ENGINE TESTS

EMBRACING THE RESULTS OF OVER ONE HUNDRED FEED-WATER TESTS AND OTHER INVESTIGATIONS ON VARIOUS KINDS OF STEAM ENGINES, CONDUCTED BY THE AUTHOR,

BY

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PREFACE.

The favor with which the author's book on "Boiler Tests," published in 1891, has been received, has led him to collect in similar form the data and results obtained on many of his engine tests. Some of the tables of results have appeared from time to time in mechanical journals and in pamphlets; also in the Transactions of the American Society of Mechanical Engineers, but a large part is now printed for the first time.

It is believed that the data here presented will prove of value to the engineering profession, to owners and intending purchasers of steam plants, and to any who are interested in the economical production of power. The book should be of special value to engineers advising intending purchasers of engines, on account of the assistance it will render in making a wise selection.

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ENGINE TESTS.

INTRODUCTION.

THE first work in the line of engine-testing with which the author was intimately connected was carried out at the Massachusetts Institute of Technology in the years 1875 to 1878. During this time he was engaged in conducting the experiments of the late George B. Dixwell on the use of superheated steam for motive power. The experiments consisted principally in investigations on a Corliss engine operated with both saturated and superheated steam; and they embraced the determination of the performance of the engine running under both of these conditions with different points of cut-off, and with different degrees of superheating, together with the determination of the effect of other changes in the conditions of operation. The engine in question, and the testing apparatus connected with it, formed the nucleus of a mechanical laboratory used in instructing the students of the Institute; and it was the first of the many steam laboratories which have been established in the colleges of this country. In the course of these investigations a board of experts, consisting of Chief Engineers Loring, Baker, and Farmer of the United States Navy, was appointed by the Bureau of Steam Engineering to examine the subject; and they conducted a series of tests on the same plant, and reported them to the Bureau. These trials were under the active charge of the author. The character of this work was such as to require from the very first the most reliable apparatus and the best methods and instruments. In preparing for it and carrying it on, the author had the best opportunity that could be afforded at that time for becoming educated in the practice of engine-testing, and the training thus acquired laid the foundation for much of the testing-work in which he has since been engaged.

This volume relates mainly to the engine tests which the author has conducted subsequent to the investigations in superheating just referred to. It has been his custom, whenever engaged upon any work relating to the performance of engines, to advocate the determination of their economy on the basis of feed-water consumption, rather than on that of the coal consumed. Whenever called upon simply for indicating, he has advocated the feed-water test, rather than to rely solely on the showing of the diagram. In a great many instances the feed-water test has thus been undertaken where it would have otherwise been omitted. By following this practice, and answering the calls which have come in the ordinary course, the author has personally obtained a considerable amount of data, which he believes to be of value to the engineering public, as well as to all who are interested in the use of power or development of economical engines, and therefore worthy of publication in permanent form, as here presented.

The author's work in engine-testing has embraced the indicating of engines for the simple purpose of valve-setting; indicating for the determination of the horse-power, or for determining the power used by various machines or departments of machinery which the engine drives; investigations upon the economy of different systems of operating engines; feed-water tests having for an object the improvement of the engine and the attainment of greater economy; and tests having in view the determination of the fulfilment or non-fulfilment of the terms of a contract guaranteeing a certain efficiency. These investigations and tests have been made on a great variety of engines, from the simple non-condensing engine with a single cylinder, to the triple expansion condensing engine; and they relate to many designs and to products of many builders. They have also covered widely varying conditions of service as to boiler pressure, cut-off, load, speed, and valve-setting, together with various conditions in regard to quality of steam, use of jackets, and the tightness of valves and pistons.

The tests reported in this volume have not been made with an organized attempt to obtain the performance of certain types and makes of engines; but they are the result of the investigations which the author has made in responding to the calls of his clients, whether it happened to be for one object or another, and whatever the class of engine or conditions of service. So far as given here they are confined mainly to stationary engines located in manufacturing establishments, and in most cases operating with a fairly uniform load. Nearly all the tests apply to engines which have a capacity of at least 100 horse-power, and they run from this size up to 1700 horse-power.

The first part of the volume is devoted to Feed-Water Tests, taking up first the simple engine, both condensing and non-condensing, and afterwards compound, and triple-expansion engines. The results of the test on each engine are given in a table by itself, and they are presented in such detail that all necessary information regarding the subject is at hand. In connection with the results is given in each case the dimensions and such information regarding the design of the engine, the conditions under which it was worked, and the character of the test, as is needed for a clear understanding of each case; and comments on the results are added where these are required. In all the engines the condition of the valves and pistons as to leakage is pointed out so far as this can be expressed in general terms. The engines selected were, as a rule, fairly tight; but in a few cases tests of leaking engines are introduced, either on account of the general interest attaching to them, or to show the wasteful effect of the leakage itself in some special instance. The tables of feed-water tests are followed by a chapter which presents a general review of the results, showing in brief the main points of information which the tests bring out. This chapter takes up the question of cylinder condensation, and analyzes the tests here reported, with the object of determining what the percentage of cylinder condensation under different conditions of running practically amounts to. The relative economy of simple, compound, and triple-expansion engines is considered, also the effects of superheating, jacketing, and piston speed, so far as the tests furnish data on these subjects.

The chapters on Feed-Water Tests are followed by one devoted to Valve-Setting and Effects produced by various conditions of operation, as illustrated by diagrams which the author has taken in his professional work. The final chapter relates to Steam-Pipe Diagrams.

In connection with the matter relating to each engine, whether feed-water tests, valve-setting, or otherwise, sample indicator diagrams are presented, usually reproduced three-fourths size, showing, so far as possible, average conditions. In the case of feed-water tests, diagrams are given from both ends of the cylinders; but in cases of valve-setting and miscellaneous diagrams, the diagrams shown are, as a rule, from only one end of the cylinder.

HOW THE FEED-WATER TESTS WERE CONDUCTED.

Before presenting the individual feed-water tests, and the review of the same as noted, it is proper to give a description of the methods employed in conducting them. This description is of a general character, applying rather to the usual practice of the author in conducting these and other engine tests than to each individual trial reported here. The principles, however, are applicable to the individual tests quite as much as to the tests as a whole. In the form thus presented, not only are the methods employed in conducting these tests described, but methods which should be adopted in the general work of testing, so far as they accord with the author's experience.

The two essential quantities to be determined in conducting a feed-water test are the weight of feed-water consumed, and the indicated horse-power developed in the cylinder.

MEASUREMENT OF THE FEED-WATER.

How the feed-water should be measured is a matter which depends somewhat upon the arrangement of the plant and the type of apparatus used for feeding the boilers, and this must in a great many cases be adapted to the local conditions. It is always best to weigh the water, and for this purpose to erect tanks and scales suitable for the work. There are instances, however, where it is impossible to do this, because it is necessary that water should be available under some head so as to fill the weighing-tank, which is generally elevated several feet above the pump; and there are cases where no water is at hand under the necessary head. A meter can be employed in such cases, or the water may be supplied through an orifice of known size arranged so as to be calibrated. In most cases, however, the system of measurement by weighing can be employed; and wherever it can be done, the method is to be followed in preference to all others. The simplest apparatus of this kind, having a capacity of say 6,000 lbs. of water per hour, consists of a small hogshead connected to the suction-pipe of the pump or injector, and an ordinary oil-barrel mounted on platform scales, the latter being supported by the hogshead on one side and by a suitable staging on the other side. The barrel is filled by means of a cold-water pipe leading from the source of supply, and this should be $1\frac{1}{2}''$ pipe for pressures not less than 25 lbs. The outlet valve of the barrel is attached to the side close to the bottom, and this should be at least $2\frac{1}{2}''$ in diameter for quick emptying. Where larger quantities of water are required, the barrel can be replaced by a hogshead, and two additional hogsheads can be coupled together for the lower reservoir. The capacity reached by this arrangement when the weighing hogshead is supplied through a $2\frac{1}{2}''$ valve under 25 lbs. pressure, and emptied through a $5''$ valve, is 15,000 lbs. of water per hour. For still larger capacity it is desirable to use rectangular tanks made for the purpose, and have the weighing-tank arranged so that the ends overhang the scales and the reservoir below, the outlet valve, consisting of a flap valve, covering an opening in the bottom 6″ or 8″ square. With rectangular tanks this system can be employed for any size of stationary engine ordinarily met with.

Where a meter is used for measurement care should be observed that water is fed through it at a uniform rate, and the instrument should be calibrated under conditions in every respect like those of the test. One method of calibrating a meter, which the author has found simple and fairly satisfactory, is to arrange the piping on the outlet side with a valve known to be tight, and provide, at a point between this valve and the meter, a tee with a branch having a flexible hose attached. A gauge is also connected to show the pressure. The valve leading to the hose need not be of the full size of the main line; for under the conditions of the calibration it discharges the water against a pressure much less than the working pressure, and, if the quantity is small, against practically no pressure. The hose is carried to two empty barrels located, preferably, outside of the building, where the water can be discharged without doing harm; and there two workmen are stationed to manipulate this end of the line. In making the calibration, the stop valve in the main is closed, and the branch valve leading to the hose is opened and so adjusted as to keep the pressure at the working point. The pump or other apparatus for feeding is at the same time adjusted to give the working-quantity of supply. This will be determined by timing the readings of the meter for, say, one minute. When the proper rate has been secured, the meter is read; and at that instant a signal is given to throw the hose into one of the barrels, the water during the preliminary operations having run to waste. When the first barrel is filled, the hose is quickly thrown into the second one; and while the second barrel is filling, the workmen tip the first one over bodily and empty it. When the second barrel is filled, the hose is quickly transferred back to the first, and immediately the second barrel is tipped over and emptied. This can be carried on as long as desired, depending upon the size of the meter and the thoroughness required. The last reading of the meter is taken when the last barrel becomes filled, accurate count having been made of the whole number. Subsequently the quantity of water contained in the two barrels is ascertained by weighing, and the rating of the meter is quickly determined by calculation.

When an engine is fitted with a surface condenser, the measurement of the feed-water can be somewhat simplified by collecting the water discharged by the air-pump. In this case the same kind of apparatus can be used for weighing; but the two tanks are reversed, the water being discharged first into the reservoir, and subsequently drawn into the weighing-tank, which is placed below it, and, after being measured, thrown away.

An approximate determination of the feed-water consumption can be made by water-glass measurement, assuming that the type of boiler is such that this method is applicable. The feed-water is shut off from the boilers for a half hour's time, or such period as is permissible, and the rate observed at which the water disappears in the gauge glasses. Subsequently the volume consumed in the observed time is computed from the known dimensions of the space occupied, and from this the weight of the water which has been evaporated. If the water line is effected to any extent by the condition of the fires, it is necessary in making these measurements to observe great care that the conditions of the fires are the same at the end of such a trial as at the beginning. In some boilers the increased activity of the fire causes the water line to rise, while the deadening of the fire has the opposite effect. With the damper wide open, and the fire barred up and in an open or free condition, there is great activity of the fire; while with the damper closed and new coal applied, there is, for the time being, a very marked reduction in the intensity of the heat. The position of the damper and the thickness and general characteristics of the fire should be the same at one time as at the other. It is best to observe this precaution in all cases, even when there is no sensible effect produced by these changes in the fire. It is also necessary that the gauge glass and the connections leading from the water column to the boiler should be clear; a condition which can be secured by blowing them out a short time (say one hour) previous to the trial. When these are obstructed a noticeable effect is produced upon the height of water shown in the glass.

It is also necessary to be assured of the tightness of the feed valves and check valves concerned, that none of the water measured escapes by leakage.

The author has, in some instances, been able to obtain a measurement of the feed-water by drawing it from the tank which is often provided in mills for fire purposes and other emergencies, and which is not regularly in use. This tank being generally of uniform cross-section, the water it contains is subject to accurate measurement. When such means is used for measuring feed-water, it is absolutely necessary to be assured that the water is not in the meantime used elsewhere than for the test, and that the valves connected with the system of regular supply do not leak.

The orifice method of measurement is one which the author has found useful in a number of cases. One instance is that of a 1000 horse-power compound condensing engine, in which the water from the hot well was used in the customary manner for feeding. A test was required to determine the coal consumption of the plant per indicated horse-power per hour, under as nearly as possible working practice. The quantity of feed-water used was desired; but it must be obtained without changing, any more than necessary, the working conditions. The hot well overflow pipe was too near the level of the suction pipe of the pump to permit of using the ordinary process of weighing; consequently, resort was had to orifice measurement. The feed-tank was supplied from the overflow of the hot well through a 4-in. pipe. The elbow on this pipe, next to the tank, was replaced by a 4-in. tee, one branch of which looked down and the other looked up. To the lower branch a pair of flanges was attached, in which was secured a horizontal plate having a hole $1\frac{3}{8}$-in. diameter; and this served as the orifice. The plate was horizontal, and the discharge from it was therefore directly downward into the tank. The upper branch of the tee contained a stand-pipe 3 ft. high; and to this pipe was attached a glass for showing the height of the water inside, the same being graduated in inches measured from the face of the orifice plate. A valve in the 4-in. supply-pipe served to regulate the height of water in the stand-pipe, and consequently the amount passing through the orifice. During the progress of the test, the head of water in the stand-pipe was maintained at such a point as to supply the required quantity of water; and a careful record was kept of the height indicated in the gauge glass. Subsequently, when the pump was stopped, the orifice was calibrated by observing the quantity of water which flowed into the tank under conditions of the average head, the contents being previously known.

Whatever method is pursued in determining the quantity of water pumped into the boilers on a feed-water test, a determination should be made of the leakage of the boilers, stop valves, safety valves, steam-pipe joints, blow-off cocks, etc., concerned in the plant, so as to correct for such leakage, and charge the engine with only that quantity of feed-water which actually passes into it as steam through the throttle valve. To accomplish this object a leakage trial should be made immediately after the engine is shut down at the close of the test, the pressure being maintained in the boilers at a point nearly, if not quite, as high as the working pressure, and no change made in the stop valves, etc., concerned, or in the drips or other avenues of escape. Observations should then be made of the height of water in the gauge glasses, taking readings at intervals of ten minutes for a period of one hour, or until successive differences in the ten-minute periods show a uniform rate of leakage. By calculating the weight of water corresponding to the volume lost, as found by this test, which can be done knowing the dimensions of the boilers, the desired correction for leakage is determined. To make this test reliable it is necessary, of course, that the throttle valve at the engine should be tight. The tightness of the throttle valve can readily be determined by observing whether steam blows from the open indicator-cock of the cylinder when the steam valve is wide open, this observation being made at the end of the cylinder which is taking steam. If it leaks, allowance should be made for this leakage. If there is considerable piping, and it pitches toward the throttle valve, it is also necessary that allowance be made for the steam which condenses in the pipes and collects at the throttle valve. In some cases it will be seen that the conditions may be such that the determination of the correction for leakage may be a difficult matter; but it is a subject which ought always to receive attention when the object of the test, as in the present instances, is to determine the quantity of steam used by the engine alone.

Whatever method of feed-water measurement is employed, it is necessary that the height of water in the gauge glass should be the same at the end of the allotted time of the test as at the beginning. It is important also that the condition of the fire should be the same at one time as at the other, because, as elsewhere noted, the height of the water may be more or less affected. For example, if the test begins just before firing and with the damper closed, or nearly closed, it should also end just before firing and with the damper likewise closed. It is better to overrun the allotted time or even to cut it short, and have these conditions right, than to overlook them in the desire to make the duration of the trial a predetermined number of hours to the exact minute. If the height of the water at the end of the test is different from what it was at the beginning, the necessary correction estimated from the corresponding volume of water is applied to the quantity weighed. This correction is determined with sufficient accuracy, in most cases, by calculation from the known exterior measurements of the boiler.

INDICATING.

It is unnecessary for the purposes of this volume to go into a description of steam-engine indicators, for the books on the subject of the indicator furnish an ample amount of information of this character. It will suffice to say that for most of the tests here reported the instruments used were either of the Tabor or the Crosby pattern, or both. The methods of applying the instruments, however, the means of driving them and manner of using them, also the methods employed in calibrating the springs, require notice.

In nearly all the indicator work on the Corliss, and similar types of slow-speed engines, the driving-rig has been some form of pantagraph, and in the large majority of cases, that form known as the "lazy-tongs," working horizontally and operated from the cross-head. The fixed end of the lazy-tongs has generally been applied to one of two wooden posts, attached to a base-board, which in turn is fastened to the floor. The second post, suitably located with reference to the first, is used for the support of a carrier-pulley, and both posts are securely fixed in position by means of three wooden braces fastened to the floor. This method of attaching the lazy-tongs has the advantage of rigidity, which is so essential to a correctly driven indicator; and the use of the carrier-pulley enables the driving-cord to be always led off in a line parallel to the direction of motion of the cross-head, whatever the position of the indicators with reference to the cord-pin of the lazy-tongs. The construction of a stand for supporting the lazy-tongs in this manner may be considered crude and clumsy for permanent use; but the author has often found permanent rigs defective from improper design or insecurity, due to gradual wear, and substituted the one described. Being made throughout of wood, it is a device which can be quickly put together, even where there is no carpenter-shop at hand and little material. As it is built in such form as to easily and positively accomplish the desired ends, it has been found most useful.

For a driving-cord, a strong braided linen fish-line having an unbraided core is used, extending a little beyond the carrier-pulley; and from this point to the indicators, pieces of annealed brass wire are used, about No. 25 B.W.G. ($\frac{1}{50}''$ in diameter). For a single cylinder two cords are thus brought into use leading from the same initial point. In the case of tandem cylinders, either four independent cords are used, or two independent cords, each having branch loops at appropriate points for connecting to the two instruments. In some cases the cords have been displaced by a light wooden rod driven by the cord pin of the lazy-tongs, and moving on guides attached to the cylinder, the direction of motion being parallel to it. A screw fastened to the rod at the proper place serves to carry the motion to the cord attached to the indicator. The use of the rod in place of the cords is especially applicable to tandem engines.

For high-speed engines the driving apparatus is some form of lever and sector, the shaft on which the lever is mounted being in many cases supported by a stand bolted to the frame of the engine. In some engines of the high-speed compound class the driving motion is derived from an eccentric fastened to the main shaft, the motion being carried from this point to the cylinder through a connecting-rod and bell-crank lever. In these cases an independent motion is used for each cylinder.

It has been the custom in making these tests to employ two indicators for each cylinder, attached as close as possible to the end of the cylinder, using the half-inch connection, a right-angle elbow, and the indicator-cock furnished with the instrument. Sometimes a straight-way valve is placed below the indicator-cock for facility in moving the same without shutting down the engine. The objections to long pipes connected by a three-way cock in the center, consist in the increased friction of the steam in passing through the greater length of the pipe with the increased number of bends, and in the collection of water in the long horizontal cavity which is thus brought into play. If two indicators are not available for an engine test, it seems better to use one instrument, and transfer it from one end to the other, than to employ the three-way cock and have the instrument fixed at the central point with the long connections.

On many of these tests "prepared" indicator paper has been used, the instrument being fitted with metallic marking-points. These marking-points are made of brass wire of suitable size, which is reduced in diameter near the marking end to about $\frac{1}{32}''$, so that by the use of a small hand-vise, such as watch-makers employ, and an oil-stone, the marking-point is readily kept in shape for tracing fine lines. The use of metallic paper is much to be preferred, as a matter of convenience, to plain sheets with the ordinary lead-pencil point, inasmuch as the sharpening of the metal point requires much the less attention.

The driving mechanism for the work referred to here has in no case been any form of reducing-wheel.

GENERAL METHOD OF CARRYING ON THE FEED-WATER TEST.

The testing apparatus being in readiness, and the engine working with the desired load, the height of water in the gauge glasses is observed, the time taken, and the position of the water in the reservoir of the weighing apparatus observed. Thereafter all the water fed is weighed. At the expiration of the time determined upon, the water in the gauge glasses and in the lower reservoir is brought to the starting-point, and the exact time observed. During the progress of the test indicator diagrams are taken every thirty minutes, and sometimes every twenty minutes, and at the same time the gauges are observed and the number of revolutions per minute counted. If the steam is superheated, the temperature of the steam is observed; and if calorimeter tests are made, these are either continuous or made at convenient intervals. Where special accuracy is required, the atmospheric pressure is determined by observation of a barometer at some time during the progress of the test. For this, it is sufficient for all practical purposes to rely upon the record of the United States signal service at the nearest station. When the test is made in a factory running ten hours per day, say five hours in the forenoon and five hours in the afternoon, the record in some instances embraces the whole period from the time the engine starts until the time of stopping. In that case the initial and final readings of the water glasses are taken just before the engine starts, and just after it stops. The duration is taken from the time the engine attains its working speed till the time the throttle valve is closed; and no further account is taken of the power developed while the engine is reaching its speed after first starting. In that case, the first set of diagrams is taken not less than five minutes after the load is put on; and the assumption is made that the loss of steam from condensation and drips during the time the engine is first starting and attaining its working speed counterbalances the deficiency of load between the time when the speed is attained and the working-load is actually applied. In factory work, the interval of time between the attainment of the working speed and the application of the full load is usually less than three minutes.

In taking diagrams from an engine with the object of determining its power, it is not desirable to limit the diagram to a single revolution. The marking-point of the indicator should be applied long enough to obtain four or five diagrams, corresponding to that number of successive revolutions, in order that the effect which the fluctuations in the governing mechanism has upon the diagrams may be provided for. In working up the diagrams, then, the mean pressure is obtained for the average diagram, and not for any single one. By pursuing this method, the average power which is determined relates to several times as many diagrams as it would if it were confined to a single revolution in each case. Instances are frequently met where the fluctuations in the cut-off for half a dozen successive diagrams varies from 2 to 5 per cent of the length of the stroke, and in such cases this matter is of considerable importance. As a convenience in working up the diagrams, a good plan to follow is to go over each one with a pencil, and trace with dotted lines the diagram which represents an average of those made by the indicator, and in the subsequent calculations to use this dotted diagram. When a load is extremely fluctuating, this system should be carried further. The period of taking the diagram should extend over at least a full minute, though it is unnecessary to make it a continuous diagram for this length of time. The marking-point can be preferably applied for three or four revolutions at the beginning of, say every ten seconds of a minute, and in that way the record applies to some twenty revolutions spread over the full period. Having these diagrams now on the same card, an average line can be dotted in by hand, using the best judgment after examining the appearance of the various diagrams and their location.

The same method is usefully applied in tests of electric railway engines. Indeed, except by some system of this kind, no fair idea of the indicated horse-power can be obtained, and no good comparison can be made between the indicated horse-power and the electrical horse-power. In these engines it is best to make the interval between the sets of diagrams thus obtained not more than ten or fifteen minutes. It should be arranged to give a signal every ten seconds while the operation is going on, so that all the indicators may be worked together for the three or four revolutions desired. Likewise, on the same signal corresponding readings are taken of the electrical instruments. This is continued until the period of time covered is two or more minutes. The diagrams being all taken on the same card, without unhooking the indicators, the means is at hand for obtaining an average for the whole period, as before pointed out.

LEAKAGE TESTS OF VALVES AND PISTONS.

The determination of the condition of an engine as to the tightness of the valves and pistons has nothing to do with the work of making a feed-water test, or of correctly ascertaining the results. When, however, it comes to analyzing the results, and ascertaining whether the engine is working with a proper degree of economy, and if not, the reasons for the waste, it is of the utmost importance that the matter of leakage should be investigated. It is always desirable, therefore, when a feed-water test is conducted, to supplement it by an inspection of the valves and pistons having this object in view. This inspection must be made when the engine is at rest. The conditions which surround the internal working parts of an engine at rest are entirely different from those of the engine in motion, and for this reason it is held by some that an examination of leakage under these circumstances gives little information which can be applied to working conditions. Those who take this view hold that under conditions of motion the quantity of leakage is reduced, and it might happen that the leakage in motion would be altogether insignificant, although very serious at rest. The author takes the ground that the only course open in this matter is to make the examination when the engine is at rest, for certainly no thorough inspection can be made when it is in motion. If it is found that there is practically no leakage at rest, it seems reasonable to conclude that the engine is tight in motion. If, however, there is leakage at rest, we can certainly say that there is a probability of leakage in motion, although it may not be possible to judge of its degree.

The leakage tests here referred to are not quantitative; that is, they do not determine the exact amount of leakage, but rather the fact as to whether leakage does or does not exist. They are intended simply to give the observer a fair idea as to the general condition of the engine.

Turning to the methods employed in testing for leakage, the steam-valves are readily disposed of. In a Corliss engine, it is necessary simply to close the two admission valves, open the two indicator-cocks, and with the starting-bar move the exhaust valves first one way and then the other, the throttle valve being open, and a full pressure of steam being admitted into the chest. When the starting-bar is moved so as to close the exhaust valve at the head of the cylinder, any leakage of steam through the steam valve at that end will be made to escape at the indicator-cock, and thus become visible. Likewise when the starting-bar is moved so as to close the exhaust valve at the crank end, the steam which leaks through the crank-end admission valve will show itself at the open cock. In making these movements of the starting-bar, care is taken that the steam valves are held unhooked. The quantity of leakage is judged by the force of the current of steam blowing out of the cock. If the valves are tight there is simply a breath of steam, or an entire absence of vapor. If they leak badly, the current will blow out of the indicator-cock with much force and noise, and rise to a height of several feet.

In testing the exhaust valves and pistons for leakage, the best method is to block the fly-wheel in such a position that the engine is taking steam with the piston at a short distance from the end of the stroke, open the throttle valve, and observe what blows through. It is well to try this if possible with the piston at different points. If the end of the exhaust pipe is open to view, as would be the case with a non-condensing engine, the steam which leaks through can be observed at the open outlet. This can also be done in the case of a condensing engine where there is a branch exhaust pipe leading to the atmosphere. Where the engine is condensing, and no such branch is provided, and there is no other opening in the exhaust pipe in front of the condenser, a pretty good idea can be obtained of the general facts by observing the amount which the condenser is heated by the steam which leaks.

With the piston in any given position in a Corliss engine, the leakage on such tests embraces the leakage of one exhaust valve, one steam valve, and the piston. To investigate the leakage of the other steam valve and the other exhaust valve, the test must be made with the piston taking steam on the opposite stroke. In either case, if the previous inspection of the two steam valves shows them to be leaking, this fact must be considered in drawing conclusions as to the leakage of the piston and exhaust valves.

There is another method of testing the leakage of piston and exhaust valves, namely, the "time method." The fly-wheel is blocked, as before, with the piston at some distance from the beginning of the stroke, the throttle valve is opened, and steam is admitted at full pressure until the cylinder is thoroughly warmed. Then the throttle valve is shut, and the length of time is observed which is required for the steam to escape through the leaking openings. To conduct the test properly, an indicator is attached to the cylinder at the end containing the steam, and a mark is made on a blank card at intervals of, say, one-quarter of a minute from the time the throttle valve is closed; and by this means the rate of fall of pressure and escape of steam is recorded. This test, like the others, is qualitative, and not quantitative. The relative condition of the engine determined from results of the time tests must be judged by comparing with other cases where known conditions of excellence prevailed. In a leaking engine the fall of pressure on a test of this kind is very rapid. If the leakage is serious, the first observation, after a quarter minute interval, might show a reduction of pressure covering nearly the whole range down to the atmosphere. On the contrary, if the engine is tight, the reduction of pressure to the atmosphere would require from five to ten minutes time. The author finds that the pressure will not fall as a rule more than fifty per cent at the expiration of one minute from the time of shutting the throttle valve, if the engine is fairly tight.

If, on leakage tests with the blocked engine, it is found that the piston and the two valves leak, whichever stroke the piston is occupying, the piston leakage can be eliminated by disconnecting the valve rods in such a way as to open both steam valves and close both exhaust valves. When this is done, the resulting leakage which is observed applies to the exhaust valves alone.

The leakage of a piston can always be inspected by removing the cylinder head and applying a pressure behind the piston. The leakage then appears at the open end of the cylinder. On large engines the operation of taking off a cylinder head is attended with considerable labor. The methods which have been described can be brought into use with great facility and save this labor, to say nothing of saving time.

The blocking of the engine which these tests require is a thing which should not be undertaken in any careless manner. In most cases the masonry foundation of the engine is so arranged that a piece of timber can be placed between the spokes of the wheel, and the two ends laid upon or against the foundation, the strain of a spoke being brought to bear upon the middle of the timber. This timber should be of ample size, say, a 12 in. or 14 in. stick of hard pine for an engine of 1000 horse-power, the points of support at the two ends being not over 8 ft. apart. The position of the arm should be brought as nearly as possible to the proper point before the block is introduced, the leeway being filled in not by subsequently moving the engine, but by the introduction of wooden filling-pieces and wedges. In the case of an engine having a shaft with two cranks and a solid bed beneath each one, the engine can be readily blocked in certain positions by standing a piece of timber endwise, reaching from the end of the crank to the floor or bed, or by putting in a number of wooden blocks laid flat, and building up to the desired height. Here, again, the crank-pin should be brought to the required position before the blocks are put in, and filling-pieces should be applied to make up the leeway, rather than move the engine and run the risk of injury by bringing up solid against the blocks.

Leakage tests of the valve in the case of single-valve engines cannot be made as satisfactorily as those in four-valve engines, for if the valve leaks excessively it is difficult to locate by these methods the exact place of the leak. The best that can be done is to place the valve on its center covering both ports, and try it under a full steam pressure. The same course can be followed in testing the piston as that described for the four-valve engines. In a leaking engine of this type, it is usually necessary to test the piston with the cylinder head removed before the investigation is complete.

It is needless to call attention, in more than a passing way, to the test of piston leakage in an engine which is single-acting. In a Westinghouse engine, for example, the leakage of the piston is revealed by simply swinging off the cover of the crank case, and observing at once what escapes from the periphery of the piston, the engine being blocked and steam pressure admitted into the cylinder.

The foregoing remarks on the subject of leakage apply to simple engines. In the case of compound engines the work is to some extent simplified. For example, in testing the leakage of the high-pressure exhaust valves and piston, the escape of steam is observed by opening the indicator-cock on the end of the low-pressure cylinder which is taking steam, and observing what blows through. Again, in testing the low-pressure exhaust valves and piston by the time method, steam is admitted into the receiver until the desired pressure is reached, then, after the cylinder has been thoroughly warmed, and the supply shut off, the drop in pressure is observed by reading the receiver gauge and keeping a record of this. A similar course is followed in testing the leakage of triple-expansion engines.

CALIBRATION OF INSTRUMENTS.

For a satisfactory comparison of the steam-pipe gauge with the initial pressure shown by the diagram, the best plan is to compare the gauge and the indicator without changing them from their working positions. This can be done at the same time that the leakage tests are in progress, as, for example, when testing the piston and exhaust valves, the fly-wheel being blocked, and the throttle valve and admission valve set wide open. By taking the reading of the steam gauge and that of the indicator at the same time (the latter being done by opening the indicator-cock, then drawing a short line on the blank card which has been applied for the purpose), not only will the error of the gauge itself be allowed for, but also the error produced by the head of water contained in the gauge pipe, if any such error exists. This comparison alone is sufficient to establish the difference in pressure between that in the main pipe (or in the boiler to which the gauge is attached) and the initial pressure in the cylinder of the working-engine, whether the gauge, or indicator, or both, are in themselves correct or in error. The gauge is then calibrated by reference to a standard, and the accuracy of the indicator is established at the particular pressure used. This single calibration is considered in many cases sufficient for determining the correct scale of the indicator in question.

The most satisfactory method of determining the correctness of the gauge, is to remove it from its place, and attach it to a dead weight testing-apparatus, of the form sold by the steam-gauge manufacturers, in which the pressure is produced by sealed weights resting upon the top of a vertical plunger of known area, the pressure being transmitted to the gauge through the medium of oil or glycerine. The convenience of this method and the portability of the apparatus, together with its extreme reliability, place it ahead of all other systems for calibrating gauges. Having made the calibration, the indication of the gauge in its working position must be corrected for the head of water in the supply-pipe of the gauge, if any exists, whether it be to increase the indication or to reduce it.

The calibration of the indicator springs used on the tests reported in this volume has in many cases been carried on by testing them under the action of dead weights, and correcting the result thus found by a percentage of allowance for the reduced tension caused by the heat of the steam in which they ordinarily work. The author's testing-apparatus consists of a scale-beam mounted on knife edges, on one end of which the weights are suspended. The movement of the beam at the other end is transmitted upward by means of a vertical adjustable rod extending to the under side of the indicator piston. The tests are made with the highest pressure to which the springs are subjected, and from this point down to the atmosphere at uniform reductions. The apparatus is operated so as to get an average reading, whether the pressure is going up or going down. This is done each time by pushing the scale-beam down as far as it will go, and drawing a line on the indicator-card, then, without changing the weight, pushing the same upward as far as it will go, and marking another line. When the lines are measured, the mean of the two is selected as the true indication. The springs are in some cases compared under different pressures with a correct steam gauge, admitting the steam directly into the indicator, and subjecting it as near as possible to its working conditions of temperature. In making calibrations under steam, difficulties are often experienced in obtaining satisfactory indications, owing to the friction of the piston of the indicator under the action of the continuously applied pressure. This is overcome, provided the pressure is maintained at a constant point, by drawing two lines with the instrument, one when the pencil-arm is pushed down with the finger as far as it will go, and the second when the arm is pushed up as far as it will go, the true indication then being taken as the mean of the two. When a set of indicator springs has once been calibrated, and their exact scales obtained, the dead-weight apparatus above referred to furnishes a much more satisfactory means for future determinations, and for showing the changes in the scale which may take place under continued use, than the steam-testing apparatus, for the reason of its greater simplicity and ease of operation, together with its freedom from the particular errors noted.

In calibrating the springs for pressures below the atmosphere, the dead-weight apparatus referred to is not applicable, and resort has been had in these tests to comparison with a mercury gauge, or with a standard vacuum gauge, the former being preferred. In making these comparisons in the shop or laboratory it is necessary to obtain a vacuum by the use of some form of pump or exhauster, and this often proves an inconvenience. For this reason the author has been in the habit of making them in the engine room where the indicators are being used, and where a vacuum is obtained by connecting the testing-apparatus with the condenser. All that is required for apparatus is the connection to the condenser, a tee for the attachment of the indicator-cock, and a mercury gauge applied to one end of the tee. With this apparatus the spring can be calibrated down to the lowest pressure to which it is subjected. It is desirable to make the calibration of an indicator spring that is used for pressures below the atmosphere under conditions of vacuum as well as under conditions of pressure; for the fact that the spring is correct when subjected to compression, as it is when a pressure is applied to it, furnishes no positive assurance that it is correct under tension, as it is when it is subjected to a vacuum.

It is of no little importance that the scale of the spring should be known within reasonable limits of error; for upon this knowledge depends the whole accuracy of the indicator work, and consequently of all the results of the tests depending upon it.

MANNER OF WORKING UP THE TESTS.

The results of the feed-water tests are computed from the hourly consumption of feed-water corrected for the leakage of the boilers, pipes, and connections, as explained, and the indicated horse-power developed. The "steam accounted for by the indicator" is determined from measurements of the diagrams and computations based thereon.

The indicator cards relating to the tests reported here have, as a rule, been measured by a polar planimeter. The average obtained by going over the line of the diagram at least twice is the reading taken.

The mean effective pressure is determined by dividing the scale of the spring by the length of the diagram expressed in inches and decimals of an inch, and multiplying the quotient by the area in square inches. The length of the diagrams, which is nearly constant, is found by selecting, say three sets out of every ten taken on the test, and obtaining the average length from those three. The horse-power is computed by multiplying the "horse-power constant" for the cylinder under consideration by the speed in revolutions per minute, and by the mean effective pressure. The horse-power constant is the power developed in the cylinder, assuming one pound mean effective pressure and a speed of one revolution per minute. It is obtained by multiplying the mean of the areas of the two sides of the piston in square inches by twice the length of the stroke in feet, and dividing the product by 33,000. The mean effective pressure used is the mean of the two measurements obtained at the two ends of the cylinder. In the detail tables, giving the data and the results of the tests here reported, the horse-power constant for each cylinder is given; and the figures of indicated horse-power in any case are the result of multiplication of this constant, the revolutions given per minute, and the mean effective pressure. For example, in the case of Engine No. 1, which has a single cylinder 23″ diameter, 5′ stroke, with one piston-rod $3\frac{1}{4}''$ in diameter, the horse-power constant is .1247, the speed 74.7 r.p.m., and the m.e.p. 33.08 lbs. The indicated horse-power, viz., 305.2, is the product of these three quantities.

The water per indicated horse-power per hour is found by simply dividing the hourly consumption of water by the indicated horse-power. In the example referred to, the hourly consumption being 8477 lbs., the feed-water per I.H.P. per hour is 8477 divided by 305.2 equals 27.77 lbs.

The method of determining the quantity of steam "accounted for by the indicator" consists in measuring the diagrams for the necessary data, and using the formula

$$\frac{13{,}750}{\text{m.e.p.}}\left[(c+e)\,Wx-(h+e)\,Wh\right]$$

in which "m.e.p." is the mean effective pressure measured from the diagrams as pointed out; "c" the proportion of the forward stroke completed either at cut-off or release, according as the determination is made at one point or another; "h" the proportion of the return stroke uncompleted at compression; "e" the proportion of the clearance space; "Wx" the weight of one cubic foot of steam at the cut-off or release pressure; and "Wh" the weight of one cubic foot of steam at the compression pressure.

The points on the diagram where these measurements are taken are illustrated in the sample diagram Fig. 1 given below. These points are located as follows: The point of cut-off is marked at the beginning of the expansion line after the steam valve has completely closed. It is at the point where the curve changes its direction from that due to the gradually closing steam valve to that of the expansion line. The point of release is marked at the end of the expansion line just before the curve begins to drop, due to the opening of the exhaust valve. Likewise the compression point is fixed at the beginning of the true compression line, or at the end of the curve formed by the gradually closing exhaust valve. The principle followed is to locate the points of cut-off and release so as to account for all the steam present in the cylinder at the instant the steam valve is closed, and for all the expanded steam present just before the exhaust valve opens. The compression point is located with the idea of obtaining a measurement of all the exhaust steam which is retained in the cylinder at the moment the exhaust valve is closed.

In all these tests the computations are made both at the cut-off point and the release point. It is desirable to do this, because there is often a considerable difference between the two quantities; and where there is such a difference, much more can be learned from the examination of the steam accounted for at cut-off than that accounted for at release. The difference in the work done during expansion is not proportional to the difference in the steam accounted for; and, consequently, the actual loss of economy due to cylinder condensation and leakage is more closely measured by the percentage which is accounted for at cut-off than by the percentage accounted for at release. Between the two, if only one computation is to be made, it is better to use the cut-off point than the release point.

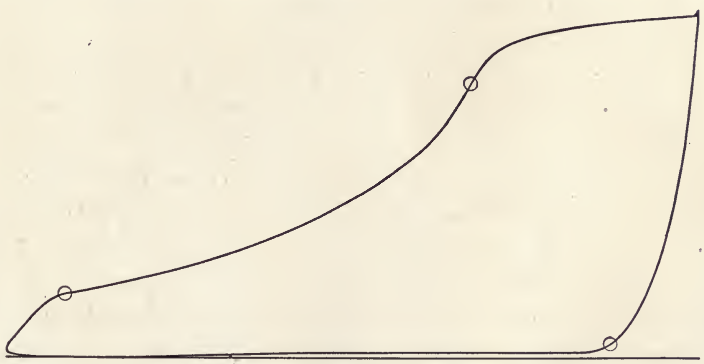


Fig. 1.

The proportions "c" and "h" are found by measuring the entire length of the diagram, first erecting perpendiculars at the extreme points, and then measuring the length up to the point marked, dividing one by the other, and ascertaining the resulting proportion expressed in a decimal. The proportion "e" for the clearance may be found either by measurement of the clearance spaces from drawings of the cylinder and valves or from actual test. The latter is to be preferred; for drawings, however correct in themselves, do not show the exact measurements of the material, especially of the ports and passages which are in the state of rough casting.

To measure the clearance by actual test, the engine is carefully set on the centre, with the piston at the end where the measurement is to be taken. Assuming, for example, a Corliss engine, the best method to pursue is to remove the steam-valve so as to have access to the whole steam-port, and then fill up the

P I a

clearance space with water which is poured into the open port through a funnel. The water is drawn from a receptacle containing a sufficient quantity, and this has previously been measured. When the whole space, including the port, is completely filled, the quantity left is measured, and the difference shows the amount that has been poured in. The measurement can be most easily made by weighing the water, and the corresponding volume determined by calculation. The proportion required in the formula is the volume in cubic inches thus found, divided by the volume of the piston displacement, also in cubic inches, and the result expressed as a decimal.

The only difficulty which arises in measuring the clearance in this way is that occurring when the exhaust valve and piston are not tight, so that, as the water is poured in, it flows away and is lost. If the leakage is serious, no satisfactory measurement can be made, and it is better to depend upon the volume calculated from the drawing. If not too serious, however, an allowance can be made by carefully observing the length of time consumed in pouring in the water; then, after a portion of the water has leaked out, fill up the space again, taking the time and measuring the quantity thus added, determining in this way the rate at which the leakage occurs. Data will thus be obtained for the desired correction.

In the tests here reported the clearance has not, as a rule, been determined by actual measurement in the manner noted, nor even in all cases by the calculation from the drawing. In cases where the proportion of clearance is assumed, the assumption is based on the known clearance of similar classes of engines, determined either by water measurement or calculation. The effect which a small error in the clearance may have upon the result of the computation of steam accounted for is not of a serious nature, unless it is a case where the cut-off is very short. For example, if the steam accounted for with a clearance of five per cent comes out $\frac{75}{100}$ of the feed-water consumption, the result with a clearance of 4 % would be $\frac{73}{100}$, changing the proportion about $\frac{2}{100}$ at cut-off and much less at release.

In compound and other multiple expansion engines the same formula for determining the steam accounted for by the indicator is used as that given above, but it must be adapted to the type of engine. The only change required in the formula is in the mean effective pressure. Here the quantity used when determining the steam accounted for in any given cylinder is the collective mean effective pressure of all the cylinders assumed to be referred to the one under consideration. In the case of the high-pressure cylinder of a compound engine the quantity to be used is the sum of the mean effective of the H. P. cylinder and a quantity representing the m.e.p. of the low-pressure cylinder referred to the high-pressure cylinder; that is, the mean effective pressure in the low-pressure cylinder multiplied by the ratio of the volume of the L. P. cylinder to the H. P. cylinder. If the ratio is 4 to 1, the m.e.p. of the low-pressure cylinder is to be multiplied by four to determine the quantity desired. Likewise the quantity to be used for computing the steam accounted for in the low-pressure cylinder is the sum of the mean effective pressure in that cylinder, and the mean effective pressure in the H. P. cylinder divided by the ratio of the volume of the L. P. cylinder to the H. P. cylinder. In the instance given it would be the mean effective in the H. P. cylinder divided by 4. In a triple expansion engine the mean effective pressure to be used for computing the steam accounted for in the L. P. cylinder is the sum of the mean effective pressure of that cylinder; that of the m. e. p. of the H. P. cylinder divided by the ratio of volume of the L. P. cylinder to that of the H. P. cylinder, and the m.e.p. of the intermediate cylinder divided by the ratio of the volume of the L. P. cylinder to that of the intermediate cylinder. Likewise the quantity to be used for the intermediate cylinder is the sum of three quantities, namely, the m.e.p. of the intermediate cylinder, the m.e.p. of the H. P. cylinder divided by the ratio of the volume of the intermediate cylinder to that of the H. P. cylinder, and the m.e.p. of the L. P. cylinder multiplied by the ratio of the volume of the L. P. cylinder to that of the intermediate cylinder.

As an example of the proper method of computing the equivalent mean effective pressure referred to either cylinder of a compound engine, we may take the case of Engine No. 32, in which the ratio of volumes of the cylinders is as 1 to 3.43, and the mean effective pressure in the two cylinders 41.26 lbs. and 9.28 lbs. respectively. The equivalent m.e.p. to be used in computing the steam accounted for in the H. P. cylinder is $46.26 + (9.28 \times 3.43) = 41.26 + 31.83 = 73.09$. For the low-pressure cylinder the quantity is $9.28 \times \frac{41.26}{3.43} = 9.28 + 12.03 = 21.31$.

As an example of this method applied to a triple expansion engine we may take the case of Engine No. 59, in which the ratios of volumes are as 1 to 2.94 to 6.5, and the mean effective pressures, 60.56 lbs., 13.22 lbs., and 10.16 lbs. respectively. The quantity to be used for the H. P. cylinder is $60.56 + (13.22 \times 2.94) + (10.16 \times 6.05) = 60.56 + 38.87 + 66.04 = 165.47$. The quantity for the intermediate cylinder is 13.22 lbs. $+ \frac{60.56}{2.94} + (10.16 \times \frac{6.5}{2.94}) = 20.6 + 13.22 + 22.46 = 56.28$. For the low-pressure cylinder the quantity is $10.16 + (13.22 \times \frac{2.94}{6.5}) + \frac{60.56}{6.5} = 9.32 + 5.98 + 10.16 = 25.46$.

The weights of steam per cubic foot used in the formulae for determining the steam accounted for in the tests under consideration are those deduced from Regnault's experiments as given in D. K. Clark's Manual.

The following examples will serve to illustrate the use of the formulae, one case being a single expansion engine and the other a triple expansion.

Engine No. 22, Simple Condensing Engine.

Clearance	2 %	
Cut-off pressure above zero	75.6	lbs.
Weight per cubic foot at cut-off pressure	.1773	"
Release pressure	15.5	"
Weight per cubic foot at release pressure	.0399	"
Mean effective pressure	37.17	"
Compression pressure	3	"
Weight per cubic foot at compression pressure	.0085	"

Proportion of direct stroke completed at cut-off	.172
Ditto at release	.903
Proportion of return stroke uncompleted at compression . . .	.048

The steam accounted for at cut-off is $\frac{13,750}{37.17}$ $[(.172+.02) \times .1773-(.048+.02) \times .0085]=369.9 \times (.03404-.00056) = 369.9 \times .03348=12.39$. The steam accounted for at release is $\frac{13,750}{37.17}$ $[(.903+.02) \times .0399-(.048+.02) \times .0085 = 369.9\ (.03682-.00056)=369.9 \times .03626=13.41$.

Engine No. 59—Triple Expansion.

H. P. Cylinder at Cut-off.

Clearance	2.5 %	
Cut-off pressure	145.2	lbs.
Weight per cubic foot at cut-off pressure	.3277	"
Compression pressure	46.8	"
Weight per cubic foot at compression pressure	.1129	"
Mean effective pressure	60.56	"
M. E. P. of all the cylinders, referred to H. P. cylinder . . .	165.47	"
Proportion of direct stroke completed at cut-off	.346	
Proportion of return stroke uncompleted at compression . . .	.006	

The steam accounted for at cut-off is $\frac{13,750}{165.47}$ $[(.346+.025) \times .3277-(.006+.025) \times .1129]=83.1 \times (.1216-.0035) = 83.1 \times .1181=9.81$.

Intermediate Cylinder at Cut-off.

Clearance	2.5 %	
Cut-off pressure	38.7	lbs.
Weight per cubic foot at cut-off pressure	.0945	"
Mean effective pressure	13.22	"
M. E. P. of all the cylinders, referred to the intermediate cylinder,	56.28	"
Compression pressure	20.7	"
Weight per cubic foot at compression pressure	.0524	"
Proportion of stroke completed at cut-off	.406	
Proportion of return stroke uncompleted at compression . . .	.008	

The steam accounted for at cut-off is, $\frac{13{,}750}{56.28} \times [\,(.406 + .025) \times .0945 - (.008 + .025) \times .0524] = 244.3 \times (.0407 - .0017) = 244.3 \times .039 = 9.53.$

L. P. Cylinder at Cut-off.

Clearance	2.5 %	
Cut-off pressure	16.0	lbs.
Weight per cubic foot at cut-off pressure	.0411	"
Mean effective pressure	10.16	"
M. E. P. of all the cylinders, referred to L. P. cylinder	25.46	"
Compression pressure	2.3	"
Weight per cubic foot at compression pressure	.0066	"
Proportion of stroke completed at cut-off	.357	
Proportion of return stroke uncompleted at compression	0	

The steam accounted for at cut-off is, $\frac{13{,}750}{25.46}\,[\,(.357 + .025) \times .0411 - (.025 \times .0066)\,] = 540.1 \times (.0157 - .00017) = 540.1 \times .01553 = 8.39.$

It is unnecessary to give the computations for the release points of these diagrams, the method being illustrated in the example given above for Engine No. 22.

The following table gives the quantity $\frac{13{,}750}{\text{m.e.p.}}$ for mean effective pressures running from 10 to 100, advancing by two-tenths of a pound; and from 100 to 200 advancing by pounds.

Table of $\frac{13750}{M.E.P.}$

M. E. P.	13750 / M. E. P.	M. E. P.	13750 / M. E. P.	M. E. P.	13750 / M. E. P.	M. E. P.	13750 / M. E. P.
10.0	1375.0	20.0	687.5	30.0	458.3	40.0	343.8
.2	1348.	.2	680.7	.2	455.3	.2	342.0
.4	1322.1	.4	674.0	.4	452.3	.4	340.3
.6	1297.1	.6	667.5	.6	449.3	.6	338.7
.8	1273.1	.8	661.1	.8	446.4	.8	337.
11.0	1250.	21.0	654.8	31.0	443.5	41.0	335.3
.2	1227.7	.2	648.6	.2	440.7	.2	333.7
.4	1206.1	.4	642.5	.4	437.9	.4	332.1
.6	1185.4	.6	636.6	.6	435.1	.6	330.5
.8	1165.3	.8	630.7	.8	432.4	.8	328.9
12.0	1145.8	22.0	625.0	32.0	429.7	42.0	327.4
.2	1127.1	.2	619.4	.2	427.	.2	325.8
.4	1108.9	.4	613.8	.4	424.4	.4	324.3
.6	1091.3	.6	608.4	.6	421.8	.6	322.8
.8	1074,2	.8	603.1	8	419.2	.8	321.3
13.0	1057.7	23.0	597.8	33.0	416.7	43.0	319.8
.2	1041.7	.2	592.7	.2	414.1	.2	318.3
.4	1026.1	.4	587.6	.4	411.7	.4	315.8
.6	1011.	.6	582.6	.6	409 2	.6	315.4
.8	996.4	.8	577.7	.8	406.8	.8	313.9
14.0	982.1	24.0	572.9	34.0	404.4	44.0	312.5
.2	968.3	.2	568.2	.2	402.	.2	311.1
.4	954.9	.4	563.5	.4	399.7	.4	309.7
.6	941.8	.6	558.9	.6	397.4	.6	308.3
.8	929.0	.8	554.4	.8	395.1	.8	306.9
15.0	916.7	25.0	550.	35.0	392.8	45.0	305.6
.2	904.6	.2	545.6	.2	390.6	.2	304.2
.4	892.9	.4	541.3	.4	388.4	.4	302.9
.6	881.4	.6	537.1	.6	386.2	.6	301.5
.8	870.2	.8	532.9	.8	384.1	.8	300.2
16.0	859.4	26.0	528.8	36.0	381.9	46.0	298.9
.2	848.8	.2	524.8	.2	379.8	.2	297.6
.4	838.4	.4	520.8	.4	377.7	.4	296.3
.6	828.3	.6	516.9	.6	375.7	.6	295.0
.8	818.4	.8	513.	.8	373.6	.8	293.8
17.0	808.8	27.0	509.2	37.0	371.6	47.0	292.5
.2	799.4	.2	505.5	.2	369.6	.2	291.3
.4	790.2	.4	501.8	.4	367.6	.4	290.0
.6	781.2	.6	498.2	.6	365.7	.6	288.8
.8	772.5	.8	494.6	.8	363.7	.8	287.6
18.0	763.9	28.0	491.1	38.0	361.8	48.0	286.4
.2	755.5	.2	487.6	.2	359.9	.2	285.2
.4	747.3	.4	484.2	.4	358.1	.4	284.1
.6	739.2	.6	480.8	.6	356.2	.6	282.9
.8	731.4	.8	477.4	.8	354.4	.8	281.7
19.0	723.7	29.0	474.1	39.0	352.6	49.0	280.6
.2	716.1	.2	470.9	.2	350.8	.2	279.4
.4	708.8	.4	467.7	.4	349.0	.4	278.3
.6	701.5	.6	464.5	.6	347.2	.6	277.2
.8	694.4	.8	461.4	.8	345.5	.8	276.1

Table of $\frac{13750}{M. E. P.}$ *(Continued).*

M. E. P.	13750 / M. E. P.	M. E. P.	13750 / M. E. P.	M. E. P.	13750 / M. E. P.	M. E. P.	13750 / M. E. P.
50.0	275.0	60.0	229.2	70.0	196.4	80.0	171.9
.2	273.9	.2	228.4	.2	195.9	.2	171.4
.4	272.8	.4	227.6	.4	195.3	.4	171.0
.6	271.7	.6	226.9	.6	194.7	.6	170.6
.8	270.6	.8	226.1	.8	194.2	.8	170.2
51.0	269.6	61.0	225.4	71.0	193.6	81.0	169.7
.2	268.5	.2	224.7	.2	193.1	.2	169.3
.4	267.5	.4	223.9	.4	192.5	.4	168.9
.6	266.4	.6	223.2	.6	192.0	.6	168.5
.8	265.4	.8	222.5	.8	191.5	.8	168.1
52.0	264.4	62.0	221.8	72.0	191.0	82.0	167.7
.2	263.4	.2	221.1	.2	190.4	.2	167.2
.4	262.4	.4	220.3	.4	189.9	.4	166.9
.6	261.4	.6	219.6	.6	189.4	.6	166.4
.8	260.4	.8	218.9	.8	188.9	.8	166.1
53.0	259.4	63.0	218.2	73.0	188.3	83.0	165.6
.2	258.4	.2	217.6	.2	187.8	.2	165.3
.4	257.5	.4	216.9	.4	187.3	.4	164.8
.6	256.5	.6	216.2	.6	186.8	.6	164.5
.8	255.5	.8	215.5	.8	186.3	.8	164.1
54.0	254.6	64.0	214.8	74.0	185.8	84.0	163.7
.2	253.6	.2	214.2	.2	185.3	.2	163.3
.4	252.7	.4	213.5	.4	184.8	.4	162.9
.6	251 8	.6	212.8	.6	184.3	.6	162.5
.8	250.9	.8	212.2	.8	183 8	.8	162.1
55.0	250.0	65.0	211.5	75.0	183.3	85.0	161.7
.2	249.1	.2	210.9	.2	182.8	.2	161.4
.4	248.2	.4	210.2	.4	182.3	.4	161.0
.6	247.3	.6	209.6	.6	181.9	.6	160.6
.8	246.4	.8	208.9	.8	181.4	.8	160.2
56.0	245.5	66.0	208.3	76.0	180.9	86.0	159.9
.2	244.6	.2	207.7	.2	180.4	.2	159.5
.4	243.8	.4	207.1	.4	180.0	.4	159.1
.6	242.9	.6	206.4	.6	179.5	.6	158.7
.8	242.1	.8	205.8	.8	179.0	.8	158.4
57.0	241.2	67.0	205.2	77.0	178.6	87.0	158.0
.2	240.4	.2	204.6	.2	178.1	.2	157.7
.4	239.5	.4	204.0	.4	177.6	.4	157.3
.6	238.7	.6	203.4	.6	177.2	.6	157.0
.8	237.8	.8	202.8	.8	176.7	.8	156.6
58.0	237.0	68.0	202.2	78.0	176.3	88.0	156.2
.2	236.2	.2	201.6	.2	175.8	.2	155.9
.4	235.4	.4	201.0	.4	175.4	.4	155.5
.6	234.6	.6	200.4	.6	174.9	.6	155.2
.8	233.8	.8	199.8	.8	174.5	.8	154.8
59.0	233 0	69.0	199.3	79.0	174.1	89.0	154.5
.2	232.2	.2	198.7	.2	173.6	.2	154.1
.4	231.4	.4	198.1	.4	173.2	.4	153.8
.6	230.7	.6	197.6	.6	172.7	.6	153.5
.8	229.9	.8	197.0	.8	172.3	.8	153.1

Table of $\frac{13750}{M.\ E.\ P.}$ *(Concluded).*

M. E. P.	13750 / M. E. P.	M. E. P.	13750 / M. E. P.	M. E. P.	13750 / M. E. P.	M. E. P.	13750 / M. E. P.
90.0	152.8	97.6	140.9	126	109.13	164	83.84
.2	152.4	.8	140.6	7	108.27	165	83.33
.4	152.1	98.0	140.3	8	107.42	6	82.83
.6	151.7	.2	140.	9	106.59	7	82.34
.8	151.4	.4	139.7	130	105.77	8	81.85
91.0	151.1	.6	139.4	1	104.96	9	81.36
.2	150.8	.8	139.1	2	104.17	170	80.88
.3	150.5	99.0	138.9	3	103.38	1	80.41
.6	150.1	.2	138.6	4	102.61	2	79.94
.8	149.8	.4	138.3	135	101.85	3	79.48
92.0	149.5	.6	138.	6	101.10	4	79.02
.2	149.2	.8	137.8	7	100.36	175	78.57
.4	148.8	100	137.5	8	99.64	6	78 13
.6	148.5	1	136.14	9	98.92	7	77.68
.8	148.2	2	134.8	140	98.21	8	77.25
93.0	147.9	3	133.5	1	97.52	9	76.82
.2	147.5	4	132.21	2	96.83	180	76.39
.4	147.2	105	130.95	3	96.15	1	75.97
.6	146.9	6	129.71	4	95.49	2	75.55
.8	146.6	7	128.5	145	94.83	3	75.14
94.0	146.3	8	127.31	6	94.18	4	74.73
.2	146.	9	126.15	7	93.54	185	74.32
.4	145.6	110	125.	8	92.91	6	73.93
.6	145.3	1	123.88	9	92.28	7	73.53
.8	145.	2	122.77	150	91.67	8	73.14
95.0	144.7	3	121.68	1	91.06	9	72.75
.2	144.4	4	120.61	2	90.46	190	72.37
.3	144.1	115	119.57	3	89.87	I	71.99
.6	143.8	6	118.54	4	89.29	2	71.62
.8	143.5	7	117.52	155	88.71	3	71.25
96.0	143.2	8	116.53	6	88.14	4	70.88
.2	142.9	9	115.55	7	87.59	195	70.51
.4	142.6	120	114.58	8	87.03	6	70.15
.6	142.3	1	113.64	9	86.48	7	69.80
.8	142.	2	112.71	160	85.94	8	69.44
97.0	141.7	3	111.79	1	85.40	9	69.10
.2	141.4	4	110.89	2	84.88	200	68.75
.4	141.2	125	110.	3	84.36		

I

PART II.

FEED-WATER TESTS.

SIMPLE ENGINES.

[*These engines are all horizontal, unjacketed, and of the automatic cut-off type, with fly-ball governor, unless otherwise specified.*]

ENGINE No. 1.

Simple Non-Condensing.

Kind of engine	Four-valve (Corliss)	
Number of cylinders	1	
Diameter of cylinder	23	in.
Diameter of piston-rod	3¼	in.
Stroke of piston	5	ft.
Clearance	2½	%
H. P. constant for 1 lb. m. e. p. one revolution per min.	.1247	
Inside diameter of steam pipe	7	in.
Inside diameter of exhaust pipe	8	in.
Condition of valves and pistons regarding leakage	Practically tight	

Data and Results of Feed-Water Test, Engine No. 1.

Character of steam	Ordinary	
Duration	5.75	hrs.
Weight of feed-water consumed	48,741	lbs.
Feed-water consumed per hour	8,477	lbs.
Pressure in steam pipe above atm.	72.3	lbs.
Mean effective pressure	33.08	lbs.
Revolutions per minute	74.7	
Indicated horse-power	305.2	H. P.
Feed-water consumed per I. H. P. per hour	27.77	lbs.

Measurements based on Sample Diagrams.

Initial pressure above atmosphere	72.8	lbs.
Steam-pipe pressure above atmosphere	73.6	lbs.
Cut-off pressure above zero	66.5	lbs.
Release pressure above zero	24.3	lbs.
Mean effective pressure	33.12	lbs.
Back pressure at mid stroke above atmosphere	2.8	lbs.
Proportion of stroke completed at cut-off	.367	
Steam accounted for at cut-off	23.32	lbs.
Steam accounted for at release	23.66	lbs.
Proportion accounted for at cut-off	.84	
Proportion accounted for at release	.852	

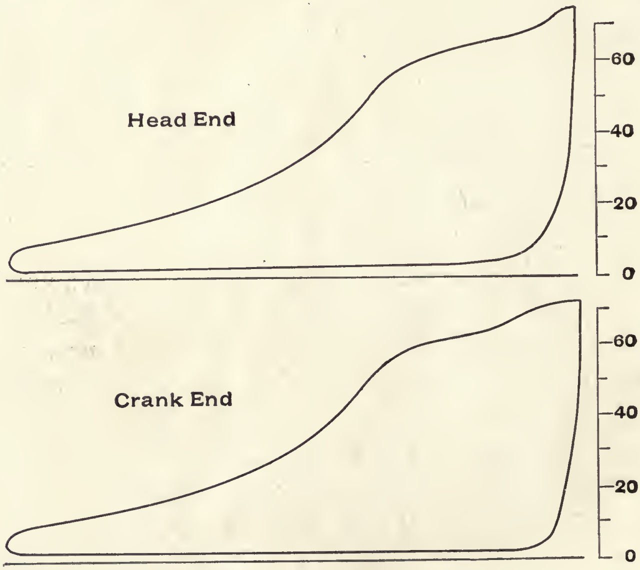
Engine No. 1 is supplied with steam in part from a number of vertical boilers, and in part from a single boiler of the horizontal return tubular type. The mixed steam showed no superheating, though probably commercially dry. The valves and pistons were all fairly tight. The load consisted of cotton machinery.

On another occasion two tests were made on this engine, the first with ordinary steam as above, and the second with superheated steam, the horizontal boiler in the latter case being out of service. The principal data and results were as follows:

Test. Character of Steam.		No. 1b. Ordinary.	No. 1c. Superheated 82°.
Mean effective pressure	lbs.	34.46	35.07
Proportion of stroke completed at cut-off		.375	.392
Feed-water consumed per I. H P. per hour	lbs.	29.34	26.83
Steam accounted for at cut-off . . .	lbs.	24.6	25.42
Steam accounted for at release . . .	lbs.	25.26	24.15
Proportion accounted for at cut-off . .		.839	.947
Proportion accounted for at release . .		.861	.900

The marked effect of superheating is indicated by comparing these two tests. By superheating the steam 82 degrees the consumption of feed-water per I. H. P. per hour was reduced about 9 per cent. A feature in these results is the effect upon the steam accounted for by the indicator. It increases between cut-off and release from 24.6 pounds to 25.26 pounds when ordinary steam is used, whereas the contrary effect is produced under the influence of the superheating, the quantity falling from 25.42 pounds to 24.15 pounds.

ENGINE No. 1



ENGINE No. 2.

Simple Non-Condensing.

Kind of engine	Four-valve (Corliss)	
Number of cylinders	1	
Diameter of cylinder	28.5	in.
Diameter of piston-rod	4	in.
Stroke of piston	59.5	in.
Clearance	3	%
H. P. constant for 1 lb. m. e. p. one revolution per min.	.1898	
Inside diameter of steam pipe	8	in.
Condition of valves and piston regarding leakage	Fairly tight.	

Data and Results of Feed-Water Test, Engine No. 2.

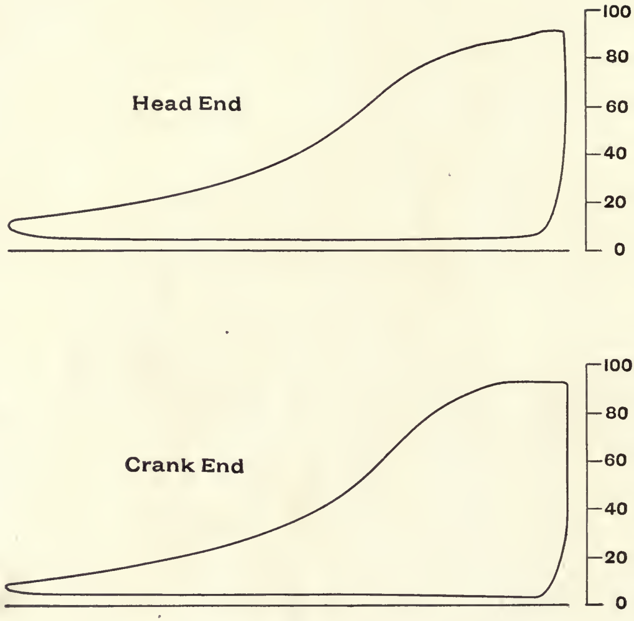
Character of steam	Ordinary	
Duration	6.08	hr.
Weight of feed-water consumed	79,467	lbs.
Feed-water consumed per hour	13,070	lbs.
Pressure in steam pipe above atmosphere	101	lbs.
Mean effective pressure	41.18	lbs.
Revolutions per minute	64.8	
Indicated horse-power	506.5	H. P.
Feed-water consumed per I. H. P. per hour	25.8	lbs.

Measurements based on Sample Diagrams.

Initial pressure above atmosphere	91	lbs.
Cut-off pressure above zero	82.9	lbs.
Release pressure above zero	26.3	lbs.
Mean effective pressure	41.18	lbs.
Back pressure at mid-stroke above atmosphere	4.2	lbs.
Proportion of stroke completed at cut-off	.315	
Steam accounted for at cut-off	21.06	lbs.
Steam accounted for at release	21.35	lbs.
Proportion accounted for at cut-off	.817	
Proportion accounted for at release.	.828	

Engine No. 2 is supplied with steam from water tube boilers. A calorimeter test showed less than one per cent of moisture. The steam valves and piston were tight. The exhaust valves leaked a small amount. The load was that of a cotton-mill.

ENGINE No. 2



ENGINE No. 3.

Simple Condensing.

Kind of engine	Four-valve (Corliss)	
Number of cylinders	2	
Diameter of each cylinder	20⅛	in.
Diameter of piston-rod	2⅞	in.
Stroke of each piston	4	ft.
Clearance	3	%
H. P. constant for 1 lb. m. e. p. one revolution per min.	.1532	
Inside diameter of steam pipe	8	in.
Inside diameter of exhaust pipe	8	in.
Condition of valves and pistons regarding leakage	Fairly tight.	

Data and Results of Feed-Water Tests, Engine No. 3.

