Construction of Electric Heaters. Heat is obtained from the electric current by placing a greater or less resistance in its path. Various forms of heaters have been employed. Some of the simplest consist merely of coils or loops of iron wire, arranged in parallel rows, so that the current can be passed through as many coils as are needed to provide the required amount of heat. In other forms, the heating material is surrounded with fire-clay, enamel, or asbestos, and in some cases the material itself has been such as to give considerable resistance to the current. A form of electric car-heater is shown in Fig. 163. Forms of radiators are shown in Figs. 164 and 165.

Calculation of Electric Heaters. The formula for the calculation of electric heaters is

$$H = I^2 R\, t \times .24,$$

in which

H = Heat, in calories;
I = Current, in amperes;
R = Resistance, in ohms;
t = Time, in seconds.

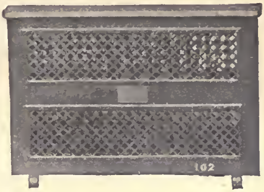


Fig. 164. Electric Radiator.

Examples. What resistance must an electric heater have, to give off 6,000 B. T. U. per hour, with a current of 20 amperes?

We have learned that 1 B. T. U. = 252 calories; so, in the present case, 6,000 × 252 = 1,512,000 calories must be provided.

Substituting the known values in the formula, we have

$$1{,}512{,}000 = 20^2 \times R \times 3{,}600 \times .24,$$

from which

$$R = \frac{1{,}512{,}000}{345{,}600} = 4.37 \text{ ohms.}$$

A heater having a resistance of 3 ohms is to supply 3,000 B. T. U. per hour. What current will be required?

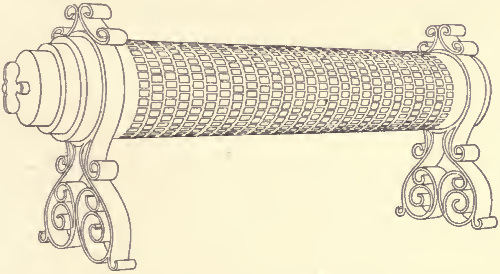


Fig. 165. Another Form of Electric Radiator.

3,000 × 252 = 756,000 calories. Substituting the known values in the formula, and solving for I, we have

$$756{,}000 = I^2 \times 3 \times 3{,}600 \times .24,$$

from which

$$I = \sqrt{291.6} = 17 + \text{ amperes.}$$

Connections for Electric Heaters. The method of wiring for electric heaters is essentially the same as for lights which require the same amount of current. A constant electromotive force or voltage is maintained in the main wire leading to the heaters. A much less voltage is carried on the return wire, and the current in passing through the heater from the main to the return, drops in voltage or pressure. This drop provides the energy which is transformed into heat.

The principle of electric heating is much the same as that involved in the non-gravity return system of steam heating. In that system, the pressure on the main steam pipes is that of the boiler, while that on the return is much less, the reduction in pressure occurring in the passage of the steam through the radiators; the water of condensation is received into a tank, and returned to the boiler by a pump.

In a system of electric heating, the main wires must be sufficiently large to prevent a sensible reduction in voltage or pressure between the generator and the heater, so that the pressure in them shall be substantially that in the generator. The pressure or voltage in the main return wire is also constant, but very low, and the generator has an office similar to that of the steam pump in the system just described—that is, of raising the pressure of the return current up to that in the main. The power supplied to the generator can be considered the same as the boiler in the first case. All the current which passes from the main to the return must flow through the heater, and in so doing its pressure or voltage falls from that of the main to that of the return.

From the generator shown in Fig. 166, main and return wires are run the same as in a two-pipe system of steam heating, and these are proportioned to carry the required current without sensible drop or loss of pressure. Between these wires are placed the various heaters, which are arranged so that when electric connection is made they draw the current from the main and discharge it into the return wire. Connections are made and broken by switches, which take the place of valves on steam radiators.

Cost of Electric Heating. The expense of electric heating must in every case be great, unless the electricity can be supplied at an exceedingly low cost. Estimated on the basis of present practice, the average transformation into electricity does not account for more than 4 per cent of the energy in the fuel which is burned in the furnace. Although under best conditions 15 per cent has been realized, it would not be safe to assume that in ordinary practice more than 5 per cent could be transformed into electrical energy. In heating with steam, hot water, or hot air, the average amount utilized will probably be about 60 per cent, so that the expense of electrical heating is approximately from 12 to 15 times greater than by these methods.

TEMPERATURE REGULATORS

The principal systems of automatic temperature control now in use, consist of three essential features; *First,* an air-compressor, reservoir, and distributing pipes; *second,* thermostats, which are

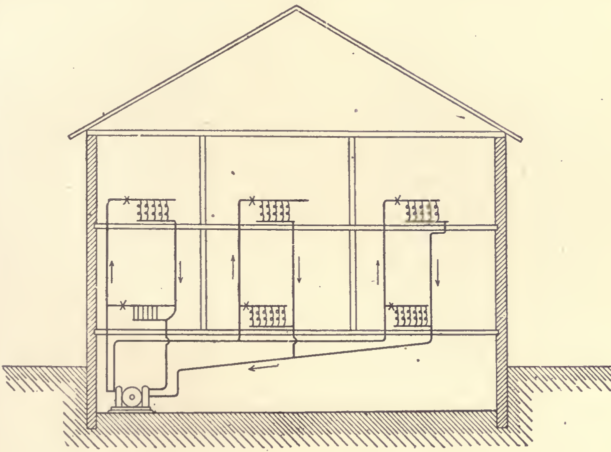


Fig. 166. General System of Wiring a House for Electric Heating.

placed in the rooms to be regulated; and *third,* special diaphragm or pneumatic valves at the radiators.

The *air-compressor* is usually operated by water-pressure in small plants and by steam in larger ones; electricity is used in some cases. Fig. 167 shows a form of water compressor. It is similar in principle to a direct-acting steam pump, in which water under pressure takes the place of steam. A piston in the upper cylinder compresses the air, which is stored in a reservoir provided for the purpose. When the pressure in the reservoir drops below a certain point, the compressor is started automatically, and continues to operate until the pressure is brought up to its working standard.

A *thermostat* is simply a mechanism for opening and closing one or more small valves, and is actuated by changes in the tempera-

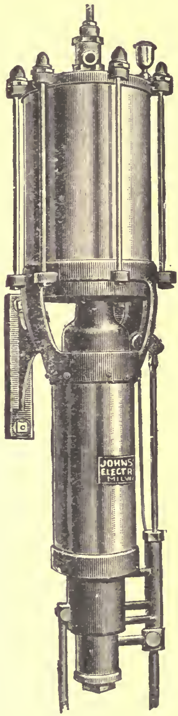


Fig. 167. Air-Compressor Operated by Water-Pressure, Automatically Controlled, and Operating to Regulate Temperature by Controlling Radiator Valves.

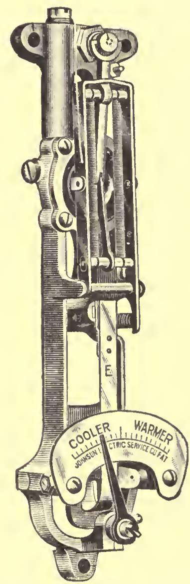


Fig. 168. Thermostat Controlling Valves on Radiators, and Operating through Expansion or Contraction of Metal Strip *E*.

ture of the air in which it is placed. Fig. 168 shows a thermostat in which the valves are operated by the expansion and contraction of the metal strip *E*. The degree of temperature at which it acts may be adjusted by throwing the pointer at the bottom one way or the other. Fig. 169 shows the same thermostat with its ornamental casing in place. The thermostat shown in Fig. 170 operates on a somewhat different principle. It consists of a vessel separated into two chambers by a metal diaphragm. One of these chambers is partially filled with a liquid which will boil at a temperature below that desired in the room. The vapor of the liquid produces considerable pressure at the normal temperature of the room, and a slight increase of heat crowds the diaphragm over and operates the small valves in a manner similar to that of the metal strip in the case just described.

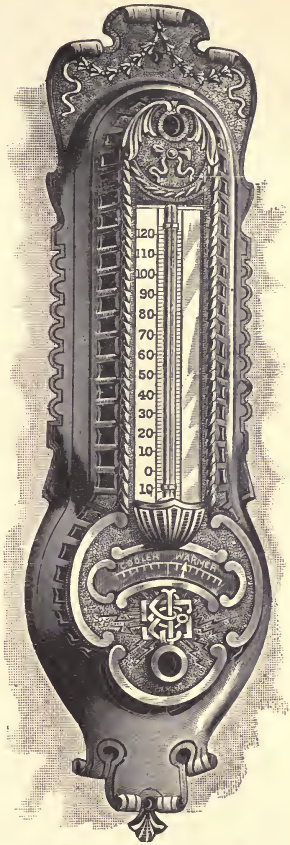


Fig. 169. Thermostat of Fig. 168 in Ornamental Casing.

The general form of a *diaphragm valve* is shown in Fig. 171. These replace the usual hand-valves at the radiators. They are similar in construction to the ordinary globe or angle valve, except that the stem slides up and down instead of being threaded and running in a nut. The top of the stem connects with a flat plate, which rests against a rubber diaphragm. The valve is held open by a spring, as shown, and is closed by admitting compressed air to the space above the diaphragm.

In connecting up the system, small concealed pipes are carried from the air-reservoir to the thermostat, which is placed upon an inside wall of the room, and from there to the diaphragm valve at the radiator. When the temperature of the room reaches the maximum point for which the thermostat is set, its action opens a small valve and admits air-pressure to the diaphragm, thus closing off the steam from the radiator. When the temperature falls, the thermostat acts in the opposite manner, and shuts off the air-pressure from the diaphragm valve, at the same time opening a small exhaust which allows the air above the diaphragm to escape. The pressure being removed, the valve opens and again admits steam to the radiator.

Diaphragm Motors. Dampers are operated pneumatically in a similar manner to steam valves. A *diaphragm motor*, so called, is acted upon by the air-pressure; and this lifts a lever which is properly connected to the damper by means of chains or levers, thus securing the desired movement.

Dampers. When mixing dampers are operated pneumatically, a specially designed thermostat for giving a graduated movement

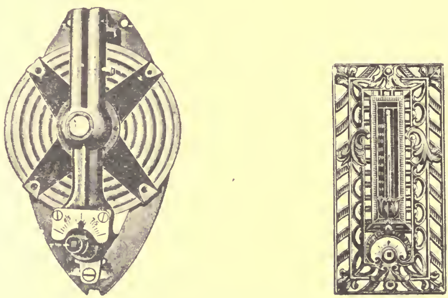


Fig. 170. Thermostat Operating through Expansion or Contraction of the Vapor of a Volatile Liquid.

to the damper should be used. By this arrangement the damper is held in such a position at all times as to admit the proper proportions of hot and cold or tempered air for producing the desired temperature in the room with which it is connected.

Large dampers which are to be operated pneumatically, should be made up in sections or louvres. Dampers constructed in this manner are handled much more easily than when made in a single piece.

It often happens, in large plants, that there are valves and dampers in places which are not easily reached for hand manipulation. These may be provided with diaphragms and connected with the air-pressure system for operation by hand-switches or cocks conveniently located at some central point in the basement or boiler room.

Telethermometer. This is a device for indicating on a dial at some central point the temperature of various rooms or ducts in different parts of a building. A special *transmitter* is placed in each of the rooms and electrically connected with a central switchboard. Then, by means of suitable switches, any room may be thrown in circuit with the *recorder*, and the temperature existing in the room at that time read from the dial.

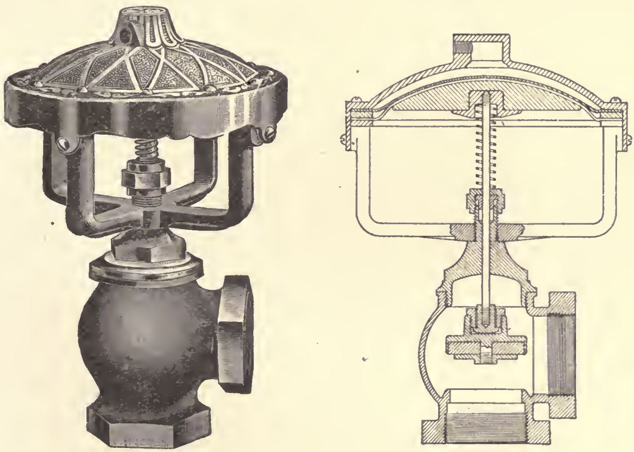


Fig. 171. Exterior View, and Section Showing Interior Mechanism of Diaphragm Valve.

Humidostat. The *humidostat* is a device to be placed in one or more rooms of a building for maintaining an even percentage of moisture in the air. The apparatus consists of two essential parts—the *humidostat* and the *humidifier*. The former corresponds to the thermostat in a system of temperature control, and operates a pneumatic valve or other mechanism connected with the humidifier when the percentage of moisture rises above or falls below certain limits. The operating medium is compressed air, the same as for temperature control; and the two devices are usually connected with the same pressure system.

The normal moisture of a room is 70 per cent, and should never exceed that. In cold weather it will be necessary to reduce the amount of moisture somewhat, owing to the "sweating" of walls and windows.

The method of moistening the air will depend somewhat upon circumstances. If the air for ventilation is delivered to the rooms at a temperature not exceeding 70 degrees, the humidifier is best placed in the main air-duct. If the air enters at a higher temperature, the humidifier must be located in the same room with the humidostat.

The moistener or humidifier may be of any one of several forms. Where steam heating is used, and where the steam is clean and odorless and free from oil from engines, a perforated pipe (or pipes) in the air-duct is the simplest and best humidifier. The outlets are properly adjusted, and then the humidostat shuts off and lets on the steam as required. Sometimes a water spray, particularly of warm water, may be used in place of steam. When neither steam jet nor water spray is advisable, an evaporating pan containing a steam coil may be used, the humidostat controlling the steam to the coil, and the water-level in the pan being kept constant by means of a ball-cock.

AIR-FILTERS AND AIR-WASHERS

In cases where the air for ventilating purposes is likely to contain soot or street dust, it is desirable to provide some form of filter for purifying it before delivering to the rooms. If the air-quantity is small and there is plenty of room between the inlet windows and the fan, screens of light cheesecloth may be used for this purpose. The cloth should be tacked to light but substantial wooden frames, which can be easily removed for frequent cleaning. These screens are usually set up in "saw-tooth" fashion in order to give as much surface as possible in the least space.

Another arrangement, used in case of large volumes of air, is to provide a number of light cloth bags of considerable length, through which the air is drawn before reaching the heater. These are fastened to a suitable frame or partition for holding them open. The great objection to filters of this kind is their obstruction to the passage of the air, especially when filled with dust, the frequent intervals at which they should be cleaned, and the great amount of filtering surface required.

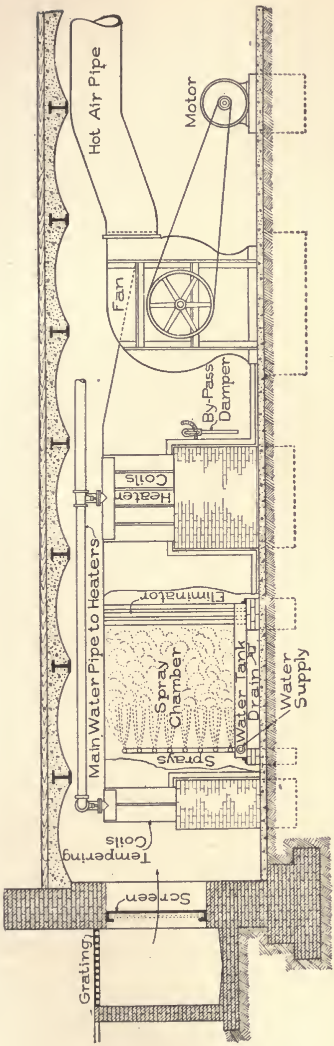


Fig. 172. One Form of Spray Filter or Air-Washer, for Removing Dust or Soot from Air Required for Ventilation.

An apparatus which is coming quite generally into use for this purpose, and which does away with the disadvantages noted above, is the *spray filter* or *air-washer,* one form of which is shown in Fig. 172. Air enters as indicated, and first passes through a tempering coil to raise it above the freezing point in winter weather; then passes through the spray-chamber, where the dirt is removed; then through an eliminator for removing the water; and then through a second heater on its way to the fan.

The water is forced through the spray-heads by means of a small centrifugal pump, either belted to the fan shaft or driven by an independent motor.

HEATING AND VENTILATION OF VARIOUS CLASSES OF BUILDINGS

The different methods used in heating and ventilation, together with the manner of computing the various proportions of the apparatus, having been taken up, the application of these systems to the different classes of buildings will now be considered briefly.

School Buildings. For school buildings of small size, the furnace system is simple, convenient, and generally effective. Its use is confined as a general rule to buildings having not more than six or eight rooms. For large ones this method must generally give way to some form of indirect steam system with one or more boilers, which occupy less space, and are more easily cared for than a number of furnaces scattered about in different parts of the basement. As in all systems that depend on natural circulation, the supply and removal of air is considerably affected by changes in the outside temperature and by winds.

The furnaces used are generally built of cast iron, this material being durable, and easily made to present large and effective heating surfaces. To adapt the larger sizes of house-heating furnaces to schools, a much larger space must be provided between the body and the casing, to permit a sufficient volume of air to pass to the rooms. The free area of the air-passage should be sufficient to allow a velocity of about 400 feet per minute.

The size of furnace is based on the amount of heat lost by radiation and conduction through walls and windows, *plus* that carried away by air passing up the ventilating flues. These quantities may be computed by the usual methods for "loss of heat by conduction through walls," and "heat required for ventilation." With more regular and skilful attendance, it is safe to assume a higher rate of combustion in schoolhouse heaters than in those used for warming residences. Allowing a maximum combustion of 6 pounds of coal per hour per square foot of grate, and assuming that 8,000 B. T. U. per pound are taken up by the air passing over the furnace, we have $6 \times 8{,}000 = 48{,}000$ B. T. U. furnished per hour per square foot of grate. Therefore, if we divide the total B. T. U. required for both warming and ventilation by 48,000, it will give us the necessary grate surface in square feet. It has been found in practice that a furnace with a firepot 32 inches in diameter, and having ample heating surface, is capable of heating two 50-pupil rooms in zero weather. The sizes of ducts and flues may be determined by rules already given under furnace and indirect steam heating.

The velocity of the warm air within the uptake flues depends upon their height and the difference in temperature between the warm air within the flues and the cold air outside. The action of the wind also affects the velocity of air-flow. It has been found by experience that flues having sectional areas of about 6 square feet for first-floor rooms, 5 square feet for the second floor, and 4½ square feet for the third, will be of ample size for standard classrooms seating from 40 to 50 pupils in primary and grammar schools. These sizes may be used for both furnace and indirect gravity steam heating.

The vent flues may be made 5 square feet for the first floor, and 6 square feet for the second and third floors. They may be arranged in banks, and carried through the roof in the form of large chimneys, or may be carried to the attic space and there gathered by means of galvanized-iron ducts connecting with roof vents of wood or copper construction.

In order to make the vent flues "draw" sufficiently in mild or heavy weather, it is necessary to provide some means for warming the air within them to a temperature somewhat above that of the rooms with which they connect. This may be done by placing a small stove made specially for the purpose, at the base of each flue. If this is done, it is necessary to carry the air down and connect with the flue just below the stove.

The cold-air supply duct to each furnace should be made ¾ the size of all the warm-air flues if free from bends, or the full size if obstructed in any way.

The inlet and outlet openings from the rooms into the flues, are commonly provided with grilles of iron wire having a mesh of 2 to 2½ inches. Both flat and square wire are used for this purpose. Mixing dampers for regulating the temperature of the rooms should be provided for each flue. The effectiveness of these dampers will depend largely upon their construction; and they should be made tight against cold-air leakage, by covering the surfaces or flanges against which they close with some form of asbestos felting. Both inlet and outlet gratings should be provided with adjustable dampers. One of the disadvantages of this system is the delivery of all the heat to the room from a single point, and this not always in a position to give the best results. The outer walls are thus left unwarmed, except as the heat is diffused throughout the room by air-currents. When there is considerable glass surface, as in most of our modern schoolrooms, draughts and currents of cold air are frequently found along the outside walls.

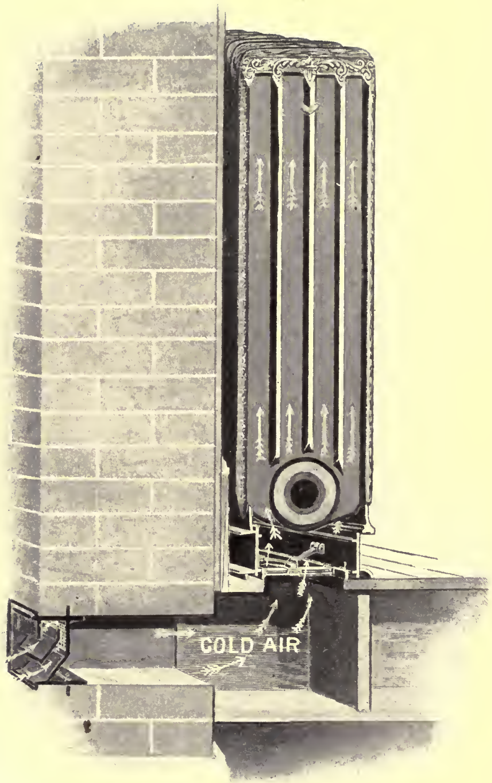
The indirect gravity system of steam heating comes next in cost of installation. One important advantage of this system over furnace heating comes from the ability to place the heating coils at the base of the flues, thus doing away with horizontal runs of air-pipe, which are required to some extent in furnace heating. The warm-air currents in the flues are less affected by variations in the direction and force of the wind where this construction is possible, and this is of much importance in exposed locations.

The method of supplying cold air to the coils or heaters is important, and should be carefully worked out. The supply should be taken from at least two sides of the building, or, if possible, from all four sides. When it is taken from four sides, each inlet should be made large enough to supply one-half the amount, or, in other words, any two should give the total quantity required. It is often possible to arrange the flues in groups so that all the heating stacks may be placed in two or more cold-air chambers, depending upon the size of the building. A cold-air trunk line may be run through the center of the basement, connecting with the outside on all four sides, and having branches supplying each cold-air chamber.

Cast-iron pin-radiators are particularly adapted to this class of work.

The *School-Pin*, having a section about 10 inches in depth and rated at 15 square feet of heating surface per section, is used quite extensively for this purpose. Stacks containing about 240 square feet of surface for southerly rooms, and 260 for those having a northerly exposure, have been found ample for ordinary conditions in zero weather.

A very satisfactory arrangement is the use of indirect heaters for warming the air needed for ventilation, and the placing of direct radiation in the rooms for heating purposes. The general construction of the indirect stacks and flues may be the same; but the heating surface can be reduced, as the air in this case must be raised only to 70 or 75 degrees in zero weather, the heat to offset that lost by conduction, etc., through walls and windows being provided by the direct surface. The mixing dampers may be omitted, and the temperature of the room regulated by opening or closing the steam valves



DIRECT-INDIRECT METHOD OF WARMING TAKING A FRESH AIR SUPPLY FROM OUTSIDE AND PASSING IT UPWARD
American Radiator Company

on the direct coils, which should be done automatically. The direct-heating surface, which is best made up of lines of $1\frac{1}{4}$-inch pipe, should be placed along the outer walls beneath the windows This supplies heat where most needed, and does away with the tendency to draughts. In mild weather, during the spring and fall, the indirect heaters may prove sufficient for both ventilation and warming.

Where direct radiation is placed in the rooms, the quantity of heat supplied is not affected by varying wind conditions, as is the case in indirect heating. Although the air-supply may be reduced at times, the heat quantity is not changed. Direct radiation has the disadvantage of a more or less unsightly appearance, and architects and owners often object to the running of mains or risers through the rooms of the building. Air-valves should always be provided with drip connections carried to a sink or dry well in the basement.

When circulation coils are used, a good method of drainage is to carry separate returns from each coil to the basement, and to place the air-valves in the drops just below the basement ceiling. A check-valve should be placed below the water-line in each return.

The gravity system has the fault of not supplying a uniform quantity of air under all conditions of outside temperature, the same as a furnace, but when properly arranged, may be made to give quite satisfactory results.

The fan or blower system for ventilation, with direct radiation in the rooms for warming, is considered to be one of the best possible arrangements.

In designing a plant of this kind, the main heating coil should be of sufficient size to warm the total air-supply to 70 or 75 degrees in the coldest weather, and the direct surface should be proportioned for heating the building independently of the indirect system. Automatic temperature regulation should be used in connection with systems of this kind, by placing pneumatic valves on the direct radiation. It is customary to carry from 3 to 8 pounds pressure on the direct system, and from 8 to 15 pounds on the main coil, depending upon the outside temperature. The foot-warmers, vestibule, and office heaters should be placed on a separate line of piping, with separate returns and trap, so that they can be used independently of the rest of the building if desired. Where there is a large assembly hall, it should be arranged so that it can be both warmed and ventilated when the rest of the building is shut off. This can be done by a proper arrangement of valves and dampers.

When different parts of the system are run on different pressures, the returns from each should discharge through separate traps into a receiver having connection with the atmosphere by means of a vent pipe. Fig. 173 shows a common arrangement for the return connections in a combination system of this kind. The different traps discharge into the vented receiver as shown; and the water is pumped back to the boiler automatically when it rises above a given level in the receiver, a pump governor being used to start and stop the pumps as required.

A water-level or seal of suitable height is maintained in the main returns, by placing the trap at the required elevation and bringing the returns into it near the bottom; a balance pipe is connected with the top for equalizing the pressure, the same as in the case of a pump governor. Sometimes a fan is used with the heating coils placed at the base of the flues, instead of in the rooms. Where this is done the radiating surface may be reduced about one-half. This system is less expensive to install, but has the disadvantage of removing the heating surface from the cold walls, where it is most needed.

With a blower type of fan, the size of the main ducts may be based on a velocity of from 1,000 to 1,200 feet per minute, and the branches on a velocity of 800 to 1,000 feet per minute.

The velocity in the vertical flues may be from 600 to 700 feet per minute, although the lower velocity is preferable.

The size of the inlet registers should be such that the velocity of the entering air will not exceed 350 to 400 feet per minute.

When the air is delivered through a register at the high velocities mentioned, some means must be provided for diffusing the entering current, in order to prevent disagreeable draughts. This is usually accomplished by the use of deflecting blades of galvanized iron, set in a vertical position and at varying angles, so that the air is thrown towards each side as it issues from the register. The size of the vent flues should be about the same as for a gravity system—that is, about 6 square feet for a standard classroom, and in the same proportion for smaller rooms.

Vent-flue heaters are not usually required in connection with a fan system, as the force of the fan is sufficient to supply the required

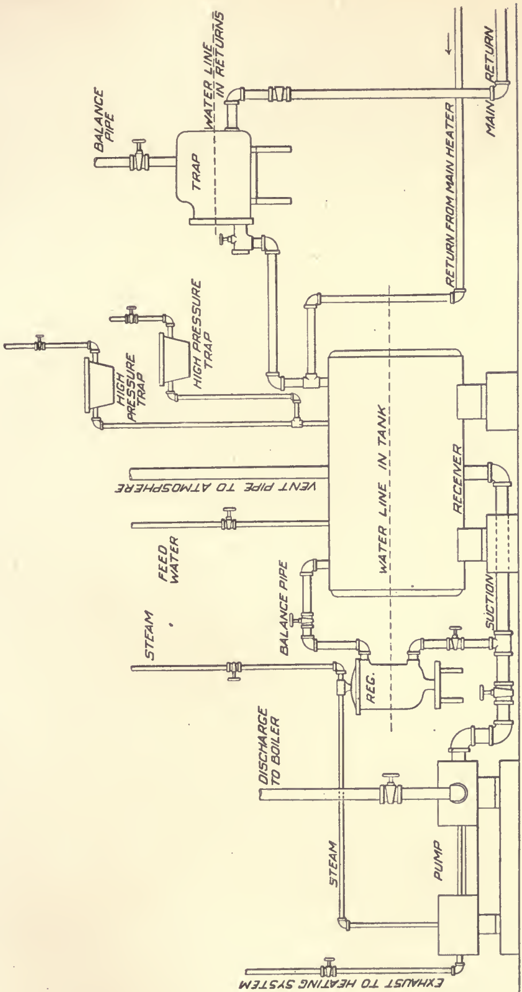


Fig. 173. General Arrangement of Return Connections Especially Adapted to Large Buildings where Different Parts of the Heating System are under Different Pressures.

quantity of air at all times without the aspirating effect of the vent flues.

The method of piping shown in Fig. 173 applies especially to buildings of large size. In the case of medium-sized buildings, it is often possible to use pin radiation for the main heater, placing the same well above the water-line of the boilers and thus returning the condensation by gravity, without the use of pumps or traps. When this arrangement is used, an engine with a large cylinder should be employed, so that the steam pressure will not exceed 15 or 18 pounds, and the whole system, including the direct surface, may be run upon the same system.

This is a very simple arrangement, and is adapted to all buildings of small and medium size where the heater can be placed at a sufficient height above the boilers.

Temperature control is usually secured automatically by placing pneumatic valves upon either the direct or supplementary heaters. Mixing dampers are sometimes used instead, in the latter case. Every fan system should be provided with a thermometer of large size for indicating the temperature of the air in the main duct just beyond the fan.

The ventilation of the toilet-rooms of a school building is a matter of the greatest importance. The first requirement is that the air-movement shall be *into* these rooms from the corridors instead of outward. To obtain this result, it is necessary to produce a slight vacuum within, and this cannot well be done if fresh air is forced into them.

One of the most satisfactory arrangements is to provide exhaust ventilation only, and to remove the greater part of the air through local vents connecting with the fixtures.

Hospitals. The best system for heating and ventilating a hospital depends upon the character and arrangement of the buildings. It is desirable in all cases to do the heating from a central plant, rather than to carry fires in the separate buildings, both on account of economy and for cleanliness.

In the case of small cottage hospitals with two or three buildings placed close together, indirect hot water affords a desirable system for the wards, with direct heat for the other rooms; but where there are several buildings, and especially if they are some distance apart, it becomes necessary to substitute steam unless the water is pumped through the mains. For large city buildings, a fan system is always desirable.

If the building is tall compared with its ground area, so that the horizontal supply ducts will be comparatively short, the double-duct system may be used with good results. Where the rooms are of good size, and the number of supply flues not great, the use of supplementary heaters at the bases of the flues makes a satisfactory arrangement. Direct radiation should never be used in the wards when it can be avoided, even in connection with an independent air-supply, as it offers too great an opportunity for the accumulation of dust in places which are difficult to reach.

It is common to provide from 80 to 100 cubic feet of air per minute per patient in ordinary wards, and from 100 to 120 cubic feet in contagious wards.

The usual ward building of a modern cottage-hospital generally contains a main ward having from 8 to 12 beds, and a number of private rooms of one bed each.

In addition to these, there are a diet kitchen, duty-room, toilet-rooms, bathrooms, linen-closets, and lockers.

For moderately sheltered locations, 30 square feet of indirect steam radiation has been found sufficient in zero weather for a single ward with one exposed wall and a single window, when upon the south side of the building.

For northerly rooms, 40 square feet should be used. In exposed locations, the heaters may be made 40 and 50 square feet for north and south rooms respectively. The standard pin-radiators rated at 10 square feet of heating surface per section, are commonly used for this purpose. In case hot water is used, the same number of sections of the deep-pin pattern rated at 15 square feet each may be employed, making a total of 45 and 60 square feet per room. For corner rooms having two exposed walls and two windows, the amount of radiation should be increased about 50 per cent over that given above.

The wards are usually furnished with fireplaces which provide for the discharge ventilation. In case the fireplaces are omitted, a special vent flue, either of brick or of galvanized iron, should be provided. These should not be less than 8 by 12 inches for single wards, and the equivalent for each bed in a large ward. Each flue of this kind should have a loop of steam pipe for producing a draught. A loop of 1-inch pipe, 10 or 12 feet in height, is usually sufficient for this purpose.

Other rooms than wards are usually heated with direct radiators, the sizes of which may be computed in the same manner as for dwelling-houses.

Steam tables for the kitchen, sterilizers, and laundry machinery, require higher pressures than is necessary for heating.

In large plants the boilers are usually run at high pressure, and the pressure reduced for heating. A good arrangement for small plants is to provide sufficient boiler power for warming and ventilating purposes, and run at a pressure of 3 to 5 pounds. In addition to this, a small high-pressure boiler carrying 70 or 80 pounds should be furnished for laundry work and water heating.

Churches. Churches may be warmed by furnaces, by indirect steam, or by means of a fan. For small buildings the furnace is more commonly used. This apparatus is the simplest of all and is comparatively inexpensive. Heat may be generated quickly, and when the fires are no longer needed, they may be allowed to go out without danger of damage to any part of the system from freezing.

It is not usually necessary that the heating apparatus be large enough to warm the entire building at one time to 70 degrees with frequent change of air. If the building is thoroughly warmed before occupancy, either by rotation or by a slow inward movement of outside air, the chapel or Sunday-school room may be shut off until near the close of the service in the auditorium, when a portion of the warm air may be turned into it. When the service ends, the switch-damper is opened wide, and all the air is discharged into the Sunday-school room. The position of the warm-air registers will depend somewhat upon the construction of the building, but it is well to keep them near the outer walls and the colder parts of the room. Large inlet registers should be placed in the floor near the entrance doors, to stop cold draughts from blowing up the aisles when the doors are opened, and also to be used as foot-warmers.

Ceiling ventilators are generally provided, but should be no larger than is necessary to remove the products of combustion from the gaslights, etc. If too large, much of the warmest and purest air will escape through them. The main vent flues should be placed in or near the floor and should be connected with a vent shaft leading outboard. This flue should be provided with a small stove or flue heater made specially for this purpose. In cold weather the natural draught will be found sufficient in most cases.