Conditions as to Use of Condenser.		Test A. All-Condensing.	Test B. Three Ends Condensing, One End Non-Condensing.
Character of steam		Ordinary	Ordinary
Duration	hrs.	4.75	4.75
Weight of feed-water consumed	lbs.	21,185.	24,671.
Feed-water consumed per hour	lbs.	4,460.	5,194.
Pressure in steam pipe above atmosphere	lbs.	67.2	69.1
Vacuum in condenser	in.	26.2	26.5
Mean effective pressure	lbs.	22.79	24.79
Revolutions per minute		60.3	60.3
Indicated horse-power	H.P.	210.5	229.
Feed-water consumed per I. H. P. per hour	lbs.	21.11	22.68

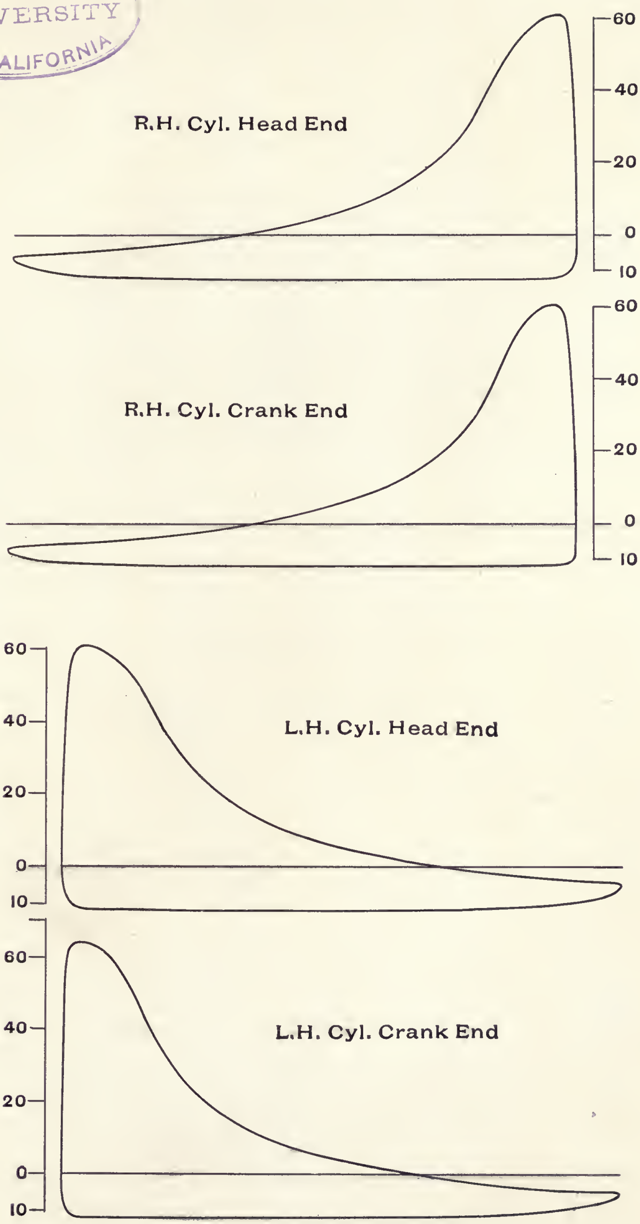
Measurements based on Sample Diagrams.

Initial pressure above atmosphere	lbs.	60.8	64.1	
			Average of Three Condensing Ends.	Non-Condensing End.
Cut-off pressure above zero	lbs.	59.9	63.	63.
Release pressure above zero	lbs.	9.3	10.5	13.5
Mean effective pressure	lbs.	23.23	26.41	19.01
Back pressure at mid-stroke above or below atmosphere	lbs.	11.9	− 12.	+ 1.
Proportion of stroke completed at cut-off	lbs.	.138	.152	.185
Steam accounted for at cut-off	lbs.	13.85	13.95	21.19
Steam accounted for at release	lbs.	14.77	14.59	24.06
Proportion accounted for at cut-off (average for the whole engine)		.654	.695	
Proportion accounted for at release		.697	.748	

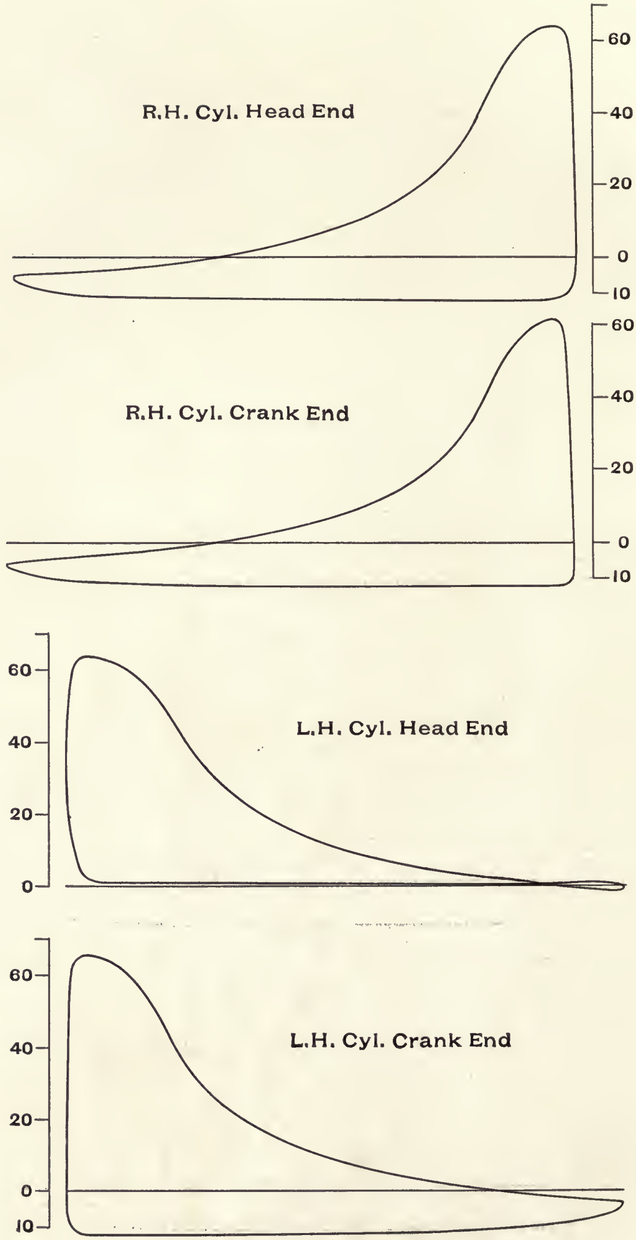
Engine No. 3 has a pair of cylinders exhausting into a jet condenser operated by a direct-connected air-pump. The exhaust passages and piping are arranged so as to run one end of one cylinder non-condensing. One test was made running both cylinders condensing, and one test running three ends condensing and one end non-condensing. The engine is supplied with steam from horizontal return tubular boilers. The quality of the steam was not tested, but it was probably commercially dry. One steam valve and the exhaust valves of one cylinder showed some leakage. The remaining valves, and the pistons, were fairly tight. The engine was employed in driving several manufactories working in connection with water-wheels.

The loss in steam due to running one end of the cylinder non-condensing is about 7%. The gain in fuel that would be produced by utilizing the exhaust steam from this end for heating feed-water for the boilers, assuming that it increases the temperature from 60 to 210 degrees, is sufficient to cover the increased steam consumption and leave a net fuel saving of some 7%.

ENGINE No. 3 a


R.H. Cyl. Head End
60
40
20
0
10
R.H. Cyl. Crank End
60
40
20
0
10
L.H. Cyl. Head End
60
40
20
0
10
L.H. Cyl. Crank End
60
40
20
0
10

ENGINE No. 3 b


R.H. Cyl. Head End
60
40
20
0
10
R.H. Cyl. Crank End
60
40
20
0
10
L.H. Cyl. Head End
60
40
20
0
L.H. Cyl. Crank End
60
40
20
0
10

ENGINE No. 4.

Simple Condensing.

Kind of engine	Four-valve (Corliss)
Number of cylinders	1
Diameter of cylinder	34.2 ins.
Diameter of piston-rod	4⅞ ins.
Stroke of piston	5 ft.
Clearance	3 %
H.P. Constant for one lb. m.e.p., one rev. per minute .	.2764
Inside diameter of steam pipe	6 ins.
Inside diameter of exhaust pipe	7 ins.
Condition of valves and piston regarding leakage . . .	Fairly tight.

Data and Results of Feed Water Test, Engine No. 4.

Character of steam	Superheated 25	deg.
Duration	10.8	hrs.
Weight of feed-water consumed	125,420	lbs.
Feed-water consumed per hour	11,613	lbs.
Pressure in steam pipe above atmosphere	83	lbs.
Vacuum in condenser	24.8	ins.
Mean effective pressure	35.53	lbs.
Rev. per min.	53.3	
Indicated horse-power	523.43	H. P.
Feed-water consumed per I. H. P. per hour	22.19	lbs.

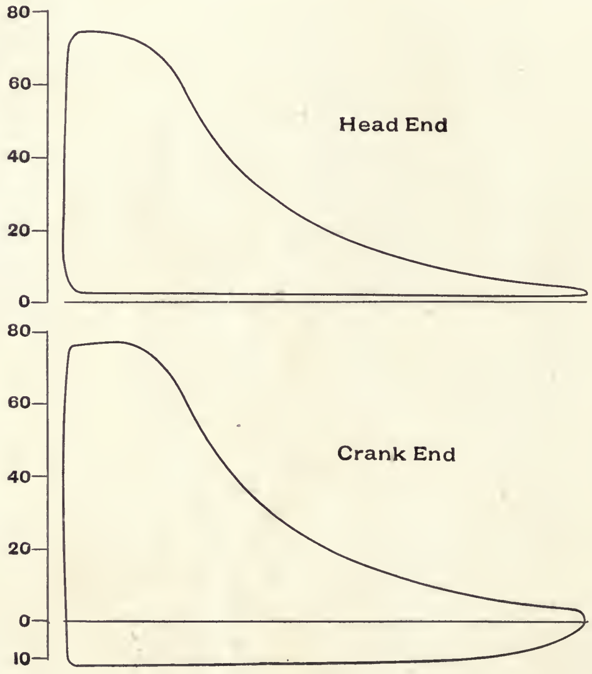
Measurements Based on Sample Diagrams: —

Initial pressure above atmosphere	76.1 lbs.
Steam-pipe pressure above atmosphere	83 lbs.

		Head End. Non-condensing.	Crank End. Condensing.
Cut-off pressure above zero	lbs.	71.9	76.7
Release pressure above zero	lbs.	17.5	18.1
Mean effective pressure	lbs.	28.22	41.88
Back pressure at mid-stroke, above or below atmosphere	lbs.	+ 2.1	— 11.7
Proportion of stroke completed at cut-off	lbs.	.237	.230
Steam accounted for at cut-off . . .	lbs.	18.79	15.18
Steam accounted for at release . . .	lbs.	18.75	15.43
Proportion accounted for at cut-off (average of two ends)	lbs.	.766	
Proportion accounted for at release .	lbs.	.77	

Engine No. 4 exhausts into a jet condenser with direct-connected air-pump. One end is run condensing, and the other end non-condensing. The boilers are of the vertical type, which superheat the steam. Steam was supplied for other purposes than power, and the amount thus used was determined and allowed for. There was slight leakage of the steam valves. The exhaust valves and piston were practically tight. The load was that of a cotton mill.

ENGINE No. 4



ENGINE No. 5.

Simple Condensing.

Kind of engine	Four-valve (Corliss)	
Number of cylinders	2	
Diameter of each cylinder	32.5	ins.
Diameter of each piston-rod	$4\frac{1}{4}$	ins.
Stroke of each piston	4.5	ft.
Clearance	3	%
H.P. Constant for one lb. M.E.P. one rev. per min. . .	.4484	
Inside diameter of steam-pipe	7	ins.
Condition of valves and pistons regarding leakage . . .	Some leakage.	

Data and Results of Feed-Water Test, Engine No. 5.

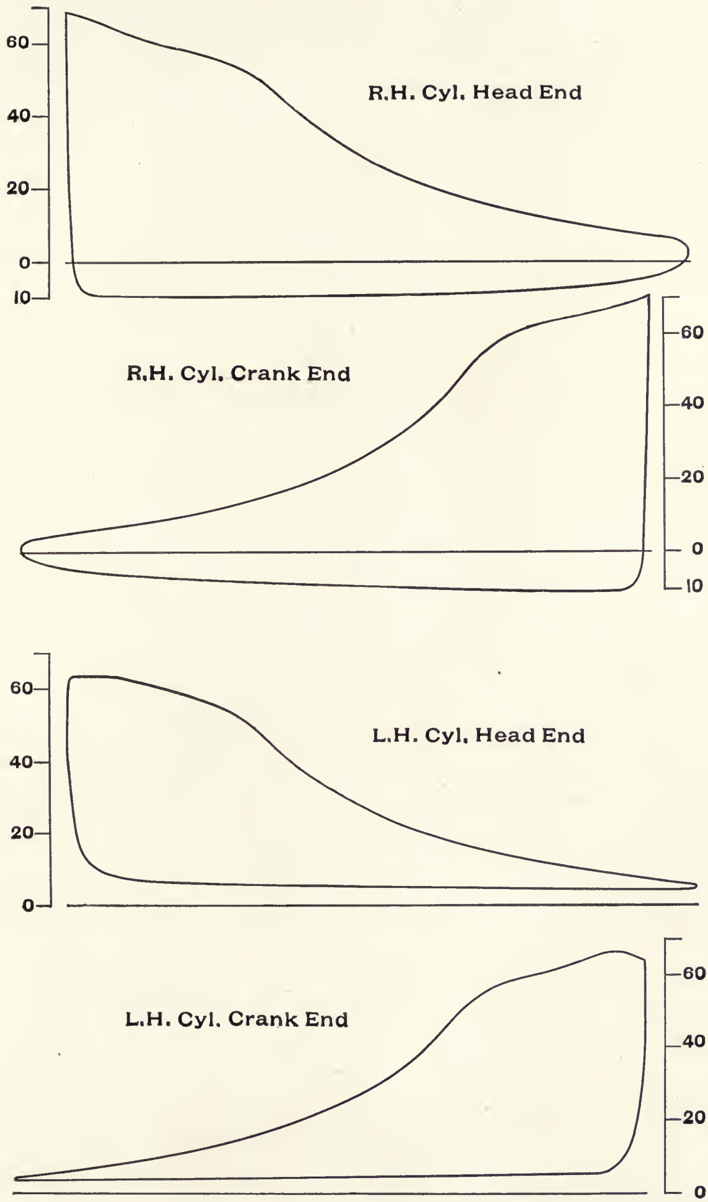
Character of steam	Ordinary	
Duration	5.55	hrs.
Weight of feed-water consumed	100,253	lbs.
Feed-water consumed per hour	18,063	lbs.
Pressure in steam pipe	71.1	lbs.
Vacuum in condenser	26.2	in.
Mean effective pressure	32.41	lbs.
Revolutions per minute	47.3	
Indicated horse-power	687.39	H. P.
Feed-water consumed per I. H. P. per hour	26.28	lbs.

Measurements Based on Sample Diagrams.

		Condensing Cylinder.	Non-Condensing Cylinder.
Initial pressure above atmosphere . . .	lbs.	70.3	65.2
Cut-off pressure above zero	lbs.	63.5	63.
Release pressure above zero	lbs.	20.1	20.
Mean effective pressure	lbs.	39.54	26.2
Back pressure at mid-stroke, above or below atmosphere	lbs.	− 9.4	+ 4.9
Proportion of stroke completed at cut-off .		.298	.308
Steam accounted for at cut-off	lbs.	16.78	24.47
Steam accounted for at release	lbs.	17.36	25.39
Proportion accounted for at cut-off (average for whole engine)		.784	
Proportion accounted for at release . . .		.813	

Engine No. 5 has a pair of cylinders, one of which exhausts into a jet condenser, with direct-connected air-pump, and the other is non-condensing. Steam is furnished from cylinder boilers, and it appeared to be commercially dry. A small amount was used for other purposes than running the engine, but the quantity thus consumed was determined, and allowance made for it. The valves and piston of one cylinder showed some leakage; those of the other cylinder were fairly tight. The load consisted of cotton machinery.

ENGINE No. 5


R.H. Cyl. Head End
60
40
20
0
10
R.H. Cyl. Crank End
60
40
20
0
10
L.H. Cyl. Head End
60
40
20
0
L.H. Cyl. Crank End
60
40
20
0

ENGINE No. 6.

Simple Condensing.

Kind of engine	Four-valve (Corliss)	
Number of cylinders	2	
Diameter of each cylinder	26¼	in.
Diameter of each piston rod	3¾	in.
Stroke of each piston	5	ft.
Clearance	3	%
H. P. Constant for one lb. m. e. p. one rev. per minute,	.3254	
Inside diameter of steam pipe	8	in.
Inside diameter of exhaust pipe	10	in.
Condition of valves and pistons regarding leakage . . .	Some leakage.	

Data and Results of Feed-Water Test, Engine No. 6.

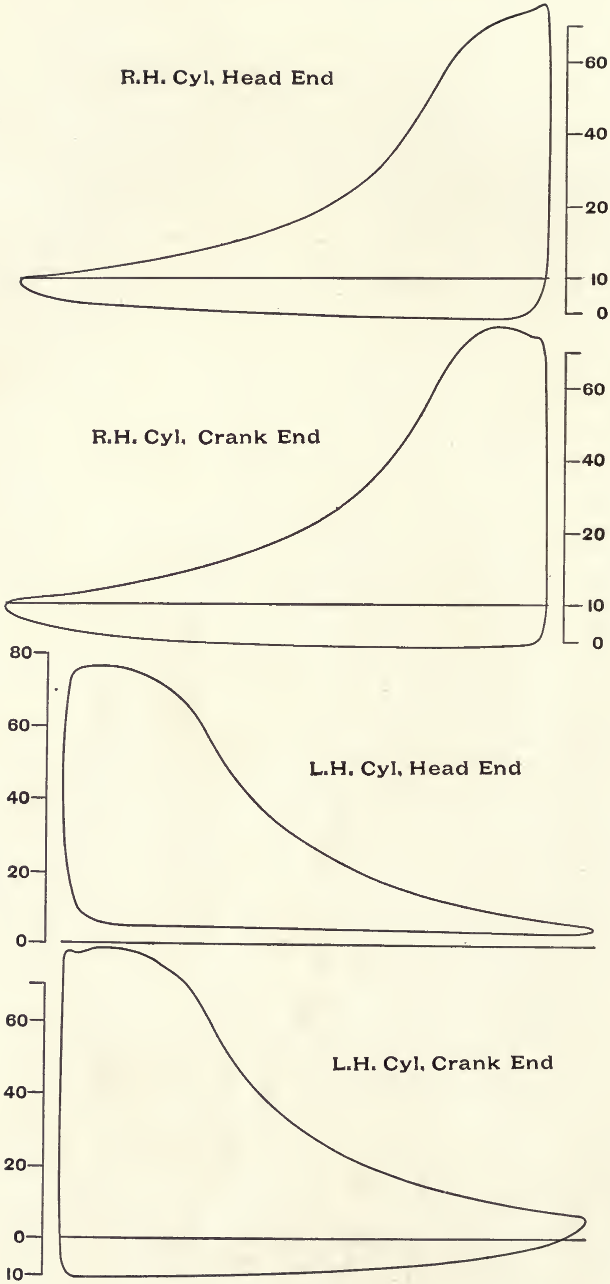
Character of steam	Ordinary	
Duration	5.08	hrs.
Weight of feed-water consumed	71,150	lbs.
Feed-water consumed per hour	14,006	lbs.
Pressure in steam pipe	84.4	lbs.
Vacuum in condenser	27.3	in.
Mean effective pressure	36.73	lbs.
Revolutions per minute	51.1	
Indicated horse-power	610.74	H. P.
Feed-water consumed per I. H. P. per hour	22.95	lbs.

Measurements based on Sample Diagrams.

		Three Ends Condensing.	Non-Condensing End.
Initial pressure above atmosphere . . .	lbs.	77.3	76.
Cut-off pressure above zero	lbs.	74.6	73.8
Release pressure above zero	lbs.	17.2	20.4
Mean effective pressure	lbs.	39.46	30.03
Back pressure at mid stroke, above or below atmosphere	lbs.	− 10.2	+ 4.5
Proportion of stroke completed at cut-off .		.233	.271
Steam accounted for at cut-off	lbs.	15.81	22.12
Steam accounted for at release	lbs.	15.56	22.63
Proportion accounted for at cut-off (average for whole engine		.757	
Proportion accounted for at release (average)		.754	

Engine No. 6 has a pair of cylinders exhausting into a jet condenser with direct-connected air-pump. One cylinder was run condensing, and one end of the other cylinder non-condensing. Steam is supplied from sectional boilers with large drum, and from all appearances it was in a commercially dry condition. Both pistons showed some leakage, but the valves were all fairly tight. The load consisted of cotton machinery.

ENGINE No. 6


R.H. Cyl, Head End
60
40
20
10
0
R.H. Cyl, Crank End
60
40
20
10
0
80
60
40
20
0
L.H. Cyl, Head End
60
40
20
0
10
L.H. Cyl, Crank End

ENGINE No. 7.

Simple Non-Condensing.

Kind of engine	Four-valve (Corliss)	
Number of cylinders	1	
Diameter of cylinder	$26\frac{3}{16}$	in.
Diameter of piston rod	$3\frac{7}{8}$	in.
Stroke of piston	4	ft.
Clearance	3	%
H. P. constant for one lb. m. e. p. one revolution per min.	.1285	
Inside diameter of steam pipe	6	in.
Condition of valves and piston regarding leakage	Some leakage.	

Data and Results of Feed-Water Test, Engine No. 7.

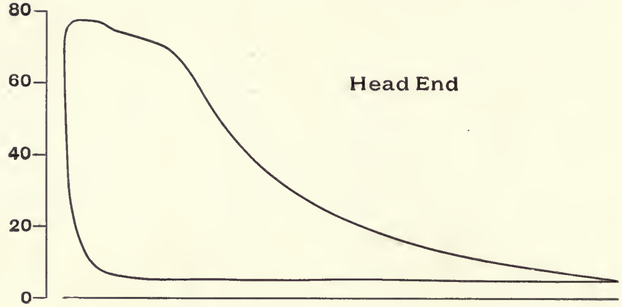
Character of steam	Ordinary	
Duration	5.1	hrs.
Weight of feed-water consumed	34,386	lbs.
Feed-water consumed per hour	6,742	lbs.
Pressure in steam pipe above atmosphere	80.5	lbs.
Mean effective pressure	27.82	lbs.
Revolutions per minute	64.7	
Indicated horse-power	232.3	H. P.
Feed-water consumed per I. H. P. per hour	29.03	lbs.

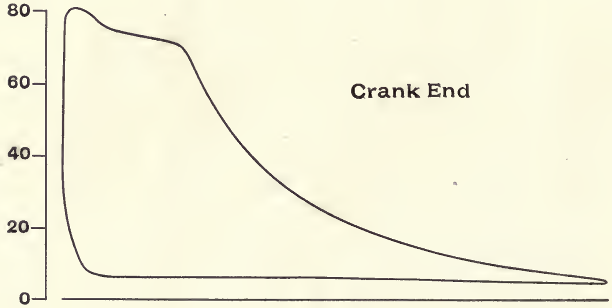
Measurements based on Sample Diagrams.

Initial pressure above atmosphere	79.6	lbs.
Cut-off pressure above zero	76.6	lbs.
Release pressure above zero	19.6	lbs.
Mean effective pressure	27.3	lbs.
Back pressure at mid stroke above atmosphere	5.4	lbs.
Proportion of stroke completed at cut-off	.237	
Steam accounted for at cut-off	21.77	lbs.
Steam accounted for at release	23.31	lbs.
Proportion accounted for at cut-off	.75	
Proportion accounted for at release	.803	

Engine No. 7 is supplied with steam from horizontal return tubular boilers, presumably in a commercially dry condition. The valves were fairly tight, but there was considerable leakage of the piston. The load consisted of cotton machinery.

ENGINE No. 7





ENGINE No. 8.

Simple Condensing.

Kind of engine	Four-valve (Corliss)	
Number of cylinders	1	
Diameter of cylinder	30	ins.
Diameter of piston rod	4⅝	ins.
Stroke of piston	6	ft.
Clearance	3	%
H.P. constant for one lb. m.e.p. one rev. per min. . .	.2543	
Condition of valves and piston regarding leakage . . .	Fairly tight.	

Data and Results of Feed-Water Tests.

Conditions as to Pressure.		Test A. Ordinary.	Test B. Extra.
Character of steam		Superhtd. 37°	Superhtd. 37°
Duration	hrs.	5.667	5.167
Weight of feed-water consumed	lbs.	40,281.	34,984.
Feed-water consumed per hour	lbs.	7,104.	6,771.
Pressure in steam-pipe above atmosphere .	lbs.	53.1	68.2
Vacuum in condenser	ins.	29.7	29.8
Mean effective pressure	lbs.	26.63	26.3
Revolutions per minute		54.1	54.1
Indicated horse-power	H.P.	366.4	361.8
Feed-water consumed per I.H.P. per hour	lbs.	19.39	18.71

Measurements based on Sample Diagrams.

Conditions as to Pressure.		Test A. Ordinary.	Test B. Extra.
Initial pressure above atmosphere . . .	lbs.	46.5	61.5
Cut-off pressure above zero	lbs.	47.0	58.6
Release pressure above zero	lbs.	11.1	9.8
Mean effective pressure	lbs.	26.84	26.39
Back pressure at mid-stroke below atm. .	lbs.	— 12.4	— 12.4
Proportion of stroke completed at cut-off .	lbs.	.247	.165
Steam accounted for at cut-off	lbs.	15.89	13.98
Steam accounted for at release	lbs.	15.19	13.72
Proportion accounted for at cut-off . . .		.819	.747
Proportion accounted for at release . .		.783	.733

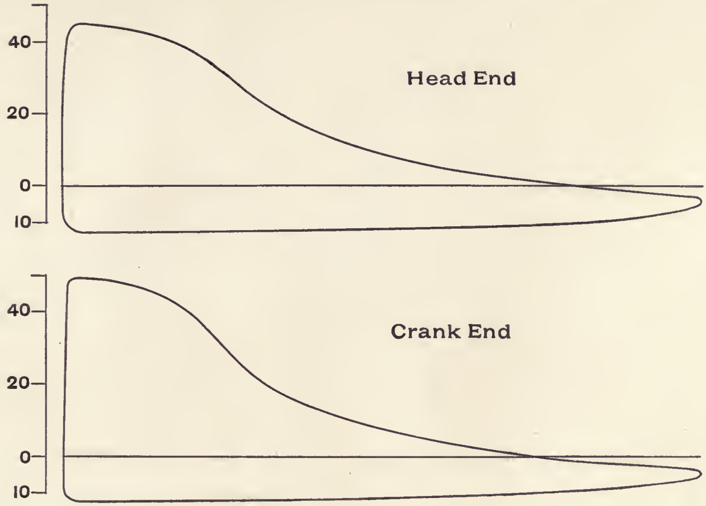
Engine No. 8 exhausts into a jet condenser with direct-connected air-pump. Steam is supplied through a 12-inch pipe, 160 feet in length, from vertical boilers which superheat. The amount of superheating at the boilers on the test was 67 degrees.

It was subsequently found that the loss of temperature between the boilers and the throttle valve was 60 degrees; so that the steam entering the cylinder was still in a slightly superheated condition. The valves and pistons were fairly tight. The load consisted of cotton machinery.

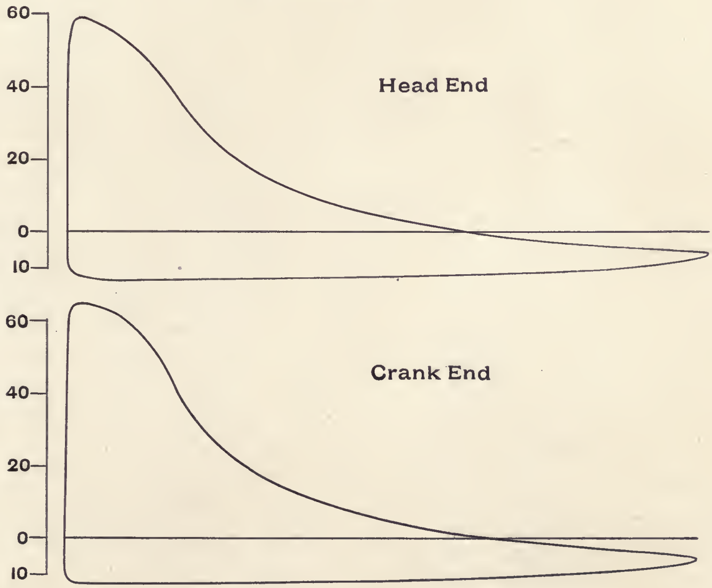
Advantage was taken of the comparatively light load to make a trial of the engine under two pressures. The other conditions of running were the same in both cases.

It appears that the increase of pressure from 53 pounds to 68 pounds was attended by a reduction in the steam consumption amounting to nearly four per cent. There is a marked increase in the cylinder condensation (and leakage), with the shortening of the cut-off and increase of pressure.

ENGINE No. 8 a



ENGINE No. 8b



ENGINE No. 9.

Simple Condensing.

Kind of engine	Four valve (Corliss)
Number of cylinders	2
Diameter of each cylinder	$30\frac{1}{8}$ ins.
Diameter of each piston rod	$4\frac{3}{8}$ ft.
Stroke of each piston	6 ft.
Clearance	3 %
H.P. constant for one lb. m.e.p. one rev. per minute . .	.515
Inside diameter of steam pipe	8 ins.
Condition of valves and pistons regarding leakage . . .	Some leakage.

Data and Results of Feed-Water Tests.

Conditions as to Use of Condenser.		Test A. All Condensing.	Test B. Three-Fourths Condensing.
Character of steam		Supd. 24°	Superhtd. 24°
Duration	hrs.	5.1	5.37
Weight of feed-water consumed	lbs.	70,565.	83,060.
Feed-water consumed per hour	lbs.	13,838.	15,467.
Pressure in steam pipe above atmosphere .	lbs.	70.8	73.4
Vacuum in condenser	ins.	26.7	26.7
Mean effective pressure	lbs.	32.00	31.44
Revolutions per minute		46.	46.
Indicated horse-power	H.P.	758.27	758.10
Feed-water consumed per I.H.P. per hour	lbs.	18.25	20.4

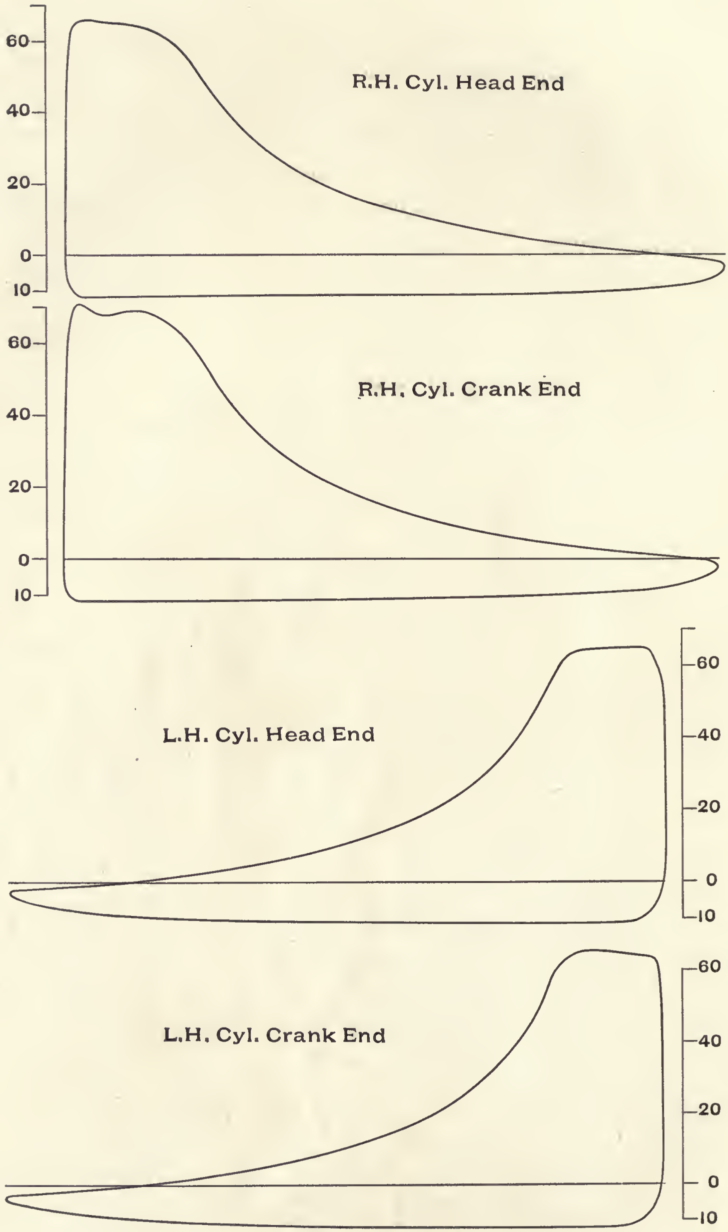
Measurements based on Sample Diagrams.

Conditions as to Use of Condenser.			Non-Condensing End.	Three Ends Condensing.
Initial pressure above atmosphere . . .	lbs.	67.7	70.7	67.8
Cut-off pressure above zero	lbs.	70.4	69.6	72.5
Release pressure above zero	lbs.	12.9	17.2	13.9
Mean effective pressure	lbs.	32.10	26.11	34.54
Back pressure at mid-stroke, above or below atmosphere	lbs.	—11.5	+4.0	—11.5
Proportion of stroke completed at cut-off .		.185	.247	.202
Steam accounted for at cut-off	lbs.	14.97	21.94	14.45
Steam accounted for at release	lbs.	14.52	21.88	14.47
Proportion accounted for at cut-off . . .		.82	.836	
Proportion accounted for at release . . .		.796	.8	

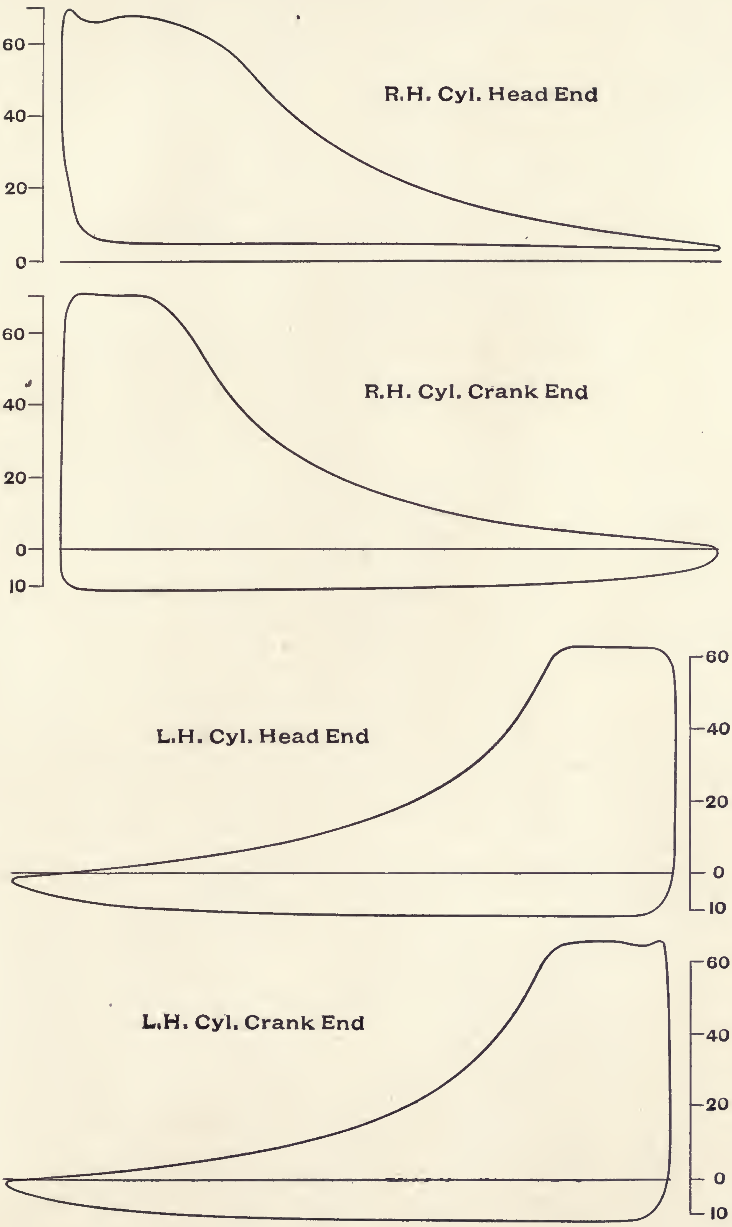
Engine No. 9 has a pair of cylinders exhausting into a jet condenser with direct-connected air-pump. Steam is supplied in a slightly superheated condition from vertical boilers. Arrangements are made so that one end of one cylinder can be run non-condensing. One test was made with this end operating non-condensing, and another when the whole engine was running condensing. The exhaust valves and steam valves of one cylinder were fairly tight. The steam valves of the other cylinder and the pistons of both cylinders showed some leakage. The load consisted of cotton machinery.

The loss of steam due to running one end of one cylinder non-condensing is 2.15 pounds per I. H. P. per hour, or 11.8% of the quantity required when running condensing.

ENGINE No. 9 a


R.H. Cyl. Head End
R.H. Cyl. Crank End
L.H. Cyl. Head End
L.H. Cyl. Crank End
60
40
20
0
10

ENGINE No.9b


R.H. Cyl. Head End
60
40
20
0
R.H. Cyl. Crank End
60
40
20
0
10
L.H. Cyl. Head End
60
40
20
0
10
L.H. Cyl. Crank End
60
40
20
0
10

ENGINE No. 10.

Simple Condensing Engine.

Kind of engine	Double Valve
Number of cylinders	2
Diameter of each cylinder	17 in.
Diameter of each piston rod	2.75 in.
Stroke of each piston	24.2 in.
Clearance	2 %
H. P. Constant for one lb. m. e. p. one rev. per minute . .	.0551 H.P.
Inside diameter of steam pipe	6 in.
Condition of valves and pistons regarding leakage	Some leakage

Data and Results of Feed-Water Tests, Engine No. 10.

Conditions as to Use of Condenser.		Condensing.	Non-Condensing.
Character of steam		Superhtd. 16°	Superhtd. 41°
Duration	hrs.	5.7	5.21
Weight of feed-water consumed	lbs.	39,299.	41,415.
Feed-water consumed per hour	lbs.	6,895.	7,952.
Pressure in steam pipe above atmosphere .	lbs.	79.	75.9
Vacuum in condenser	ins.	23.6	
Mean effective pressure	lbs.	39.36	36.82
Revolutions per minute		154.7	152.9
Indicated horse-power	I.H.P.	336.2	310.1
Feed-water consumed per I.H.P. per hour	lbs.	20.51	25.64

Measurements based on Sample Diagrams.

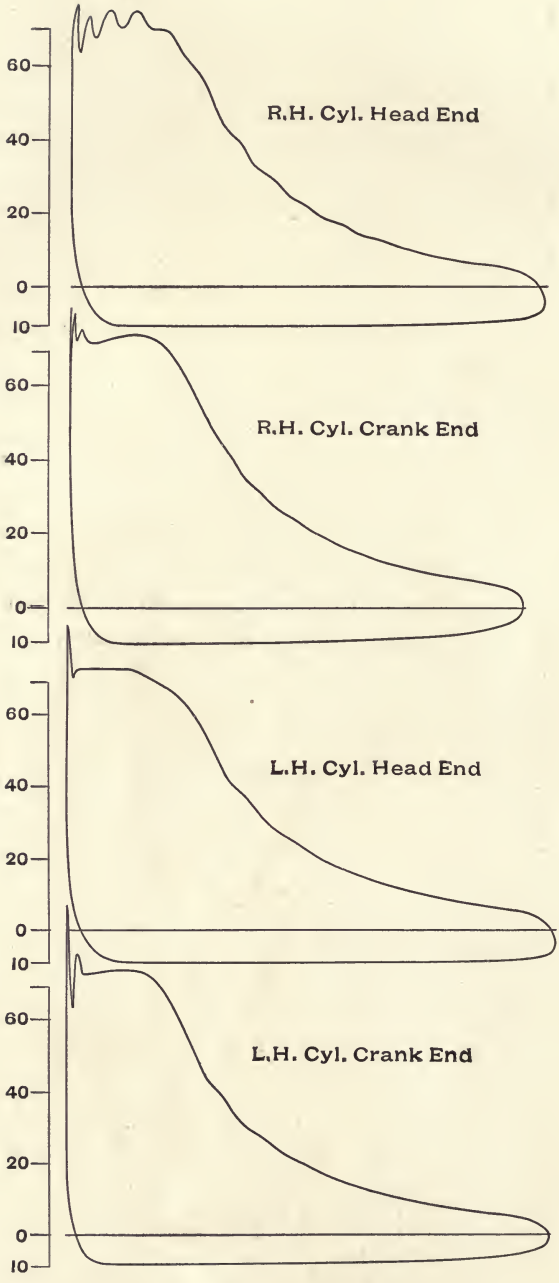
Conditions as to Use of Condenser.		Condensing.	Non-Condensing.
Initial pressure above atmosphere . . .	lbs.	74.5	76.4
Steam-pipe pressure above atmosphere . .	lbs.	78	79
Cut-off pressure above zero	lbs.	72.9	72.8
Release pressure above zero	lbs.	19.7	25.2
Mean effective pressure	lbs.	42.15	37.43
Back pres. at mid stroke above or below atm.	lbs.	— 10.1	+ 1.0
Proportion of stroke completed at cut-off .		.262	.337
Steam accounted for at cut-off	lbs.	15.82	20.78
Steam accounted for at release	lbs.	15.53	20.42
Proportion accounted for at cut-off . . .		.771	.811
Proportion accounted for at release . . .		.757	.795

Engine No. 10 has a pair of cylinders, with condenser of the siphon type, which is supplied with water by means of a belt pump operated by the engine. The main valves are balanced slides. The cut-off valve rides on a seat in the interior of the main value, which is of box pattern. The cut-off valve is controlled by a shaft governor. One test was made with the engine running condensing, and one running non-condensing. Steam is furnished by superheating vertical boilers, which are 190 feet distant from the throttle valves, the connecting pipe being 10 inches in diameter. The loss of temperature from the boilers to the engine amounted to 54 degrees. The pistons and cut-off valves were practically tight. The main valves showed some leakage. The engine worked in connection with water-wheels, and supplied power to a cotton-mill.

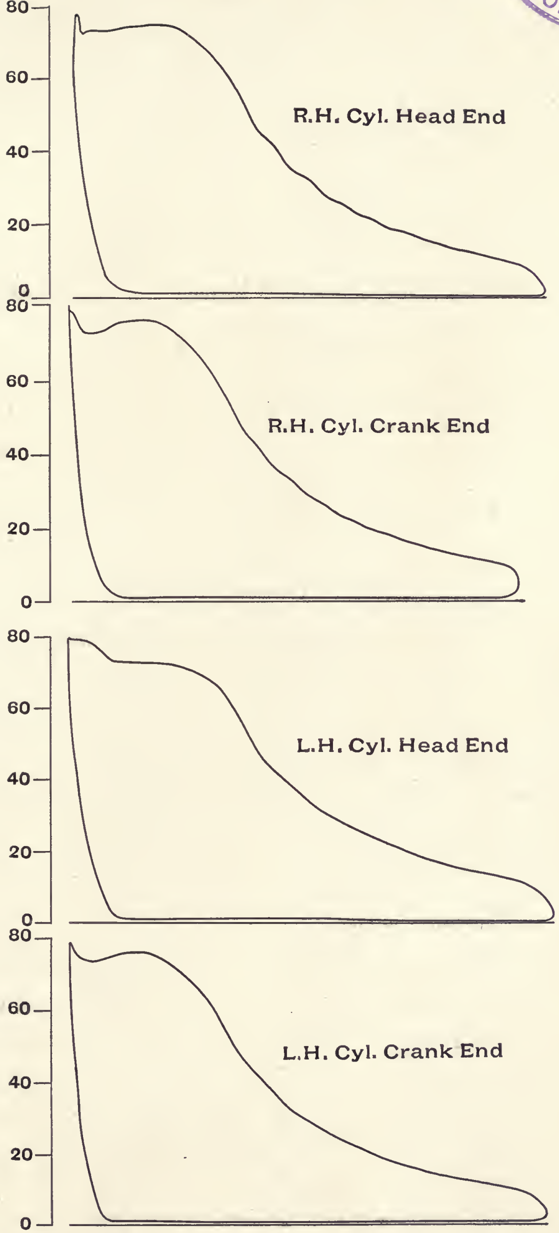
From these results it appears that the consumption of steam when the engine was run condensing was 5.13 lbs. per I. H. P. per hour less than when run non-condensing, or 20%.

In making this comparison it should be observed that there was a comparatively poor vacuum, both in the cylinders and in the condenser, which acted unfavorably upon the condensing result; and this was further influenced in the same direction by the relatively small amount of superheating.

ENGINE No.10a



ENGINE No. 10b



ENGINE No. 11.

Simple Non-Condensing Engine.

Kind of engine	Four valve	
Number of cylinders	1	
Diameter of cylinder	$16\frac{5}{8}$	ins.
Diameter of piston rod	$2\frac{3}{4}$	ins.
Stroke of piston	32	ins.
Clearance	4	%
H.P. constant for one lb. m. e. p. one rev. per min.	.0346	H.P.
Inside diameter of steam pipe	5	in.
Condition of valves and piston regarding leakage	Considerable leakage	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	5.47	hrs.
Weight of feed-water consumed	10,277	lbs.
Feed-water consumed per hour	1,879	lbs.
Pressure in steam-pipe above atmosphere	61	lbs.
Mean effective pressure	18.19	lbs.
Revolutions per minute	79.8	
Indicated horse-power	50.2	I.H.P.
Feed-water consumed per I. H. P. per hour	37.43	lbs.

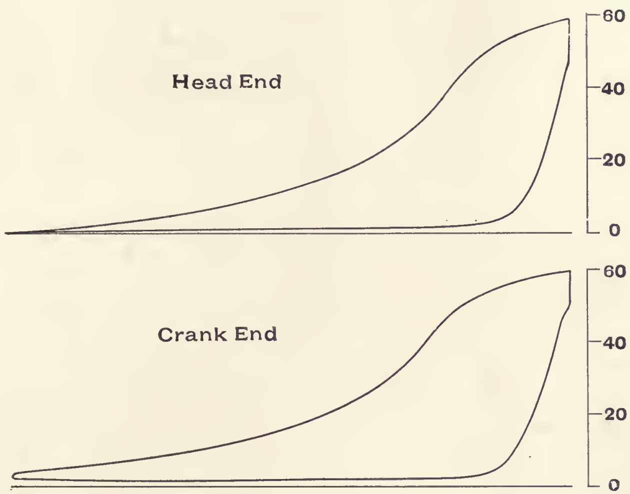
Measurements based on Sample Diagrams.

Initial pressure above atmosphere	58.6	lbs.
Cut-off pressure above zero	56.1	lbs.
Release pressure above zero	16.7	lbs.
Mean effective pressure	18.29	lbs.
Back pressure at mid stroke above atmosphere	1.1	lbs.
Proportion of stroke completed at cut-off	.234	
Steam accounted for at cut-off	21.52	lbs.
Steam accounted for at release	27.42	lbs.
Proportion accounted for at cut-off	.575	
Proportion accounted for at release	.732	

Engine No. 11 is controlled by a shaft governor. It has piston valves provided with means for adjustment to take up wear. Steam is supplied from return tubular boilers, probably in a commercially dry condition. The piston and one steam valve were fairly tight. The other steam valve and both exhaust valves leaked. The engine was employed in driving a machine-shop.

The effect of leakage, low pressure, and light load is seen in the excessive consumption of steam shown on this test.

ENGINE No. 11



ENGINE No. 12.

Simple Condensing Engine.

Kind of engine	Four valve (Corliss)	
Number of cylinders	1	
Diameter of cylinder	24¼	in.
Diameter of piston rod	3¼	in.
Stroke of piston	4	ft.
Clearance	3	%
H. P. constant for one lb. m. e. p. one revolution per min.	.112	H.P.
Inside diameter of steam pipe	6	in.
Inside diameter of exhaust pipe	7	in.
Condition of valves and piston regarding leakage	Considerable leakage	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	5.07	hrs.
Weight of feed-water consumed	30,920	lbs.
Feed-water consumed per hour	6,099	lbs.
Pressure in steam pipe above atmosphere	70.2	lbs.
Vacuum in condenser	21	in.
Mean effective pressure	33.06	lbs.
Revolutions per minute	70.2	
Indicated horse-power	258.2	I.H.P.
Feed-water consumed per I. H. P. per hour	23.62	lbs.

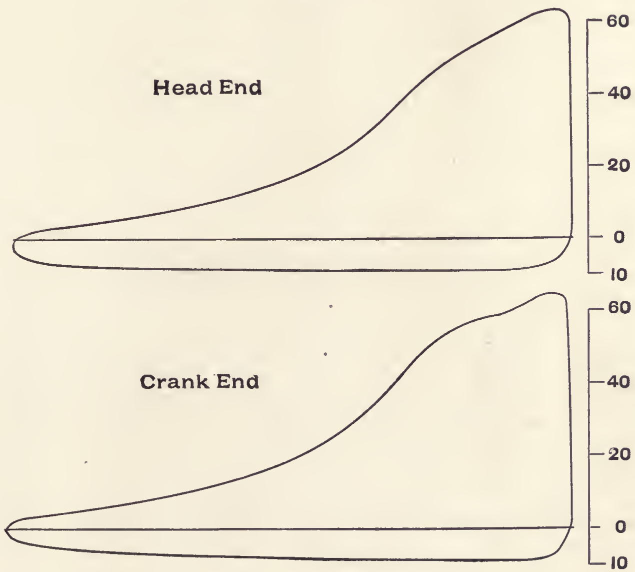
Measurements based on Sample Diagrams.

Initial pressure above atmosphere	63.6	lbs.
Cut-off pressure above zero	56.1	lbs.
Release pressure above zero	17.8	lbs.
Mean effective pressure	33.31	lbs.
Back pressure at mid stroke, below atmosphere	8.5	lbs.
Proportion of stroke completed at cut-off	.29	
Steam accounted for at cut-off	16.99	lbs.
Steam accounted for at release	17.75	lbs.
Proportion accounted for at cut-off	.719	
Proportion accounted for at release	.751	

Engine No. 12 exhausts into a jet condenser having a direct connected air-pump. The joints about the air-pump were out of repair, and the condenser was rendered somewhat inefficient. Steam is supplied from vertical boilers, which do not superheat, but which appeared to furnish steam in a commercially dry condition. One of the steam valves leaked, but the remaining valves were practically tight. The piston leaked badly. The load consisted of cotton machinery.

Leakage and the poor vacuum are evidently accountable for the comparatively low result obtained here.

ENGINE No. 12



ENGINE No. 13.

Simple Non-Condensing Engine.

Kind of engine	Single valve	
Number of cylinders	1	
Diameter of cylinder	14.5	in.
Diameter of piston rod	2¼	in.
Stroke of piston	13	in.
Clearance	10	%
H. P. constant for one lb. m. e. p. one revolution per min.	.0108	H.P.
Inside diameter of steam pipe	4	in.
Inside diameter of exhaust pipe	6	in.
Condition of valves and piston regarding leakage	Considerable leakage	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	2.5	hr.
Weight of feed-water consumed	4,350	lbs.
Feed-water consumed per hour	1,740	lbs.
Pressure in steam pipe above atmosphere	102.5	lbs.
Mean effective pressure	20.07	lbs.
Revolutions per minute	246	
Indicated horse-power	53.26	I.H.P.
Feed-water consumed per I. H. P. per hour	32.67	lbs.

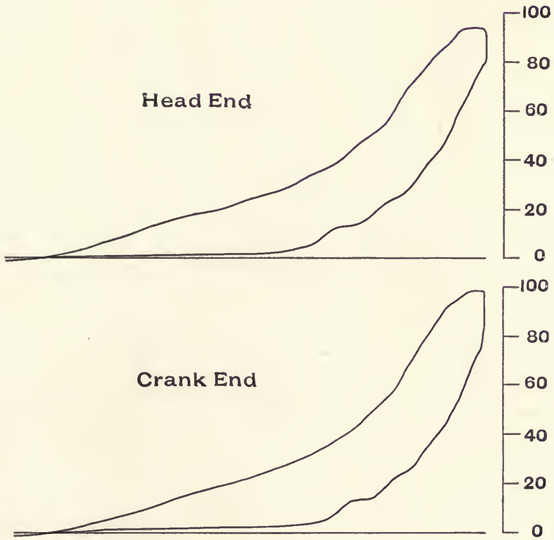
Measurements based on Sample Diagrams.

Initial pressure above atmosphere	98.1	lbs.
Cut-off pressure above zero	97.0	lbs.
Release pressure above zero	40.8	lbs.
Mean effective pressure	20.07	lbs.
Back pressure at mid stroke above atmosphere	+2.6	lbs.
Proportion of stroke completed at cut-off	.119	
Steam accounted for at cut-off	17.72	lbs.
Steam accounted for at release	22.59	lbs.
Proportion accounted for at cut-off	.539	
Proportion accounted for at release	.391	

Engine No. 13 is of the high-speed class, with a shaft governor. The valve is of the piston type, unpacked. Steam is supplied from a water-tube boiler, and is presumed to be in a commercially dry condition. The piston was fairly tight. The valve at one end was fairly tight, but at the other end it leaked badly. The load consisted of a dynamo furnishing current for electric lighting.

The leaking of the piston valve is evidently responsible in some degree for the comparatively poor showing on this engine.

ENGINE No. 13



ENGINE No. 14.

Simple Non-Condensing Engine.

Kind of engine	Single valve	
Number of cylinders	1	
Diameter of cylinder	8.5	in.
Diameter of piston rod	$1\frac{3}{8}$	in.
Stroke of piston	10	in.
Clearance	8	%
H. P. constant for 1 lb. m. e. p. one revolution per min.	.0028	H.P.
Inside diameter of steam pipe	$2\frac{1}{2}$	in.
Inside diameter of exhaust pipe	$3\frac{1}{2}$	in.
Condition of valves and piston regarding leakage	Considerable leakage	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	$2\frac{1}{2}$	hrs.
Weight of feed-water consumed	2,357	lbs.
Feed-water consumed per hour	942.8	lbs.
Pressure in steam pipe above atmosphere	105.3	lbs.
Mean effective pressure	30.58	lbs.
Revolutions per minute	315	
Indicated horse-power	27.35	I.H.P.
Feed-water consumed per I. H. P. per hour	34.44	lbs.

Measurements based on Sample Diagrams.

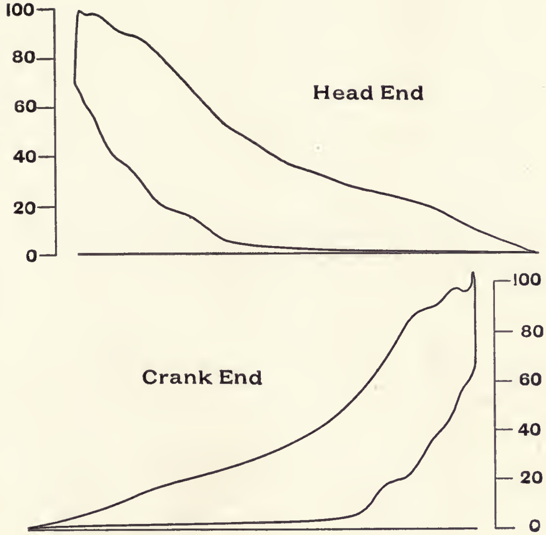
Initial pressure above atmosphere	99.6	lbs.
Cut-off pressure above zero	92.5	lbs.
Release pressure above zero	34.1	lbs.
Mean effective pressure	30.58	lbs.
Back pressure at mid stroke above atmosphere	1.7	lbs.
Proportion of stroke completed at cut-off	.194	
Steam accounted for at cut-off	17.92	lbs.
Steam accounted for at release	21.17	lbs.
Proportion accounted for at cut-off	.52	
Proportion accounted for at release	.615	

Engine No. 14 is of the high-speed class, controlled by a shaft governor. It is provided with a piston valve which is unpacked. Steam is supplied from a water-tube boiler, probably in a commercially dry condition. The piston was fairly tight. The valve leaked badly. The load consisted of a dynamo furnishing current for electric lighting.

The boiler plant in this case is the same as that of Engine No. 13.

The inferior economy exhibited here can be attributed in the main to leakage.

ENGINE No. 14



ENGINE No. 15.

Simple Condensing Engine.

Kind of engine	Four valve (Corliss)	
Number of cylinders	2	
Diameter of each cylinder	23	in.
Diameter of each piston rod	$3\frac{3}{8}$	in.
Stroke of each piston	5	ft.
Clearance	3	%
H. P. constant for 1 lb. m. e. p. one revolution per min.	.249	H.P.
Inside diameter of steam pipe	6	in.
Inside diameter of exhaust pipe	10	in.
Condition of valves and pistons regarding leakage	Fairly tight	

Data and Results of Feed-Water Test.

Character of steam	Superheated 59°	
Duration	5.63	hrs.
Weight of feed-water consumed	74,247	lbs.
Feed-water consumed per hour	13,187	lbs.
Pressure in steam pipe above atmosphere	77.6	lbs.
Vacuum in condenser	27.9	in.
Mean effective pressure	40.49	lbs.
Revolutions per minute	61	
Indicated horse-power	615.1	I.H.P.
Feed-water consumed per I. H. P. per hour	21.44	lbs.

Measurements Based on Sample Diagrams.

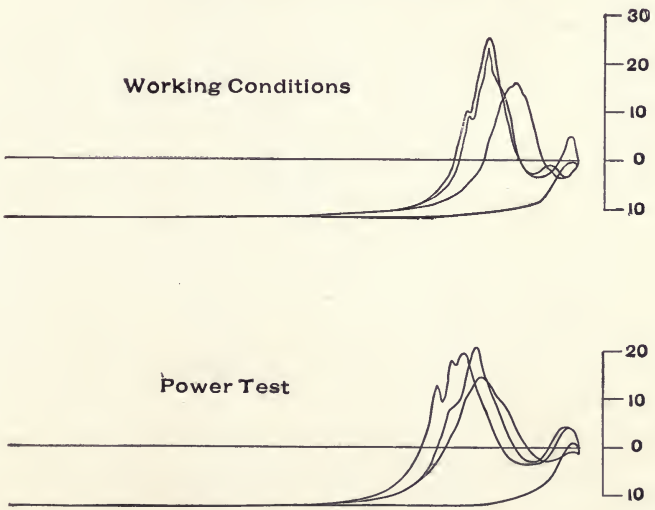
		Condensing Cylinder.	Non-Condensing Cylinder.
Initial pressure above atmosphere	lbs.	76.7	73.2
Cut-off pressure above zero	lbs.	80.4	80.1
Release pressure above zero	lbs.	20.3	23.6
Mean effective pressure	lbs.	45.33	36.05
Back pressure at mid stroke, above or below atmosphere	lbs.	− 11.9	+ 2.6
Proportion of stroke completed at cut-off		.264	.298
Steam accounted for at cut-off	lbs.	16.62	22.41
Steam accounted for at release	lbs.	15.93	22.29
Proportion accounted for at cut-off		.895	
Proportion accounted for at release		.874	

Engine No. 15 has a pair of cylinders provided with a jet-condenser and direct connected air-pump. The exhaust piping is arranged so as to run one cylinder condensing and the other non-condensing, as was done on the test. Steam is supplied from vertical superheating boilers. A small quantity of steam was drawn from the boilers and used for other purposes than running the engine; but the quantity is insignificant, and no allowance is made for it. The steam valves and pistons of both cylinders were practically tight. There was a slight amount of leakage in all the exhaust valves. The engine was employed in driving a cotton-mill.

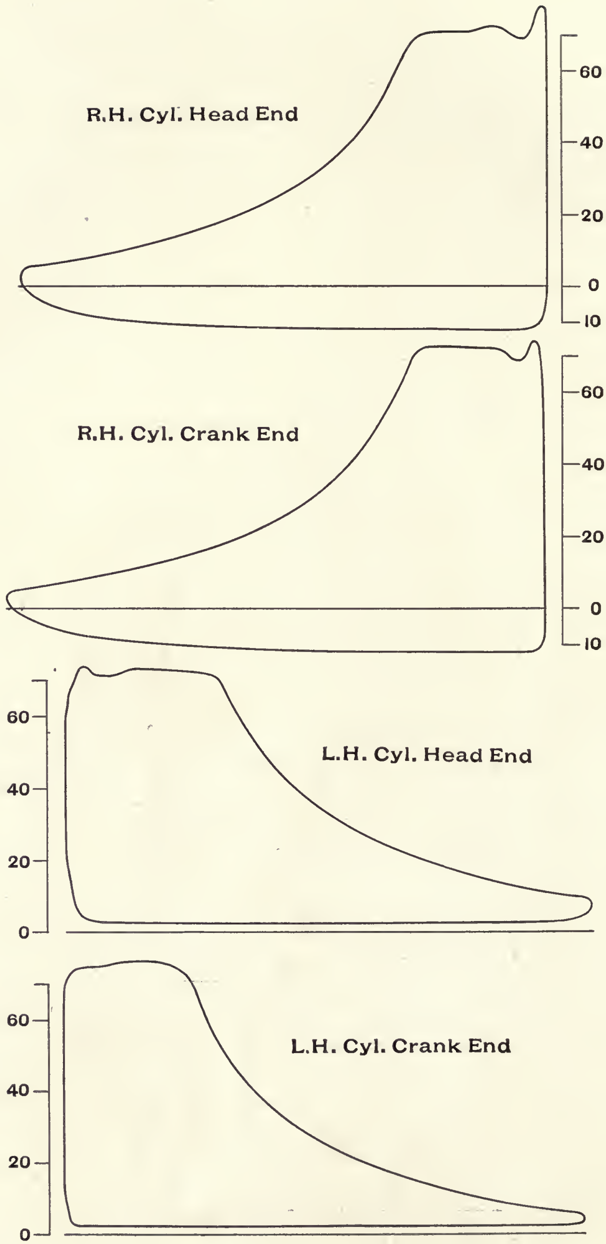
A test was made on this engine to determine the amount of power used by the air-pump, which had a vertical plunger 22 in. diameter and 12-in. stroke. The connecting-rod on the condensing side was disconnected, and cards were taken from the other cylinder first with air-pump in operation and then with air-pump stopped. The load driven in both cases was the shafting of the mill. The difference in the two results was 10.8 horse-power, or 1.8% of the working power of the engine.

Sample indicator diagrams from the pump cylinder are appended, the first taken under the working conditions, and the second obtained on the power test.

ENGINE No. 15 AIR PUMP



ENGINE No. 15


R.H. Cyl. Head End
60
40
20
0
10
R.H. Cyl. Crank End
60
40
20
0
10
L.H. Cyl. Head End
60
40
20
0
L.H. Cyl. Crank End
60
40
20
0

ENGINE No. 16.

Simple Non-Condensing Engine.

Kind of engine	Single valve	
Number of cylinders	2	
Diameter of each cylinder	9.5	in.
Diameter of each indicator rod	.375	in.
Stroke of each piston	9	in.
Clearance	14.1	%
H. P. constant for 1 lb. m. e. p. one revolution per min. . .	.00322 H.P.	
Inside diameter of steam pipe	3½	in.
Inside diameter of exhaust pipe	3½	in.
Condition of valves and pistons regarding leakage	Some leakage	

Data and Results of Feed-Water Tests.

Letter by Which Tests are Designated		A.	B.	C.
Character of steam		Ordinary	Ordinary	Ordinary
Duration	hrs.	2.908	2.983	3.067
Weight of feed-water consumed .	lbs.	4,248.3	3,451.9	2,854.76
Feed-water consumed per hour .	lbs.	1,460.9	1,157.2	930.8
Pressure in steam pipe above atmosphere	lbs.	91 7	92.5	92.1
Mean effective pressure . . .	lbs.	39.49	30.76	22.33
Revolutions per minute		352.2	353.9	356.7
Indicated horse-power	I.H.P.	44.81	35.08	25.66
Feed-water consumed per I. H. P. per hour	lbs.	32.6	32.99	36.27

Measurements based on Sample Diagrams.

Letter by Which Tests are Designated		A.	B.	C.
Initial pressure above atmosphere	lbs.	84.7	85.3	82.7
Cut-off pressure above zero . .	lbs.	79.1	77.1	76.4
Release pressure above zero . .	lbs.	38.3	33.8	30.6
Mean effective pressure . . .	lbs.	39.57	30.55	22.29
Back pressure at mid stroke above atmosphere	lbs.	+ 2.1	+2.8	+4.
Proportion of stroke completed at cut-off		.353	.278	.206
Steam accounted for at cut-off .	lbs.	22.92	21.51	19.92
Steam accounted for at release .	lbs.	23.27	22.89	24.07
Proportion accounted for at cut-off		.703	.652	.549
Proportion accounted for at release		.714	.694	.664

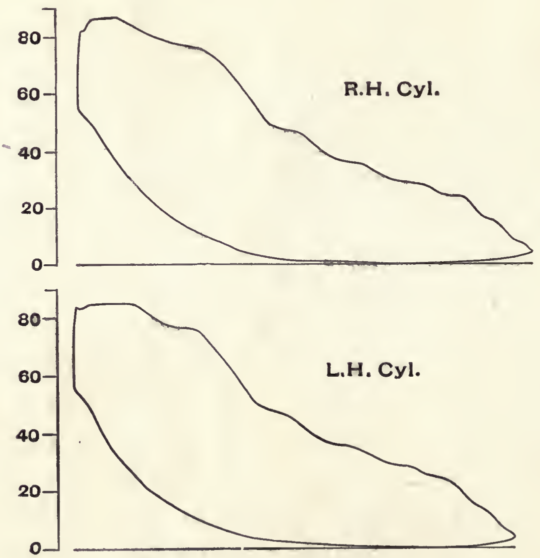
Engine No. 16 has a pair of vertical, single-acting cylinders with working parts inclosed in a chamber partly filled with oil. The valve, which is common to both cylinders, is of the piston type with ring packing. Steam is supplied by a vertical boiler having only a small amount of steam-heating surface. At a point near the throttle valve a calorimeter test showed the presence of 3% of moisture, no allowance for which is made in the record of results. The pistons were practically tight. The valve leaked a small amount. The load consisted of a Prony brake applied to the fly-wheel.

The tests were three in number, made with different loads.

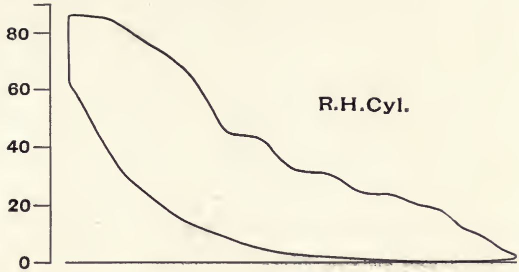
In these tests it appears that the economy of the engine was not materially affected by reducing the load from 44.81 H. P. to 35.08 H. P. A further reduction, however, increased the consumption.

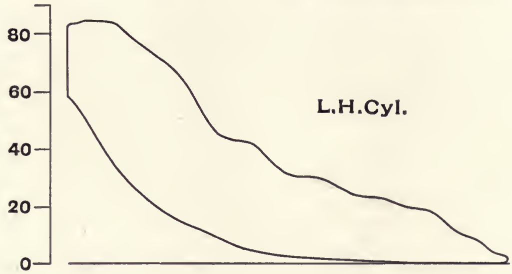
In connection with this series of tests, experiments were made on the effect of a reduction of speed. It was found that with a speed of 201.1 revolutions per minute, the steam consumption per horse power per hour was increased 10 per cent.