The same general rules are to be followed in the case of indirect steam as have been described for furnace heating. The stacks are placed beneath the registers or flues, and mixing dampers provided. If there are large windows, flues should be arranged to open in the window-sills, so that a sheet of warm air may be delivered in front of the windows, to counteract the effects of cold down-draughts from the exposed glass. These flues may usually be made 3 or 4 inches in depth, and should extend the entire width of the window. Small rooms, such as vestibules, library, pastor's room, etc., are usually heated with direct radiators. Rooms which are used during the week are often connected with an independent heater so that they may be warmed without running the large boilers, as would otherwise be necessary.

When a fan is used, it is desirable, if possible, to deliver the air to the auditorium through a large number of small openings. This is often done by constructing a shallow box under each pew, running its entire length, and connecting it with the distributing ducts or a plenum space by means of a pipe from below. The air is delivered at a low velocity through a long slot, as shown in Fig. 174.

The warm-air flues in the window-sills should be retained, but may be made shallower, and the air forced in at a high velocity.

If the auditorium has a sloping floor, a plenum space may be provided between the upper or raised portion and the main floor. Sometimes a shallow basement 3 or 4 feet in height, with a cemented floor, and extending under the entire auditorium, is used as an air or plenum space.

If the basement is of good height and used for storage or other purposes, it is necessary to carry galvanized-iron ducts at the ceiling under the center of each double row of pews, and to connect with each pair by means of branch uptakes. The size of these should be equal to 3 or 4 square inches for each occupant.

Another method is to supply the air through a small register in the end of each pew. This simplifies the pew construction somewhat, but otherwise is not so satisfactory as the preceding method.

If the special pew construction is too expensive, or for any other reason cannot well be used, and the fan is to be retained, the greater part of the air is best introduced through wall registers placed about 8 feet above the floor, with exhaust openings at or near the floor. By this arrangement the air is thrown horizontally toward the center of the church, and much of it falls to the breathing level without rising to the upper part of the room.

Halls. The treatment of a large audience hall is similar to that of a church, the warming being usually done in one of the three ways already described. Where a fan is used, the air is commonly delivered through wall registers placed in part near the floor, and partly at a height of 7 or 8 feet above it. They should be made of ample size, so that there will be freedom from draughts. A part of the vents should be placed in the ceiling, and the remainder near the floor. All ceiling vents, in both halls and churches, should be provided with dampers having means for holding them in any desired position. If indirect gravity heaters are used, it will generally be necessary to place heating coils in the vent flues for use in mild weather; but if the fresh air is supplied by means of a fan, there will usually be pressure enough in the room to force the air out without the aid of other means. When the vent air-ways are restricted, or the air is impeded in any way, electric ventilating fans are often used. These give especially good results in warmer weather, when natural ventilation is sluggish. The temperature may be regulated either by using the double-duct system or by shutting off or turning on a greater or less number of sections in the main heater. After an audience hall is once warmed and filled with people, very little heat is required to keep it comfortable, even in the coldest weather.

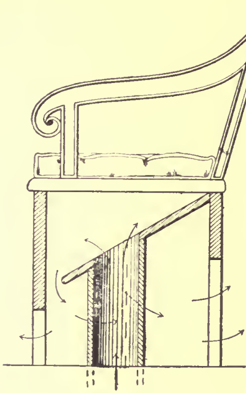


Fig. 174. An Approved Method of Delivering Warm Air to the Auditorium of a Church.

Theaters. In designing heating and ventilating systems for theaters, a wide experience and the greatest care are necessary to secure the best results. A theater consists of three parts: the body of the house, or auditorium; the stage and dressing-rooms, and the foyer, lobbies, corridors, stairways, and offices. Theaters are usually located in cities, and surrounded with other buildings on two or more sides, thus allowing no direct connection by windows with the external air; for this reason artificial means are necessary for providing suitable ventilation, and a forced circulation by means of a fan is the only satisfactory means of accomplishing this. It is usually advisable to create a slight excess of pressure in the auditorium, in order that all openings shall allow for the discharge rather than the inward leakage of air.

The general and most approved method of air-distribution is to force it into closed spaces beneath the auditorium and balcony floors, and allow it to discharge upward through small openings among the seats. One of the best methods is through chair-legs of special latticed design, which are placed over suitable openings in the floor; in this way the air is delivered to the room in small streams, at a low velocity, without draughts or currents. The discharge ventilation should be largely through ceiling vents, and this may be assisted if necessary by the use of ventilating fans. Vent openings should also be provided at the rear of the balconies, either in the wall or in the ceiling, and these should be connected with an exhaust fan either in the basement or in the attic, as is most convenient.

The close seating of the occupants produces a large amount of animal heat, which usually increases the temperature from 6 to 10 degrees, or even more; so that, in considering a theater once filled and thoroughly warmed, it becomes more of a question of cooling than one of warming to produce comfort.

The dressing-rooms should be provided with a generous supply of fresh air, sufficient to change the entire contents once in 10 minutes at least, and should have discharge flues of sufficient size to carry away this amount of air at a velocity not exceeding 300 feet per minute, unless connected with an exhaust fan, in which case the velocity may be doubled. The foyer, corridors, dressing-rooms, etc., are generally heated by direct radiators, which may be concealed by ornamental screens if desired.

Office Buildings. This class of buildings may be satisfactorily warmed by direct steam, hot water, or, where ventilation is desired, by the fan system. Probably direct steam is used more frequently than any other system for this purpose. Vacuum systems are well adapted to the conditions usually found in this type of building, as most modern office buildings have their own light and power plants, and the exhaust steam can thus be utilized for heating purposes. The piping may be either single or double. If the former is used, it is better to carry a single main riser to the upper story, and run drops to the basement, as by this means the steam and water flow in the same direction, and much smaller pipes can be used than would be the case if risers were carried from the basement upward.

Special provision must be made for the expansion of the risers or drops in tall buildings. They are usually anchored at the center, and allowed to expand in both directions. The connections with the radiators must not be so rigid as to cause undue strains or to lift the radiators from the floor.

It is customary, in most cases, to make the connections with the end farthest from the riser; this gives a length of horizontal pipe which has a certain amount of spring, and will care for any vertical movement of the riser that is likely to occur. Forced hot-water circulation is often used in connection with exhaust steam. The water is warmed by the steam in large heaters similar to feed-water heaters and is circulated through the system by means of centrifugal pumps. This has the usual advantage of hot water over steam, inasmuch as the temperature of the radiators may be regulated to suit the conditions of outside temperature.

When a fan system is used the arrangement of the air-ways is usually somewhat different from any of those yet described. Owing to the great height of these buildings, and the large number of small rooms which they contain, it is impossible to carry up separate flues from the basement. One of the best arrangements is to construct false ceilings in the corridor-ways on each floor, thus forming air-ducts which may receive their supply through one or more large up-takes extending from the basement to the top of the building. These corridor air-ways may be tapped over the door of each room, the openings being provided with suitable regulating dampers for gauging the air-supply to each. Adjustable deflectors should be placed in the main air-shafts for proportioning the quantity to be delivered to each floor. If both supply and discharge ventilation are to be provided, the fresh air may be carried in galvanized-iron ducts within the ceiling spaces, and the remainder used for conveying the exhausted air to uptakes leading to a discharge fan placed upon the roof of the building. In both of these cases, it is assumed that heat is supplied to the rooms by direct radiation, and that the air-supply is for ventilation only.

Apartment Houses. These are warmed by furnaces, direct steam, and hot water. Furnaces are more often used in the smaller houses, as they are cheaper to install, and require a less skilful attendant to operate them. Steam is probably used more than any other system in blocks of larger size. A well-designed single-pipe connection, with automatic air-valves dripped to the basement, is probably the most satisfactory in this class of work. People who are more or less unfamiliar with steam systems are apt to overlook one of the valves in shutting off or turning on steam; and where only one valve is used, the difficulty arising from this is avoided. Where pet-cock air-valves are used, they are often left open through carelessness; and the automatic valves, unless dripped, are likely to give more or less trouble.

Greenhouses and Conservatories. Buildings of this class are heated in some cases by steam and in others by hot water, some florists preferring one and some the other. Either system, when properly designed and constructed, should give satisfaction, although hot water has its usual advantage of a variable temperature. The methods of piping are, in a general way, like those already described, and the pipes may be located to run underneath the beds of growing plants or above, as bottom or top heat is desired. The main is generally run near the upper part of the greenhouse and to the farthest extremity, in one or more branches, with a pitch upward from the heater for hot water and with a pitch downward for steam. The principal radiating surface is made of parallel lines of $1\frac{1}{2}$ inch or larger pipe, placed under the benches and supplied by the return current. Figs. 175, 176, and 177 show a common method of running the piping in greenhouse work. Fig. 175 shows a plan and elevation of the building with its lines of pipe; and Figs. 176 and 177 give details of the pipe connections of the outer and inner groups of pipes respectively.

Any system of piping which gives free circulation and which is adapted to the local conditions, should give satisfactory results. The radiating surface may be computed from the rules already given. As the average greenhouse is composed almost entirely of glass, we

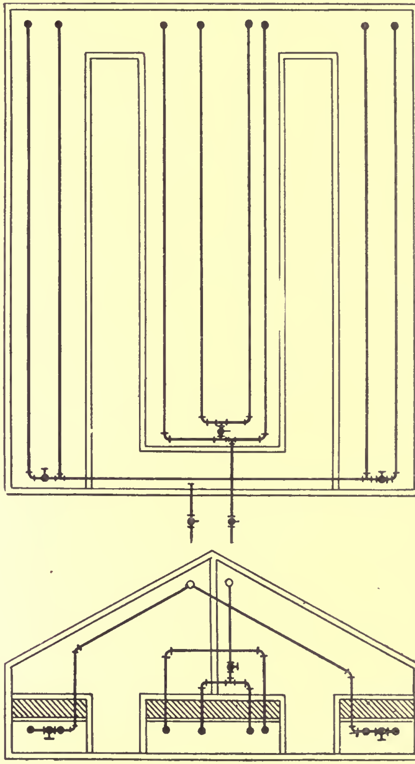


Fig. 175. Plan and Elevation Showing One Method of Running Piping in a Greenhouse

may for purposes of calculation consider it such; and if we divide the total exposed surface by 4, we shall get practically the same result as if we assumed a heat loss of 85 B. T. U. per square foot of surface per hour, and an efficiency of 330 B. T. U. for the heating coils; so that we may say, in general, that the square feet of radiating surface required equals the total exposed surface, divided by 4 for steam coils, and by 2.5 for hot-water. These results should be increased from 10 to 20 per cent for exposed locations.

CARE AND MANAGEMENT

The care of furnaces, hot-water heaters, and steam boilers has been discussed in connection with the design of these different systems of heating, and need not be repeated. The management of the heating and ventilating systems in large school buildings is a matter of much importance, especially in those using a fan system. To obtain the best results, as much depends upon the skill of the operating engineer as upon that of the designer.

Beginning in the boiler-room, he should exercise special care in the management of his fires, and the instruction given in "Boiler Accessories" should be carefully followed; all flues and smoke passages should be kept clear and free from accumulations of soot and ashes by means of a brush or steam jet. Pumps and engine should be kept clean and in perfect adjustment, and extra care should be taken when they are in rooms through which the air-supply is drawn, or the odor of oil will be carried to the rooms. All steam traps should be examined at regular intervals to see that they are in working order; and upon any sign of trouble, they should be taken apart and carefully cleaned.

The air-valves on all direct and indirect radiators should be inspected often; and upon the failure of any room to heat properly, the air-valve should first be looked to as a probable cause of the difficulty. Adjusting dampers should be placed in the base of each flue, so that the flow to each room may be regulated independently. In starting up a new plant, the system should be put in proper balance by a suitable adjustment of these dampers; and, when once adjusted, they should be marked, and left in these positions. The temperature of the rooms should never be regulated by closing the inlet registers. These should never be touched unless the room is to be unused for a day or more.

In designing a fan system, provision should be made for *air-rotation*; that is, the arrangement should be such that the same air may be taken from the building and passed through the fan and

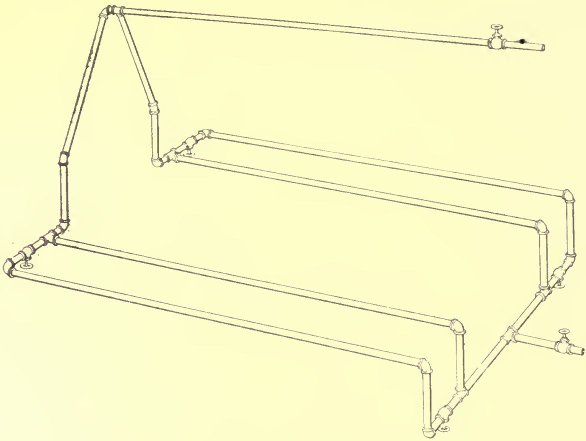


Fig. 176. Connections of Outer Groups of Pipes of Greenhouse Shown in Fig. 175.

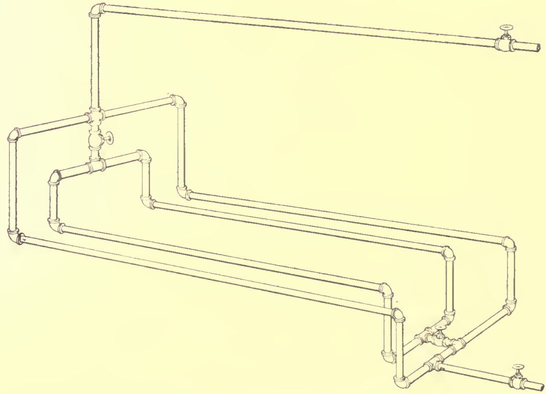


Fig. 177. Connections of Inner Groups of Pipes of Greenhouse Shown in Fig. 175.

heater continuously. This is usually accomplished by closing the main vent flues and the cold-air inlet to the building, then opening the class-room doors into the corridor-ways, and drawing the air down the stair-wells to the basement and into the space back of the main heater through doors provided for this purpose. In warming up a building in the morning, this should always be done until about fifteen minutes before school opens. The vent flues should then be opened, doors into corridors closed, cold-air inlets opened wide, and the full volume of fresh air taken from out of doors.

At night time the dampers in the main vents should be closed, to prevent the warm air contained in the building from escaping. The fresh air should be delivered to the rooms at a temperature of from 70 to 75 degrees; and this temperature must be obtained by proper use of the shut-off valves, thus running a greater or less number of sections on the main heater. A little experience will show the engineer how many sections to carry for different degrees of outside temperature. A dial thermometer should be placed in the main warm-air duct near the fan, so that the temperature of the air delivered to the rooms can be easily noted.

The exhaust steam from the engine and pumps should be turned into the main heater; this will supply a greater number of sections in mild weather than in cold, owing to the less rapid condensation.

TYPE OF MODERN AMERICAN BATH ROOM WITH LATEST APPROVED FITTINGS.
The Federal Company.

PLUMBING.

PART I.

PLUMBING FIXTURES.

Bath Tubs. There are many varieties of bath tubs in use at the present time, ranging from the wooden box lined with zinc or copper which was in common use a number of years ago and is still to be found in the old houses, to the finest crockery and enameled tubs which are now used in the best modern plumbing. In selecting a tub we should choose one with as little woodwork about it as possible. Those lined with zinc or copper are hard to keep clean and are liable to leak and are, therefore, undesirable from a sanitary standpoint. The plain cast iron tub, painted, is the next in cost. This makes a serviceable and satisfactory tub if

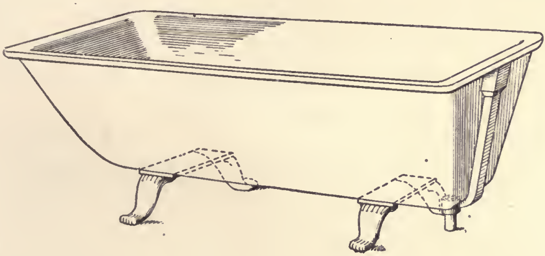


Fig. 1.

kept painted; it is used quite extensively in asylums, hospitals, etc. One of this type is shown in Fig. 1. These are sometimes galvanized instead of being painted.

The "steel-clad" tub shown in Fig. 2 is a good form for a low-priced article. This tub is formed of sheet steel and has a lining of copper. This form is light and easy to handle; it is an open fixture the same as the cast iron tub and requires no casing. It is provided with cast iron legs and a wooden cap. Probably the most common form to be found in the average house at the present time is the porcelain lined iron tub as shown in Fig. 3.

This has a smooth interior finish and is easily kept clean. It will not, however, stand the hard usage of those above described as the lining is likely to crack if struck by any hard substance.

In Fig. 4 is shown a crockery or porcelain tub arranged for needle and shower baths. This is a most sanitary article in every respect and requires no woodwork of any kind; being made of one

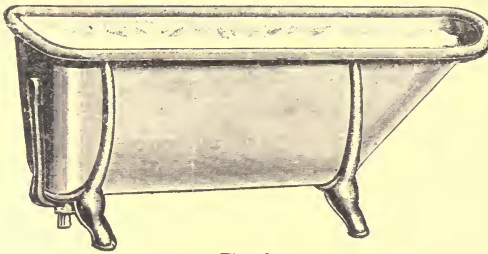


Fig. 2.

piece, there is no chance for dirt to collect. It is a heavy tub and requires great care in handling. This material is very cold to the touch until it has become thoroughly warmed by the hot water. Fig. 5 shows a seat bath and Fig. 6 a foot bath, both of which are

Fig. 3.

very convenient and should be placed in all well equipped bath rooms if the expense does not prohibit their use.

Water Closets. There is a great variety of water closets from which to choose, many operating upon the same principle but varying slightly in form and finish. The best are made of porcelain, the bowl and trap being in one piece without corners or crevices so that they are easily kept clean. The top of the bowl is provided with a wooden rim and cover. The general arrangement of seat and flushing tank is shown in Fig. 7. A section through the bowl is shown in Fig. 8. This type is known as a

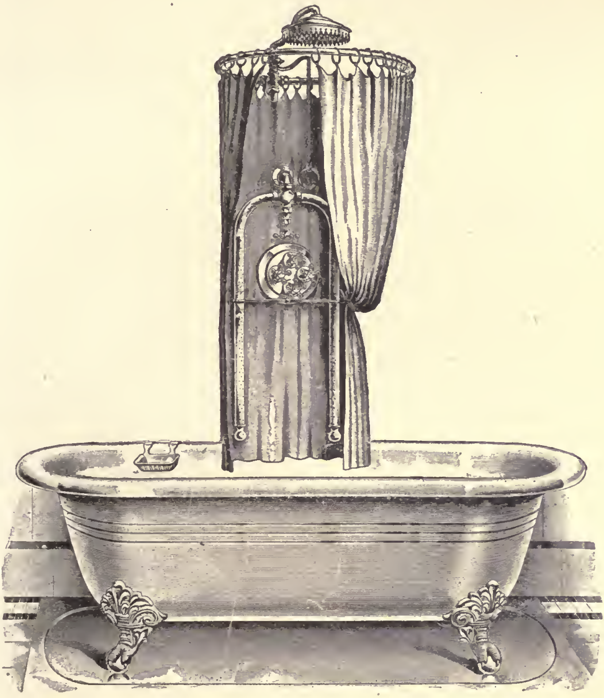


Fig. 4.

syphon closet, and those made on this principle are probably the most satisfactory of any in present use. They are made in different forms by various manufacturers but each involves the principle which gives it its name. Water stands in the bottom as shown, thus forming a seal against gases from the sewer.

When the closet is flushed, water rushes down the pipe and fills the small chamber at the rear which discharges in a jet at the bottom as shown by the arrow. The syphon action thus set up draws the entire contents of the bowl over into the soil pipe. In the meantime a part of the water from the tank fills the hollow rim of the bowl and is discharged in a thin stream around the

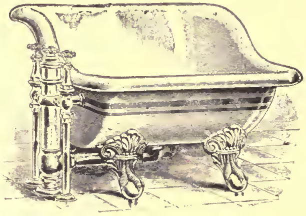


Fig. 5.

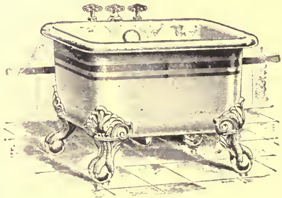


Fig. 6.

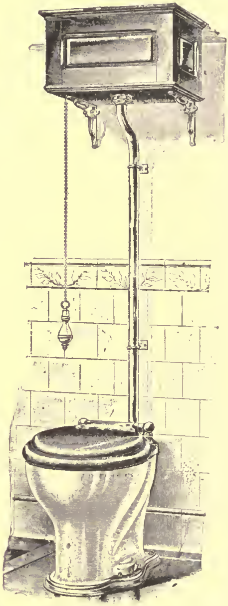


Fig. 7.

entire perimeter which thoroughly washes the inside of the bowl each time it is flushed. Fig. 9 shows a form called the "wash-out" closet. In this case the whole of the water is discharged through the flushing rim but with greater force at the rear which washes the contents of the upper bowl into the lower which overflows into the soil pipe. This is a good form of closet and is widely used. A similar form, but without the upper bowl is shown in Fig. 10. This is known as the "wash down" closet and operates in the manner already described. The water enters the bowl through the flushing rim and discharges its contents by

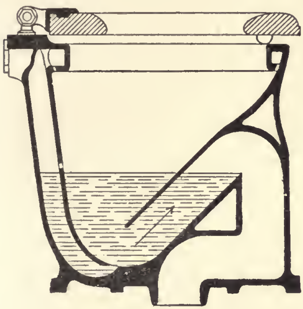


Fig. 8.

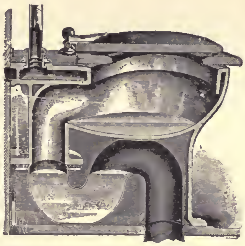


Fig. 9.

overflowing into the soil pipe. This is a simple form of closet and easily kept clean.

One of the simplest closets is the "hopper" shown in Fig. 11. This consists of a plain bowl of porcelain or cast iron tapering to

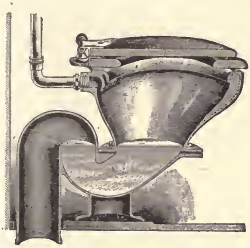


Fig. 10.

Fig. 11.

an outlet about 4″ in diameter at the bottom. It is connected directly with the soil pipe as shown. The trap may be placed either above the floor or below as desired. They are provided with a flushing rim at the top similar to that already described. This type of closet is the cheapest but at the same time the least satisfactory of any of the different kinds shown.

It is sometimes desirable to place a closet in a location where there would be danger of freezing if the usual form of flushing tank was used. Fig. 12 shows an arrangement which may be used in a case of this kind. The valve and water connections are placed below the frost line and a pipe not shown in the cut is carried up to the rim of the bowl. When the rim is shut down the

Fig. 12.

valve is opened by means of the chain attached to it and water flows through the bowl while in use. When released, the weight on the lever closes the valve and raises the wooden rim to its original position. Any water which remains in the flush pipe is drained to the soil pipe through a small drip pipe which is seen in the cut.

Urinals. A common form of urinal is shown in Fig. 13. The partitions and slab at the back are either of slate or marble and the bowl of porcelain. They may be flushed like a closet. Fig. 14 shows a section through the bowl and indicates the manner of flushing, partly through the rim and partly at the back. The trap or seal is shown at the bottom. Another form is shown in Fig. 15. In this case the bowl remains partly filled with water which forms a seal as shown. It is flushed both through the rim and the passage at the back. In action it is the same as the syphon closet shown in Fig. 8 and the bowl is drained each

Fig. 13.

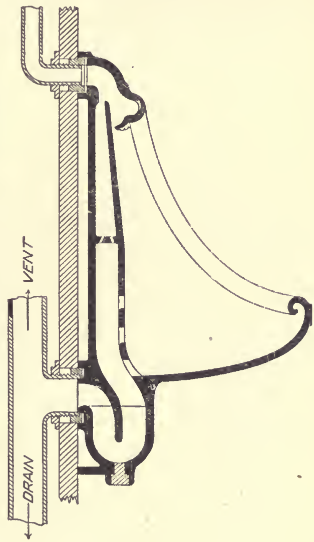


Fig. 14.

time it is flushed, but immediately fills with water to the level indicated.

An automatic flushing device is illustrated in Fig. 16. When the water line in the tank reaches a given level, the float lever releases a catch and flushes the urinal. The intervals of flushing can be regulated by adjusting the cock shown in the inlet pipe, near the bottom of the tank.

A simple form of urinal commonly used in schools and public buildings is shown in Fig. 17. This is flushed by means of small streams of water which are discharged through the perforated pipe near the top of the slab at the back and run down in a thin sheet to the gutter at the bottom.

Lavatories. Bowls and lavatories can be had in almost any form. Fig. 18 shows a simple corner lavatory, made of porcelain and provided

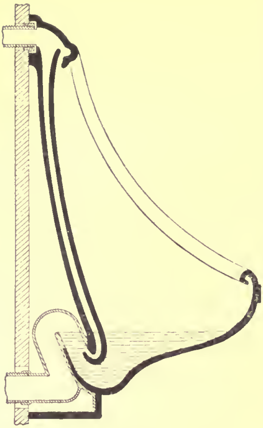


Fig. 15.

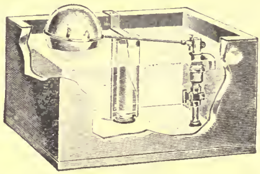


Fig. 16.

Fig. 17.

with hot and cold water faucets. It has an overflow, shown by the small openings at the back and a rubber plug for closing the drain at the bottom.

The lavatory shown in Fig. 19 is provided with marble slabs and is more expensive. Fig. 20 shows a section through the bowl. The waste pipe is at the back, which brings the plug and chain well out of the way. A pattern still more elaborate is shown in Fig. 21, and a section through the bowl in Fig. 22. The waste pipe plug in this case is in the form of a hollow tube and acts as an overflow when closed and as a strainer when open. It is held open by means of a slot and

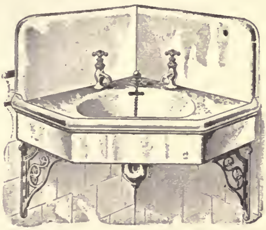


Fig. 18.

Fig. 19.

pin near the top. Fig. 23 shows a bowl so arranged that either hot, cold or tepid water may be drawn through the same opening which is placed well down in the bowl where it is out of the way.

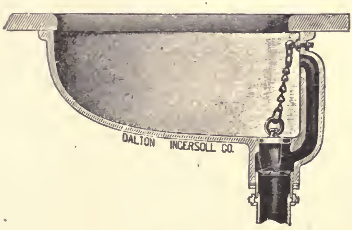


Fig. 20.

Sinks. Sinks are made of plain wood, and of wood lined with sheet metal, such as copper, zinc or galvanized iron. They are also made of sheet steel, cast iron, either plain, galvanized or enameled, and of soapstone and porcelain. Each has its advantages and disadvantages. The wooden sink is liable to leak, and is difficult to keep thoroughly clean. The lined sink is most satisfactory when new, but holes are quite easily cut or punched through the lining and it then becomes very objectionable from a sanitary standpoint as the greasy water and vegetable matter which works through the opening causes the woodwork to decay rapidly and to give off in the process a gas which is not only unhealthful but tends to destroy the lining of the sink from the underside so that its destruction is rapid after a leak is once started. The cast iron sink is satisfactory. The appearance is improved by galvanizing, but this soon wears off on the inside. Enameled sinks are easily kept clean but likely to become cracked or broken from hard usage or from extremes of hot or cold; the porcelain sink has the same defects;

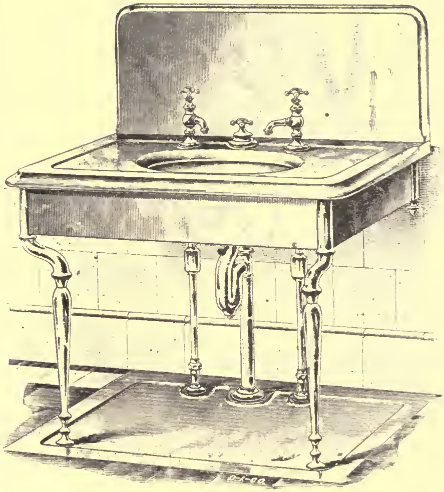


Fig. 21.

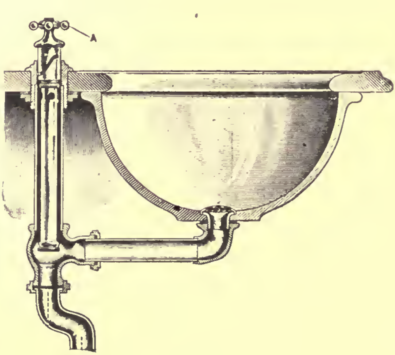


Fig. 22.

they are both however well adapted to places where they will receive careful usage.

Taking all points into consideration the soapstone sink may perhaps be considered the most satisfactory for all-around use.

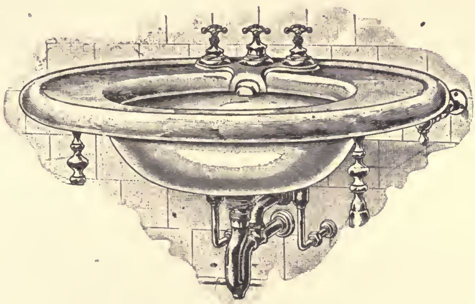


Fig. 23.

It will not absorb moisture ; is not affected by the action of acids; oil or grease will not enter the pores and it is not injured by hot water nor liable to crack.

Fig. 24 shows the ordinary cast iron sink, made to be set in a wooden casing ; this is not to be recommended however, and it is

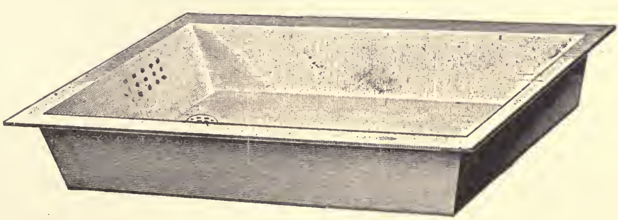


Fig. 24.

much better to support them upon iron brackets or legs. Fig. 25 shows an enameled sink mounted in this way. A porcelain sink with dish racks is shown in Fig. 26. This is a good form for a pantry sink which is used only for washing cutlery, glassware, crockery, etc., and is not subjected to hard usage. A slop sink is shown in Fig. 27. This, as will be noticed, is provided with an extra large waste pipe and trap to prevent clogging. These sinks are made of cast iron with different finishes and of porcelain.

Set tubs for laundry use are made of soapstone, slate, cast

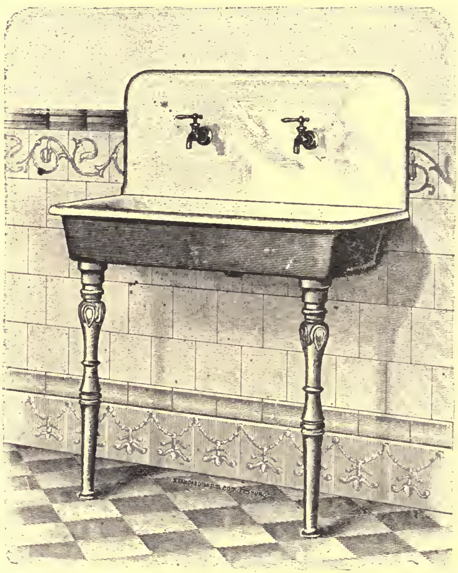


Fig. 25.

iron (enameled or galvanized) and of porcelain. What has been said in regard to kitchen sinks applies equally well in this case.

A set of enameled tubs is shown in Fig. 28.

Traps. A trap is a loop or water seal placed in a pipe to prevent the gases from the drain or sewer from passing up through the waste pipes of the fixtures into the rooms. A common form made up of cast iron pipe and known as a "running trap" is shown in Fig. 29. A trap of this form is placed in the main drain pipe of a building outside of all the connections to prevent gases from the main sewer or cesspool from entering the building. A removable cover is placed on top of the trap to give access for cleaning.

The floor trap shown in section in Fig. 30 is made both of brass and of lead. It is commonly used for kitchen sinks and is placed on the floor just beneath the fixture. It is provided with a removable trap screw or clean-out for use when it is desired

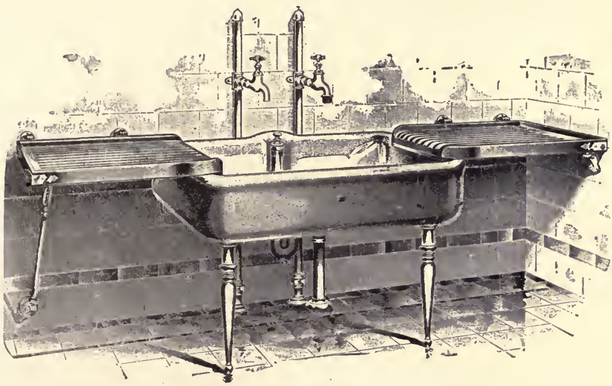


Fig. 26.

to remove grease or sediment from the interior. Fig. 31 shows a common form for lavatories, which consists simply of a loop in the waste pipe. These are usually made of brass and nickle plated when used with open fixtures. A trap for similar purposes is shown in Figs. 32 and 33.

Figs. 34 and 35 show a form known as the centrifugal trap on account of the rotary or whirling motion given to the water by the peculiar arrangement of the inlet and outlet. This motion carries all solid particles to the outside and discharges them with the water, thus keeping the trap clear of sediment. Where there is likely to be a large amount of grease in the water as in the case of waste from a hotel or restaurant it becomes necessary to use a special form of separating trap to prevent the waste pipes from becoming clogged. A grease trap designed for this purpose is shown in Fig. 36. Its action is readily seen as the fatty matter will be separated, first by dropping into a large body of cold water and then being driven against the center partition before an outlet can be gained. The grease then rises to the surface where it cools and can then be easily removed as often as necessary.

Sometimes a cellar or basement is drained into a sewer which

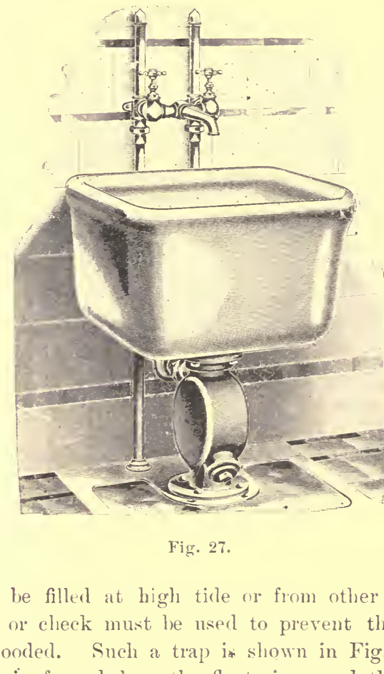


Fig. 27.

is liable to be filled at high tide or from other causes and a special trap or check must be used to prevent the cellar from becoming flooded. Such a trap is shown in Fig. 37. When water flows in from below, the float rises, and the rubber rim pressing against the valve seat prevents any passage through the trap; the cut shows the valve closed by the action of high water.

Tanks or cisterns for flushing closets or other fixtures are usually made of wood and lined with zinc or copper. These are generally placed inside a finished casing. A common form is shown in Fig. 38. The arrangement of valves for supplying water to the tank and for flushing the fixtures is shown in Fig. 39. The large float or ball cock regulates the flow of water into the tank from the street main or house tank. When the water in the tank

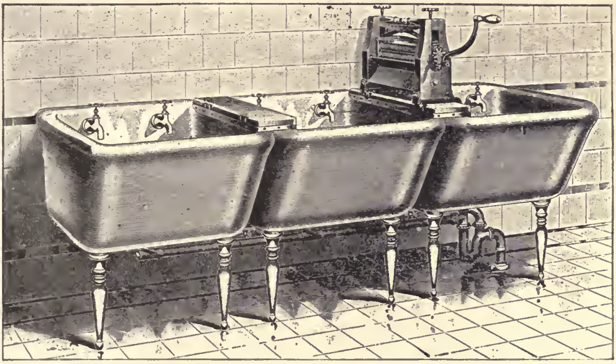


Fig. 28.

falls below a certain level the float drops and opens a valve, thus admitting more water, and closes again when the tank is filled. The closet is flushed by pulling a chain attached to the lever at the right which opens the valve in the bottom of the tank and admits water to the flushing pipe. In this form the valve remains open only while the lever is held down by the chain, the weight on the other end of the lever closing the valve as soon as the chain is released. Another form which is partially automatic is shown in Fig. 40. When the chain is pulled it raises the central valve from its seat and allows the water to flow down the flush pipe

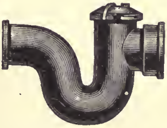


Fig. 29.

until the tank is nearly empty. When empty, the strong suction seals the valve which remains closed until the chain is again pulled. In this type of valve a single pull of the chain is sufficient to flush the closet without further attention.

A purely automatic flushing device is shown in Fig. 41.

The chain in this case is attached to the rim of the seat so that when it is pressed down, the valve in the compartment at the bottom, connecting with the flush pipe is closed and at the same time

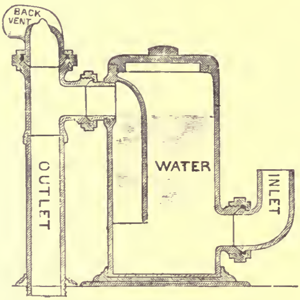


Fig. 30.

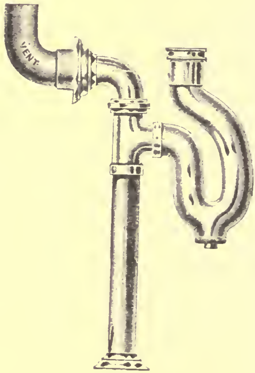


Fig. 31.

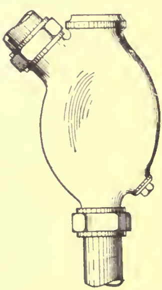


Fig. 32.

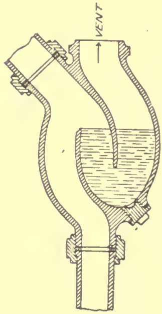
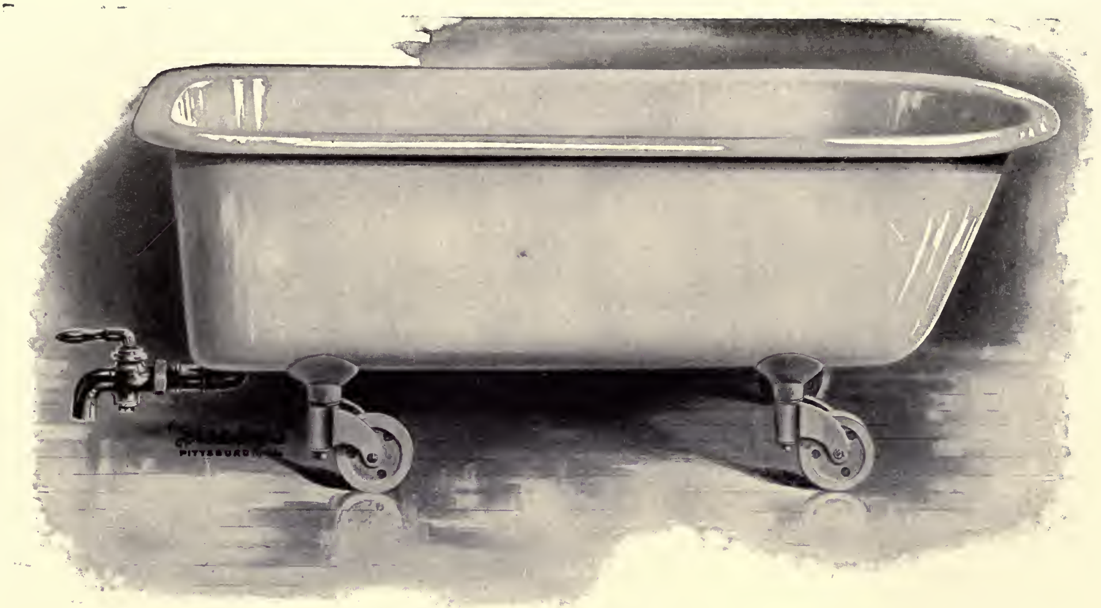


Fig. 33.

communication is opened between the two compartments. When the pull on the chain is released the valve connecting the flush pipe is opened and the opening between the compartments closed



Standard Bath Tub for Hospital Use.

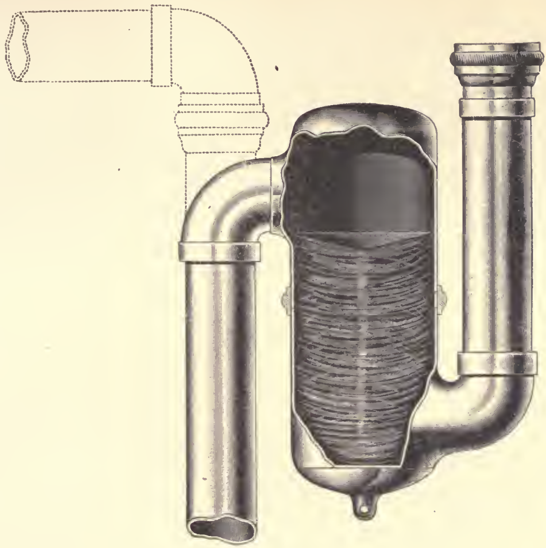


Fig. 34.

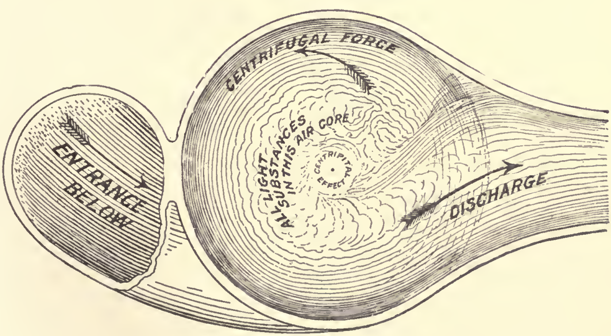

CENTRIFUGAL FORCE
ALL LIGHT SUBSTANCES IN THIS AIR CORE
CENTRIPITAL EFFECT
ENTRANCE BELOW
DISCHARGE

Fig. 35.

so that the water in the lower portion of the tank flows through the flush pipe into the closet automatically, and when empty no more can be admitted until the lever is again pulled down and the valve in the partition opened.

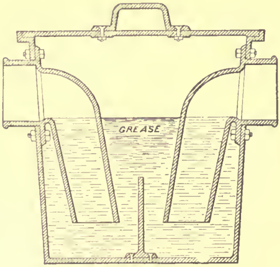


Fig. 36.

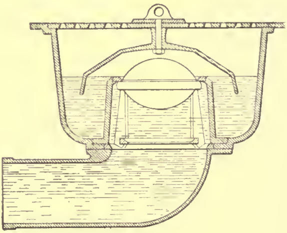


Fig. 37.

Faucets. There are many different forms of faucets in use. The most common is the compression cock shown in Fig. 42. This has a removable leather or asbestos seat which requires renewing from time to time as it becomes worn. Fig. 43 shows a similar form, in which the valve seat is free to adjust itself, being held in place by a spring. Another

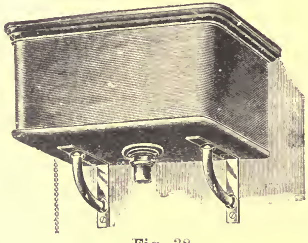


Fig. 38.

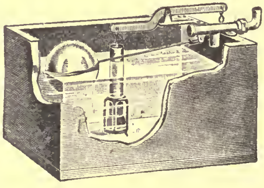


Fig. 39.

style often used in hotels and other public places is the self-closing faucet. These are fitted with springs in such a way that they remain closed except when held open. Two different forms are shown in Figs. 44 and 45.

There are various arrangements for mixing the hot and cold water for bowls and bath tubs before it is discharged. This is accomplished by having both faucets connect with a common nozzle. Such a device for a lavatory is shown in Fig. 46.

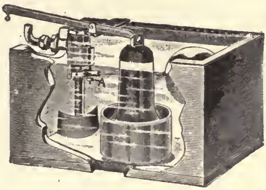


Fig. 40.

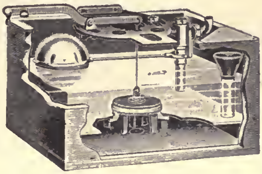


Fig. 41.

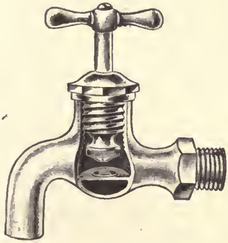


Fig. 42.

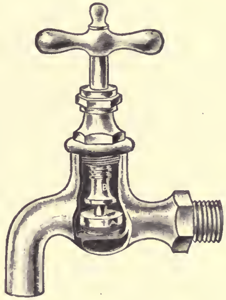


Fig. 43.

SOIL AND WASTE PIPES.

Cast-Iron Pipe. There are many different forms of soil pipes and fittings, and one can best acquaint himself with these by looking over the catalogues of different manufacturers. Figs. 47 and 48 show two lengths of soil pipe; the first is the regular pattern, having only one hub, and the second is a length of double-hub pipe; this can be used to good advantage where many short pieces are required.

Figs. 49 to 57 show some of the principal soil pipe fittings. Figs. 49, 50, 51, 52 and 53 show quarter, sixth, eighth, sixteenth and return bends respectively, and by the use of these almost any desired angle can be obtained. Different lines of pipe may be connected by means of the Y and T-Y branches shown in Figs.

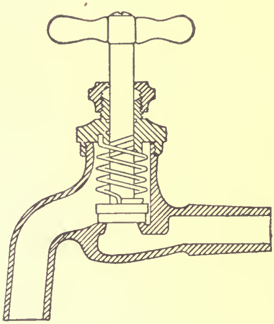


Fig. 44.

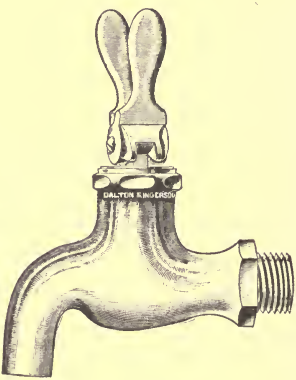


Fig. 45.

54, 55, 56 and 57. The T-Y fitting, Fig. 56, is used in place of the Y branch, Fig. 54, in cases where it is desired to connect two pipes which run perpendicular to each other.

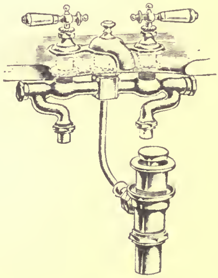


Fig. 46.

The double T-Y, Fig. 57, is convenient for use in double houses where a single soil pipe answers for two lines of closets.

Pipe Joints. Great care should be given to making up the joints in a proper manner, as serious results may follow any defective workmanship which allows sewer gas to escape into the building. In making up a joint, first place the ends of the pipes in position and fasten them rigidly, then pack the joint with the best picked oakum. In packing the oakum around the hub, the first layer must be twisted into a small rope so that it will drive in with ease and still not pass through to the inside of the pipe where the ends join.

In a 4-inch pipe the packing should be about 1 inch in thickness and calked perfectly tight so that it will hold water of itself without the lead. Just before the packing is driven tightly into

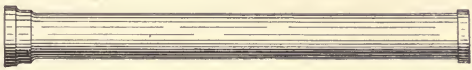


Fig. 47.

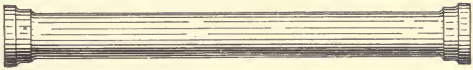


Fig. 48.

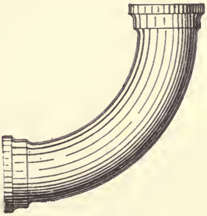


Fig. 49.

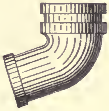


Fig. 50.

Fig. 51.

Fig. 52.

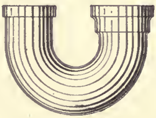


Fig. 53.

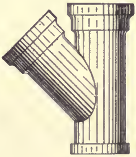


Fig. 54.

the hub, the joint should be examined to see that the space around the hub is the same, so that the lead will flow evenly and be of the same thickness at all points, as the expansion and contraction will work an imperfect joint loose much sooner than one in which the lead is of an even thickness all the way around. Only the best of clean soft lead should be used for this purpose. In calking in the lead after it has been poured, great care must be exercised, as the pipe, if of standard grade, is easily cracked and will stand but little shock from the calking chisel and hammer.

Fig. 58 shows a section through the calked joint of a cast iron soil pipe.

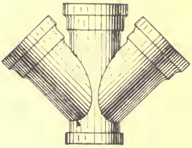


Fig. 55.

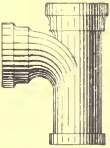


Fig. 56.

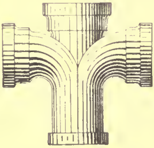


Fig. 57.

Wrought Iron Pipe. This is used but little in connection with the waste pipes except for the purpose of back venting where it may be employed with screwed joints the same as in steam work. It is sometimes used where only small drain pipes are

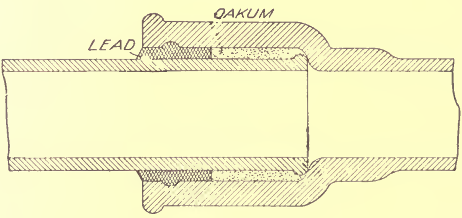


Fig. 58.

necessary, but is not desirable as it is likely to become choked with rust or to be eaten through by moisture from the outside.

Brass Pipe. Brass pipe, nickle plated, is largely used for connecting open fixtures, such as lavatories or bath tubs, with the soil pipe. It is common to use this for the exposed portions of the connections and to use lead for that part beneath the floor or in partitions. The various fittings are also made of brass and finished in a similar manner.

Lead Pipe. For sinks, bath tubs, laundry tubs, etc., nothing is better for carrying off the waste water than lead pipe, for the reason that it has a smooth interior surface which offers a small resistance to the flow of water, and does not easily collect dirt or sediment. It can also be bent in easy curves which is an advantage over fittings which make abrupt turns; this is especially important in pipes of small size.

Pipe Joints. There are two common methods of making joints in lead pipe, known as the "cup joint" and the "wipe joint." The first is suitable only on small pipes or very light pressures. This is made by flanging the end of one of the pipes and inserting the other, then filling in the flange with solder by means of a soldering iron, see Fig. 59. In making this joint great

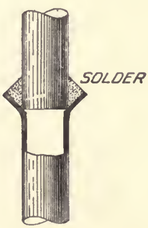


Fig. 59.

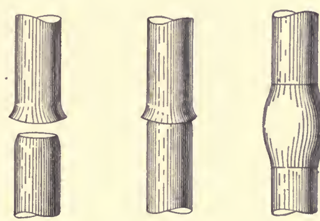


Fig. 60.

care should be taken that the ends of the pipes are round and fit closely so there will be no chance for the solder to run through inside the pipe and form obstructions for the collection of sediment.

The different stages of a wipe joint are shown in Fig. 60. The ends of the pipes are first cleaned and then fitted together as shown in the second stage. The solder is melted in a small cast iron crucible and is carefully poured on the joint or thrown on with a small stick called a "spatting stick." As the solder cools it becomes pasty and the joint can be worked into shape by means of the stick or a soft cloth, or both, depending upon the kind of joint and stage of operation. The final shape and smooth finish is given with the cloth. The ability to make a joint of this kind can be attained only by practice. and printed directions are of little value as compared with observation and actual practice. This is the strongest and most satisfactory joint that can be made between two lead pipes or a lead and brass or copper pipe. In the latter case the brass or copper should be carefully tinned as far as the joint is to extend by means of a soldering iron.

Where lead waste pipes are to be connected with cast iron soil pipes a brass ferule should be used. Different forms of these are shown in Figs. 61 and 62. The lead pipe is wiped to the finished end of the ferule while the other end is calked into the hub of the cast iron pipe in the manner already described. The ferule should be made heavy so as not to be injured in the proc-

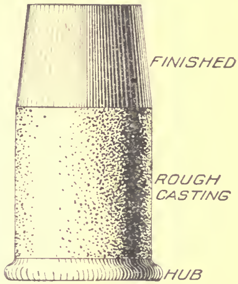


Fig. 61.

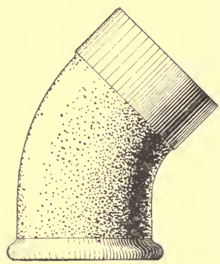


Fig. 62.

ess of calking. Cup joints should never be used for this purpose.

Tile Pipes. Nothing but metal piping should be used inside of a building, but in solid earth, starting from a point about 10 feet away from the cellar wall, we may use salt-glazed, vitrified, or terra cotta pipe for making the connection with the main sewer. This pipe is made in convenient lengths and shapes and is easily handled. Various fittings are made similar in form to those already described for cast iron. In laying tile pipe each piece should be carefully examined to see that it is smooth, round, and free from cracks. The ends should fit closely all around, and each length of pipe should fit into the next the full length of the hub. In making the joints nothing but the best hydraulic cement should be used, and great care should be taken that this is pressed well into the space between the two pipes. All cement that works through into the interior should be carefully removed by means of a swab or brush made especially for this purpose. The earth should be filled in around a pipe of this kind before the cement is set or else the joints are likely to crack. Fine soil should be filled in around the pipe to a depth of 3 or 4 inches, and rammed down solid, and the ditch may then be filled in without regard to the pipe. No tile pipe should be used inside of a house or nearer than about 10 feet for the reason it might not stand the pressure in case a stoppage should occur in the sewer. This kind of pipe is not intended to carry a pressure and when used in this way is seldom entirely filled with water. Joints between iron and tile piping are made with cement in the manner described for two sections of tile.

Cesspools. It is often desired to install a system of plumbing in a building in the country or in a village where there is no system of sewerage with which to connect. In this case it becomes necessary to construct a cesspool. This is always undesirable, but if properly constructed and placed at a suitable distance from the house and in such a position that it cannot drain into a well or other source of water supply it may be used with comparative safety. Especial care should be taken in the construction, and when in use it should be regularly cleaned. One form of cesspool is shown in Fig. 63. This consists of two brick chambers located at some distance from the building and in a position where the ground slopes away from it if possible. The larger chamber has a clean-out opening in the top which should be provided with an air-tight cover. An ordinary cast iron cover may be made sufficiently tight by covering it over with 3 or 4 inches of earth packed solidly in place. A vent pipe should be carried from the top to such a height that all gases will be discharged at an elevation sufficient to prevent any harm.

The smaller chamber is connected with the first by means of a soil pipe as shown. This chamber is arranged for absorbing the liquids and for this purpose is provided with lengths of porous tile radiating from the bottom as shown in the plan. The house drain connects with the larger chamber, which fills to the level of the overflow, then the liquid portion of the sewage drains over into chamber No. 2 and is absorbed through the porous tile branches. The solid part remains in chamber No. 1, and can be removed from time to time. A suitable trap should of course be placed in the house drain in the same manner as though connected with a street sewer. The safety of the cesspool will depend much upon its location, its general construction and care and the nature of the soil.

TRAPS AND VENTS.

Traps. The best method of connecting traps, and their actual value under all conditions, are matters upon which there is

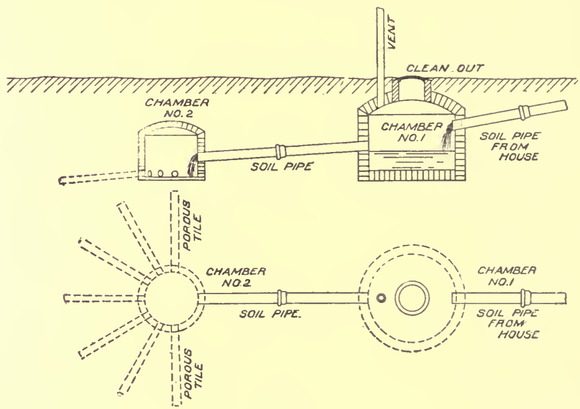


Fig. 63.

much difference of opinion. Cities also vary in their requirements to a greater or less extent, so that it will be possible to show in a general way only the various principles involved and to illustrate what is considered good practice, in the average case, at the present time.

A separate trap should in general be placed in the waste pipe from each fixture, although several of a kind, such as lavatories, etc., are often drained through a common trap, as shown in Fig. 64.

In addition to the traps at the fixtures a main or running trap is placed in the main soil pipe outside of all the connections;

this is sometimes placed in a manhole just outside the building, but more commonly in the cellar before passing through the wall; the former method is much to be preferred, as the trap may be cleaned without admitting gases or odors to the house. The running trap has been shown in Fig. 29, and is provided with a removable cap for cleaning.

The agencies which tend to destroy the water seal of traps

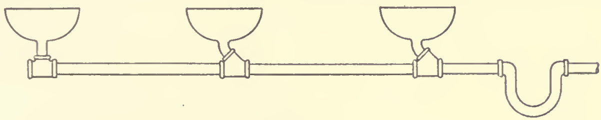


Fig. 64.

are siphonage, evaporation, back pressure, capillary action, leakage and accumulation of sediment.

Siphonage. This can best be illustrated by a few simple diagrams showing the principles involved. In Fig. 65 is shown a U tube with legs of equal length and filled with water. If we

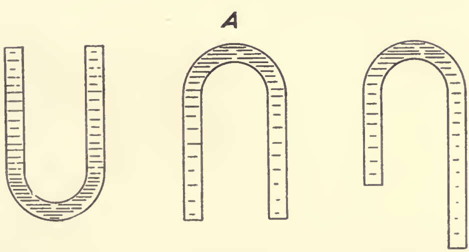


Fig. 65. Fig. 66. Fig. 67.

invert the tube, as shown in Fig. 66, the water will not run out, because the legs are of equal length, and contain equal weights of water, which pull downward from the top with the same force, tending to form a vacuum at the point A. If one of the legs is lengthened, as in Fig. 67, so that the column of water is heavier on one side than on the other, it will run out, while atmospheric pressure will force the water in the shorter tube up over the bend, as there would be no pressure to resist this action should the column of water break at this point. This action is also assisted by the adhesion of the particles of water to each other. The column of water in the tube may be likened to a piece of flexible rope hanging over a pulley; when equal lengths hang over each side it will remain stationary, but if drawn over one side slightly, so that one end is heavier than the other, the whole rope will be drawn over the pulley toward the longer and heavier end. The first cause, due to atmospheric pressure, is the principal reason for the action of siphons, but the latter assists it to some extent. If the shorter leg of the siphon be dipped in a vessel of water, as shown in Fig. 68, the atmospheric pressure, which before acted on the bottom of the water in the tube, is transferred to the surface of the water in the vessel, and the flow through the tube will continue until the water level in the vessel falls slightly below the end of the tube and admits air pressure, which breaks the siphon action. Fig. 69 shows the same principle applied to the trap of a sink or bowl. If the bowl is well filled with water, sc that when the plug is removed from the bottom, the waste pipe for some distance below the trap is filled with a solid column of water, a siphon action will be set up like the one just described, and the trap will be drained. Frequently a sufficient amount of water runs down from the fixture and sides of the pipe above the trap to partially restore the seal. This direct action of the water of a fixture in breaking its own trap seal by siphoning is called "self-siphonage."

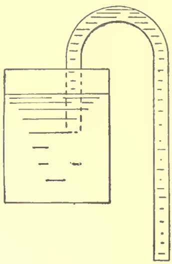


Fig. 68.

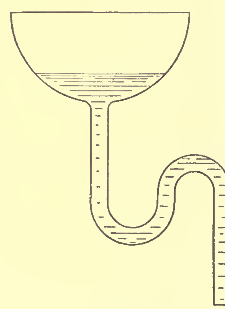


Fig. 69.

A more common form, where two or more fixtures connect with the same waste pipe, is shown in Fig. 70. In this case the seal of the lower closet is broken by the discharge of the upper. The falling column of water leaves behind it a partial vacuum in the soil pipe, and the outer air tends to rush into the pipe through the way of least resistance, which is often through the trap seals of the fixtures below. The friction of the rough sides of a tall soil pipe, even though it be open at the roof, will sometimes cause more resistance to air flow than the trap seals of the fixtures, with the result that they are broken, and gases from the drain are free to enter the building.

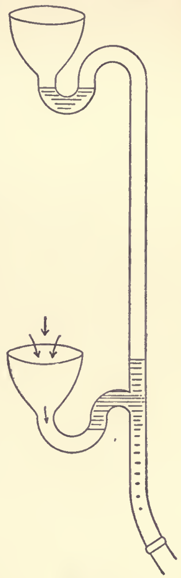


Fig. 70.

Three methods have been employed to prevent the destruction of the seal by siphonage. The first method devised was what is known as "back venting," and this is largely in use at the present time, although careful experiments have shown that in many cases it is not as effective as it was at first supposed to be, and is considered by some authorities to be a useless complication. It is, however, called for in the plumbing regulations of many cities, and will be taken up briefly in connection with other methods.

Back Venting. This consists in connecting a vent pipe at or near the highest part of the trap, as shown in Fig. 71. The action of this arrangement is evident; in place of the waste pipe receiving the air necessary to fill it, through the basin, after the solid column of water has passed down, it is drawn in through the vent pipe, as shown by the arrows, and the seal remains, or should remain, unbroken. It also prevents "self-siphonage" by breaking the column of water and admitting atmospheric pressure at the highest point or crown of the trap. The vent not only prevents the seal from being broken, as described, but allows any gases which may form in the waste pipe to escape above the roof of the house. In order to be effective, the back vent should be large, but even when of the same size as the waste pipe, the flushing of a closet will oftentimes break the seal, especially if the vent pipe is of considerable length. The vent often becomes choked, either with the accumulation of sediment near the trap or by frost or snow at the top; in this case its effect is of course destroyed. Another disadvantage of the back vent is the hastening of evaporation from the trap and the unsealing of fixtures which are not often used.

The second method of guarding against the loss of seal by

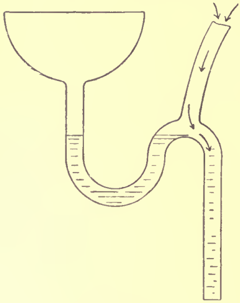


Fig. 71.

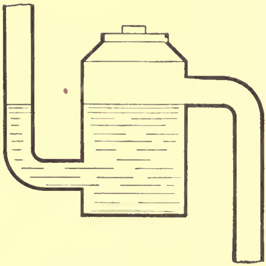


Fig. 72.

siphonage is to make the body of the trap so large that a sufficient quantity of water will always adhere to its sides after siphoning to restore a seal. The pot or cesspool trap shown in Fig. 72 is based on this principle.

The third method consists in the use of a trap of such form that it will not siphon, and will at the same time be self-cleaning. Among other types the centrifugal trap, shown in Figs. 34 and 35, is claimed to fulfil these conditions. The pot trap, while less affected by the siphoning action, is more or less objectionable on account of retaining much of the sediment and solid part of the sewage which falls into it.

Local Vents. A local vent is a pipe connected directly with a closet or urinal for carrying off any odor when in use. It has no connection with the soil pipe, unless the trap seal becomes broken, and is not provided for the purpose of carrying off gases from the sewer. A urinal provided with a local vent is shown in Fig. 73.

Sometimes a small register face back of the fixture, and connecting with a flue in the wall, is used in place of the regular local vent. In order for a vent flue of either form to be of any value, it must be warmed to insure a proper circulation of air through it. This is done in some cases by placing a gas-jet at the bottom of the flue, in others a steam or hot water pipe is run through a portion of the flue, and in still others the vent is carried up beside a chimney flue, from which it may receive sufficient warmth to assist the circulation to some extent.

Main or Soil Pipe Vent. It is customary to vent the main soil pipe by carrying it through the roof of the building, and leaving the end open. This is shown in Fig. 74. On gravel roofs which drain toward the center, the soil pipe is sometimes stopped on a level with the roof, and serves as a rain leader. In other cases the roof water may be led to the soil pipe in the cellar. If the latter method is used, the water should pass through a deep trap before connecting with the drain. These arrangements tend both to flush out the soil pipe and trap and prevent the accumulation of sediment.

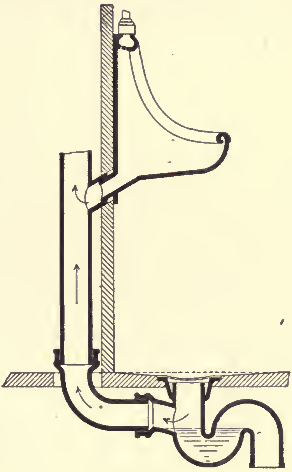


Fig. 73.

Fresh Air Inlets. The fresh air inlet shown just above the running trap Fig. 74 is to cause a circulation of air through the soil pipe, as shown by the arrows. The connection should be made just inside of the trap, so that the entire length of the drain will be swept by the current of fresh air. It is sometimes advised to extend the fresh air pipe up to the roof, because foul air may at times be driven out by heavy flushing of the drain pipe, but where this is done there is much less chance for circulation, as the inlet and outlet are nearly on a level, and the columns of air in them are more likely to be balanced. By carrying the inlet six or eight feet above the ground both objections are overcome to some extent, unless this brings it near a window, which, of course, would not be safe. The main trap does not require a back vent, for should it be siphoned under ordinary conditions, it will always be filled

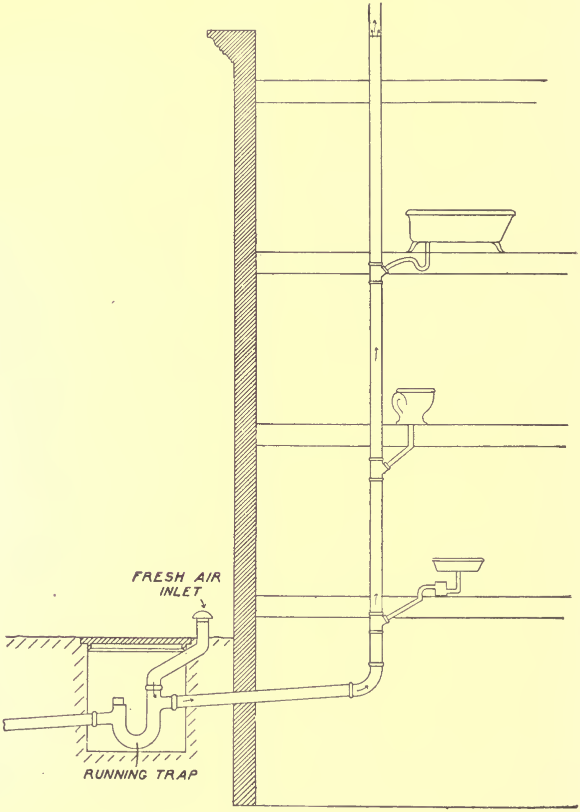
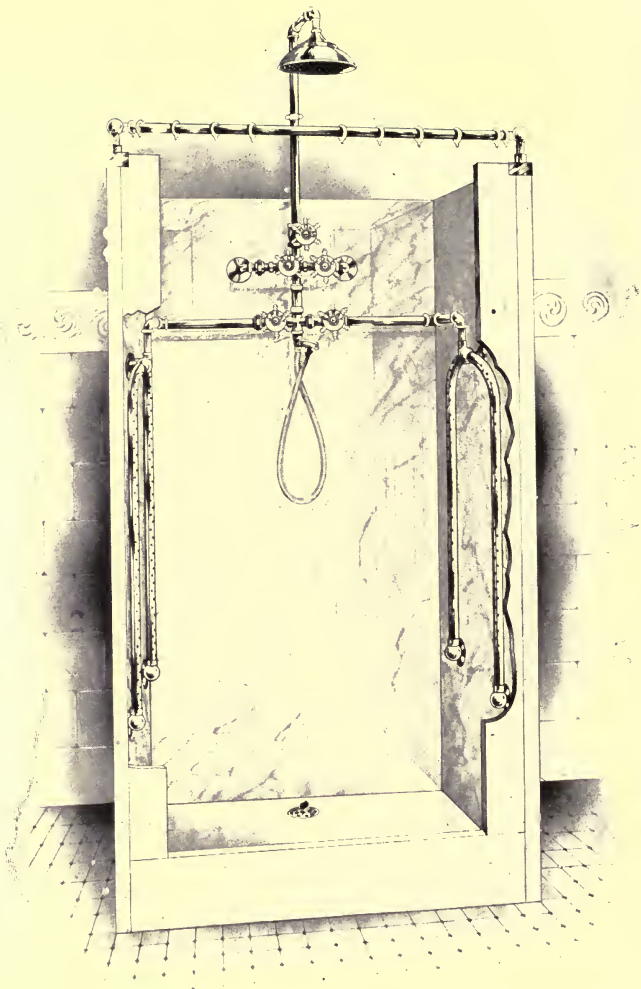


Fig. 74.

again within a few minutes; and if the main soil pipe is open at the top and all fixtures are properly tapped, no harm would come from the slight leakage of gas into the drain under these condi-



NICKEL PLATED BRASS SHOWER BATH.
The Federal Company.

tions, and some engineers recommend the omission of the running trap.