ENGINE No. 16a

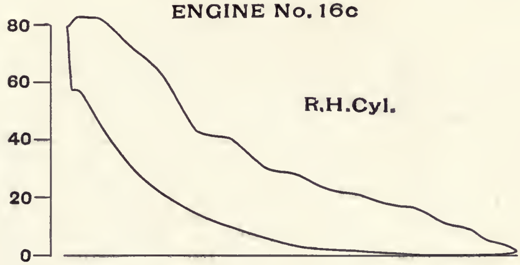


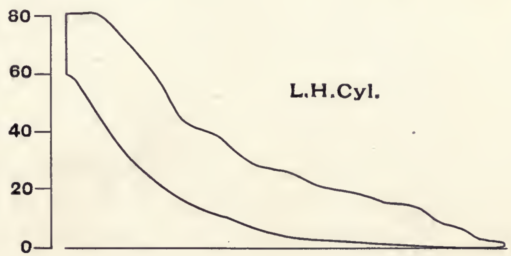
ENGINE No. 16b





ENGINE No. 16c





ENGINE No. 17.

Simple Condensing Engine.

Kind of engine	Four valve
Number of cylinders	1
Diameter of each cylinder	18 ins.
Diameter of piston-rod	$2\frac{5}{8}$ ins.
Stroke of piston	30 ins.
Clearance	5 %
H.P. Constant for one lb. m.e.p. one rev. per min.	.031 H.P.
Condition of valves and pistons regarding leakage	Fairly tight

Data and Results of Feed-Water Tests.

Conditions Regarding Use of Condenser. Test.		Condensing. A.	Non-Condensing. B.
Character of steam		Ordinary	Ordinary
Duration	hrs.	4.1	4.
Weight of feed-water consumed. . .	lbs.	19.298.	24,201.
Feed-water consumed per hour . . .	lbs.	4,707.	6,050.2
Pressure in steam pipe above atmosphere	lbs.	67.	67.6
Vacuum in condenser	ins.	25.5	
Mean effective pressure	lbs.	33.75	33.34
Rev. per min		165.6	164.4
Indicated horse-power	I.H.P.	213.2	209.1
Feed-water consumed per I.H.P. per hour	lbs.	22.08	28.93

Measurements Based on Sample Diagrams.

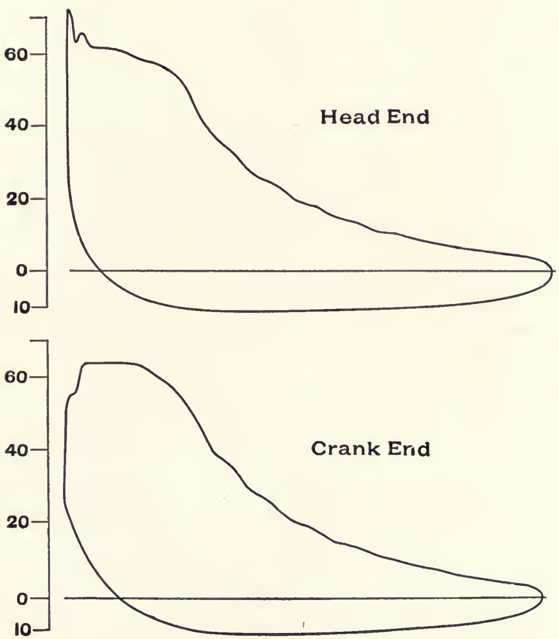
Conditions Regarding Use of Condenser. Test.		Condensing A.	Non-Condensing B.
Cut-off pressure above zero	lbs.	62.1	64.5
Release pressure above zero	lbs.	18.0	28.2
Mean effective pressure	lbs.	33.99	33.75
Back pressure at mid stroke, above or below atmosphere	lbs.	—10.3	+1.5
Proportion of stroke completed at cut-off		.264	.385
Steam accounted for at cut-off . . .	lbs.	17.11	23.75
Steam accounted for at release . . .	lbs.	17.5	23.54
Proportion accounted for at cut-off (average of two ends)		.77	.82
Proportion accounted for at release .	lbs.	.79	.81

Engine No. 17 has balanced slide valves. The condenser is of the siphon pattern supplied with injection water under a natural head. Steam is taken from return tubular boilers, and it is presumed to be commercially dry. The valves were practically tight, but the pistons showed a small amount of leakage. The load was cotton machinery.

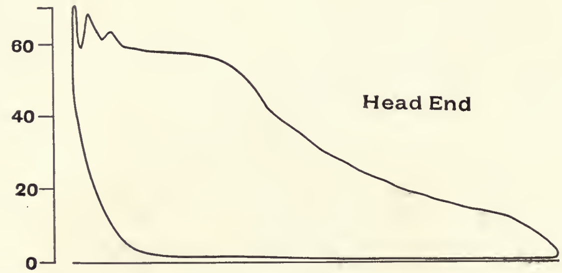
Two tests were made, one with the condenser in operation, and the other with the engine exhausting into the atmosphere.

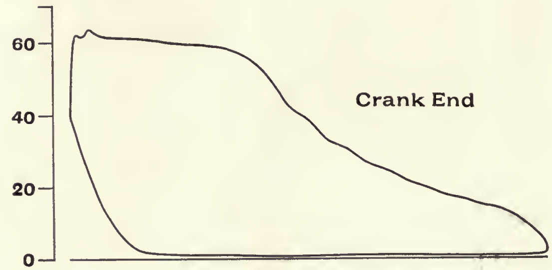
From these figures it appears that the use of the condenser secured a reduction in the weight of steam consumed amounting to 24%. This comparison is made under conditions of a comparatively low boiler pressure.

ENGINE No. 17a



ENGINE No. 17b





ENGINE No. 18.

Simple Condensing Engine.

Kind of engine	Four valve (Corliss)
Number of cylinders	2
Diameter of each cylinder	20¼ ins.
Diameter of each piston rod	3 ins.
Stroke of each piston	4 ft.
Clearance	3.4 %
H.P. Constant for one lb. m.e.p., one rev. per minute	.0772
Inside diameter of steam pipe	6 ins.
Condition of valves and pistons regarding leakage	Fairly tight

Data and Results of Feed-Water Tests, Engine No. 18.

Test. Cylinders in Use.		A. One.	B. Both.
Character of steam		Ordinary	Ordinary
Duration	hrs.	5.867	5.844
Weight of feed-water consumed	lbs.	24,310.	25,045.
Feed-water consumed per hour	lbs.	4,143.5	4,285.6
Pressure in steam pipe above atmosphere .	lbs.	84.5	59.
Vacuum in condenser	ins.	26.4	25.9
Mean effective pressure	lbs.	43.2	21.84
Revolutions per minute		61.16	61.8
Indicated horse-power	I.H.P.	204.02	208.45
Feed-water consumed per I.H.P. per hour	lbs.	20.31	20.56

Measurements based on Sample Diagrams.

Test. Cylinders in Use.		A. One.	B. Both.
Initial pressure above atmosphere . . .	lbs.	82.7	56.8
Steam-pipe pressure above atmosphere . .	lbs.	85.3	60.5
Cut-off pressure above zero	lbs.	89.3	64.9
Release pressure above zero	lbs.	18.0	8.7
Mean effective pressure	lbs.	42.51	21.72
Back pres. at mid stroke below atmosphere .	lbs.	— 11.7	—11.9
Proportion of stroke completed at cut-off .		.188	.111
Steam accounted for at cut-off	lbs.	14.73	13.73
Steam accounted for at release	lbs.	14.8	14.61
Proportion accounted for at cut-off . . .		.725	.668
Proportion accounted for at release . . .		.729	.711

Engine No. 18 has a pair of cylinders with a jet condenser operated by a direct connected air-pump. Steam is furnished by return tubular boilers, and calorimeter tests showed that the percentage of moisture varied from $\frac{1}{2}$ to 1 per cent. The steam valves were fairly tight. The piston and exhaust valves of one cylinder were absolutely tight. Those of the other cylinder leaked a trifle. The load consisted of cotton machinery.

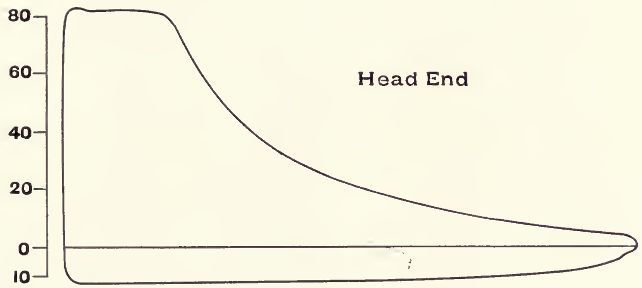
Two tests were made, one with both cylinders in operation and the other with a single cylinder, and in both the load was practically the same. The tests were made with different boiler pressures.

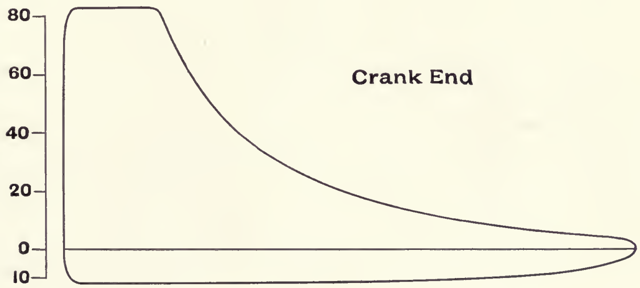
In this case it appears that the economy of feed-water consumption was practically the same whether one cylinder was used or the whole engine. As would be expected, however, the proportion of steam accounted for by the indicator was the least in the case of the earlier expansion.

In a series of experiments, of which these formed a part, a test was made to determine the effect of increasing the boiler pressure 20 lbs. above the normal, one cylinder being in use. In one case the pressure was 85.8 lbs., and in the other 105.7; and the mean effective pressure was, in round numbers, 41 lbs. in both cases. The cut-off occurred at $\frac{17}{100}$ of the stroke in one, and $\frac{13}{100}$ of the stroke in the other. The steam consumption with the high pressure was 19.5 lbs. per I. H. P. per hour, and with the low pressure 19.2. In other words, there was a trifling loss due to the increase of pressure. This engine was not absolutely tight, and doubtless leakage affected the results, so that the advantage of the increase of pressure was to some extent counteracted.

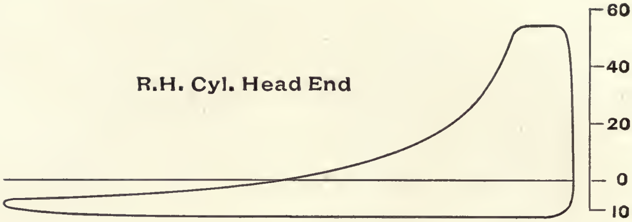
On the last mentioned test the steam accounted for was .67.

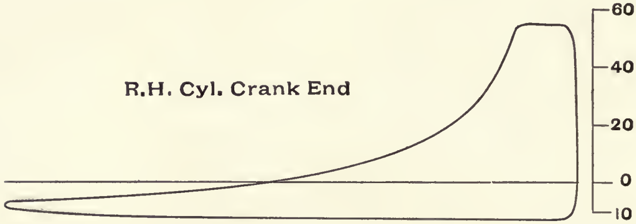
ENGINE No. 18a

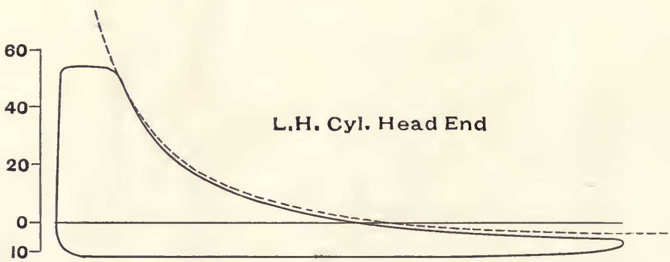


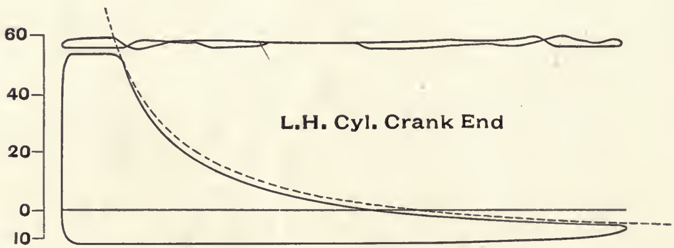


ENGINE No. 18b









ENGINE No. 19.

Simple Condensing Engine.

Kind of engine	Single Valve	
Number of cylinders	1	
Diameter of cylinder	18.5	in.
Diameter of piston rod	2⅞	in.
Stroke of piston	30.	in.
Clearance	7.5	%
H. P. Constant for one lb. m. e. p. one rev. per minute	.0405 H.P.	
Condition of valves and pistons regarding leakage	Some leakage	

Data and Results of Feed-Water Test.

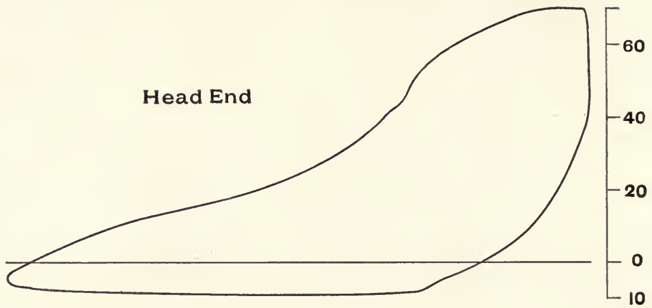
Character of steam	Ordinary	
Duration	5.011	hrs.
Weight of feed-water consumed	27,838.1	lbs.
Feed-water consumed per hour	5,555.4	lbs.
Pressure in steam pipe above atmosphere	74.5	lbs.
Vacuum in condenser	24.8	ins.
Mean effective pressure	39.05	lbs.
Revolutions per minute	129.33	
Indicated horse-power	204.59	H. P.
Feed-water consumed per I. H. P. per hour	27.15	lbs.

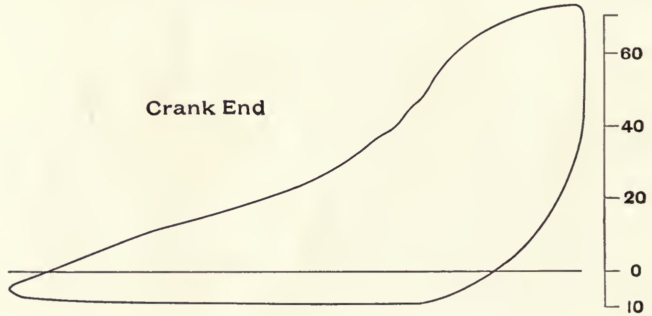
Measurements based on Sample Diagrams.

Initial pressure above atmosphere	69.3	lbs.
Steam-pipe pressure	73.2	lbs.
Cut-off pressure above zero	66.8	lbs.
Release pressure above zero	29.0	lbs.
Mean effective pressure	38.1	lbs.
Back pressure at mid stroke, below atmosphere	—8.9	lbs.
Proportion of stroke complete at cut-off	.303	
Steam accounted for at cut-off	19.17	lbs.
Steam accounted for at release	18.97	lbs.
Proportion accounted for at cut-off	.706	
Proportion accounted for at release	.699	

Engine No. 19 has a single unpacked piston valve, controlled by a shaft governor. The engine is provided with a jet condenser operated by an independent air-pump, driven by steam. The steam used by the condenser was determined separately. and allowance made for it in the record. Steam is supplied by horizontal return tubular boilers, and it is presumed that it was in a commercially dry condition. The piston of the engine was tight, but the valve placed at the middle of its throw showed considerable leakage. The load was cotton machinery.

ENGINE No. 19





ENGINE No. 20.

Simple Condensing.

Kind of engine	Four valve	
Number of cylinders	2	
Diameter of each cylinder	28	ins.
Diameter of each piston rod	4	ins.
Stroke of each piston	5	ft.
Clearance	3	%
H.P. constant for one lb. m.e.p. one rev. per minute	.3694 H.P.	
Inside diameter of steam pipe	9	ins.
Inside diameter of exhaust pipe	10	ins.
Condition of valves and pistons regarding leakage	Some leakage	

Data and Results of Feed-Water Tests.

Test. Conditions Regarding Use of Condenser.		A. Condensing.	B. Non-Condensing.
Character of steam		Ordinary.	Ordinary
Duration	hrs.	10.08	9.83
Weight of feed-water consumed	lbs.	102,947.	133,925.
Feed-water consumed per hour	lbs.	10,213.	13,620.
Pressure in steam pipe above atmosphere	lbs.	68.1	65.1
Vacuum in condenser	ins.	23.	
Mean effective pressure	lbs.	23.12	23.81
Revolutions per minute		52.	50 5
Indicated horse-power	I.H.P.	444.	451.6
Feed-water consumed per I.H.P. per hour	lbs.	23.	30.16

Measurements based on Sample Diagrams.

Test. Conditions Regarding Use of Condenser.		A. Condensing	B. Non-Condensing.
Initial pressure above atmosphere	lbs.	63.6	62.5
Steam-pipe pressure	lbs.	68.1	65.7
Cut-off pressure above zero	lbs.	71.2	70.1
Release pressure above zero	lbs.	9.5	16.8
Mean effective pressure	lbs.	23.25	23.97
Back pressure at mid stroke, above or below atmosphere	lbs.	—11.1	+1.5
Proportion of stroke completed at cut-off		.119	.222
Steam accounted for at cut-off	lbs.	14.13	21.8
Steam accounted for at release	lbs.	14.37	23.17
Proportion accounted for at cut-off		.615	.722
Proportion accounted for at release		.625	.766

Engine No. 20 has a pair of cylinders with gridiron unbalanced slide valves. It is fitted with a jet condenser, operated by an independent air-pump driven by steam. The engine was supplied from horizontal return tubular boilers, and the steam on subsequent occasions was found to contain 1.2 % of moisture. In the matter of leakage, the engine was in fair condition, though every valve, and the pistons as well, showed a small amount of leakage. The load was made up largely of rubber grinding machinery.

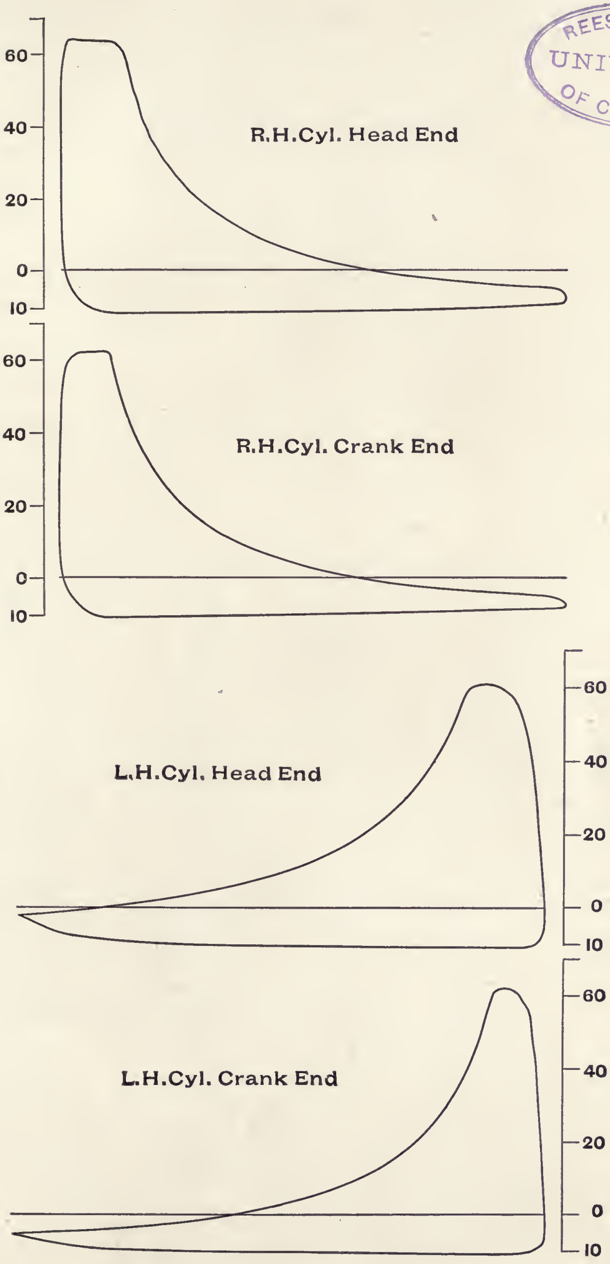
Two tests were made, one with the condenser in operation, and the other with the engine exhausting into the atmosphere, the condenser being stopped. Independent tests were made, showing the quantity of steam used by the air-pump; and allowance has been made for it. Besides the engine, the boiler supplied the feed-pump and a tank-pump, The steam thus used has not been allowed for.

The air-pump, which had a single steam cylinder 16″ in diameter and 24″ stroke, when making 56.3 single strokes per minute, was found to use 1682 lbs. of steam per hour. The power developed amounted to 12.8 H. P.; consequently the air-pump consumed 131.7 lbs. of steam per I. H. P. per hour. On the condensing test the air-pump used over 13 per cent of the entire quantity consumed by the engine.

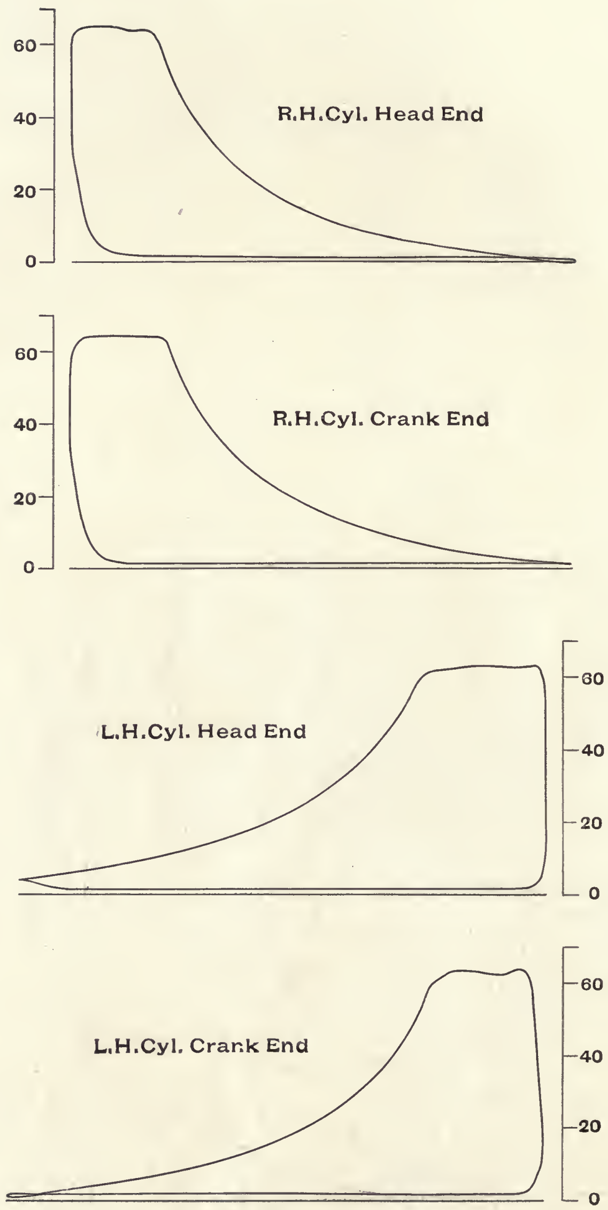
The quantity of steam used by the engine and air-pump working condensing was about 12 % less than that used when the engine was running non-condensing, and the quantity used by the engine alone about 24 % less.

In explanation of the comparatively low proportion of feed-water accounted for on the condensing test, it is probable that allowance made for steam used by the condenser was less than the actual quantity, owing to the fact that ordinarily the cylinder drips were kept partially open. On the condenser test, these were closed. Probably the actual consumption of feed-water was somewhat below the 23 lbs. given in the table, and the actual proportions referred to were somewhat higher. It should also be noted that the portion unaccounted for includes the steam used by the boiler-feed and tank-pump on both tests, probably 2 or 3 % of the whole.

ENGINE No. 20a



ENGINE No. 20b



ENGINE No. 21.

Simple Non-Condensing Engine.

Kind of engine	Four valve
Number of cylinders	1
Diameter of cylinder	11½ ins.
Diameter of piston rod	1⅝ ins.
Stroke of piston	20 ins.
Clearance	10 %
H. P. constant for one lb. m. e. p. one revolution per min.	.0104 H.P.
Inside diameter of steam pipe	4 in.
Condition of valves and piston regarding leakage . . .	Considerable leakage

Data and Results of Feed-Water Test.

Character of steam	Superheated 4°
Duration	8 hrs.
Weight of feed-water consumed	10,341 lbs.
Feed-water consumed per hour	1,292 lbs.
Pressure in steam pipe above atmosphere	64.5 lbs.
Mean effective pressure	15.7 lbs.
Revolutions per minute	198.3
Indicated horse-power	32.26 I.H.P.
Feed-water consumed per I. H. P. per hour	40.04 lbs.

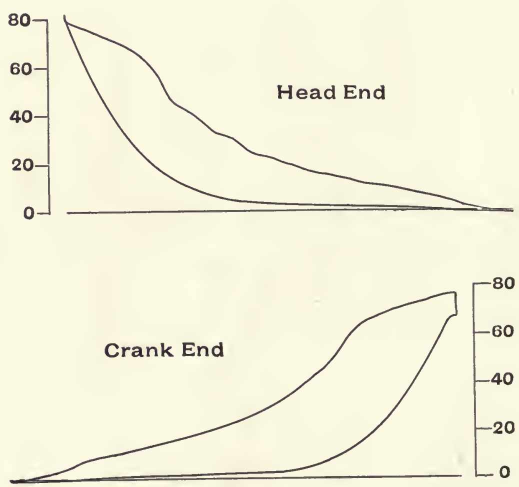
Measurements based on Sample Diagrams.

Initial pressure above atmosphere	63.1 lbs.
Corresponding steam-pipe pressure	67 lbs.
Cut-off pressure above zero	60.4 lbs.
Release pressure above zero	22.4 lbs.
Mean effective pressure	15.4 lbs.
Back pressure at mid stroke, above atmosphere	2 lbs.
Proportion of stroke completed at cut-off	.238
Steam accounted for at cut-off	24.85 lbs.
Steam accounted for at release	25.13 lbs.
Proportion of feed-water accounted for at cut-off	.62
Proportion of feed-water accounted for at release	.627

Engine No. 21 has balanced slide valves. Steam is furnished from a horizontal return tubular boiler of special design, which is provided with a considerable amount of steam-heating surface. The steam was superheated at the boiler 30°, and at a point near the engine 4°. One of the exhaust valves and the piston were fairly tight. The other steam valve and the other exhaust valve leaked very badly. The load consisted of a dynamo furnishing a steady current for electric lighting.

It is evident that leakage of the valves referred to had much to do with the poor showing.

ENGINE No. 21



ENGINE No. 22.

Simple Condensing Engine.

Kind of engine	Four valve	
Number of cylinders	1	
Diameter of cylinder	$34\frac{7}{16}$	ins.
Diameter of piston rod	5	ins.
Stroke of piston	5	ft.
Clearance	2	%
H.P. constant for one lb. m. e. p. one rev. per min.	.2791	H.P.
Inside diameter of steam pipe	14	ins.
Inside diameter of exhaust pipe	14	ins.
Condition of valves and piston regarding leakage	Some leakage	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	5.367	hrs.
Weight of feed-water consumed	60,879	lbs.
Feed-water consumed per hour	11,343	lbs.
Pressure in steam-pipe	82.3	lbs.
Vacuum in condenser	27.9	ins.
Mean effective pressure	37.23	lbs.
Revolutions per minute	59.9	
Indicated horse-power	613.4	I.H.P.
Feed-water consumed per I. H. P. per hour	18.49	lbs.

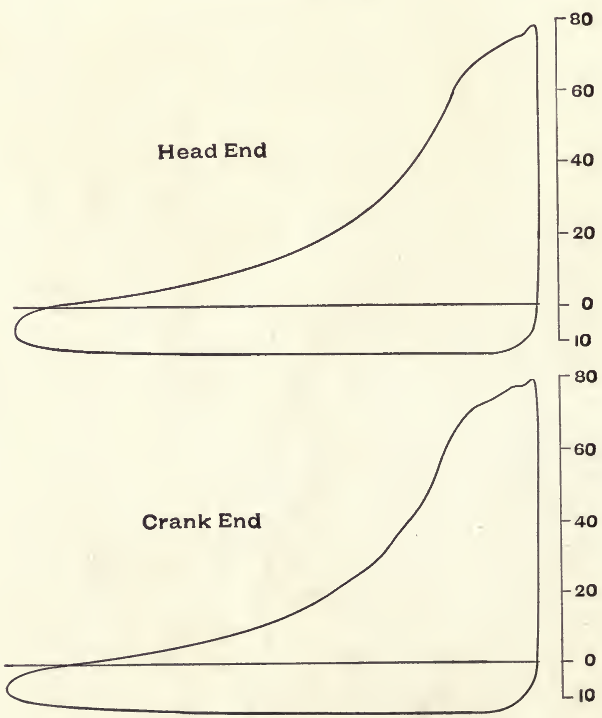
Measurements based on Sample Diagrams.

Initial pressure above atmosphere	81.9	lbs.
Corresponding steam-pipe pressure	84.2	lbs.
Cut-off pressure above zero	75.6	lbs.
Release pressure above zero	15.5	lbs.
Mean effective pressure	37.17	lbs.
Back pressure at mid stroke above or below atmosphere	13.6	lbs.
Proportion of stroke completed at cut-off	.172	
Steam accounted for at cut-off	12.39	lbs.
Steam accounted for at release	13.41	lbs.
Proportion of feed-water accounted for at cut-off	.669	
Proportion of feed-water accounted for at release	.725	

Engine No. 22 has slide valves of the gridiron type. It is provided with a siphon condenser, the injection water for which is furnished under natural head. Steam is supplied to the engine from water tube boilers through a 16-inch pipe 331 feet in length. At a point near the engine it is drained by means of a trap, which discharges to waste. On the test 143 lbs. of water were discharged per hour, and no allowance has been made for this. At a point between the trap and the engine the steam was found by calorimeter test to contain $2\frac{1}{2}$ per cent of moisture. The exhaust valves were practically tight. The steam valves and pistons showed some leakage. The engine worked in connection with a water-wheel driving cotton machinery.

In examining the results of this test, which in view of the long distance which the steam had to travel between the boilers and the engine, shows excellent economy, the part which the vacuum played cannot be overlooked. This was phenominally low, the back pressure at the middle of the stroke being only about one pound above a perfect vacuum.

ENGINE No. 22



ENGINE No. 23.

Simple Non-Condensing Engine.

Kind of engine	Single valve	
Number of cylinders	1	
Diameter of cylinder	8	ins.
Diameter of piston rod	$1\frac{7}{16}$	ins.
Stroke of piston	12	ins.
Clearance	14	%
H. P. constant for 1 lb. m. e. p. one revolution per min.	.00301	H.P.
Inside diameter of steam pipe	3	ins.
Inside diameter of exhaust pipe	$3\frac{1}{2}$	ins.
Condition of valves and piston regarding leakage	Fairly tight	

Data and Results of Feed-Water Tests.

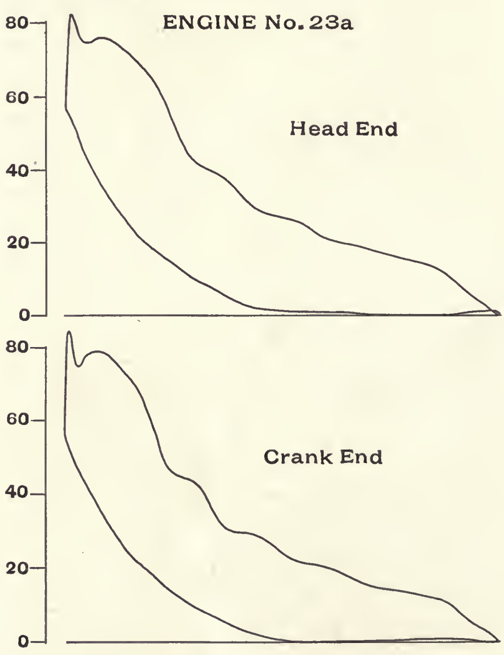
Tests.		A.	B.	C.
Character of steam		Ordinary	Ordinary	Ordinary
Duration	hrs.	3	4	4
Weight of feed-water consumed	lbs.	2,140	4,035	4,833
Feed-water consumed per hour	lbs.	713.3	1,008.8	1,208.2
Pressure in steam pipe above atmosphere	lbs.	83	82.4	81.9
Mean effective pressure	lbs.	24.53	34.86	42.99
Revolutions per minute		303.7	307.8	304.5
Indicated horse-power	I.H.P.	22.45	32.33	39.44
Feed-water consumed per I. H. P. per hour	lbs.	31.78	31.2	30.63

Measurements based on Sample Diagrams.

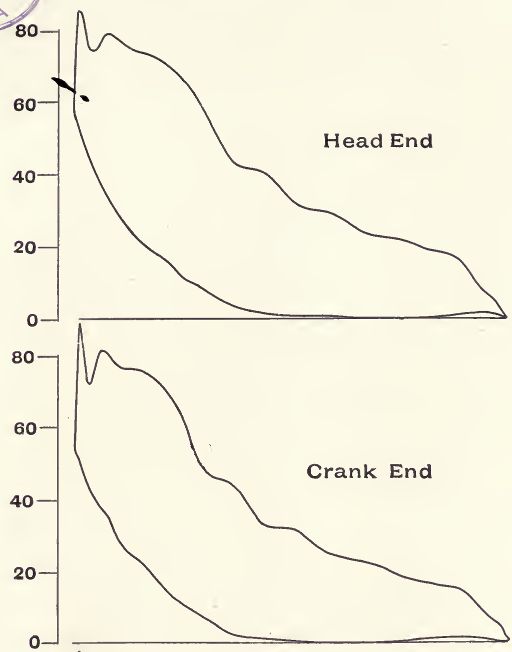
Tests.		A.	B.	C.
Initial pressure above atmosphere	lbs.	78.8	77.3	79.5
Corresponding steam-pipe pressure	lbs.	83	82	81.8
Cut-off pressure above zero	lbs.	76.2	75.5	78.1
Release pressure above zero	lbs.	27.4	32.5	37.3
Mean effective pressure	lbs.	24.61	34.79	43.12
Back pressure at mid stroke above atmosphere	lbs.	.8	.7	.7
Proportion of stroke completed at cut-off		.203	.312	.378
Steam accounted for at cut-off	lbs.	20.4	22.48	23.33
Steam accounted for at release	lbs.	21.66	21.76	22.24
Proportion of feed water accounted for at cut-off		.642	.72	.762
Proportion of feed water accounted for at release		.681	.698	.726

Engine No. 23 has a single-slide valve which is balanced by means of a pressure-plate riding on the back, and the cut-off is made automatic through the action of a shaft governor. Steam is supplied from a horizontal return tubular boiler. A calorimeter test showed that it was practically dry. The piston was fairly tight. The valve showed some leakage. The load consisted of a Sturtevant Blower. A series of tests were made under conditions of different loads, but practically constant boiler pressure.

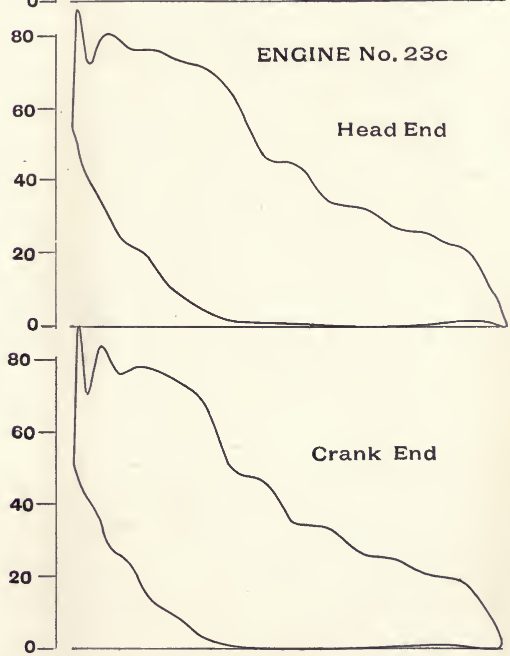
Another test in the same series with a load of 28.44 I. H. P., which is intermediate between the first and the second, gave a feed-water consumption of 31.46 lbs. per I. H. P. per hour, and the proportions of steam accounted for were respectively .685 and .694. In this series of tests the gradual improvement in the economy as the load is increased is a noticeable feature, as is also the uniform increase in the proportion of steam accounted for at the cut-off. Another point to be noticed is that as the cut-off becomes later, the amount of steam present at the release compared with that at cut-off is gradually reduced. In the first experiment the steam at release is the greater of the two, while in the last it is the smaller.



ENGINE No. 23b



ENGINE No. 23c



ENGINE No. 24.

Simple Non-Condensing Engine.

Kind of engine	Single valve	
Number of cylinders	1	
Diameter of cylinder	14.5	ins.
Diameter of piston rod	$2\frac{1}{4}$	ins.
Stroke of piston	13	ins.
Clearance	12	%
H. P. constant for 1 lb. m. e. p. one revolution per min.	.0107	HP.
Inside diameter of steam pipe	5	ins.
Condition of valves and piston regarding leakage	Fairly tight	

Data and Results of Feed-Water Test.

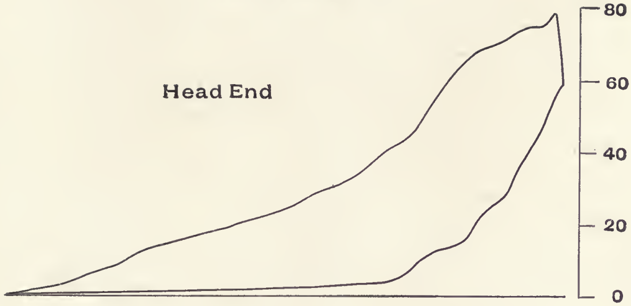
Character of steam	Ordinary	
Duration	4.45	hrs.
Weight of feed-water consumed	8,983	lbs.
Feed-water consumed per hour	2,018.6	lbs.
Pressure in steam pipe above atmosphere	80.3	lbs.
Mean effective pressure	23.22	lbs.
Revolutions per minute	248.4	
Indicated horse-power	61.7	I.H.P.
Feed-water consumed per I. H. P. per hour	32.71	lbs.

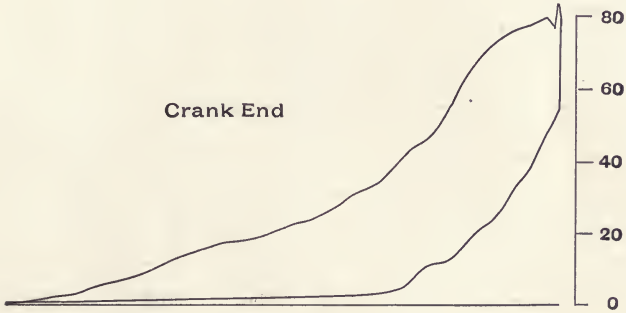
Measurements based on Sample Diagrams.

Initial pressure above atmosphere	78.4	lbs.
Corresponding steam-pipe pressure	80.5	lbs.
Cut-off pressure above zero	74.1	lbs.
Release pressure above zero	29.6	lbs.
Mean effective pressure	23.43	lbs.
Back pressure at mid stroke above atmosphere	2.6	lbs.
Proportion of stroke completed at cut-off	.211	
Steam accounted for at cut-off	21.66	lbs.
Steam accounted for at release	23.57	lbs.
Proportion of feed-water accounted for at cut-off	.662	
Proportion of feed-water accounted for at release	.721	

Engine No. 24 is of the high-speed type, with an unpacked piston valve controlled by a shaft governor. Steam was supplied from a horizontal return tubular boiler in what was believed to be a commercially dry state. The valve was new, well fitted, and fairly tight. The piston was also fairly tight. The load consisted mainly of machinery for the manufacture of woolen yarns.

ENGINE No. 24





ENGINE No. 25.

Simple Condensing Engine.

Kind of engine	Four valve (Corliss)	
Number of cylinders	2	
Diameter of each cylinder	26	ins.
Diameter of each piston rod	3¾	ins.
Stroke of each piston	5	ft.
Clearance	3	%
H. P. constant for one lb. m. e. p. one revolution per minute each cylinder	.16	H.P.
Inside diameter of steam pipe	8	ins.
Inside diameter of exhaust pipe	10	ins.
Condition of valves and pistons regarding leakage	Some leakage	

Data and Results of Feed Water Test.

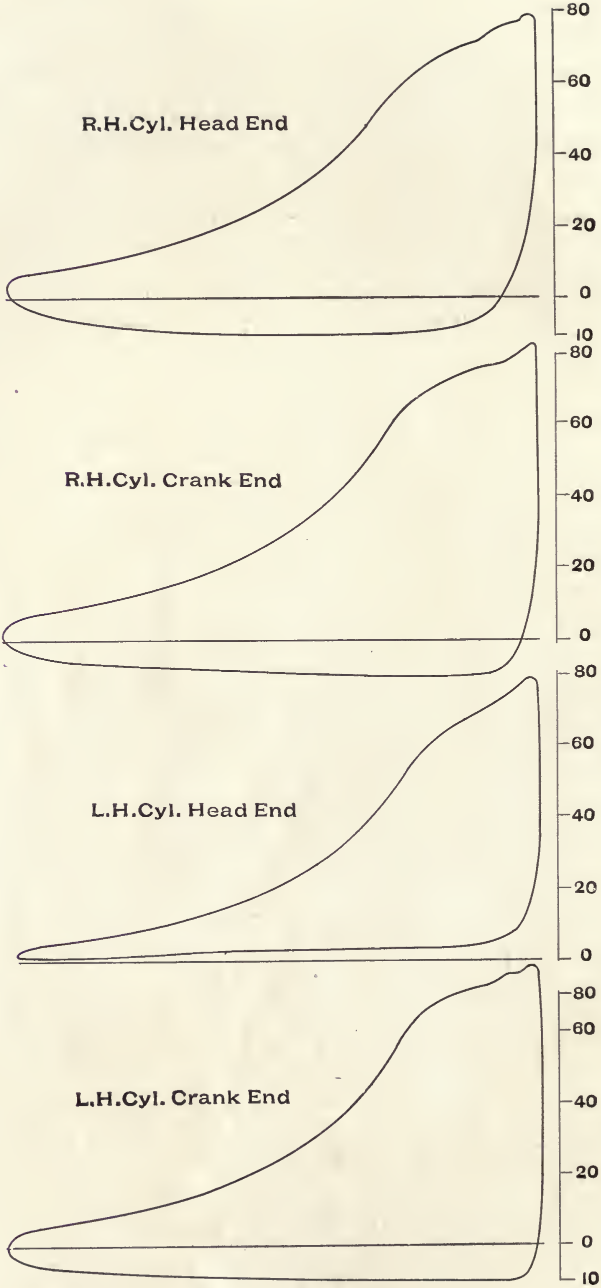
Character of steam	Ordinary	
Duration	1.75	hrs.
Weight of feed-water consumed	24,416	lbs.
Feed-water consumed per hour	13,948	lbs.
Pressure in steam pipe above atmosphere	82.9	lbs.
Vacuum in condenser	25.9	ins.
Mean effective pressure	36.29	lbs.
Revolutions per minute	53.9	
Indicated horse-power	625.6	I.H.P.
Feed-water consumed per I. H. P. per hour	22.29	lbs.

Measurements Based on Sample Diagrams.

		Three Ends Condensing.	One End Non-Condensing.
Initial pressure above atmosphere	lbs.	79.9	79.7
Corresponding steam-pipe pressure	lbs.	83.4	83.4
Cut-off pressure above zero	lbs.	73.2	66.2
Release pressure above zero	lbs.	19.2	17.6
Mean effective pressure	lbs.	40.25	23.94
Back pressure at mid stroke, above or below atmosphere	lbs.	− 10.	+ 3.5
Proportion of stroke completed at cut-off		.245	.24
Steam accounted for at cut-off	lbs.	15.49	21.17
Steam accounted for at release	lbs.	15.59	22.23
Proportion of feed-water accounted for at cut-off, average		.737	
Proportion of feed-water accounted for at release, average		.740	

Engine No. 25 has a pair of horizontal cylinders exhausting into a jet condenser which is operated by a direct connected air-pump. The cylinders were arranged for running one end of one cylinder non-condensing, and it was under these conditions that the tests were made. Steam is furnished by cylinder boilers, and it is presumed that it was in a commercially dry condition. When the water was carried at an unusually low point, a small portion of the shell became steam-heating surface, and the steam was found to be slightly superheated. Two of the steam valves showed some leakage. The pistons also leaked a small amount. The remaining valves were fairly tight. The load consisted of cotton machinery.

ENGINE No. 25


R.H.Cyl. Head End
80
60
40
20
0
10
R.H.Cyl. Crank End
80
60
40
20
0
L.H.Cyl. Head End
80
60
40
20
0
L.H.Cyl. Crank End
80
60
40
20
0
10

ENGINE No. 26.

Simple Non-Condensing Engine.

Kind of engine	Four valve	
Number of cylinders	1	
Diameter of cylinder	16¼	ins.
Diameter of piston rod	2¾	ins.
Stroke of piston	3	ft.
Clearance	6	%
H.P. Constant for one lb. m.e.p. one rev. per min.	.03717	H.P.
Condition of valves and piston regarding leakage	Leakage Test A	

Data and Results of Feed-Water Tests.

Test.	A.	B.
Character of steam	Ordinary	Ordinary
Duration . . . . hrs.	3.117	3.
Weight of feed-water consumed. . . lbs.	7,207.	6,221.
Feed-water consumed per hour . . . lbs.	2,312.2	2,073.7
Pressure in steam pipe above atmosphere . . . . lbs.	74.2	74.
Mean effective pressure . . . . . . lbs.	25.17	24.89
Revolutions per minute	75.8	76.03
Indicated horse-power . . . . . . I.H.P.	70.9	70.6
Feed-water consumed per I.H.P. per hour, lbs.	32.61	29.37

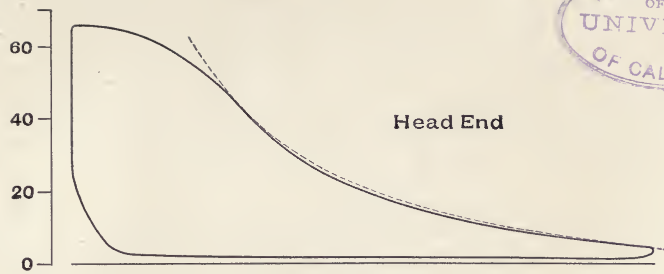
Measurements based on Sample Diagrams.

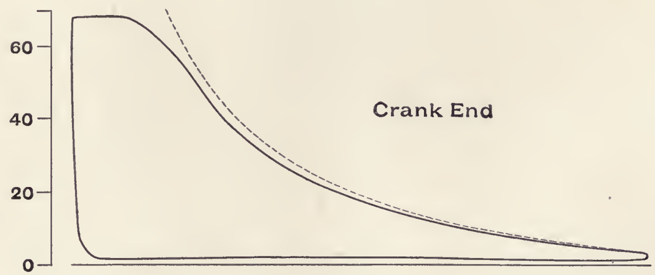
Test.	A.	B.
Initial pressure above atmosphere . . . lbs.	66.2	66.5
Corresponding steam-pipe pressure . . . lbs.	74.2	74.
Cut-off pressure above zero . . . . lbs.	64.7	65.5
Release pressure above zero . . . . . lbs.	18.5	18.3
Mean effective pressure . . . . lbs.	25.17	24.89
Back pres. at mid stroke above atmosphere . lbs.	1.9	1.7
Proportion of stroke completed at cut-off .	.237	.223
Steam accounted for at cut-off . . . . lbs.	22.55	21.87
Steam accounted for at release . . . . lbs.	24.65	24.64
Proportion of feed-water accounted for at cut-off	.692	.745
Proportion of feed-water accounted for at release	.755	.839

Engine No. 26 has double poppet steam valves, and plain slide exhaust valves. Steam is drawn from horizontal return tubular boilers. The load was a machine shop. The valves and piston were practically tight on test B. On test A the exhaust valve leaked badly, and during the interval between the two it was repaired.

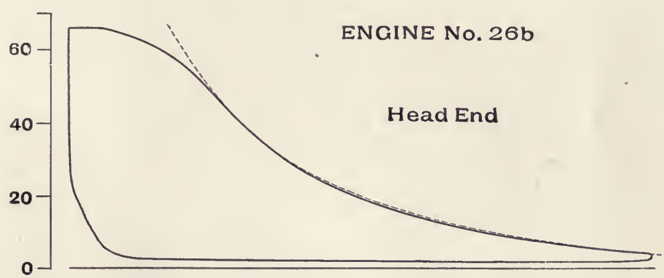
The effect of exhaust valve leakage on the economy of the engine is here clearly revealed. The tighter engine used about 10% less steam. The effect of the leakage upon the lines of the diagrams is hardly noticeable.

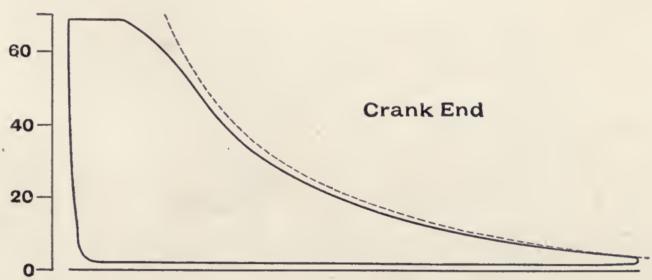
ENGINE No. 26a





ENGINE No. 26b





ENGINE No. 27.

Simple Non-Condensing Engine.

Kind of engine	Single valve	
Number of cylinders	1	
Diameter of cylinder	12^1	ins.
Diameter of piston rod	2_4	ins.
Stroke of piston	20	ins.
Clearance	8	%
H. P. constant for 1 lb. m. e. p. one revolution per minute	.01219	H.P.
Condition of valves and piston regarding leakage	Considerable leakage	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	3.083	hrs.
Weight of feed-water consumed	4,374.5	lbs.
Feed-water consumed per hour	1,418.9	lbs.
Pressure in steam pipe above atmosphere	72.2	lbs.
Mean effective pressure	18.15	lbs.
Revolutions per minute	172.3	
Indicated horse-power	38.1	I.H.P.
Feed-water consumed per I. H. P. per hour	37.21	lbs.

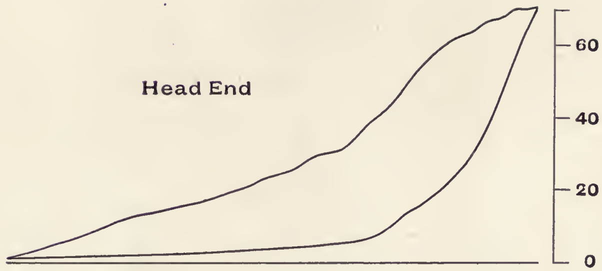
Measurements based on Sample Diagrams.

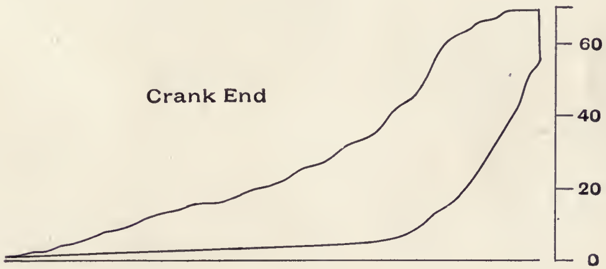
Initial pressure above atmosphere	68.8	lbs.
Corresponding steam-pipe pressure	72.2	lbs.
Cut-off pressure above zero	66.5	lbs.
Release pressure above zero	26.4	lbs.
Mean effective pressure	18.15	lbs.
Back pressure at mid stroke above atmosphere	3.5	lbs.
Proportion of stroke completed at cut-off	.219	
Steam accounted for at cut-off	21.13	lbs.
Steam accounted for at release	26.74	lbs.
Proportion of feed-water accounted for at cut-off	.568	
Proportion of feed-water accounted for at release	.719	

Engine No. 27 is of the high-speed class, with unpacked piston valve operated through a shaft governor. The boiler is of the horizontal return tubular type. The leakage of the engine was confined mainly to the valve. The load consisted of machine tools.

The engine being located at a distance of some 75 ft. from the boiler, the condition of the steam was not so favorable for economy as it might otherwise have been. Doubtless this explains in part the poor showing.

ENGINE No. 27





ENGINE No. 28.

Simple Condensing Engine.

Kind of engine	Four valve (Corliss)	
Number of cylinders	1	
Diameter of cylinder	32¼	ins.
Diameter of piston rod	4½	ins.
Stroke of piston	5	ft.
Clearance	2½	%
H. P. Constant for one lb. m.e.p., one rev. per minute	.2451	H.P.
Condition of valves and piston regarding leakage	Some leakage	

Data and Results of Feed-Water Test.

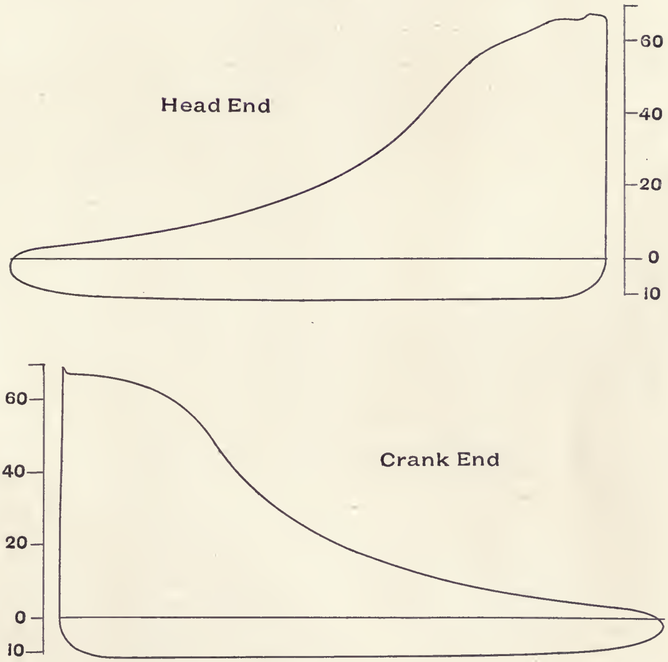
Character of steam	Ordinary	
Duration	5.1	hrs.
Weight of feed-water consumed	55,001	lbs.
Feed-water consumed per hour	10,784.5	lbs.
Pressure in steam pipe above atmosphere	70.1	lbs.
Vacuum in condenser	24.2	ins.
Mean effective pressure	38.26	lbs.
Revolutions per minute	59.13	
Indicated horse-power	554.4	I.H.P.
Feed-water consumed per I. H. P. per hour	19.45	lbs.

Measurements based on Sample Diagrams.

Initial pressure above atmosphere	67.5	lbs.
Corresponding steam-pipe pressure	70.1	lbs.
Cut-off pressure above zero	61.5	lbs.
Release pressure above zero	17.6	lbs.
Mean effective pressure	38.26	lbs.
Back pressure at mid stroke, below atmosphere	11.6	lbs.
Proportion of stroke completed at cut-off	.271	
Steam accounted for at cut-off	15.20	lbs.
Steam accounted for at release	15.27	lbs.
Proportion of feed-water accounted for at cut-off	.781	
Proportion of feed-water accounted for at release	.785	

Engine No. 28 exhausts into a jet condenser, with direct connected air-pump. The boilers are of the horizontal return tubular type. The steam valves were practically tight. There was some small amount of leakage of the piston, and a trifling leakage of the exhaust valves. The load was cotton machinery.

ENGINE No. 28



ENGINE No. 29.

Simple Condensing Engine.

Kind of engine	Four valve (Corliss)	
Number of cylinders	2	
Diameter of each cylinder	28	ins.
Diameter of each piston rod	4	ins.
Stroke of each piston	5	ft.
Clearance	2½	%
H. P. Constant for one lb. m. e. p. one rev. per minute	.1846	H.P.
Condition of valves and pistons regarding leakage	Some leakage	

Data and Results of Feed-Water Test.

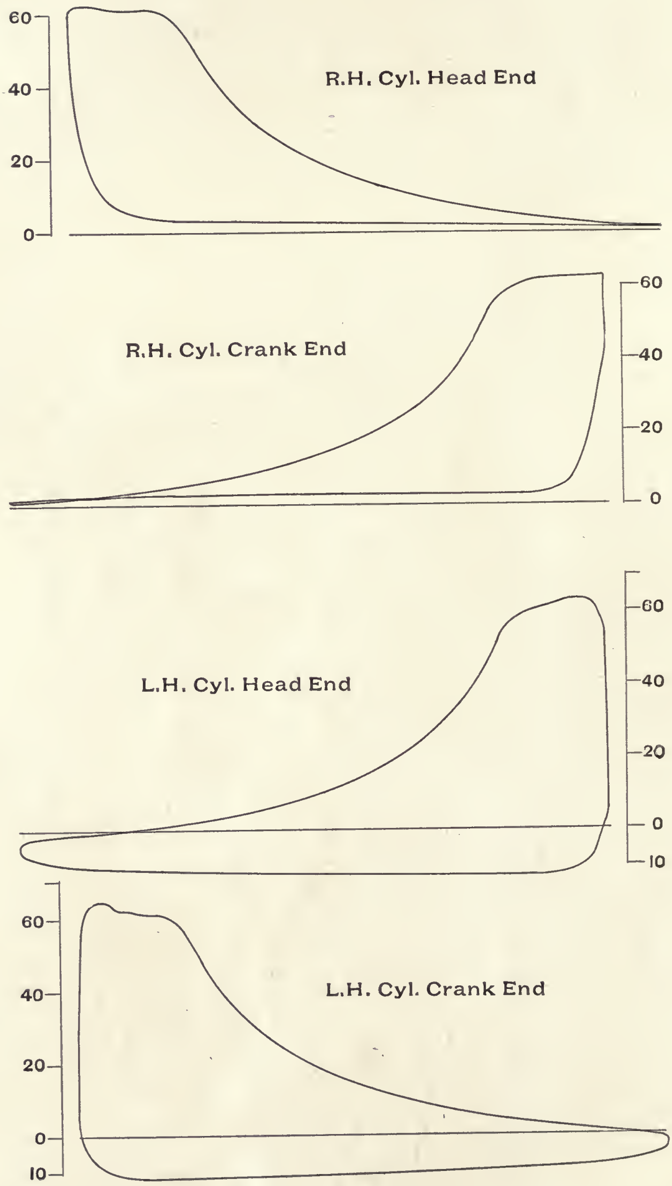
		R. H. Cylinder.	L. H. Cylinder.
Character of steam		Ordinary	
Duration	hrs.	5.32	
Weight of feed-water consumed	lbs.	76,053	
Feed-water consumed per hour	lbs.	14,295.7	
Pressure in steam-pipe above atmosphere .	lbs.	67.1	
Vacuum in condenser	ins.		27.3
Mean effective pressure	lbs.	19.26	31.26
Revolutions per minute		60.27	60.27
Indicated horse-power	I.H.P.	214.3	347.8
Indicated horse-power, whole engine .	I.H.P.	562.1	
Feed-water consumed per I. H. P. per hour	lbs.	25.43	

Measurements Based on Sample Diagrams.

		R. H. Cylinder	L. H. Cylinder.
Initial pressure above atmosphere . . .	lbs.	61.9	63.7
Corresponding steam-pipe pressure . . .	lbs.	67.1	
Cut-off pressure above zero	lbs.	67.7	67.6
Release pressure above zero	lbs.	16.6	13.9
Mean effective pressure	lbs.	19.26	31.26
Back pressure at mid stroke, above or below atmosphere	lbs.	+2.8	—12.1
Proportion of stroke completed at cut-off .		.202	.187
Steam accounted for at cut-off	lbs.	21.82	14.46
Steam accounted for at release	lbs.	24.31	14.86
Steam accounted for at cut-off, both cylinders, average	lbs.	17.27	
Steam accounted for at release, both cylinders, average	lbs.	18.37	
Proportion of feed-water accounted for at cut-off		.679	
Proportion of feed-water accounted for at release		.722	

Engine No. 29 has a pair of cylinders, one of which is non-condensing, and the other exhausts into a jet condenser with direct connected air-pump. Steam is drawn from horizontal return tubular boilers, and is presumably in a commercially dry state. There was a small amount of leakage in the valves of both cylinders, not only steam valves but exhaust valves, and some piston leakage. The engine operated a cotton-mill, working in connection with water-wheels.

ENGINE No. 29


R.H. Cyl. Head End
60
40
20
0
R.H. Cyl. Crank End
60
40
20
0
L.H. Cyl. Head End
60
40
20
0
10
L.H. Cyl. Crank End
60
40
20
0
10

ENGINE No. 30.

Simple Condensing Engine.

Kind of engine	Four valve	
Number of cylinders	2	
Diameter of cylinder	16	ins.
Diameter of piston rod	$2\frac{3}{8}$	ins.
Stroke of piston	3	ft.
Clearance	5	%
H. P. constant for one lb. m. e. p. one rev. per min., each	.0361	H.P.
Inside diameter of steam pipe	5	ins.
Inside diameter of exhaust pipe	6	ins.
Condition of valves and pistons regarding leakage	Some leakage	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	5.	hrs.
Weight of feed-water consumed	22,055	lbs.
Feed-water consumed per hour	4,411	lbs.
Pressure in steam pipe above atmosphere	83.4	lbs.
Vacuum in condenser	26.7	ins.
Mean effective pressure	31.27	lbs.
Revolutions per minute	90.4	
Indicated horse-power	205.9	I.H.P.
Feed-water consumed per I. H. P. per hour	21.42	lbs.

Measurements based on Sample Diagrams.

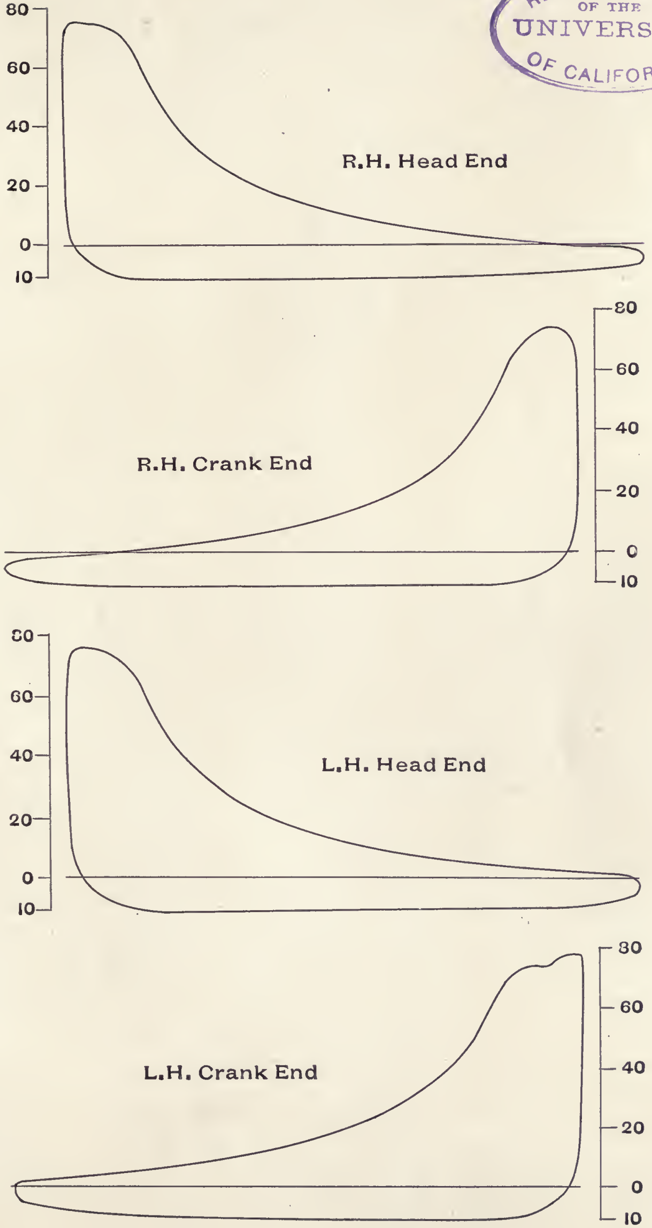
Initial pressure above atmosphere	74.4	lbs.
Cut-off pressure above zero	75.0	lbs.
Release pressure above zero	14.9	lbs.
Mean effective pressure	31.16	lbs.
Back pressure at mid stroke below atmosphere	—11.15	lbs.
Proportion of stroke completed at cut-off	.139	
Steam accounted for at cut-off	13.81	lbs.
Steam accounted for at release	16.45	lbs.
Proportion of feed-water accounted for at cut-off	.645	
Proportion of feed-water accounted for at release	.769	

Engine No. 30 has a pair of cylinders each having two steam valves and two exhaust valves, all being slide valves. The condenser is of the jet type operated by an independent air-pump driven by steam taken from the engine-pipe. The quantity of steam used by the condenser was determined by an independent test and allowed for. Steam is furnished by vertical water tube boilers, and a separator is fitted to the main steam pipe.

No water collected in the separator, and the steam is presumed to be commercially dry. The valves and pistons of each cylinder showed some leakage. The load consisted of dynamos furnishing current for electric lighting.

Engine No. 30 belongs to the same plant as Nos. 35 and 36, and it is supplied with steam from the same boiler plant.

ENGINE No. 30


80
60
40
20
0
10
R.H. Head End
80
60
40
20
0
10
R.H. Crank End
80
60
40
20
0
10
L.H. Head End
80
60
40
20
0
10
L.H. Crank End

ENGINE No. 31.

Simple Non-Condensing Engine.

Kind of engine	Four valve (Corliss)	
Number of cylinders	2	
Diameter of each cylinder	16	ins.
Diameter of each piston rod	$2\frac{3}{4}$	ins.
Stroke of each piston	42	ins.
Clearance	2.5	%
H.P. constant for one lb. m.e.p. one rev. per minute . .	.042	H.P.
Inside diameter of steam pipe	7	ins.
Condition of valves and pistons regarding leakage . . .	Fairly tight	

Data and Results of Feed-Water Tests.

Character of Load.		A. Light Load.	B. Heavy Load.
Character of steam		Ordinary	Ordinary
Duration	hrs.	4.	2.
Weight of feed-water consumed	lbs.	10,897.	17,746.
Feed-water consumed per hour	lbs.	2,724.2	8,873.
Pressure in steam pipe above atmosphere .	lbs.	101.8	98.6
Mean effective pressure	lbs.	5.03	48.4
Revolutions per minute		87.6	84.9
Indicated horse-power	I.H.P.	37.02	342.43
Feed-water consumed per I.H.P. per hour	lbs.	73.63	25.91

Measurements based on Sample Diagrams.