Where a house drains into a cesspool instead of a sewer, it is far more necessary that the system should be trapped against it as it gives off a constant stream of the foulest gases. The usual form of running trap serves to protect the house, but the cesspool should have an independent vent pipe leading to some unobjectionable point and carried well up above the surface of the ground.

Disposal of Sewage. In cities and towns having a system of sewers, or where there is a large stream of running water near by, the matter is a simple one. In the first case, the house drain is merely extended to the sewer, into which it should discharge at as high a point as possible, and at an acute angle with the direction of flow. When the drain connects with a stream it should be carried out some distance from the shore and discharge under water, an opening for ventilation being provided at the bank. Where there are neither sewers nor streams, the cesspool must be used. When the soil is sufficiently porous the method shown in Fig. 63 may be employed. Sometimes the sewage is collected in a closed cistern and discharged periodically through a flush tank into a series of small tiles laid to a gentle grade, from 8 to 12 inches below the surface. By extending these tiles over a sufficient area and allowing from 40 to 70 feet of tile for each person, a complete absorption of the sewage takes place by the action of the atmosphere and the roots.

PIPE CONNECTIONS.

The Bath Room. There are different methods of connecting up the fixtures in a bath room, depending upon the general arrangement, type, the kind of trap used, etc. Fig. 75 shows a set of fixtures connected up with vented traps. Both the soil and vent pipes are carried above the roof with open ends. No trap or fixture should be vented into a chimney, as is quite commonly done; this may work satisfactorily when the flues are warm, but in summer time, when the fires are out, there are quite likely to be down drafts, which cause the gases to be carried into the rooms through stoves or fireplaces. The vent pipe, although usually carried through the roof independently, is sometimes connected with the soil pipe above the highest fixture; the soil pipe is often made a larger size through the attic space and above the roof in order to increase the upward flow of air through it. Fig. 76 shows a set of bath room connections in which non-siphoning traps are used without back venting; this is a simpler and less expensive method of making the connections and is especially recommended by some engineers. Its efficiency of course depends upon the proper working of the traps.

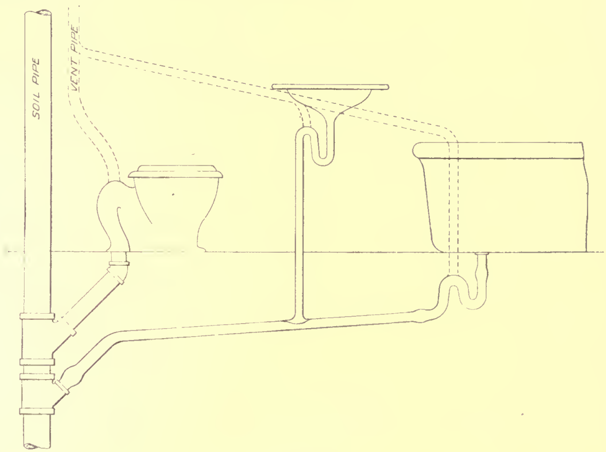


Fig. 75.

The bath room itself should be well lighted, and if possible, in a location where it will receive the sun. It should be arranged so that it may be heated to a higher temperature than other rooms in the house if desired, and it should also be thoroughly ventilated, the vent register being placed 5 or 6 feet above the floor in order that it may carry off any steam which rises from the bath tub. The walls, doors, etc., should be finished in a way to make them as nearly waterproof as possible; some form of good enamel paint answers well for this purpose. Paper should never be used on the walls, nor carpets on the floors, which should be of hard wood. Where the expense is not a matter of importance, glazed tile may be used for the floor and walls. Means should always be provided for ventilating the bathroom without opening the door into the other rooms, and the greatest care should be taken to keep not only the fixtures, but the room itself, in the most perfect order.

Urinal Connections. The common form of urinal connection

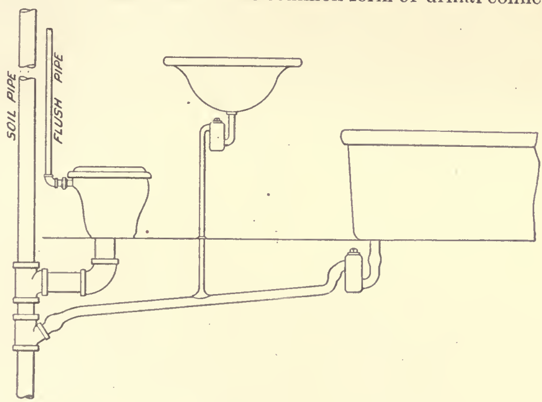


Fig. 76.

is shown in Fig. 14. The overflow from the trap ends in a tee, the lower outlet of which connects with the soil pipe and the upper with the vent pipe. Where several urinals are erected side by side it is usual to omit the individual traps, using the direct outlet connection shown in Fig. 77. These connect with a common waste pipe and drain through a single trap to the soil pipe.

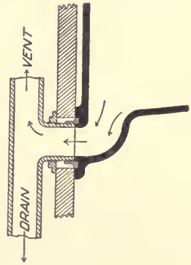


Fig. 77.

Kitchen Sink Connections. Fig. 78 shows the usual method of making the connections for a kitchen sink. The waste and vent are of lead, connected with the main cast-iron soil and vent pipes by means of brass ferules and wiped joints.

Soil and Waste Pipes. The various fixtures have been taken up, together with the different kinds of traps which are used in connection with them, and also the general methods of making the various connections for waste and vent. We will next take up some of the points in regard to the manner of running and supporting the different pipes, together with the proper sizes to be used under different conditions.

The waste pipes of necessity contain more foul matter and therefore more harmful gases than the fixtures, so that especial care must be taken in their arrangement and construction. It is advisable to keep all piping as simple as possible, using as few connections as is consistent with the proper working of the system.

The fixtures on each floor should be arranged to come directly over each other, so as to avoid the running of horizontal pipes across or between the floor beams. The sizes of pipes commonly used require such a sharp grade that there is not sufficient space, in ordinary building construction, between the floor boards and ceiling lath below for horizontal runs of much length. One soil pipe is usually sufficient for buildings of ordinary size, and in cold climates is necessarily carried down inside the building to prevent freezing. One or more waste pipes from sinks, bathtubs, etc., are usually required in addition to the soil pipe These may be connected directly with the soil pipe (through traps), if located near it, or may be carried to the basement vertically and then joined with the main drain pipe inside the running trap. These should also be placed on the inside wall of the house, and, if necessary to conceal them, the boxing used should be put together in such a manner that it may be easily removed for inspection.

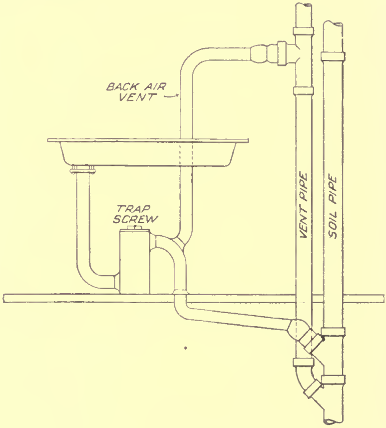


Fig. 78.

The main soil pipe should also be placed where it can be seen, so that leaks may be easily discovered; it is commonly run along the basement wall and supported by suitable brackets or hangers. If carried beneath the cellar floor, it should run in a brick trench with removable covers. In running all lines of pipe, whether vertical or horizontal, they should be securely supported and, in the case of the latter, properly graded. Some of the various kinds of hangers and supports used are shown in Figs. 79 and 80. The grade of the pipes should be as sharp and as uniform as possible. The velocity in the pipes should be at least two feet per second to thoroughly clean them and prevent clogging. Generally speaking, the pitch of the pipes should not be in any case less than 1 foot in 50. In running lines of soil pipe, it is best to

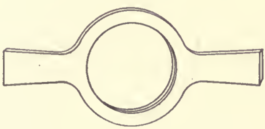


Fig. 79.

Fig. 80.

set the joints ready for calking in the exact positions they are to occupy and resting upon the supports which are intended to hold them permanently. In this way there is less liability of sagging or loosening of the joints after calking. In the running of vertical pipes, care should be taken to have them as straight as possible from the lowest fixture to the roof.

It is very necessary that the pipes be given such an alignment that the water entering them will meet with no serious obstructions. Where vertical pipes join those which are horizontal, they should be given a bend which will turn the stream gradually into the latter, thus preventing any resistance and the resulting accumulation of deposits. Horizontal pipes may be joined with vertical pipes without a bend, as the discharge will be sufficiently free without it. However, it is customary to use a Y or T branch, giving a downward direction to the flow when connecting a closet or other fixture where there is likely to be much solid matter in the sewage. Offsets should always be avoided as far as possible, as they obstruct the flow of both water and air.

Pipe Sizes. The most important requirements in the case of discharge pipes are that they carry away the waste matter as thoroughly as possible without stoppage of flow or eddying, and that they be well ventilated. In order to accomplish this they must be given such sizes as experience has shown to be the best. When water having solid matter in suspension half fills a pipe, the momentum or force for clearing the pipe will be much greater than when it forms only a shallow stream in one of a larger size, so that in proportioning the sizes of soil pipes and drains care must be taken that they are not made larger than necessary, for if the

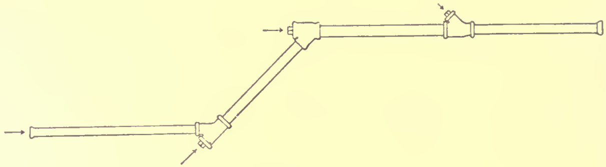


Fig. 81.

stream becomes too shallow the pipes will not be properly flushed and deposits are likely to accumulate. The amount of water used in a house of ordinary size, even when increased by the roof water from a heavy rain, will easily be cared for by a 4-inch pipe having a good pitch. While a pipe of this size would seem to be sufficient, it is found by experience that it is likely to become clogged at times by substances which through carelessness find their way into the drain, so that it seems best to use a somewhat larger size. For city buildings in general, it is recommended that the main drain should not be less than 5 or 6 inches in diameter, and in ordinary dwelling houses not less than 5 inches. The vertical soil pipes need not be larger than 4 inches, except in very high buildings.

Waste pipes may vary from $1\frac{1}{4}$ inches to 2 inches. The waste from a single bowl or lavatory should be $1\frac{1}{4}$ inches in diameter, from a bathtub, kitchen sink or laundry tub $1\frac{1}{2}$ inches, from a slop sink $1\frac{3}{4}$ inches. Smaller pipes should never be used. In laying out the lines of piping, provision should be made for clearing the pipes in case of stoppage. Fig. 81 shows how this may be done. Clean-out plugs are left at the points indicated by the arrows, so that flexible sticks or strips of steel may be inserted to dislodge any obstruction which may occur.

The fresh-air inlet to the main drain pipe has already been referred to. This should be located away from windows, where foul air would be objectionable; in cities they may be placed at the curb line and covered with a grating. Sometimes they are arranged as shown in Fig. 82. The opening is made in the usual way, and a hood placed over the inlet, and a pipe leading from this is carried through the roof. When the circulation of air is upward through the main soil pipe the opening acts in the usual way, that is, as a fresh-air inlet, but should there be a reversal of the current from any reason, which would discharge foul air from the sewer, it would be caught by the overhanging hood and carried upward through the connecting vent pipe to a point above the roof. A general layout for house drainage is shown in Fig. 83.

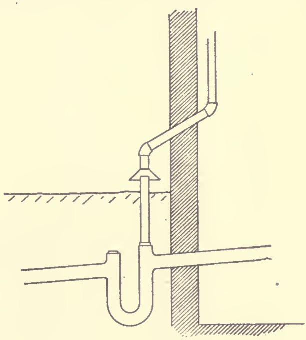


Fig. 82.

PLUMBING FOR VARIOUS BUILDINGS.

Dwelling Houses. The bathroom fixtures, laundry tubs and kitchen sink, with the possible addition of a slop sink, make up the usual fixtures to be provided for in the ordinary dwelling house. In houses of larger size these may be duplicated to some extent, but the general methods of connection are the same as have already been described and need not be taken up again in detail.

Apartment Houses. These are usually made up of duplicate flats, one above the other, so that the plumbing fixtures may be the same for each. It is customary to place the bathrooms in the same position on each floor, so that a single soil pipe will care for all.

Hotels. Here, as in the case just described, the bathrooms are placed one above another, so that a single soil pipe may care for each series, and the problem then becomes that of duplicating the layout for an apartment house. In addition to the private baths there is a public lavatory or toilet-room, usually on the first floor or in the basement. This is fitted up with closets,

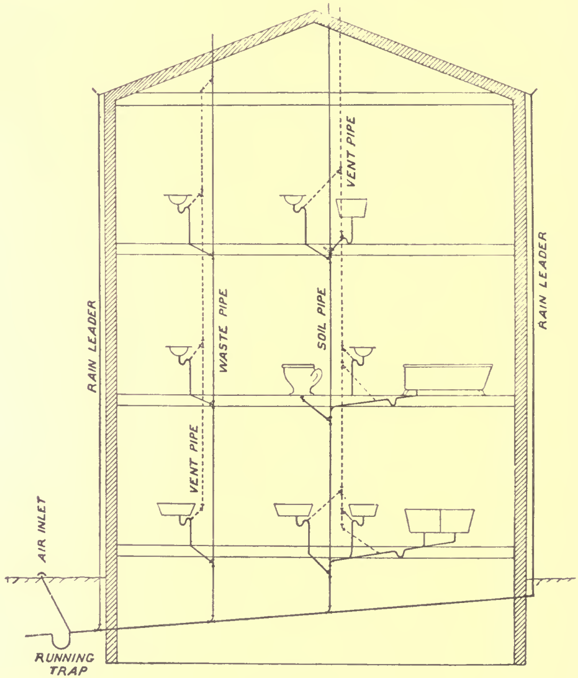


Fig. 83.

urinals and bowls. The closet seats and urinals are placed side by side, with dividing partitions, and connect with a common soil pipe running back of them and having a good pitch. Each fixture should have its own trap. The flushing of the fixtures is often made automatic, so that pressing down the wooden rim of a closet seat will throw a lever which on being released will flush the closet. Urinals are commonly made to flush at regular intervals by some of the devices already shown. The lavatories are made up in long rows, as shown in Fig. 84.

Railroad Stations. The plumbing of a railroad station is similar to that of a hotel, although even greater care should be taken to make the fixtures self-cleansing, as the patrons are likely to include many of the lowest and most ignorant class of people. Special attention should be given to both the local ventilation of the fixtures and the general ventilation of the room.

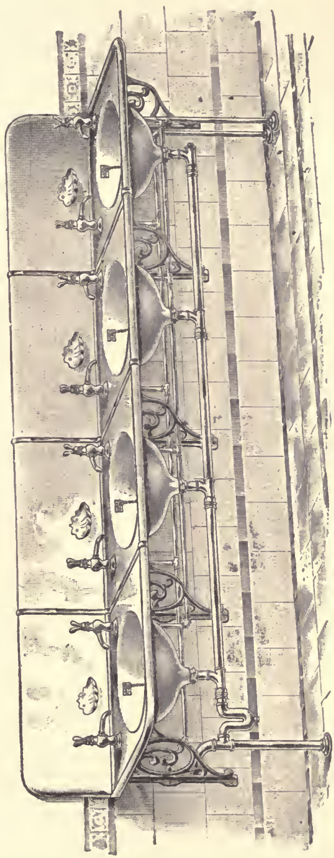


Fig. 84.

Schoolhouses. The same general rules hold in the case of school buildings as in hotels and railroad stations. As the pupils are under the direct supervision of teachers and janitors it is not necessary to have the fixtures automatic to as great an extent as in the cases just described, and it is customary to flush the closets by means of tanks, and pull chains or rods, the same as in private dwellings. The urinals may be automatic or a small stream of water may be allowed to flow through them continuously during school hours. A good form for this class of work is shown in Fig. 85.

Shops and Factories. Some simple type of fixture which can be easily cared for is best in buildings of this kind.

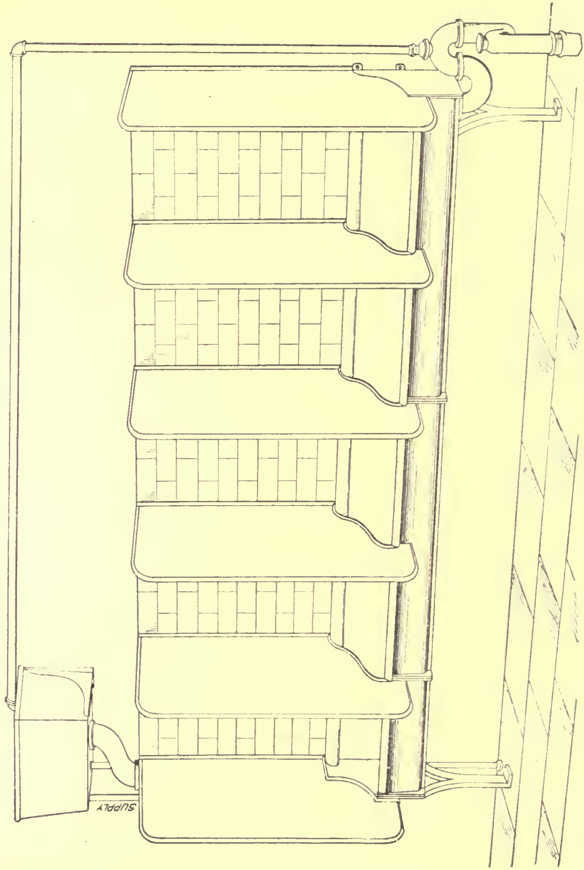


Fig. 85.

TESTING AND INSPECTION.

All plumbing work of any importance should be given two tests; the first, called the "roughing test," applies only to the soil, waste and vent pipes, and is made before the fixtures are connected. The best method of making this test is to plug the main drain pipe just outside the running trap, and also all openings for the connections of fixtures, etc., and then fill the entire system with water. This may be done in small systems through the main vent pipe on the roof, and in larger ones by making a temporary connection with the water main. If any leaks are present they are easily detected in this way. In cold weather, when there would be danger of freezing, compressed air under a pressure of at least ten pounds per square inch may be used in place of water. Leaks in this case must be located by the sound of the issuing air. The water test is to be preferred in all cases, as it is easier to make, and small leaks are more easily detected.

The final test is made after the fixtures are in and all work is completed. There are two ways of making this test, one known as the "peppermint test," and the other as the "smoke test." In making either of these, the system should first be flushed with water, so that all traps may be sealed. If peppermint is used, 4 to 6-ounces of oil of peppermint, depending upon the size of the system, are poured down the main vent pipe, and then a quart or two of hot water to vaporize the oil. The vent pipe is then closed, and the inspector must carefully follow along the lines of piping and locate any leaks present by the odor of the escaping gas. Another and better way is to close the vent pipe and vaporize the oil in the receiver of a small air pump, and then force the gas into the system under a slight pressure. The receiver is provided with a delicate gage, so that after reaching a certain pressure (which must not be great enough to break the trap seals) the pump may be stopped and the pressure noted. If, after a short time, the pressure remains the same, it is known that the system is tight; if, however, the pressure drops, then leaks are present and must be located, as already described. Ether is sometimes used in place of peppermint for this purpose.

In making the smoke test the system is sealed, and the vent pipes closed in the same manner as for the test just described; smoke from oily waste or some similar substance is then forced into the pipes by means of a bellows. When the system is filled with smoke, and a slight pressure produced, the fact is shown by a float, which rises and remains in this position if the joints are tight. If there are leaks, the float falls as soon as the bellows are stopped. Leaks may be detected in this way, both by the odor of the smoke and by the issuing jets from leaks of any size. Special machines are made for both the peppermint and smoke tests.

The water test is preferable for roughing in, and the smoke test for the final. Every system of plumbing should be tested at least once a year.

SEWERAGE AND SEWAGE PURIFICATION.

An abundant supply of pure water is a necessity in every town and city; and such a supply having been secured brings up the question of its disposal after being used. This is plainly the reverse of its introduction. As it was distributed through a network of conduits, diminishing in size, with its numerous branches, so it may be collected again by similar conduits, increasing in size, as one after another they unite in a common outlet.

This fouled water is called *sewage*, and the conduits which collect it constitute a sewerage system. In general, sewage is disposed of in two ways; either it must be turned into a body of water so large as to dilute it beyond all possibility of offence, and where it cannot endanger human life by polluting a public water-supply, or it must be purified in some manner.

The conduits which carry water collected from street surfaces during and after rains, or ground water collected from beneath the surface, are called drains. When one set of conduits removes sewage and another carries surface and ground water, it is said that the *separate* system of sewerage is in use. Where one system conveys both sewage and drainage water it is called the *combined* system. Various modifications of these two systems are possible, both for whole cities and for limited areas within the same town or city.

A sanitary sewerage system cannot be installed until a public water-supply has been provided. It is needed as soon as that is accomplished, for while the wells can then be abandoned the volume of waste water is greatly increased by the water-works system. Its foulness is also much increased through the introduction of water-closets. Without sewers and with a public water-supply cesspools must be used, and with these begins a continuous pollution of the soil much more serious than that which commonly results from closets and the surface disposal of slops.

Among the data which should first be obtained in laying out a sewerage system are:

First.—The area to be served, with its topography and the general character of the soil.—A contour map of the whole town or city, showing the location of the various streets, streams, ponds or lakes, and contour lines for each 5 feet or so of change in elevation, is necessary for the best results. The general character of the soil can usually be obtained by observation and inquiry among residents or builders who have dug wells or cellars, or have observed work of this kind which was being done. The kind of soil is important as affecting the cost of trenching and its wetness or dryness, and this, together with a determination of the ground-water level, will be useful in showing the extent of underdraining necessary.

Second.—Whether the separate or combined system of sewerage, or a compromise between the two is to be adopted.—These points will depend almost wholly upon local conditions. The size and cost of combined sewers is much greater than the separate system, since the surface drainage in times of heavy rainfall is many times as great as the flow of sanitary sewage. In older towns and cities it sometimes happens that drains for removing the surface water are already provided, and in this case it is only necessary to put in the sanitary sewers; or again, the latter may be provided, leaving the matter of surface drainage for future consideration.

If the sewage must be purified, the combined system is out of the question, for the expense of treating the full flow in times of maximum rainfall would be enormous. Sometimes more or less limited areas of a town may require the combined system, while the separate system is best adapted to the remainder; and again it may be necessary to take only the roof water into the sewers. As already stated, local conditions and relative costs are the principal factors in deciding between the separate and combined systems.

Third.— Whether subsoil drainage shall be provided.— In most cases this also will depend upon local conditions. It is always an advantage to lower the ground-water level in places where it is sufficiently high to make the ground wet at or near the surface during a large part of the year. In addition to rendering the soil dry around and beneath cellars, the laying of underdrains is of such aid in sewer construction as to warrant their introduction for this purpose alone. This is the case where the trenches are so wet as to render the making and setting of cement joints difficult. The aim in all good sewer work is to reduce the infiltration of ground water into the pipes to the smallest amount; but in very wet soil, tight joints can be made only with difficulty, and never with absolute certainty. Cases have been known where fully one-half the total volume of sewage discharged consisted of ground water which had worked in through the joints.

Fourth.— The best means for the final disposal of the sewage.— Until recently it was turned into the nearest river or lake where it could be discharged with the least expense. The principal point to be observed in the disposal of sewage is that no public water-supply shall be endangered. At the present time no definite knowledge is at hand regarding the exact length of time that disease germs from the human system will live in water. The Massachusetts legislature at one time said that no sewer should discharge into a stream within 20 miles of any point where it is used for public water-supply, but it is now left largely in the hands of the State Board of Health. There may be cases where sewage disposal seems to claim preference to water supply in the use of a stream, but each case must be decided on its own merits. Knowing the amount of water and the probable quantity and character of the sewage, it is generally easy to determine whether all of the crude sewage of a city can safely be discharged into the body of water in question. Averages in this case should never be used; the water available during a hot and dry summer, when the stream or lake is at its lowest, and the banks and beds are exposed to the sun, is what must be considered. Where sewage is discharged into large bodies of water, either lakes or the ocean, it is generally necessary to make a careful study of the prevailing currents in order to determine the most available point of discharge, in order to prevent the sewage becoming stagnant in bays, or the washing ashore of the lighter portions. Such studies are commonly made with floats, which indicate the direction of the existing currents.

Fifth.— Population, water consumption and volume of sewage for which provision should be made, together with the rainfall data, if surface drainage is to be installed.—The basis for population studies is best taken from the census reports, extending back many years. By means of these the probable growth may be estimated for a period of from 30 to 50 years. In small and rapidly growing towns it must be remembered that the rate of increase is generally less as the population becomes greater.

It is desirable to design a sewerage system large enough to serve for a number of years, 20 or 30 perhaps, although some parts of the work, such as pumping or purification works, may be made smaller and increased in size as needed.

The pipe system should be large enough at the start to serve each street and district for a long period, as the advantages to be derived from the use of city sewers are so great that all houses are almost certain to be connected with them sooner or later. It is often necessary to divide a city into districts in making estimates of the probable growth in population. Thus the residential sections occupied by the wealthiest classes will be comprised of a comparatively small population per acre, due to the large size of the lots. The population will grow more dense in the sections occupied by the less wealthy, the well-to-do and finally the tenement sections. In manufacturing districts the amount of sewage will vary somewhat, depending upon the lines of industry carried on.

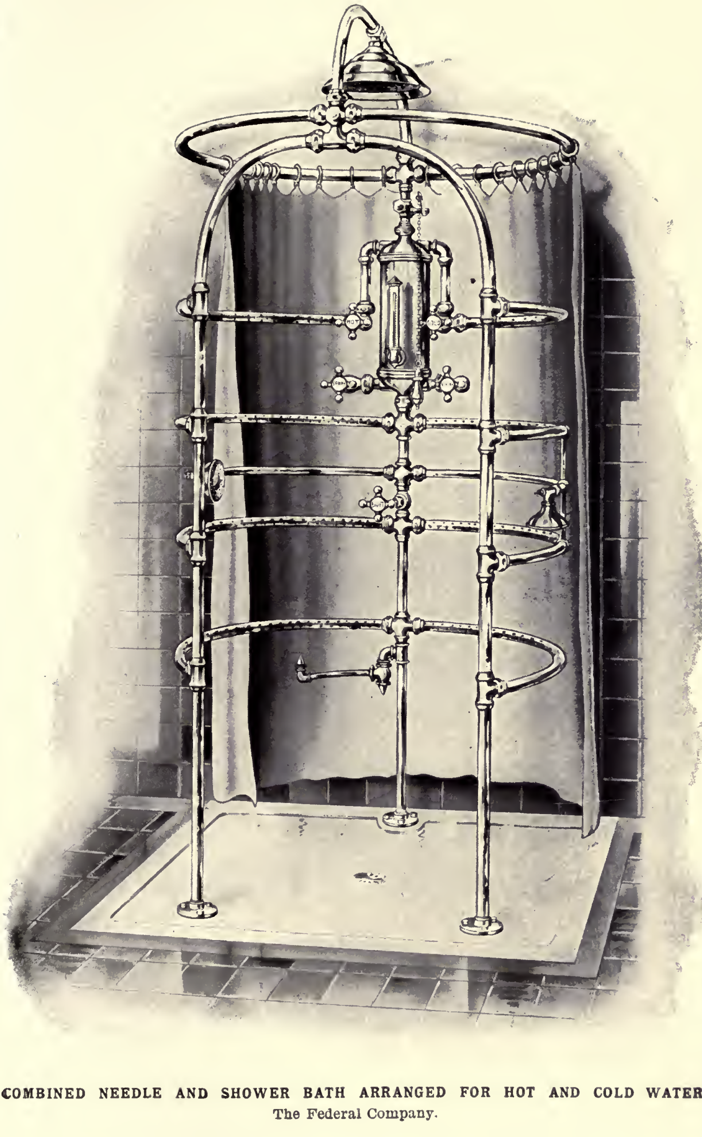
The total water consumption depends mainly upon the population, but no fixed rule can be laid down for determining it beforehand. It is never safe to allow less than 60 gallons per day per capita as the average water consumption of a town if most of the people patronize the public water-supply. In general it is safer to allow 100 gallons. The total daily flow of sewage is not evenly distributed through the 24 hours. The actual amount varies widely during different hours of the day. In most towns there should be little if any sewage, if the pipes are tight enough to prevent inward leakage, between about 10 o'clock in the evening and 4 in the morning. From $\frac{2}{3}$ to $\frac{3}{4}$ of the daily flow usually occurs in from 9 to 12 hours, the particular hours varying in different communities. This is not of importance in designing the pipe system, but only affects the disposal.

Rainfall data is usually hard to obtain except in the cities and larger towns. In cases of this kind the data of neighboring town or cities may be used if available. Monthly or weekly totals are of little value, as it is necessary to provide for the heaviest rains, as a severe shower of 15 minutes may cause more inconvenience and damage, if the sewers are not sufficiently large, than a steady rain extending over a day or two. A maximum rate of 1-inch per hour will usually cover all ordinary conditions. The proportion which will reach the sewers during a given time will depend upon local conditions, such as the slope of land, whether its surface is covered with houses and paved streets, cultivated fields or forests, etc.

Sixth.— Extent and cost of the proposed system.— This is a matter largely dependent upon the local treasury, or the willingness of the people to pay general taxes or a special assessment for the benefits to be derived.

DESIGN AND CONSTRUCTION.

The first step is to lay out the pipe or conduit system. For this the topographical map already mentioned will be found useful. This, however, should be supplemented by a profile of all the streets in which sewers are to be laid, in order to determine the proper grades. In laying out the pipe lines, special diagrams and tables which have been prepared for this purpose may be used. In the separate system it is generally best to use 8″ pipe as the smallest size to lessen the risk of stoppage, although 6″ pipe is ample for the volume of sanitary sewage from an ordinary residence street of medium length. Pipe sewers are generally made of vitrified clay, with a salt-glazed surface. Cement pipe is also used in some cities. The size of pipe sewers is limited to 30 inches in diameter, owing to the difficulty and expense of making the larger pipe and the comparative ease of laying brick sewers of any size from 24 or 30 inches up. In very wet ground,



COMBINED NEEDLE AND SHOWER BATH ARRANGED FOR HOT AND COLD WATER.
The Federal Company.

cast iron pipe with lead joints is used, either to prevent inward leakage or settling of the pipe.

The pipes should be laid to grade with great care and a good alignment should be secured. Holes should be dug for the bells of the pipe, so that they will have solid bearings their entire length. If rock is encountered in trenching, it will be necessary to provide a bed for the pipe which will not be washed into fissures by the stream of subsoil water which is likely to follow the sewer when the ground is saturated.

Underdrains. Where sewers are in wet sand or gravel, underdrains may be laid beneath or alongside the sewer. These are usually the ordinary agricultural tiles, from 3 inches in diameter upward. They have no joints, being simply hollow cylinders, and are laid with their ends a fraction of an inch apart, wrapped with a cheap muslin cloth to keep out the dirt until the matter in the trench becomes thoroughly packed about them. These drains may empty into the nearest stream, provided it is not used for a public water-supply.

Manholes. These should be placed at all changes of grade and at all junctions between streets. They are built of brick and afford access to the sewer for inspection; in addition to this they are sometimes used for flushing. They are provided with iron covers which are often pierced with holes for ventilation.

Sewer Grades. The grades of sewers should be sufficient where possible, to give them a self-clearing velocity. Practical experiments show that sewers of the usual sections will remain clear with the following minimum grades: Separate house connections, 2 per cent; (2-feet fall in each 100 feet of length) small street sewers, 1 per cent; main sewers, 0.7 per cent. These grades may be reduced slightly for sewers carrying only rain or quite pure water.

The following formula may be used for computing the minimum grade for a sewer of clear diameter equal to "d" inches and either circular or oval in section.

$$\text{Minimum grade, in per cent} = \frac{100}{5d+50}.$$

Flushing Devices. Where very low grades are unavoidable and at the head of branch sewers, where the volume of flow is small, flushing may be used with advantage.

In some cases water is turned into the sewer through a manhole, from some pond or stream, or from the public water-works system. Generally, however, the water is allowed to accumulate before being discharged, by closing up the lower side of the manhole until the water partially fills it, then suddenly releasing it and allowing the water to rush through the pipe. Instead of using clear water from outside for this purpose, it may be sufficient at some points on the system to simply back up the sewage, by closing the manhole outlet, thus flushing the sewer with the sewage itself. Where frequent and regular flushing is required, automatic devices are often used. These usually operate by means of a self-discharging siphon, although there are other devices operated by means of the weight of a tank which fills and empties itself at regular intervals.

House Connections. Provision for house connections should be made when the sewers are laid, in order to avoid breaking up the streets after the sewers are in use. Y branches should be put in at frequent intervals, say from 25 feet apart upwards, according to the character of the street. When the sewer main is deep down, quarter bends are sometimes provided, and the house connection pipe carried vertically upwards to within a few feet of the surface to avoid deep digging when connections are made. Where house connections are made with the main, or where two sewers join, the direction of flow should be as nearly in the same direction as possible, and the entering sewer should be at a little higher level in order to increase the velocity of the inflowing sewage.

Depth of Sewers Below the Surface. No general rule can be followed in this matter except to place them low enough to secure a proper grade for the house connections, which are to be made with them. They must be kept below a point where there would be trouble from freezing, but the natural depth is usually sufficient to prevent this in most cases.

Ventilation of Sewers. There is more or less difference in opinion in regard to the proper method of ventilating sewer mains. Ventilation through house soil pipes is generally approved where the sewers and house connections are properly constructed and operated, and where the houses on a given street are of a uniform height, so that the tops of all the soil pipes will be above the highest windows. Where the houses are uneven in height, or where the sewerage system or connections are not well designed or constructed, it is recommended that main traps should be placed on all soil pipes, and that air inlets and air outlets be placed on the sewers at intervals of from 300 to 400 feet.

The Combined System. The principal differences between this and the separate system are in the greater size of conduits and the use of catch-basins or inlets for the admission of surface water. They are generally of brick, stone or concrete, or a combination of these materials, instead of vitrified pipe.

Another difference is the provision for storm overflows, by means of which the main sewers when overcharged in times of heavy rainfall may empty a part of their contents into a nearby stream. At such times the sewage is diluted by the rain-water, while the stream which receives the overflow is also of unusually large size.

Size, Shape and Material. The actual size of the sewer, and also to a large extent its shape and the material of which it is constructed, depends upon local conditions. Where the depth of flow varies greatly it is desirable to give the sewer a cross-section designed to suit all flows as fully as possible.

The best form to meet these requirements is that of an egg with its smaller end placed downward. With this form the greatest depth and velocity of flow is secured for the smallest amount of sewage, thus reducing the tendency to deposits and stoppages. Where sewers have a flow more nearly constant and equal to their full capacity the form may be changed more nearly to that of an ellipse. For the larger sewers brick is the most common material, both because of its low cost and the ease with which any form of conduit is constructed. Stone is sometimes used on steep grades, especially where there is much sand in suspension, which would tend to wear away the brick walls. Concrete is used where leakage may be expected or where the material is liable to movement, but is more commonly used as a foundation for brick construction.

A catch-basin is generally placed at each street corner and provided with a grated opening for giving the surface water access to a chamber or basin beneath the sidewalk, from which a pipe leads to the sewer. Catch-basins may be provided with water traps to prevent the sewer air from reaching the street, but traps are uncertain in their action, as they are likely to become unsealed through evaporation in dry weather. To prevent the carrying of sand and dirt into the sewers, catch-basins should be provided with silt chambers of considerable depth, with overflow pipes leading to the sewer. The heavy matter which falls to the bottoms of these chambers may be removed by buckets and carted away at proper intervals.

Storm Overflows. The main point to be considered in the construction of storm overflows is to ensure a discharge into another conduit when the water reaches a certain elevation in the main sewer. This may be carried out in different ways, depending upon the available points for overflow.

Pumping Stations. The greater part of the sewerage systems in the United States operate wholly by gravity, but in some cases it is necessary to pump a part or the whole of the sewage of a city to a higher level. The lifts required are usually low, so that high-priced machinery is not required. In general the sewage should be screened before it reaches the pumps.

Where pumping is necessary, receiving or storage chambers are sometimes used to equalize the work required of the pumps, thus making it possible to shut down the plant at night. Such reservoirs should be covered, unless in very isolated localities. The force main or discharge pipe from the pumps is usually short, and is generally of cast iron put together in a manner similar to that used for water-supply systems.

Tidal Chambers. Where sewage is discharged into tide water it is often necessary to provide storage or tidal chambers, so that the sewage may be discharged only at ebb tides. These are constructed similar to other reservoirs, except that they must have ample discharge gates, so that they may be emptied in a short time. They are sometimes made to work automatically by the action of the tide.

SEWAGE PURIFICATION.

Before taking up this subject in detail it is well to consider what sewage is, from a chemical standpoint.

When fresh, it appears at the mouth of an outlet sewer as a milky-looking liquid with some large particles of matter in suspension, such as orange peels, rags, paper and various other articles not easily broken up. It often has a faint, musty odor and in general appearance is similar to the suds-water from a family laundry. Nearly all of the sewage is simply water, the total amount of solid matter not being more than 2 parts in 1,000, of which half may be organic matter. It is this 1 part in 1,000 which should be removed, or so changed in character as to render it harmless.

The two systems of purification in most common use are "chemical precipitation" and the "land treatment." Mechanical straining, sedimentation and chemical precipitation are largely removal processes, while land treatment by the slow process of infiltration, or irrigation, changes the decaying organic matter into stable mineral compounds.

Sedimentation. This is effected by allowing the suspended matter to settle in tanks. The partially clarified liquid is then drawn off leaving the solid matter, called "sludge" at the bottom for later disposal. This system requires a good deal of time and large settling tanks; therefore it is suitable only for small quantities of sewage.

Mechanical Straining. This is accomplished in different ways with varying degrees of success. Wire screens or filters of various materials may be employed. Straining of itself is of little value except as a step to further purification. Beds of coke from 6 to 8 inches in depth are often used with good results.

Chemical Precipitation. Sedimentation alone removes only such suspended matter as will sink by its own weight during the comparatively short time which can be allowed for the process.

By adding certain substances chemical action is set up, which greatly increases the rapidity with which precipitation takes place.

Some of the organic substances are brought together by the formation of new compounds, and as they fall in flaky masses they carry with them other suspended matter.

A great number and variety of chemicals have been employed for this purpose, but those which experience has shown to be most useful are lime, sulphate of alumina and some of the salts of iron.

The best chemical to use in any given case depends upon the character of the sewage and the relative cost in that locality. Lime is cheap, but the large quantity required greatly increases the amount of sludge. Sulphate of alumina is more expensive, but is often used to advantage in connection with lime. Where an acid sewage is to be treated, lime alone should be used.

The chemicals should be added to the sewage and thoroughly mixed before it reaches the settling tank ; this may be effected by the use of projections or baffling plates placed in the conduit leading to the tank. The best results are obtained by means of long, narrow tanks, and they should be operated on the continuous rather than the intermittent plan. The width of the tank should be about one-fourth its length. In the continuous method the sewage is constantly flowing into one part of the tank and discharging from another. In the intermittent system a tank is filled and then the flow is turned into another, allowing the sewage in the first tank to come to rest. In the continuous plan the sewage generally flows through a set of tanks without interruption until one of the compartments needs cleaning. The clear portion is drawn off from the top, the sludge is then removed, and the tank thoroughly disinfected before being put in use again. The satisfactory disposal of the sludge is a somewhat difficult matter. The most common method is to press it into cakes, which greatly reduces its bulk and makes it more easily handled. These are sometimes burned but are more often used for fertilizing purposes. In some cases peat or other absorbent is mixed with the sludge and the whole mass removed in bulk. In other instances it is run out on the surface of coarse gravel beds and reduced by draining and drying. In wet weather little drying takes place and during the cold months the sludge accumulates in considerable quantities. This process also requires considerable manual labor, and in many cases suitable land is not available for the purpose. The required capacity of the settling tanks is the principal item in determining the cost of installing precipitation works.

In the treatment of house sewage provision must be made for about $\frac{1}{12}$ the total daily flow, and in addition to this, allowance must be made for throwing out a portion of the tanks for cleaning and repairs. In general, the tank capacity should not be much less than $\frac{1}{8}$ the total daily flow.

In the combined system it is impossible to provide tanks for the total amount, and the excess due to storm water must discharge into natural water courses or pass by the works without treatment.

Broad Irrigation or Sewage Farming. Where sewage is applied to the surface of the ground upon which crops are raised the process is called "sewage farming." This varies but little from ordinary irrigation where clean water is used instead of sewage. The land employed for this purpose should have a rather light and porous soil, and the crops should be such as require a large amount of moisture. The application of from 5,000 to 10,000 gallons of sewage per day per acre is considered a liberal allowance. On the basis of 100 gallons of sewage per head of population this would mean that one acre would care for a population of from 50 to 100 people.

Sub-Surface Irrigation. This system is employed only upon a small scale and chiefly for private dwellings, public institutions and for small communities where for any reason surface disposal would be objectionable. The sewage is distributed through agricultural drain tiles laid with open joints and placed only a few inches below the surface. Provision should be made for changing the disposal area as often as the soil may require by turning the sewage into sub-divisions of the distributing pipes.

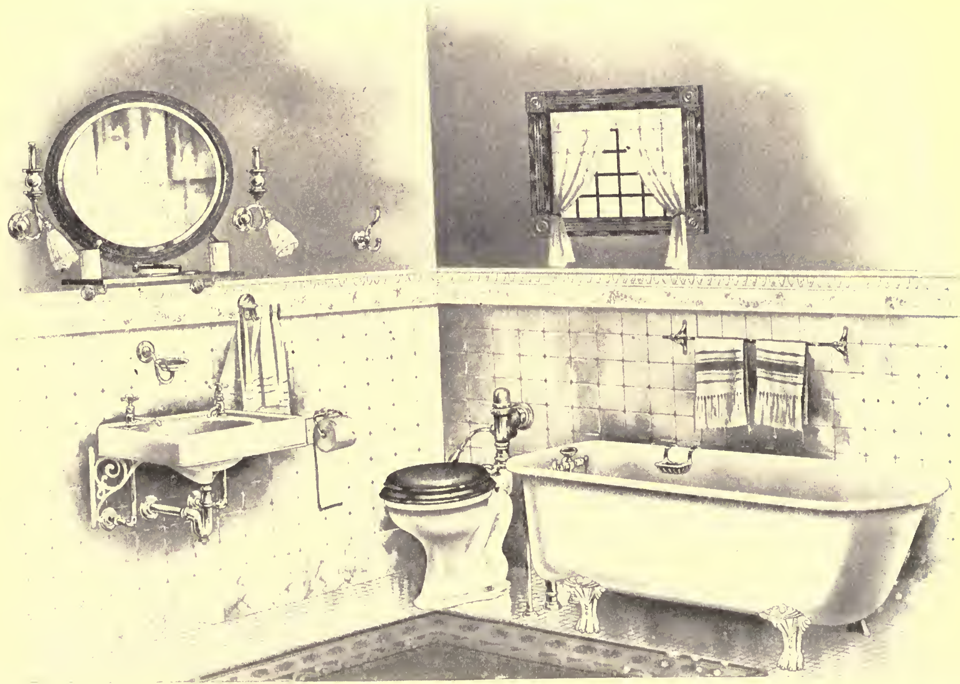
Intermittent Filtration. This method and the broad irrigation already described are the only purification processes in use on a large scale which can remove practically all the organic matter from sewage without being supplemented by some other method. The process is a simple one and consists in running the sewage out through distributing pipes onto beds of sand 4 or 5 feet in thickness with a system of pipes or drains below for collecting the purified liquid. In operation the sewage is first turned on one bed and then another, thus allowing an opportunity for the liquid portion to filter through. As the surface becomes clogged it is raked over or the sludge may be scraped off together with a thin layer of sand. The best filtering material consists of a clean, sharp sand with grains of uniform size such that the free space between them will equal about one-third the total volume. When the sewage is admitted to the sand only a part of the air is driven out, so there is a store of oxygen left upon which the bacteria may draw. This is not a mere process of straining but the formation of new compounds by the action of the oxygen in the air, thus changing the organic matter into inorganic. Much depends upon the size and quality of the sand used. The grains that have been found to give the best results range from .1 to .5 of an inch in diameter. The work done by a filter is largely determined by the finer particles of sand and that used should be of fairly uniform quality, and the coarser and finer particles should be well mixed. The area and volume of sand or gravel required are so large that the transportation of material any great distance cannot be considered. Usually the beds are constructed on natural deposits, the top soil or loam being removed. The sewage should be brought into the beds so as to disturb their surface as little as possible, and should be distributed evenly over the whole bed.

The under drains should not be placed more than 50 feet apart, usually much less, and should be provided with manholes at the junctions of the pipes. Before admitting the sewage to the beds it is usually best to screen it sufficiently to take out paper, rags and other floating matter. The size of each bed should be such as to permit an even distribution of sewage over its surface.

Where the filtration area is small, it must be divided so as to permit of intermittent operation; that is, if a bed is to be in use and at rest for equal periods, then two or more beds would be necessary, the number depending on the relative periods of use and rest. Some additional area should also be provided for emergency, or for use while the beds are being scraped. If a large area is laid out, so that the size of the beds is limited only by convenience in use, then an acre may be taken as a good size.

The degree of purification depends upon various circumstances, but with the best material practically all of the organic matter can be removed from sewage by intermittent filtration at a rate of about 100,000 gallons per day.

There is often much opposition to sewage purification by those living or owning property near the plants; but experience has shown that well-conducted plants are inoffensive both within and without their enclosures. The employees about such works are as healthy as similar classes of men in other occupations. The crops raised on sewage farms are as healthful as those of the same kind raised elsewhere, and meat and milk from sewage farms are usually as good as when produced under other conditions. Good design and construction, followed by proper methods of operation, are all that are needed to make sewage purification a success. No one system can be said to be the best for all localities. The special problems of each case must be met and solved by a selection from among the several systems and the combinations of systems, and parts chosen that are best adapted to the conditions at hand.



REPRESENTATIVE TYPE OF MODERN AMERICAN BATHROOM

PLUMBING.

PART II.

DOMESTIC WATER SUPPLY.

Hydraulics of Plumbing. Although the principles of Hydraulics and Hydrostatics are discussed in "Mechanics," it will be well to review them briefly, showing their application to the various problems under the head of "Water Supply."

If several open vessels containing water are connected by pipes, the water will eventually stand at the same level in all of them, regardless of the length or the size of the connecting pipes.

The pressure exerted by a liquid at any given point is the same in all directions, and is proportional to the depth.

A column of water at 60° temperature having a sectional area of one square inch and a height of one foot, weighs .43 pound, and the pressure exerted by a liquid is usually stated in pounds per square inch, the same as in the case of steam. If a closed vessel is connected, by means of a pipe, with an open vessel at a higher level, so that it is 10 feet, for example, from the bottom of the first vessel to the surface of the water in the second, the pressure on each square inch of the entire bottom of the lower vessel will be $10 \times .43 = 4.3$ pounds, and the pressure per square inch at any given point in the vessel or connecting pipe will be equal to its distance in feet from the surface of the water in the upper vessel multiplied by .43. If a pipe is carried from a reservoir situated on the top of a hill to a point at the foot of the hill a hundred feet below the surface of the water, a pressure of $100 \times .43 = 43$ pounds per square inch will be exerted at the lower end of the pipe, provided it is closed. When the pipe is opened and the water begins to flow, the conditions are changed and the pressure in the different parts of the pipe varies with the distance from the open end.

In order for a liquid to flow through a pipe there must be a certain pressure or "head" at the inlet end. The total head causing the flow is divided into three parts, as follows: 1st, the *velocity* head: the height through which a body must fall in a vacuum to acquire the velocity with which the water enters the pipe. 2d, the *entry* head: that required to overcome the resistance to entrance into the pipe. 3d, the *friction* head: due to the frictional resistance to flow within the pipe. In the case of long pipes and low heads the sum of the velocity and entry heads is so small that it may be neglected.

Table I shows the pressure of water in pounds per square inch for elevations varying in height from 1 to 135 feet.

Table II gives the drop in pressure due to friction in pipes of different diameters for varying rates of flow. The figures given are for pipes 100 feet in height. The frictional resistance in smooth pipes having a constant flow of water through them is proportional to the length of pipe. That is, if the friction causes a drop in pressure of 4.07 pounds per square inch in a $1\frac{1}{4}$-inch pipe 100 feet long, which is discharging 20 gallons per minute, it will cause a drop of $4.07 \times 2 = 8.14$ pounds in a pipe 200 feet long; or $4.07 \div 2 = 2.03$ pounds in a pipe 50 feet long, acting under the same conditions. The factors given in the table are for pipes of smooth interior, like lead, brass or wrought iron.

Example.—A $1\frac{1}{2}$-inch pipe 100 feet long connected with a cistern is to discharge 35 gallons per minute. At what elevation above the end of the pipe must the surface of the water in the cistern be to produce this flow?

In Table II we find the friction loss for a $1\frac{1}{2}$-inch pipe discharging 35 gallons per minute to be 5.05 pounds. In Table I we find a pressure of 5.2 pounds corresponds to a head of 12 feet, which is approximately the elevation required.

How many gallons will be discharged through a 2-inch pipe 100 feet long where the inlet is 22 feet above the outlet? In Table I we find a head of 22 feet corresponds to a pressure of 9.53 pounds. Then looking in Table II we find in the column of Friction Loss for a 2-inch pipe that a pressure of 9.46 corresponds to a discharge of 100 gallons per minute.

Tables I and II are commonly used together in examples.

A house requiring a maximum of 10 gallons of water per minute is to be supplied from a spring which is located 600 feet distant, and at an elevation of 50 feet above the point of dis-

TABLE I.

Head in feet.	Pressure pounds per square inch.	Head in feet.	Pressure pounds per square inch.	Head in feet.	Pressure pounds per square inch.
1	.43	46	19.92	91	39.42
2	.86	47	20.35	92	39.85
3	1.30	48	20.79	93	40.28
4	1.73	49	21.22	94	40.72
5	2.16	50	21.65	95	41.15
6	2.59	51	22.09	96	41.58
7	3.03	52	22.52	97	42.01
8	3.46	53	22.95	98	42.45
9	3.89	54	23.39	99	42.88
10	4 33	55	23.82	100	43.31
11	4.76	56	24.26	101	43.75
12	5.20	57	24.69	102	44.18
13	5.63	58	25.12	103	44.61
14	6.06	59	25.55	104	45.05
15	6.49	60	25.99	105	45.48
16	6.92	61	26.42	106	45.91
17	7.36	62	26.85	107	46.34
18	7.79	63	27.29	108	46.78
19	8.22	64	27.72	109	47.21
20	8.66	65	28.15	110	47.64
21	9.09	66	28.58	111	48.08
22	9.53	67	29.02	112	48.51
23	9.96	68	29.45	113	48.94
24	10.39	69	29.88	114	49.38
25	10.82	70	30.32	115	49.81
26	11.26	71	30.75	116	50.24
27	11.69	72	31.18	117	50.68
28	12.12	73	31.62	118	51.11
29	12.55	74	32.05	119	51.54
30	12.99	75	32.48	120	51.98
31	13.42	76	32.92	121	52.41
32	13.86	77	33.35	122	52.84
33	14.29	78	33.78	123	53.28
34	14.72	79	34.21	124	53.71
35	15.16	80	34.65	125	54.15
36	15.59	81	35.08	126	54.58
37	16.02	82	35.52	127	55.01
38	16.45	83	35.95	128	55.44
39	16.89	84	36.39	129	55.88
40	17.32	85	36.82	130	56.31
41	17.75	86	37.25	131	56.74
42	18.19	87	37.68	132	57.18
43	18.62	88	38.12	133	57.61
44	19.05	89	38.55	134	58.04
45	19.49	90	38.98	135	58.48

charge. What size of pipe will be required? From Table I we find an elevation or head of 50 feet will produce a pressure of 21.65 pounds per square inch. Then if the length of the pipe were only 100 feet, we should have a pressure of 21.65 pounds available to overcome the friction in the pipe, and could follow along the line corresponding to 10 gallons in Table II until we came to the

TABLE II.

Gallons discharged per minute.	½ in.		¾ in.		1 in.		1¼ in.		1½ in.		2 in.		2½ in.		3 in.	
	Velocity in feet per second.	Friction loss in pounds.	Velocity in feet per second.	Friction loss in pounds.	Velocity in feet per second.	Friction loss in pounds.	Velocity in feet per second.	Friction loss in pounds.	Velocity in feet per second.	Friction loss in pounds.	Velocity in feet per second.	Friction loss in pounds.	Velocity in feet per second.	Friction loss in pounds.	Velocity in feet per second.	Friction loss in pounds.
5	8.17	24.6	3.63	3 3	2.04	.84	1.31	.31	.91	.12						
10	16.3	96.0	7.25	13.0	4.08	3.16	2.61	1.05	1.82	.47	1.02	.12				
15			10.9	28.7	6.13	6.98	3.92	2.38	2.73	.97	1.53	.27				
20			14.5	50.4	8.17	12.3	5.22	4.07	3.63	1.66	2.04	.42				
25			18.1	78.0	10.2	19.0	6.53	6.40	4.54	2.62	2.55	.67	1.63	.21	1.13	.10
30					12.3	27.5	7.84	9.15	5.45	3.75	3.06	.91				
35					14.3	37 0	9.14	12 04	6.36	5.05	3.57	1.25				
40					16.3	48.0	10.4	16.10	7.26	6.52	4.09	1.60				
45							11.7	20.2	8.17	8.15	4.60	2.02				
50							13.1	24.9	9.08	10.0	5.11	2.44	3 26	.81	2.27	.35
75							19.6	56.1	13.6	22.4	7.66	5.32	4.90	1.80	3.40	.74
100									18.2	39.0	10.2	9.46	6.53	3.20	4.54	1.31
125											12.8	14.9	8.16	4.89	5.67	1.99
150											15.3	21.2	9 80	7.00	6.81	2.85
175											17.1	28.1	11.4	9.46	7.94	3.85
200											20.4	37.5	13.1	12.47	9.08	5.02

friction loss corresponding most nearly to 21.65, and take the size of pipe corresponding. But as the length of the pipe is 600 feet, the friction loss will be six times that given in Table II for given sizes of pipe and rates of flow; hence we must divide 21.65 by 6 to obtain the available head to overcome friction, and look for this quantity in the table, $21.65 \div 6 = 3.61$, and Table II shows us that a 1-inch pipe will discharge 10 gallons per minute with a friction loss of 3.16 pounds, and this is the size we should use.

EXAMPLES FOR PRACTICE.

1. What size pipe will be required to discharge 40 gallons per minute, a distance of 50 feet, with a pressure head of 19 feet?

Ans. 1¼ inch.

2. What head will be required to discharge 100 gallons per minute through a $2\frac{1}{2}$-inch pipe 700 feet long?

Ans. 52 feet.

PIPING.

Wrought iron, lead and brass are the principal materials used for water pipes. Wrought-iron pipe is the cheapest and easiest to lay, but is objectionable on account of rust and the consequent discoloration of water passing through it. When it is employed for this purpose it is customary to use galvanized pipe, that is, pipe which has been covered with a thin coating of zinc or zinc and tin. This prevents rust from forming where the zinc is unbroken, but at the joints where threads are cut, and at other places where the zinc becomes loosened, as by bending, the pipe is likely to be eaten away more or less rapidly, depending upon the quality of the water. Zinc, when taken into the system, is poisonous, and for this reason galvanized pipes should not ordinarily be used for drinking water.

TABLE III.

Nominal inside diameter.	Actual outside diameter.	Thickness.	Actual inside diameter.	Internal circumference.	External circumference.	Length of pipe per square foot of inside surface.	Length of pipe per square foot of outside surface.	Internal area.	External area.	Length of pipe containing 1 cubic foot.	Weight per foot.	Number of threads per inch of screw.	Gallons per foot of length.
in.	in.	in.	in.	in.	in.	feet	feet	in.	in.	feet	pounds		
												27	.0006
1/8	.40	.068	.27	.85	1.27	14.1	9.44	.05	.13	2500.	.24	18	.0026
1/4	.54	.088	.36	1.14	1.69	10.5	7.05	.10	.23	1385.	.42	18	.0057
3/8	.67	.091	.49	1.55	2.12	7.67	5.65	.19	.36	751.5	.56	14	.0102
1/2	.84	.109	.62	1.95	2.65	6.13	4.50	.30	.55	472.4	.84	14	.0230
3/4	1.05	.113	.82	2.59	3.29	4.63	3.63	.53	.86	270.0	1.12	11½	.0408
1	1.31	.134	1.05	3.29	4.13	3.68	2.90	.86	1.35	166.9	1.67	11½	.0638
1¼	1.66	.140	1.38	4.33	5.21	2.77	2.30	1.49	2.16	96.2	2.26	11½	.0918
1½	1.90	.145	1.61	5.06	5.96	2.37	2.01	2.04	2.83	70.6	2.69	11½	.1632
2	2.37	.154	2.06	6.49	7.46	1.85	1.61	3.35	4.43	42.3	3.66	8	.2550
2½	2 87	.204	2.47	7.75	9.03	1.54	1.33	4.78	6.49	30 1	5.77	8	.3673
3	3.50	.217	3.06	9.63	10.1	1.24	1.09	7.39	9.62	19.5	7.54	8	.4998
3½	4.00	.226	3.55	11.1	12.5	1.07	.95	9 88	12.5	14.5	9.05	8	.6528
4	4.50	.237	4 02	12.6	14.1	.95	.85	12.7	15.9	11.3	10.7	8	.8263
5	5.56	.259	5.04	15.8	17.4	.75	.63	20.0	24.3	7.2	14.5	8	1.469
6	6.62	.280	6.06	19.0	20.8	.63	.57	28.9	34.4	4.9	18.7	8	1.999

Table III gives the various dimensions of wrought-iron pipe. In using pipe of this kind, it is well to allow something in size for possible choking by rust or sediment. While galvanized pipe does not rust, for a time at least, there is likely to be a roughness which causes an accumulation of more or less sediment.

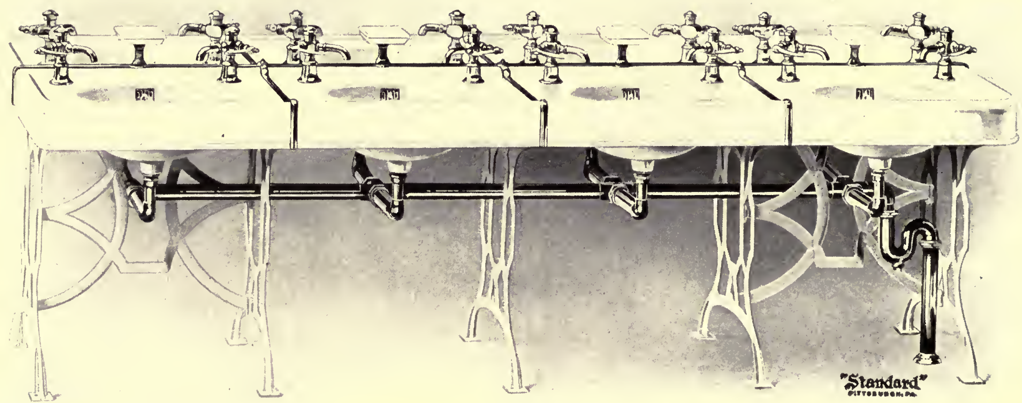
TABLE IV.

Lead Pipe.

Internal diameter.	External diameter.	Thickness.	Weight per foot.	Mean bursting pressure (pounds per square inch).	Safe working pressure.
$\frac{3}{8}$	.75	.18	1 lb. 12 oz.	1968	492
$\frac{3}{8}$	.55	.087	10	1085	271
$\frac{1}{2}$	1.00	.25	3	1787	446
$\frac{1}{2}$	.63	.065	10	625	156
$\frac{5}{8}$	1.10	.23	3 8	1548	387
$\frac{5}{8}$	.84	.10	1 4	708	177
$\frac{3}{4}$	1.33	.29	4 14	1462	365
$\frac{3}{4}$	.93	.09	1 3	505	126
1	1.60	.30	6	1230	307
1	1.18	.09	1 8	325	81
$1\frac{1}{4}$	1.80	.275	6 12	962	240
$1\frac{1}{4}$	1.44	.095	2	322	80
$1\frac{1}{2}$	2.08	.29	8	742	185
$1\frac{1}{2}$	1.74	.12	3	245	61
$1\frac{3}{4}$	2.12	.19	5	460	116
$1\frac{3}{4}$	2.0	.125	3 10	318	79
2	2.60	.30	10 11	611	152
2	2.18	.09	4	200	50

Iron pipe having a lining of tin $\frac{1}{16}$ inch or more in thickness is now manufactured, but being a comparatively new product, its wearing qualities have not yet been thoroughly tested.

Lead Pipe is the best and most widely used for domestic water supply. Although poisonous under certain conditions, as



PORCELAIN ENAMELED PANDORA DESIGN DOUBLE SECTIONAL LAVATORY
Standard Sanitary Mfg. Co.

when new and bright and when used with very pure water, it usually becomes coated with a scale which makes it practically harmless. It is more costly than iron pipe, and requires more skill in laying and making up the joints. It is less likely to burst from the action of frost, as it is a soft metal and stretches with the expansion of the ice in the pipe. When it does break under pressure it generally occurs in small holes not over an inch long, which are easily repaired without removing any part of the pipe, while in the case of iron pipe the cracks generally extend the entire length of the section in which the

TABLE V.

Tin-lined Lead Pipe.

Internal diameter.	A A A Weight per foot.		A A Weight per foot.		A Weight per foot.		B Weight per foot.		C Weight per foot.		D Weight per foot.		D light Weight per foot.		E Weight per foot.		E light Weight per foot.	
	lb.	oz.	lb.	oz.	lb.	oz.	lb.	oz.	lb.	oz.	lb.	oz.	lb.	oz.	lb.	oz.	lb.	oz.
3/8	1	8	1	5	1	2	1	0	0	13	0	10			0	8		
1/2	3	0	2	0	1	12	1	4	1	0	0	13			0	11	0	9
5/8	3	8	2	12	2	8	2	0	1	12	1	8	1	4	1	0	0	12
3/4	4	8	3	8	3	0	2	4	2	0	1	12	1	8	1	4	1	0
1	6	0	4	12	4	0	3	4	2	8	2	0			1	8		
1¼	6	12	5	12	4	12	3	12	3	0	2	8			2	0		
1½	9	0	8	0	6	4	5	0	4	4	3	8			8	4		
2	10	12	9	0	7	0	6	0	5	4	4	0						

water is frozen, and new pipe will be required. Lead pipe is commonly made in six different thicknesses or weights, designated as AAA, AA, A, B, C and D, in which AAA is the heaviest and D the lightest. Table IV gives the principal properties of the heaviest and lightest weight for lead pipe of different diameters.

Tin-lined lead pipe is used to some extent for conveying water for domestic purposes. The principal objection to this pipe lies in the difficulty experienced in making the joints. Tin melts at a considerably lower temperature than lead, so that in making wipe joints it is likely to melt before the lead and block up the passage through the pipe. Another objection is due to the fact that the tin lining and the outer lead covering are simply pressed together, and it often happens that in bending the pipe the lining pulls away from the lead, thus both obstructing and weakening the pipe. When used for hot water, the uneven expansion of the two metals may separate the two layers, and so cause the same difficulties already mentioned.

Table V gives some of the properties of tin-lined lead pipe.

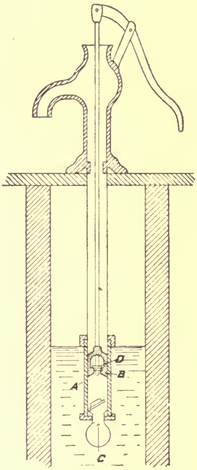


Fig. 1.

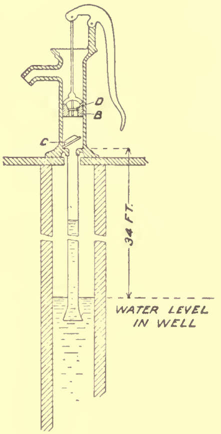


Fig. 2.

The strength of tin-lined pipe is about the same as that of lead pipe, the greater strength of the tin being offset by the lighter weight of the pipe made in this way.

Brass Pipe. Brass is one of the best materials for hot-water pipes, and should be used where the cost is not the controlling feature. It is commonly employed for connecting pumps and boilers and for the steam-heating coils inside laundry-water heaters. It is often used for the connections between the kitchen hot-water tank and range, and when nickel plated is extensively employed in connection with bathroom fixtures. The sizes and thicknesses are approximately the same as wrought-iron pipe.

PUMPS.

The principle upon which the pump operates has already been taken up in the Instruction Paper, "Mechanics." The more common forms are known as the "lift pump," the "suction pump" and a combination of the two called the "deep well pump."

Fig. 1 shows a pump of the first kind. In this pump A is the cylinder, B the plunger, C the bottom valve and D the plunger valve. When the plunger is drawn up, a vacuum is formed in the cylinder, and water flows in through C to fill it. When the plunger is forced down, valve D opens and allows the water to flow through the plunger while C remains closed. As this operation is repeated, the water is raised by the plunger at each stroke until the entire length of the pump barrel is filled, and it will then flow from the spout in an intermittent stream.

In the suction pump shown in Fig. 2, the cylinder and valves are the same, but they are placed at the top of the well and are connected with the water below by means of a pipe, as shown. When the pump is operated, a vacuum is formed in the cylinder and pipe below the plunger, and the pressure of the atmosphere upon the surface of the water forces it up the pipe and fills the chamber, after which the action becomes the same as in the case of a lift pump. The pressure of the atmosphere is approximately 15 pounds per square inch, which corresponds to the weight of a column of water 34 feet high, which is the height that the water may be raised theoretically by suction.