Character of Load.		A. Light Load.	B. Heavy Load.
Initial pressure above atmosphere . . .	lbs.	80.5	91.6
Cut-off pressure above zero	lbs.	81.9	94.8
Release pressure above zero	lbs.		31.4
Mean effective pressure	lbs.	5.4	49.26
Back pres. at mid stroke above atmosphere .	lbs.	2.	2.6
Proportion of stroke completed at cut-off .		.041	.323
Steam accounted for at cut-off	lbs.	28.16	20.63
Steam accounted for at release	lbs.		21.55
Proportion of feed-water accounted for at cut-off		.382	.796
Proportion of feed-water accounted for at release			.832

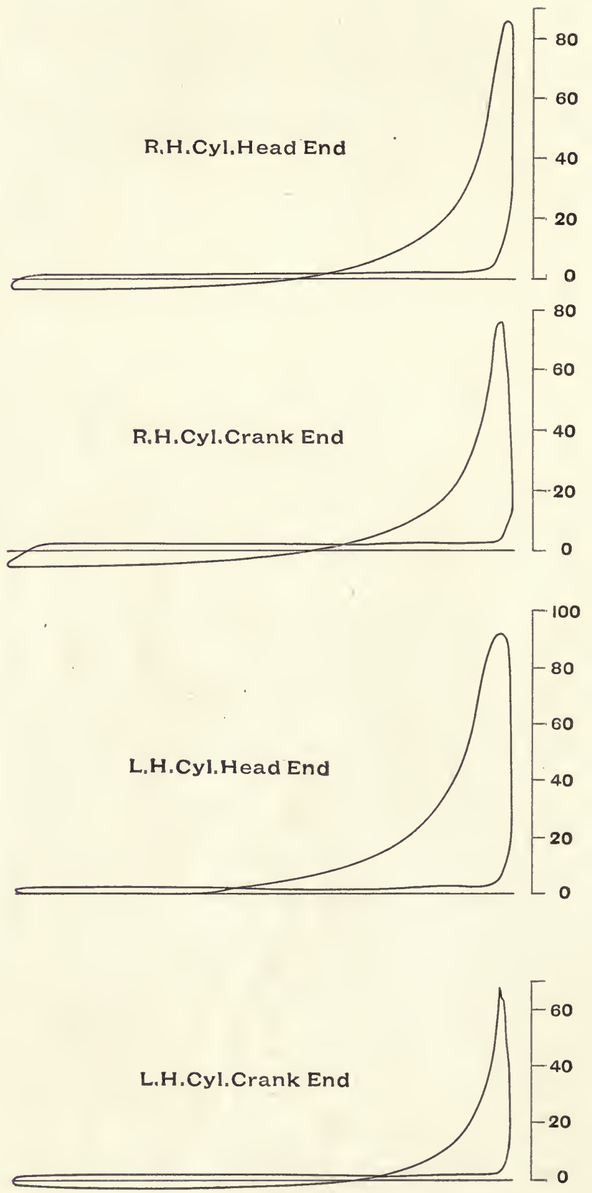
Engine No. 31 has a pair of cylinders drawing steam from horizontal return tubular boilers. There was only a small amount of leakage in any of the valves and pistons. The engine was employed in driving a line-shaft to which were belted dynamos supplying current for electric lighting.

The tests reported in the principal table were two in number, one of which was made with a friction load consisting of the shafting and empty dynamos, and the other with a full load.

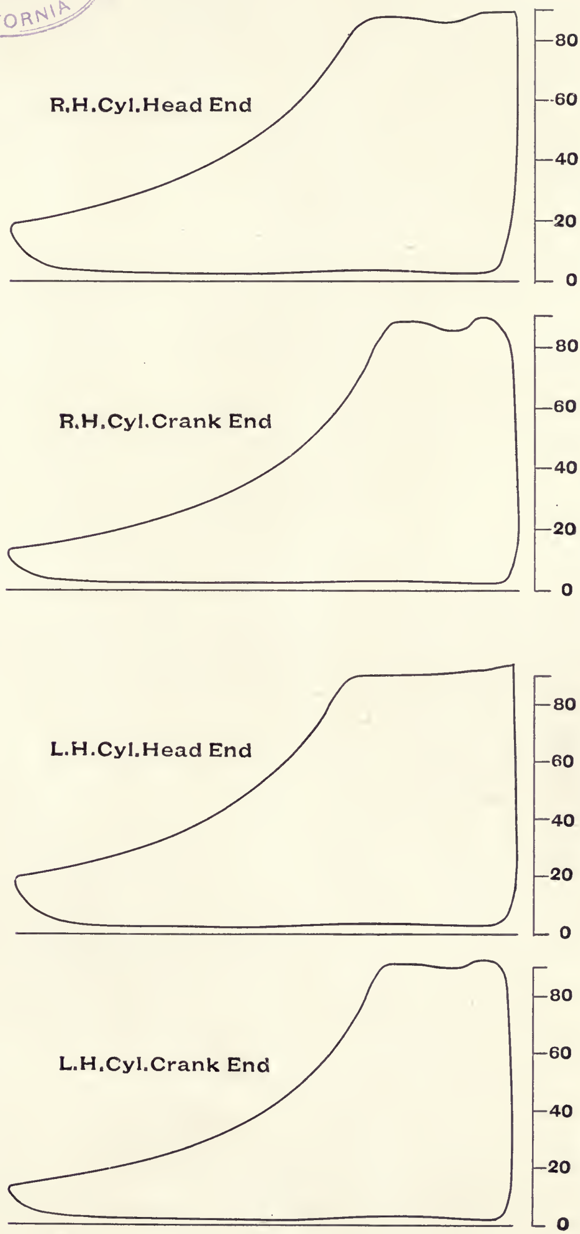
Tests on the same engine at intermediate loads gave the following principal results :

Indicated Horse Power.	Feed-Water per I.H.P. per Hour.	Proportion of Stroke Completed at Cut-off.	Proportion of Feed-Water Acct. for at Cut-off.
100.4	38.38	.084	.509
146.2	31.43	.121	.588
222.2	25.83	.178	.709
287.1	25.39	.231	.745

ENGINE No. 31a



ENGINE No. 31b



FEED-WATER TESTS.

COMPOUND ENGINES.

[*These engines are all of the automatic cut-off type, with fly-ball governor, unless otherwise stated.*]

ENGINE No. 32.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve (Corliss)	
Number of cylinders		1	1
Diameter of cylinders	ins.	26	48
Diameter of piston rod	ins.	4	5
Stroke of piston	ft.	4	4
Clearance	%	3	3
H. P. constant for 1 lb. m. e. p. one revolution per min.	H.P.	.159	.5454
Ratio of areas of cylinders		1	3.43
Inside diameter of steam pipe	ins.	8	14
Inside diameter of exhaust pipe	ins.	14	14
Condition of valves and pistons regarding leakage		Fairly tight	Tight

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	4.5	hrs.
Weight of feed-water consumed	*44,436	lbs.
Feed-water consumed per hour	9,874	lbs.
Pressure in steam pipe above atmosphere	94.8	lbs.
Pressure in receiver	6.4	lbs.
Vacuum in condenser	27.2	ins.
Revolutions per minute	52.3	
Mean effective pressure, H. P. cylinder	41.14	lbs.
Mean effective pressure, L. P. cylinder	9.27	lbs.
Indicated horse-power, H. P. cylinder	342.1	H.P.
Indicated horse-power, L. P. cylinder	264.43	H.P.
Indicated horse-power, whole engine	606.53	H.P.
Feed-water consumed per I. H. P. per hour	*16.28	lbs.

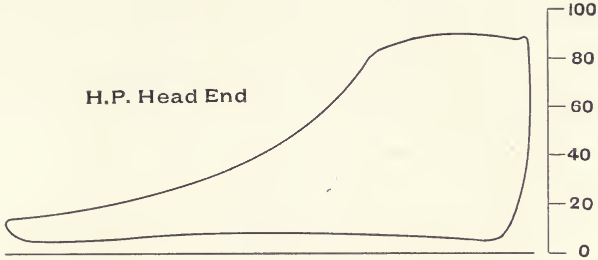
Measurements based on Sample Diagrams.

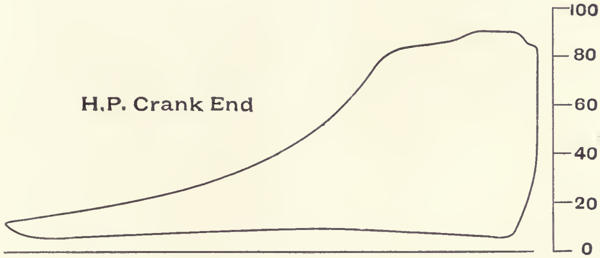
		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere	lbs.	89.4	6.0
Corresponding steam-pipe or receiver pressure	lbs.	94.2	6.3
Cut-off pressure above zero	lbs.	91.9	12.6
Release pressure above zero	lbs.	28.4	7.5
Mean effective pressure	lbs.	41.26	9.28
Back pressure at mid stroke, above or below atmosphere	lbs.	+ 8.1	− 10.8
Proportion of stroke completed at cut-off		.305	.544
Steam accounted for at cut-off	lbs.	12.60	11.78
Steam accounted for at release	lbs.	12.84	12.98
Proportion of feed-water accounted for at cut-off		.774	.723
Proportion of feed-water accounted for at release		.789	.797

* Includes steam used by circulating-pump.

Engine No. 32 is a cross-compound, having unjacketed horizontal cylinders and unjacketed receiver. A surface condenser is employed, and the air-pump is operated by direct connection with the engine. The circulating-pump is a duplex steam pump 9″ x 10″ x 12″, and the steam it used is included in that reported. The engine is furnished with steam from sectional boilers, and it is presumed to be in a commercially dry condition. In the matter of leakage the engine was in excellent condition throughout with the exception of the piston in the high-pressure cylinder, which leaked a small amount. The load consisted of a cotton-mill. The feed-water consumption was determined by measuring the water discharged by the air-pump.

ENGINE No. 32


H.P. Head End
100
80
60
40
20
0


H.P. Crank End
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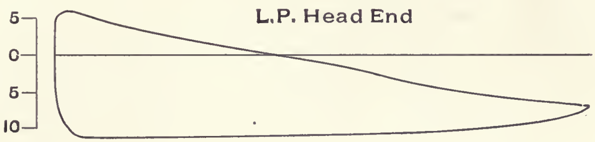
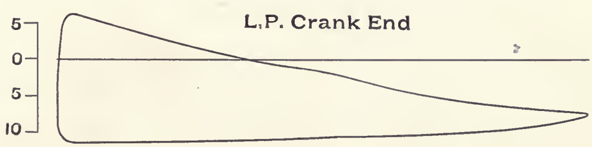
80
60
40
20
0


L.P. Head End
5
0
5
10


L.P. Crank End
5
0
5
10

ENGINE No. 33.

Compound Condensing Engine.

	H. P. Cylinder.	L. P. Cylinder.
Kind of engine	Single valve.	
Number of cylinders	1	1
Diameter of cylinder . . . . ins.	12	20
Stroke of piston . . . . ins.	12	12
Clearance . . . . %	33	9
H. P. Constant for one lb. m. e. p. one rev. per min. . . . . H.P.	.00342	.00952
Ratio of areas of cylinders . . .	1	2.78
Condition of valves and pistons regarding leakage	Tight.	Tight.

Data and Results of Feed-Water Tests.

Test. Conditions regarding use of Condenser.	A. Condensing	B. Non-Condensing
Character of steam	Ordinary	Ordinary
Duration . . . . hrs.	8	8
Weight of feed-water consumed . lbs.	34,555	41,562
Feed-water consumed per hour . lbs.	4,319.4	5,195
Press. in steam pipe above atmos. lbs.	129.3	128
Vacuum in condenser . . . . ins.	25	
Revolutions per minute	300.	296.1
Mean effective pressure, H.P. cyl. lbs.	53.53	57.21
Mean effective pressure, L.P. cyl. lbs.	20.73	20.34
Indicated horse-power, H.P. cyl. H.P.	109.84	115.85
Indicated horse-power, L. P. cyl. H.P.	118.41	114.69
Indicated H. P., whole engine . H.P.	228.25	230.54
Feed-water cons. per I. H.P. per hr. lbs.	18.92	22.53

The above are the totals and averages for the two engines.

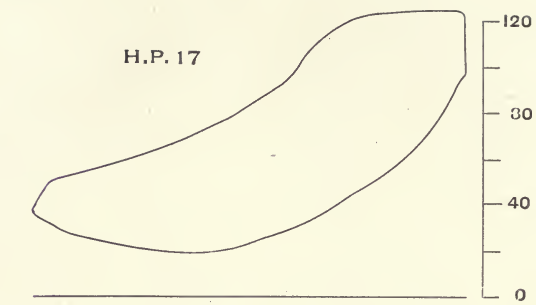
Measurements based on Sample Diagrams.

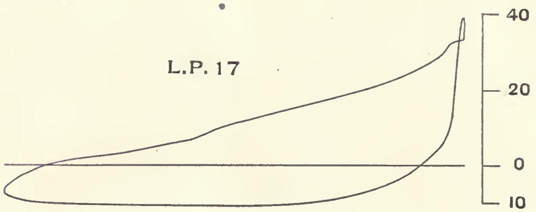
Tests.	H.P. Cyl.	L.P. Cyl.	H.P. Cyl.	L.P. Cyl.
Initial pressure above atmosphere lbs.	122.9	33.9	121.5	52.8
Corresponding steam-pipe pressure lbs.	132.		136.	
Cut-off pressure above zero . . lbs.	122.	27.1	124.3	35.8
Release pressure above zero . . lbs.	67.9	17.8	82.3	27.4
Mean effective pressure . . . lbs.	53.4	20.71	56.44	20.25
Back pressure at mid stroke above or below atmosphere . . lbs.	+20.3	−11.	+29.4	+1.1
Proportion of stroke completed at cut-off	.38	.521	.532	.66
Steam accounted for at cut-off . lbs.	15.21	12.14	20.13	16.54
Steam accounted for at release . lbs.	16.24	13.21	28.2	17.76
Proportion of feed water accounted for at cut-off	.804	.642	.894	.734
Proportion of feed water accounted for at release	.859	.700	.925	.789

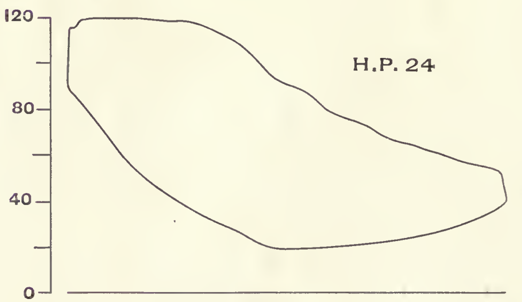
Engine No. 33 consists of two independent engines which were tested simultaneously. These engines are single-acting with vertical unjacketed cylinders, and provided with a single piston valve fitted with ring packing, one valve serving for both high- and low-pressure cylinders. A jet condenser is used which is common to both engines; and it is operated by an independent air-pump, which takes steam from the main supply pipe. The boiler feed-pump is also supplied from the main pipe. The quantity of steam used by these two pumps was determined by independent tests and allowed for. Steam is furnished by water tube boilers; and a calorimeter test showed in one case $\frac{7}{10}$ of 1% of moisture, and in the other $1\frac{1}{10}$%. The valves and pistons of both engines were practically tight. The load consisted of dynamos employed in electric lighting. One test was made with the engines running condensing, and another running non-condensing, the condenser being stopped.

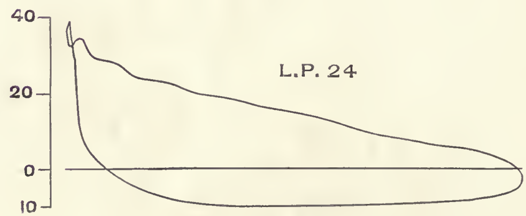
The difference in economy of these engines, due to the use of a condenser not allowing for steam used by air-pump, is represented by 3.61 lbs. of feed-water per I. H. P. per hour, which, in round numbers, is 20% of the quantity used when the engine was run condensing. The results of these tests cannot be passed by without noticing the marked difference in the porportion of steam accounted for at the cut-off under the two conditions of operation; and the loss of steam between the high-pressure cylinder and low-pressure cylinder in both cases.

ENGINE No. 33a

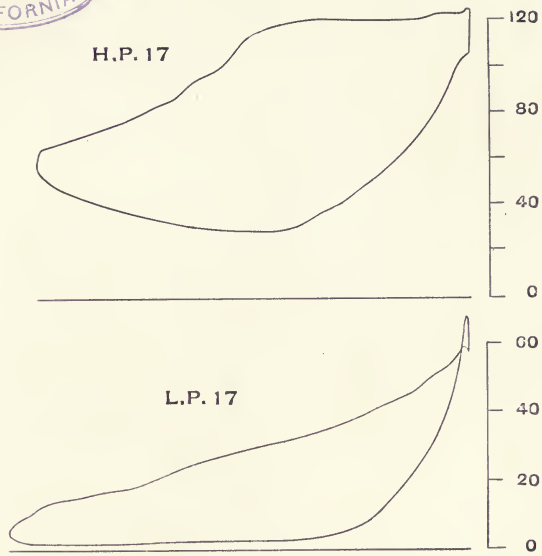


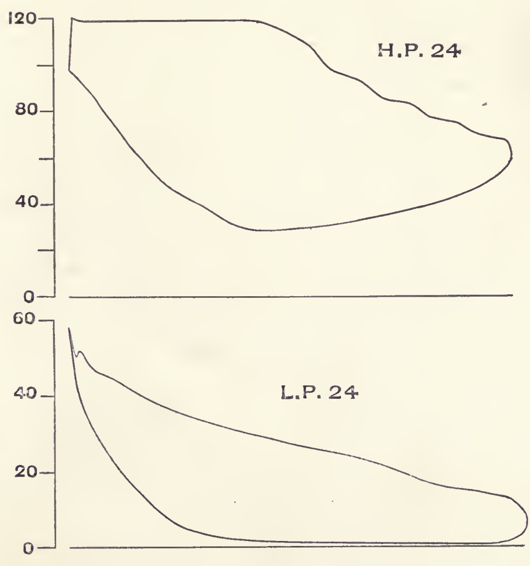






ENGINE No. 33b





ENGINE No. 34.

Compound Condensing Engine.

		H.P. Cylinder.	L.P. Cylinder.
Kind of engine		Four valve (Corliss)	
Number of cylinders		1	1
Diameter of cylinder	ins.	22	44
Diameter of piston rod	ins.	$3\frac{3}{8}$	$3\frac{5}{8}$
Stroke of piston	ft.	5	5
Clearance	%	2.2	5.9
H. P. constant for one lb. m. e. p. one rev. per min.	H.P.	.1138	.4592
Ratio of areas of cylinders		1	4.04
Inside diameter of steam pipe	ins.	7	13
Inside diameter of exhaust pipe	ins.	9	16
Condition of valves and pistons regarding leakage		Some leakage	Practically tight

Data and Results of Feed-Water Test.

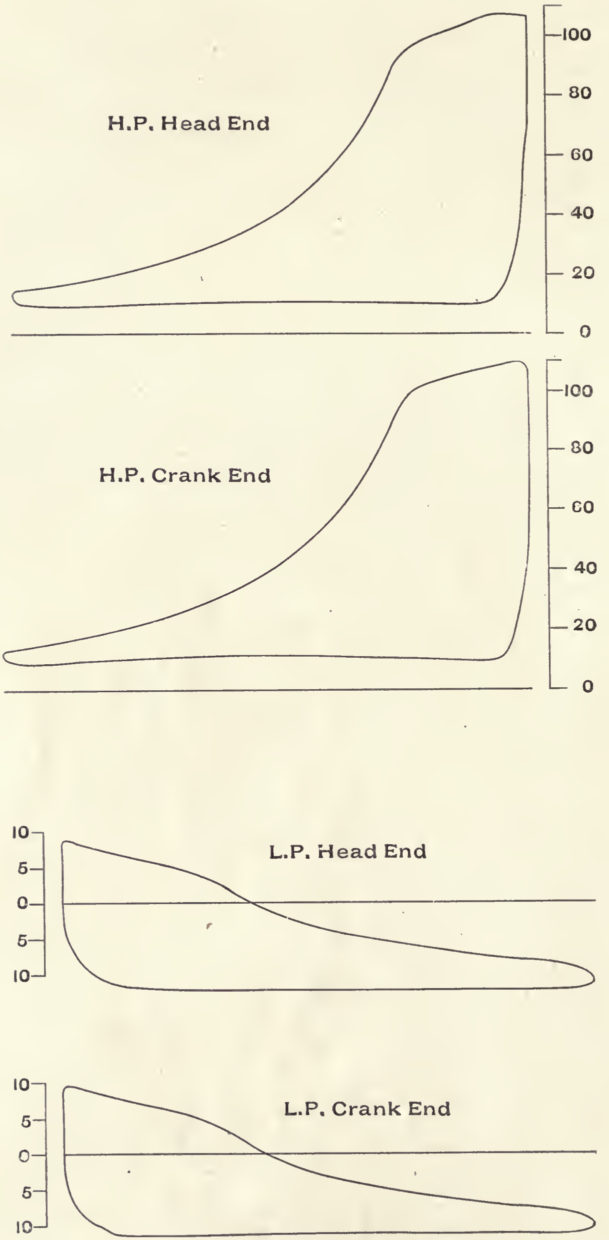
Character of steam	Ordinary	
Duration	4	hrs.
Weight of feed-water consumed	33,813	lbs.
Feed-water consumed per hour	8,453.2	lbs.
Pressure in steam pipe above atmosphere	116.1	lbs.
Pressure in receiver above atmosphere	7.5	lbs.
Vacuum in condenser	25.5	ins.
Revolutions per minute	68.08	
Mean effective pressure H. P. cylinder	41.48	lbs.
Mean effective pressure L. P. cylinder	10.08	lbs.
Indicated horse-power H. P. cylinder	321.39	H.P.
Indicated horse-power L. P. cylinder	315.09	H.P.
Indicated horse-power, whole engine	636.48	H.P.
Feed-water consumed per I. H. P. per hour	13.28	lbs.

Measurements based on Sample Diagrams.

		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere	lbs.	108.8	9.1
Corresponding steam-pipe or receiver pressure	lbs.	114.2	7.4
Cut-off pressure above zero	lbs.	104.	16.7
Release pressure above zero	lbs.	29.1	6.8
Mean effective pressure	lbs.	41.26	10.05
Back pressure at mid stroke above or below atmosphere	lbs.	+11.3	−11.8
Proportion of stroke completed at cut-off		.26	.325
Steam accounted for at cut-off	lbs.	10.48	10.06
Steam accounted for at release	lbs.	11.3	10.98
Proportion of feed-water accounted for at cut-off		.789	.758
Proportion of feed-water accounted for at release		.852	.827

Engine No. 34 is a cross compound with horizontal jacketed cylinders and unjacketed receiver. Steam supplied to the low-pressure cylinder first circulates through the jacket space, entering at the bottom at a central opening. The jackets are drained into tanks, which are emptied by means of pumps operated by the engine. The water of condensation from this source during the tests amounted to 600 lbs. per hour, or about 7 % of the total quantity of steam used by the engine. The condenser is of the jet type with a direct connected air-pump. Steam is supplied from horizontal return tubular boilers. A calorimeter test showed that the amount of moisture was $\frac{2}{10}$ of 1 %. The exhaust valves and pistons of both cylinders, and the steam valves of the low-pressure cylinder were found to be practically tight. The steam valves of the high-pressure cylinder showed some leakage. The load consisted of cotton machinery.

ENGINE No. 34


H.P. Head End
100
80
60
40
20
0
H.P. Crank End
100
80
60
40
20
0
L.P. Head End
10
5
0
5
10
L.P. Crank End
10
5
0
5
10

ENGINE No. 35.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Single valve.	
Number of cylinders		1	1
Diameter of cylinder	ins.	13	22
Diameter of piston rod	ins.	$1\frac{15}{16}$	$2\frac{1}{4}$
Stroke of piston	ins.	18	18
Clearance	%	7	10
Horse-power constant for one lb. m.e.p. one revolution per minute	H.P.	.0119	.0344
Ratio of areas of cylinders		1	2.89
Inside diameter of steam pipe	ins.	$4\frac{1}{2}$	
Inside diameter of exhaust pipe	ins.		6.
Condition of valves and pistons regarding leakage		Considerable leakage	Considerable leakage

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	5	hrs.
Weight of feed-water consumed	16,375	lbs.
Feed-water consumed per hour	3,275	lbs.
Pressure in steam pipe above atmosphere	105.2	lbs.
Vacuum in condenser	28	ins.
Revolutions per minute	197.1	
Mean effective pressure H. P. cylinder	32.57	lbs.
Mean effective pressure L. P. cylinder	10.63	lbs.
Indicated horse-power H. P. cylinder	76.4	H.P.
Indicated horse-power L. P. cylinder	72.1	H.P.
Indicated horse-power, whole engine	148.5	H.P.
Feed-water consumed per I. H. P. per hour	22.05	lbs.

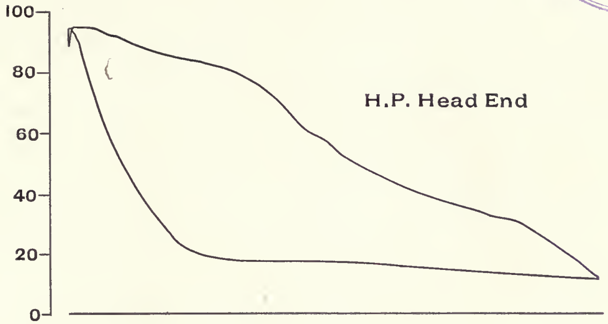
Measurements based on Sample Diagrams.

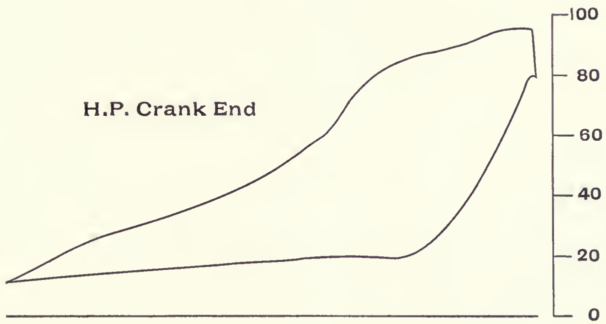
		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere	lbs.	96	11.5
Corresponding steam-pipe or receiver pressure	lbs.	104	
Cut-off pressure above zero	lbs.	84.6	16.9
Release pressure above zero	lbs.	43.6	11.
Mean effective pressure	lbs.	33.21	10.82
Back pressure at mid stroke above or below atmosphere	lbs.	+18	−9.5
Proportion of stroke completed at cut-off		.382	.505
Steam accounted for at cut-off	lbs.	13.73	13.93
Steam accounted for at release	lbs.	14.95	14.68
Proportion of feed-water accounted for at cut-off		.623	.632
Proportion of feed-water accounted for at release		.678	.666

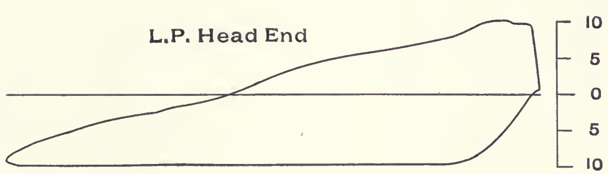
Engine No. 35 is a horizontal cross-compound unjacketed engine, provided with a shaft governor operating on the cut-off of the high-pressure cylinder. The valves are of the piston type without packing. A jet condenser is used operated by an independent air-pump driven with steam taken from the engine pipe. The quantity thus used, as also that consumed by the boiler feed-pump, was determined by independent tests and allowed for. Steam is supplied from vertical water-tube boilers, and a separator placed in the steam pipe secured what was believed to be commercially dry steam. The steam showed no superheating. The valve in each cylinder was found to leak badly. The piston of the high-pressure cylinder was fairly tight. Owing to the leakage of the low-pressure valve no leakage observations could be made upon the low-pressure piston. The load consisted of dynamos furnishing current for electric lighting.

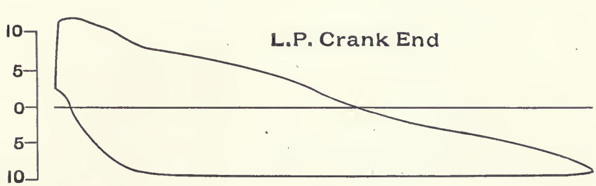
There is a close agreement between the steam accounted for by the indicator in the two cylinders, which might be surprising in view of the fact that the cylinders are unjacketed, were it not known that the steam valve of the high-pressure cylinder showed considerable leakage. Some of the steam shown on the low-pressure diagram was undoubtedly due to this cause.

ENGINE No. 35


H.P. Head End
100
80
60
40
20
0


H.P. Crank End
100
80
60
40
20
0


L.P. Head End
10
5
0
5
10


L.P. Crank End
10
5
0
5
10

ENGINE No. 36.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve	(Corliss)
Number of cylinders		1	1
Diameter of cylinder	ins.	16	32
Diameter of piston rod	ins.	3	$3\frac{11}{16}$
Stroke of piston	ft.	4	4
Clearance	%	2	4
H. P. constant for one lb. m. e. p. one revolution per minute	H.P.	.0479	.1937
Ratio of areas of cylinders		1.00	4.04
Inside diameter of steam pipe . . .	ins.	6	
Inside diameter of exhaust pipe . .	ins.		12
Condition of valves and pistons regarding leakage		Practically tight	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	5.05	hrs.
Weight of feed-water consumed	27,133	lbs.
Feed-water consumed per hour	5,373	lbs.
Pressure in steam pipe above atmosphere	126.8	lbs.
Pressure in receiver above atmosphere	8	lbs.
Vacuum in condenser	27.4	ins.
Revolutions per minute	74.9	
Mean effective pressure, H. P. cylinder	58.29	lbs.
Mean effective pressure, L. P. cylinder	11.79	lbs.
Indicated horse-power, H. P. cylinder	211.6	H.P.
Indicated horse-power, L. P. cylinder	170.9	H.P.
Indicated horse-power, whole engine	382.5	H.P.
Feed-water consumed per I. H. P. per hour	14.05	lbs.

Measurements based on Sample Diagrams.

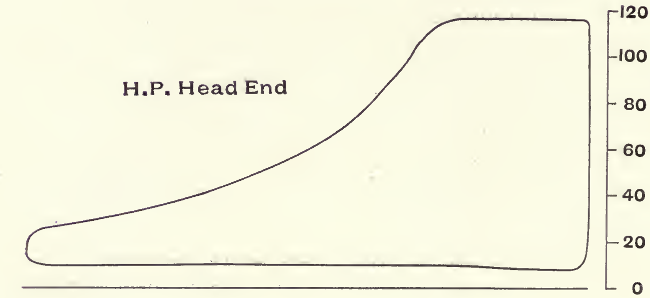
		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere . .	lbs.	116.5	7
Cut-off pressure above zero	lbs.	120.7	18.4
Release pressure above zero	lbs.	39.4	7.4
Mean effective pressure	lbs.	59.4	11.97
Back pressure at mid stroke, above or below atmosphere	lbs.	+9.9	−13.
Proportion of stroke completed at cut-off		.295	.337
Steam accounted for at cut-off . . .	lbs.	10.78	8.98
Steam accounted for at release . . .	lbs.	12.	10.42
Proportion of feed-water accounted for at cut-off		.767	.64
Proportion of feed-water accounted for at release		.853	.741

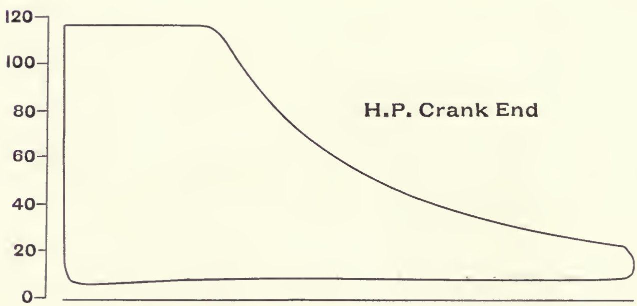
Engine No. 36 is a cross-compound horizontal engine with steam jacketed cylinders and a jet condenser operated by a direct connected air-pump. The jacket spaces in each cylinder form a thoroughfare through which the steam is supplied to the respective steam chests, the steam first entering the bottom of the jacket at a central point. During the test the drain-pipes provided for carrying off the water of condensation were closed, and all this water passed over into the cylinder. Whatever effect the jackets might otherwise have produced was thus nullified, and the engine may be considered as practically unjacketed. Steam is supplied from vertical water tube-boilers, and a separator is provided in the main steam pipe. For a short period during the test, water accumulated in the separator, and its quantity was determined and allowed for. For the balance of the test there was no accumulation, and the steam is presumed to be commercially dry. The valves and pistons of both cylinders were practically tight. The load consisted of dynamos supplying current for electric lighting.

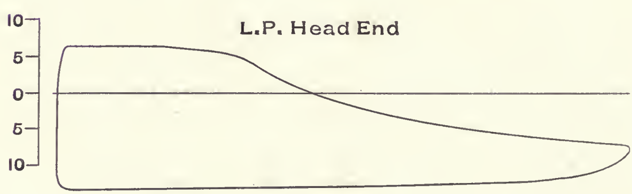
Engine No. 36 belonged to the same plant as Nos. 30 and 35.

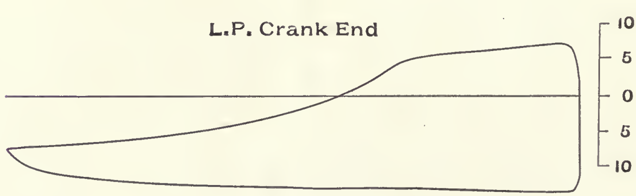
The behavior of the steam in its passage through the cylinders which the analysis of the indicator diagrams reveals is of unusual interest. The increase in the amount of steam shown at release over cut-off is very large in both cylinders, and the loss of steam which the low-pressure cylinder shows is a marked feature. These actions may be attributed to the effect of the jacket-water in the cylinders combined with the cooling action which always takes place when steam passes from a high to a low-pressure cylinder, where no means is provided for reducing cylinder condensation. In this case the quantities are unaffected by steam which leaked, all the valves and pistons being practically tight.

ENGINE No. 36


H.P. Head End
120
100
80
60
40
20
0


H.P. Crank End
120
100
80
60
40
20
0


L.P. Head End
10
5
0
5
10


L.P. Crank End
10
5
0
5
10

ENGINE No. 37.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve	(Corliss)
Number of cylinders		1	1
Diameter of cylinder	ins.	16½	32
Diameter of piston rod	ins.	2¾	{ 2¾ 3¾
Stroke of piston	ft.	4½	4½
Clearance	%	2½	2½
H. P. constant for 1 lb. m. e. p. one revolution per minute	H.P.	.0573	.2162
Ratio of areas of cylinders		1	3.774
Condition of valves and pistons regarding leakage		Fairly tight.	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	0.833	hrs.
Weight of feed-water consumed	3,122.	lbs.
Feed-water consumed per hour	3,746	lbs.
Pressure in steam pipe above atmosphere	108	lbs.
Pressure in receiver above atmosphere	2	lbs.
Vacuum in condenser	27	ins.
Revolutions per minute	59	
Mean effective pressure, H. P. cylinder	45.47	lbs.
Mean effective pressure, L. P. cylinder	9.83	lbs.
Indicated horse-power, H. P. cylinder	154.28	H.P.
Indicated horse-power, L. P. cylinder	125.85	H.P.
Indicated horse-power, whole engine	280.13	H.P.
Feed-water consumed per I. H. P. per hour	13.37	lbs.

Measurements based on Sample Diagrams.

		H. P. Cylinder.	L. P. Cylinder.
Steam accounted for at cut-off . . .	lbs.	9.8	10.48
Steam accounted for at release . . .	lbs.	10.78	10.94
Proportion of feed-water accounted for at cut-off		.732	.784
Proportion of feed-water accounted for at release		.806	.818

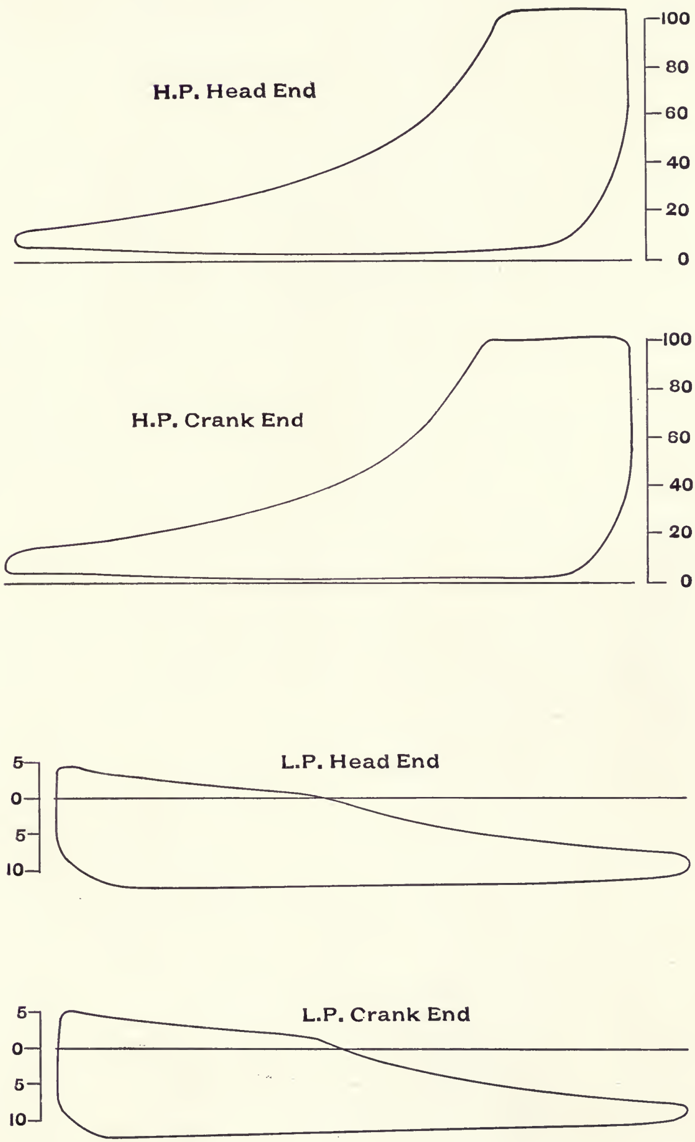
Engine No. 37 is a tandem horizontal compound with cylinders and heads steam jacketed. The condenser is of the jet type, operated with an air-pump connected to the engine.

Steam is supplied from vertical boilers, which gave steam that was at times slightly superheated, and at other times in its ordinary condition. The feed-water was measured on the test by water-glass observations, the water being first pumped to a high point, then shut off, and the test continued until the boilers needed replenishing. The water drained from the jackets amounted to 248 lbs. per hour, or in round numbers, 7% of the total quantity used by the engine. The load consisted mainly of rubber grinding machinery.

The variable character of the load, and the short duration of the test, make the results less accurate than they would be if the load had been steady and the water had been measured for a longer period. The sample indicator diagrams which are here presented, owing to the fluctuating load, must be regarded as showing the general distribution of the steam in the cylinders rather than precise average samples of the work.

When the jackets were shut off, the distribution of the steam was affected in a noticeable degree. The difference between the steam shown at release and cut-off was greatly increased.

ENGINE No. 37


H.P. Head End
100
80
60
40
20
0
H.P. Crank End
100
80
60
40
20
0
L.P. Head End
5
0
5
10
L.P. Crank End
5
0
5
10

ENGINE No. 38.

Compound Condensing Engine.

	H. P. Cylinder.	L. P. Cylinder.
Kind of engine	Four valve (Corliss)	
Number of cylinders	1	1
Diameter of cylinder . . . . ins.	22	44
Diameter of piston rod . . . . . ins.	3.5	3.5
Stroke of piston . . . . ft.	5	5
Clearance . . . . %	2.5	2.5
H. P. Constant for one lb. m.e.p., one rev. per minute . . . . H.P.	.0114	.0459
Ratio of areas of cylinders	1	4.03
Condition of valves and pistons regarding leakage	Practically tight	Excessive leakage

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	8.58	hrs.
Weight of feed-water consumed	118,927	lbs.
Feed-water consumed per hour	13,861	lbs.
Pressure in steam pipe above atmosphere	108.9	lbs.
Pressure in receiver	8.4	lbs.
Vacuum in condenser	23.6	ins.
Revolutions per minute	62.14	
Mean effective pressure, H. P. cylinder	61.53	lbs.
Mean effective pressure, L. P. cylinder	9.87	lbs.
Indicated horse-power, H. P. cylinder	434.6	H.P.
Indicated horse-power, L. P. cylinder	281.4	H.P.
Indicated horse-power, whole engine	716	H.P.
Feed-water consumed per I. H. P. per hour	19.36	lbs.

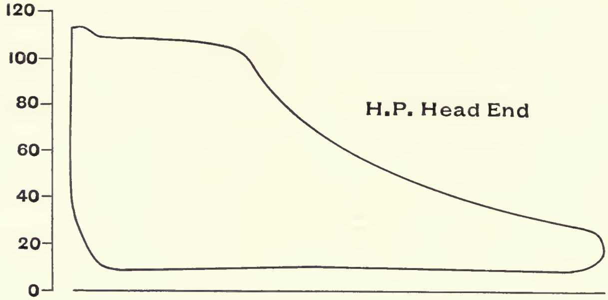
Measurements Based on Sample Diagrams.

	H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere . . lbs.	112	8
Cut-off pressure above zero . . . . lbs.	111	16.8
Release pressure above zero . . . . lbs.	45.7	6.2
Mean effective pressure . . . . . lbs.	62.08	9.84
Back pressure at mid stroke above or below atmosphere . . . . . . lbs.	+10	−11
Proportion of stroke completed at cut-off	.377	.381
Steam accounted for at cut-off . . . lbs.	13.07	8.42
Steam accounted for at release . . . lbs.	12.76	8.67
Proportion of feed-water accounted for at cut-off	.675	.435
Proportion of feed-water accounted for at release	.659	.448

Engine No. 38 is a horizontal cross compound. The cylinders are steam-jacketed, and the intermediate receiver, which is a chamber 30″ in diameter and 8′ high, is also jacketed. The arrangement of the jacket-piping is such that the drain pipe of the high-pressure jacket supplies the low-pressure jacket, and the drain pipe of this supplies the receiver jacket, and their sizes are so proportioned that there is a continual reduction of pressure from one point to the next, and consequently a continuous circulation. The engine is fitted with a jet condenser operated by a direct connected air-pump. Steam is furnished by horizontal return tubular boilers located at a distance of some 200 feet. The water of condensation which collects in the steam pipe is carried back to a feed tank in the boiler-room, and steam used by the feed pump exhausts into the same tank. There was some leakage of joints in the steam piping which has not been allowed for. The valves and pistons of the high-pressure cylinder were practically tight. The valves of the low-pressure cylinder were tight, but the piston contained a loosely fitting packing ring and leaked very badly. The load consisted of cordage machinery.

The interest in this test centers upon the effect which was produced by excessive leakage through the low-pressure piston. In a well jacketed engine the steam accounted for by the indicator is nearly as great in the low-pressure cylinder as in the high pressure cylinder. In this case there is a reduction from .675 to .435, or 24% of the total weight of steam consumed, and this is evidently due to the leakage referred to.

ENGINE No. 38


H.P. Head End
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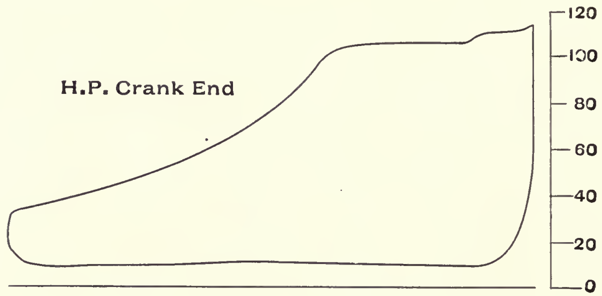
H.P. Crank End
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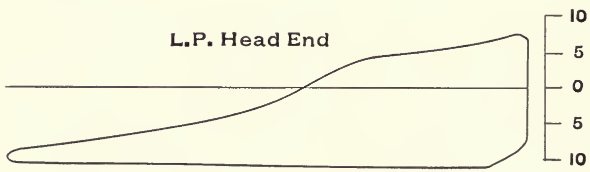
L.P. Head End
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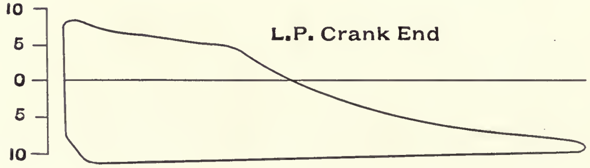
L.P. Crank End
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0
5
10

ENGINE No. 39.

Compound Condensing Engine.

	H. P. Cylinder.	L. P. Cylinder.
Kind of engine	Single valve	
Number of cylinders	1	1
Diameter of cylinder . . . . ins.	13	26
Diameter of piston rod . . . . . ins.	$1\frac{15}{16}$	$2\frac{3}{4}$
Stroke of piston . . . . ins.	18	18
Clearance . . . . %	7	10
Horse-power constant for one lb. m.e.p. one revolution per minute . . . H.P.	.0019	.048
Ratio of areas of cylinders	1	4.03
Condition of valves and pistons regarding leakage	Considerable leakage	Considerable leakage

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	2.85	hrs.
Weight of feed-water consumed	11,325	lbs.
Feed-water consumed per hour	3,973.7	lbs.
Pressure in steam pipe above atmosphere	120.6	lbs.
Vacuum in condenser	27	ins.
Revolutions per minute	195.3	
Mean effective pressure H. P. cylinder	43.75	lbs.
Mean effective pressure L. P. cylinder	12.68	lbs.
Indicated horse-power H. P. cylinder	101.7	H.P.
Indicated horse-power L. P. cylinder	118.9	H.P.
Indicated horse-power, whole engine	220.6	H.P.
Feed-water consumed per I. H. P. per hour	18.01	lbs.

Measurements based on Sample Diagrams.

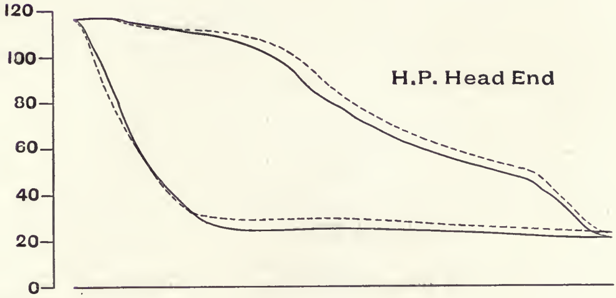
	H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere . . lbs.	115.5	15
Cut-off pressure above zero . . . . lbs.	104.9	20.6
Release pressure above zero . . . . lbs.	55.4	10.3
Mean effective pressure . . . . . . lbs.	43.1	12.69
Back pressure at mid stroke, above or below atmosphere . . . . lbs.	+ 23	− 10.5
Proportion of stroke completed at cut-off	.396	.387
Steam accounted for at cut-off . . . lbs.	11.33	12.49
Steam accounted for at release . . . lbs.	12.3	12.49
Proportion of feed-water accounted for at cut-off	.629	.694
Proportion of feed-water accounted for at release	.683	.694

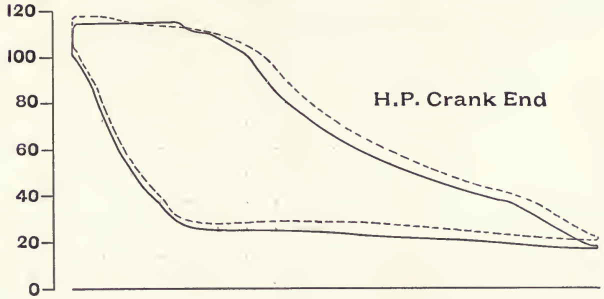
Engine No. 39 is a single valve, cross compound, unjacketed engine, with a shaft governor operating on the cut-off of the high-pressure cylinder. The valves are of the piston type provided with an inefficient ring packing. A jet condenser is used, operated by an independent air-pump driven with steam taken from the engine pipe. The quantity thus used was determined by an independent test and allowed for. Steam is supplied from vertical water-tube boilers, and a separator placed in the steam pipe secured what was believed to be commercially dry steam without superheating. The valves and pistons all leaked a considerable amount. The load consisted of dynamos furnishing current for electric lighting. With the exception of the low-pressure cylinder and the valves, this engine is the same as No. 35. During the interval between the tests the engine had been provided with new valves fitted with packing and a complete new low-pressure cylinder of larger size.

Referring to the test on Engine No. 35, the figures given here show an improvement, due largely to a better distribution of the steam, which was accomplished by a change of proportion in the steam cylinders. The increase in the size of the low-pressure cylinder enabled this cylinder to do a larger proportion of the work, with corresponding advantage. The reduction in the quantity of feed water consumed per horse power per hour amounted to 18.3%; and the reduction in the steam accounted for by the diagrams at cut-off, which is 17.5%, furnishes a reason for the change. In view of the leakage of the valves and pistons, it is not surprising that the proportion of steam accounted for is low; and this is true in the case of both engines.

To make a ready comparison of the diagrams in the two cases under consideration, showing the general effect of the change of cylinders, diagrams from Engine No. 35, taken with the same load, are superposed in dotted lines upon those relating to No. 39, which are represented in full lines.

ENGINE No. 39


120
100
80
60
40
20
0
H.P. Head End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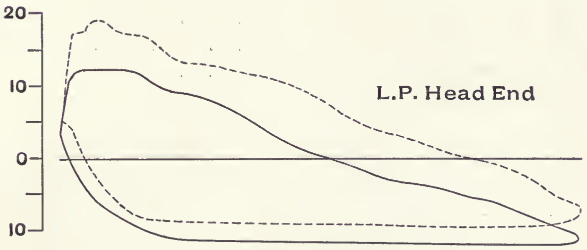
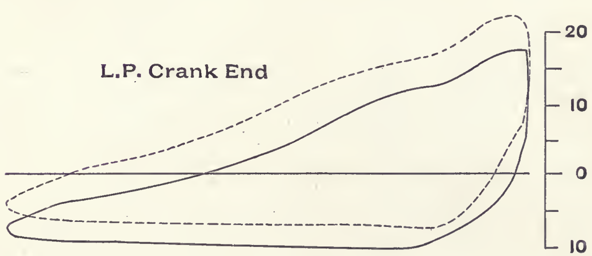
H.P. Crank End


20
10
0
10
L.P. Head End


L.P. Crank End
20
10
0
10

ENGINE No. 40.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder
Kind of engine		Single valve	
Number of cylinders		1	1
Diameter of cylinders	ins.	18	30
Stroke of piston	ins.	16	16
Clearance	%	33	9
H. P. constant for 1 lb. m. e. p. one revolution per min.	H.P.	.0103	.0285
Ratio of areas of cylinders		1	2.78
Condition of valves and pistons regarding leakage		Practically tight	Practically tight

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	1.527	hrs.
Weight of feed-water consumed	9,660	lbs.
Feed-water consumed per hour	6,326.1	lbs.
Pressure in steam pipe above atmosphere	126	lbs.
Vacuum in condenser	21.1	ins.
Revolutions per minute	228	
Mean effective pressure, H. P. cylinder	63.9	lbs.
Mean effective pressure, L. P. cylinder	30.4	lbs.
Indicated horse-power, H. P. cylinder	149.7	H.P.
Indicated horse-power, L. P. cylinder	197.9	H.P.
Indicated horse-power, whole engine	347.6	H.P.
Feed-water consumed per I. H. P. per hour	18.2	lbs.

Measurements based on Sample Diagrams.

		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere	lbs.	111.6	49
Cut-off pressure above zero	lbs.	114.8	29.7
Release pressure above zero	lbs.	82.4	25.7
Mean effective pressure	lbs.	63.4	30.1
Back pressure at mid stroke, above or below atmosphere	lbs.	+24.5	−7.
Proportion of stroke completed at cut-off		.595	.795
Steam accounted for at cut-off	lbs.	16.58	17.58
Steam accounted for at release	lbs.	15.59	16.11
Proportion of feed-water accounted for at cut-off		.911	.965
Proportion of feed-water accounted for at release		.857	.885

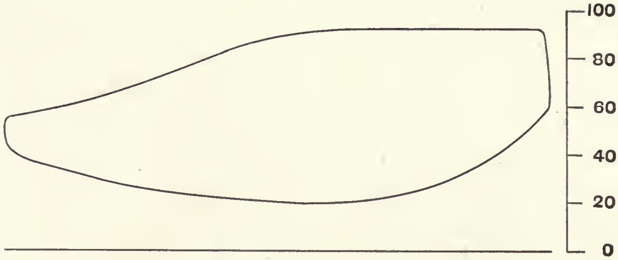
Engine No. 40 is a vertical single-acting engine with unjacketed cylinders and a single piston valve fitted with ring packing, one valve serving for both cylinders. The condensing apparatus is a surface condenser with air-pump operated by steam. During the test the exhaust from the air-pump escaped to the atmosphere. This pump was of insufficient size to give a proper vacuum. Steam is furnished by horizontal return tubular boilers. It was found by calorimeter test that at a point near the engine it contained one-half of 1% of moisture. The pistons and the valve were practically tight, although in this class of engines there is always some escape of water by the piston rings into the crank case. The load consisted of a centrifugal pump.

The feed-water was measured by collecting the water discharged from the surface condenser. The quantity thus determined does not include that referred to above, which leaked from the steam cylinders into the crank case, and which there is no ready means of determining.

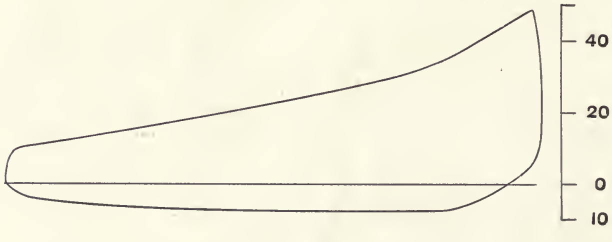
The consumption of feed-water here given was less than the actual amount of steam which passed through the engine, owing to the fact above noted that some of the steam which was condensed in the cylinders passed into the crank case and failed to be measured. This accounts for the large proportions which the steam accounted for at cut-off and release bears to the feed-water consumption. In view of the late cut-off, the high back pressure in the small cylinder, the excellent quality of the steam furnished to the engine, and the tightness of the valve, all of which tend to reduce the losses shown by an analysis of the diagram, these proportions must necessarily be large. The leakage referred to could hardly be expected to exceed 5%. Assuming it to be 5%, the feed-water consumption would stand 19.1 lbs., and the proportions of steam accounted for at cut-off in the two cylinders .87 and .92 respectively.

ENGINE No. 40

H.P. Cyl.



L.P. Cyl.



ENGINE No. 41.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Single valve	
Number of cylinders		1	1
Diameter of cylinder	ins.	11½	18½
Diameter of piston rod	ins.	2	2
Stroke of piston	ins.	13	13
Clearance	%	7	10
H. P. constant for one lb. m. e. p. one revolution per minute	H.P.	.00672	.01755
Ratio of areas of cylinders		1	2.61
Inside diameter of steam pipe	ins.	4	5
Inside diameter of exhaust pipe	ins.	5	7
Condition of valves and pistons regarding leakage		Considerable leakage	

Data and Results of Feed-Water Tests.

Test. Character of Load.		A. Light Load.	B. Heavy Load.
Character of steam		Ordinary	Ordinary
Duration	hrs.	3.5	4.8
Weight of feed-water consumed.	lbs.	7,203.5	18,043.
Feed-water consumed per hour	lbs.	2,058	3,759
Pressure in steam pipe above atmos.	lbs.	129.7	130.1
Vacuum in condenser	ins.	25.9	25.5
Revolutions per minute		306	298.5
Mean effective pressure, H. P. cylinder	lbs.	25.02	48.5
Mean effective pressure, L. P. cylinder	lbs.	7.27	19
Indicated horse-power, H. P. cylinder	H.P.	51.5	97.3
Indicated horse-power, L. P. cylinder	H.P.	39.	99.5
Indicated horse-power, whole engine	H.P.	90.5	196.8
Feed-water consumed per I.H.P. per hr.	lbs.	22.74	19.1

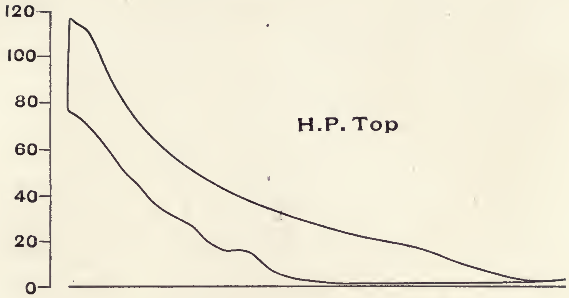
Measurements based on Sample Diagrams, heavy load Test.

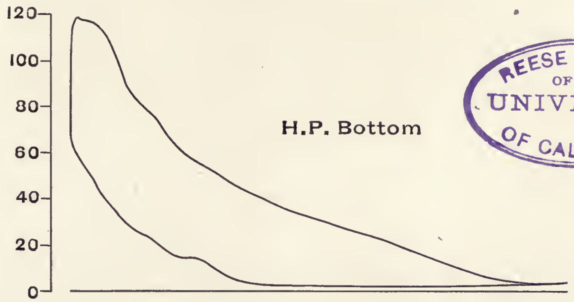
Test.		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere	lbs.	130	20.3
Corresponding steam-pipe or receiver pressure	lbs.	132	
Cut-off pressure above zero	lbs.	115.2	24.6
Release pressure above zero	lbs.	63.1	15.
Mean effective pressure	lbs.	48.9	19.2
Back pressure at mid stroke above or below atmosphere	lbs.	+20.3	− 11.1
Proportion of stroke completed at cut-off		.382	.409
Steam accounted for at cut-off	lbs.	12.03	9.97
Steam accounted for at release	lbs.	13.43	12.76
Proportion of feed-water accounted for at cut-off		.629	.522
Proportion of feed-w. acc'd for at release		.703	.668

Engine No. 41 is a vertical cross-compound unjacketed high-speed engine, having unpacked piston valves, one for each cylinder, and controlled by a shaft governor operating on the cut-off of the high-pressure cylinder. A jet condenser is used, operated by an independent air-pump driven by steam taken from the main pipe. The quantity of steam used by the condenser was determined by an independent test and allowed for. Allowance was also made for steam condensed in the large service main, which was designed for supplying several other engines besides the one tested. Steam was furnished by horizontal return tubular boilers, and a calorimeter test showed that it contained $\frac{4}{10}$ of 1% of moisture. The valve of the high-pressure cylinder was found to leak quite badly. That of the low-pressure cylinder was reasonably tight. The leakage of the valves interfered with a determination of the condition of the pistons. The load consisted of dynamos furnishing current for electric lighting. The tests were made with two different loads, other conditions remaining the same.

If the results of this test are compared with those made on a four-valve engine such as No. 36, which showed a much more economical performance, the effect of various features in the design of the engine are apparent. Engine No. 41 had a single valve, which secured less perfect distribution of steam than the four valves of the other engine. It had larger percentages of clearance space, and finally, the type of valve used permitted a much larger amount of leakage than occurred in the other engine. Engine No. 41, however, had the advantage of more rapid reciprocations; but this, it appears, did not have sufficient effect to overcome the losses due to the causes mentioned.

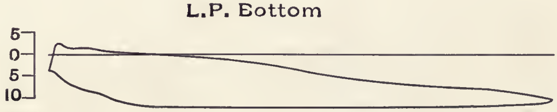
ENGINE No. 41a



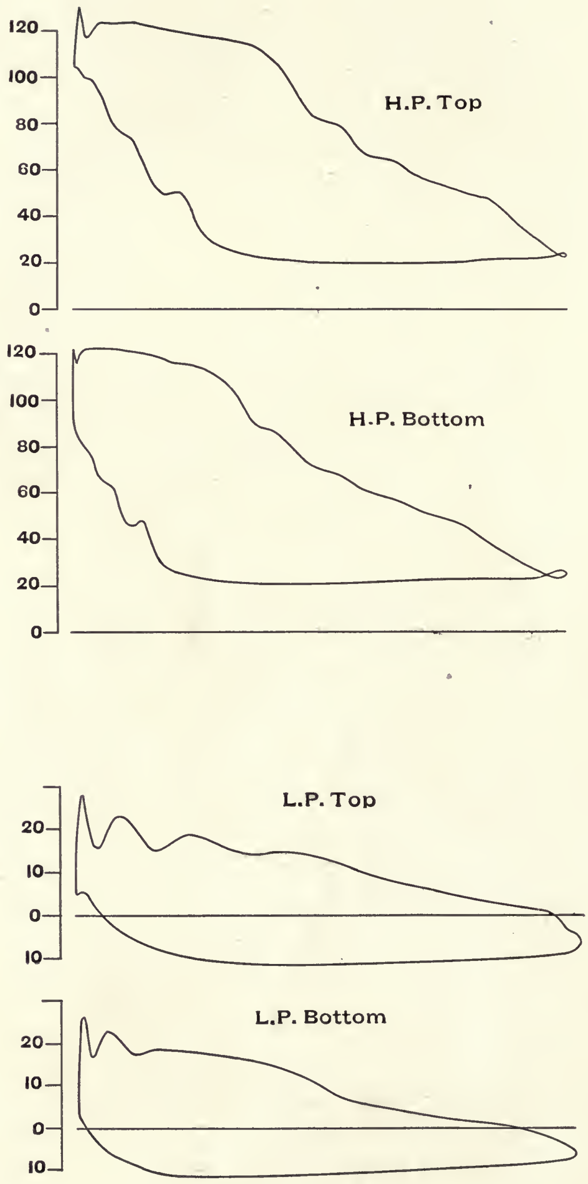


L.P. Top

L.P. Bottom



ENGINE No. 41b



ENGINE No. 42.

Compound Non-Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Single valve	
Number of cylinders		1	1
Diameter of cylinders	ins.	11½	18½
Diameter of piston rod	ins.	2	2
Stroke of piston	ins.	13	13
Clearance	%	7	10
H. P. Constant for one lb. m. e. p. one rev. per min.	H.P.	.00672	.01755
Ratio of areas of cylinders		1	2.61
Condition of valves and pistons regarding leakage		Considerable leakage	

Data and Results of Feed-Water Tests.

Test. Conditions regarding Load.		A. Light Load.	B. Heavy Load.
Character of steam		Ordinary	Ordinary
Duration	hrs.	5	5
Weight of feed-water consumed	lbs.	10,228	15,369
Feed-water consumed per hour	lbs.	2,045.6	3,842.2
Press. in steam pipe above atmos.	lbs.	126.5	128
Revolutions per minute		300.2	292.7
Mean effective pressure, H.P. cyl.	lbs.	20.36	42.16
Mean effective pressure, L.P. cyl.	lbs.	.86	13.56
Indicated horse-power, H. P. cyl.	H.P.	41.05	82.89
Indicated horse-power, L. P. cyl.	H.P.	4.53	69.59
Indicated H. P., whole engine	H.P.	45.58	152.48
Feed-water cons. per I. H. P. per hr.	lbs.	44.89	25.2

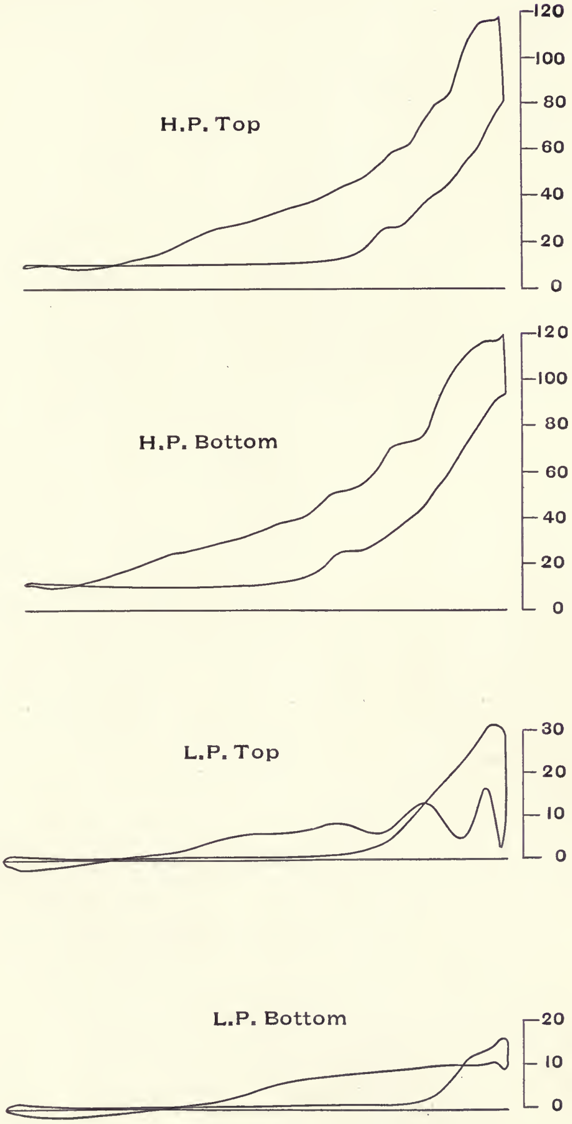
Measurements based on Sample Diagrams.

Test. Conditions regarding Load.		H.P. Cyl.	L.P. Cyl.	H.P. Cyl.	L.P. Cyl.
Initial pressure above atmosphere	lbs.	116.4	10.3	120	30
Corresponding steam-pipe or receiver pressure	lbs.	125.		128.	
Cut-off pressure above zero	lbs.	112.4	19.5	120.3	53.5
Release pressure above zero	lbs.	40 9		64.7	20.6
Mean effective pressure	lbs.	20.2	.7	42.27	14.1
Back pressure at mid stroke above atmosphere	lbs.	10.7	1.	29.9	1.8
Proportion of stroke completed at cut-off		.107	.526	.389	.424
Steam accounted for at cut-off	lbs.	10.21	28.26	15.56	13.82
Steam accounted for at release	lbs.	26.43		17.16	16.66
Proportion of feed-water accounted for at cut-off		.228	.629	.617	.548
Proportion of feed-water accounted for at release		.589		.681	.661

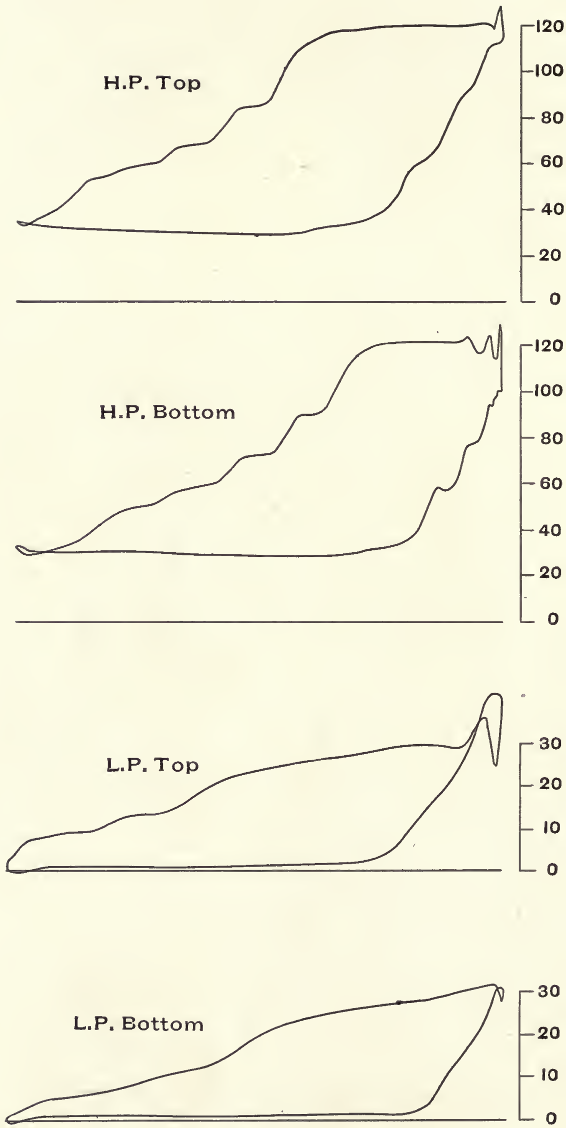
Engine No. 42 is of the vertical cross-compound unjacketed high-speed class. It is a duplicate of Engine No. 41, being located in the same power house, and forming a part of the same plant. It was supplied with steam from a different portion of the service main, the water condensed in which returned back to the boiler. Unlike engine No. 41 it was run non-condensing. The valves and pistons leaked to about the same extent as in the other engine, and the load was of the same character. The tests were two in number, one being made with a very light load.