When the surface of the water is a greater distance than this below the point of discharge, a pump similar to that shown in Fig. 3 must be used. A is a cylinder with plunger and valves similar to those of a suction pump. The cylinder is supported in the well at some point less than 34 feet above the surface of the water; E is an air chamber connecting with the upper part of the pump cylinder, and F a discharge pipe leading from the bottom of the air chamber E. The action is as follows: water is pumped into the bottom of the air chamber, and as it rises and seals the end of the discharge pipe, the air in the upper part of the chamber is compressed, and as soon as sufficient pressure is obtained the water is forced out through the discharge pipe F. The pressure required in the air chamber depends upon the height to which the water is raised.

The Hydraulic Ram. This is a device for automatically raising water from a lower to a higher level, the only requirements within certain limits being that the ram shall be placed at a given distance from the spring or source of supply and at a lower level, depending upon the height to which the water is to be raised and the length of the pipe through which it is to be forced. The distance from the source or spring to the ram should be at least from 25 to 50 feet, in order to secure the required velocity for proper operation. A difference in level of 2 feet, or even less, is sufficient to operate the ram; but the greater the difference, the more powerful is its operation. For ordinary purposes, where the water is to be conveyed from 50 to 60 rods, about $\frac{1}{10}$ to $\frac{1}{14}$ of the total amount used can be raised and discharged at an elevation ten times as great as the fall from the spring to the ram.

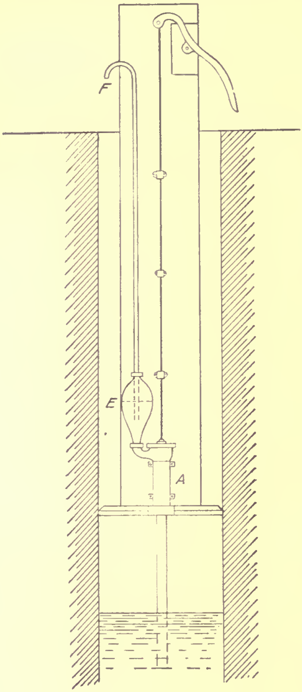


Fig. 3.

In Fig. 4, A represents the source or spring, B the supply pipe, C a valve opening upward, D an air chamber, E a valve closing when raised, and F the discharge pipe. When the water in the pipe is at rest, the valve E drops by its own weight and allows the water to flow through it. As soon as a sufficient velocity is reached by the water, its momentum or force raises the valve against its seat and closes it. The water being thus suddenly arrested in its passage flows into the chamber D, where its sudden influx compresses the air in the top of the chamber, and this in turn forces the water upward through the discharge pipe F. As soon as the water in the pipe B becomes quiet, the valve E again opens and the operation is repeated. Bends in either the drive or discharge pipe should be avoided if possible. If elbows are necessary, the extra long turn pattern should be used in order to give as little resistance as possible. These machines are made of iron and

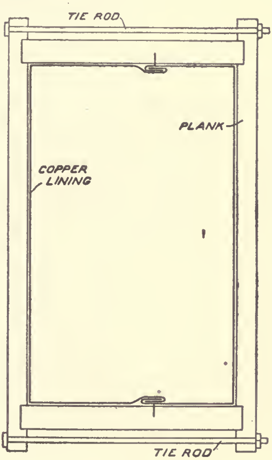


Fig. 5.

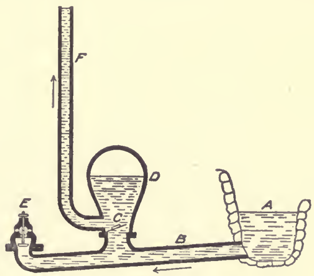


Fig. 4.

brass. The valve and stem are of bronze, on account of its wearing qualities.

Cisterns and Tanks. Water cisterns and tanks are made of various materials and in different shapes and sizes, according to the special uses for which they are required. A durable and satisfactory tank may be made of heavy woodwork or plank bolted together with iron rods and nuts and then lined with some sheet metal, such as copper, lead or zinc. Copper or lead makes the best lining, as the zinc has a greater tendency to corrode and become leaky. If copper is used, it should be tinned on the outside. Fig. 5 shows a wooden tank in plan, with the method of locking the joints in the copper lining. All nails should be so placed as to be covered by the copper, and the joints soldered with the best quality of solder, which should be allowed to soak into the seams. If the tank is lined with lead, a good weight should be used (about six pounds per square foot) and the joints carefully wiped by an experienced workman. If used for the storage of drinking water, this form of lining is open to the same objections as lead pipe, but if kept filled at all times, and especially if the water contains mineral matter to any extent, there is very little danger, as a coating is soon formed over the surface of the lead, protecting it from the action of the water.

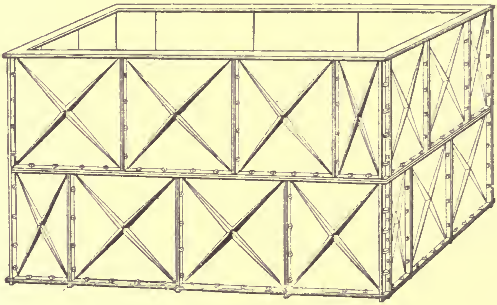


Fig. 6.

Cast-iron sectional tanks can be had in almost any size or shape. A tank of this form is shown in Fig. 6. It is made up of plates which are planed and bolted together, the joints being made tight with cement. The sections are made in convenient sizes, so that they may be handled easily and conveyed without difficulty through small openings to any part of the house. These tanks are easily set up, and are practically indestructible. Wrought-iron tanks are often used, but are not as easily handled as either of the kinds just described. Table VI will be found useful in computing the size of cylindrical tanks.

COLD-WATER SUPPLY.

Systems. There are two general methods of supplying a building with water, one known as the "direct supply" system, and the other as the "indirect" or "tank" system.

In the direct system each fixture is connected with the supply pipe and is under the same pressure as the street main, unless a reducing valve is introduced. This system is not always desirable, as the street pressure in many places is likely to vary, especially where the water is pumped into the mains. A variable pressure is injurious to the fixtures, causing them to leak much sooner than if subjected to a steady pressure. Where the pressure in the street main exceeds 40 pounds per square inch, a reducing valve should be used if the direct system is to be employed.

TABLE VI.

Capacity of Cisterns, in Gallons, for each 10 inches in Depth.

Diameter in feet.	Gallons.	Diameter in feet.	Gallons.	Diameter in feet.	Gallons.
2.0	19.5	6.0	176.3	10	489.6
2.5	30.5	6.5	206.8	11	592.4
3.0	44.6	7.0	239.9	12	705.0
3.5	60.0	7.5	275.4	13	827.4
4.0	78.3	8.0	313.3	14	959.6
4.5	99.1	8.5	353.7	15	1101.6
5.0	122.4	9.0	396.5	20	1958.4
5.5	148.1	9.5	461.4	25	3059.4

The following factors for changing a given quantity of water from one denomination to another will often be found useful :

Cubic feet $\times$ $62\frac{1}{2}$ = Pounds
Pounds $\div$ $62\frac{1}{2}$ = Cubic feet
Gallons $\times$ 8.3 = Pounds
Pounds $\div$ 8.3 = Gallons
Cubic feet $\times$ 7.48 = Gallons
Gallons $\div$ 7.48 = Cubic feet

For domestic purposes the indirect system is much better. In this case the connection with the street main is carried directly to a tank placed in the attic or at some point above the highest fixture, and all the water used in the house discharged into it. The supply of water is regulated by a ball-cock in the tank which shuts it off when a certain level is reached. All the plumbing fixtures are supplied from the tank, and are therefore under a constant pressure. This pressure depends upon the distance of the fixture below the tank. The pipes and fixtures in a house supplied with the tank system will last much longer and

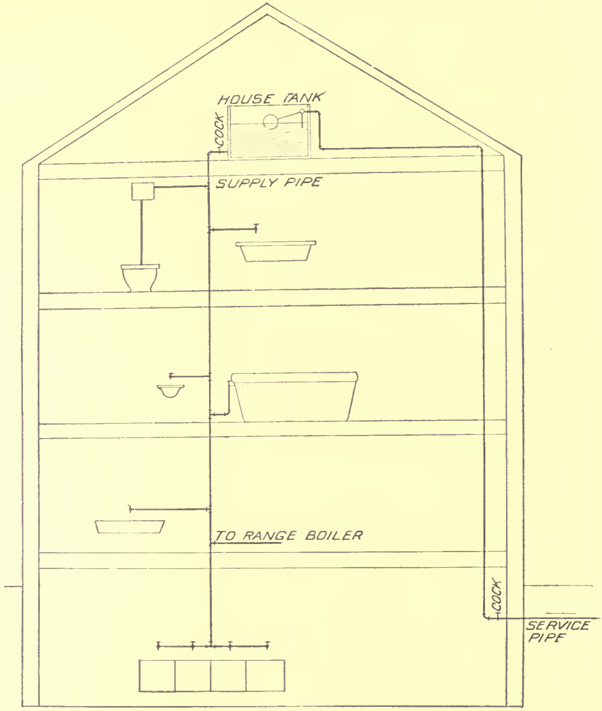


Fig. 7.

give much better results than if connected directly with the street main. The tank is also found useful for storage purposes in case of repairs to the street mains, which is often a matter of much inconvenience.

Fig. 7 shows the general arrangement of the cold-water pipes of an indirect supply system. On the right is shown the service pipe, which is carried directly from the street to the attic, and then connected with a ball-cock located inside the house tank. A supply pipe is taken from the bottom of the tank and carried downward through the building for supplying the various fixtures. A stopcock should be placed in the supply pipe for closing off the tank connections in case of repairs to the house-piping or fixtures.

Tank Overflow Pipe. In order to prevent any possibility of overflow, every house tank should be supplied with an overflow pipe of sufficient size to carry off easily the greatest quantity of water that may be discharged into it. The overflow from a house tank should never be connected directly with a sewer or soil pipe, even if provided with traps, for the water may seldom flow

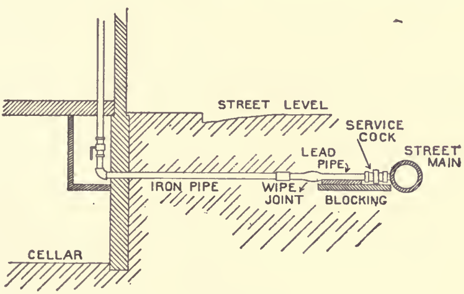


Fig. 8.

through this pipe, thus allowing the trap to become unsealed through evaporation. It is much better to let the end of the overflow pipe be open to the atmosphere or drop over some fixture which is in constant use.

Service Pipe Connections. Fig. 8 shows the usual method of connecting the service pipe with the street main. The service cock is connected directly with the main, and should be carefully blocked, so that any pressure of earth from above will not break the connection or strain the cock. To do this properly, the earth under the pipe should be rammed down solid after the connections are made, and the pipe at this point should be supported on sound wooden blocks. If galvanized iron is used for the service pipe, it should in all cases be connected to the main service cock with a short piece of lead pipe two or three feet long, for the reason that lead will give or sag with the pressure of the earth without breaking. The remainder of the pipe should be carefully embedded in the earth, to prevent uneven strains at any particular point. Connections between the lead and iron pipes should be made by means of brass ferrules and wiped joints. A stopcock should be placed in the service pipe just inside the cellar wall, and in a position where it will be accessible in case of accident. A drip should be connected with the stopcock for draining the pipes when water is shut off.

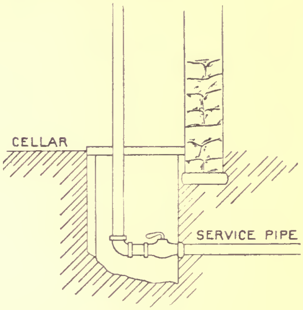


Fig. 9.

In protecting pipes against freezing it is well to pack them in hair, felt, granulated cork or dry shavings where they pass through the floor. This is shown in Fig. 8. When the service pipe comes in below the cellar floor, it may be arranged as shown in Fig. 9. The cock should be placed about 18 inches below the cellar bottom in a wooden box with hinged cover, so that it may be easily reached.

In many cities and in certain elevated situations the pressure in the mains is not sufficient to carry the water to the house tanks in the attics of the higher buildings, and it becomes necessary to use some form of automatic pump for this purpose. The screw pump shown in Fig. 10 is especially adapted to uses of this kind when equipped with an electric motor and automatic starting and stopping devices. A float in the tank operates an electric switch by means of a chain and weights, as shown. A centrifugal or rotary pump is also satisfactory for this work.

Another device which may be attached to a steam pump is shown in Fig. 11. When the water line in the tank reaches a given height, the float closes a butterfly valve in the discharge pipe, thus increasing the pressure within it; this in pressure acts on the bottom of a piston by means of a connecting pipe, and in raising the piston, shuts off the steam supply to the pump. When the water line in the tank is lowered, the float falls and the butterfly valve opens, relieving the pressure in the pipe and allowing the steam valve to open by the action of the counterweights attached to the lever arm of the valve, as shown. The automatic valve is shown in section in Fig. 12. Another means of raising water to an elevation for domestic purposes, especially in the country, is by the use of a windmill. A large storage tank is placed at a suitable height so that a sufficient supply may be pumped on windy days to last over intervening periods of calm weather.

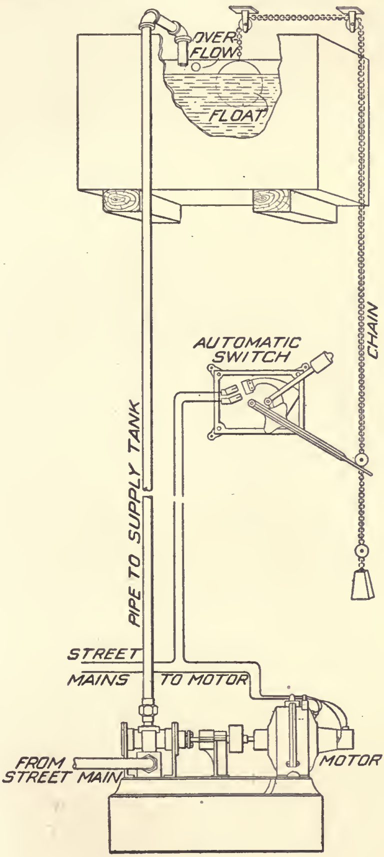


Fig. 10.

HOT-WATER SUPPLY.

All modern systems of plumbing include a hot-water supply to the various sinks, bowls, bathtubs and laundry-tubs throughout the house.

Fig. 13 shows the usual arrangement of a kitchen boiler and water-back with the necessary pipe connections. The boiler is

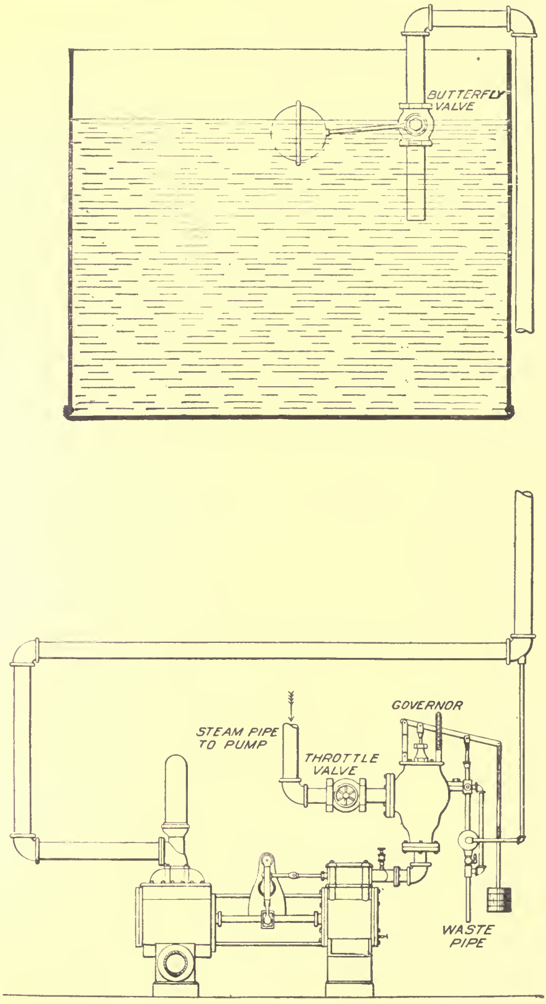

BUTTERFLY
VALVE
STEAM PIPE
TO PUMP
THROTTLE
VALVE
GOVERNOR
WASTE
PIPE

Fig. 11.

commonly made of copper and supported upon a cast-iron base. It may be located in the kitchen near the range, or may be concealed in a nearby closet. The "water-back," so called, is a special casting placed so as to form one side of the fire box in

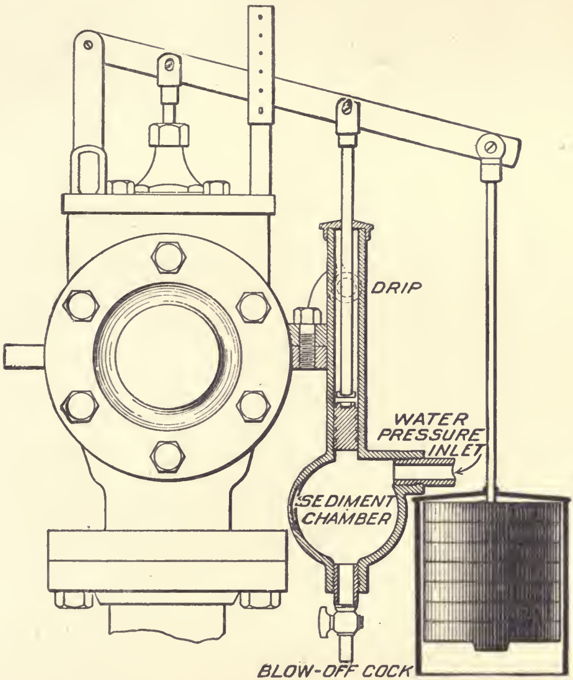


Fig. 12.

the range. The cold-water supply pipe to the boiler usually enters at the top and is carried down to a point near the bottom, as shown by the dotted lines. Connection is made between the bottom of the boiler and the lower chamber of the water-back. The upper chamber is connected at a point about one-third of the way up in the side of the boiler, as shown. The circulation of water through the boiler and supply pipes is the same as already described for hot-water-heating systems. The range fire in contact with the water-back heats the water within it, which causes it to rise through the pipe connected with the upper chamber and

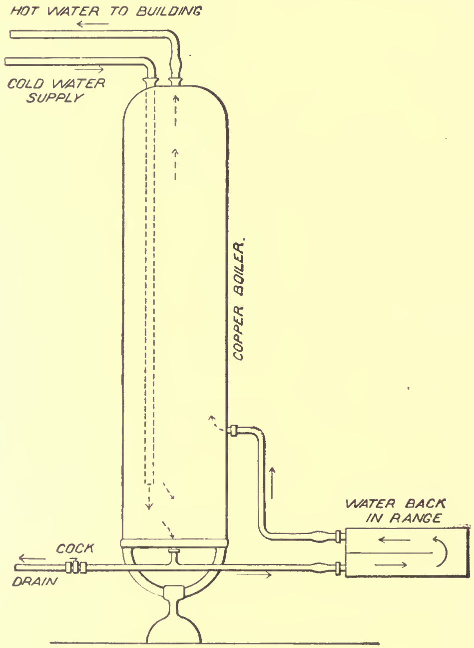


Fig. 13.

flow into the boiler or tank; in the meantime cooler water flows in at the lower connection to take its place, and the circulation thus set up is constant as long as there is a fire in the range.

The "boiler," so called, is not a heater, but only a storage tank. As the water becomes heated it rises to the top of the tank and is carried to the different fixtures in the building through a pipe or pipes connected at this point. The cold-water supply pipe is connected with the house tank so that the pressure in the boiler is that due to the height of the tank above it. When any of the hot-water faucets are open, the pressure of the cold water in the supply pipe forces out the hot water at the top of the boilers and rushes in to take its place. There is no connection between the circulation through the water-back and the pressure in the cold-water supply pipe. The circulation is due only to the difference in temperature between the water in the pipe leading from the top of the water-back and the water in the lower part of the boiler, and difference in elevation of the connections with the boiler. The nearer the top of the boiler the discharge from the water-back is connected, the more rapid will be the circulation and the greater the quantity of water which will be heated in a given time. The cold-water supply simply furnishes a pressure to force the hot water through the pipes to the different fixtures, and replaces any water that is drawn from the boiler.

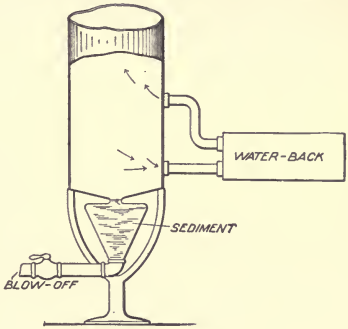


Fig. 14.

Care should always be taken to have the pipes between the water back and the boiler free from sediment or any other obstruction. If the water-back from any cause should become shut off from the boiler, an explosion would be likely to occur if there was a hot fire in the range. Freezing of the pipes is sometimes a cause of accident. The sediment which accumulates more or less rapidly should be regularly blown off through the blow-off cock provided for this purpose at the bottom of the boiler. The best time for doing this is in the morning, before the fire is started. The device shown in Fig. 14 is intended to prevent the sediment from collecting in the pipes or from being drawn into the water-back, making the water roily when a large amount is drawn off at one time. It consists of a small cylinder or chamber connected to the bottom of the boiler in such a way that the sediment will fall into it and not be disturbed by the circulation of the water through the pipes.

Double Water-back Connections. It is often desirable to connect a boiler with two water-backs, one in the kitchen range and another in a laundry stove in the cellar for summer use. Fig. 15 shows the common method of making the connections. In this case either may be used separately, or both together without any adjustment of valves. The blow-off cock at the bottom

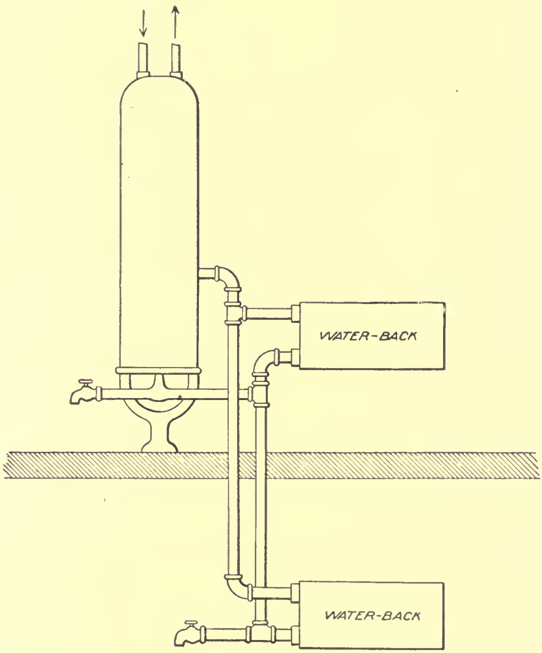
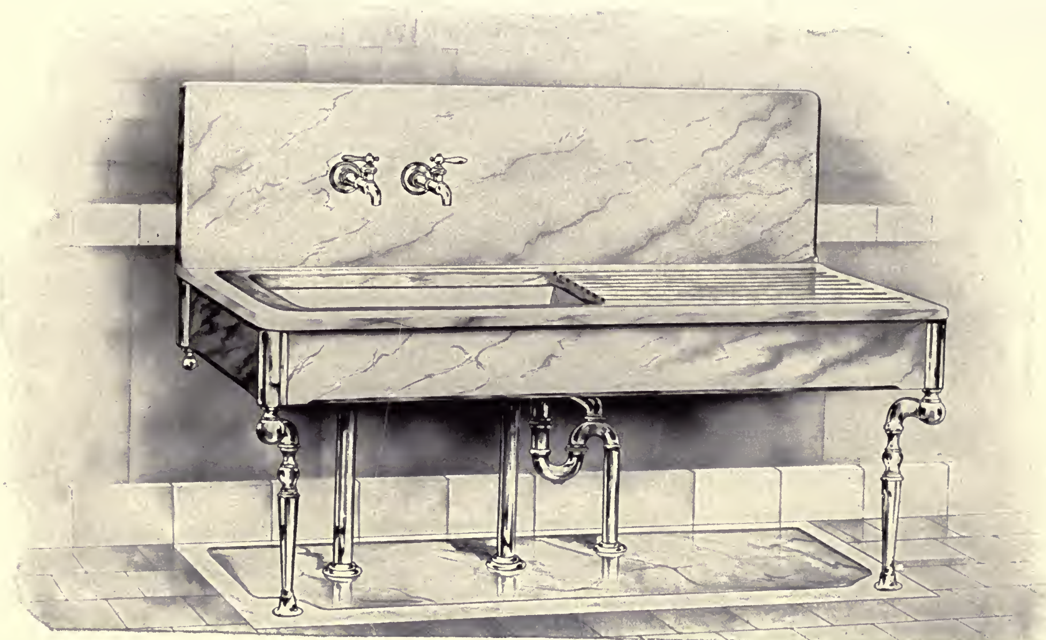


Fig. 15.

of the lower water-back should be opened quite often to clear it of sediment, as it will collect much faster at this point than at the bottom of the boiler.

Double Boiler Connections. It quite frequently happens that the kitchen boiler does not have sufficient capacity for the entire house, and it is not desirable to use a larger boiler on account



ITALIAN MARBLE KITCHEN SINK.
The Federal Company.

of the limited space in the kitchen. In such cases a second boiler may be connected with the laundry stove if one is provided, and the water pipes from both boilers be connected together at some point so that they may both discharge hot water into the same general supply.

Stopcocks should be placed in the pipe connections as shown, so that either boiler may be shut off for repairs without interfering with the operation of the other. Waste cocks should always be used for this purpose, so that when closed there will be a connection between the boiler and the atmosphere. This will prevent damage to the boiler in case those in charge should forget to open the cocks when starting up a fire in the stove with which the boiler is connected.

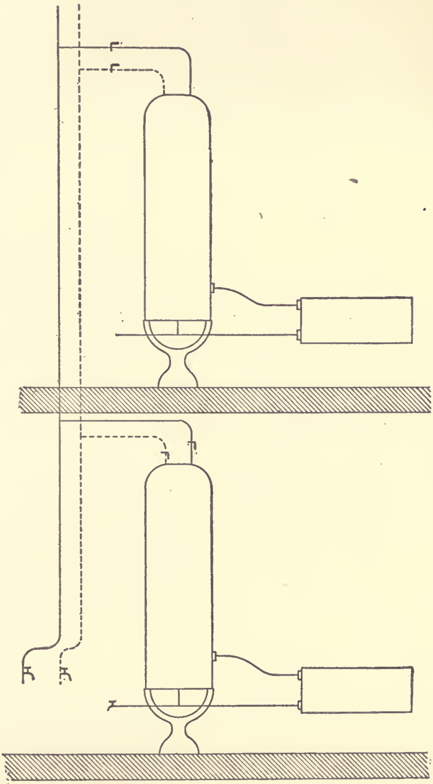


Fig. 16.

Circulation Pipes. It is often desirable to produce a continuous circulation in the distributing pipes so that hot water may be drawn from the faucets at once, without waiting for the cooler water in the pipe between the boiler and the faucet to run out.

This is accomplished by connecting a small pipe with the hot-water pipe near the faucet, and connecting it with the bottom of the boiler as shown in Fig. 17. This makes a circuit, and a constant circulation is produced by the difference in temperature of the water in the supply and circulation pipes.

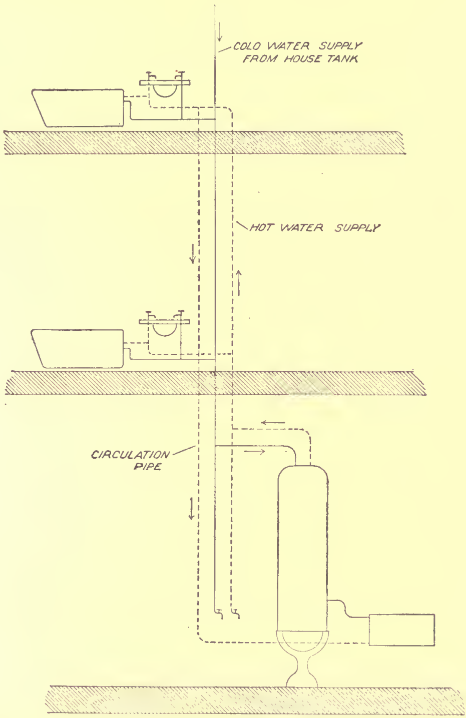


Fig. 17.

Pipe Connections. Brass or copper pipe with screwed fittings should always be used for making the connections between the boiler and water back. Where unions are used they should have ground joints without packing. Lead pipe is too soft to stand the high temperature to which these pipes are sometimes subjected.

Laundry Boilers. In laundries, hotels, etc., where a large amount of hot water is used, it is necessary to have a larger storage tank and a heater with more heating surface than can be

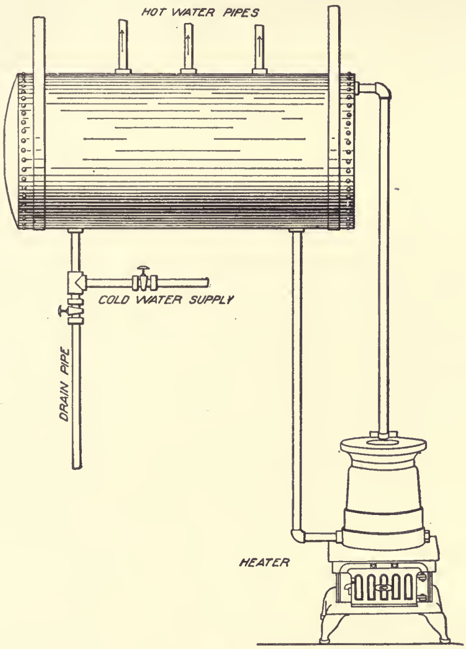


Fig. 18.

obtained in the ordinary range water-back. Fig. 18 shows an arrangement for this purpose.

The boiler may be of wrought iron or steel of any size desired, and is usually suspended from the ceiling by means of heavy strap iron. The heaters used are similar to those employed for hot water warming. The method of making the connections is indicated in the illustration.

The capacity of the heater and tank depends entirely upon the amount of water used. In some cases a large storage tank and a comparatively small heater are preferable, and in others the reverse is more desirable.

The required grate surface of the heater may be computed as follows: first determine or assume the number of gallons to be heated per hour, and the required rise in temperature. Reduce gallons

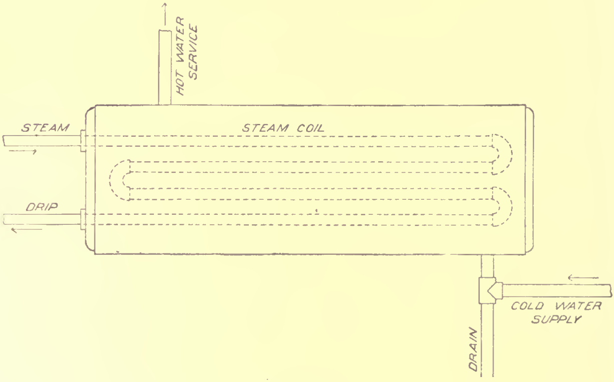


Fig. 19.

to pounds by multiplying by 8.3, and multiply the result by the rise in temperature to obtain the number of thermal units. Assuming a combustion of five pounds of coal per square foot of grate, and an efficiency of 8,000 thermal units per pound of coal, we have

$$\text{Grate Surface in sq. ft.} = \frac{\text{gal. per hour} \times 8.3 \times \text{rise in temp.}}{5 \times 8{,}000}$$

Example.— How many square feet of grate surface will be required to raise the temperature of 200 gallons of water per hour from 40 degrees to 180 degrees?

$$\frac{200 \times 8.3 \times (180 - 40)}{5 \times 8000} = 5.8 \text{ square feet}$$

In computing the amount of water required for bathtubs it is customary to allow from 20 to 30 gallons per tub, and to consider that the tub may be used three or four times per hour as a maximum during the morning. This will vary a good deal, depending upon the character of the building. The above figures are based on apartment hotel practice.

Boilers with Steam Coils. In large buildings where steam is available, the water for domestic purposes is usually warmed by placing a steam coil of brass or copper pipe in the storage tank. This may be a trombone coil made up with brass fittings, or a spiral consisting of a single pipe. Heaters of these types are shown in Figs. 19 and 20. The former must be used in tanks which are placed horizontally, and the latter in vertical tanks. If the steam is used at boiler pressure, the condensation may return directly to the boiler by gravity; but if steam at a reduced pressure is used, it must be trapped to the receiver of a return pump or to the sewer.

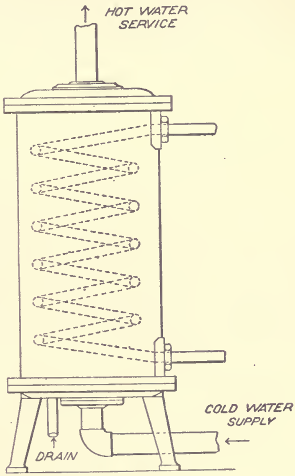


Fig. 20.

The cold water is supplied near the bottom of the tank, and the service pipes are taken off at the top. A drip pipe should be connected with the bottom, for draining the tank to the sewer. Gate valves should be provided in all pipe connections for shutting off in case of repairs. Sometimes a storage tank is connected with a steam-heating system for winter use, and cross connected with a coal-burning heater for summer use where steam is not available. Such an arrangement is shown in Fig. 21.

The efficiency of a steam coil surrounded by water is much greater than when placed in the air. A brass or copper pipe will give off about 200 thermal units per square foot of surface per hour for each degree difference in temperature between the steam and the surrounding water. This is assuming that the water is circulating through the heater so that it moves over the coil at a moderate velocity. In assuming the temperature of the water we must take the average between that at the inlet and outlet.