These tests bring out very forcibly the wastefulness of a non-condensing compound engine of this type when carrying an extremely light load. In the case of the first test the load was so small that the low-pressure cylinder contributed only about 10% of the whole power, which is so small as to be immaterial; and consequently, the engine showed simply the economy due to a non-condensing cylinder of this type carrying a high back pressure, and working at a comparatively early cut-off. The effect of valve leakage is revealed by the small proportion of steam accounted for by the indicator. Compared with the condensing engine of the same type, No. 41, there is a marked advantage due to the use of the condenser; and this appears to be especially true in the case of the light load. Comparing the two heavy-load tests the reduced consumption of feed-water is 6.1 lbs. per I. H. P. per hour, or about 24%. In the light-load test there is a remarkable increase in the steam accounted for at release of the high-pressure cylinder over that shown at cut-off. It is due largely no doubt to valve leakage.

ENGINE No. 42a



ENGINE No. 42b



ENGINE No. 43.

Compound Condensing Engine.

		H.P. Cylinder.	L.P. Cylinder.
Kind of engine		Four valve (Corliss)	
Number of cylinders		1	1
Diameter of cylinder	ins.	27.9	48.3
Diameter of piston rod	ins.	5	6
Stroke of piston	ft.	5	5
Clearance (assumed)	%	2.5	2.5
H. P. constant for one lb. m. e. p. one rev. per min.	H.P.	.1828	.556
Ratio of areas of cylinders		1	3.04
Inside diameter of steam pipe . . .	ins.	12	
Condition of valves and pistons regarding leakage		Fairly tight	Fairly tight

Data and Results of Feed-Water Test.

Character of steam	Superh'd 44.5	degs.
Duration	19.08	hrs.
Weight of feed-water consumed	257,351	lbs.
Feed-water consumed per hour	13,488	lbs.
Pressure in steam pipe above atmosphere	119.8	lbs.
Pressure in receiver above atmosphere	11.0	lbs.
Vacuum in condenser	25.7	ins.
Revolutions per minute	70.03	
Mean effective pressure H. P. cylinder	39.04	lbs.
Mean effective pressure L. P. cylinder	13.28	lbs..
Indicated horse-power H. P. cylinder	499.9	H.P.
Indicated horse-power L. P. cylinder	517.2	H.P.
Indicated horse-power, whole engine	1,017.1	H.P.
Feed-water consumed per I. H. P. per hour	13.26	lbs.

Measurements Based on Sample Diagrams.

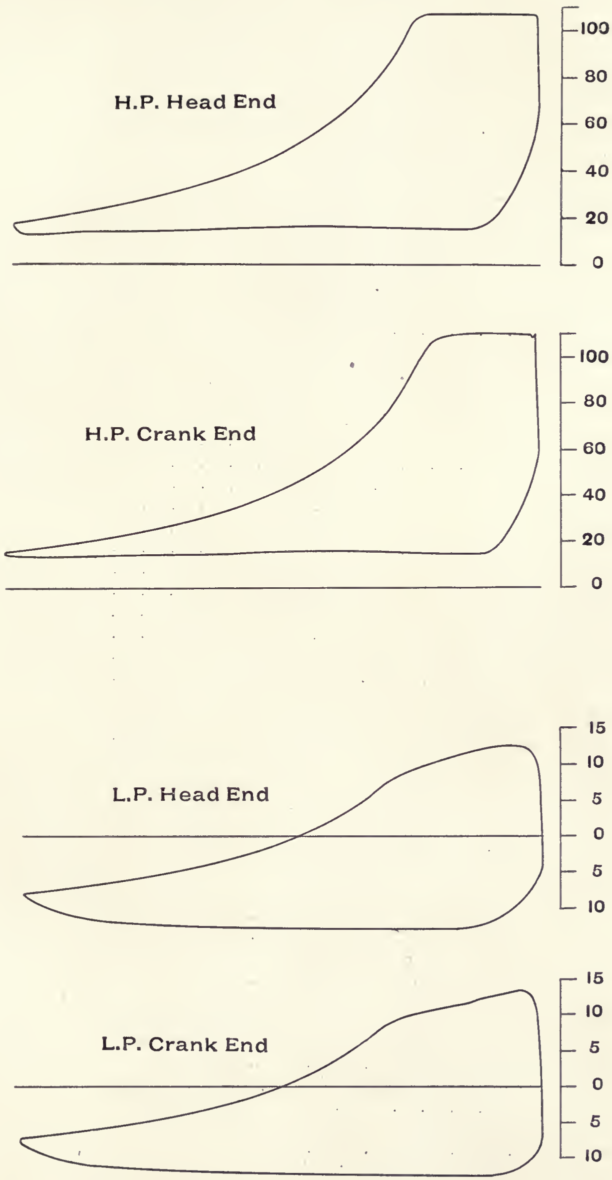
		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere . . .	lbs.	109	12.9
Corresponding steam-pipe or receiver press.	lbs.	113	
Cut-off pressure above zero	lbs.	114.8	22.1
Release pressure above zero	lbs.	31.4	7.4
Mean effective pressure	lbs.	40.03	13.21
Back pressure at mid stroke, above or below atmosphere	lbs.	+ 16.	− 12.2
Proportion of stroke completed at cut-off .		.236	.312
Steam accounted for at cut-off	lbs.	10.1	9.39
Steam accounted for at release	lbs.	12.0	10.21
Proportion of feed-water accounted for at cut-off		.762	.708
Proportion of feed-water accounted for at release		.908	.77

Engine No. 43 is a horizontal cross compound with jacketed cylinders and unjacketed receiver. It exhausts into a jet condenser provided with a direct connected air-pump. The jacket space of either cylinder, which is confined to the barrel of the cylinder, forms a thoroughfare through which the steam passes to the top chest, the steam entering at the bottom. The spaces are drained by traps. Steam is supplied by vertical boilers which superheat. The valves and pistons of the H. P. cylinder leaked a small amount, but those of the L. P. cylinder were practically tight. The load consisted of cotton machinery.

The test reported is the collective result of four independent trials of 4.5 to 5 hours each. . . . .

A noticeable feature in these results is the increase in the steam accounted for at release H. P. cylinder over that shown at cut-off, viz., .146. In working out these figures the clearance was assumed at $2\frac{1}{2}$ %. If the clearance were in reality 1 % more (i. e., 3.5 %) the increase is reduced to .123. Even this is notable.

ENGINE No. 43


H.P. Head End
100
80
60
40
20
0
H.P. Crank End
100
80
60
40
20
0
L.P. Head End
15
10
5
0
5
10
L.P. Crank End
15
10
5
0
5
10

ENGINE No. 44.

Compound Non-Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Single valve	
Number of cylinders		1	1
Diameter of cylinder	ins.	11	19
Stroke of piston	ft.	11	11
Clearance	%	33	9
H. P. constant for 1 lb. m. e. p. one revolution per minute	H.P.	.00264	.00788
Ratio of areas of cylinders		1	2.98
Inside diameter of steam pipe . . .	ins.	5	
Condition of valves and pistons regarding leakage		Practically tight	Practically tight

Data and Results of Feed-Water Test.

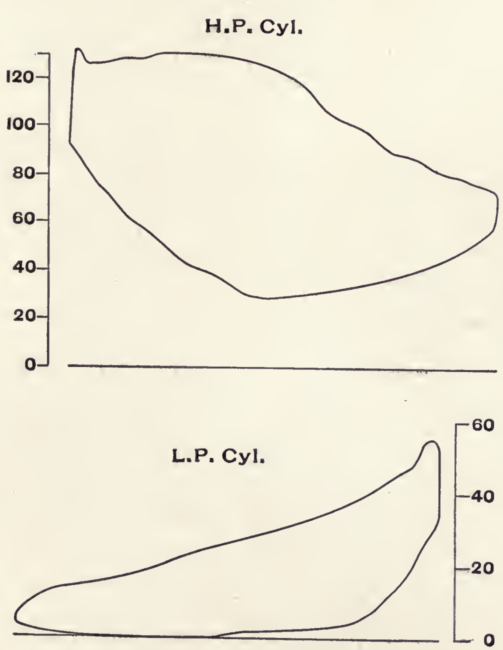
Character of steam	Ordinary	
Duration	5.33	hrs.
Weight of feed-water consumed	13,397	lbs.
Feed-water consumed per hour	2,511.8	lbs.
Pressure in steam pipe above atmosphere	135.9	lbs.
Revolutions per minute	296.3	
Mean effective pressure, H. P. cylinder	67.2	lbs.
Mean effective pressure, L. P. cylinder	23.3	lbs.
Indicated horse-power, H. P. cylinder	53.6	H.P.
Indicated horse-power, L. P. cylinder	56	H.P.
Indicated horse-power, whole engine	109.66	H.P.
Feed-water consumed per I. H. P. per hour	22.91	lbs.

Measurements based on Sample Diagrams.

		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere . .	lbs.	130	55
Corresponding steam-pipe pressure .	lbs.	132	
Cut-off pressure above zero	lbs.	128.2	35.5
Release pressure above zero	lbs.	89.5	28.4
Mean effective pressure	lbs.	67.1	23.6
Back pressure at lowest point above atmosphere	lbs.	30	0
Proportion of stroke completed at cut-off		.605	.662
Steam accounted for at cut-off . . .	lbs.	18.39	14.96
Steam accounted for at release . . .	lbs.	18.56	16.61
Proportion of feed-water accounted for at cut-off		.803	.653
Proportion of feed-water accounted for at release		.81	.725

Engine No. 44 is a vertical cross-compound, single-acting, unjacketed, high-speed engine, having a single piston valve fitted with ring packing, the speed being controlled by a shaft governor. Steam is supplied by a horizontal return tubular boiler, which by calorimeter test contained some 2% of moist-ture. Drip pockets in the main pipe and at the throttle valve, which were trapped, intercepted most of the entrained water which would otherwise have passed into the engine. The load consisted of an electric generator with constant output. The valve and pistons were very nearly tight.

ENGINE No. 44



ENGINE No. 45.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Double valve	
Number of cylinders		1	1
Diameter of cylinder	ins.	14	28
Diameter of piston rod	ins.	2⅜	2⅜
Stroke of piston	ins.	24	24
Clearance	%	3.6	6.4
H.P. constant for one lb. m.e.p. one revolution per minute	H.P.	.01839	.07436
Ratio of areas of cylinders		1	4.04
Condition of valves and pistons regarding leakage		Considerable leakage	

Data and Results of Feed-Water Tests.

Test.		A.	B.	C.
Character of steam		Ordinary	Ordinary	Ordinary
Duration	hrs.	4	4	3
Weight of feed-water consumed	lbs.	19,014	15,366	6,375
Feed-water consumed per hour	lbs.	4,753.7	3,841.6	2,125
Pressure in steam-pipe above atmos.	lbs.	119.9	120.4	117.6
Pressure in receiver	lbs.	10.1	4.5	below at.
Vacuum in condenser	ins.	25.1	25.5	25.6
Revolutions per minute		161.75	162.75	170.1
Mean effective pressure, H. P. cylinder	lbs.	57.4	49.97	25.89
Mean effective pressure, L. P. cylinder	lbs.	10.39	7.85	3.35
Indicated horse-power, H. P. cylinder	H.P.	170.74	149.55	80.98
Indicated horse-power, L. P. cylinder	H.P.	124.97	95	42.41
Indicated horse-power, whole engine	H.P.	295.71	244.55	123.39
Feed-water consumed per I. H. P. per hour	lbs.	16.07	15.71	17.22

Measurements based on Sample Diagrams.

Test.		A		C	
		H.P.	L.P.	H.P.	L.P.
Initial pressure above atmosphere	lbs.	117.4	9.5	118	5.1
Corresponding steam-pipe or receiver pressure	lbs.	120	10.5	119	
Cut-off pressure above zero	lbs.	114.5	18.6	121.8	7.5
Release pressure above zero	lbs.	42.5	7.7	14.1	3.7
Mean effective pressure	lbs.	57.95	10.37	24.97	3.47
Back pressure at mid stroke above or below atmospere	lbs.	+10.9	−11.6	−3.6	−12.4
Proportion of stroke completed at cut-off		.336	.338	.044	.336
Steam accounted for at cut-off	lbs.	12.08	9.06	6.21	8.67
Steam accounted for at release	lbs.	12.75	10.69	11.32	12.82
Proportion of feed-water accounted for at cut-off		.751	.563	.361	.503
Proportion of feed-water accounted for at release		.793	.665	.657	.744

ENGINE No. 45 (Continued).

Data and Results of Feed-Water Tests.

Test.		D.	E. Non-Condensing.
Character of steam		Ordinary	Ordinary
Duration	hrs.	3	3
Weight of feed-water consumed	lbs.	13,356.5	18,614
Feed-water consumed per hour	lbs.	4,452.2	6,204.7
Pressure in steam pipe above atmos.	lbs.	118.9	118
Pressure in receiver above atmosphere	lbs.	14.1	27.8
Vacuum in condenser	ins.	25.8	
Revolutions per minute		164.77	165.66
Mean effective pressure, H. P. cylinder	lbs.	46.94	52.4
Mean effective pressure, L. P. cylinder	lbs.	10.99	8.81
Indicated horse-power, H. P. cylinder	H.P.	142.23	158.54
Indicated horse-power, L. P. cylinder	H.P.	134.65	108.47
Indicated horse-power, whole engine	H.P.	276.88	267.1
Feed-water consumed per I.H.P. per hr.	lbs.	16.07	23.24

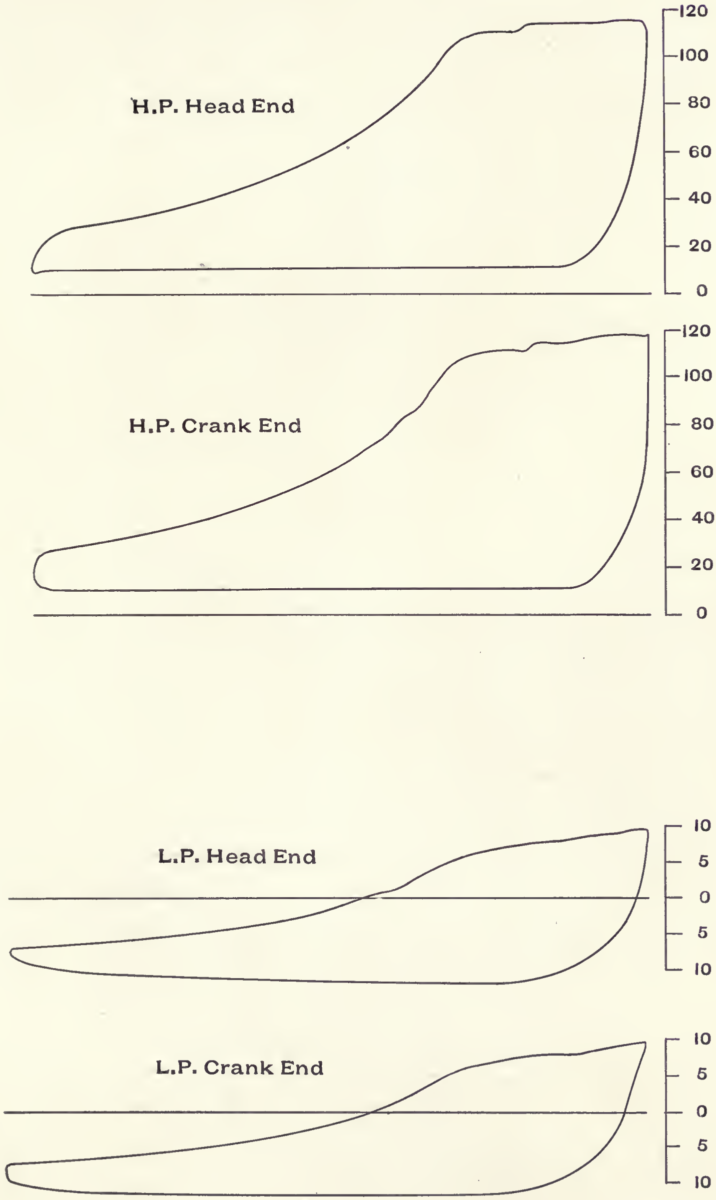
Measurements based on Sample Diagrams.

Test.		D. Condensing.		E. Non-Condensing.	
		H.P. Cy.	L.P. Cy.	H.P. Cy.	L.P. Cy.
Initial pressure above atmosphere	lbs.	117.6	14.1	113.1	27.8
Corresponding steam-pipe or receiver pressure		118	14.5	115	28.5
Cut-off pressure above zero	lbs.	113.1	22.2	110.8	31.9
Release pressure above zero	lbs.	38.3	7.0	58.2	14.9
Mean effective pressure	lbs.	47.65	11.07	50.94	8.77
Back pressure at mid stroke above or below atmosphere	lbs.	+16.	−12.2	+28.7	+1.2
Proportion of stroke completed at cut-off		.3	.239	.487	.406
Steam accounted for at cut-off	lbs.	11.16	8.82	18.46	15.83
Steam accounted for at release	lbs.	12.06	10.71	19.1	18.3
Proportion of feed-water accounted for at cut-off		.694	.548	.794	.681
Proportion of feed-water accounted for at release		.750	.666	.821	.787

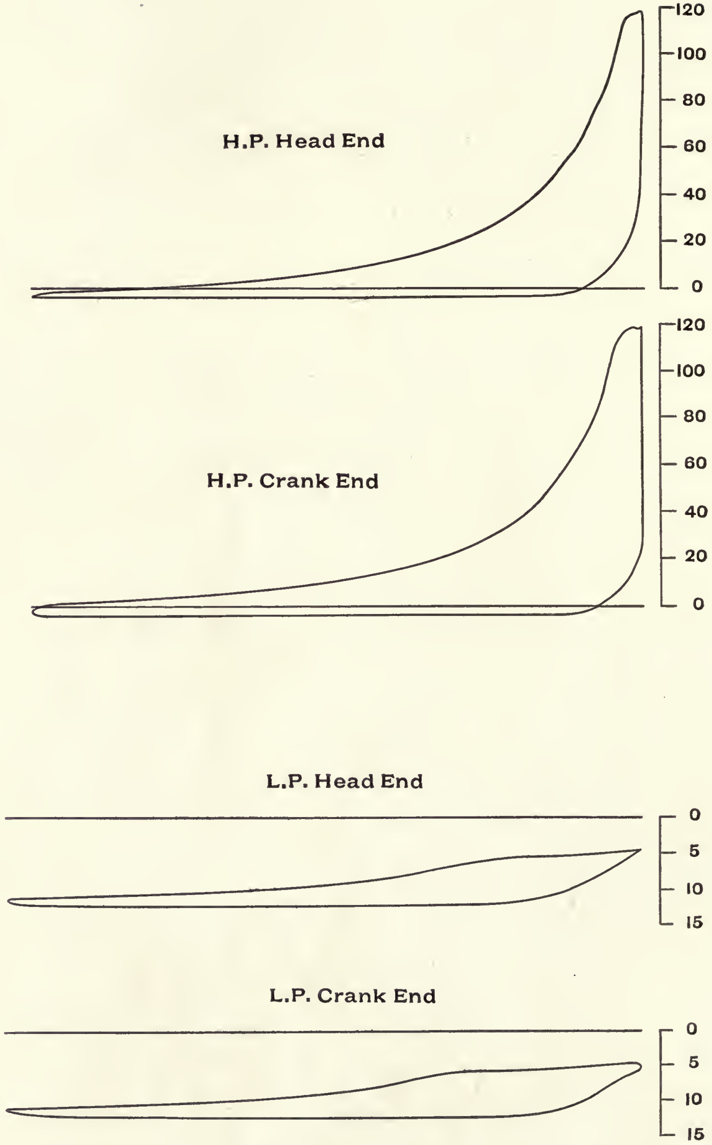
Engine No. 45 is a horizontal cross-compound with unjacketed cylinders and unjacketed receiver. There is a shaft governor operating on the cut-off of the H. P. cylinder. The main valves are balanced slides. The cut-off valve rides on a seat in the interior of the main valve, which is of box pattern. The engine exhausts into a surface condenser, with independent air-pumps, the latter exhausting to waste. The feed-water consumption was found by weighing the water discharged by the air-pump. Steam was drawn from the main service of a large plant, and a calorimeter test showed that it was practically dry. The main valve of the H. P. cylinder leaked quite badly. The other valves and pistons leaked a small amount. The engine supplied power to dynamos for electric lighting. A series of tests was made with different loads, and in one case the engine was run non-condensing.

Considering the wide changes of load in the tests A, B, and C, viz., from 295. H. P. to 123. H. P., the small difference in economy, 15.71 to 17.22, is noteworthy. Probably the leakage of the valves of the H. P. cylinder affected the matter, but to what extent can only be conjectured. The economy is at best much below that obtained from some of the four-valve engines, and excessive leakage is the only thing which satisfactorily explains it. The results of tests D and E, condensing and non-condensing, are respectively 16.07 lbs. and 23.24 lbs., from which it appears that the consumption when running condensing was 30.9% less than when running non-condensing.

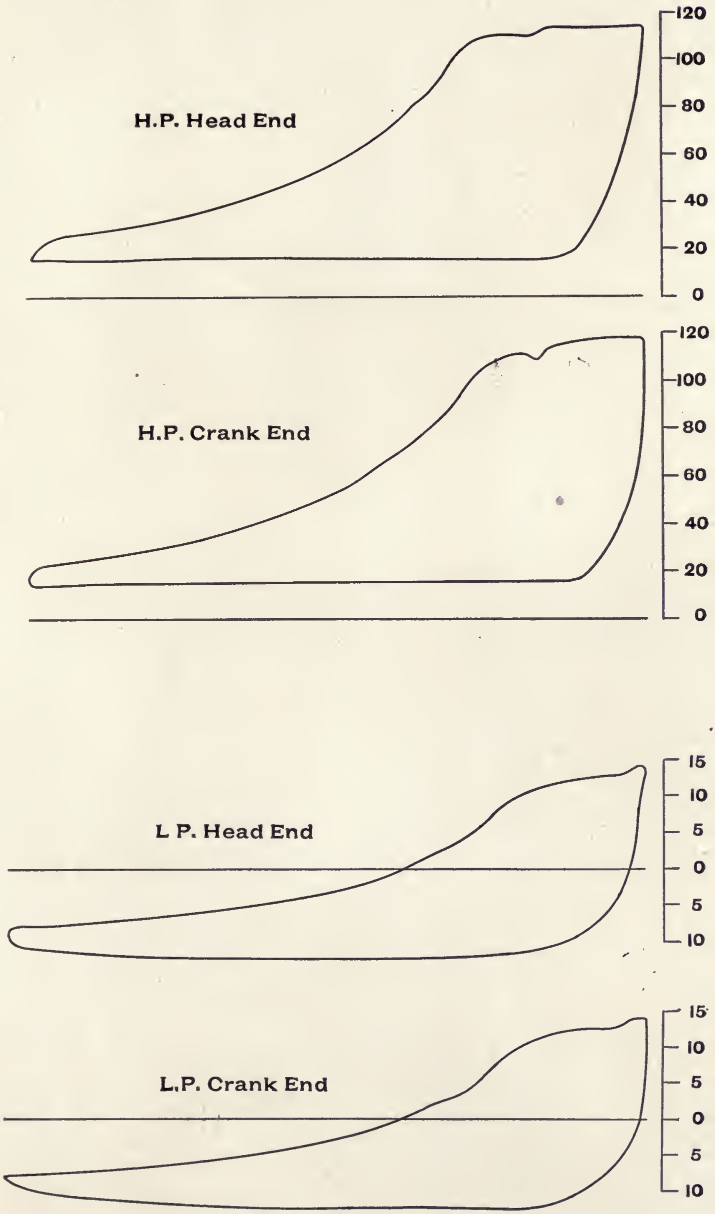
ENGINE No. 45a



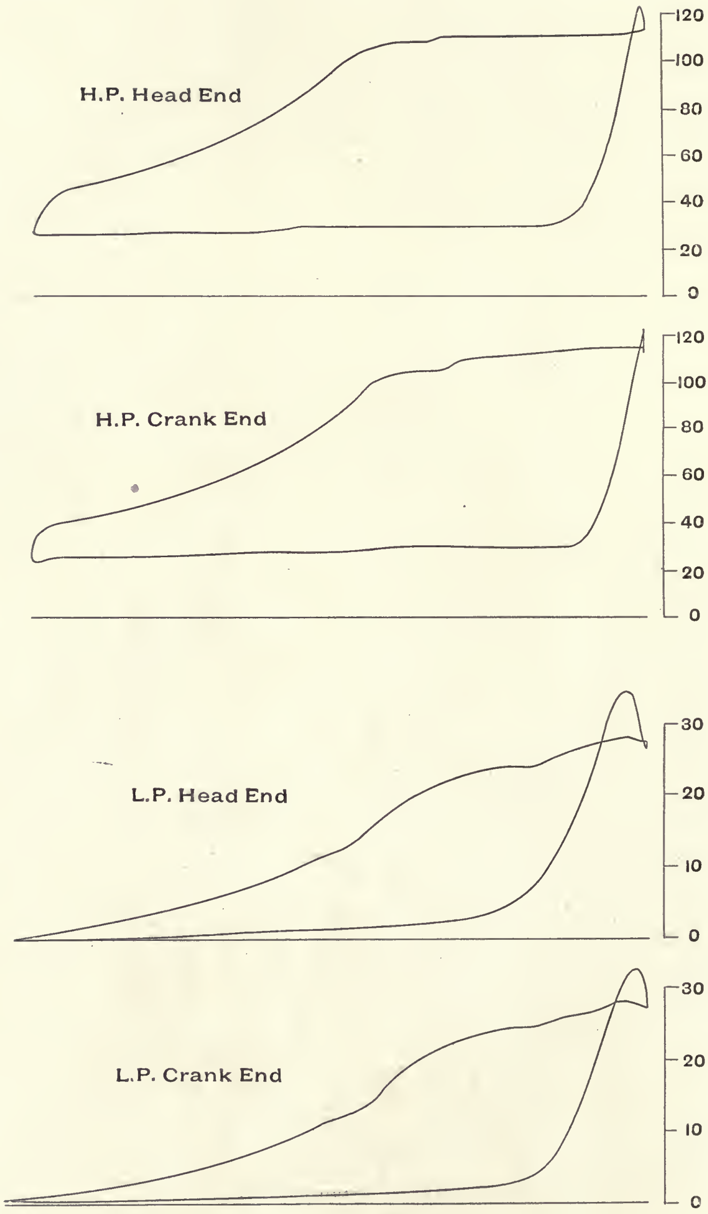
ENGINE No. 45c



ENGINE No. 45d



ENGINE No. 45e



ENGINE No. 46.

Compound Non-Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve	
Number of cylinders		1	1
Diameter of cylinders	ins.	17.5	28
Diameter of piston rod	ins.	$3\frac{3}{8}$	$3\frac{5}{8}$
Stroke of piston	ins.	48	48
Clearance	%	4.1	5.8
H. P. Constant for one lb. m. e. p. one rev. per min.	H.P	.0572	.148
Ratio of areas of cylinders		1	2.587
Inside diameter of steam pipe	ins.	7	
Inside diameter of exhaust pipe	ins.	8	12
Condition of valves and pistons regarding leakage		Practically tight	

Data and Results of Feed-Water Tests.

Test.		A.	B.
Character of steam		Ordinary	Ordinary
Duration	hrs.	8.55	7.87
Weight of feed-water consumed	lbs.	65,591	82,697
Feed-water consumed per hour	lbs.	7,671.2	10,507.9
Press. in steam pipe above atmos.	lbs.	128.7	135.5
Pressure in receiver above atmos.		27.2	29.5
Revolutions per minute		101.02	99.06
Mean effective pressure, H.P. cyl.	lbs.	34.83	52.42
Mean effective pressure, L.P. cyl.	lbs.	9.75	14.19
Indicated horse-power, H. P. cyl.	H.P.	201.3	278.84
Indicated horse-power, L. P. cyl.	H.P.	145.6	207.85
Indicated H. P., whole engine	H.P.	346.9	486 69
Feed-water cons. per I.H.P. per hr.	lbs.	22.11	21.59

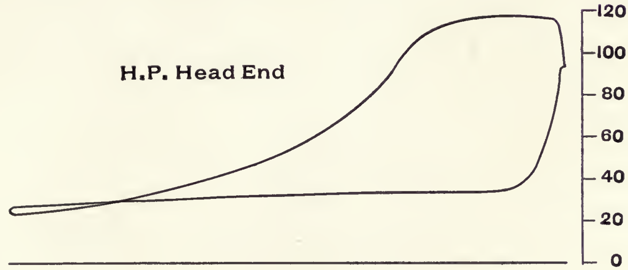
Measurements based on Sample Diagrams.

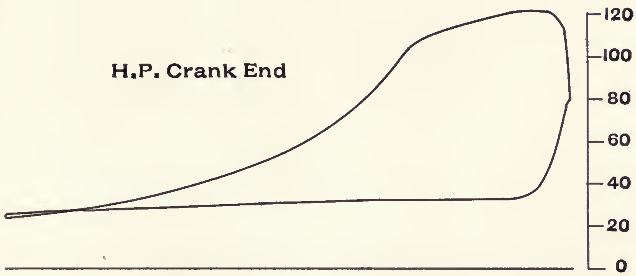
		H.P.Cyl.	L.P. Cyl.	H.P.Cyl.	L.P. Cyl.
Initial pressure above atmosphere	lbs.	118.8	28.01	125.3	30.3
Corresponding steam-pipe pressure	lbs.	129.2		136.	
Cut-off pressure above zero	lbs.	112.37	34.31	116.52	34.91
Release pressure above zero	lbs.	39 81	13.88	51.76	18.26
Mean effective pressure	lbs.	34.66	9.66	49.03	14.22
Back pressure at mid stroke above atmosphere	lbs.	32.0	2.2	34.3	2.9
Proportion of stroke completed at cut-off		.309	.338	.419	.474
Steam accounted for at cut-off	lbs.	17.83	15.81	17.62	16.03
Steam accounted for at release	lbs.	20.34	18.61	18.43	17.54
Proportion of feed-water accounted for at cut-off		.806	.715	.816	.743
Proportion of feed-water accounted for at release		.919	.842	.854	.813

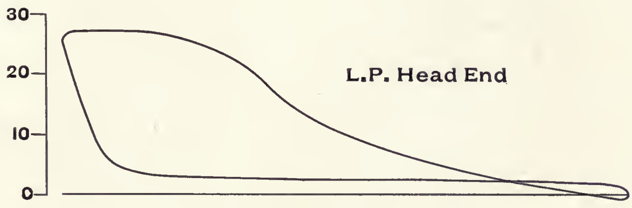
Engine No. 46 is a cross-compound, with horizontal unjacketed cylinders and unjacketed receiver. The valves are all plain slides. The steam was drawn from horizontal water-tube boilers, and contained 0.7 % of moisture by calorimeter test. The load was miscellaneous iron-working machinery. Tests were made with two different loads. The valves and pistons were all in excellent condition as regards leakage.

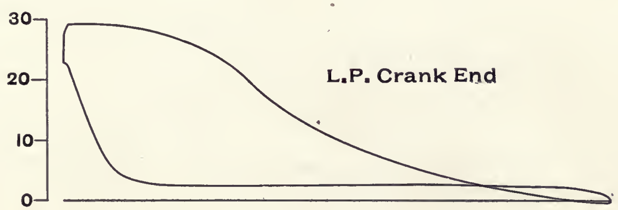
It is evident from an analysis of the diagrams in these tests, that the economy was as high as could be expected under the conditions of boiler pressure, ratio of cylinder areas, and cut-off. For higher economy a higher pressure, larger ratio of cylinder area, and earlier cut-off are required. It must be noted, however, that the distribution is not the most perfect, and there is rather a high back pressure in the L. P. cylinder.

ENGINE No. 46a

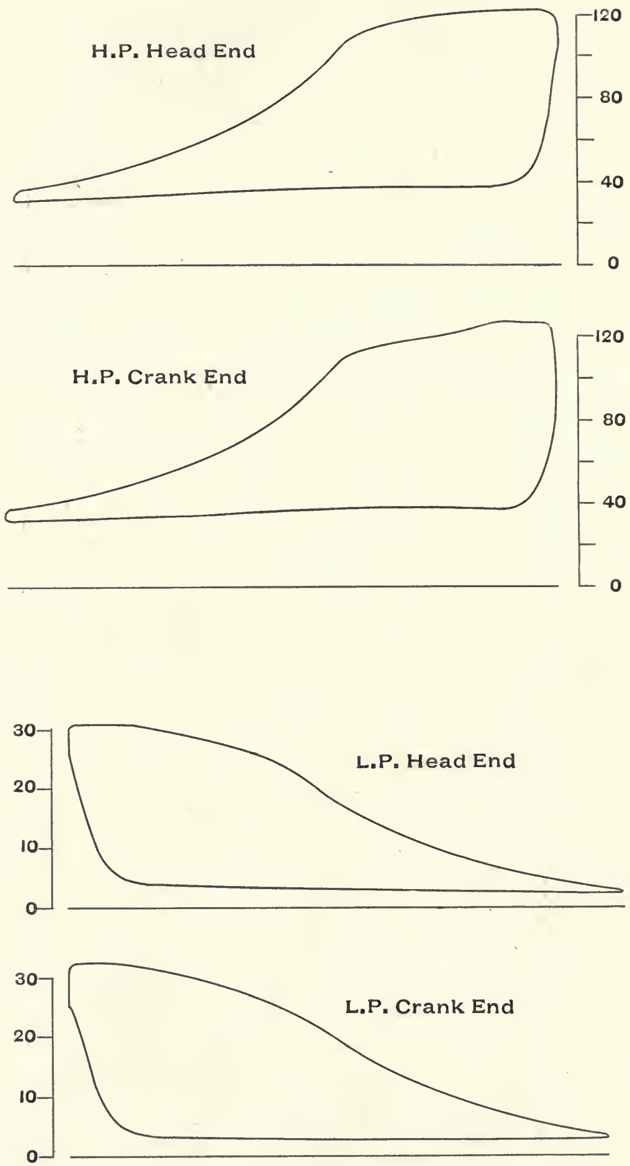








ENGINE No. 46b



ENGINE No. 47.

Compound Condensing Engine.

		H.P. Cylinder.	L.P. Cylinder.
Kind of engine		Four valve	
Number of cylinders		1	1
Diameter of cylinder	ins.	$18\frac{1}{32}$	$44\frac{5}{16}$
Diameter of piston rod	ins.	one $\begin{cases} 4\frac{1}{16} \\ 4\frac{3}{4} \end{cases}$	$4\frac{1}{16}$
Stroke of piston	ft.	6	6
Clearance	%	2.3	1.8
H. P. constant for one lb. m. e. p. one rev. per min.	H.P.	.08728	.5584
Ratio of areas of cylinders		1	6.398
Inside diameter of steam pipe	ins.	8	14
Inside diameter of exhaust pipe	ins.	9	16
Condition of valves and pistons regarding leakage		Practically tight	Considerable leakage

Data and Results of Feed-Water Tests.

Test. Conditions regarding use of jackets.		A. Jackets off.	B. Jackets on.	C. Jackets on.	D. Jackets on.
Character of steam		Ordinary			
Duration	hrs.	4.817	4.833		
Weight of dry steam consumed	lbs.	42,147.	42,643.		
Dry steam consumed per hour	lbs.	8,749.8	8,823.3	8,596.6	8,707.
Pressure in steam pipe above atmosphere	lbs.	150.7	151.1	150.5	151.4
Pressure in receiver above atmosphere	lbs.	10.6	9.4	15.0	19.4
Vacuum in condenser	ins.	27.	27.3	27.1	28.0
Revolutions per minute		60.31	60.54	60.59	60.33
Mean effective pressure, H. P. cylinder	lbs.	67.79	66.15	59.9	57.13
Mean effective pressure, L. P. cylinder	lbs.	9.87	10.63	10.78	11.39
Indicated horse-power, H. P. cylinder	H.P.	366.8	349.52	316.77	300.85
Indicated horse-power, L. P. cylinder	H.P.	332.52	359.31	364.69	383.71
Indicated horse-power, whole engine	H.P.	689.32	708.33	681.45	684.56
Dry steam consumed per I.H.P. per hr.	lbs.	12.69	12.45	12.61	12.72

Measurements Based on Sample Diagrams. Engine No. 47.

Test.		A.		B.	
		H.P.Cyl.	L.P. Cyl.	H.P.Cyl.	L.P.Cyl.
Initial pressure above atmosphere	lbs.	145.2	9.8	146.1	9.
Corresponding steam-pipe or receiver pressure	lbs.	150.	10.6	149.2	9 5
Cut-off pressure above zero . .	lbs.	149.2	20.	149.3	18.9
Release pressure above zero . .	lbs.	42.5	5.3	41.	5.4
Mean effective pressure . . .	lbs.	68.19	9.78	66.06	10.55
Back pressure at mid stroke, above or below atmosphere .	lbs.	+10.9	−12.1	+10.1	−12.6
Proportion of stroke completed at cut-off		.281	.25	.263	.282
Steam accounted for at cut-off .	lbs.	10.00	8.69	9.22	9.13
Steam accounted for at release .	lbs.	10.34	9.54	9.59	9.61
Proportion of feed-water accounted for at cut-off . . .		.788	.685	.740	.733
Proportion of feed-water accounted for at release . .		.815	.752	.770	.771

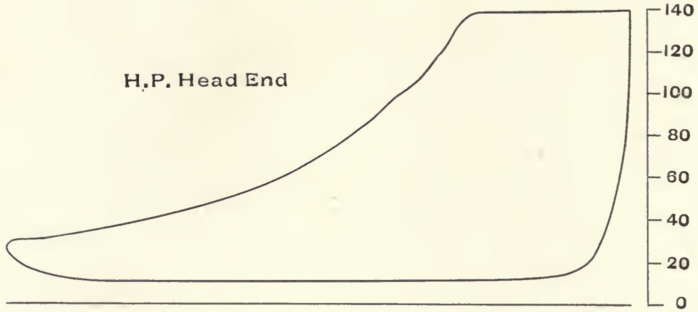
Engine No. 47 is a horizontal tandem compound, with jacketed cylinders and a reheater. The condenser is of the siphon type with water supplied by gravity. The jacketing applies to heads and barrel of the H. P. cylinder, and to the heads but not the barrel of the L. P. cylinder. The reheater is of the tubular type, and contains a sufficient area of surface to superheat the steam passing to the L. P. cylinder, although some of that entering the heater remains in a condensed state, and is drawn off by a trap. The valves are all of the gridiron type. The steam is furnished by horizontal tubular boilers, and it was found by calorimeter test to be practically dry. The valves and piston of the H. P. cylinder were found in good condition. The steam valve at the head end of the L. P. cylinder leaked badly, and the crank-end valve a considerable amount; but as near as could be judged under these circumstances the exhaust valves and piston were fairly tight. The load was cotton machinery. Three tests were made with different receiver pressures, and one test was made with steam shut off from jackets and reheater.

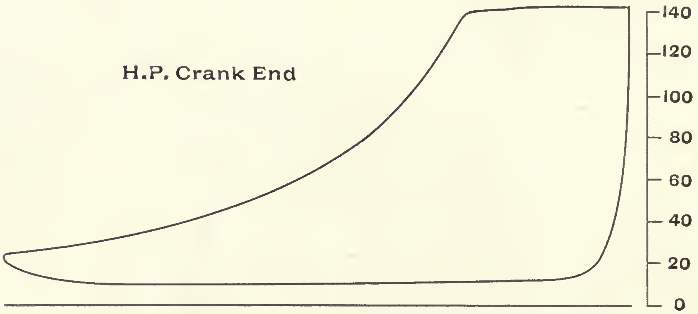
On test B the water condensed in the jackets and reheater tubes amounted to 681 lbs. per hour, or 7.7% of the total. This is included in the total quantity given in the table.

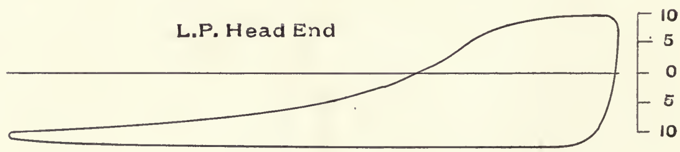
A noticeable feature of these results is the systematic increase in the steam consumption per H. P. per hour as the receiver pressure was raised. This may have been due to the increased leakage of the L. P. steam valves.

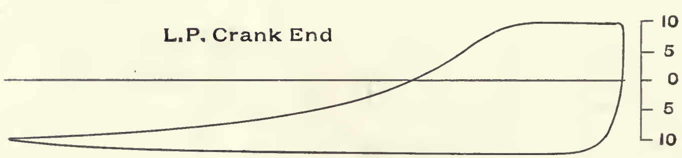
A comparison of the jacket tests reveals a gain due to the jackets of 0.24 lbs. per H. P. per hour, or about 2%. Although this is not a marked difference, it is evident from an analysis of the diagrams that the jackets had a considerable effect upon the distribution of the steam, especially in increasing the power developed by the L. P. cylinder and the quantity of steam accounted for by the diagram for that cylinder.

ENGINE No. 47a

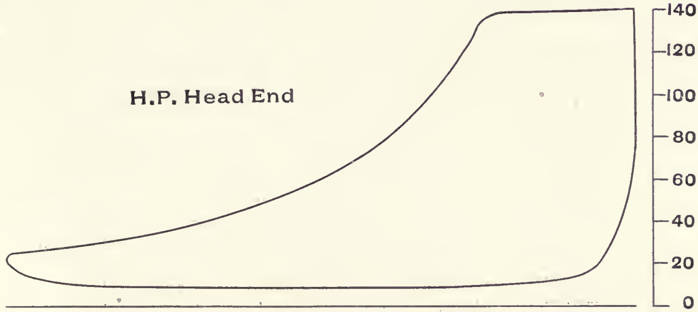


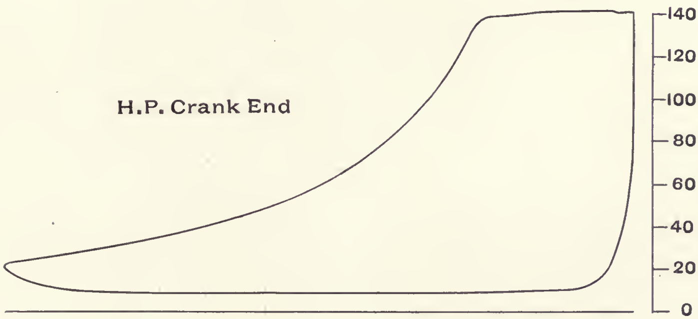


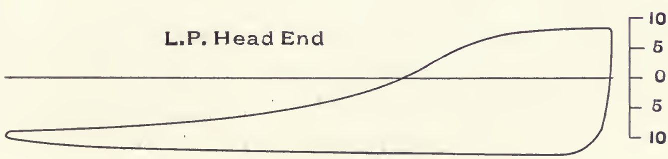


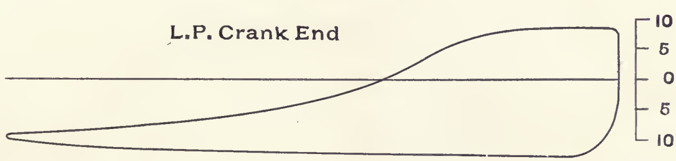


ENGINE No. 47b









ENGINE No. 48.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve	
Number of cylinders		1	1
Diameter of cylinder	ins.	$28\frac{1}{16}$	54
Diameter of piston rod	ins.	5	two 7
Stroke of piston	ft.	5	5
Clearance	%	4	6
Horse-power constant for one lb. m.e.p. one revolution per minute	H.P.	.185	.682
Ratio of areas of cylinders		1	3.69
Condition of valves and pistons regarding leakage		Fairly tight	

Data and Results of Feed-Water Tests.

Test. Conditions regarding use of reheater.		A. Reheater on	B. Reheat'r off.	C. Reheater on.
Character of steam		Sup'd 12°	Sup'd 13°	Sup'd 20°
Duration	hrs.	5.0	5.0	5.0
Total weight of feed-water consumed	lbs.	101,465.	97,856.	105,355.
Total feed-water consumed per hour	lbs.	20,293.	19,571.2	21,071.
Feed-water consumed per hour by air-pump		2,063.	2,030.	1,744.
Feed-water consumed per hour by engine alone	lbs.	18,230.	17,541.2	19,327.
Pressure in steam pipe above atmos.	lbs.	125.9	121.5	100.2
Pressure in receiver above atmosphere	lbs.	11.4	10.1	10.2
Vacuum in condenser	ins.	27.	27.	27.4
Revolutions per minute		77.	76.69	76.68
Mean effective pressure, H. P. cylinder	lbs.	46.11	46.47	45.29
Mean effective pressure L. P. cylinder	lbs.	12.08	11.34	12.22
Indicated horse-power, H. P. cylinder	H.P.	656.51	658.90	642.16
Indicated horse-power, L. P. cylinder	H.P.	634.62	593.31	639.57
Indicated horse-power, whole engine	H.P.	1,291.13	1,252.21	1,281.73
Total feed-water consumed per I.H.P. per hour	lbs.	* 15.72	15.63	16.44
Feed-water per I. H. P. per hour, engine alone	lbs.	14.12	14.01	15.08

* This refers to steam used by the engine alone.

Measurements based on Sample Diagrams.

Test.		A.		B.		C.	
		H.P.Cy.	L.P.Cy.	H.P.Cy.	L.P.Cy.	H.P.Cy.	L.P.Cy.
Initial pressure above atmosphere . . .	lbs.	119.5	13.1	113.3	11.8	94.8	10.
Corresponding steam pipe or receiver pressure	lbs.	127.	12.3	121.	9.8	100.	10.3
Cut-off pres. above zero	lbs.	115.3	22.	112 5	20.7	93.	18.8
Release pres. above zero	lbs.	38.2	8.3	37.7	7.7	39.	8.5
Mean effective press. .	lbs.	46.62	12.29	46.99	11.5	45.11	12.25
Back pressure at mid stroke above or below atmosphere .	lbs.	+18.5	—10.7	+16.5	—10.7	+13.7	—11.3
Proportion of stroke completed at cut-off		.294	.326	.31	.326	.432	.416
Steam accounted for at cut-off	lbs.	11.18	10.66	11.99	10.33	12.41	12.18
Steam accounted for at release . . .	lbs.	12.27	11.55	12.63	11.	13.04	12.25
Proportion of feed water accounted for at cut-off . . .		.792	.755	.856	.737	.823	.808
Proportion of feed water accounted for at release . . .		.869	.818	.901	.785	.865	.812

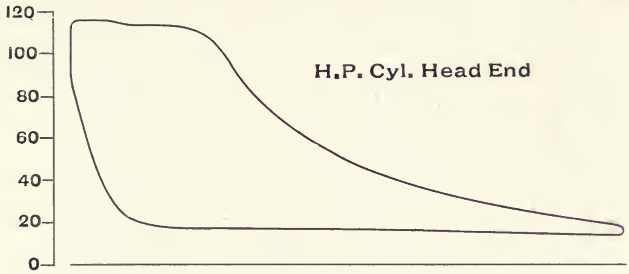
Engine No. 48 is a horizontal cross compound, with unjacketed cylinders and a reheating receiver. The valves are plain slides. The condenser is of the jet type with steam-driven air-pump, the steam used for which was determined and allowed for. The boilers are of the vertical fire-tube type, furnishing slightly superheated steam. The crank end exhaust valve of the H. P cylinder leaked considerably, but with this exception the valves and pistons were practically tight. The load was cotton machinery.

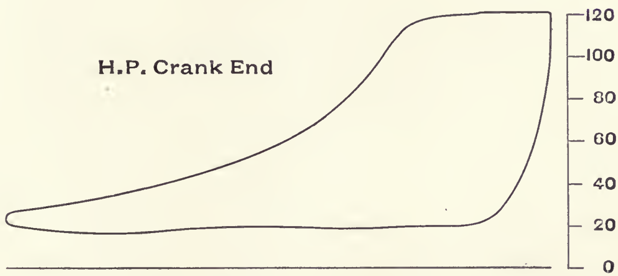
A comparison of tests A and C shows the effect of two widely different pressures upon the economy, one being 125.9 lbs., and the other 100.2 lbs. The reduction of pressure increased the consumption from 14.12 lbs. per I. H. P. per hour to 15.08 lbs., or nearly 7 %.

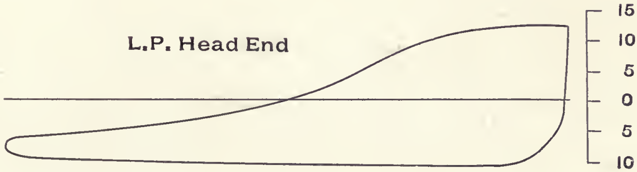
Test B as compared with test A exhibits the effect of shutting off the reheater. There is a slight loss of economy; but as the difference is within the limits of errors of measurement and accidental difference of condition, the most that can be said is that the economy produced by the reheater was in this case inappreciable. On test A there was 648 lbs. of steam per hour condensed and drawn off from the reheater tubes. This is 4 % of the quantity used by the engine. The effect of the heat derived from this source is seen in the increased power developed by the L. P. cylinder, and the increase in the amount of steam accounted for in the L. P. cylinder as compared with that in the H. P. cylinder.

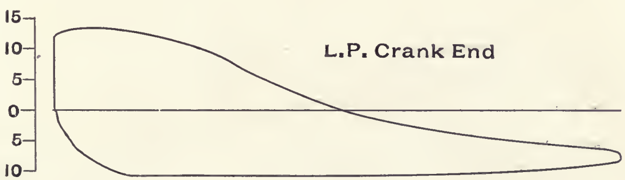
The comparatively large amount of steam used by the air-pump is noticeable, being about 10 % of the entire quantity on Test A.

ENGINE No. 48a

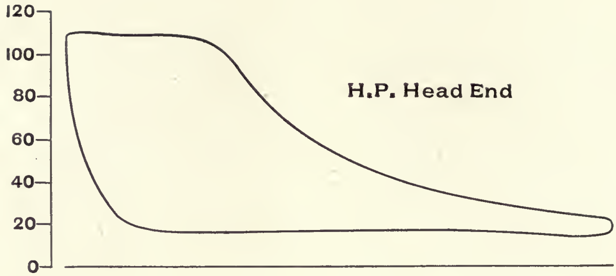


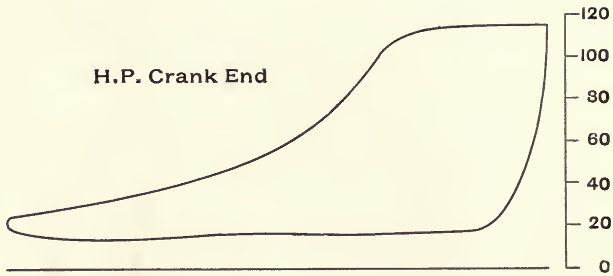


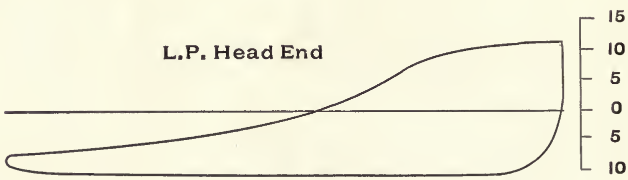


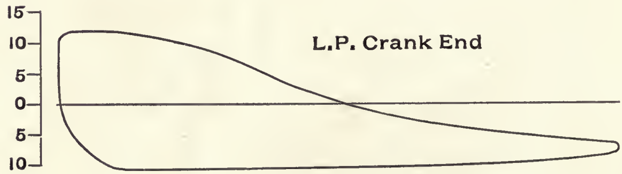


ENGINE No. 48b

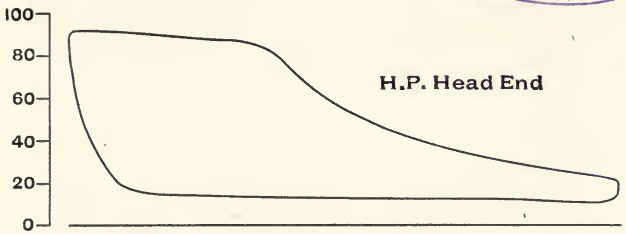


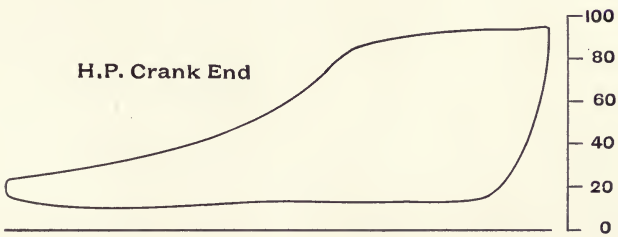


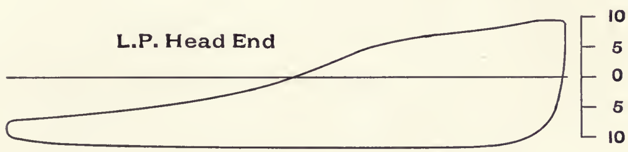


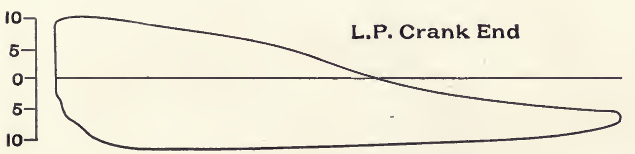


ENGINE No.48c









ENGINE No. 49.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve (Corliss)	
Number of cylinders		1	1
Diameter of cylinder	ins.	24	44
Diameter of piston rod	ins.	$5\frac{5}{16}$	$5\frac{5}{8}$
Stroke of piston	ft.	5	5
Clearance	%	3	$4\frac{1}{2}$
H. P. constant for 1 lb. m. e. p. one revolution per minute	H.P.	.1337	.457
Ratio of areas of cylinders		1	3.43
Condition of valves and pistons regarding leakage		Some leakage	Fairly tight

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	4.75	hrs.
Weight of feed-water consumed	58.832	lbs.
Feed-water consumed per hour	12,385.6	lbs.
Pressure in steam pipe above atmosphere	115.4	lbs.
Pressure in receiver above atmosphere	6.8	lbs.
Vacuum in condenser	28.4	ins.
Revolutions per minute	71.3	rev.
Mean effective pressure, H. P. cylinder	47.93	lbs.
Mean effective pressure, L. P. cylinder	12.71	lbs.
Indicated horse-power, H. P. cylinder	457.9	H.P.
Indicated horse-power, L. P. cylinder	415	H.P.
Indicated horse-power, whole engine	872.9	H.P.
Feed-water consumed per I. H. P. per hour	14.18	lbs.

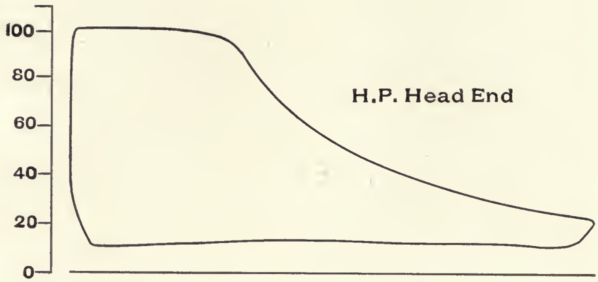
Measurements based on Sample Diagrams.

		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere . .	lbs.	100.9	9.3
Corresponding steam-pipe and receiver pressure	lbs.	114	10.6
Cut-off pressure above zero	lbs.	104.4	19.
Release pressure above zero	lbs.	36.3	8.6
Mean effective pressure	lbs.	49.03	12.71
Back pressure at mid stroke above or below atmosphere	lbs.	+12.7	—11.8
Proportion of stroke completed at cut-off		.33	.404
Steam accounted for at cut-off . . .	lbs.	12.28	10.82
Steam accounted for at release . . .	lbs.	12.95	11.78
Proportion of feed-water accounted for at cut-off		.866	.763
Proportion of feed-water accounted for at release		.913	.831

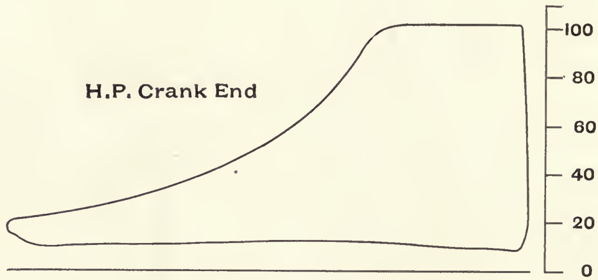
Engine No. 49 is a cross compound horizontal engine with jacketed cylinders, and a jet condenser operated by a direct connected air-pump. The jacket spaces are of the kind which allow the steam to pass through them before entering the steam chest of either cylinder, and the water of condensation drains to waste. Steam is supplied from water-tube boilers through a pipe about 100 feet in length, which is trapped near the throttle valve. Steam lost by condensation in this pipe, and that used for certain heating purposes, was determined independently, and allowed for. The front-end steam valve of the high-pressure cylinder leaked badly. With this exception, the valves and pistons throughout were in fairly good condition. The load was cotton machinery.

Considering the proportion which is borne by the steam accounted for to the feed-water consumption in the high-pressure cylinder, the result of this test, 14.18 lbs. per. I. H. P. per hour, must be considered as an exceptionally good performance.

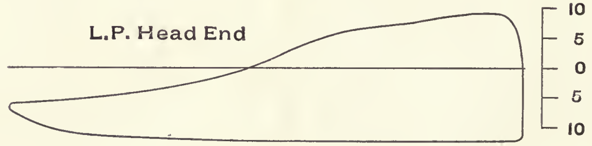
ENGINE No.49



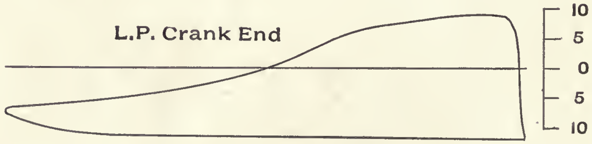
H.P. Head End
100
80
60
40
20
0



H.P. Crank End
100
80
60
40
20
0



L.P. Head End
10
5
0
5
10



L.P. Crank End
10
5
0
5
10

ENGINE No. 50.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder
Kind of engine		Four valve (Corliss)	
Number of cylinders		1	1
Diameter of cylinders	ins.	$24\frac{1}{8}$	$43\frac{3}{8}$
Diameter of piston rod	ins.	$4\frac{11}{16}$	$5\frac{11}{16}$
Stroke of piston	ft.	6	6
Clearance	%	2.5	4
H. P. constant for 1 lb. m. e. p. one revolution per min.	H.P.	.1631	.5577
Ratio of areas of cylinders		1	3.42
Condition of valves and pistons regarding leakage		Some leakage	Fairly tight

Data and Results of Feed-Water Test.

Character of steam	Superheated 30°	
Duration	5	hrs.
Weight of feed-water consumed	53,003	lbs.
Feed-water consumed per hour	10,600.7	lbs.
Pressure in steam pipe above atmosphere	108.1	lbs.
Pressure in receiver above atmosphere	13.4	lbs.
Vacuum in condenser	27.2	ins.
Revolutions per minute	61.4	
Mean effective pressure, H. P. cylinder	35.77	lbs.
Mean effective pressure, L. P. cylinder	12.86	lbs.
Indicated horse-power, H. P. cylinder	358.2	H.P.
Indicated horse-power, L. P. cylinder	440.2	H.P.
Indicated horse-power, whole engine	798.4	H.P.
Feed-water consumed per I. H. P. per hour	13.28	lbs.

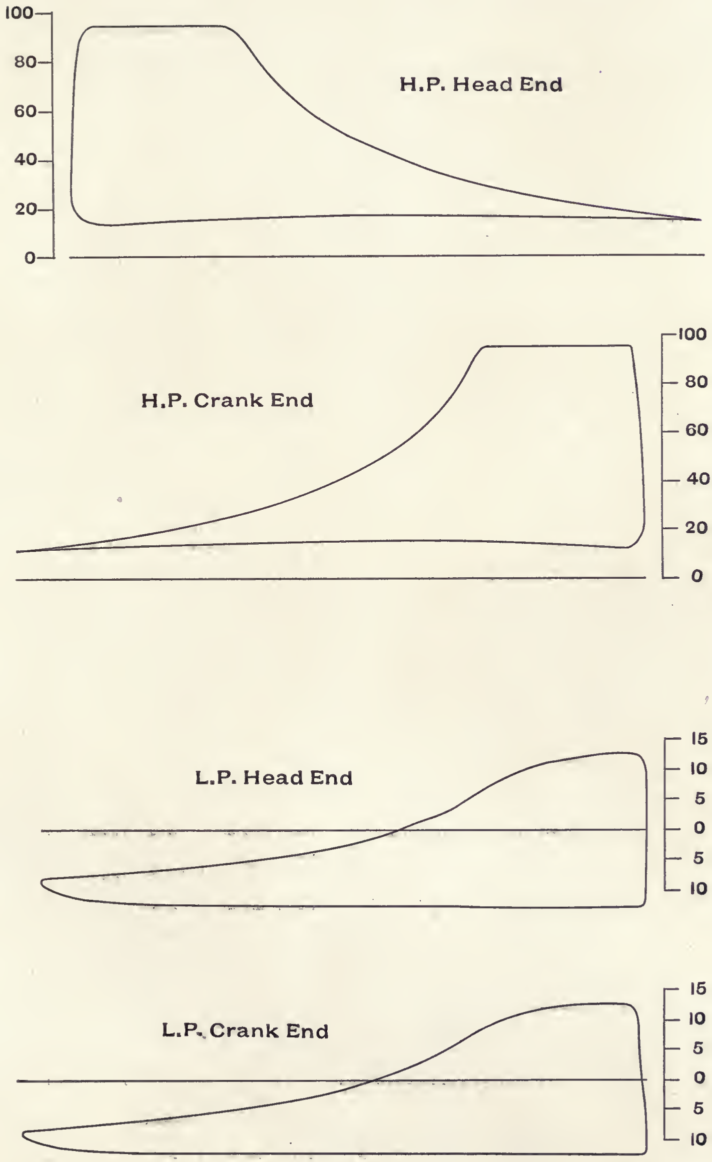
Measurements based on Sample Diagrams.

		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere	lbs.	94.9	12.8
Corresponding steam-pipe and receiver pressure	lbs.	108	13.4
Cut-off pressure above zero	lbs.	102.6	22.4
Release pressure above zero	lbs.	28.1	6.9
Mean effective pressure	lbs.	35.78	12.72
Back pressure at mid stroke, above or below atmosphere	lbs.	+16.3	−12.4
Proportion of stroke completed at cut-off		.274	.27
Steam accounted for at cut-off	lbs.	11.81	10.23
Steam accounted for at release	lbs.	11.97	10.87
Proportion of feed-water accounted for at cut-off		.889	.77
Proportion of feed-water accounted for at release		.901	.819

Engine No. 50 is a cross compound engine with horizontal steam jacketed cylinders and a jet condenser operated by a direct connected air-pump. The jacket of the L. P. cylinder forms a thoroughfare through which the steam is supplied to the steam chest, the steam being admitted through the bottom. The jacket of the H. P. cylinder drains into the receiver. The drip of the receiver and of the low-pressure jacket passes to a pump operated by the engine, and thence to flue heaters, or "regenerators" as they are called, which are located in the flue of the boilers. The steam generated in these heaters returns to the receiver. By this means the low-pressure cylinder receives benefit from some of the heat which would otherwise escape from the boilers to the chimney. Steam is supplied from vertical boilers, which superheat. The exact amount of superheating was not determined; but from the operation of boilers of similar type, the temperature was probably 30° above the normal. The piston of the high-pressure cylinder leaked considerable, but the piston of the low-pressure cylinder and the valves of both were in good condition. The load was that of a cotton mill.

The results of this test are interesting on account of the means provided for reheating the steam and re-evaporating the jacket water for the use of the low-pressure cylinder, employing for this purpose the heat of the waste gases of the boilers. Comparing this test with that made on Engine No. 49, where no such provision was made, the difference is quite marked, being .9 of a pound per I. H. P. per hour, or nearly 7%. It is difficult to determine by this comparison how much, if any, effect was produced by the reheating process, because of the difference in the condition of the steam; and, furthermore, because there was quite a difference in the degree of expansion, the cut-off in the high-pressure cylinder of one engine being .33, and in the other .27. The effect of the reheating is not sufficiently marked to show in the analysis of the diagrams. It appears that there was no more steam accounted for in the low-pressure cylinder in one case than in the other. A greater quantity might be expected if there was a marked effect produced by the reheating.

ENGINE No. 50


H.P. Head End
100
80
60
40
20
0
H.P. Crank End
100
80
60
40
20
0
L.P. Head End
15
10
5
0
5
10
L.P. Crank End
15
10
5
0
5
10

ENGINE No. 51.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve	
Number of cylinders		1	1
Diameter of cylinder	ins.	18	48¼
Diameter of piston rod	ins.	3⅝	4⅝
Stroke of piston	ft.	4	4
Clearance	%	2	2¾
H. P. constant for one lb. m. e. p. one revolution per minute	H.P.	.0604	.4412
Ratio of areas of cylinders		1	7.3
Condition of valves and pistons regarding leakage		Some leakage	Some leakage

Data and Results of Feed-Water Tests.

Test.		A.	B.	C.
Character of steam, degs. superh'g		15.7	16.4	12.2
Duration	hrs.	5	5	5
Weight of feed-water consumed	lbs.	41,210	39,583	39,174
Feed-water consumed per hour		8,242	7,916.6	7,834.8
Pressure in steam pipe above atmos.		149.7	150.4	150.2
Pressure in receiver	lbs.	5.4	9.1	12.9
Vacuum in condenser	ins.	26.9	26.4	26.6
Revolutions per minute	rev.	80.04	80.14	80
Mean effective pressure, H. P. cylinder	lbs.	72.44	65.89	61.9
Mean effective pressure, L. P. cylinder	lbs.	9.03	9.59	10.2
Indicated horse-power, H.P. cylinder	H.P.	350.9	318.9	299.3
Indicated horse-power, L.P. cylinder	H.P.	390.6	339.2	359.8
Indicated horse-power, whole engine	H.P.	670.5	658.1	659.1
Feed-water cons. per I. H. P. per hour	lbs.	12.29	12.03	11.89

Measurements Based on Sample Diagrams.