Example.—How many square feet of heating surface will be required in a brass coil to heat 100 gallons of water per hour from 38 degrees to 190 degrees, with steam at 5 pounds pressure?

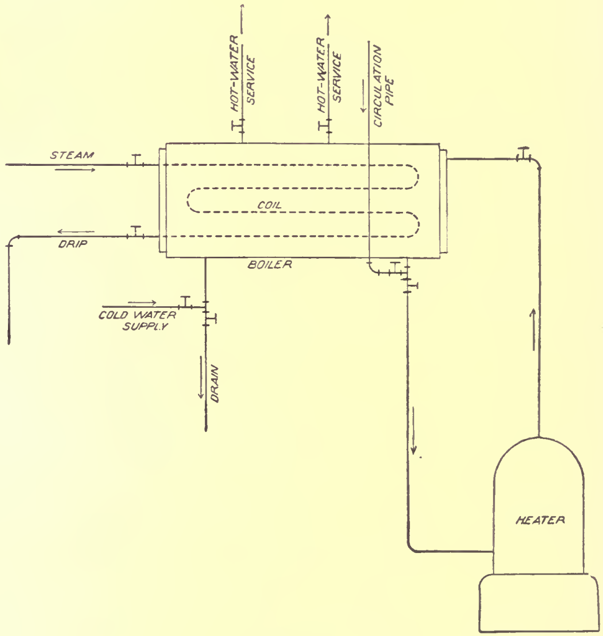


Fig. 21.

Water to be heated $= 100 \times 8.3 = 830$ pounds.
Rise in temperature $= 190 - 38 = 152$ degrees.
Average temperature of water in contact with the coils

$$= \frac{190 + 38}{2} = 114 \text{ degrees}$$

Temperature of steam at 5 pounds pressure = 228 degrees.

The required B. T. U. per hour = 830 × 152 = 126,160.

Difference between the average temperature of the water and the temperature of the steam = 228 — 114 = 114 degrees.

B. T. U. given up to the water per square foot of surface per hour = 114 × 200 = 22,800, and

$$\frac{126,160}{22,800} = 5.5 \text{ square feet. Ans.}$$

EXAMPLES FOR PRACTICE.

1. How many linear feet of 1-inch brass pipe will be required to heat 150 gallons of water per hour from 40 to 200 degrees, with steam at 20 pounds pressure?
Ans. 21.3 feet.

2. How many square feet of grate surface will be required in a heater to heat 300 gallons of water per hour from 50 to 170 degrees?
Ans. 7.4 square feet.

3. A hot-water storage tank has a steam coil consisting of 30 linear feet of 1-inch brass pipe. It is desired to connect a coal-burning heater for summer use which shall have the same capacity. Steam at 5 pounds pressure is used, and the water is raised from 40 to 180 degrees. How many square feet of grate surface are required? Ans. 5.9 sq. ft.

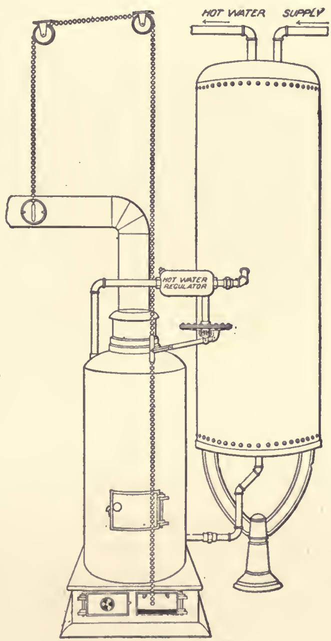


Fig. 22.

4. A hotel has 30 bathtubs, which are used three times apiece between the hours of seven and nine in the morning. The

hot-water system has a storage tank of 400 gallons. Allowing 20 gallons per bath, and starting with the tank full of hot water, how many square feet of grate surface will be required to heat the additional quantity of water within the stated time, if the temperature is raised from 50 to 130 degrees? If steam at 10 pounds pressure is used instead of a heater, how many square feet of heating coil will be required?

Ans. { 11.6 sq. ft. grate.
15.3 sq. ft. coil.

Temperature Regulators. Hot-water storage tanks having special heaters or steam coils should be provided with some means for regulating the temperature of the water. Fig. 22 shows a simple form attached to a coal-burning heater. It consists of a

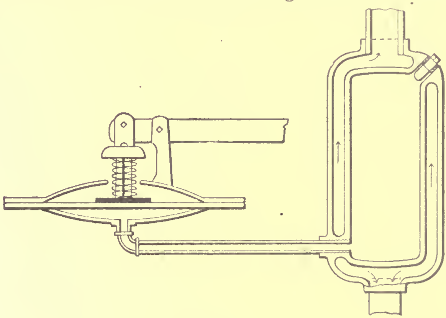


Fig. 23.

casting about nine inches long, tapped at the ends to receive a 2-inch pipe, and containing within it a second shell called the steam generator. (See Fig. 23.) The outer shell is connected with the circulation pipe as shown in Fig. 22. The generator is filled with kerosene, or a mixture of kerosene and water, depending upon the temperature at which it is wished to have the regulator operate. The inner chamber connects with the space below a flexible rubber diaphragm. The boiling point of the mixture in the generator is lower than that of water alone, and depends upon the proportion of kerosene used, so that when the temperature of the water in the outer chamber reaches this point, the mixture boils, and its vapor creates a pressure which forces down the diaphragm and closes the draft door of the heater with which it is connected.

A form of regulator for use with a steam coil is shown in Fig. 24. This consists of a rod made up of two metals having different coefficients of expansion, and so arranged that this difference in expansion will produce sufficient movement, when the water reaches a given temperature, to open a small valve. This

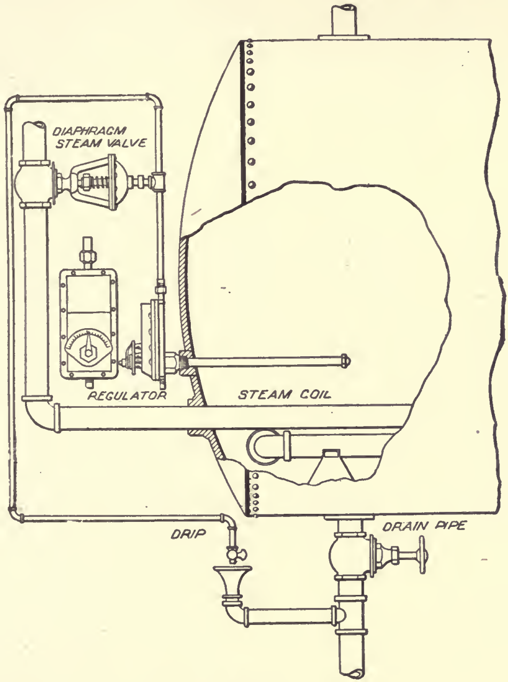


Fig. 24.

allows water pressure from the street main with which it is connected, to flow into a chamber above a rubber diaphragm, thus closing the steam supply to the coil. When the water cools, the rod contracts, and the pressure is released above the diaphragm, allowing the valve to open and thus again admit steam to the coil.

GAS FITTING.

Next to heating and ventilation and plumbing there is no part of interior house construction requiring so much attention as the gas piping and gas fitting.

Gas piping in buildings should be installed according to carefully drawn specifications, and only experienced workmen should be employed. The gas fitter should work from an accurate sketch plan showing the location of all gas service and distributing pipes in the building and the locations of the meter and shut-off cock. The plan should also indicate the exact location and size of the risers and the position of the lights in the different rooms.

Service Pipe and Meter. The service pipe by which the gas is conveyed to a building is always put in by the gas company. The size of this pipe is governed by the number of burners to be supplied, but it should never in any case, even for the smallest house, be less than 1 inch in diameter. This may be slightly larger than is necessary, but the cost is only a little more and the liability of stoppages is much less; this also allows for the future addition of more burners, which is often a matter of much convenience. Service and distributing pipes for water, or naphtha gas, should be from 15 to 20 per cent larger than for coal gas. The material for the main service pipe, from the street to the house, should be either lead or wrought iron. As a rule, wrought-iron pipe with screwed joints is preferable to lead, because it is less likely to sag in the trench, thus causing dips for the accumulation of water of condensation. Care must be observed in the use of wrought-iron pipe to protect it by coating with asphalt, or coal tar, to prevent corrosion. The pipe should also be well supported in the trench. Service pipes should preferably rise from the street gas main toward the house in order to allow all condensation to run back into the mains. This, however, cannot always be done, owing to the relative levels of the street main and the meter in the house. The latter should be placed in a cool, well-lighted position, at or below the level of the lowest burner, which is usually in the cellar. If the meter is below the gas main, the service pipe must grade toward the house and should be provided with a drip pipe, or "siphon," before connecting with the meter.

When water accumulates in the siphon, the cap is removed and the pipe drained. The gas company usually supplies and sets the meter, which should be of ample size for the number of lights burned.

A stopcock, or valve, is placed by the company in the service pipe, so that the gas may be shut off from each building separately. This is usually placed outside near the curb in the case of buildings requiring a pipe $1\frac{1}{2}$ inches in diameter, or larger. In the case of theaters or assembly halls it is often required by law as a safeguard in case of fire. The meter is connected with both the service pipe and the main house pipe by means of short connections of extra heavy lead pipe. A cock is placed near the meter, and in large buildings this is arranged so that a lock may be attached to it when the gas is shut off by the company. Gate valves are preferable for gas mains, as they give a free opening equal to the full size of the pipe.

PIPES.

Distributing Pipes. The distributing pipes inside of a house are usually of wrought iron, except where exposed in rooms, or

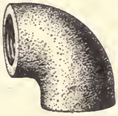


Fig. 25.

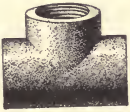


Fig. 26.

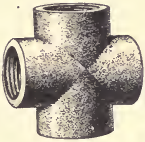


Fig. 27.

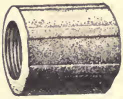


Fig. 28.

carried along walls lined with enameled brick, or tile, in which case they may be of polished bass, or copper. The chief requirements for wrought-iron distributing pipes are that they be carefully welded and perfectly circular in section. The first is important in order to avoid splitting when cutting or threading them on the pipe bench.

All gas pipes are put together with screwed joints, a thread being cut upon the outside. When the pipe is irregular in section the threading will be more or less imperfect, and as a result the joints will be defective. A good gas fitter must examine all pipe as it is delivered at the building, and observe the section either by means of the eye or by the use of calipers. Plain wrought-iron pipe is likely to rust upon the inside, especially where the gas supplied is imperfectly purified, and for this reason it is often advisable to use rustless, or galvanized pipe, for the smaller sizes.

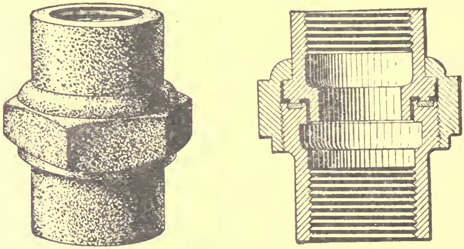


Fig. 29.

Fittings and Joints. The fittings used in gas piping are similar to those employed in steam work, such as couplings, elbows, tees, crosses, etc. (see Figs. 25, 26, 27 and 28). Other fittings not so extensively used are the union, the flange union, the running socket and right and left couplings. Fig. 29 shows a screwed union and Fig. 30 a flange. These fittings are of cast iron, or of malleable iron, the latter being preferred for the smaller sizes. Fittings may be either galvanized, or rustless, as in the case of pipe, and it is especially necessary that they be free from sand holes. In making pipe joints the gas fitter should make use of red lead, or red and white lead mixed, to make up for any possible imperfections in the threads; this, however, should be used sparingly so that the pipe may not be choked or reduced in size. The use of gas fitters' cement should be prohibited. It is important that each length should be tightly screwed into the fitting before the next length is put on. It is always a wise precaution

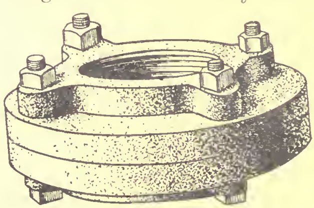


Fig. 30.

to examine each length of pipe before it is put in place, to make sure it is free from imperfections of any kind.

Running Pipes and Risers. All large risers should be exposed, and it is desirable to keep all piping accessible as far as possible so that it may be easily reached for repairs. All horizontal pipes should be run with an even though slight grade toward the riser, and all sags in the pipes must be avoided to prevent the collection of water, and for this reason they should be well supported. Floor boards over all horizontal pipes should be fastened down with screws so that they may be removed for inspection of the pipes. When it becomes necessary to trap a pipe, a drip with a drain cock must be put in, but this should always be avoided under floors or in other inaccessible places. When pipes under floors run across the timbers, the latter should be cut into near the ends, or where supported upon partitions, in order to avoid weakening the timbers. All branch outlet pipes should be taken from the side or top of the running lines, and bracket pipes should be run up from below instead of dropping from above. Never drop a center pipe from the bottom of a running line; always take such an outlet from the side of the pipe. Where possible it is better to carry up a main riser near the center of the building, as the distributing pipes will be smaller than if carried up at one end. Where this is done the timbers will not require so much cutting, and the flow of gas will be more uniform throughout the system.

When a building has different heights of post it is always better to have an independent riser for each height rather than to drop a system of piping from a higher to a lower post and grading to a lower point and establishing drip pipes. Drips in a building should be avoided if possible and the whole system of piping be so arranged that any condensed gas will flow back through the system and into the service pipe. All outlet pipes should be securely fastened in position, so that there will be no possibility of their moving when the fixtures are attached. Center pipes should rest on a solid support fastened to the floor timbers near the top. The pipe should be securely fastened to the support to prevent movement sidewise. The drop must be perfectly plumb and pass through a guide fastened near the bottom of the timbers in order to hold it rigidly in position. (See arrangement, Fig. 31.)

Unless otherwise directed, outlets for brackets should be placed $5\frac{1}{2}$ feet from the floor except in the cases of hallways and bathrooms, where it is customary to place them 6 feet from the floor. Upright pipes should be plumb, so that nipples which project through the walls will be level; the nipples should not

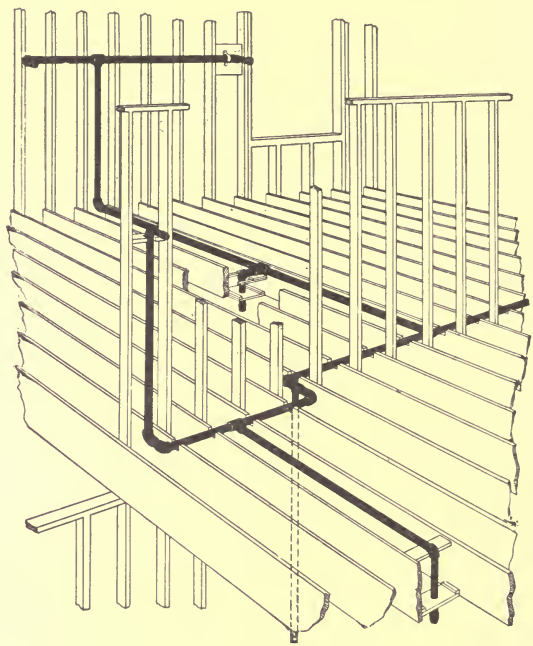


Fig. 31.

project more than $\frac{3}{4}$ inch from the face of the plastering. Lathes and plaster together are usually about $\frac{3}{4}$ inch thick, so the nipples should project about $1\frac{1}{2}$ inches from the face of the studding.

Gas pipes should never be placed on the bottoms of floor timbers that are to be lathed and plastered, because they are inaccessible in case of leakage or alterations.

Pipe Sizes. All risers and distributing pipes, and all branches to bracket and center lights should be of sufficient size to supply the total number of burners indicated on the plans. Mains and branches should be proportioned according to the number of lights they are to supply, and not the number of outlets.

No pipe should be less than $\frac{3}{8}$ inch in diameter, and this size should not be used for more than two-bracket lights. No pipe for a chandelier should be less than $\frac{1}{2}$ inch up to four burners, and it should be at least $\frac{3}{4}$ inch for more than four burners. The following table gives sizes of supply pipes for different numbers of burners and lengths of run.

TABLE VII.

Size of Pipe. Inches.	Greatest Length of Run. Feet.	Greatest Number of Burners to be Supplied.
$\frac{3}{8}$	20	2
$\frac{1}{2}$	30	4
$\frac{3}{4}$	50	15
1	70	25
$1\frac{1}{4}$	100	40
$1\frac{1}{2}$	150	70
2	200	140
$2\frac{1}{2}$	300	225
3	400	300
4	500	500

Testing Gas Pipes. As soon as the piping is completed, it should be tested by means of an air pump; a manometer or mercury gage is used to indicate the pressure. In the case of large buildings, it is better to divide the piping into sections, and test each separately. All leaks revealed must be repaired at once, and the test repeated until the whole system is air tight at a pressure of from 15 to 20 inches of mercury, or $7\frac{1}{2}$ to 10 pounds per square inch.

The final test is of great importance. This test is to provide against future troubles and dangers from leaks resulting from sand holes in the fittings, split pipe, imperfect threads, loose joints or outlets left without capping. If the building is new, a careful inspection should first be made to see that all outlets are closed, then the valve in the service pipe closed and the air pump attached to any convenient side-light. To the same outlet or an adjacent one attach the mercury column gage used by gas fitters, and having a column from 15 to 20 inches in height. Care must be taken that there are no leaks in the gage or its connections; a tight-closing valve must be placed between the gas pipe and the temporary connections with the pump, so that it may be shut off immediately after the pump stops, thus preventing any leakage through the pump valves or hose joints. When all is ready, pump the system full of air until the mercury rises to a height of at least 12 inches in the gage; then close the intermediate valve between the pump and the piping. Should the mercury column "stand" for five minutes, it is reasonable to assume that the pipes are sufficiently tight for any pressure to which they will afterward be subjected.

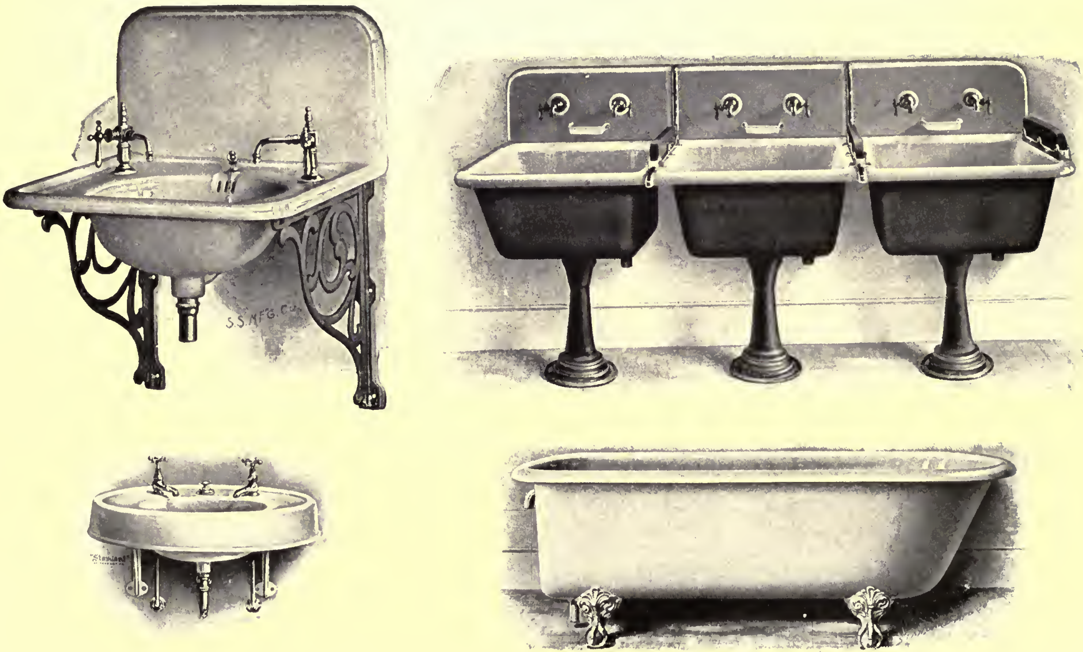
If the mercury rises and falls with the strokes of the pump, it indicates a large leak or open outlet near the pump. But should there be a split pipe or an aggregation of small leaks, the mercury will run back steadily between the strokes of the pump, though more slowly than it rose. Should it rise well in the glass and sink at the rate of 1 inch in five seconds, small leaks in fittings or joints may be expected.

A leak that cannot be detected by the sound of issuing air may usually be found by applying strong soap-water with a brush over suspected joints or fittings; the leak in this case being indicated by the bubbles blown by the escaping air. Sometimes it is necessary to use ether in the pipes for locating leaks, if the pipes are in partitions or under floors. The ether is put into a bend of the connecting hose, or in a cup attached to the pump, and forced in with the air. By following the lines of the pipe, the approximate position of a leak may be determined by the odor of escaping ether.

If the house is an old one or has been finished, the meter should be taken out and the bottom of the main riser capped. Next remove all fixtures and cap the outlets. Then use ether to locate the leaks before tearing up floors or breaking partitions.

GAS FIXTURES.

Burners. Illuminating gas is a complex mixture of gases, of which various chemical compounds of carbon and hydrogen



REPRESENTATIVE MODERN BATH ROOM AND LAVATORY FITTINGS.

form the principal light-giving properties. Gas always contains more or less impurities, such as carbonic oxide, carbonic acid, ammonia, sulphureted hydrogen and bisulphides of carbon. These are partly removed by purifying processes before the gas leaves the works.

When the gas-jet is lighted, the hydrogen is consumed in the lower part of the flame, producing sufficient heat to render the minute particles of carbon incandescent. The hydrogen, in the process of combustion, combines with the oxygen from the air, forming an invisible vapor of water, while the carbon unites with the oxygen, forming carbonic acid.

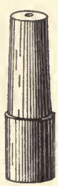


Fig. 32.

Fig. 33.

Fig. 34.

Various causes tend to render combustion incomplete: there may be excessive pressure of gas, lack of air or defective burners. An excess of pressure at the burners causes a reduction of the amount of illumination; on the other hand, if the pressure is insufficient, the heat of the flame will not raise the carbon to a white heat, and the result will be a smoky flame. It therefore follows that for every burner there is a certain pressure and corresponding flow of gas which will cause the brightest illumination.

There is a great variety of burners upon the market, among which the following are the principal types:

The single-jet burner, the bat's-wing burner, the fish-tail burner, the Argand burner, the regenerative burner and the incandescent burner.

The Single-jet burner (Fig. 32) is the simplest kind, having only one small hole from which the gas issues. It is suitable only where a very small flame is required.

The Bat's-wing or slit burner (Fig. 33) has a hemispherical tip with a narrow vertical slit from which the gas spreads out in a thin, flat sheet, giving a wide and rather low flame, resembling in shape the wing of a bat, from which it is named. The common kind of slit burners are not suitable for use with globes, as the flame is likely to crack the glass.

The Union-jet or *Fish-tail* burner (Figs. 34 and 35) consists of a flat tip slightly depressed or concave in the center, with two small holes drilled, as shown in Fig. 35. Two jets of equal size issue from these holes, and by impinging upon each other produce a flat flame longer and narrower in shape than the bat's-wing, and not unlike the tail of a fish. Neither of these burners require a chimney, but the flames are usually encased with glass globes.

Fig. 35.

The Argand burner (Fig. 36) consists of a hollow ring of metal connected with the gas tube, and perforated on its upper surface with a series of fine holes, from which the gas issues, forming a round flame. This burner requires a glass chimney. As an intense heat of combustion tends to increase the brilliancy of the flame, it is desirable that the burner tips shall be of a material that will cool the flame as little as possible. On this account

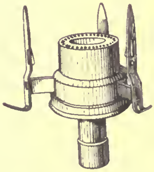


Fig. 36.

Fig. 37.

Fig. 38.

metal tips are inferior to those made of some nonconducting material, such as lava, adamant, enamel, etc. Metal tips are also objectionable because they corrode rapidly, and thus obstruct the passage of the gas. Figs. 37 and 38 show lava tips for bat's-wing and fish-tail burners. Burner tips should be cleaned occasionally, but care should be taken not to enlarge the slits or holes.

In all *regenerative* burners the high temperature due to the combustion in a gas flame is used to raise the temperature of the gas before ignition, and of the air before combustion. These powerful burners are used for lighting streets, stores, halls, etc.

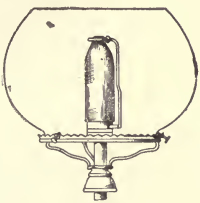


Fig. 39.

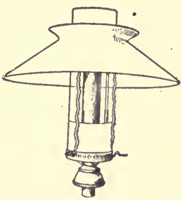


Fig. 40.

In the incandescent burner the heat of the flame is applied in raising to incandescence some foreign material, such as a basket of magnesium or platinum wires, or a funnel-shaped asbestos wick or mantel chemically treated with sulphate of zirconium and other chemical elements. A burner of this kind is shown in Fig. 39, where the mantel may be seen supported over the gas flame by a wire at the side. Fig. 40 shows another form of this burner in which a chimney and shade are used in place of a globe. Burners of this kind give a very brilliant white light when used with water gas unmixed with naphtha gases. The mantel, however, is very fragile, and is likely to lose its incandescence when exposed to an atmosphere containing much dust.

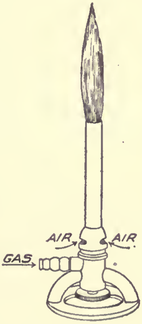


Fig. 41.

The Bunsen burner shown in Fig. 41 is a form much used for laboratory work. It burns with a bluish flame, and gives an intense heat without smoke or soot. The gas before ignition is mixed with a certain quantity of air, the proportions of gas and air being regulated by the thumbscrew at the bottom, and by screwing the outer tube up or down, thus admitting a greater or less quantity of air at the openings indicated by the arrows. This same principle is utilized in a burner for brazing, the general form of which is shown in Fig. 42. A flame of this kind will easily melt brass in the open air.

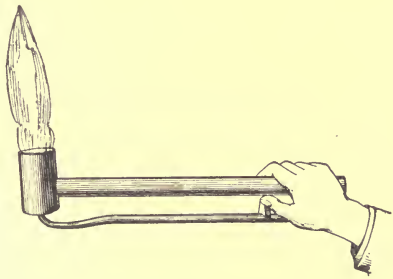


Fig. 42.

Cocks. It is of greatest importance that the stopcocks at the fixtures should be perfectly tight. It is rare to find a house piped for gas where the pressure test could be successfully applied without first removing the fixtures, as the joints of folding

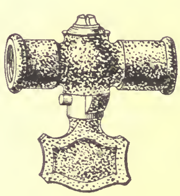


Fig. 43.

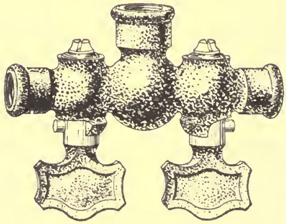


Fig. 44.

brackets, extension pendants, stopcocks, etc., are usually found to leak more than the piping. The old-fashioned, "all-around" cock should never be allowed under any conditions whatever; only those provided with stop pins should be used. Various forms of cocks with stop pins are shown in Figs. 43, 44 and 45. All joints should be examined and tightened up occasionally to prevent their becoming loose and leaky.

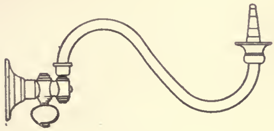


Fig. 46.

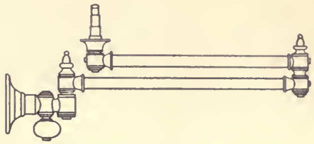


Fig. 47.

Brackets and Chandeliers. Poor illumination is frequently caused by ill-designed or poorly constructed brackets or chandeliers. Gas fixtures, almost without exception, are designed solely from an artistic standpoint, without regard to the proper conditions for obtaining the best illumination. Fixtures having too many scrolls or spirals may, in the case of imperfectly purified gas, accumulate a large amount of a tarry deposit which in time hardens and obstructs the passages. Another fault is the use of very small tubing for the fixtures, while a third defect consists in the many leaky stopcocks of the fixtures, caused either by defec-

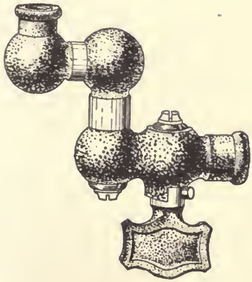


Fig. 45.

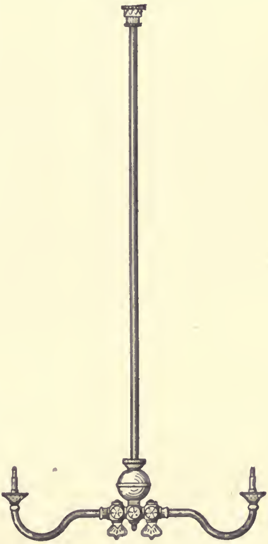


Fig. 48.

tive workmanship, or by the keys becoming worn and loose. Common forms of brackets are shown in Figs. 46 and 47, the latter being an extensive bracket. There is an endless variety of chandeliers used, depending upon the kind of building, the finish of the room and the number of lights required. Figs. 48, 49 and 50 show common forms for dwelling houses, Fig. 50 being used for halls and corridors.

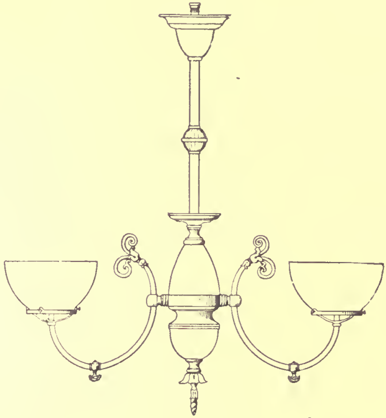


Fig. 49.

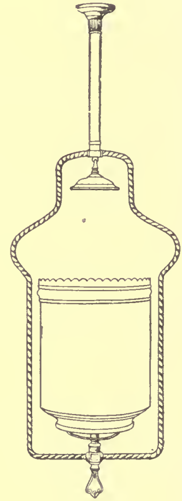


Fig. 50.

Globes and Shades. Next to the burners, the shape of the globes or shades surrounding the flame affects the illuminating power of the light. In order to obtain the best results, the flow of air to the flame must be steady and uniform. Where the supply is insufficient the flame is likely to smoke ; on the other hand, too strong a current of air causes the light to flicker and become dim through cooling.

Globes with too small openings at the bottom should not be used. Four inches should be the smallest size of opening for an ordinary burner. All glass globes absorb more or less light, the loss varying from 10 per cent for clear glass to 60 per cent or more for colored or painted globes. Clear glass is therefore much more economical, although where softness of light is especially desired the use of opal globes is made necessary.

COOKING AND HEATING BY GAS.

Cooking by gas as well as heating is now very common and there is a great variety of appliances for its use in this way. Cooking by gas is less expensive and less troublesome than by coal, oil or wood and is more healthful on account of the absence of waste heat, smoke and dust. A gas range is always ready for use and is instantly lighted by applying a match to the burner. The fire, when kindled, is at once capable of doing its full work; it is easily regulated and can be shut off the moment it has been used, so that if properly managed there is no waste of fuel as in the case of coal or wood. The kitchen in the summertime may be kept comparatively cool and comfortable. Gas stoves are made in all sizes, from the simple form shown in Fig. 51 to the most elaborate range for hotel use. A range for family use, with ovens and water heater, is shown in Fig. 52. Figs. 53 and 54 show the forms of burners used for cooking, the former being a griddle burner and the latter an oven burner.

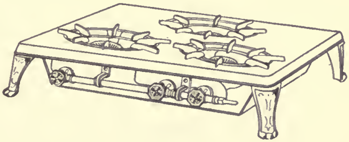


Fig. 51.

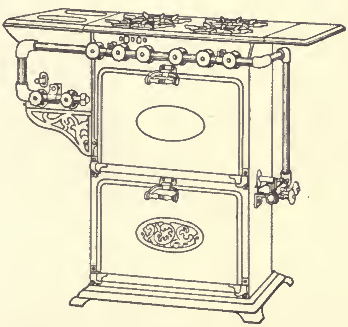


Fig. 52.

A broiler is shown in Fig. 55; the sides are lined with asbestos, and the gas is introduced through a large number of small openings. The asbestos becomes heated and the effect is the same as a charcoal fire upon both sides.

Heating by Gas. Gas as a fuel has not been used to any great extent for the warming of whole buildings, its application

Fig. 54.

being usually confined to the heating of single rooms. Unlike cooking by gas, a gas fire for heating is not as cheap as a coal fire when kept burning constantly. In other ways it is effective and convenient. It is especially adapted to the warming of small apartments and single rooms where heat is only wanted occasion-

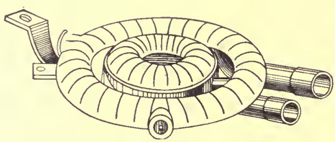


Fig. 53.

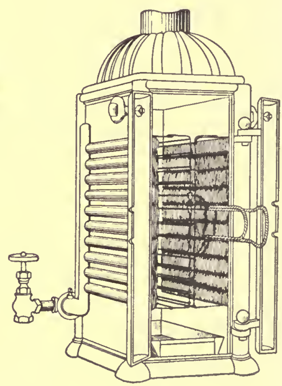


Fig. 55.

ally and for brief periods of time. In the case of bedrooms, bathrooms or dressing-rooms, a gas fire is preferable to other modes of warming and fully as economical. It may be used on cold winter days as a supplementary source of heat in houses heated by stoves or by furnaces. Again, a gas fire may be used as a substitute for the regular heating apparatus in a house, in the spring or fall, when the fire in the furnace or boiler has not yet been started. It is often employed as the only means for heating smaller bedrooms, guest rooms, bathrooms, and for temporary heating in summer hotels where fires are required only on occasional cold days.