Test.		A.		C.	
		H.P.	L.P.	H.P.	L.P.
Initial pressure above atmosphere	lbs.	143.9	5.2	143.	14.5
Corresp. steam-pipe and receiver pres.	lbs.	151	*5.5	150.5	*12.8
Cut-off pressure above zero	lbs.	149	15.4	145.9	24.7
Release pressure above zero	lbs.	42.5	5.2	43.	5.2
Mean effective pressure	lbs.	72.7	9.08	62.27	10.23
Back pressure at mid stroke above or below atmosphere	lbs.	+6	−13.1	+16.5	−13
Proportion of stroke completed at cut-off		.285	.323	.285	.176
Steam accounted for at cut-off	lbs.	9.54	9.07	9.21	7.96
Steam accounted for at release	lbs.	9.62	8.88	9.58	8.43
Proportion of feed-water accounted for at cut-off		.77	.732	.769	.654
Proportion of feed-water accounted for at release		.776	.716	.8	.704

* Not corrected.

Note. — The weight of steam condensed in all the jackets averaged, for the three trials, 9.5 % of the total steam consumed; and this is included in the quantities given.

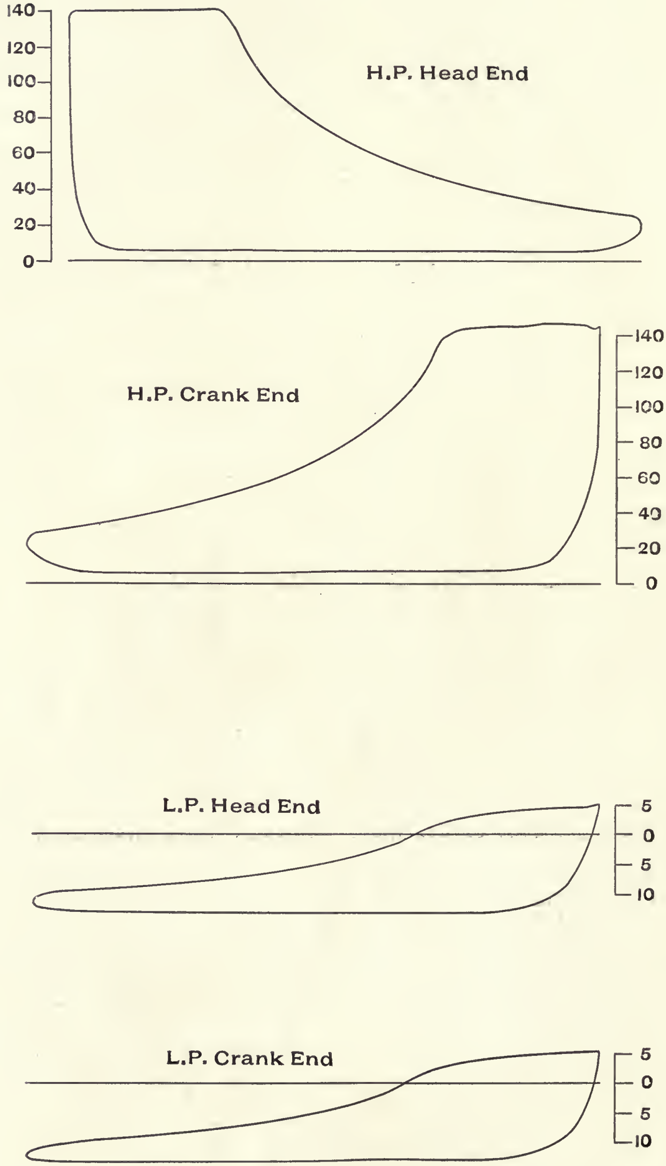
Engine No. 51 is a horizontal cross compound, with jacketed cylinders and a reheater. The condenser is of the siphon type, and water is supplied by gravity. Both the barrel and the heads of the high-pressure cylinder are jacketed, but only the barrel of the L. P. cylinder. The reheater has sufficient surface to superheat the steam, although not sufficient to prevent some water condensing in the bottom of the shell, from which it is drawn away by a trap. The jackets are drained by traps which discharge to waste. The valves are all of the gridiron type. Steam is supplied from vertical boilers which superheat. The piston of the H. P. cylinder was found to show some leakage. The low-pressure exhaust valve at the crank end leaked quite badly. The piston of the L. P. cylinder and the remaining valves were in good condition. Three tests were made, using three different receiver pressures.

These tests are of interest on account of the unusual ratio of volumes of the cylinders. This ratio is about the same as that which is common practice between the low-pressure and high-pressure cylinder of a triple expansion engine. This large ratio taken in conjunction with the high initial pressure, and the fact that the steam was slightly superheated, furnishes an explanation for the economical results obtained, which are unusual.

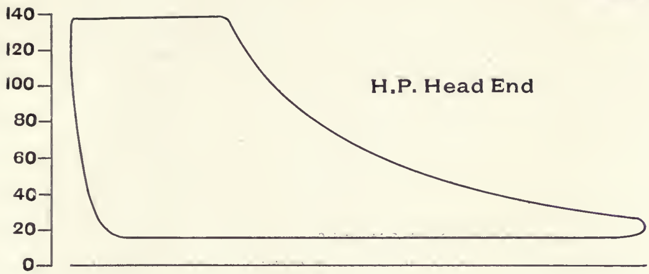
Comparing the three tests together, it appears that there was a gradual improvement produced by increasing the receiver pressure, the best result being obtained when that pressure was the highest.

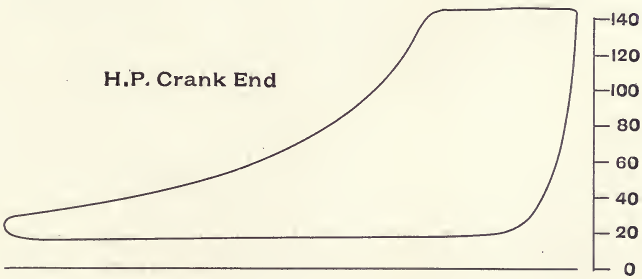
In this connection, it is noticeable that as the receiver pressure increased, and the cut-off in the low-pressure cylinder became less, the steam accounted for by the low-pressure cylinder became less.

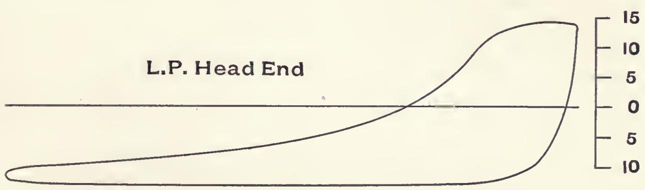
ENGINE No.51a

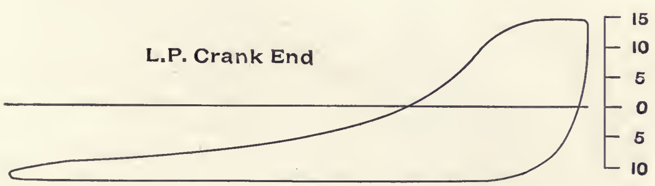


ENGINE No.51c









ENGINE No. 52.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve	
Number of cylinders		2	2
Diameter of cylinder	ins.	$14\frac{1}{16}$	$36\frac{5}{32}$
Diameter of piston rod	ins.	$3\frac{1}{4}$	one $3\frac{1}{4}$ one $4\frac{1}{2}$
Stroke of piston	ft.	4	4
Clearance	%	2	$2\frac{1}{2}$
H. P. Constant for one lb. m.e.p., one rev. per minute	H.P.	.0367 each	.2456 each
Ratio of areas of cylinders		1	6.69 both
Condition of valves and pistons regarding leakage		Some leakage	Some leakage

Data and Results of Feed-Water Tests.

Test. Jackets on or off.		A. On.	B. On.	C. On.	D. Off.
Character of steam		Ordinary	Ordinary	Ordinary	Ordinary
Duration	hrs.	5.0	5.06	5.03	4.5
Weight of feed-water consumed	lbs.	47,045.0	49,720.0	48,389.0	43,321.0
Feed-water consumed per hour	lbs.	9,409.0	9,812.5	9,614.4	9,626.9
Pres. in steam pipe above atmos.	lbs.	144.2	144.1	143.8	144.1
Pres. in receiver above atmos.	lbs.	4.9	8.6	12.2	12.3
Vacuum in condenser	ins.	25.3	25.2	25.0	24.9
Revolutions per minute . . .	rev.	77.45	76.65	76.86	78.89
Mean effective pressure, H.P. cyl.	lbs.	61.12	60.82	57.29	60.53
Mean effective pressure, L.P. cyl.	lbs.	9.76	10.6	11.0	9.66
Indicated horse-power, H.P. cyl.	H.P.	347.63	342.48	323.48	350.79
Indicated horse-power, L.P. cyl.	H.P.	371.22	398.84	415.29	374.41
Indicated H. P. whole engine .	H.P.	718.85	741.32	738.77	725.2
Feed-water consumed per I.H.P. per hour	lbs.	13.09	13.23	13.01	13.27

Measurements based on Sample Diagrams.

Test. Jackets on or off.		C. On.	C. On.	D. Off.	D. Off.
		H. P.	L. P.	H. P.	L. P.
Initial pressure above atmosphere . .	lbs.	139.7	13.0	139.9	11.7
Corresponding steam-pipe and receiver pressure above atmosphere . .	lbs.	145.0	12.3	144.5	12.8
Cut-off pressure above zero	lbs.	140.0	22.4	140.0	21.4
Release pressure above zero	lbs.	41.3	5.6	44.0	5.2
Mean effective pressure	lbs.	57.64	11.07	60.3	9.58
Back pressure at mid stroke above or below atmosphere	lbs.	+13.8	−12.2	+14.1	−12.2
Proportion of stroke completed at cut-off		.268	.236	.293	.213
Steam accounted for at cut-off . . .	lbs.	8.48	10.23	9.75	8.71
Steam accounted for at release . . .	lbs.	9.26	10.62	10.27	9.79
Proportion of feed-water acc. for at cut-off		.651	.786	.734	.655
Proportion of feed-water accounted for at release		.712	.816	.774	.73

Engine No. 52 is a pair of horizontal tandem compounds, having jacketed cylinders and reheating receivers, the condensers being of the siphon type, to which water is supplied by gravity. The H. P. cylinders are jacketed all over, but the L. P. cylinders have only the barrels jacketed. As in the case of Engine No. 51, the reheaters are provided with sufficient surface to superheat the steam passing to the low-pressure cylinders, the water which remains being trapped. The valves are all of the gridiron type. Steam is supplied by horizontal return tubular boilers, and at the throttle valves it contained .1 of 1 % of moisture. The H. P. pistons leaked to some extent, and the head-end exhaust valve of the left-hand low-pressure cylinder leaked a considerable amount. The pistons of the H. P. cylinder and the remaining valves were fairly tight. The load was cotton machinery.

Three trials were made with three different receiver pressures, and a fourth trial with steam shut off from the jackets and the reheating tubes.

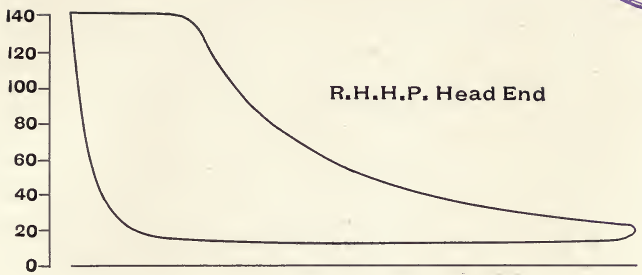
Comparing the results of these tests with those made on Engine No. 51, which is of the same general type and of about the same power, but having only half the number of cylinders, there is a striking difference. This engine did not have the benefit of superheated steam as did Engine No. 51, and this difference in the conditions must be taken into account; but it is hardly possible that the whole of the difference, which is about 9 %, could be produced in this way. There may be some difference, also, in the amount of leakage of the two engines. Engine No. 51 had the benefit of the best vacuum. Making all allowances for these differences, it is quite certain that the size of the cylinders had some effect upon the results. The action of the steam in the cylinders is quite different in No. 52 from what it is in No. 51; but it will be noticed that steam accounted for in the low-pressure cylinders is greater than that shown in the high pressure cylinders, whereas in Engine No. 51 the contrary is true.

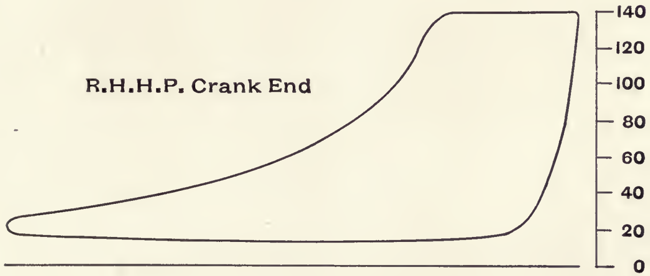
Comparing Test "C" with jackets on, and Test "D" with jackets off, the difference in the economy shown is only .26 of a pound, or about 2 %. The nature of the action which the jackets produced is shown in the analysis of the diagrams. With the jackets on, the steam accounted for in the low-pressure cylinder is the greatest; whereas, with the jackets off, the steam accounted for in that cylinder is the least.

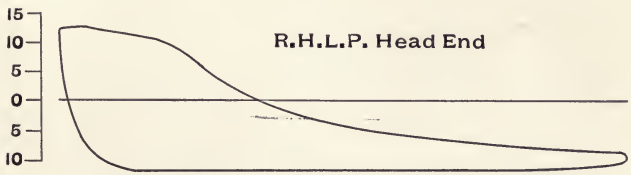
Another noticeable thing in the action of the jackets is in the distribution of the power between the cylinders. With jackets on, the low-pressure cylinders developed 92 horse-power more than the high-pressure cylinders, or about 30 %; whereas, with jackets off, the increase was only 24 horse-power, or about 7 %.

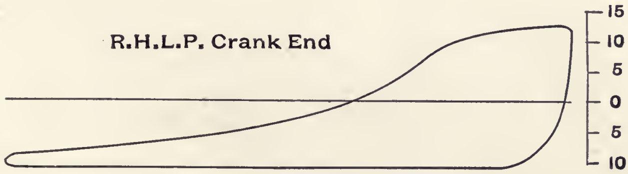
NOTE. — The quantity of steam condensed in the jackets on the first three trials was respectively, 11.4 %, 10.8 %, and 10.8 % of the total quantity consumed; and these are included in the figures given in the tables.

ENGINE No. 52c

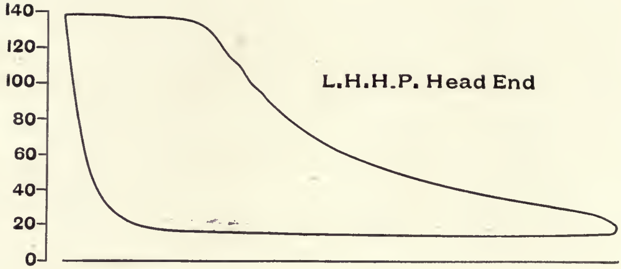


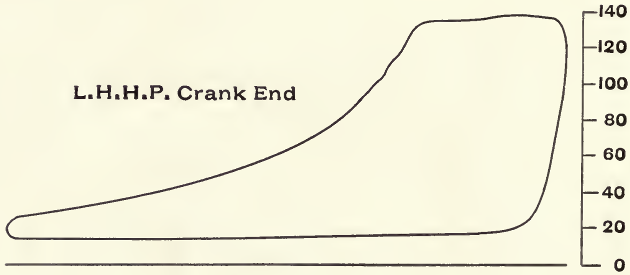


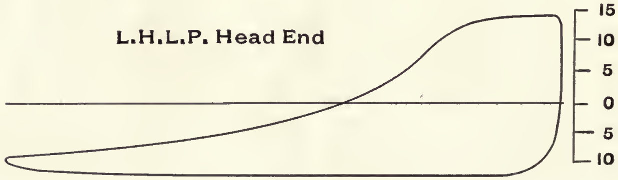


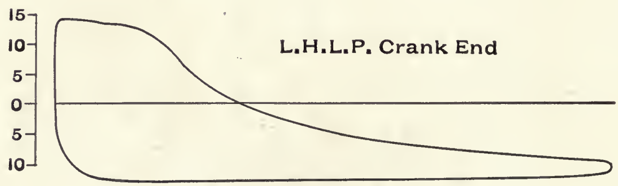


ENGINE No. 52c









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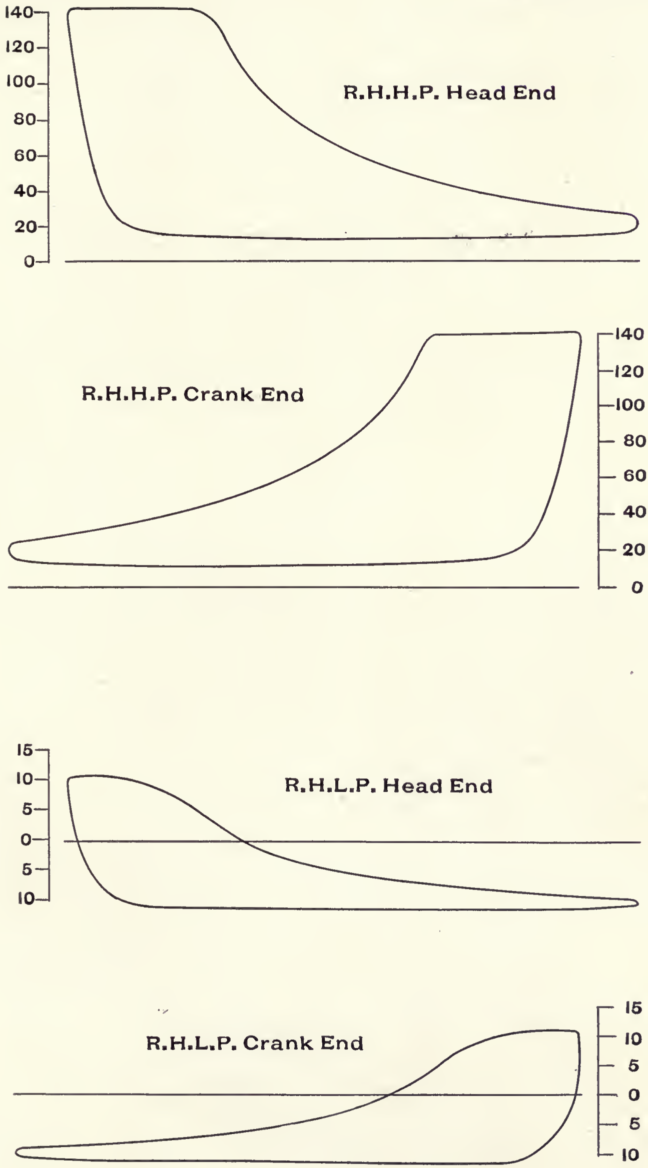
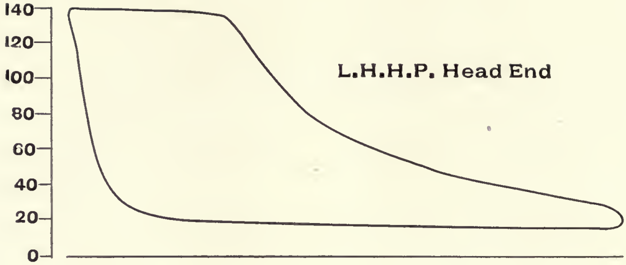
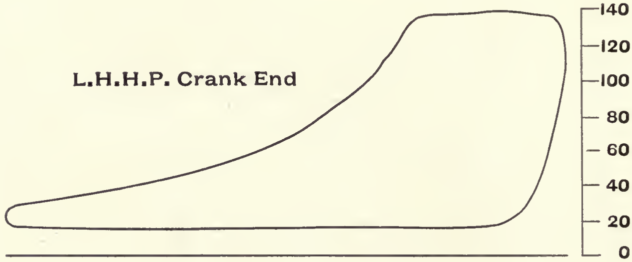
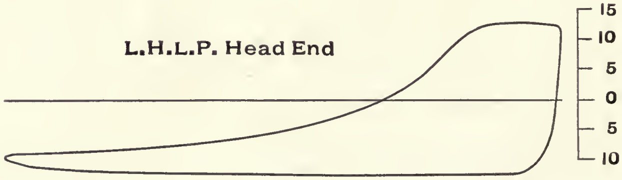
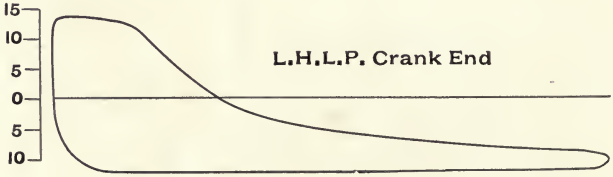


ENGINE No. 52d









ENGINE No. 53.

Compound Condensing Engine.

		H.P. Cylinder.	L.P. Cylinder.
Kind of engine		Four valve (Corliss)	
Number of cylinders		1	1
Diameter of cylinder	ins.	18	30
Diameter of piston rod	ins.	$3\frac{1}{8}$	one $\begin{cases} 3\frac{1}{8} \\ 4\frac{1}{2} \end{cases}$
Stroke of piston	ft.	4	4
Clearance	%	3	3
H. P. constant for one lb. m. e. p., one rev. per min.	H.P.	.0608	.1685
Ratio of areas of cylinders		1	2.77
Condition of valves and pistons regarding leakage		Fairly tight	Fairly tight

Data and Results of Feed-Water Tests.

Character of steam	Ordinary	
Duration	3.0	hrs.
Weight of feed-water consumed	14,195.0	lbs.
Feed-water consumed per hour	4,731.7	lbs.
Pressure in steam pipe above atmosphere	114.9	lbs.
Pressure in receiver above atmosphere	15.4	lbs.
Vacuum in condenser	25.6	ins.
Revolutions per minute	65.5	rev.
Mean effective pressure, H. P. cylinder	35.07	lbs.
Mean effective pressure, L. P. cylinder	14.27	lbs.
Indicated horse-power, H. P. cylinder	142.1	H.P.
Indicated horse-power, L. P. cylinder	157.6	H.P.
Indicated horse-power, whole engine	299.7	H.P.
Feed-water consumed per I. H. P. per hour	15.78	lbs.

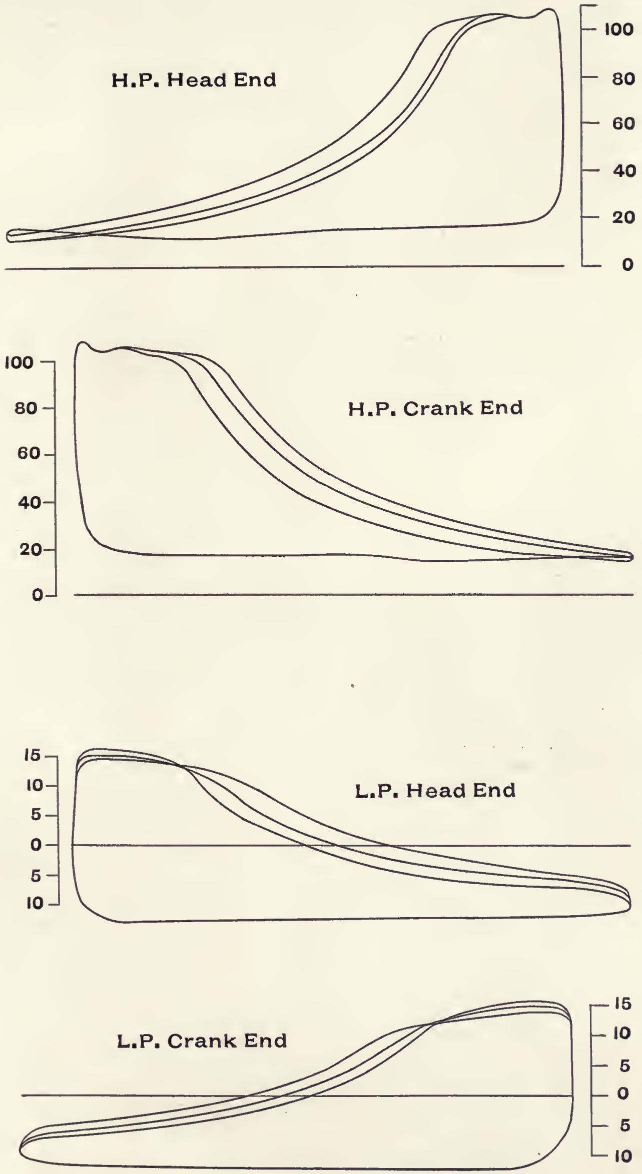
Measurements based on Sample Diagrams.

		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above zero	lbs.	111.9	14.6
Cut-off pressure above zero	lbs.	111.7	33.2
Release pressure above zero	lbs.	30.8	8.7
Mean effective pressure	lbs.	36.65	14.52
Back pressure at mid stroke, above or below atmosphere	lbs.	+ 16.7	− 11.8
Proportion of stroke completed at cut-off .		.238	
Steam accounted for at cut-off	lbs.	11.31	13.01
Steam accounted for at release	lbs.	9.47	10.76
Proportion of feed-water accounted for at cut-off		.717	.60
Proportion of feed-water accounted for at release		.825	.682

Engine No. 53 is a horizontal tandem compound, with unjacketed cylinders and jet condenser operated by an independent air-pump. The steam from the air-pump was taken from an auxiliary boiler. The boilers are of the water-tube vertical type, furnishing steam slightly superheated. The steam passes through a reservoir at the engine, which is drained by a trap discharging to waste. The valves and pistons of both cylinders were found to be in fairly good condition throughout. The load on the engine was rubber-grinding machinery and somewhat variable in its character.

The economy shown by this test, 15.78 lbs. per I. H. P. per hour, is rather low compared with other compound engines of this type. The explanation of this result is found in part, at least, in the small ratios of volumes of the two cylinders. The diagrams show the variable character of the load.

ENGINE No. 53


H.P. Head End
100
80
60
40
20
0
H.P. Crank End
100
80
60
40
20
0
L.P. Head End
15
10
5
0
5
10
L.P. Crank End
15
10
5
0
5
10

ENGINE No. 54.

Compound Non-Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Single valve	
Number of cylinders		1	1
Diameter of cylinder	ins.	12	20
Diameter of piston rod	ins.	2¼	2¼
Stroke of piston	ins.	13	13
Clearance	%	11	8
H.P. constant for one lb. m.e.p., one revolution per minute	H.P.	.0073	.0205
Ratio of areas of cylinders		1	2.81
Condition of valves and pistons regarding leakage		Some leakage	

Data and Results of Feed-Water Tests.

Test.		A.	B.	C.
Character of steam		Ordinary	Ordinary	Ordinary
Duration	hrs.	5.0	5.0	5.0
Weight of feed-water consumed	lbs.	17,918.0	19,815.0	25,730.0
Feed-water consumed per hour	lbs.	3,583.6	3,963.0	5,146.0
Pressure in steam pipe above atmos.	lbs.	166.9	166.8	164.6
Pressure in receiver above atmosphere	lbs.	28.2	46.3	60.6
Revolutions per minute	rev.	275.7	271.2	273.4
Mean effective pressure, H. P. cylinder	lbs.	29.06	48.37	52.31
Mean effective pressure, L. P. cylinder	lbs.	7.94	16.5	24.58
Indicated horse-power, H. P. cylinder	H.P.	58.5	95.77	104.41
Indicated horse-power, L. P. cylinder	H.P.	44.88	91.74	137.77
Indicated horse-power, whole engine	H.P.	103.37	187.51	242.18
Feed-water consumed per I. H. P. per hour	lbs.	24.99	21.14	21.25

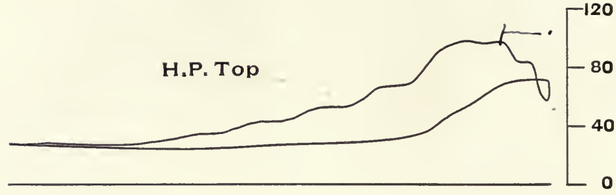
Measurements based on Sample Diagrams, Test B.

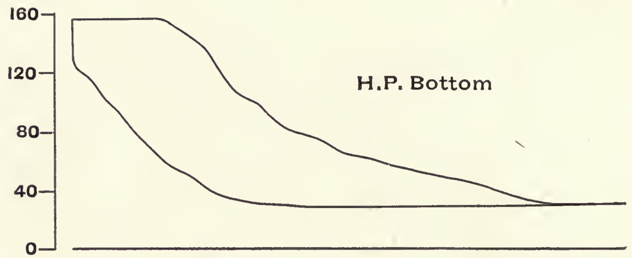
		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere	lbs.	157.3	43.0
Corresponding steam-pipe and receiver pressure	lbs.	168.5	46.0
Cut-off pressure above zero	lbs.	147.5	39.9
Release pressure above zero	lbs.	77.4	23.9
Mean effective pressure	lbs.	47.98	16.36
Back pressure at mid stroke above atmosphere	lbs.	41.6	2.00
Proportion of stroke completed at cut-off		.479	.499
Steam accounted for at cut-off	lbs.	15.73	14.71
Steam accounted for at release	lbs.	15.41	14.51
Proportion of feed-water accounted for at cut-off		.744	.696
Proportion of feed-water accounted for at release		.728	.686

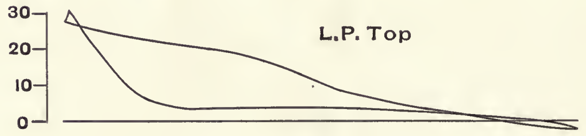
Engine No. 54 is a vertical cross compound with unjacketed cylinders. Each cylinder has a single balanced slide valve, and the speed is controlled by a shaft governor operating on the H. P. valve. The steam is drawn from water-tube boilers through a considerable length of pipe, having headers and separators which were thoroughly drained, and a calorimeter attached near the engine showed that it was practically dry. Steam condensed from the pipes was trapped and properly allowed for. The load consisted of two dynamos located on the engine shaft. The valves of both cylinders leaked a small amount, and the piston of the H. P. cylinder leaked considerably at full pressure. The low-pressure piston was tight. Three tests were made with three different loads.

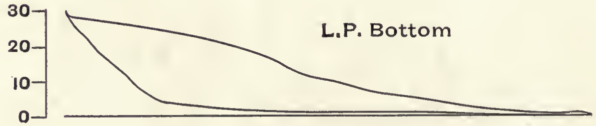
In these tests there is substantial agreement between the results of tests "B" and "C"; the former being made under conditions of a medium load, and the latter under what would be considered an overload. This reveals the advantage of compounding in engines of this class, where by this means the benefits of expansion in the engine as a whole are realized without suffering the losses produced in either cylinder due to early cut-offs.

ENGINE No. 54a

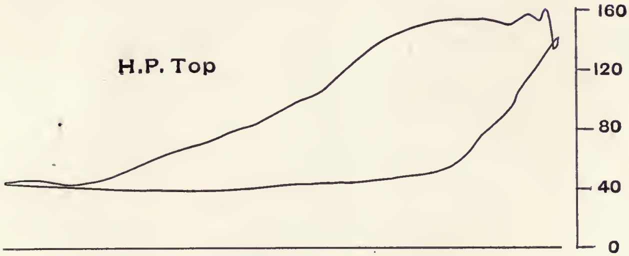


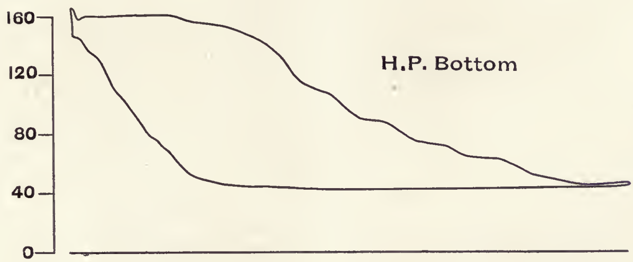


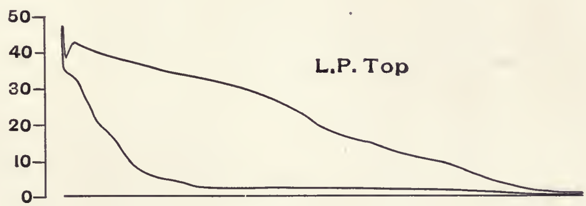


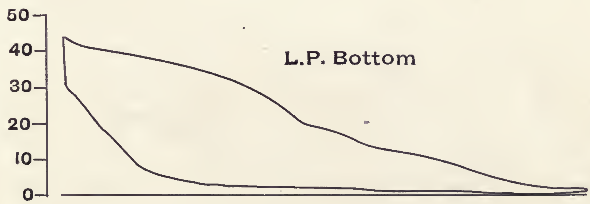


ENGINE No. 54b

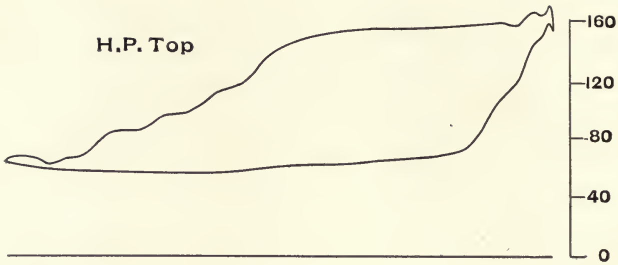


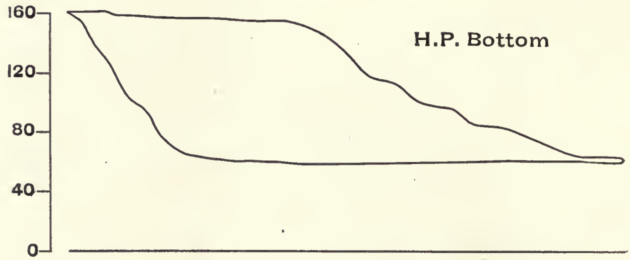


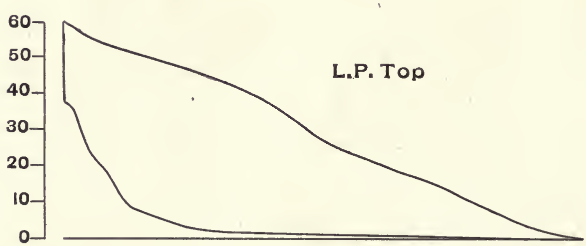


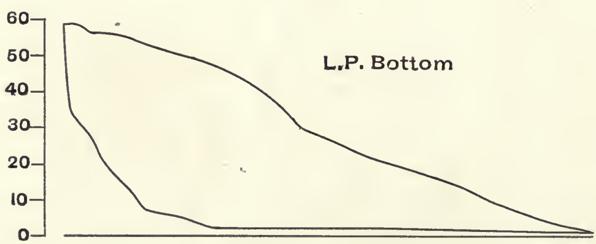


ENGINE No. 54c









ENGINE No. 55.

Compound Condensing Engine.

	H. P. Cylinder.	L. P. Cylinder.
Kind of engine	Four valve (Corliss)	
Number of cylinders	1	1
Diameter of cylinder . . . . . ins.	28	56
Diameter of piston rod . . . . ins.	5¾	6½
Stroke of piston . . . . . . ft.	5	5
Clearance . . . . %	3.1	4.3
H.P. Constant for one lb. m. e. p., one rev. per min. . . . . H.P.	.1827	.7413
Ratio of areas of cylinders . . .	1	4.06
Condition of valves and pistons regarding leakage	Practically tight	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	9.5	hrs.
Weight of feed-water consumed	216,002.45	lbs.
Feed-water consumed per hour	22,737.1	lbs.
Pressure in steam pipe above atmosphere	151.5	lbs.
Pressure in receiver above atmosphere (not verified)	11.8	lbs.
Vacuum in condenser	26.8	ins.
Revolutions per minute	75.18	rev.
Mean effective pressure, H. P. cylinder	61.76	lbs.
Mean effective pressure, L. P. cylinder	15.53	lbs.
Indicated horse-power, H. P. cylinder	848.27	H.P.
Indicated horse-power, L. P. cylinder	865.58	H.P.
Indicated horse-power, whole engine	1,713.85	H.P.
Feed-water consumed per I. H. P. per hour	13.27	lbs.

Measurements based on Sample Diagrams.

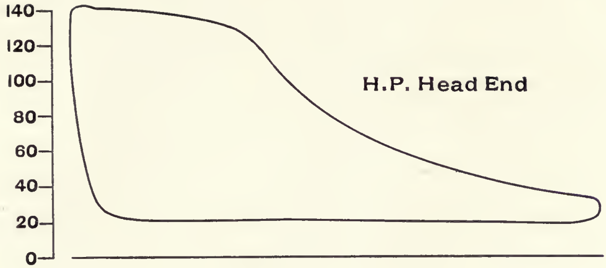
	H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere . . lbs.	143.0	15.5
Corresponding steam-pipe or receiver pressure . . . . lbs.	151.0	*11.8
Cut-off pressure above zero . . . . lbs.	152.3	22.0
Release pressure above zero . . . . lbs.	44.4	9.5
Mean effective pressure . . . . . lbs.	61.53	15.84
Back pressure at mid stroke above or below atmosphere . . . . lbs.	+ 20.0	− 11.5
Proportion of stroke completed at cut-off	.326	.421
Steam accounted for at cut-off . . . lbs.	10.84	10.98
Steam accounted for at release . . . lbs.	10.84	10.85
Proportion of feed-water accounted for at cut-off	.817	.828
Proportion of feed-water accounted for at release	.817	.818

* Not verified.

Engine No. 55 is a cross compound with horizontal unjacketed cylinders and a reheating receiver. The steam is exhausted into a surface condenser operated by an independent steam-driven air and circulating pump. The quantity of steam used by the condenser was determined independently and allowed for. The steam is taken from vertical water-tube boilers in a slightly superheated state. The valves and pistons of both cylinders were found to be in excellent condition throughout, with practically no leakage. The load was an electric generator located on the main shaft, furnishing current for motors in a cotton mill. The steam condensed in the reheater coil amounted to three and one half per cent of the total weight of steam passing the throttle valve. This is included in the quantities given in the table.

A noticeable feature in these results is the close agreement between the four quantities given for steam accounted for by the indicator. Three of these are practically equal, and the fourth differs only one per cent.

ENGINE No.55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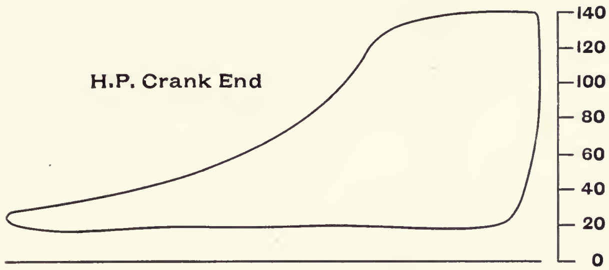
H.P. Head End


H.P. Crank End
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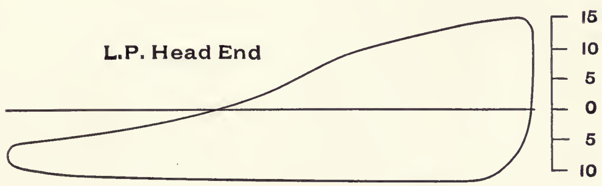
L.P. Head End
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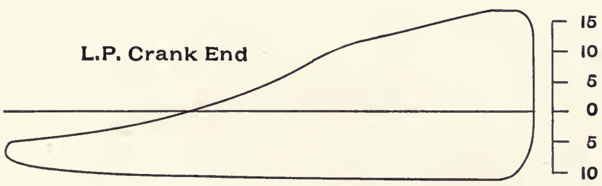
L.P. Crank End
15
10
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0
5
10

ENGINE No. 56.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve (Corliss)	
Number of cylinders		1	1
Diameter of cylinder	ins.	$22\frac{1}{8}$	42
Diameter of piston rod	ins.	$4\frac{1}{2}$	$\begin{cases} 4\frac{1}{2} \\ 5\frac{1}{2} \end{cases}$
Stroke of piston	ft.	3.5	3.5
Clearance	%	3	4
H. P. constant for 1 lb. m. e. p. one revolution per minute	H.P.	.0799	.2897
Ratio of areas of cylinders		1	3.63
Condition of valves and pistons regarding leakage		Some leakage	Some leakage

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	5.0	hrs.
Weight of feed-water consumed	60,636	lbs.
Feed-water consumed per hour	12,127.3	lbs.
Pressure in steam pipe above atmosphere	107.8	lbs.
Pressure in receiver above atmosphere	11.0	lbs.
Vacuum in condenser	25.2	ins.
Revolutions per minute	120.2	rev.
Mean effective pressure, H. P. cylinder	45.07	lbs.
Mean effective pressure, L. P. cylinder	11.30	lbs.
Indicated horse-power, H. P. cylinder	432.93	H.P.
Indicated horse-power, L. P. cylinder	394.12	H.P.
Indicated horse-power, whole engine	827.05	H.P.
Feed-water consumed per I. H. P. per hour	14.67	lbs.

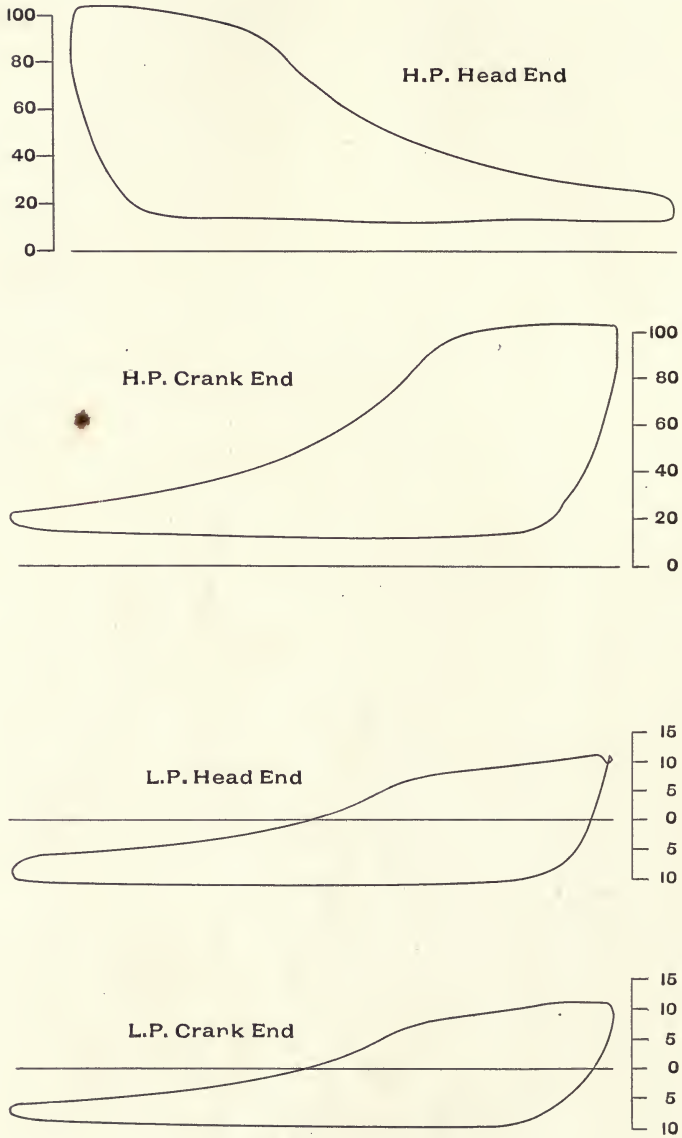
Measurements based on Sample Diagrams.

		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere	lbs.	102.2	11.1
Corresponding steam-pipe and receiver pressure	lbs.	107.0	11.0
Cut-off pressure above zero	lbs.	99.0	20.3
Release pressure above zero	lbs.	36.0	8.5
Mean effective pressure	lbs.	44.93	11.29
Back pressure at mid stroke above or below atmosphere	lbs.	+11.6	−10.5
Proportion of stroke completed at cut-off		.331	.362
Steam accounted for at cut-off	lbs.	11.87	10.46
Steam accounted for at release	lbs.	12.91	11.79
Proportion of feed-water accounted for at cut-off		.813	.716
Proportion of feed-water accounted for at release		.88	.806

Engine No. 56 is a tandem compound with horizontal jacketed cylinders and reheating receiver. Steam is supplied to the bottom of each cylinder, and the jacket spaces form a thoroughfare through which it passes to the steam chest at the top. The jackets are drained by traps which discharge to waste. A jet condenser is used, with an independent steam-driven air-pump, which is supplied from an independent boiler. Steam is taken from horizontal return tubular boilers, and it contained 0.8 % of moisture at a point near the throttle valve. The valves and pistons were found to be in fair condition, but not the best. The load consisted of an electric generator placed on the driving-shaft, which for the test supplied current to a water rheostat.

This is an example of a Corliss engine running at comparatively high rotative speed and piston speed as well, which is generally considered to be one of the conditions which contribute to good economy. The result, however, is nothing unusual. The conclusion cannot fairly be drawn from this test that such a speed produces no advantage; for there were other conditions pertaining to the work, such as the pressure and vacuum, which were unfavorable to economy.

ENGINE No.56



H.P. Head End
100
80
60
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20
0
H.P. Crank End
100
80
60
40
20
0
L.P. Head End
15
10
5
0
5
10
L.P. Crank End
15
10
5
0
5
10

ENGINE No. 57.

Compound Condensing Engine.

		H. P. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve (Corliss)	
Number of cylinders		1	1
Diameter of cylinder	ins.	28	56
Diameter of piston rod	ins.	5½	6
Stroke of piston	ft.	5	5
Clearance	%	2.6	3.7
Horse-power constant for one lb. m.e.p. one revolution per minute	H.P.	.1844	.7425
Ratio of areas of cylinders		1	4.03
Condition of valves and pistons regarding leakage		Practically tight	

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	5.0	hrs.
Weight of feed-water consumed	94,545.	lbs.
Feed-water consumed per hour	18,909.	lbs.
Pressure in steam pipe above atmosphere	133.00	lbs.
Pressure in receiver above atmosphere	13.60	lbs.
Vacuum in condenser	25.20	ins.
Revolutions per minute	66.04	rev.
Mean effective pressure, H. P. cylinder	52.27	lbs.
Mean effective pressure, L. P. cylinder	14.37	lbs.
Indicated horse-power, H. P. cylinder	636.61	H.P.
Indicated horse-power, L. P. cylinder	704.64	H.P.
Indicated horse-power, whole engine	1341.25	H.P.
Feed-water consumed per I. H. P. per hour	14.10	lbs.

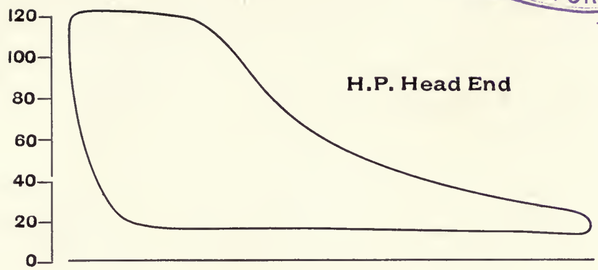
Measurements Based on Sample Diagrams.

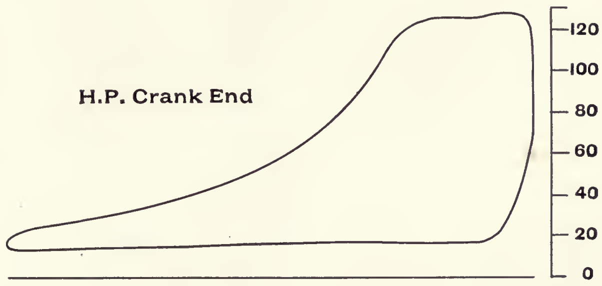
		H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere	lbs.	125.2	13 9
Corresponding steam-pipe and receiver pressure	lbs.	133.0	13.1
Cut-off pressure above zero	lbs.	121.3	20.8
Release pressure above zero	lbs.	38.9	10.1
Mean effective pressure	lbs.	51.87	14.5
Back pressure at mid stroke above or below atmosphere	lbs.	+16.3	−11.4
Proportion of stroke completed at cut-off		.294	.429
Steam accounted for at cut-off	lbs.	9 82	11.22
Steam accounted for at release	lbs.	10.34	11.57
Proportion of feed-water accounted for at cut-off		.696	.796
Proportion of feed-water accounted for at release		.733	.821

Engine No. 57 is a cross compound with unjacketed horizontal cylinders and a reheating receiver. The condenser is of the siphon type, the water for which is supplied by an independent steam pump which takes steam from the main pipe and exhausts into the receiver. Steam is furnished by vertical water-tube boilers in a slightly superheated condition. The load was cotton machinery. The leakage tests showed that the valves and pistons were all in excellent condition throughout, excepting the exhaust valve at the crank end of the low-pressure cylinder, which leaked a considerable amount.

A test was made of the steam consumed by the condenser pump when exhausting into the condenser; and it was found that it used, under these circumstances, 1,176 lbs. per hour, or .9 of a pound per I. H. P. per hour. When exhausting into the receiver, as it did on the test, the consumption was considerably greater; but a large proportion of it was utilized by increasing the power developed in the low-pressure cylinder. It is estimated that .5 of a pound per I. H. P. per hour is chargeable to the condenser pump when used as it was on the main test. The effect of exhausting the pump into the receiver in this way is indicated in the analysis of the diagrams, which shows a considerably larger amount of steam accounted for in the low-pressure cylinder than that shown in the H. P. cylinder.

ENGINE No. 57


120
100
80
60
40
20
0
H.P. Head End


H.P. Crank End
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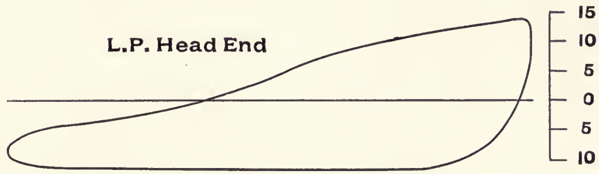
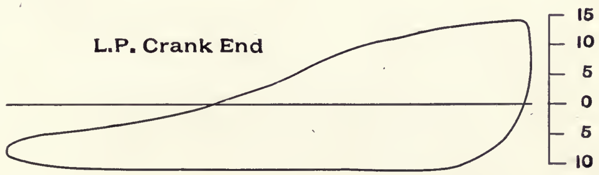
100
80
60
40
20
0


L.P. Head End
15
10
5
0
5
10


L.P. Crank End
15
10
5
0
5
10

ENGINE No. 58.

Compound Condensing Engine.

	H. P. Cylinder.	L. P. Cylinder.
Kind of engine	Four valve (Corliss)	
Number of cylinders	1	1
Diameter of cylinder . . . . ins.	26	50
Diameter of piston rod . . . . . ins.	5	5.5
Stroke of piston . . . . ft.	4	4
Clearance . . . . %	4	4.8
H. P. constant for one lb. m. e. p., one revolution per minute . . . . H.P.	.1269	.4719
Ratio of areas of cylinders	1	3.72
Condition of valves and pistons regarding leakage	Practically tight	Practically tight

Data and Results of Feed-Water Tests.

Test Load.	A. Constant.	B. Variable.
Character of steam	Ordinary	Ordinary
Duration . . . . hrs.	2.5	3.0
Weight of feed-water consumed . . lbs.	34,040.0	34,239.0
Feed-water consumed per hour . . . lbs.	13,616.0	11,413.0
Pressure in steam pipe above atmos. . lbs.	136.2	128.9
Pressure in receiver above atmosphere lbs.	16.8	12.8
Vacuum in condenser . . . . ins.	26.2	26.2
Revolutions per minute	78.0	78.0
Mean effective pressure, H. P. cylinder lbs.	*47.38	
Mean effective pressure, L. P. cylinder lbs.	*15.24	
Indicated horse-power, H. P. cylinder H.P.	*468.97	
Indicated horse-power, L. P. cylinder H.P.	*560.19	
Indicated horse-power, whole engine . H.P.	1,030.06	843.44
Feed-water consumed per I.H.P. per hr. lbs.	13.21	13.53

Measurements based on Sample Diagrams, Test A.

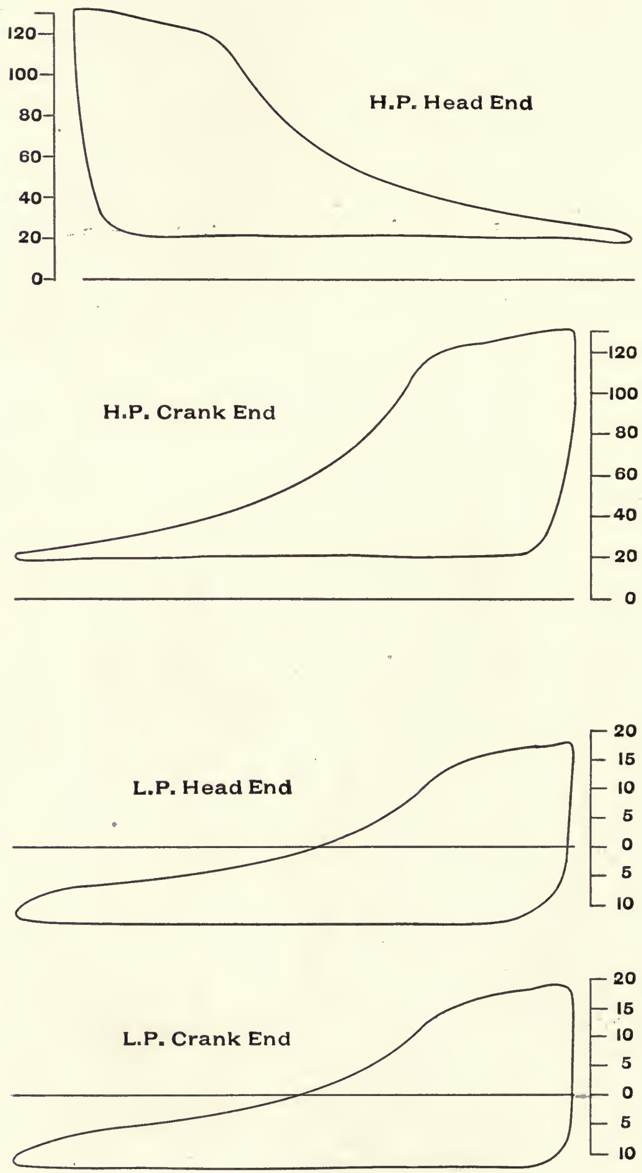
	H. P. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere . . lbs.	131.9	18.2
Corresponding steam-pipe and receiver pressure . . . . lbs.	136.0	16.7
Cut-off pressure above ~~zero~~ atmos . . . lbs.	120.7	125.2 12
Release pressure above zero . . . . lbs.	37.9	8.5
Mean effective pressure . . . . . . lbs.	47.85	15.17
Back pressure at mid stroke, above or below atmosphere . . . . lbs.	+21.2	−12.8
Proportion of stroke completed at cut-off	.293	.274
Steam accounted for at cut-off . . . lbs.	10.56	9.48
Steam accounted for at release . . . lbs.	10.92	9.69
Proportion of feed-water accounted for at cut-off	.8	.718
Proportion of feed-water accounted for at release	.826	.733

Engine No. 58 is a cross compound with horizontal unjacketed cylinders and a reheating receiver. The condenser is of the jet type with an independent steam driven air-pump, the quantity of steam used by which was determined and allowed for. The steam is taken from water-tube boilers, and at the throttle valve was found to contain .2 of one per cent of moisture. The load was an electric generator carried by the fly-wheel shaft, and on Test "A" the current was consumed in a water rheostat, while that of Test "B" was used by the motors of an electric street railroad, and the load was variable. The valves and pistons were in an unusually tight condition throughout.

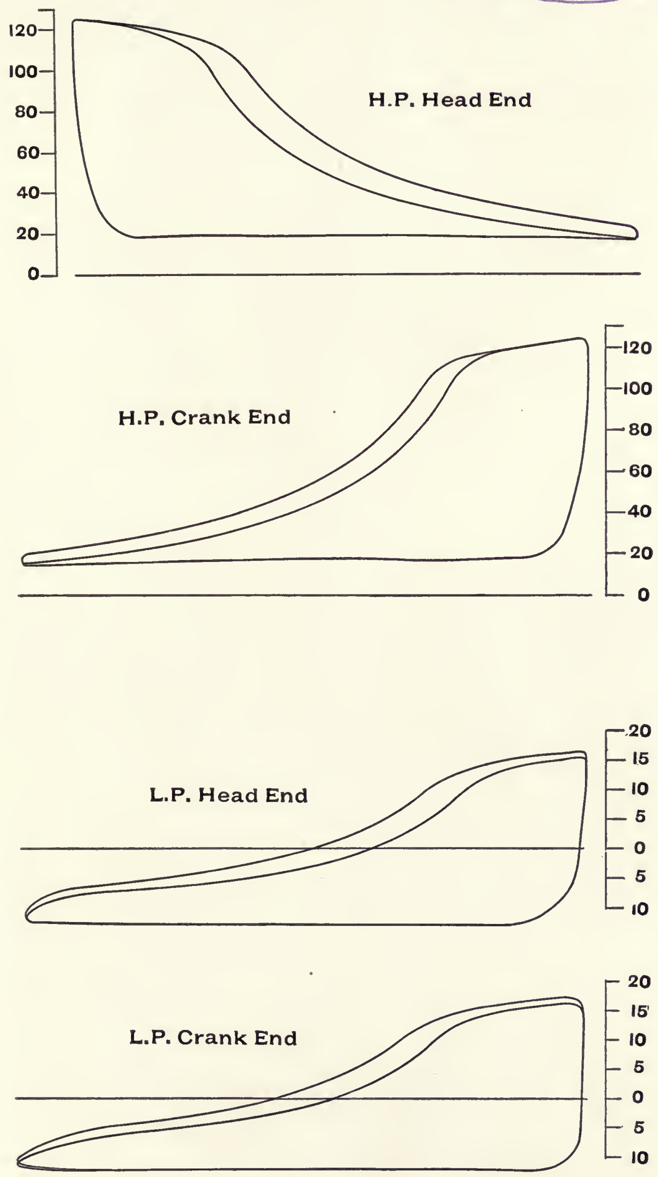
The weight of steam condensed in the reheater coil on test "A" was 500 lbs. per hour, or about .5 of a pound per I. H. P. per hour; and this is included in the quantities given in the tables.* In the diagrams which are appended for the variable load test, the two extreme lines are reproduced which were taken for a period covering ten consecutive revolutions. During the whole trial with variable load, the maximum variation of the load was shown by the extreme readings of the ammeter. The highest was 1,456 amperes and the lowest 624. The next highest readings were 1,300 amperes, and the next lowest 669.

* These figures were determined from ten sets of diagrams, the average horse-power developed by the whole engine being 1029.16. The average horse-power used for working up the results (1030.6) was determined from the average electrical readings, using the efficiency corresponding to the readings when the ten sets were obtained. On the variable load test the horse-power was determined from the electrical readings by using the average efficiency found by independent tests made with a steady load, this load being the average load of the main trial.

ENGINE No. 58a



ENGINE No.58b



TRIPLE EXPANSION ENGINES.

ENGINE No. 59.

Triple Expansion Engine.

		H. P. Cylinder.	Int. Cylinder.	L. P. Cylinder.
Kind of engine		Four valve (Corliss)		
Number of cylinders		1	1	2
Diameter of cylinders	ins.	20	34	36
Diameter of piston rod	ins.	$4\frac{5}{8}$	$4\frac{5}{8}$	one $\begin{cases} 4\frac{5}{8} \\ 6 \end{cases}$
Stroke of piston	ft.	5	5	5
Clearance	%	2.5	2.5	2.5
H. P. Constant for one lb. m.e.p., one rev. per minute	H.P.	.0928	.2727	.3017 each
Ratio of areas of cylinders		1	2.94	6.5 two
Condition of valves and pistons regarding leakage		Consid. leakage	Practic'ly tight	Practic'ly tight

Data and Results of Feed-Water Test.

Character of steam	Superheated 39°	
Duration	10.375	hrs.
Weight of feed-water consumed	131,461	lbs.
Feed-water consumed per hour	12,670.9	lbs.
Pressure in steam pipe above atmosphere	151	lbs.
Pressure in first receiver above atmosphere	33	lbs.
Pressure in second receiver above atmosphere	4	lbs.
Vacuum in condensers	27	ins.
Revolutions per minute	65.24	
Mean effective pressure, H. P. cylinder	59.59	lbs.
Mean effective pressure, intermediate cylinder	13.19	lbs.
Mean effective pressure, L. P. cylinder	10.19	lbs.
Indicated horse-power, H. P. cylinder	360.9	H.P.
Indicated horse-power, intermediate cylinder	234.7	H.P.
Indicated horse-power, L. P. cylinder	401.2	H.P.
Indicated horse-power, whole engine	996.8	H.P.
Feed-water consumed per I. H. P. per hour	12.71	lbs.

Measurements Based on Sample Diagrams.

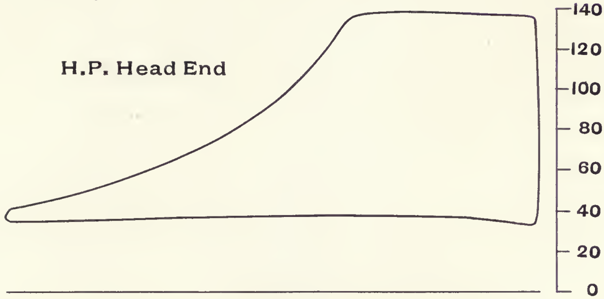
		H. P. Cylinder.	Int. Cylinder.	L. P. Cylinder.
Initial pressure above atmosphere	lbs.	142.	32.9	4.1
Corresp. steam-pipe pressure	lbs.	152.		
Cut-off pressure above zero	lbs.	145.2	38.7	16
Release pressure above zero	lbs.	53.	17.4	6.7
Mean effective pressure	lbs.	60.56	13.22	10.16
Back pressure at mid stroke above or below atmosphere	lbs.	+32.6	+4.8	−12.5
Proportion of stroke completed at cut-off		.346	.406	.357
Steam accounted for at cut-off	lbs.	9.81	9.53	8.39
Steam accounted for at release	lbs.	10.42	10.45	9.95
Proportion of feed-water accounted for at cut-off		.773	.741	.66
Proportion of feed-water accounted for at release		.82	.822	.783

Engine No. 59 is a horizontal four-cylinder engine, arranged in the manner of a pair of tandem compound engines. The cylinders nearest the beds are the low-pressure cylinders, of which there are two. The high-pressure cylinder is in front of one of the low-pressure cylinders, and the intermediate cylinder in front of the other. The cylinders are jacketed on the system which allows the steam which is supplied to the cylinder to first pass through the jacket space, each jacket thus being filled with steam having the initial pressure of supply. The jackets are drained into receivers, and these are provided with pumps operated by the engine. They discharge the water into reheaters placed in the flue of the boilers. The steam which is formed in the reheaters is supplied to the receiver between the intermediate and the low-pressure cylinders. This receiver is provided with a coil of live steam pipe presenting 33 square feet of exterior surface. The total quantity of water condensed in the jackets and withdrawn from them amounted to 691 lbs. per hour, or about 5 % of the total quantity of steam supplied to the engine. About one-half of this was re-evaporated in the reheater and utilized in the low-pressure cylinders. The condensers, of which there are two, are of the jet type, and operated by direct connected air-pumps. Steam is supplied from vertical tubular boilers, and on the test it was superheated 39° at a point near the boilers. With the exception of the high-pressure piston, which leaked quite badly, the valves and pistons were all in a practically tight condition.

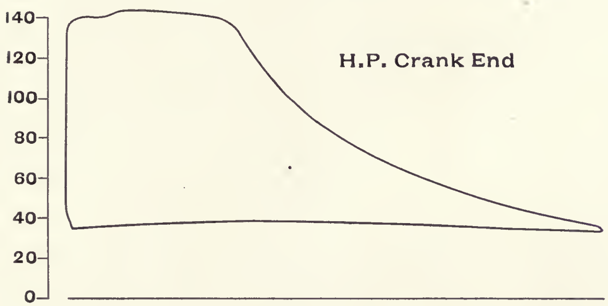
The load on the engine consisted of cotton machinery. The loss of steam which, referring to the analysis of the diagrams, took place between the intermediate cylinder and the low-pressure cylinders is noticeable in view of the arrangements made to reheat the steam in the receiver, and augment the supply by means of the jacket-water re-evaporated in the flue heaters. It shows the powerful action of cylinder condensation, and the necessity of employing more efficient means for overcoming the loss.

ENGINE No. 59

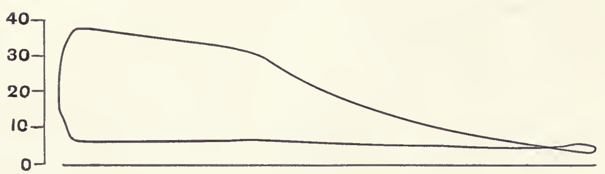
H.P. Head End



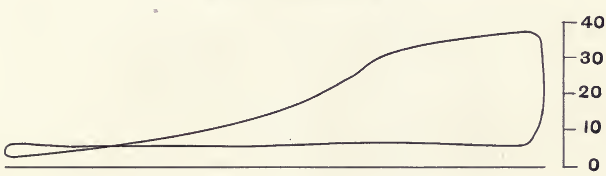
H.P. Crank End



Int^{m} Head End

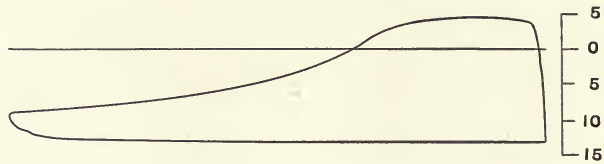


Int^{m} Crank End

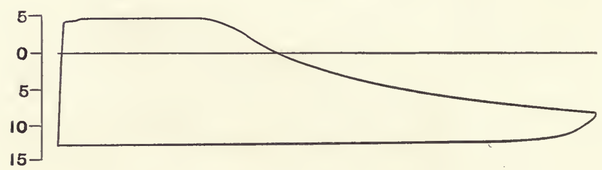


ENGINE No. 59

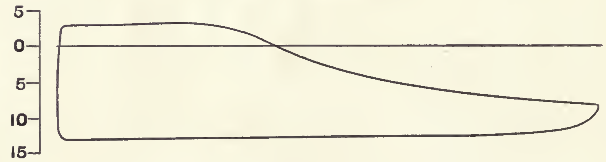
R.H.L.P. Head End



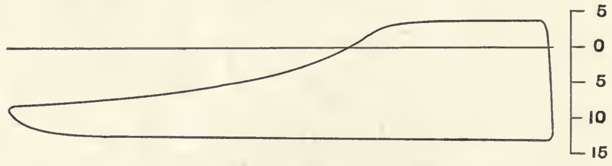
R.H.L.P. Crank End



L.H.L.P. Head End



L.H.L.P. Crank End



ENGINE No. 60.

Triple Expansion Engine.

		H. P. Cyl.	Int. Cyl.	L. P. Cyl.
Kind of engine		Four valve (Corliss)		
Number of cylinders		1	1	1
Diameter of cylinder	ins.	28	48	74
Diameter of piston rod	ins.	two 4	two 4	two 4
Stroke of piston	ft.	5	5	5
Clearance	%	1.4	1.5	.8
Ratio of areas of cylinders		1	2.98	7.11
Condition of valves and pistons regarding leakage		Fairly tight	Fairly tight	Fairly tight

Data and Results of Feed-Water Test.