The most common form of heater is that shown in Fig. 56. This is easily carried from room to room and may be connected with a gas-jet, after first removing the tip, by means of rubber tubing. The heater is simply a large burner surrounded by a sheet-iron jacket or funnel. Another and more powerful form is the gas radiator, shown in Fig. 57. This is arranged with a flue for conducting the products of combustion to the chimney, as shown in the section Fig. 58. Each section of the radiator consists of an outer and an inner tube with the gas flame between the

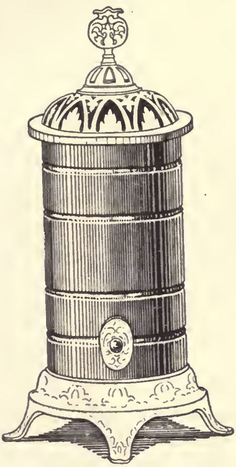


Fig. 56.

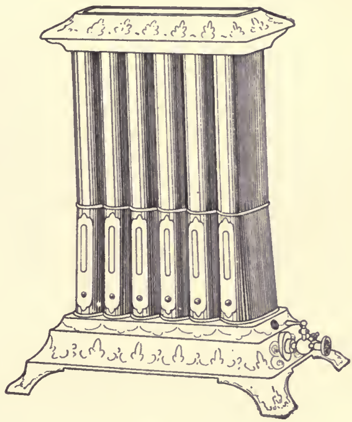


Fig. 57.

two. This space is connected with the flue, while the air to be heated is drawn up through the inner tube, as shown by the arrows.

Fig. 59 shows an asbestos incandescent grate, and Fig. 60 a grate provided with gas logs made of metal or terra-cotta and asbestos. The gas issues through small openings among the logs, and gives the appearance of an open wood fire.

Hot-water Heaters. The use of gas cooking ranges makes it necessary to provide separate means for heating water. This is accomplished in several ways. The range shown in Fig. 52 has a boiler attached which is provided with a separate burner.

Fig. 61 shows a gas heater attached to the ordinary kitchen boiler. A section through the heater is shown in Fig. 62. This consists of a chamber surrounded by an outer jacket with an air space between. Circulation pipes, through which the water passes, are hung in the inner chamber just above a powerful gas-burner placed at the bottom of the heater.

A heater of different form for heating larger quantities of

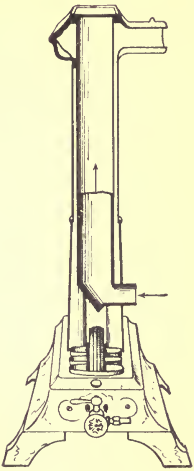


Fig. 58.

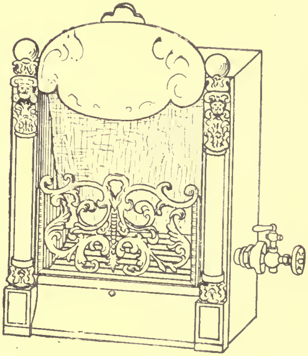


Fig. 59.

Fig. 60.

water is shown in Figs. 63 and 64. This consists, as in the case just described, of a circulation coil suspended above a series of burners. The supply of gas admitted to the burners is regulated by an automatic valve, which is opened more or less as the flow of water through the heater is increased or diminished. When no water is being used, the gas is shut off from the burners, and only a small "pilot light," which takes its supply from above the automatic valve, is left burning. As soon as a faucet in any part of the building is opened and a flow of water started through the heater, the automatic valve opens, admitting gas to the main burn-

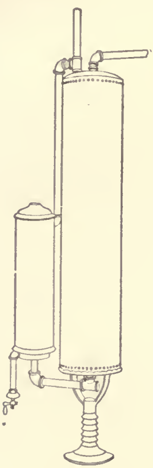


Fig. 61.

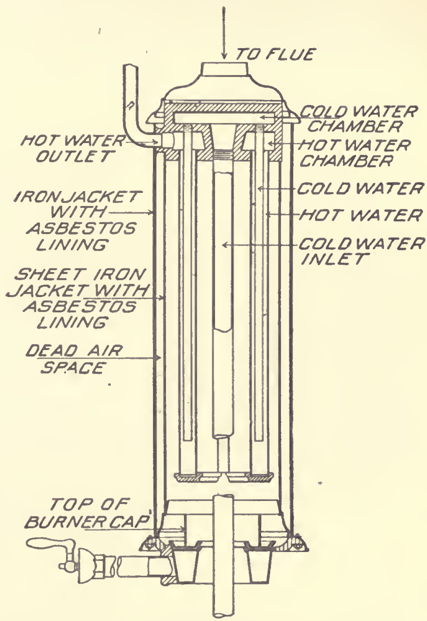


Fig. 62.

ers, which is ignited by the pilot light, and in a few moments hot water will flow from the faucet. The heater shown has a capacity of 9 gallons per minute from a temperature of 55 to 130°.

Another type is that known as the instantaneous water heater, one form of which is shown in Fig. 65. This is made especially for bathrooms, and will produce a continuous stream of hot water whenever desired. The heater is shown in section in Fig. 66, in which A is the gas valve, B the water valve, D the pilot light, FF the burners, I a conical heating ring, J a disc to retard and spread the rising heat, K a perforated copper screen, and L a revolving water distributer. In this heater the water is exposed directly to the heated air and gases in addition to its passing over the heated surface of the ring I. The upward arrows show the path of the heat, and the downward arrows the passage of the water.

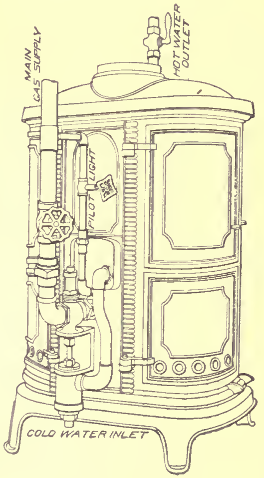


Fig. 63.

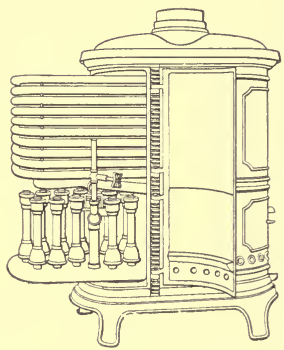


Fig. 64.

GAS METERS.

The meter should be placed in such a position that it is easily accessible and may be read without the use of an artificial light. It is connected into the system between the service pipe and main riser to the building, the connections being made as shown in Fig. 67.

Different meters vary but little in the arrangement of the dials. In large meters there are often as many as five dials, but those used for dwelling houses usually have but three. Fig. 68 shows the common form of index of a dry meter. The small index hand, D, on the upper dial is not taken into consideration when reading the meter, but is used merely for testing. The three dials, which record the consumption of gas, are marked A, B and C, and each complete revolution of the index hand denotes 1,000, 10,000 and 100,000 cubic feet respectively. It should be noted that the index hands on the three dials do not move in the same direction:

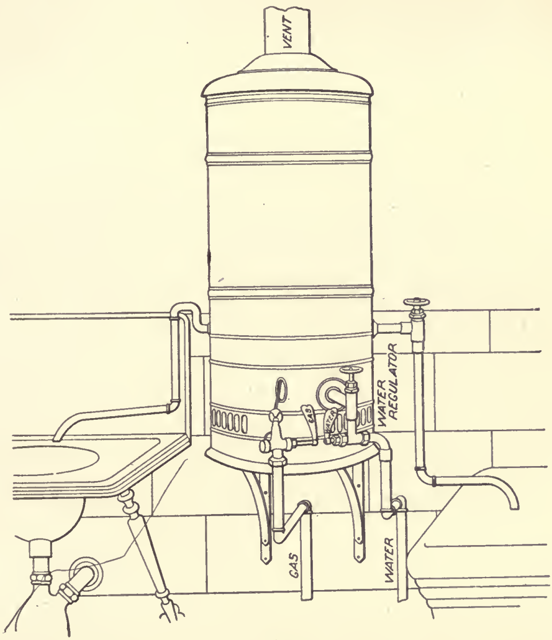


Fig. 65.

A and C move with the hands of a watch, and B in the opposite direction. The index shown in Fig. 68 should be read 48,700. Suppose after being used for a time, the hands should have the position shown in Fig. 69. This would read 64,900, and the amount of gas used during this time would equal the difference in the readings: 64,900 — 48,700 = 16,200 cubic feet.

GAS MACHINES.

While the manufacture of gas for cities and towns is a matter beyond the scope of gas fitting, it may not be out of place to take up briefly the operation of one of the forms of gas machines

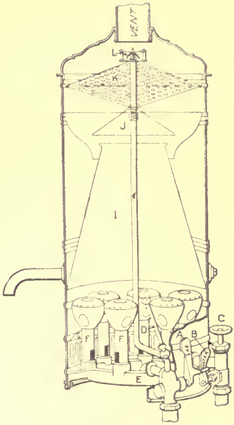


Fig. 66.

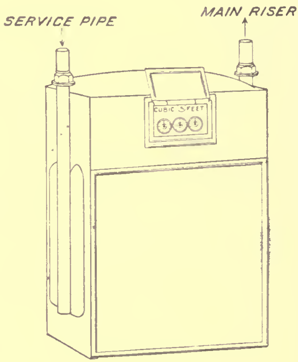


Fig. 67.

which are used for supplying private residences or manufacturing plants.

The general arrangement of the apparatus is shown in Fig.

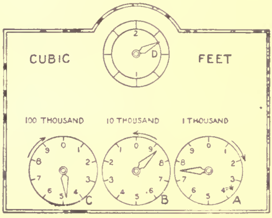


Fig. 68.

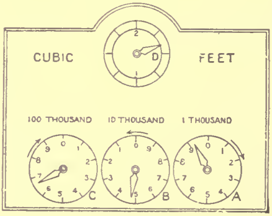


Fig. 69.

70, which consists of a generator, containing evaporating pans or chambers, and an automatic air pump, together with the necessary piping for air and gas. The gas made by these machines is com-

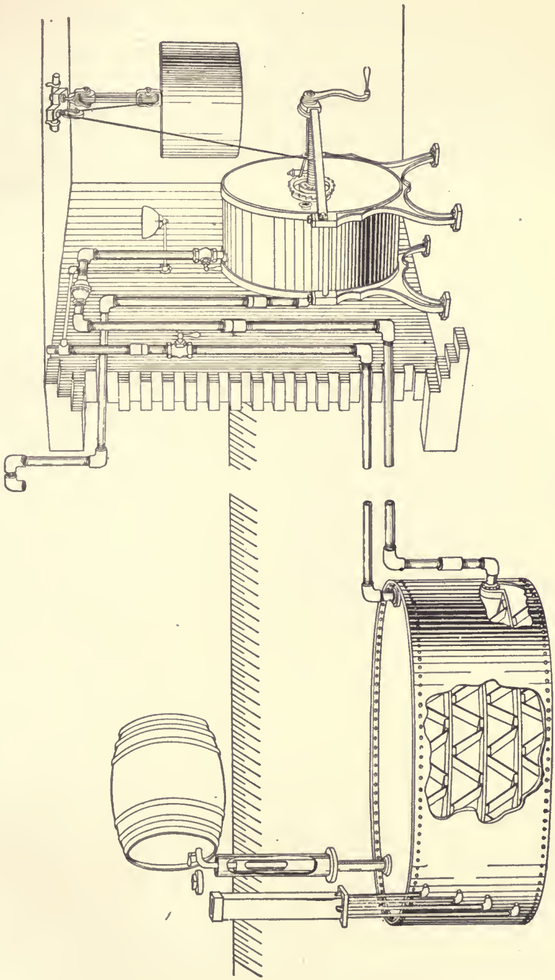


Fig. 70.

monly known as carbureted air gas, being common air impregnated with the vapors of gasoline. It burns with a rich bright flame similar to coal gas, and is conducted through pipes and fixtures in the same manner.

Referring to Fig. 70, the automatic air pump is seen in the cellar of the house, and connected to it and running underground are the air and gas pipes connecting it with the generator, which may be a hundred feet or more away if desired. When the machine is in operation, the pump forces a current of air through the generator, where it becomes carbureted, thus forming an illuminating gas that is returned through the gas pipe to the house, where it is distributed to the fixtures in the usual way. The operation is automatic, gas being generated only as fast and in such quantities as required for immediate consumption. The process is continuous while the burners are in use, but stops as soon as the lights are extinguished. Power for running the air compressor is obtained by the weight shown at the right, which must be wound up at intervals, depending upon the amount of gas consumed. An air compressoı to be run by water power is shown in Fig. 71.

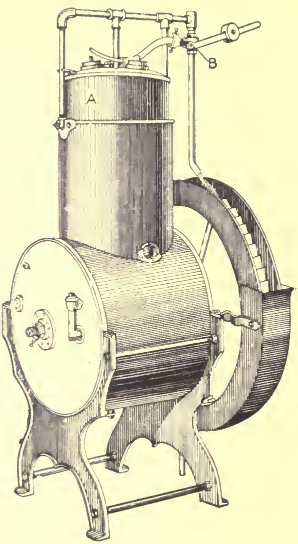


Fig. 71.

The action of this machine is entirely automatic. the supply of water being controlled by the rising and falling of the holder A. which, being attached by a lever to the valve B, regulates the amount of water supplied to the wheel in exact proportion to the number of burners lighted. If all the burners are shut off. the pressure accumulating in the holder A raises it and shuts the water off. If a burner is lighted, the holder falls slightly, allowing just enough water to fall upon the wheel to furnish the amount of gas required. A pump or compressor of this kind requires about two gallons of water per hour for each burner. The advan-

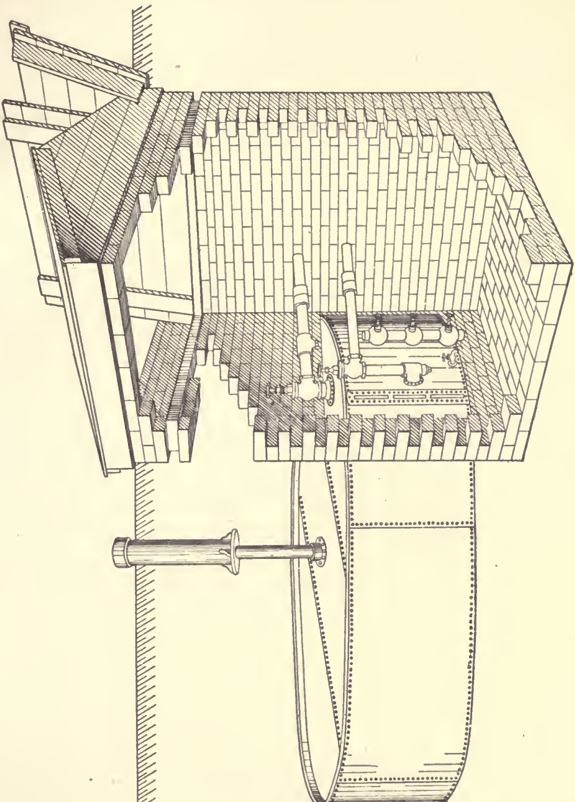


Fig. 72.

tages of a water compressor over one operated by a weight are that it requires no attention, never runs down and is ready for immediate use at all times.

The generator is made up of a number of evaporating pans or chambers placed in a cylinder one above another. These chambers are divided by supporting frames into winding passages, which give an extended surface for evaporation. Fig. 72 shows the generator when set with a brick pit and manhole at one side. It is supplied with mica gages for showing the amount of gasoline in each pan, and with tubes and valves for distributing it to the different pans as required. In small plants the generator is usually buried without the pit being provided, but for larger plants the setting shown in Fig. 72 is recommended. Carbureted air gas of standard quality contains 15 per cent of vapor to 85 per cent of air. A regulator or mixer for supplying gas having these proportions is shown in section in Fig. 73. It consists of a cast-iron case in which is suspended a sheet-metal can, B, filled with air and closely sealed. The balance beam E, to which this is hung, is supported by the pin H, on agate bearings. Since the weight of the can B is exactly balanced by the ball on the beam E, movement of B can only be caused by a difference in the weight or density of the gas inside the chamber A and surrounding the can B. If the gas becomes too dense, B rises and opens the valve C, thus admitting more air; and if it becomes too light, C closes and partially or wholly shuts off the air, as may be required.

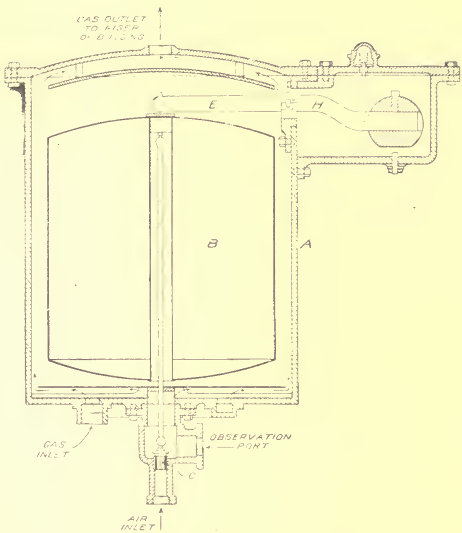


Fig. 73.

REVIEW QUESTIONS.

PRACTICAL TEST QUESTIONS.

In the foregoing sections of this Cyclopedia numerous illustrative examples are worked out in detail in order to show the application of the various methods and principles. Accompanying these are examples for practice which will aid the reader in fixing the principles in mind.

In the following pages are given a large number of test questions and problems which afford a valuable means of testing the reader's knowledge of the subjects treated. They will be found excellent practice for those preparing for College, Civil Service, or Engineer's License. In some cases numerical answers are given as a further aid in this work.

REVIEW QUESTIONS

ON THE SUBJECT OF

MACHINE DESIGN.

PART I.

The drawings made in accordance with the problems below should be traced in ink on tracing cloth 18 by 24 inches in size, and having a border line $\frac{1}{2}$ inch inside the edge of the paper.

PROBLEMS.

1. Suppose a 30-inch pulley is substituted for the 42-inch in the problem given, and that the pulley on the motor remains $10\frac{1}{2}$ inches as before, how fast must the motor run to give the rope the same speed, 150 feet per minute?

2. Will the horse-power of the motor be changed with this new condition? Explain fully.

3. Calculate the width of double belt for above condition.

4. What is the torque on the motor shaft for above condition?

5. Calculate the size of shaft in the small pulley for above condition.

6. Calculate the size of shaft in the 30-inch pulley for above condition.

7. Design and draw both pulleys for above condition, making complete working drawings, and giving all calculations in full.

8. Taking the original problem as given in the text, suppose it is desired to increase the large gear to 45 inches diameter, calculate the load on the tooth, and a suitable pitch and face to take this load.

9. How many teeth must the pinion have to give the same speed of rope, 150 feet per minute, assuming that the motor runs 470 revolutions per minute, for condition in Problem 8?

10. Calculate the bore of pinion for this case.

11. Design and draw the gears for the conditions of Problems 8 and 9, giving all calculations in full.

12. When there is but 3,000 pounds on the rope, what are the tensions in each end of the brake strap, assuming that the size of drum and other conditions remain the same?

13. How much pressure on the foot lever would it take to hold this load of 3,000 pounds on the rope?

14. Suppose we put a bearing 9 inches long on the drum shaft; the distance, center to center of bearings, would then be 3 feet $5\frac{3}{4}$ inches, gears, drum, brake, and load being same as in the original problem of the text. Calculate the diameter of the drum shaft.

15. Suppose the height of bracket, center to base, to be 15 inches; length and diameter of bearing, as in Problem 14; and that we use a separate bracket for the drum bearings, not connected with the pinion-shaft bearings. Design and draw such a bracket

16. Calculate, design, and draw all the parts for a machine similar to that of the text, from the following data:

Load on rope . . . . 4,000 pounds.
Speed of rope . . . . 175 feet per minute.
Length of rope to be reeled in . . . . 250 feet.

NOTE. Problems 12, 13 and 15 are comparatively simple, following closely the steps of the text in their solution.

Problem 16, likewise, is supposed to be worked out on the same lines as the text, but is wholly original in its nature, being based on entirely new data. It is not expected that this problem will be attempted except by well-advanced students who can give considerable time to working it out completely. It will be found, however, an excellent exercise in original and yet simple design.

REVIEW QUESTIONS

ON THE SUBJECT OF

HEATING AND VENTILATION.

PART I.

---

1. What advantage does indirect steam heating have over direct heating? What advantages over furnace heating?

2. What are the causes of heat loss from a building?

3. Why is hot water especially adapted to the warming of dwellings?

4. What proportion of carbonic acid gas is found in outdoor air under ordinary conditions?

5. A room in the N. E. corner of a building of fairly good construction is 18 feet square and 10 feet high; there are 5 single windows each 3 by 10 feet in size. The walls are of brick 12 inches in thickness. With an inside temperature of 70 degrees, what will be the heat loss per hour in zero weather?

6. State four important points to be noted in the care of a furnace.

7. A grammar school building, constructed in the most thorough manner, has 4 rooms, one in each corner, each being 30 ft. by 30 ft. and 14 ft. high, and seating 50 pupils. The walls are of wooden construction, and the windows make up $\frac{1}{3}$ of the total exposed surface. The basement and attic are warm. How many pounds of coal will be required per hour for both heating and ventilation in zero weather, if 8,000 B. T. U. are utilized from each pound of coal?

8. What two distinct types of furnaces are used? What are the distinguishing features?

9. What is meant by the efficiency of a furnace? What efficiencies are obtained in ordinary practice?

10. What are the principal parts of a furnace? State briefly the use of each.

11. A brick house of the best construction, 20 ft. by 40 ft., has 3 stories, each 10 feet high. The walls are 12 inches in thickness; and $\frac{1}{4}$ the total exposed wall is taken up by windows, which are double. The basement is warm, but the attic is cold. The house is to be warmed to 70 degrees when it is ten degrees below zero outside. How many square feet of grate surface will be required, assuming usual efficiencies of coal and furnace?

12. A high school is to be provided with tubular boilers. What H. P. will be required for warming and ventilation in zero weather if there are 600 occupants, and the heat loss through walls and windows is 1,500,000 B. T. U. per hour?

13. What are the three essential parts of any heating system?

14. Is direct-steam heating adapted to the warming of school-houses and hospitals? Give the reason for your answer.

15. The heat loss from a dwelling-house is 280,000 B. T. U. per hour. It is to be heated with direct steam by a type of sectional boiler in which the ratio of heating surface to grate surface is 28. What will be the most efficient rate of combustion, and how many square feet of grate surface will be required?

16. What is the use of a blow-off tank? Show by a sketch how the connections are made.

17. How are the sizes of single-pipe risers computed?

18. What weight of steam will be discharged per hour through a 6-inch pipe 300 feet long, with an initial pressure of 10 pounds, and a drop of $\frac{3}{4}$ pound in its entire length?

19. What is an air-valve? Upon what principles does it work?

20. What size of steam pipe will be required to discharge 2,400 pounds of steam per hour a distance of 900 feet, with an initial pressure of 60 pounds, and a drop in pressure of 5 pounds?

21. What objection is there to a single-pipe riser system? How is this sometimes overcome in large buildings?

22. What patterns of valves should be used for radiators? What conditions of construction must be observed in making the connections between the radiator and riser?

REVIEW QUESTIONS

ON THE SUBJECT OF

HEATING AND VENTILATION.

PART II.

---

1. How would you obtain the sizes of the cold-air and warm-air pipes connecting with indirect heaters in dwelling-house work?

2. What is an aspirating coil, and what is its use?

3. What efficiencies may be allowed for indirect heaters in schoolhouse work? How would you compute the size of an indirect heater for a room in a dwelling-house?

4. How is the size of a direct-indirect radiator computed?

5. A schoolroom on the third floor is to be supplied with 2,400 cubic feet of air per minute. What should be the area of the warm-air supply flue?

6. What is the chief objection to a mixing damper, and how may this be overcome?

7. How many square feet of indirect radiation will be required to warm and ventilate a schoolroom when it is 10 degrees below zero, if the heat loss through walls and windows is 42,000 B. T. U., and the air-supply 120,000 cubic feet per hour?

8. What is the difference in construction between a steam radiator and one designed for hot water? Can the steam radiator be used for hot water? State reasons for answer.

9. How may the piping in a hot-water system be arranged so that no air-valves will be required on the radiators?

10. What efficiency is commonly obtained from a direct hot-water radiator? How is this computed?

11. How should the pipes be graded in making the connections with indirect hot-water heaters? Where should the air-valve be placed?

12. Describe briefly one form of grease extractor.

13. What is the office of a pressure-reducing valve in an exhaust-steam heating system?

14. Upon what principle does a pump governor operate?

15. What type of pipe fittings should always be used in hot-water work?

16. How is the water of condensation returned to the boilers in exhaust steam heating?

17. How many cubic feet of air per hour will be discharged through a flue 2 feet by 3 feet, and 60 feet high, if the air in the flue has a temperature of 80 degrees and the outside air 60 degrees?

18. In a hot-water heating system, what causes the water to flow through the pipes and radiators? How does the height of the radiator above the boiler affect the flow?

19. What precaution should always be taken before starting a fire under a steam boiler?

20. What is the free opening in square feet through a register 24 inches by 48 inches?

21. Why are return pumps or return traps necessary in exhaust-steam heating plants?

22. What efficiency may be obtained from indirect hot-water radiators under usual conditions? What is the common method of computing indirect hot-water surface for dwelling-house work?

23. State briefly how a return trap operates.

24. What is the use of an expansion tank, and what should be its capacity?

25. Describe the action of one form of damper regulator.

26. What is the principal difference between a hot-water heater and a steam boiler? What type of heater is best adapted to the warming of dwelling-houses?

27. Upon what four conditions does the size of a pipe to supply any given radiator depend?

28. What is the use of an exhaust head?

29. A hospital ward requires 60,000 cubic feet of air per hour for ventilation, and the heat loss through walls and windows is 140,000 B. T. U. per hour. How many square feet of indirect steam radiation will be required in zero weather?

30. For what purpose is a back-pressure valve used?

REVIEW QUESTIONS

ON THE SUBJECT OF

HEATING AND VENTILATION.

PART III.

---

1. A main heater contains 1,040 square feet of heating surface made up of wrought-iron pipe, and is used in connection with a fan which delivers 528,000 cubic feet of air per hour. The heater is 20 pipes deep, and has a free area, between the pipes, of 11 square feet. If air is taken at zero, to what temperature will it be raised with steam at 5 pounds' pressure.

2. An 8-foot fan used for schoolhouse ventilation runs at a speed of 124 r. p. m. What horse-power of engine is required? What horse-power would be required if the fan were speeded up to 134.6 r. p. m.?

3. What precaution must be taken in connecting the radiators in tall buildings?

4. Give the size of heater from Table XXXI which will be required to raise 672,000 cubic feet of air per hour, from 10° below zero to 95°, with a steam pressure of 20 pounds. If the air-quantity is raised to 840,000 cubic feet per hour through the same heater, what will be the resulting temperature with all other conditions the same?

5. A fan running at 150 revolutions produces a pressure of $\frac{1}{2}$ ounce. If the speed is increased to 210 revolutions, what will be the resulting pressure?

6. A certain fan is delivering 12,000 cubic feet of air per minute, at a speed of 200 revolutions. It is desired to increase the amount to 18,000 cubic feet. What will be the required speed? If the original power required to run the fan was 4 H. P., what will be the final power due to the increased speed?

7. What size fan will be required to supply a schoolhouse

having 300 pupils, if each is to be provided with 3,000 cubic feet of air per hour? What speed of fan will be required, and what H. P. of engine?

8. What advantages has the plenum method of ventilation over the exhaust method?

9. A church is to be warmed and ventilated by means of a fan and heater. The air-supply is to be 300,000 cubic feet per hour. The heat loss through walls and windows is 200,000 B. T. U., when it is zero. How many square feet of heating surface will be required, and how many rows of pipe deep must the heater be, with steam at 5 pounds' pressure?

10. A schoolhouse requiring 600,000 cubic feet of air per hour is to be supplied with a cast-iron sectional heater of the pin type. How many square feet of radiating surface will be required to raise the air from 10° below zero to 70° above, with a steam pressure of 10 pounds?

11. What velocities of air-flow in the main duct and branches are commonly used in connection with a fan system?

12. A main heater is to be designed for use in connection with a fan. How many square feet of radiation will be required to warm 1,000,000 cubic feet of air per hour, from a temperature of 10° below zero to 70° above, with a steam pressure of 5 pounds and a velocity of 800 feet per minute between the pipes of the heater? How many rows of pipe deep must the heater be?

13. State in a brief manner the essential parts of a system of automatic temperature control.

14. What advantage does an indirect steam-heating system have over furnace heating in schoolhouse work?

15. The air in a restaurant kitchen is to be changed every 10 minutes by means of a disc fan. The room is 60 by 30 by 10 feet. Give size and speed of fan, and H. P. of motor.

16. What forms of heating are best adapted to the warming of apartment houses?

17. Give an approximate method for finding the heating surface required for greenhouses, both for steam and hot water.

18. How does the cost of electric heating compare with that by steam and hot water?

19. Describe briefly the construction of an electric heater, and the principle upon which it works.

20. A school building of 4 rooms is to be supplied with 600,000 cubic feet of air per hour. The heat loss from the building is 300,000 B. T. U. per hour in zero weather. Give the square feet of grate surface required in the furnaces.

21. What is a double-duct system as applied to forced-blast heating? What are its advantages?

22. What is a thermostat? Give the principles upon which two different kinds operate.

23. Describe briefly the connections to be made in a system of electric heating. In what way do they correspond to the piping in a system of steam heating?

24. State certain points to be observed in the introduction of air for the ventilation of churches and theaters.

25. A shop 100 feet long, 50 feet wide, and having 5 stories, each 10 feet high, is to be warmed by forced blast using steam at 80 pounds' pressure. The full amount of air passed through the heater is to be taken from out of doors, and the entire air of the building changed 3 times an hour. Give linear feet of 1-inch pipe required for heater, and size of fan and engine.

26. In what cases would you use a disc fan in preference to a blower?

27. The heat loss from a room is 12,000 B. T. U. per hour. How many kilowatt-hours will be required to furnish the necessary heat?

28. What is one of the best systems for the heating and ventilation of school buildings of large size?

29. What form of heating system would you recommend for a four-room school?

30. A factory 250 feet long by 50 feet wide has two stories, each 10 feet high. Each floor is to have a separate fan and heater, but the fans are to be driven by the same electric motor. The lower floor is to be supplied with air from out of doors, and is to have a complete change of air every 20 minutes. On the upper floor the air is to be returned to the heater from the room, and the entire contents is to pass through the heater every 20 minutes. Exhaust steam is to be used in both heaters. Give sizes of fans, heaters, and motor.

31. What is a telethermometer?

32. Describe two methods of moistening air.

REVIEW QUESTIONS

ON THE SUBJECT OF

PLUMBING.

PART I.

---

1. What causes a trap to "siphon," and in what three ways may it be prevented?

2. What size of soil pipe should be used for an ordinary-sized dwelling, and what pitch should be given to the horizontal portion?

3. What quantity of water per capita should be allowed in designing a sewerage system?

4. What form of cross-section of conduit gives a maximum velocity of flow to small quantities of sewage?

5. Describe the manner of making house connections with the main sewer.

6. Show by sketch the general method of running the waste and vent pipes in a dwelling house, and indicate the proper location of traps.

7. What are the two principal methods of sewage purification?

8. Describe the method of making up the joints in cast iron soil pipe.

9. In what way may the seal of a trap be broken besides siphonage?

10. What two tests are usually given to a system of plumbing? State the use of each.

11. What grade should be given to main sewers and branches?

12. Give two methods of flushing sewers.

13. Describe briefly some of the usual arrangements in the plumbing of hotels.

14. What is sewage farming? Describe the process briefly.

15. What is the difference between a "cup joint" and a "wipe joint?" State the conditions under which you would use each.

16. What is the use of a fresh-air inlet in connection with a soil pipe, and how is it connected?

17. Describe the "Smoke Test."

18. Should a trap or fixture be vented into a chimney? Give the reasons for your answer.

19. What material is commonly used for sewer pipes of different sizes?

20. When are underdrains required and how are they constructed?

21. What precautions should be taken in back venting traps?

22. What chemicals are commonly used in the precipitation of sewage?

23. How should you connect a lead pipe with a cast or wrought iron pipe?

24. Define the "separate" and "combined" systems of sewerage.

25. What is the principal point to be observed in the disposal of sewage? What precautions should be taken when it is discharged into a stream?

26. What is the sedimentation process?

27. What precautions should be taken in locating a cesspool? Describe briefly one form of construction.

28. Name some of the most important data to be obtained before laying out a system of sewerage.

29. In designing a system of surface drains what maximum conditions should be provided for?

30. Under what conditions may sub-surface irrigation be used to advantage?

REVIEW QUESTIONS

ON THE SUBJECT OF

PLUMBING.

PART II.

---

1. A hotel requires a water supply of 200 gallons of water per minute during a certain part of the day. It receives its supply from a reservoir 1,000 feet distant, and located 116 feet above the house tank, in the attic of the building. What size of wrought-iron pipe will be required to bring the water from the reservoir?

Ans. 3 inch.

2. What is the best kind of pipe for domestic water supply under ordinary conditions? When may it be objectionable?

3. A 1-inch pipe is to discharge 40 gallons of water per minute from a cistern placed directly above it. What must be the elevation if we assume the friction in the pipe and bends to be equivalent to 100 feet?

Ans. 111 feet.

4. A house tank is situated 15 feet above a faucet upon the fifth floor of the building. If the stories are 8 feet high, what will be the difference in pressure in pounds per square inch between this faucet and one in the basement?

Ans. 17.3 pounds.

5. Describe the action of an hydraulic ram.

6. A pump has a steam cylinder 6 inches in diameter and a water cylinder 5 inches in diameter. What steam pressure will be required to raise water to an elevation of 135 feet, neglecting friction in the pipe?

Ans. 40.3 pounds.

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Notes

Billing