Character of steam	Ordinary	
Duration	72.0	hrs.
Weight of feed-water consumed	518,811.0	lbs.
Feed-water consumed per hour	7,205.7	lbs.
Pressure in steam pipe above atmosphere	125.6	lbs.
Pressure in first receiver above atmosphere	30.3	lbs.
Pressure in second receiver above atmosphere	.8	lbs.
Vacuum in condenser	25.3	ins.
Revolutions per minute	20.99	rev.
Mean effective pressure, H. P. cylinder	49.85	lbs.
Mean effective pressure, intermediate cylinder	15.4	lbs.
Mean effective pressure, L. P. cylinder	7.57	lbs.
Indicated horse-power, H. P. cylinder	191.3	H.P.
Indicated horse-power, intermediate cylinder	176.04	H.P.
Indicated horse-power, L. P. cylinder	206.39	H.P.
Indicated horse-power, whole engine	573.73	H.P.
Feed-water consumed per I. H. P. per hour	12.55	lbs.

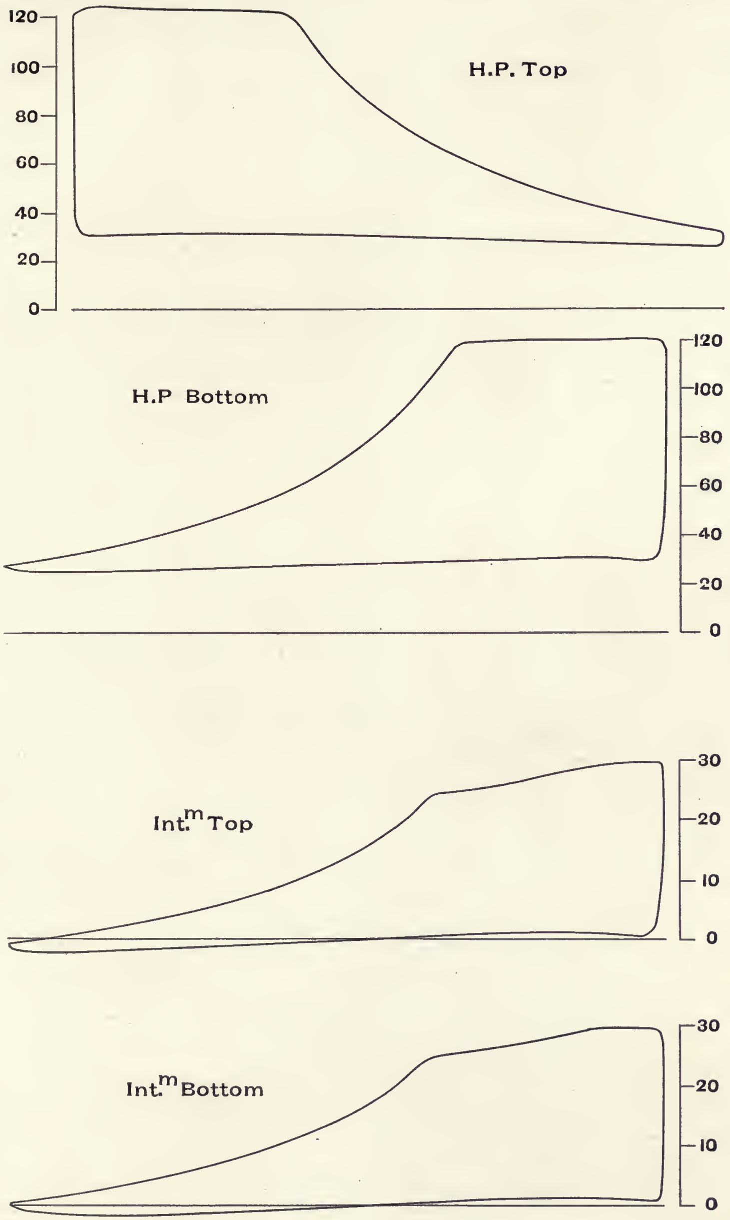
Measurements based on Sample Diagrams.

		H. P. Cyl.	Int. Cyl.	L. P. Cyl.
Initial pressure above atmosphere	lbs.	124.3	30.4	0.2
Corresponding steam-pipe and receiver pressure	lbs.	129.7	30.1	1.0
Cut-off pressure above zero	lbs.	134.1	30.3	11.5
Release pressure above zero	lbs.	44.7	14.3	5.8
Mean effective pressure	lbs.	50.07	15.41	7.59
Back pressure at mid stroke, above or below atmosphere	lbs.	+29.4	0.0	−11.9
Proportion of stroke completed at cut-off		.338	.362	.479
Steam accounted for at cut-off	lbs.	9.48	9.53	9.45
Steam accounted for at release	lbs.	9.96	9.97	9.91
Proportion of feed-water accounted for at cut-off		.756	.759	.753
Proportion of feed-water accounted for at release		.793	.794	.789

Engine No. 60 is a vertical triple expansion pumping-engine with jacketed cylinders and two reheaters. Only the barrels of the cylinders are jacketed, the heads being unjacketed, except so far as the valve chests, which are located in the heads, furnish a substitute. The jacket of the low-pressure cylinder is supplied with steam at a reduced pressure. The remaining jackets and the reheaters are supplied with boiler steam. The engine is furnished with steam from horizontal return tubular boilers, and at a point near the throttle valve the percentage of moisture was found to be .3 %. There was no undue leakage of the valves and pistons, but they were not in a perfectly tight condition. The load of each cylinder is that of a direct-acting pump, the diameter of each plunger being 36,″ and the total head, expressed in pounds, 53.4 lbs. per square inch. The jackets consumed 955 lbs. of steam per hour, which is 12.7 % of total used by the engine; and this is included in the quantities given in the tables. When the engine was at rest, the jackets consumed 163.5 lbs. per hour, being the loss due to radiation.

The analysis of the diagrams in this test shows a remarkably close agreement between the steam accounted for by the indicator in the various cylinders. There is a variation of less than 1 % between the quantities shown at the cut-off in the three cylinders, and a similarly close agreement in the three quantities shown at the release.

ENGINE No. 60


H.P. Top
120
100
80
60
40
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H.P Bottom
120
100
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Int.m Top
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10
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Int.m Bottom
30
20
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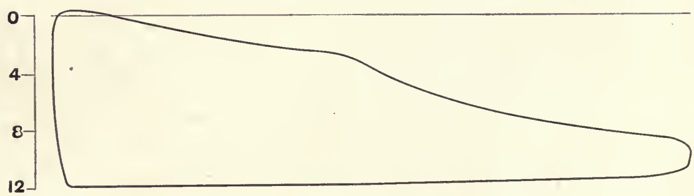
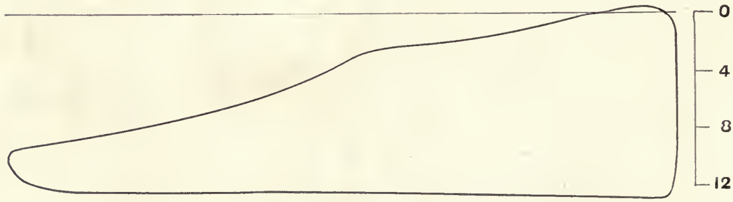
0

ENGINE No. 60

L.P. Top



L.P. Bottom



Summary of Feed-Water Tests.

SIMPLE ENGINES.

Number.	Kind of Engine.	Condensing or Non-Condensing.	Quality of Steam.	Leakage Conditions.	Pressure.	Rev. per Min.	Indicated Horse-Power.	Cut-off.	Proportion of Feed-Water accounted for at Cut-off.	Feed-Water per I. H. P. per Hour.
1 a	Four-valve(c)	Non-Cond.	Ordinary	Fairly tight	72.3	74.7	305.2	.367	.84	27.77
1 b	" " "	" "	"	" "	81.3	74.6	320.5	.375	.839	29.34
1 c	" " "	" "	Superh'd 82°	" "	79.6	74.1	324.	.392	.947	26.83
2	" " "	" "	Ordinary	" "	101.	64.8	506.5	.315	.817	25.8
3 a	" " "	Condensing	"	" "	67.2	60.3	210.5	.138	.654	21.11
3 b	" " "	¾ Cond.	"	" "	69.1	60.3	229.	.16	.695	22.68
4	" " "	½ "	Superh'd 25°	" "	83.	53.3	523.4	.233	.766	22.19
5	" " "	½ "	Ordinary	Some leakage	71.1	47.3	687.4	.303	.784	26.28
6	" " "	¼ "	"	" "	84.4	51.1	610.7	.252	.757	22.95
7	" " "	Non-Cond.	"	" "	80.5	64.7	232.3	.237	.75	29.03
8 a	" " "	Condensing	Superh'd 37°	Fairly tight	53.1	54.1	366.4	.247	.819	19.39
8 b	" " "	"	" 37°	" "	68.2	54.1	361.8	.165	.747	18.71
9 a	" " "	"	" 24°	Some leakage	70.8	46.	758.3	.185	.82	18.25
9 b	" " "	¾ Cond.	" 24°	" "	73.4	46.	758.1	.225	.836	20.4
10 a	Double valve	Condensing	" 16°	" "	79.	154.7	336.2	.262	.771	20.51
10 b	" "	Non-Cond.	" 41°	" "	75.9	152.9	310.1	.337	.811	25.64
11	Four-valve	" "	Ordinary	Considerable l'k'ge	61.	79.8	50.2	.234	.575	37.43
12	Four-valve(c)	Condensing	"	" "	70.2	70.2	258.2	.29	.719	23.62
13	Single valve	Non-Cond.	"	" "	102.5	246.	53.3	.119	.539	32.67
14	" "	" "	"	" "	105.3	315.	27.3	.194	.52	34.44
15	Four-valve(c)	½ Cond.	Superh'd 59°	Fairly tight	77.6	61.	615.1	.281	.895	21.44
16 a	Single valve	Non-Cond.	Ordinary	Some leakage	91.7	352.2	44.8	.353	.703	32.6
16 b	" "	" "	"	" "	92.5	353.9	35.1	.278	.652	32.99
16 c	" "	" "	"	" "	92.1	356.7	25.7	.206	.549	36.27
17 a	Four-valve	Condensing	"	Fairly tight	67.	165.3	213.2	.264	.77	22.68

SIMPLE ENGINES (CONTINUED).

Number.	Kind of Engine.	Condensing or Non-Condensing.	Quality of Steam.	Leakage Conditions.	Pressure.	Rev. per Min.	Indicated Horse-Power.	Cut-off.	Proportion of Feed-Water accounted for at Cut-off.	Feed-Water per I. H. P. per Hour.
17 b	Four-valve	Non-Cond.	Ordinary	Fairly tight	67.6	164.4	209.1	.385	.82	28.93
18 a	Four-valve(c)	Condensing	"	" "	84.5	61.2	204.	.188	.725	20.31
18 b	" " "	"	"	" "	59.	61.8	208.4	.111	.668	20.56
19	Single valve	"	"	Some leakage	74.5	129.3	204.6	.303	.706	27.15
20 a	Four-valve	"	"	" "	68.1	52.	444.	.119	.615	23.
20 b	" "	Non-Cond.	"	" "	65.1	50.5	451.6	.222	.722	30.16
21	" "	" "	Superh'd 4°	Considerable l'k'ge	64.5	198.3	32 3	.238	.62	40.04
22	" "	Condensing	Ordinary	Some leakage	82.3	59.9	613.4	.172	.669	18.49
23 a	Single "	Non-Cond.	"	Fairly tight	83.	303.7	22.4	.203	.642	31.78
23 b	" "	" "	"	" "	82.4	307.8	32.3	.312	.72	31.2
23 c	" "	" "	"	" "	81.9	304.5	39.4	.378	.762	30.63
24	" "	" "	"	" "	80.3	248.4	61.7	.211	.662	32.71
25	Four-valve(c)	$\frac{4}{4}$ Cond.	"	Some leakage	82.9	53.9	625.6	.243	.737	22.29
26 a	Four-valve	Non-Cond.	"	Considerable l'k'ge	74.2	75.8	70.9	.237	.692	32.61
26 b	" "	" "	"	Fairly tight	74.	76.3	70.6	.223	.745	29.37
27	Single valve	" "	"	Considerable l'k'ge	72.2	172.5	38.1	.219	.568	37.21
28	Four-valve(c)	Condensing	"	Some leakage	70.1	59.1	554.4	.271	.781	19.45
29	" " "	½ Cond.	"	" "	67.1	60.3	562.1	.194	.679	25.43
30	Four-valve	Condensing	"	" "	83.4	90.4	205.9	.139	.645	21.42
31 a	Four-valve(c)	Non-Cond.	"	Fairly tight	101.8	89.6	37.	.041	.382	73.63
31 b	" " "	" "	"	" "	98.6	84.9	342.4	.323	.796	25.91
31 c	" " "	" "	"	" "	. . .	. . .	100.4	.084	.509	38.28
31 d	" " "	" "	"	" "	. . .	. . .	146.2	.121	.588	31.43
31 e	" " "	" "	"	" "	. . .	. . .	222.2	.178	.709	25.83
31 f	" " "	" "	"	" "	. . .	. . .	287.1	.231	.745	25.39

COMPOUND ENGINES.

Number.	Kind of Engine.	Condensing or Non-Condensing.	Quality of Steam.	Leakage Conditions.	Pressure.	Rev. per Min.	Indicated Horse-Power.	Cut-off. H.P.Cyl.	Proportion of Feed-Water accounted for at Cut-off. H. P. Cyl.	Feed-Water per I. H. P. per Hour.
32	Four-valve(c)	Condensing	Ordinary	Fairly tight	94.8	52.3	606.5	.305	.774	16.28
33 a	Single valve	"	"	Tight	129.3	300.	228.2	.38	.804	18.92
33 b	" "	Non-Cond.	"	"	128.	296.1	230.5	.532	.894	22.53
34	Four-valve(c)	Condensing	"	Some leakage	116.1	68.1	636.5	.26	.789	13.28
35	Single valve	"	"	Considerable l'k'ge	105.2	197.1	148.5	.382	.623	22.05
36	Four-valve(c)	"	"	Practically tight	126.8	74.9	382.5	.295	.767	14.05
37	" " "	"	"	Fairly tight	108.	59.	280.1	. . .	.732	13.37
38	" " "	"	"	Excessive leakage	108.9	62.1	716.	.377	.675	19.36
39	Single valve	"	"	Considerable l'k'ge	120.6	195.3	220.6	.396	.629	18.01
40	" "	"	"	Practically tight	126.	228.	347.6	.595	.911	18.2
41 a	" "	"	"	Considerable l'k'ge	129.7	306.	90.5	. . .	. . .	22.74
41 b	" "	"	"	" "	130.1	298.5	196.8	.382	.629	19.1
42 a	" "	Non-Cond.	"	" "	126.5	300.2	45.6	.107	.228	44.89
42 b	" "	" "	"	" "	128.	292.7	152.5	.389	.617	25.2
43	Four-valve(c)	Condensing	Sup'h'd 44.5°	Fairly tight	119.8	70.	1017.1	.236	.762	13.26
44	Single valve	Non-Cond.	Ordinary	Practically tight	135.9	296.3	109.7	.605	.803	22.91
45 a	Double valve	Condensing	"	Considerable l'k'ge	119.9	161.7	295.7	.336	.751	16.07
45 b	" "	"	"	" "	120.4	162.7	244.5	. . .	. . .	15.71
45 c	" "	"	"	" "	117.6	170.1	123.4	.044	.361	17.22
45 d	" "	"	"	" "	118.9	164.8	276.9	.3	.694	16.07
45 e	" "	Non-Cond.	"	" "	118.	165.7	267.1	.487	.794	23.24
46 a	Four-valve	" "	"	Practically tight	128.7	101.	346.9	.309	.806	22.11
46 b	" "	" "	"	" "	135.5	99.	486 7	.419	.816	21.59
47 a	" "	Condensing	"	Considerable l'k'ge	150.7	60.3	689 3	.281	.788	12.69
47 b	" "	"	"	" "	151.1	60.5	708 3	.263	.74	12.45
47 c	" "	"	"	" "	150.5	60.6	681.4	. . .	. . .	12.61
47 d	" "	"	"	" "	151.4	60.3	684.6	. . .	. . .	12.72

COMPOUND ENGINES (CONTINUED).

Number.	Kind of Engine.	Condensing or Non-Condensing.	Quality of Steam.	Leakage Conditions.	Pressure.	Rev. per Min.	Indicated Horse-Power.	Cut-off. H.P.Cyl.	Proportion of Feed-Water accounted for at Cut-off. H. P. Cyl.	Feed-Water per I. H. P. per Hour.
48 a	Four-valve	Condensing	Superh'd 12°	Fairly tight	125.9	77.	1106.7	.294	.792	14.12
48 b	" "	"	" 13°	" "	121.5	76.7	1073.2	.31	.856	14.01
48 c	" "	"	" 20°	" "	100.2	76.7	1098.6	.432	.823	15.08
49	Four-valve(c)	"	Ordinary	Some leakage	115.4	71.3	872.9	.33	.866	14.18
50	" " "	"	Superh'd 30°	" "	108.1	61.4	798.4	.274	.889	13.28
51 a	Four-valve	"	" 15.7°	" "	149.7	80.	670.5	.285	.77	12.29
51 b	" "	"	" 16.4°	" "	150.4	80.1	658.1	. . .	. . .	12.03
51 c	" "	"	" 12 2°	" "	150.2	80.	659.1	.285	.769	11.89
52 a	" "	"	Ordinary	" "	144.2	77.4	718 9	. . .	. . .	13.09
52 b	" "	"	"	" "	144.1	76.6	741.3	. . .	. . .	13.23
52 c	" "	"	"	" "	143.8	76.9	738.8	.268	.651	13.01
52 d	" "	"	"	" "	144.1	78.9	725.2	.293	.734	13.27
53	Four-valve(c)	"	"	Fairly tight	114.9	65.5	299.7	.238	.717	15.78
54 a	Single valve	Non-Cond.	"	Some leakage	166.9	275.7	103.4	. . .	. . .	24.99
54 b	" "	" "	"	" "	166.8	271.2	187.5	.479	.744	21.14
54 c	" "	" "	"	" "	164.6	273.4	242.9	. . .	. . .	21.25
55	Four-valve(c)	Condensing	"	Practically tight	151.4	75.2	1713.8	.326	.817	13.27
56	" " "	"	"	Some leakage	107.8	120.2	676.2	.331	.813	14.6
57	" " "	"	"	" "	133.	66.	1539.9	.294	.696	14.1
58 a	" " "	"	"	Practically tight	136.2	78.	1030.1	.293	.8	13.21
58 b	" " "	"	"	" "	128.9	78.	843.4	. . .	. . .	13.53
				TRIPLE EXPANSION ENGINES.						
59	Four-valve(c)	Condensing	Superh'd 39°	Considerable l'k'ge	151.	65.2	996.8	.346	.773	12.71
60	" " "	"	Ordinary	Fairly tight	125.6	21.	573.7	.338	.756	12.55

REVIEW OF FEED-WATER TESTS.

REVIEW OF FEED-WATER TESTS.

It could hardly be expected that conclusive information upon the subjects which are of most interest in connection with the operation of steam engines could be obtained from a large number of tests made on engines of various sizes, running under different conditions of service, and located in plants which are not always best adapted for experimental purposes or research, like the tests under consideration. Such tests, however, cannot but bring out some points on these subjects which are of considerable practical value, if for no other reason than that the tests were, in the main, conducted under those very circumstances of practical operation which alone could give results of that nature.

The tests furnish information in regard to cylinder condensation, leakage of valves and pistons, the effect of pressure and speed, the economy of condensing and superheating, the relative economy of simple, compound, and triple expansion engines, the effect of steam jacketting and reheating, the effect of different ratios of cylinder areas in compound engines, and some miscellaneous questions; and these are discussed in the order named.

I. CYLINDER CONDENSATION AND LEAKAGE.

Cylinder condensation and leakage is that part of the feed-water consumption which is not accounted for by the indicator diagram. It is necessary to put these two losses in one class. There is no way of separating them either absolutely or approximately. The only practicable thing to do is to test the valves and pistons for leakage with the engine at rest; and if under these conditions they prove to be tight, it is fair to assume that the leakage under conditions of running is practically nothing, and the part not accounted for by the diagram is wholly or substantially cylinder condensation. If a similar engine, working under similar conditions, is found by such tests to leak, and then a comparison is made between the loss in the tight engine and that in the leaking engine, an inference can be drawn as to the extent of the leakage and how much the loss amounts to in percentage of the whole consumption. Practically, it may be said that it is unnecessary to know the absolute amount of cylinder condensation, for it is seldom that an engine is found in an absolutely tight condition; and, after all, the important thing to know is what the cylinder condensation and leakage amount to when the engine is in ordinarily good working condition. These tests furnish satisfactory evidence on this point, especially those made on simple engines. Selection may properly be made from the list of simple engines, those of the larger sizes of the four-valve type, using ordinary steam, taking those which are tight or leaking only a small amount; namely, those numbered 1, 2, 3, 5, 6, 7, 17, 18, 20, 22, 25, 28, 29, 30, and 31. These are tabulated as follows, being arranged in the order of the point of cut-off: —

No. of Engine.	Cut-off.	Proportion of Feed-Water Accounted for at Cut-off.	Cylinder Condensation and Leakage.
31 A	.041	.382	.618
31 C	.084	.509	.491
18 B	.111	.668	.332
31 D	.121	.588	.412
20 A	.119	.615	.385
3 A	.138	.654	.346
30	.139	.645	.355
3 B	.160	.695	.305
22	.172	.669	.331
31 C	.178	.709	.291
18 A	.188	.725	.275
29	.194	679	.321
20 B	.222	.722	.278
31 F	.231	.745	.255
7	.237	.750	.250
25	.243	.737	.263
6	.252	.757	.243
17 A	.264	.770	.230
28	.271	.781	.219
5	.303	.784	.216
2	.315	.817	.183
31 B	.323	.796	.204
1 A	.367	.840	.160
1 B	.375	.839	.161
17 B	.385	.820	.180

These proportions of cylinder condensation and leakage can better be examined by referring to the accompanying chart on which they are plotted, ordinates or verticals representing percentages of cut-off, and abscissae or horizontals, percentages of condensation and leakage. The curved line drawn through them represents the mean curve of condensation and leakage deduced from these results. It clearly shows that the percentage rapidly increases as the point of cut-off becomes earlier, and at the very earliest cut-off the condensation and leakage bears a very large proportion to the total consumption of steam.

The best series of tests on this subject, using the same engine, are those made on Engine No. 31, which was a pair of Corliss non-condensing engines having cylinders 16 in. diameter and 42 in. stroke. This engine was practically tight, and the percentages of condensation (and leakage) over a range of cut-off from 4% to 32% was from 62% down to 20%.

That leakage has an important effect upon the economy of an engine is well shown by a comparison of the results obtained from engines which leaked excessively with those shown on the chart. For example, in Engine No. 19, which is of the single-valve type showing considerable leakage compared with Corliss engines, the proportion of steam accounted for is .706 at 30% cut-off, giving condensation and leakage of 29.4%. The condensation and leakage shown on the average curve of the chart at that cut-off is about 20%, so that the difference between 20 and 29.4 must be due in a large degree to leakage. In Engine No. 26 a test with a leaking exhaust valve showed condensation and leakage of 30.8%. When the valve had been repaired and made tight, the condensation and leakage dropped down to 25.5%, the difference being due almost entirely to a reduction in leakage. The saving in actual feed-water consumption was about 10%.

In the compound engine tests which can be compared for the purpose of studying cylinder condensation, the range of cut-off in the high-pressure cylinder is hardly sufficient to serve as a basis for satisfactory conclusions. The tests which can be

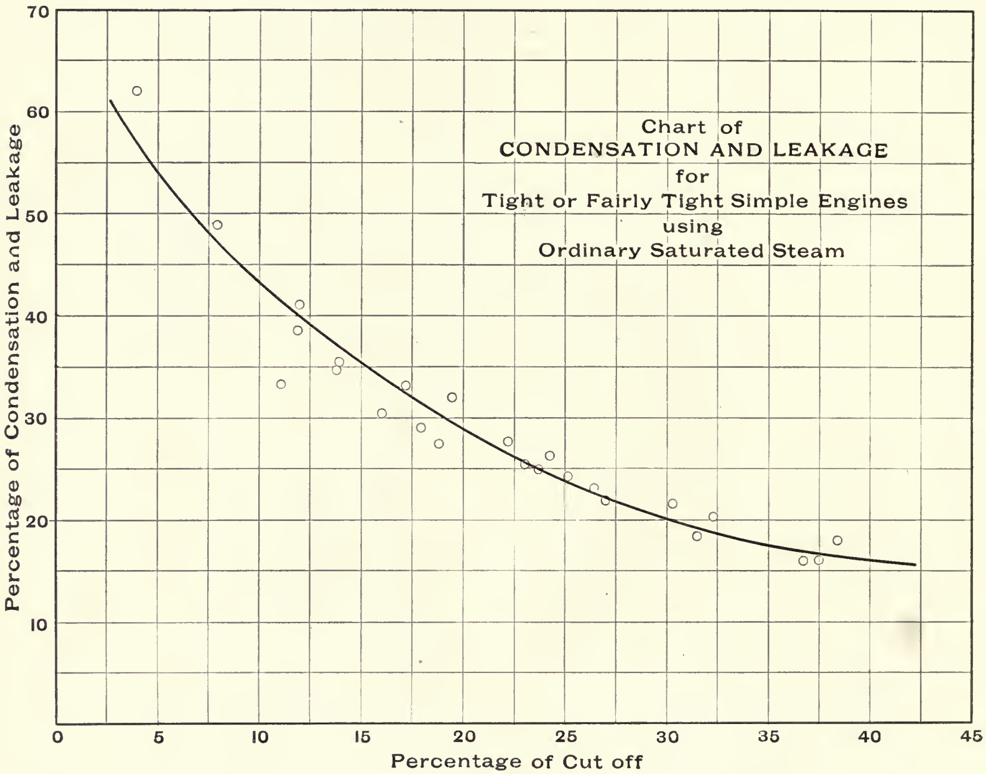


Chart of
CONDENSATION AND LEAKAGE
for
Tight or Fairly Tight Simple Engines
using
Ordinary Saturated Steam
Percentage of Condensation and Leakage
70
60
50
40
30
20
10
0
5
10
15
20
25
30
35
40
45
Percentage of Cut off

fairly compared for this purpose are those numbered 32, 34, 36, 49, 53, 55, 56, and 58A. The range of cut-off in the high-pressure cylinder is from .238 to .331, and the range of steam accounted for at cut-off in the same cylinder is from .717 to .866. These are tabulated below:

No. of Engine.	Cut-off, H. P. Cyl.	Proportion of Feed-Water Accounted for at Cut-off, H. P. Cyl.
32	.305	.774
36	.295	.767
49	.330	.866
53	.238	.717
55	.326	.817
56	.331	.813
58 A	.293	.800
Average,	.303	.793

The average cylinder condensation and leakage in these cases is 100–79.3 = 20.7%, and the average cut-off in the high-pressure cylinder is .303. If these be compared with the curve of condensation and leakage given on the chart for the simple engines, it will be seen that this average falls closely upon that curve. The point on the curve for 30% cut-off is 20.5%. The engines selected are those in which the high-pressure cylinder is unjacketed, or of the class in which the jacket space forms a thoroughfare through which steam is supplied to the chest. It may be inferred from the close agreement between the average of these tests and the indications of simple engines in the matter, that the curve of condensation and leakage for simple engines applies also to the high-pressure cylinder of compound engines where these are unjacketed, and where the valves and pistons are fairly tight.

Condensation and leakage in the low-pressure cylinder of the compound engines reported above is affected to a considerable extent by the conditions regarding jacketing and reheating, Where there is neither jacketing nor reheating, the condensation and leakage is much greater in the low-pressure cylinder than in the H. P. cylinder. For example, reference may be made to Engines No. 32, 36, 46A, 47A, 48B, 52D, and 53, in which the proportions of cylinder condensation and leakage are as follows:

No. of Engine.	Condensation and Leakage Cut-off H. P. Cyl.	Condensation and Leakage Cut-off, L. P. Cyl.
32	.226	.28
36	.233	.36
46 A	.194	.29
47 A	.212	.32
48 B	.144	.26
52 D	.266	.35
53	.283	.40
Average,	.222	.323

The average of these shows 10% more condensation and leakage in the L. P. cylinder than in the H. P. cylinder.

That leakage in itself produces an important effect in compound engines is exhibited in the case of one engine reported, that of No. 38, where the feed-water consumption per I. H. P. per hour is 19.36 lbs. The leakage here was in the low-pressure piston; and although the cylinders of the engine were jacketed, and the receiver also jacketed, the condensation and leakage at cut-off in the H. P. cylinder was 32.5% against 17.5% by the simple-engine curve, and 57.0% at cut-off in the L. P. cylinder; the increase between the two cylinders being 24.5%.

If we may judge from indications of one test, that of Engine No. 26, the effect of leakage upon the consumption of steam and economy of the engine may be exceedingly marked, and at the same time have so little influence upon the lines of the diagram that it may be scarcely noticeable. In this instance, the form and position of the expansion line with reference to a hyperbolic curve drawn through the cut-off point of the diagram is so nearly the same that careful measurements would hardly distinguish a difference, although in one case the exhaust valve leaked no less than 10%. Practically the same effect, or rather want of effect, has been noticed in one other case where a broken packing-ring in a piston caused a leakage which amounted to a still larger quantity. A close examination of the expansion line of the diagrams, before and after, failed to reveal any clearly defined difference. This is Engine No. 78. In a case like that of the compound Engine No. 38, it cannot be inferred from this that the influence of excessive leakage could not be revealed by a study of the indicator diagrams. Here it did produce a marked effect in the distribution of the load between the cylinders, cutting down the power developed in the L. P. cylinder, and increasing it to a corresponding extent in the H. P. cylinder. At the same time, it caused a noticeable "drop" in pressure at the high-pressure release.

In studying the effect of cylinder condensation and leakage, and the extent of the loss which it produces, the quantity shown at the cut-off point of the diagram is selected in preference to that shown at the release, in the belief that at the cut-off point the full extent of the loss is the more truthfully indicated. If the steam accounted for at both points is identical, the loss is the same at one point as at the other, and it does not matter which point is selected. If the quantity accounted for is larger at the release than it is at the cut-off, which owing to re-evaporation during expansion frequently occurs, the apparent loss due to condensation and leakage is less at the release than it is at the cut-off. Sometimes there is as much as 10 per cent less apparent condensation and leakage at the release than at the cut-off. In cases like this the release percentage does not show the full extent of loss, because the work recovered on account of re-evaporation is in no sense proportional to the increase in the steam accounted for at that point. Neither does the loss at cut-off in such cases represent the true loss, and the reason is the same; but the loss at cut-off furnishes a closer indication of the true loss than the loss at release, and a better basis for the study of the question of cylinder condensation.

It will be noticed in the table giving the quantities upon which the chart of cylinder condensation and leakage is based, that no distinction is made between engines which are condensing and those running non-condensing. It is probable that the transfer of heat from the steam to the metal of the cylinder, under the action of the comparatively low temperature of the condenser, is different from that which occurs in the non-condensing engine; and if a suitable investigation were made, this difference would appear in the percentage of cylinder condensation. Whatever this difference may be, it is not sufficiently marked to be noticeable in the tests referred to in the chart; and consequently the results are used indiscriminately, whether the engines are condensing or non-condensing.

II. EFFECT OF PRESSURE ON THE ECONOMY.

Other things being the same, it is a well recognized principle in steam-engineering that the higher the pressure the more economical the consumption of steam. But circumstances attending its use are not always the same; and consequently, in examining the results obtained from different engines, such as those here reported, it does not follow that in any individual case where the pressure is highest the economy is necessarily the greatest. For example, two tests are reported on Engine No. 18, which show practically the same economy as measured by the feed-water consumed per I. H. P. per hour; yet the pressure in one case is 84 lbs., and in the other case 59 lbs. It is evident that the difference in the cut-off which accompanied the change of pressure exerted such an influence that the benefit which might have been derived from a higher pressure was counterbalanced.

There are two instances among the simple engines which may be examined to show the importance due to increased pressure. Test No. 1 A and test No. 2 is one of these instances. Here an increase of the pressure from 72.3 lbs. to 101 lbs., accompanied by a slight shortening of the cut-off, had a marked effect in improving the economy, the consumption of feed-water being reduced from 27.8 lbs. to 25.8 lbs. Another comparison may be made between test No. 7 and test No. 31 F. Here, with the same cut-off, an increase of pressure from 80.5 lbs. to 98.6 lbs. was evidently the principal cause of a reduction in the consumption from 29.03 lbs. per I. H. P. per hour to 25.31 lbs.

In the list of the compound engines, there are two tests which can be compared for this purpose, — those numbered 32 and 36. In No. 32, with a pressure of 94.8 lbs., and the cut-off in the high pressure cylinder .305, the feed-water consumption is 16.28 lbs. per I. H. P. per hour. In test No. 36, with 126.8 lbs. pressure, and the cut-off at .295, or practically the same as in the other case, the consumption is reduced to 14.05 lbs.

That an increased pressure in the same engine is advantageous under some circumstances, is clearly shown by test 48 A and 48 C. In the latter the pressure was 100.2 lbs., and the consumption of feed-water 15.08 lbs., while in the former the pressure was 125,9 lbs., and with practically the same load, the consumption was 14.12 lbs. In this case the benefit due to the increase of pressure was largely enhanced by the increased expansion obtained, the cut-off in the H. P. cylinder dropping from .432 to .294.

III. EFFECT OF SPEED UPON ECONOMY.

The speeds, expressed in revolutions per minute, vary in these tests from a minimum of 21 to a maximum of 356.7. With such a wide range, there is reason for expecting information as to the economy to be derived from increasing the rotative speed, but the tests furnish no conclusive evidence on this subject. The high-speed engines are all, or nearly all, of a different class from the low-speed ones, and the nature of the design and construction is such that certain features which are necessary for the highest economy are sacrificed in order to obtain the desired increase of speed. Many of the high-speed engines have a single valve which performs all the functions of the four valves in a slow-speed engine. The result is that these functions are not so perfectly performed in the engines which run at high speed, and there is a loss of economy. Furthermore, the valves in the high-speed engines are generally of some balanced type; and valves of this kind are not so well adapted to tight construction, and do not maintain so tight a condition as those of the four valves in slow-speed engines. Again, the high-speed engines usually require larger clearance spaces in the cylinders than those of the slow-speed class. For these various reasons, the high-speed engine is handicapped at the outset with conditions which are unfavorable to economy; and if the effect of the high speed is advantageous, the advantage must be so great as to overcome the losses noted if it is to show in favor of the high-speed engine when it is subjected to a test. An examination of the tests reported, taking those engines which are run at the highest speeds, shows that in every case they are less economical than the slow-speed engines, and in every case the reason appears in one or more of the points mentioned. There is a further reason for the comparatively low economy of the high-speed engines reported, in the fact that in almost all cases the engines given are of comparatively small size; and this, no doubt, has an important influence in making the engine less economical than it would otherwise be.

There is one case given where a Corliss engine was run at a speed of one hundred and twenty revolutions per minute, — that of Engine No. 56; and this may be compared with the other Corliss engines running a lower speed. The comparison, however, is not a very satisfactory one; because the valves and pistons were not in the best condition in regard to leakage, and the boiler pressure was rather lower than that obtained on other engines with which it could be compared. Looking at the proportion of steam accounted for by the indicator, which is .813, there is nothing in this indication to show any marked improvement due to the high rotative speed, if such existed.

IV. ECONOMY OF CONDENSING.

It is held, in the popular mind, that the economy of condensing is, in round numbers, 25%. This percentage usually relates to simple engines, and it refers to the economy as measured by the difference in the coal consumption produced by a condenser. The evidence of some of the tests here given shows that this belief is not well founded, unless it be in special cases. The economy due to condensing ought to be reckoned on the basis of coal consumption, and not alone on the basis of feed-water consumption; because a non-condensing engine is usually accompanied by a feed-water heater, and some of the loss of economy produced by running non-condensing is made up by the saving of coal due to warming the feed-water. If the feed-water is heated by the exhaust steam of the non-condensing engine from a temperature of 100°, which is that of the ordinary hot well, to a temperature of 210°, the non-condensing engine can be credited with about 11% less coal consumption. This matter should properly be taken into account when considering the economy produced by a condenser.

In the list of simple engines, a number of comparisons are made on the same engine when carrying the same load, one test being made with the engine condensing and the other non-condensing. In the case of Engine No. 10, where such a comparison was made, the feed-water consumed when running non-condensing was 25.64 lbs. per I. H. P. per hour, and when running condensing, 20.51 lbs., the difference being 5.13 lbs., or 20% of the larger quantity. In Engine No. 17, which was tried in the same manner, the consumption running non-condensing was 28.93 lbs., and condensing, 22.08 lbs., the difference being 6.85, or 24% of the larger quantity. In Engine No. 20, a similar test was made; and the consumption in one case was 30.16 lbs., and in the other 23 lbs., the difference being 7.16 lbs., or 24% of the larger quantity. The average of these three comparisons gives a saving produced by condensing of 22.3%. If we allow for the steam or power used by an economical condenser, it will be seen that the net economy of condensing is, at best, not much over 20%; and this is on the basis of steam consumption. If, furthermore, we allow for the difference produced by heating the feed-water to the extent mentioned above, the saving of fuel would be reduced to about 11%. In some cases in practice where these conditions exist, the difference in favor of condensing might be greater, owing to the evaporative economy of the boilers being improved by reducing the work upon them; but all that could be fairly expected on the basis of these three engines, other things being the same, would be not much over 10%.

There is another method of looking at this subject; and that is, to compare the best performance of engines running condensing with the best results running non-condensing. The best non-condensing result, in the list of simple engines using ordinary steam, is that of Engine No. 31 F, working at about 100 lbs. pressure at .231 cut-off, and developing 287.1 indicated horse-power. This result is 25.39 lbs. The best result obtained from a condensing engine, using ordinary steam, is that of No. 22, working at a pressure of 82.3 lbs. at .172 cut-off, the engine developing 613.4 I. H. P. This is 18.49 lbs. per I. H. P. per hour. Comparing these two figures, there is a difference in favor of the condensing engine of 6.9 lbs., or 27.2% of the larger quantity. Allowing for steam or power used by the condensing apparatus, the net economy in feed-water consumption is not, at best, over 25%; and further allowance for the gain due to heating the feed-water, as estimated in the former case, would bring the coal-saving down to about 17%. It appears, therefore, that the tests here given on simple engines do not confirm the popular impression that the saving produced by condensing is 25%.

The economy of condensing, as compared with non-condensing, depends to some extent on the type of air-pump and condenser employed. There are four principal classes of these: —

1. The jet condenser and direct-connected air-pump, which uses power supplied by the main engine.

2. The siphon type of condenser, in which the water is supplied by gravity, and no air-pump is required.

3. The siphon condenser, in which the water is supplied by an independent pump.

4. The jet condenser, with air-pump driven by an independent engine or other motor.

In all of these, with the exception of the second, the expenditure of power or the consumption of steam must be charged against the saving due to condensing. Those that use steam can be arranged to utilize a portion of that steam in cases where the exhaust from the independent engine or pump is carried through a feed-water heater, and the heat returned to the boiler. In test No. 15, which was provided with a jet condenser and direct-connected air-pump, the amount of power used by the air-pump was found to be 1.8% of the working power of the engine. In Engine No. 57, which was provided with a siphon condenser supplied with water by an independent pump, the quantity of steam used by the pump, when exhausting into the condenser, was 6.7% of the total consumption of steam by the engine. In Engine No. 19, which was provided with a jet condenser operated by an independent steam-driven air-pump, the consumption of steam by the pump when exhausting into the air was 13% of the total quantity used by the engine. In Engine No. 20, which was fitted with a similar condenser, the quantity of steam used by the air-pump when exhausting into the condenser was over 13% of the total quantity. When an independent steam-driven air-pump is used, and the heat of the exhaust steam is returned to the boilers so far as possible by heating the feed-water, it is probable that from one-half to two-thirds of the steam is saved; and in a case where the air-pump uses 12% of the entire quantity, the actual loss of coal due to the air-pump would be not over 4 or 5%. From these considerations it appears that in cases where an air-pump or other condenser pump is required, the percentage to be charged to the condenser on this account is, in the best instances, about 2%; and in cases where the exhaust steam from the motor is not properly utilized, it may be so great as to largely offset the economy otherwise resulting from the use of the condenser.

The tests furnish some data as to the economy produced by a condenser in compound engines. In Engine No. 33, with practically the same load, the use of the condenser reduced the consumption of steam per I. H. P. per hour from 22.53 lbs. to 18.92 lbs., or 16%. In Engine No. 45, the use of the condenser, with a nearly constant load, reduced the consumption from 23.24 lbs. to 16.07 lbs., or 31%. A comparison may be made between Engines 41 and 42. The latter (42 B), running non-condensing, used 25.2 lbs. per I. H. P. per hour; and the former (41 B), running condensing, used 19.1 lbs. The reduction due to condensing here is 24%. Engine No. 46, which is run non-condensing, may be compared with Engine No. 48, which is run condensing, making allowance for the difference in the condition of the steam. In this instance the condenser appears to have reduced the feed-water consumption about 30%. In all these no account is taken of the steam used by the condensing apparatus, the percentages given being the gross savings. Throwing out Engine No. 33, which may be regarded as of special design, and possibly not useful for general comparison, it appears that the effect of the condenser on the compound engines is considerably greater than in the case of the simple engines. It will be seen, however, that the advantage of the condenser in compound engines depends largely upon the boiler pressure; and a comparison made on an engine like No. 41, which is running at 130 lbs., would show very differently from what it would in an engine like No. 54, for example, which is run at 167 lbs. The effect of the vacuum on the low-pressure cylinder is much more telling when the boiler pressure is low, and less work is done in the high-pressure cylinder, than it is when the boiler pressure is high.

At pressures ranging between 120 and 140 lbs., it would appear from these records that a 4-valve compound engine running non-condensing would use not over 21.5 lbs. of feed-water per I. H. P. per hour; and a similar engine running condensing, with the usual proportions of cylinders, would use not over 14 lbs. The difference between the two is 7.5 lbs., or about 35% in favor of the condensing engine. Allowing, say, 2% for power used by a direct-connected air-pump, and making further allowance, as in the case of the simple engines mentioned, for the effect of a feed-water heater, the net saving of fuel in favor of the condensing engine is about 25%.

The effect on a pair of condensing engines produced by running one end of one cylinder non-condensing is shown in two cases. In Engine No. 3 the effect was to increase the consumption of feed-water from 21.11 lbs. to 22.68 lbs., or about 7%. In Engine No. 9 the increased consumption amounted to about 12%. The object of running an engine in this manner is to utilize a portion of the steam for heating the feed-water, or for other uses to which exhaust steam can be adapted. If its use is confined to heating feed-water, and the amount is 110°, or that corresponding to the instances heretofore noted, an advantage would be produced, provided the increased consumption did not exceed 11%. If in these two engines the exhaust steam from the single end were used for that purpose, there would be a net gain corresponding to about 7% in Engine No. 3, and a net loss corresponding to 1% in Engine No. 9.

V. EFFECT OF SUPERHEATING.

The effect which superheating has upon the economy of an engine is clearly shown in the case of Engine No. 1, where test No. 1 C was made with the steam superheated 82°, and test No. 1 B under practically the same conditions, except that the steam was practically dry. This was a simple non-condensing engine. The economy produced by the superheating was sufficient to reduce the feed-water consumption from 29.34 lbs., per I. H. P. per hour, to 26.83 lbs., or 8.6% or about 1% for each 10° of superheating. This may be examined further by comparing this and other simple engines which use superheated steam with those using ordinary steam. The effect can best be studied by comparing the cylinder condensation and leakage, in the case of the engines using superheated steam, with the curve of condensation and leakage given on the chart for simple engines using ordinary steam. For example, in the case of test No. 1 C the cut-off is .392, the proportion of feed-water accounted for at cut-off .947, and the cylinder condensation and leakage 5.3%. On the curve for ordinary steam referred to, the condensation and leakage at a cut-off of .392 is 16.7%. The difference between 5.3 and 16.7, which is 11.4%, represents the reduction in the condensation due to the superheating, as determined by this method of comparison. Pursuing the matter in the same way for the remaining engines, we have the following table:

No.	Degrees of Super-Heating.	Cut-off.	Proportion of Feed-Water Accounted for at Cut-off.	Proportion of Cylinder Condensation and Leakage.	Cylinder Condensation and Leakage derived from Curve for ordinary Steam.	Proportion Reduced by Super-Heating.
1 C	82	.392	.947	.053	.167	.114
4	25	.233	.766	.234	.255	.021
8 A	37	.247	.819	.181	.243	.062
8 B	37	.165	.747	.253	.334	.081
9 A	24	.185	.820	.180	.307	.127
9 B	24	.225	.836	.164	.261	.097
15	59	.281	.895	.105	.215	.110
Average,	41°					.087

From this comparison it appears that with steam superheated 41° (generally at a point near the boilers, and a considerable distance from the engine), the proportion of condensation and leakage was reduced an average of 8.7%. Assuming as a criterion the relation between the actual saving in the case of No. 1 Engine, and the reduced proportion of cylinder condensation, which was about .8, this reduction in the cylinder condensation corresponds to an actual saving of feed-water of 7%. Assuming that if the engines had been supplied with ordinary steam, this steam would have contained 1% of moisture, corresponding in round numbers to, say, 20° of superheating, the difference in the quality of the steam in the two cases, expressed as superheating, is about 60°. According to this calculation, therefore, the effect of the superheating is to reduce the feed-water consumption 7% for a superheating of 60°, or a trifle over 1% for each 10°; and this practically corroborates the evidence furnished by the tests on Engine No. 1.

The compound engine which shows the highest economy of any in the list, is one which is supplied with superheated steam; and although this fact may be considered as one reason for the high result, there were other conditions which were favorable, and the exact effect of the superheating is a matter of conjecture.

Incidentally, it should be noted that superheating produces a marked effect in the character of the expansion line of the indicator diagram. In Engine No. 1 this is clearly revealed by a comparison of the steam accounted for by the indicator at cut-off and release. In test No. 1 B, where the engine was running with ordinary steam, the proportion accounted for at cut-off is .839, and that at release is .861, which is an increase of .022. On the other hand, on test No. 1 C, where the steam was superheated 82°, the proportion accounted for at cut-off was .947, and at release .900, there being a reduction here of .047. This change is evidently due to the reduced condensation produced by the superheating, and the consequent reduction in the amount of re-evaporation during expansion.

VI. RELATIVE ECONOMY OF SIMPLE, COMPOUND AND TRIPLE EXPANSION ENGINES.

In comparing the economy of a compound or other multiple expansion engine with that of a simple engine, the question may be raised, What should be the conditions of the comparison? One method of comparing the two would be to select those running under the same boiler pressure and quality of steam, and with similar provisions in regard to jacketing. This method may be interesting and valuable for scientific research; but for practically showing the advantages of compound engines, it is of little importance, because one of the principal objects in compounding is to enable the economy due to large expansions and high pressures to be obtained without the sacrifice which such expansions produce when carried on during the single stage which occurs in one cylinder. The nearest approach to a comparison of this kind, derived from the tests reported, is that of compound Engine No. 32, which was made with a boiler pressure of 94.8 lbs. Here the engine was unjacketed, and no provision was made for re-heating between the cylinders. If we compare this with the very best result obtained from a simple condensing engine, that of No. 22, there appears, even under these circumstances, a marked difference in favor of the compound engine. These figures are 18.49 for the simple engine, and 16.28 for the compound; and the difference is 2.21 lbs., or about 12%. Comparing this, again, with simple Engine No. 28, which is running at 70 lbs. pressure on a consumption of 19.45 lbs. of feed-water per I. H. P. per hour, the difference is 3.17 lbs., or 16.3%.

A fairly satisfactory comparison between compound engines and simple engines, where no jacketing or re-heating is provided, can be made by using compound Engine No. 36. This engine was jacketed; but the jackets were not drained, and consequently, under the circumstances, they were ineffective. In this engine the consumption of feed-water was 14.05 lbs. per I. H. P. per hour when running at a pressure of 106.8 lbs. If we compare this with No. 28, simple engine, the difference is 5.4 lbs., or 27.8%.

A general comparison between the compound and simple engines may be made without regard to the matter of pressure or the use of jackets and re-heaters, and without regard to the quality of the steam, omitting the three engines which have an excessively high ratio of cylinder areas. The engines selected are those of the Corliss or other 4-valve type. Such a comparison is made in the following tables.

Simple Condensing Engines.

Number.	Feed-Water Consumed per I. H. P. per Hour.
3 A	21.11
8 A	19.39
8 B	18.71
9 A	18.25
18 A	20.31
18 B	20.56
22	18.49
28	19.45
30	21.42
Average,	19.74

Compound Condensing Engines.

Number.	Feed-Water Consumed per I. H. P. per Hour.
32	16.28
34	13 28
36	14.05
37	13.37
43	13.26
48 A	14.12
48 B	14.01
48 C	15.08
49	14.18
50	13.28
53	15.78
55	13.27
56	14.60
57	14.10
58 A	13.21
Average,	14.12

The average of the results on the simple condensing engines is 19.74 lbs., and of those on the compound condensing engines, 14.12 lbs. The difference is 5.62 lbs., or 28.5% of the feed-water consumption of the simple engines.

In the case of non-condensing compound engines of the 4-valve type, there is only one engine in this class, Engine No.

46. Test No. 46 B on this engine gave a feed-water consumption of 21.59 lbs. This may be compared with Engine No. 2, which was run non-condensing at a pressure of 101 lbs., and gave 25.8 lbs. consumption. Here the economy due to the compound engine is 4.21 lbs., or 16.3%. This is rather unfavorable to the compound engine on account of the relatively small difference in the boiler pressures.

Referring to the single-valve engines running condensing, comparison may be made between Engine No. 41 and Engine No. 19. Engine No. 19, the simple engine, used 27.15 lbs. per I. H. P. per hour, and Engine No. 41 B, compound, used 19.1 lbs., the difference being 8.05 lbs., or an economy of 29.6%. There are no single-valve engines of the non-condensing class from which to make a fair comparison between the compound and simple engines, owing to the great difference in the sizes; but the results obtained on engines of this kind, disregarding their size, are of the same kind as those already discussed.

The results of the tests on the two triple expansion engines which are given, show an average consumption of 12.63 lbs. of water per I. H. P. per hour. This is below the average of 14.12 lbs. for the various compound condensing engines which are tabulated, and it is below the result obtained from any individual engine given in that table. It is better to the extent of 10%, compared with the average. This result is not, however, so good as that obtained from the special compound Engine No. 51, where the ratio of cylinder areas is about the same as the ratio between the low-pressure cylinder and the high-pressure cylinder of triple expansion engines.

VII. ECONOMY OF STEAM JACKETING AND RE-HEATING IN COMPOUND ENGINES.

There are two compound engines given where the effect of shutting off the steam from the jackets and re-heater tubes was tested, these being No. 47 and No. 52. In each of these cases, the difference in the feed-water consumption per I. H. P.

per hour was 2%. Both of these are cases where the ratio of area of the two cylinders was unusually large, and the re-heating surface in the receiver was also unusually large, being sufficient to superheat the steam that passed into the low-pressure cylinder. Whatever value jacketing and re-heating may have in a compound engine, it may be reasonably expected that it would show to the best advantage where the expansion is carried to the greatest extent; and consequently the conditions of these two cases are as favorable to a good showing for the jackets as they could be in most engines of the compound type. It would appear then, that 2% is the most that can be expected for the saving of steam due to jackets and re-heaters in ordinary compound engines of the types referred to.

There are none of the tests of the other compound engines which furnish much actual data on the subject; but it may be said that the superficial indications of the results of the tests where the engines are jacketed, furnish little ground for the belief that jacketing had much effect upon the economy. Take the case of Engine No 58 A, which had no jackets, but which was fitted with a re-heating receiver. The consumption of feed-water was 13.21 lbs. per I. H. P. per hour, and this is lower than any result given where the engine was provided with jackets. No doubt the unusually tight condition of the valves and pistons in this case had a favorable effect; but if jacketing is necessary for good economical results and the advantage it produces is a marked one, its absence in Engine No. 58 should have produced a much more noticeable effect.

Beyond the saving in steam consumption produced by jackets, which in Engines No. 47 and 53 amounted to 2%, there is a further saving in fuel which cannot be overlooked, which may be obtained by returning the hot water condensed in the jackets to the boilers. The temperature of this water is ordinarily about 300°, and its quantity on the tests noted was 7.7% in one case, and 11% in the other, averaging 9.3% for the two. If the temperature of the main supply of feed-water is 100°, the return of this water to the boilers would add about 19° to the temperature of the feed-water, and increase the efficiency of the boilers a little less than 2%. If the temperature of the main feed-water was at a higher point, the effect of the heat returned from the jackets would be correspondingly less. If we make this for an average case, 1½%, we should have the combined economy of the jackets due to both causes about 3½%.

There is one test of a compound engine which was made to determine the effect of shutting off the steam from the re-heater, in a case where the cylinders were unjacketed. This relates to Engine No. 48. Test A was made with the re-heater on, and test B with the re-heater off. The figures show that the engine was the most economical in the latter case, the difference between .11 of a pound or .7 of 1%; so that in this one instance it would seem that the use of the re-heater produced a loss in steam consumption instead of a gain. If allowance is made for the heat which could be returned from the water of condensation to the boilers, the advantage from this source would be nearly 1%, so that there was a slight advantage in fuel economy due to the use of the re-heater.

Whatever the actual economy due to jacketing or to re-heating or to both, which from the evidence of these tests appears to be rather small, there is no question but that the action of the jacket and the re-heater produces a powerful influence on the steam in its passage through the cylinders. The effect upon the indicator diagrams is very marked. The use of these appliances makes the engine more powerful in view of the fact that it increases the work done by the low-pressure cylinder for a given amount performed by the high-pressure cylinder. In Engine No. 47, the low-pressure cylinder developed 34 horse-power less than the high-pressure cylinder when the jackets and re-heater were off, and 10 horse-power more than the H. P. cylinder when the jackets were on. In Engine No. 52, the low-pressure cylinder developed 24 horse-power more than the H. P. cylinder when the jackets and re-heater were off, and 92 horse-power more when the jackets were on. In Engine No. 48, the low-pressure cylinder developed 56 horse-power less than the H. P. cylinder when the re-heater was off, and 19 horse-power less when the re-heater was on.

The effect of the jacketing and re-heating is also seen to be very marked when comparison is made between the steam accounted for in the two cylinders. In Engine No. 47, with the jackets off, the steam accounted for in the L. P. cylinder at cut-off is 10.8% less than in the H. P. cylinder; whereas with the jackets on, the difference is only 1%. In Engine No. 52, with jackets off, the steam accounted for in the L. P. cylinder is 8.4% less than in the H. P. cylinder. When the jackets were on, it was 12.9% more than in the H. P. cylinder. In Engine No. 48, the steam accounted for in the L. P. cylinder with the re-heater off, was 11.6% less than that in the H. P. cylinder; whereas, when the re-heater was on, it was only 4.2% less. In Engine No. 55 the effect of the re-heater on the diagrams is seen to be considerable, from the fact that the steam accounted for in the L. P. cylinder is 1.3% more than that accounted for in the H. P. cylinder.

VIII. EFFECT OF RATIO OF CYLINDER AREAS IN COMPOUND ENGINES.

In most of the compound engines given, where these are of the Corliss or other 4-valve type, the ratio of cylinder areas is between 3.5 and 4. Three cases are given, however, where the ratio is about 7 to 1, these being Engines 47, 51, and 52. The engines with the large ratio of cylinder area show more economical results than the others. The difference is not so noticeable in No. 52 as it is in Nos. 47 and 51. In both these cases, however, the pressure is higher than it is in most of the tests given with the lower ratios; and this higher pressure furnishes one reason for the better result. There is one case of a pressure of 151 lbs. in an engine having a low ratio with which these may be compared, and that is Engine No. 55. This engine gives a horse-power for 13.27 lbs. of feed-water per hour. Engine No. 47 B gives 12.45 lbs., while Engine No. 51 C, gives 11.89 lbs. Taking the average of the last two, which is 12.17, there is a difference between the two cases in favor of the larger ratio of areas of 1.1 lbs. or 8%. Engine No. 55, as will be seen, does not give so well-formed diagrams, there being considerable wiredrawing in the H. P. cylinder; and the result obtained on this engine is not so good as it would have been if these conditions had been better. Making due allowance for this, however, and further allowance for the fact that Engine No. 51 was supplied with slightly superheated steam, there appears to be a noticeable advantage in the use of the higher ratio of cylinder area for an engine running at 150 lbs. pressure. It is a noteworthy fact that with the high ratio of area, an excellent steam distribution, and a slight amount of superheating, the most economical result given in the whole list of tests is produced, — Engine No. 51 C producing a horse-power for 11.89 lbs. of feed-water per hour.

IX. MISCELLANEOUS.

The tests furnish some indication as to the loss of economy produced by light loads, especially in non-condensing engines. In Engine No. 16, which is a single-valve, single-acting engine of the high-speed class, the consumption of steam per horse-power per hour was increased from 32.6 lbs. to 36.27 lbs., by reducing the horse-power developed from 44.8 H. P. to 25.7 H. P. In Engine No. 23, which is of the single-valve high-speed class, the consumption increased from 30.63 lbs. to 31.78 lbs., corresponding to a reduction of load from 39.4 H. P. to 22.2 H. P. In Engine No. 31, which is of the Corliss type, the consumption was fairly constant with a load varying from 222 H. P. to 342 H. P., but with lighter loads it rapidly fell off; and with the load of the idle engine and shafting, which was 37 horse-power, the consumption rose to 73.63 lbs. per I. H. P. per hour. In Engine No. 42, which is a single-valve high-speed compound, the consumption was increased from 25.2 lbs. to 44.89 lbs. by reducing the load from 152.5 H. P. to 45.6 H. P.

In Engine No. 54, which is a single-valve compound, the feed-water consumption was nearly constant for loads of 242.9 H. P. and 187.5 H. P.; but it was increased from 21.14 lbs. to 24.99 lbs. by dropping the load to 103.4 H. P. In Engine No. 41, a single-valve compound condensing, the consumption was increased from 19.1 lbs. to 22.74 lbs. by reducing the load from 196.8 H. P. to 90.5 H.P. In Engine No. 45, which is a double-valve compound condensing, the consumption was increased from 15.71 lbs. to 17.22 lbs. by reducing the load from 244.5 H. P. to 123.4 H. P.

Very little information of definite character is furnished by the tests as to the effect of size of cylinder on economy. Most of the smaller engines given are of the single-valve class, with shaft governors, running at high speed; and although these generally show less economy than the larger engines, it would hardly be fair to attribute it to the smaller size of cylinder when other differences of condition are known to be of much importance. Two cases are given for Corliss engines which seem to show that a considerable difference of size has no appreciable effect. These are Engine No. 2, having a 28.5″ x 59.5″ non-condensing cylinder, and Engine No. 31, which had 2–16″ x 42″ cylinders. The former gave a horse-power for 25.8 lbs. of feed-water per hour; and the latter, when working at about the same cut-off, for 25.9 lbs. per hour, or practically the same result. Cylinder condensation and leakage is 2.1% greater in the case of the smaller engine; and this fact furnishes a slight indication that the smaller engine was the more wasteful.

It needs but a glance at the results of the various tests to show that the 4-valve engines are more economical as a type than those having a less number of valves; and this is true whether they are simple or compound, and whether condensing or non-condensing. The single-valve compound non-condensing Engine No. 54, compared with the 4-valve compound non-condensing Engine No. 46, shows a better result, some 2%; but it will be observed that the former works under a pressure of 165 lbs., while the pressure in the latter case is 135.

As the economy of non-condensing compound engines is greatly affected by the boiler pressure, the single-valve engine in this case has an undoubted advantage, which more than makes up for the difference produced by the valve. The superior economy of the 4-valve type is evidently due in part to the better distribution of the steam in the cylinders, as revealed by more perfectly formed diagrams; and, in some cases to the tighter condition of the valves and pistons.

One test is given that shows the loss in economy due to the variable load produced in electric railway service. This is Engine No. 58 B, which is a Corliss compound condensing engine. Compared with test No. 58 A, which was made with the same engine working under a steady load, the loss is only 2.5%. On the test with the variable load, the average power was 843.4 H. P., while that with the steady load was 1030.1 H. P. It is evident that the difference in economy shown was caused to a considerable extent, if not wholly, by the fact that in the variable load the engine is at times underloaded, and not working to its best economical advantage. This was probably an unusually favorable showing for a variable load in the service mentioned, for the reason that the range of variation was less than occurs in much work of this kind. An examination of the indicator diagrams gives some idea of the extent of the variation.

One method of reducing the loss of steam where compound engines are used, is to exhaust the air-pump into the receiver of the engine. This virtually converts the air-pump from a simple engine to a compound engine. The effect of thus utilizing the exhaust steam of the air-pump is seen in test No. 57. The effect is shown by the large increase of the amount of steam accounted for in the low-pressure cylinder, as compared with that in the high-pressure cylinder. The increase is from .696 to .80, or .104. In Engine No 55, which is of similar type except in this particular, the increase is only .013; and in Engine No. 58 A, also similar in type, there is a falling off of .08. In Engine No. 57, the steam used by the air-pump when exhausting into the condenser amounted to .9 of a pound per

I. H. P. per hour, and when exhausting into the receiver it was, of course, a much larger quantity; but in spite of this the extra power produced by the use of the steam in the low-pressure cylinder was such that the entire consumption of the engine and condenser was only 14.1 lbs. per I. H. P. per hour.

One test on a compound engine is given, where the water drained from jackets and receiver was pumped into a flue heater, and the steam produced by its re-evaporation brought back to the receiver and used in the low-pressure cylinder. This is Engine No. 50. Under the circumstances of a comparatively low boiler pressure, which was 108.1 lbs., the economical result obtained, which was 13.28 lbs. per I. H. P. per hour, must be considered excellent. The engine, however, was supplied with superheated steam; and this condition is, no doubt, accountable, in some degree at least, for the result obtained. It is doubtful whether the re-heating had any marked effect; because it appears that the steam accounted for in the L. P. cylinder is .77 as against .889 in the H. P. cylinder, showing a loss between the two of .119. If this is compared with Engine No. 49, which is supplied with ordinary steam, and had no re-heating feature, there is a difference between the two cylinders of .106; so that there is no more loss in this case between the cylinders than in Engine No. 50, which had the re-heating system.

The evidence of the tests furnish some data upon the effect of varying the receiver pressure in a compound engine, but this data is not conclusive as applied to other engines. In the case of Engine No. 51, three tests made with nearly the same load and with a receiver pressure, ranging from 5.4 lbs. above the atmosphere to 12.9 lbs., the cut-off in the low-pressure cylinder being gradually shortened as the pressure increased, showed a gradual reduction in the feed-water consumed per I. H. P. per hour. With the lowest pressure, it was 12.29 lbs., and with the highest, 11.89 lbs. In Engines No. 47 and 52, where similar tests were made with three different receiver pressures, practically the same result is produced at the two extreme pressures. In one case, the intermediate pressure gave a slight reduction, whereas in the other, the intermediate pressure gave a slight increase in the consumption.

IN CONCLUSION.

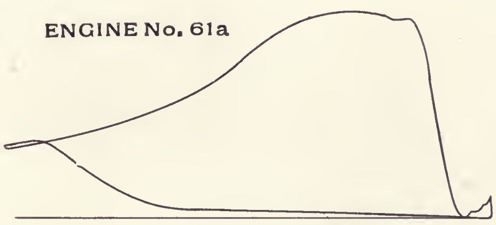
A careful study of these tests should be of service to engineers in designing new plants or re-organizing old ones, inasmuch as they show, within the limits covered, what designs and practices should be avoided, and what conditions should be observed in order to secure desired results in the best manner.

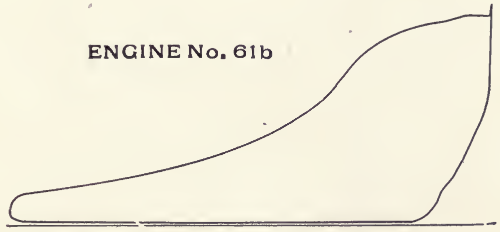
VALVE SETTING.

ENGINE No. 61.

Double valve, 6″ x 14″. Speed, 210 revolutions per minute.

This is an automatic cut-off engine with slide valves and shaft governor. The main valve is of the box pattern, with balance plates on the back face. Steam is admitted into the interior of the box before it passes through the ports into the cylinder. The cut-off valve rides on a seat inside, and is operated by a separate eccentric, which is shifted by the action of a shaft governor. The diagrams here given show the effect of moving the eccentric which operates the main valve an angular distance of 43° on the shaft. This represents a distance of $1\frac{1}{2}''$ on a shaft 4″ in diameter. No. 61*a* was taken before, and No. 61*b* after, the change.



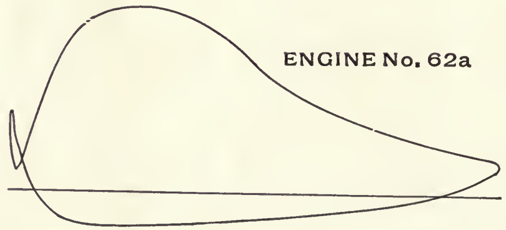


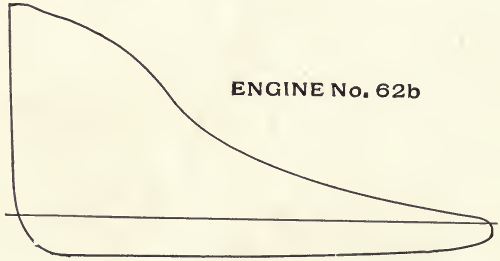
ENGINE No. 62.

Four valve, 16″ x 48″. Speed, 82 revolutions per minute.

This engine is of the 4-valve type, the steam valves being slide valves, and the exhaust, Corliss valves. They are operated by separate eccentrics. Changes in the setting of the valves consisted in moving the steam-valve eccentric ahead 2″ measured on the circumference of the 8″ shaft, moving the exhaust eccentric ahead $\frac{5}{8}$″, adjusting the tappet which operates the steam valve so as to obtain earlier admission, and shortening the exhaust-valve rod two turns to obtain earlier release. The diagrams were taken from the head end, No. 62 *a* before, and No. 62*b* after, the change.

In connection with these changes feed-water tests were made which showed a saving of 8 % on the steam used by the plant of which this engine formed a part; the total power of the plant being somewhat more than twice the power developed by this engine.



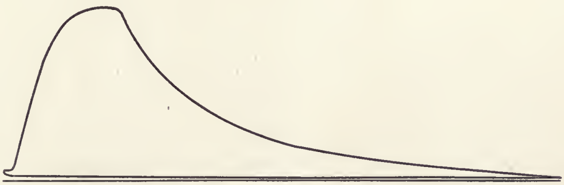


ENGINE No. 63.

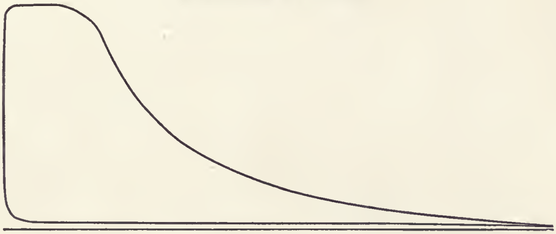
Four valve (Corliss), 18″ x 48″. Speed, 57 revolutions per minute.

This engine is of the ordinary Corliss type with single eccentric. The changes in the valves consisted in moving the eccentric forward ½ inch on a 10″ shaft, and shortening the steam-valve rod 4 turns, or 4 threads. The diagrams were taken from the head end, No. 63*a* before, and No. 63*b* after, the change.

ENGINE No. 63a



ENGINE No. 63b

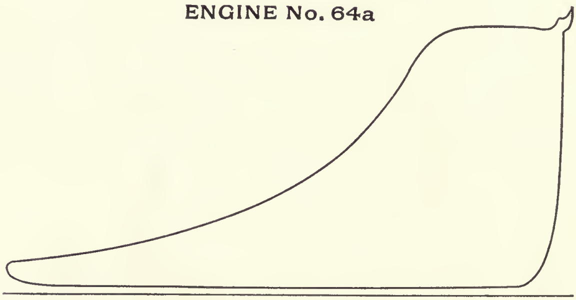


ENGINE No. 64.

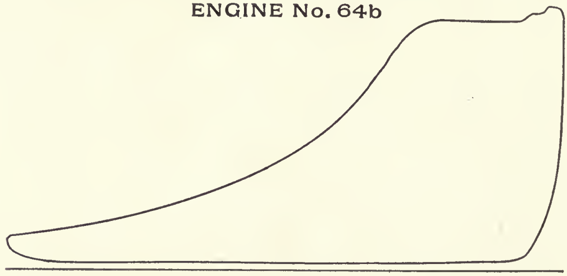
Four valve (Corliss), 23″ x 48″. Speed, 51 revolutions per minute.

The steam-valve rod was lengthened 6 turns, or 6 threads, to reduce the lead. The diagrams are from the crank end, No. 64*a* being taken before, and No. 64*b* after, the adjustment.

ENGINE No. 64a



ENGINE No. 64b

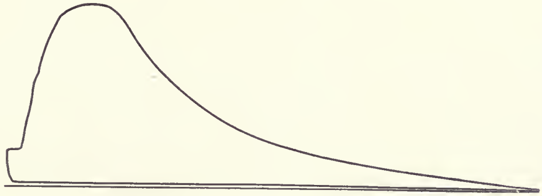


ENGINE No. 65.

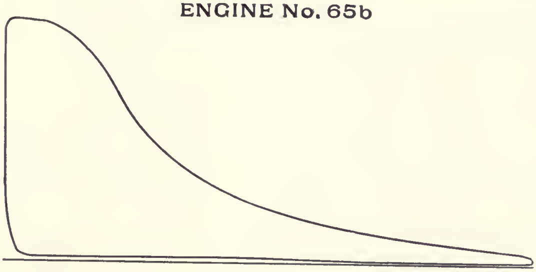
Four valve (Corliss), 14″ x 36″. Speed, 67 revolutions per minute.

The eccentric was moved forward $\frac{3}{8}$″ on the 7″ shaft. The steam-valve rod was shortened 6 turns or 6 threads. The diagrams are from the head end, No. 65*a* being taken before, and No. 65*b* after, the change.

ENGINE No. 65a



ENGINE No. 65b

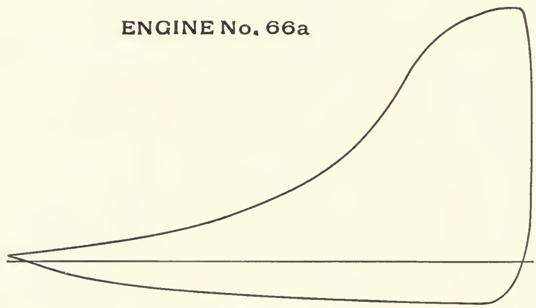


ENGINE No. 66.

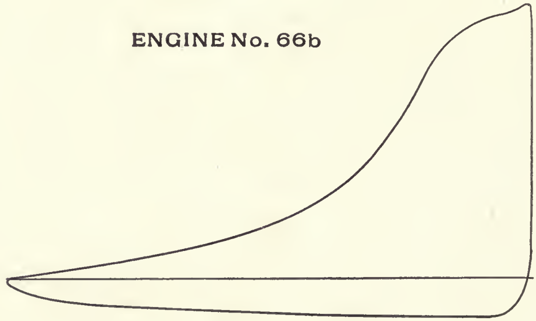
Four valve (Corliss), 26″ x 60″. Speed, 51 revolutions per minute.

The changes in the valve-setting consisted in moving the eccentric forward $\frac{1}{2}''$ on the 12″ shaft and shortening the exhaust-valve rod $2\frac{1}{2}$ turns, or 5 threads, so as to secure earlier release. The diagrams are from the head end of the cylinder, No. 66*a* being taken before, and No. 66*b* after, the change.

ENGINE No. 66a



ENGINE No. 66b

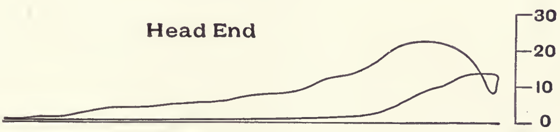


ENGINE No. 67.

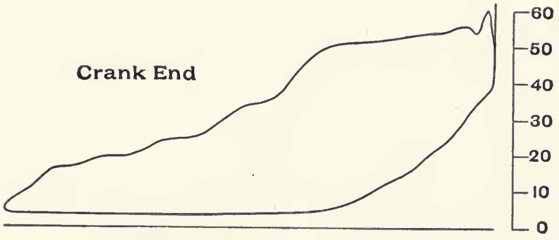
Single valve, 8″ x 10″. Speed, 326 revolutions per minute.

This engine is of the automatic cut-off type with shaft governor, shifting eccentric, and balanced slide valve. These diagrams show the effect of unequal adjustment of the lap of the valve. The first set, 67*a* and 67*b*, was taken with the engine loaded. The mean effective pressure at the head end is 8 lbs., and at the crank end, 32.4 lbs. The second set, 67*c* and 67*d*, was taken with a friction load. Here the mean effective pressure at the head end is a minus quantity, and at the crank end a plus quantity. The third set, 67*e* and 67*f*, was taken under the same conditions of load as the second, after equalizing the lap. With this adjustment the mean effective pressure at the head end was 1.9 lbs., and at the crank end, 3.4 lbs.

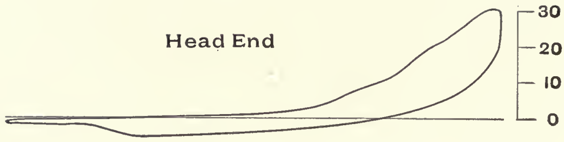
ENGINE No. 67a



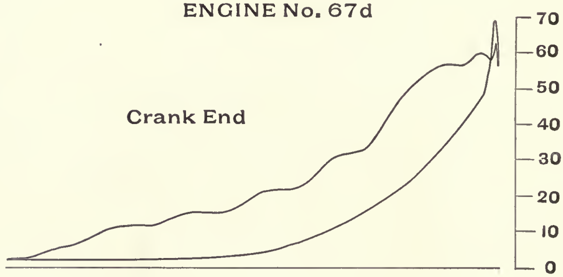
ENGINE No. 67b



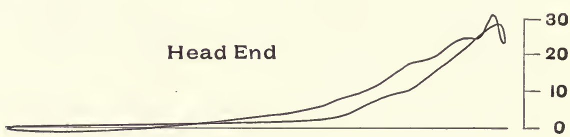
ENGINE No. 67c



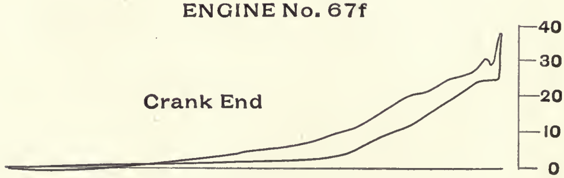
ENGINE No. 67d



ENGINE No. 67e



ENGINE No. 67f

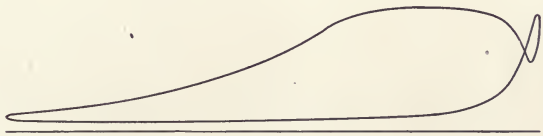


ENGINE No. 68.

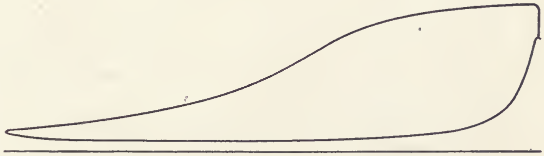
Four valve, 13″ x 36″. Speed, 61 revolutions per minute.

This engine has double poppet valves for admission, and slide valves for exhaust. The valves are operated by a train of gears and cams. The adjustments consisted in moving forward the cam which operates the steam valve so as to produce earlier admission. The diagrams are taken from the head end, No. 68*a* before, and 68*b* after, the changes.

ENGINE No. 68a



ENGINE No. 68b



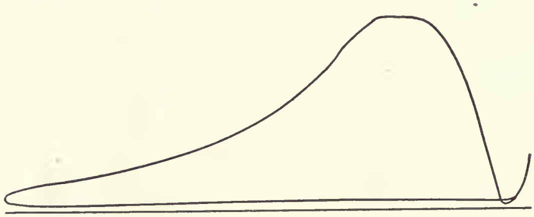
ENGINE No. 69.

Four valve, 11″ x 30″. Speed, 80 revolutions per minute.

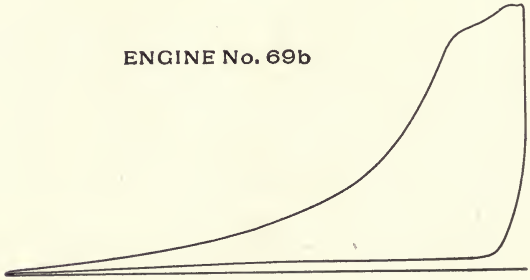
This engine has double-poppet admission valves, and slide valves for exhaust; all driven by a train of gears.

The steam-valve cam was moved forward $\frac{1}{4}''$ on its shaft, and the exhaust cam $\frac{1}{8}''$. The diagrams are taken from the crank end, No. 69*a* before, and No. 69*b* after, the adjustments.

ENGINE No. 69a



ENGINE No. 69b



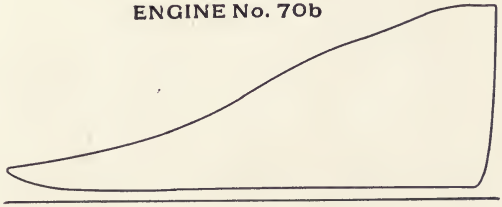
ENGINE No. 70.

Four valve, 18″ x 42″. Speed, 55 revolutions per minute.

In this engine the steam valves are double poppet, and the exhaust valves, slides. The mechanism is driven by means of bevel gears. The adjustment of the valves consisted in moving the driving-gear forward on the shaft two teeth. The total number of teeth on this gear was 44. The diagrams are from the head end, No. 70*a* being taken before, and No. 70*b* after, the changes.

ENGINE No. 70a

ENGINE No. 70b

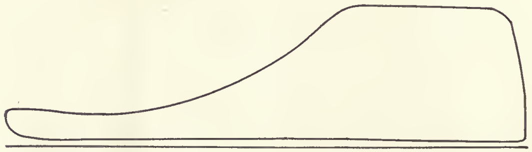


ENGINE No. 71.

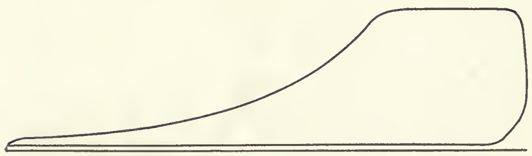
Four valve, 14″ x 35″. Speed, 49 revolutions per minute.

This engine has two double-poppet steam valves, and slide valves for the exhaust; all driven through a train of gears. The changes consisted in moving the stem of the steam valve in so as to clear the driving-cam, and setting the gear forward on the shaft 2 teeth. The diagrams were taken from the head end, No. 71*a* before, and No. 71*b* after, the adjustments.

ENGINE No. 71a



ENGINE No. 71b



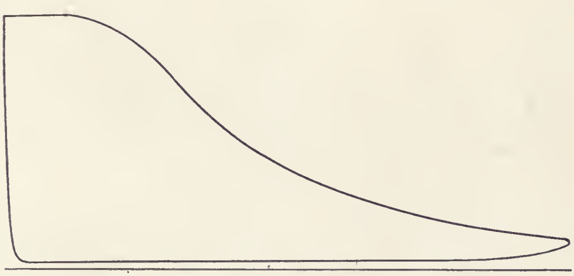
ENGINE No. 72.

Four valve, 16″ x 36″. Speed, 72 revolutions per minute.

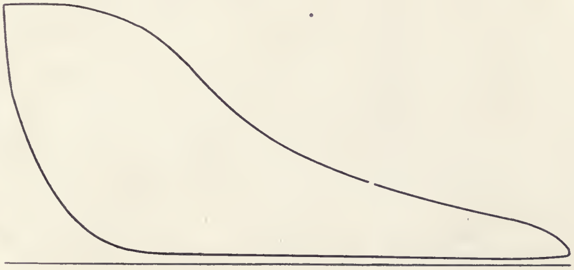
This engine has two double-poppet steam valves, and slide valves for the exhaust. They are driven through a train of gears.

The gear which drives the valves was changed, with a view to securing compression of the exhaust steam up to the initial pressure, being moved forward, thereby hastening the release as well as the compression. This change was made with the object of studying the effect of compression upon the actual economy of the engine under conditions of practically the same load. So far as this test showed anything, under these conditions, there was in reality a slight increase in the amount of feed-water consumed per horse-power per hour, attending the earlier compression. The diagrams were taken from the crank end, No. 72*a* before, and No. 72*b* after, the change.

ENGINE No. 72a



ENGINE No. 72b



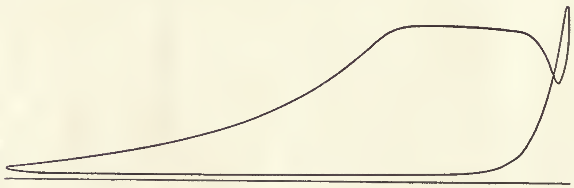
ENGINE No. 73.

Four valve, 26″ x 48″. Speed, 50 revolutions per minute.

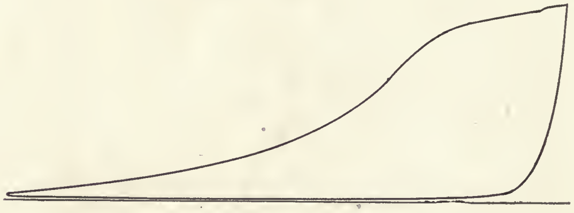
The steam valves in this engine are slide valves, and the exhaust are Corliss valves.

The change here consisted in moving the driving-gear for the steam valves one tooth ahead, the total number being 42, and in shortening the exhaust rod so as to reduce the lap on the exhaust valve. The diagrams were taken from the head end, No. 73*a* before, and No. 73*b* after, the changes.

ENGINE No. 73a



ENGINE No. 73b

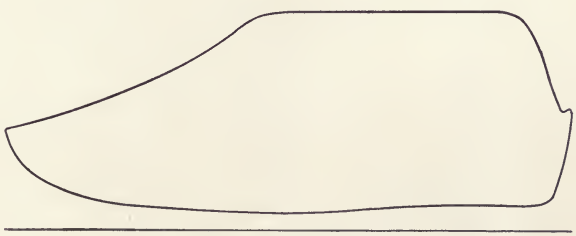


ENGINE No. 74.

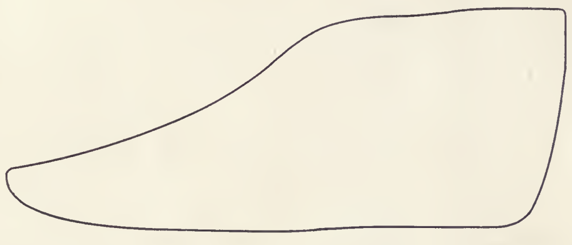
Four valve, 18″ x 48″. Speed, 64 revolutions per minute.

All the valves in this engine are slides driven by double eccentrics. The adjustments consisted in advancing the steam-valve eccentric $1\frac{1}{4}''$ on the 9″ shaft, and the exhaust eccentric $\frac{1}{2}$ inch. The diagrams were taken from the head end, No. 74*a* before, and No. 74*b* after, the changes.

ENGINE No. 74a



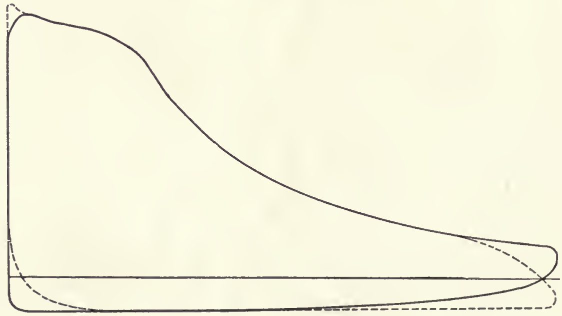
ENGINE No. 74b



ENGINE No. 75.

Four valve (Corliss), 30″ x 48.″ Speed, 80 revolutions per minute. The setting of the valves was changed by the introduction of a separate eccentric for driving the exhaust valves, and the adjustment of this eccentric so as to obtain early compression. With a single eccentric the engine operated unsatisfactorily on account of the noisy action of the piston and valves, there being decided and annoying sounds at each end of the stroke, which could be distinctly heard by a person standing at a distance of 20 feet from the cylinder. When the additional eccentric had been applied and the valves readjusted, the troublesome sounds so far disappeared that it was necessary for the observer to hold his ear close to the cylinder to be aware of any disturbance. The diagrams were taken from the head end, and for ready comparison, they are superimposed, the full line being taken with single eccentric and the dotted line after changing to double eccentrics and resetting the valves.

ENGINE No. 75

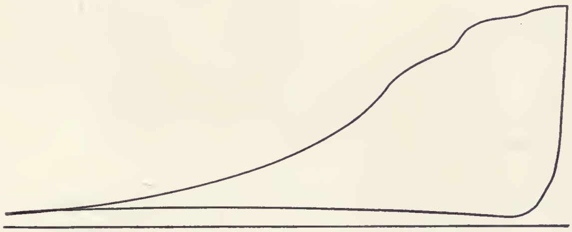


ENGINE No. 76.

Four valve, 20″ x 50.″ Speed, 64 revolutions per minute.

This engine has four slide valves, all operated by means of a train of gears. The diagram here given is presented as a curiosity, showing the effect of admission of steam to the cylinder subsequent to the cut-off, due to the rebounding of the valve after it had once closed. The diagram was taken from the crank end of the cylinder.

ENGINE No. 76



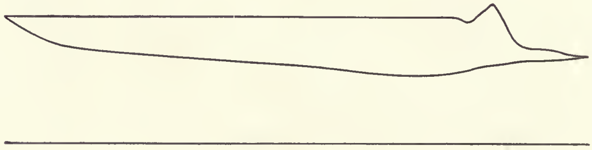
ENGINE No. 77.

Elevator Engine, 8″ x 10.″

This is introduced as a curiosity, and at the same time it reveals the wasteful character of this class of engine. There is an absence of expansion and exceedingly high back pressure, both of which are required by the exigencies of the service and type of valve mechanism which that service necessitates. This diagram also illustrates the effect of improper location of the indicator on the cylinder. In this case it was placed at a short distance from the end of the stroke, so that the piston ring covered the hole until it had moved a certain distance on the forward stroke. The hump on the diagram is caused by this defective location.

The diagram was taken on an upward trip.

ENGINE No. 77

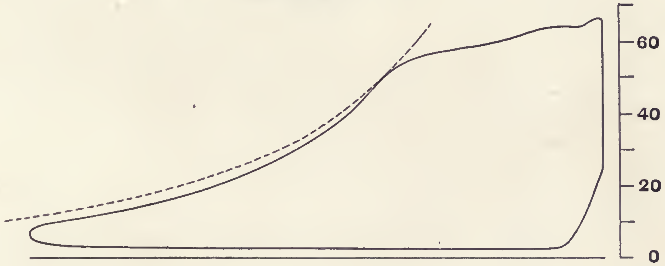


ENGINE No. 78.

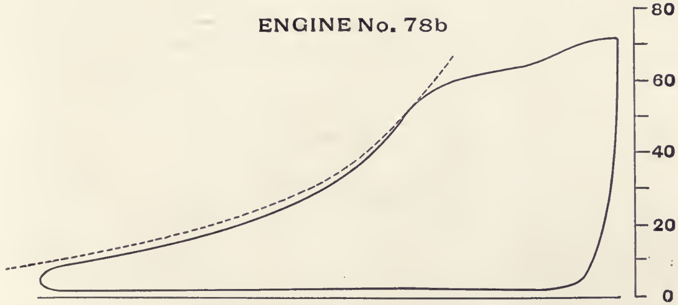
Four valve (Corliss), 23″ x 60.″ Speed, 74 rev. per min.

These diagrams furnish another instance showing the influence, or want of influence, of leakage. In this case the trouble was with the piston. Diagram No. 78*a* was taken with the piston leaking, the packing ring being broken, and No. 78*b* was taken when the ring had been renewed, and the piston made tight. Feed-water tests made under both conditions showed that the leaking engine used 34.5 lbs. of steam per I. H. P. per hour, and the tight engine, 27.7 lbs. The difference is about 20 %. The boiler pressure was higher after the repairs than before, but this does not affect the general features. So far as the expansion line is concerned, the leakage of the piston had no appreciable effect. There is a noticeable difference in the compression lines, but in the leaking engine this alone would not prove the leakage in question. The diagrams are from the crank end.

ENGINE No. 78a



ENGINE No. 78b

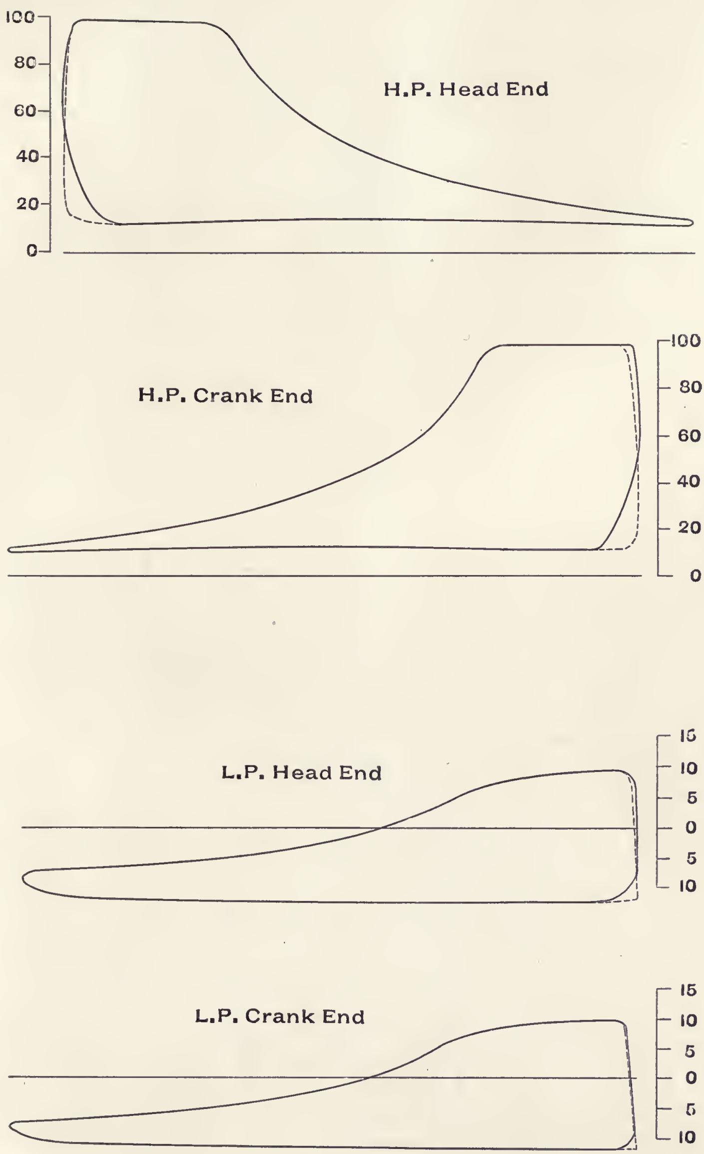


ENGINE No. 79.

Four valve (Corliss), cross compound, 24″ and 44″x 72.″ Speed, 61 revolutions per minute.

The main object in changing the adjustment of the valves in this engine was to secure a greater amount of compression, and a more quiet operation of the engine. Previous to the changes there was considerable knocking in the main connections when the centers were passed, and internal noises in both cylinders. The effect of the changes was to almost wholly overcome these defects in the running qualities. The adjustments consisted in moving the eccentric of the high-pressure cylinder forward $\frac{11}{16}$″ on the 12″ shaft and the eccentric of the low-pressure cylinder forward $1\frac{1}{8}$.″ The steam-valve rods of the high-pressure cylinder were both shortened two threads, so as to give earlier admission. The exhaust rods of the same cylinder were lengthened 8 threads each, so as to increase the compression. The steam-valve rods of the L. P. cylinder were shortened three threads each, so as to give earlier admission; and the exhaust rods were each lengthened 6 threads, so as to obtain earlier compression. To better reveal the effect of the changes, the diagrams are superimposed, the dotted lines being taken before, and the full lines after, the adjustments.

ENGINE No. 79

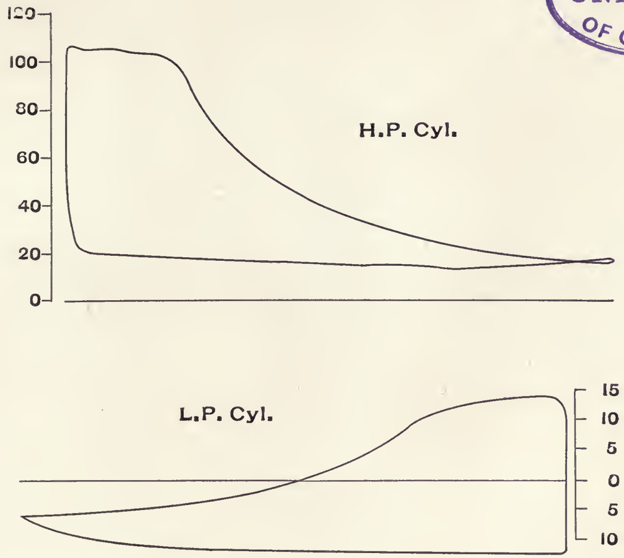

H.P. Head End
100
80
60
40
20
0
H.P. Crank End
100
80
60
40
20
0
L.P. Head End
15
10
5
0
5
10
L.P. Crank End
15
10
5
0
5
10

ENGINE No. 80.

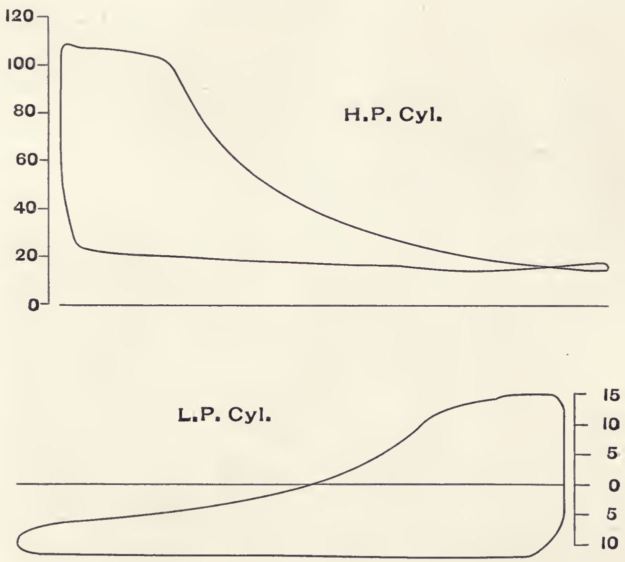
Four valve (Corliss) tandem compound, 18″ and 30″ x 48″. Speed, 63 revolutions per minute.

The low-pressure cylinder of this engine was operated by double eccentrics. The diagrams here given show the effect produced by advancing the eccentric which drives the exhaust valves of the low-pressure cylinder $3\frac{1}{2}$″on the 12″ shaft. At the same time the exhaust rod on the high-pressure cylinder was lengthened 2 threads, so as to give greater compression. The diagrams were taken at the head end of both cylinders, No. 80*a* before, and No. 80*b* after, the adjustments.

ENGINE No. 80a



ENGINE No. 80b

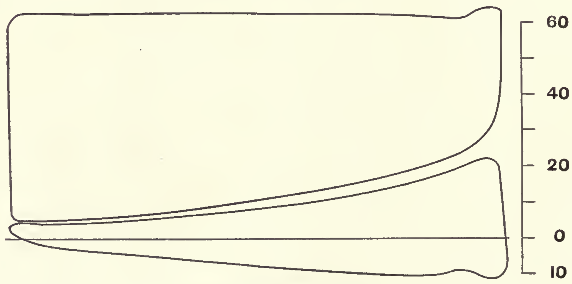


ENGINE No. 81.

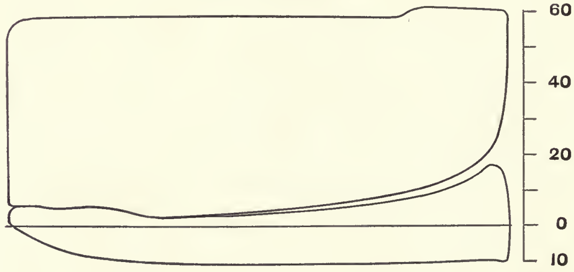
Compound duplex direct acting pumping engine.

These diagrams show the effect of increasing the throw of the valves (which are slide valves), thereby giving the engine the benefit of wider opening of ports. The improvement is shown mainly in the increased effect of the vacuum in the low-pressure cylinder. The effect of the change on the duty performed by the pump was marked, and the consumption of coal was much reduced. No. 81*a* was taken before, and No. 81*b* after, the change.

ENGINE No. 81a



ENGINE No. 81b

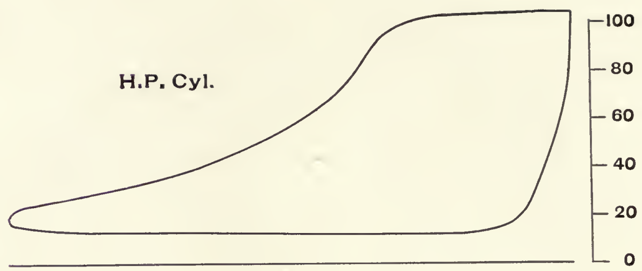


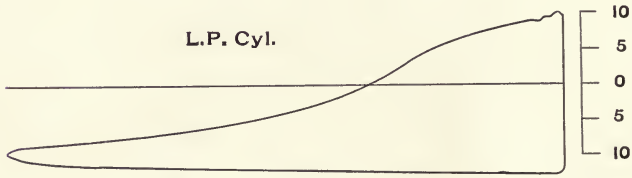
ENGINE No. 82.

Four valve cross compound, 24″ and 46″ x 48″. Speed, 75 revolutions per minute.

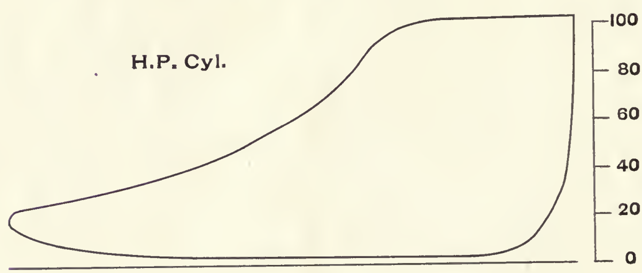
The high-pressure cylinder of this engine is of the four-valve type. The low-pressure cylinder has slide valves with cut-off adjustable by hand. These diagrams show the effect upon the distribution of the load between the cylinders produced by changing the cut-off in the low-pressure cylinder. In one case, No. 82*a*, it was set at the $\frac{1}{4}$ mark, and the other, No. 82*b*, at the $\frac{5}{8}$ mark. In the former the power developed by the high-pressure cylinder was 408 H. P., and by the low-pressure cylinder 300 H. P., while in the latter the quantities were respectively 480 and 222. The diagrams are from the crank end.

ENGINE No. 82a





ENGINE No. 82b



L.P. Cyl.

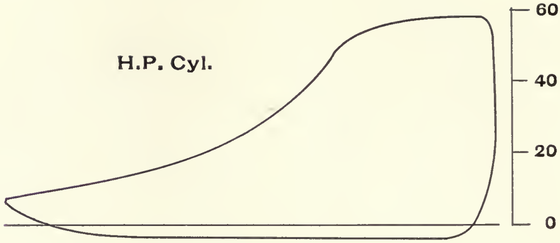
ENGINE No. 83.

Canadian cross-compound engine, 20″ and 36″ x 42″. Speed, 76 revolutions per minute.

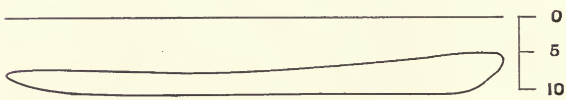
The high-pressure cylinder in this engine has Corliss valves and the usual automatic cut-off. The low-pressure cylinder has a plain slide valve with no means of adjusting the cut-off save by shifting the eccentric. These diagrams show the effect upon the distribution of the load between the cylinders produced by changing the low-pressure cut-off by the eccentric adjustment. When the steam followed in the low-pressure cylinder to nearly full stroke No. 83*a*, the power developed in the high-pressure cylinder was 167 H. P., and in the low-pressure cylinder, 60 H. P. When the eccentric was advanced in the low-pressure cylinder, No. 83*b*, these quantities became respectively 149 H. P. and 80 H. P.

ENGINE No. 83a

H.P. Cyl.

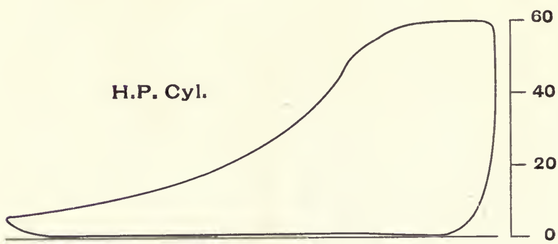


L.P. Cyl.

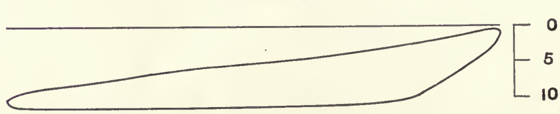


ENGINE No. 83b

H.P. Cyl.



L.P. Cyl.

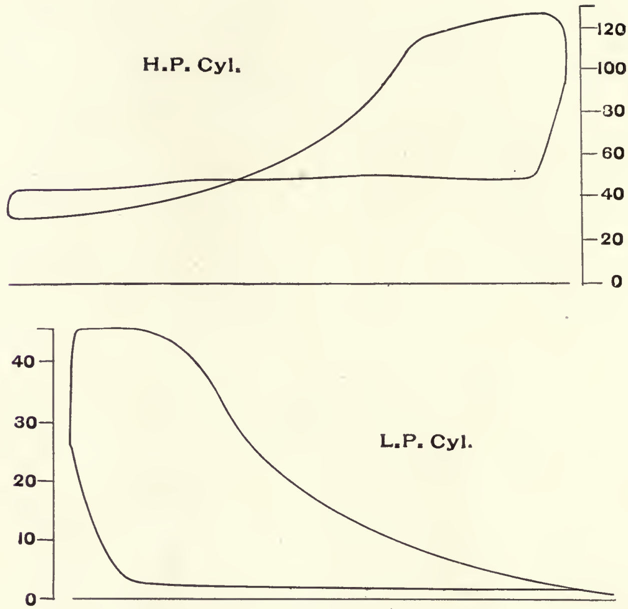


ENGINE No. 84.

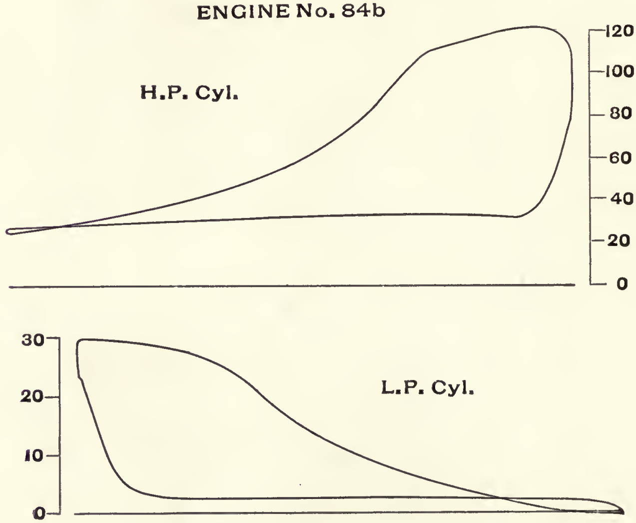
Four-valve cross compound, $17\frac{1}{2}''$ and $28'' \times 48''$. Speed, 100 revolutions per minute.

This engine is a non-condensing compound. The valves are all slide valves, and the steam and exhaust are operated by independent eccentrics. The diagrams show the effect produced by changing the cut-off on the low-pressure cylinder, and thereby the pressure in the intermediate receiver. Diagram 84 *a* was taken with a receiver pressure of 44 lbs., and diagram 84 *b* with a receiver pressure of $27\frac{1}{2}$ lbs. They are all from the crank end.

ENGINE No. 84a



ENGINE No. 84b

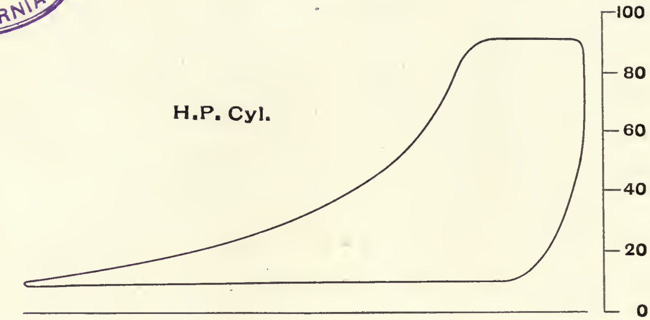


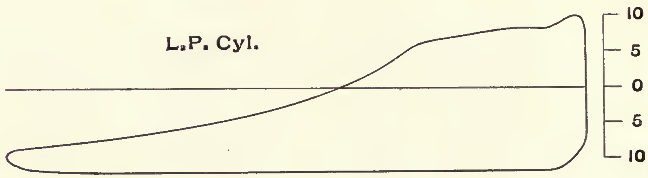
ENGINE No. 85.

Four-valve (Corliss) cross-compound, 24″ and 34″ x 48″. Speed, 61 revolutions per minute.

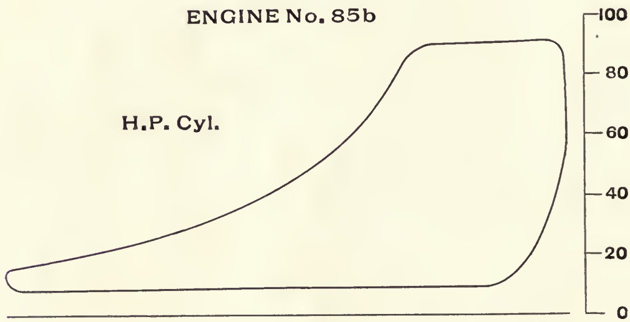
In this engine the governor operated on the cut-off of the high-pressure cylinder. The cut-off of the low-pressure cylinder was under the control of a pressure regulator set so as to maintain a constant pressure in the receiver, irrespective of the load or other conditions. Steam was withdrawn from this receiver for heating purposes; in this case for the heating of feed-water for the plant of boilers which supplied the engine, and the diagrams given were taken under two conditions of running; first, No. 85 *a*, when all the steam was used for power, and second, No. 85 *b*, when the steam was withdrawn from the receiver as noted.

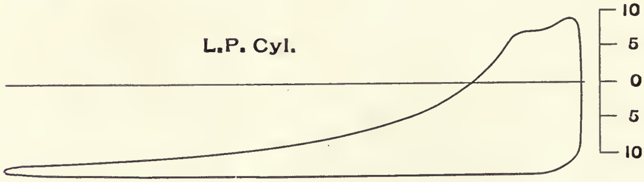
ENGINE No. 85a





ENGINE No. 85b



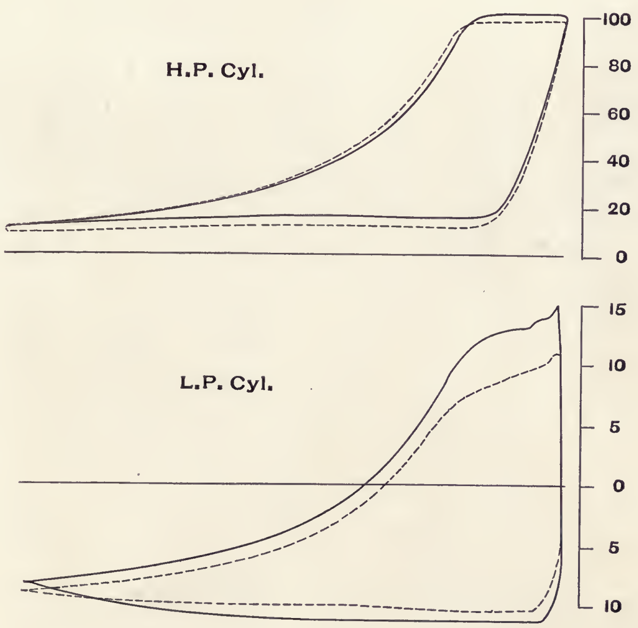


ENGINE No. 86.

Four-valve cross compound, 20" and 36″ x 48″. Speed, 65 revolutions per minute.

The condenser of this engine is the siphon type, and the injection water is supplied by an independent direct-acting steam pump. This is arranged so as to exhaust into the receiver or into the condenser, as desired. The diagrams given were taken under both of these conditions of running the pump; those with the full lines being taken when the pump was exhausting into the receiver, and those with dotted lines when the same was turned into the condenser. The comparatively poor vacuum in the latter is due to the air leakage through the packing around the valve stems and piston rod of the pump.

ENGINE No. 86



ENGINE No. 87.

Single-valve cross compound, 15″ and 23″ x 15″. Speed, 260 revolutions per minute.

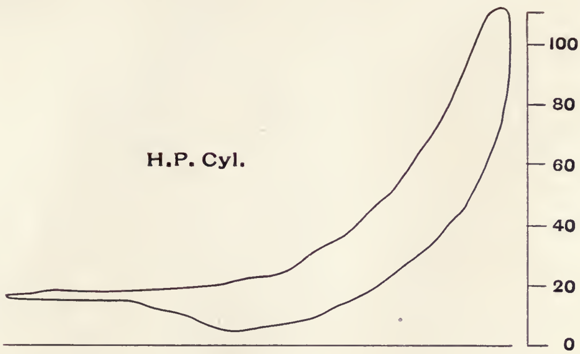
This engine has unpacked piston valves, one for each cylinder, with a shaft governor operating on the high-pressure valve.

These diagrams are given to show the effect of a break in the casting of the high-pressure steam-chest, which allowed steam to pass directly into the low-pressure chest without going through the high-pressure cylinder. The two cylinders and the chest were all made in one casting. Diagrams No. 87*a* and 87*b* were taken with a load of 79.5 I. H. P., and No. 87*c* and 87*d* with a load of 131.1, I. H. P. In the former the low-pressure cylinder developed 155.2 H. P., and in the latter 141.3 H. P. In the former the high-pressure cylinder produced a *resistance* or negative power equivalent to 75.7 horse-power, while in the latter this was reduced to 10.2 horse-power. The difference in these quantities gives the respective horse-powers as stated. In diagram 87*a* the upper line, which ordinarily is the steam and expansion line, is here the compression line, and the lower line is the one that is made during the admission, expansion, and release. The boiler pressure here is 75 lbs., and the compression of the exhaust carries the back pressure up to 120 lbs. The point of cut-off takes place at the very beginning of the stroke, and evidently there is no steam admitted save that which comes from the compression of the exhaust.

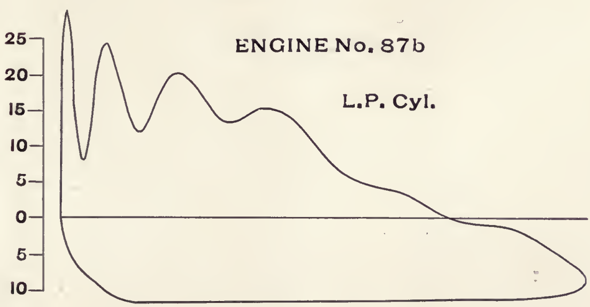
Diagrams 87*e* and 87*f* were taken from an engine of the same size and make, in which there was no defect such as that mentioned; and a comparison with these will show the effect produced by the disordered condition. In these diagrams the high-pressure cylinder developed 71.4 I. H. P., and the low-pressure 53.3, making a total for the engine of 124.7 I. H. P. This is about the same power as that shown by diagrams 87*c* and 87*d*.

These diagrams are all from the crank ends.

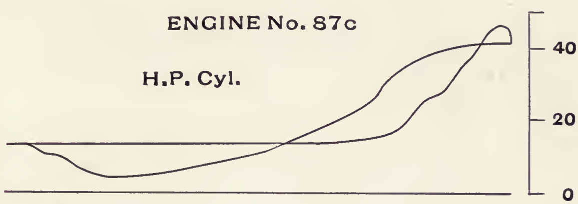
ENGINE No. 87a



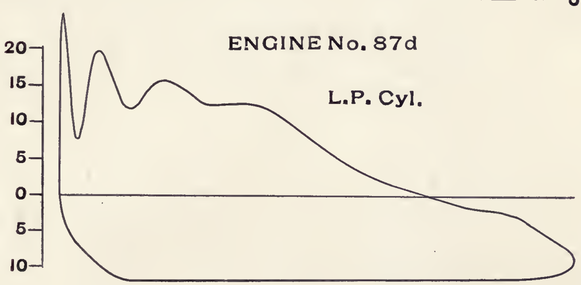
ENGINE No. 87b



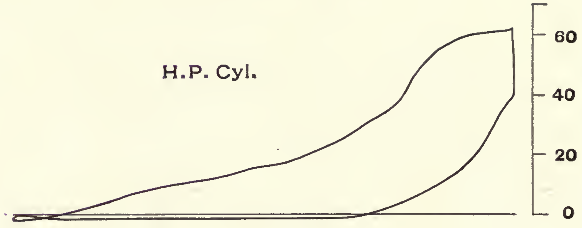
ENGINE No. 87c



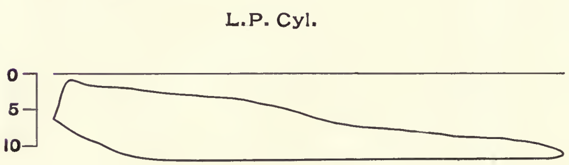
ENGINE No. 87d



ENGINE No. 87e



ENGINE No. 87f

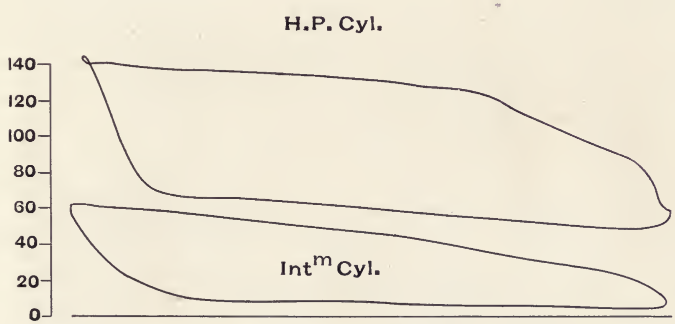


ENGINE No. 88.

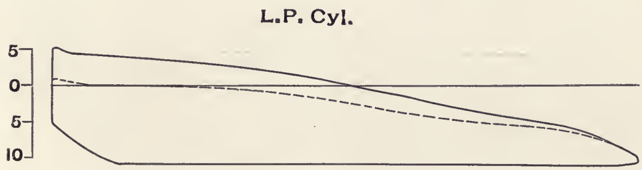
Marine triple expansion engine, 15″, 23″, and 40″ x 30″. Speed, 83½ revolutions per minute.

These diagrams show the effect produced by leakage of the low-pressure piston. The dotted line on the low-pressure diagram is the one taken when the leakage was going on, and the full line the one taken with a tight piston. The diagrams 88*a* from the intermediate and high-pressure cylinders are those taken with the tight engine. The effect of stopping the leakage, which was due to the weakness of the springs under the packing-rings, was to raise the pressure in the receiver. The increase was 4 lbs. Another effect was to increase the speed of the engine when running at full capacity from 81 revolutions per minute to 84. Still another effect was to increase the power developed from 410 I. H. P. to 442 I. H. P.

ENGINE No. 88a



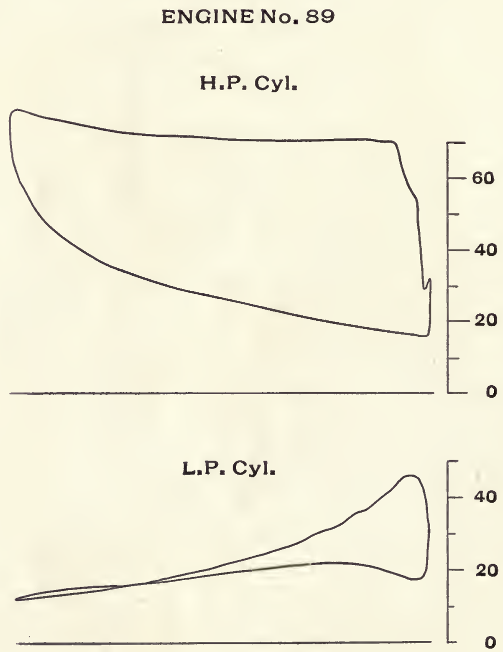
ENGINE No. 88b



ENGINE No. 89.

Compound high-speed non-condensing engine, 6″ and 12″ x 12″. Speed, 201 revolutions per minute.

These diagrams are given simply as curiosities. The high-pressure cylinder is doing nearly all the work, and the conditions under which the steam is distributed are about as wasteful as could occur. There is no cut-off in the high-pressure cylinder; the terminal pressure is the highest of any part of the diagram, the release is late, and the back pressure on the low-pressure diagram is excessive.



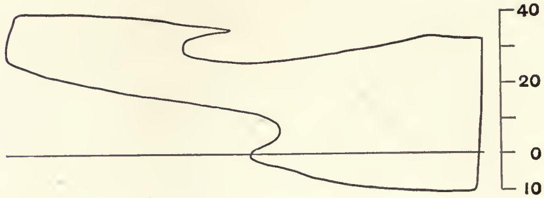
ENGINE No. 90.

Diagrams 90 *a* and 90 *b* are introduced partly as curiosities and partly to show the general features of diagrams obtained from a steam-driven air-pump operating an independent condenser. Here the cylinder was 10″ x 10″. Diagram 90 *a* was taken when the pump exhausted into the condenser, and diagram 90 *b* when it exhausted into the atmosphere. The peculiarity of these diagrams lies in the fact that the pump takes steam at full stroke, exhausts at a higher pressure than the pressure of admission; and the return stroke is made, for a portion at least, under the wasteful conditions of a very high back pressure. Another curiosity is the stopping of the piston at about the middle of the stroke, and the rebounding of the same before it proceeds on its course.

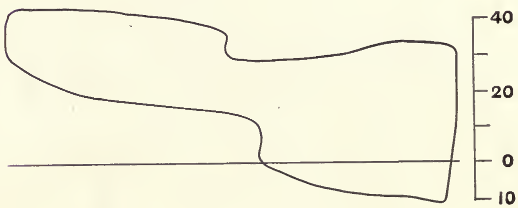
When this pump was running non-condensing the exhaust steam was measured by collecting and condensing it in a barrel of water. It was found to use 717 lbs. of steam per hour, at a speed of 61.2 double strokes per minute, or 103.3 lbs. of steam per I. H. P. per hour, the power developed being 6.94 H. P. This performance represents, as might be expected, a very wasteful use of steam; but it should be stated that in a plant properly arranged the heat of the steam can be utilized in warming the feed-water, and the loss is reduced to a comparatively small quantity.

ENGINE No. 90a

Head End



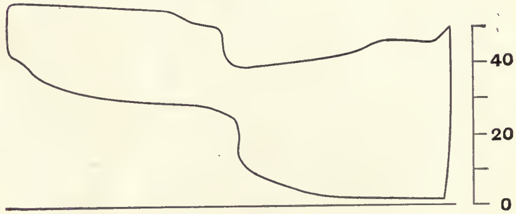
Crank End



ENGINE No. 90b

Head End

Crank End



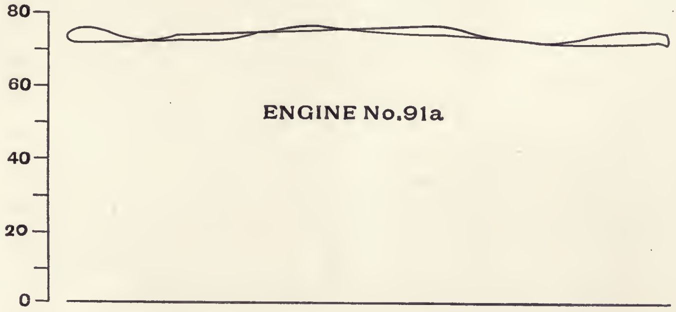
STEAM-PIPE DIAGRAMS.

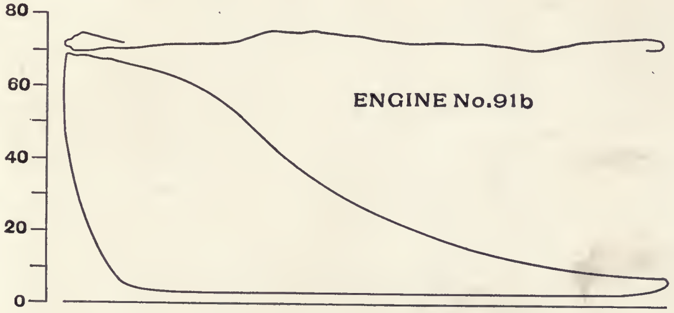
STEAM-PIPE DIAGRAMS.

THE effect which a running-engine has upon the pressure in the steam pipe, as shown by an indicator diagram taken from the pipe, is a matter which not only possesses interest from an engineering point of view, but it has a bearing on an important question relating to steam-pipe design. The fluctuations of pressure in the pipe caused by the intermittent flow of steam into an automatic cut-off engine is sufficient to set up vibrations in the pipe; and these extend from the engine through the whole distance back to the boiler unless the pipe is well anchored, and sometimes in spite of what appears to be good anchorage. When we consider the relatively small weight of the substance which is traveling through the pipe, it is difficult to realize the powerful effect which these fluctuations have upon its stability. It is not, however, the substance itself which is the potent factor in the matter, but the effect of the unbalanced pressure acting between the two ends of a section of pipe produced by the sudden and intermittent reduction of pressure at the end nearest the engine. If the reduction is 10 lbs. and the diameter of the pipe is 8″, there is an unbalanced pressure of 10 lbs. per square inch upon an area of about 50 square inches, or a total force of 500 lbs. acting in the direction of the length of the section. Such a force would have in a measure the effect of a 500 lb. blow upon the pipe, which, of course, is a serious matter. These fluctuations can be overcome to some extent by avoiding short right-angle elbows, and employing long-turn bends in their place. They can be overcome more effectually by introducing in the steam pipe as near as possible to the engine a reservoir having considerable volume relative to the size of the cylinder, and passing the steam through the large space thus provided. The fluctuations will then occur mainly in that part of the pipe which lies between the reservoir and the cylinder, and the reservoir serves to prevent them to a large extent from extending back to the boiler. The steam-pipe diagrams here given show the desirability of employing some means for reducing the extent of these fluctuations, and in one instance the beneficial effect of a reservoir is clearly revealed.

ENGINE No. 91.

Diagrams 91*a* and 91*b* were taken from a 9″ steam pipe supplying a 28″ x 48″ Corliss non-condensing engine running at a speed of 100 revolutions per minute. The pipe is a branch from a long 12″ pipe leading to the boilers, and its length measured from the 12″ is about 30 feet. The pipe contains 6 short right-angle elbows. Diagram 91*a* is complete for the entire revolution of the engine, and reveals the pulsations produced by the admission at both ends of the cylinder. Diagram 91*b* relates to one stroke. The indicator diagram from the cylinder taken on the same stroke is also shown.



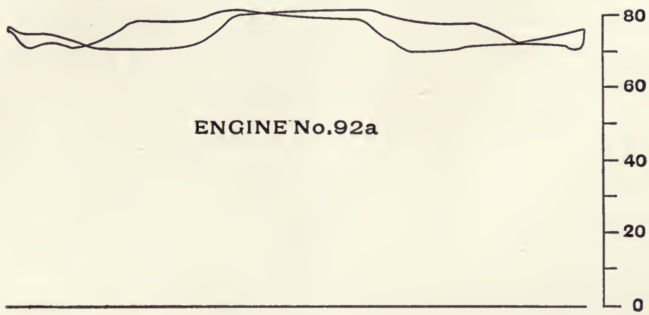


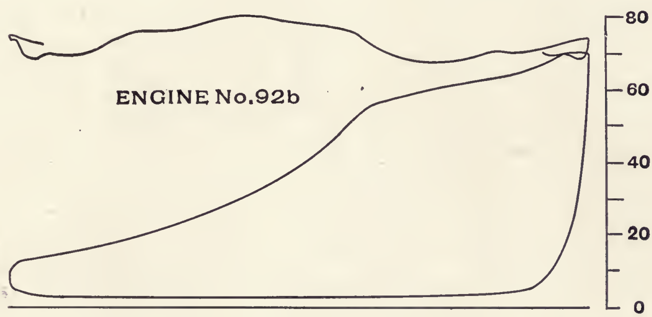
It will be seen that just before the beginning of the stroke the pressure in the steam pipe drops; and it is maintained nearly constant until the cut-off takes place, when it immediately rises.

Afterwards the pressure gradually falls, and a short time before the opposite end of the stroke is reached it rises again. Subsequently when the very end is reached it falls abruptly, coincident with the admission of steam to the other end of the cylinder.

ENGINE No. 92.

Diagrams 92*a* and 92*b* are from an 8″ steam pipe supplying a 23″ x 60″ non-condensing Corliss engine running at a speed of 75 revolutions per minute. The pipe is 82 feet in length measured from the nearest boiler to the throttle valve, and it contains 5 short right-angle elbows. Diagram 92*a* applies to a complete revolution, and 92*b* simply to the forward stroke taken while the piston was moving from





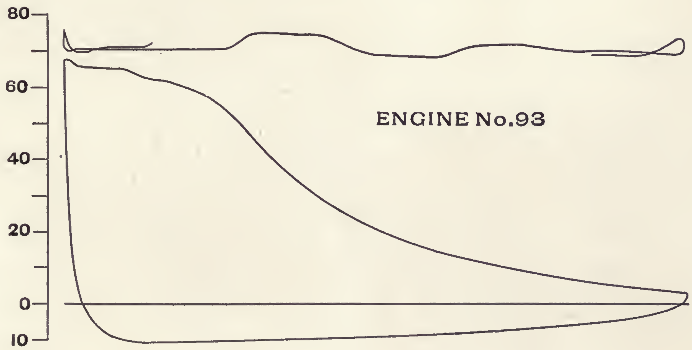
the head end of the cylinder. The indicator diagram from the head end of the cylinder is also given. Referring to the latter, it appears that just prior to the beginning of the stroke the pressure rises in the steam pipe. Coincident with the movement of the piston forward during the admission, the pressure in the pipe gradually falls up to the point of cut-off, and when this occurs it rises to a point some 10 lbs. above the line of average pressure. At a point just beyond the middle of the stroke the pressure gradually falls, until just before the end of the forward stroke it suddenly rises again preparatory to the beginning of the return stroke.

This diagram is a curiosity for the reason that the pressure rises at the very beginning of the stroke, when presumably the cylinder is taking steam, whereas under the ordinary circumstances it would be expected that the operation would be reversed and the pressure would fall. The probability is that owing to the compression of the exhaust steam into the clearance space the quantity of live steam admitted is very small.

Another curiosity in this diagram is the fall of pressure from the middle to nearly the end of the stroke. During this period there is no steam being drawn out of the pipe, and the only explanation of this action is the assumption of a sort of rebounding of the steam within the pipe due to the intermittent character of the flow. In this matter as well as in the conformation of the diagram throughout, there are many points which, to say the least, are obscure.

ENGINE No. 93.

This diagram is from a 7″ pipe supplying a Corliss condensing engine, 32″ x 54″, making 47 revolutions per minute. The engine diagram given is from the crank end of the cylinder, and the steam-pipe diagram refers to one stroke of the piston, that is, the one made from the crank end to the front end. This engine was one cylinder of a pair, and the steam pipe consisted of a 10″ main leading from the boilers and a 7″ branch to each cylinder. The distance from the 10″ to the throttle valve was 20 feet, and it contained two right-angle elbows. The other cylinder was in operation when the diagram was being taken.

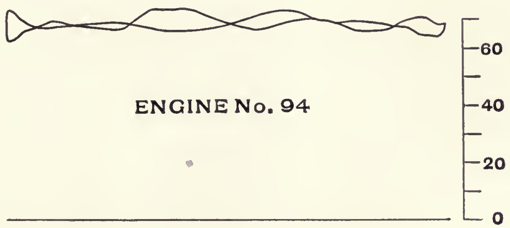


On the steam-pipe diagram it appears that the pressure rises just before the beginning of the stroke, and immediately after it drops back to nearly the same point, and remains nearly constant until the steam is cut off from the cylinder, when it rises. Just before the middle of the stroke the pressure falls again, this action being due presumably to the other cylinder taking steam, followed by another rise in the pressure at about the time of the cut-off in the other cylinder. Just prior to the beginning of the return stroke the pressure rises as before, and again drops soon after the beginning of the return stroke, when the other end of the cylinder begins to take steam.

Here is another curiosity. At the very beginning of the steam-pipe diagram the pressure increases in a marked degree at the time when apparently the cylinder begins to take steam, and then immediately drops back. The reason for this action is difficult of explanation.

ENGINE No. 94.

Diagram No. 94 is from the steam pipe of the right-hand cylinder of a pair of double-valve engines, 17″ x 24″, running at a speed of 154 revolutions per minute.

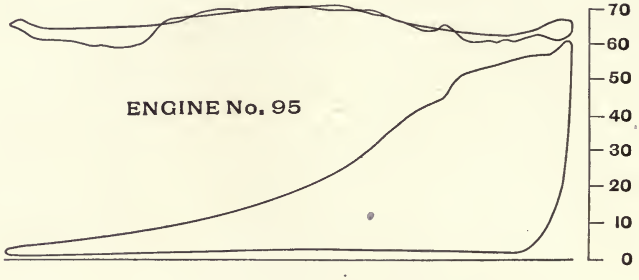


The main pipe here was 140 feet in length, 10″ in diameter, and had 3 short right-angle elbows. The branch pipe for the two cylinders were each 6″ in diameter and 8′ in length, and each contained 2 right-angle elbows. This is the same engine as that referred to as No. 10 in the section on Feed-Water Tests, but it was taken with an indicator having a different scale from the diagrams given in connection with the results of those tests.

It will be seen in this diagram that the effect of the closing of the valves at the points of cut-off are clearly revealed, but that in other respects the various operations are not clearly defined. Considering that the reciprocations are somewhat rapid, and that the diagram shows the effect of the fluctuations produced by both cylinders, it is difficult to make a close study of its various features.

ENGINE No. 95.

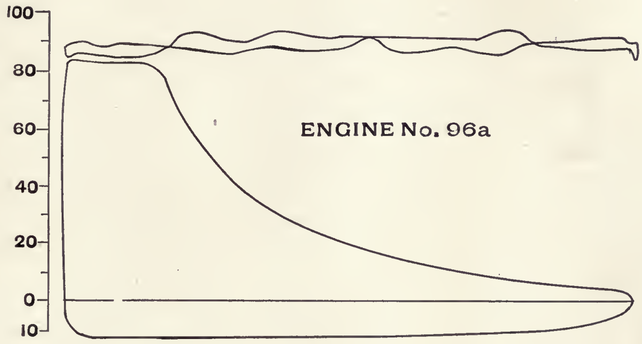
Diagram No. 95 is from the steam pipe of a 20″ x 50″ four-valve engine making 65 revolutions per minute.

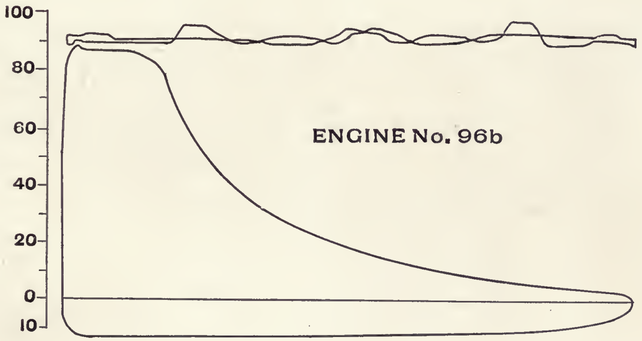


The pipe is 5″ in diameter and 36′ long, and it contains 4 short right-angle elbows. This is the same engine as the one numbered 76 under the head of Valve Setting. The features in this diagram conform in the main to what would be expected from the known operations of the steam. The pressure drops at the beginning of the stroke, and rises at the point of cut-off; and when the opposite end of the stroke is reached it drops again, coincident with the opening of the steam valve, and rises again when the cut-off at the other end takes place. One feature here is noticeable; and that is, that the effect of the subsequent admission after the cut-off, which is shown on the diagram taken from the cylinder, the same as in No. 76, is clearly revealed on the steam-pipe diagram, where there is a second fall of pressure just beyond the point where the rise occurs due to the regular cut-off. The fall of pressure commencing at the middle of the stroke and continuing to near the end is a feature of this diagram the same as in some of the preceding ones which have been referred to, though here it takes place more gradually than in some.

ENGINE No. 96.

Diagrams 96*a* and 96*b* are from the head end of a Corliss condensing engine, 20″ x 48″ running at a speed of 60 revolutions per minute.

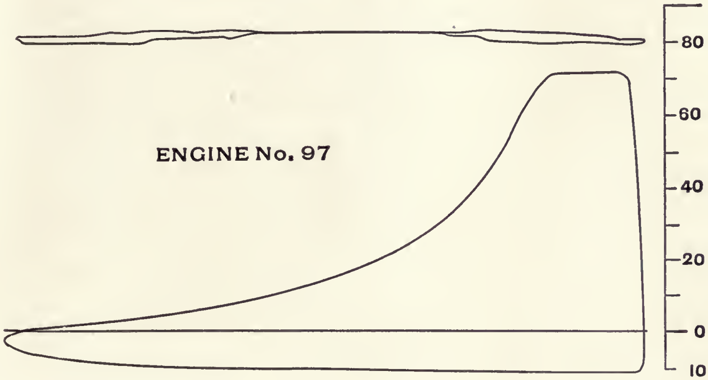




This engine is one of a pair; but when these diagrams were taken, the second cylinder was out of use, and the throttle valve closed. The main pipe here is 10″ in diameter and 33′ in length. The branches are 6″ in diameter, and the one leading to the left-hand cylinder is 10′ in length, and that to the right-hand cylinder 15′ in length. Each of these branches has two short right-angle elbows. The diagrams were taken from the left-hand cylinder. Diagram 96*a* was taken when the steam was passing through the pipe above referred to. When diagram 96*b* was taken the 10″ pipe was shut off at the boiler end, and steam was furnished through twenty-five feet of 8″ pipe and one 45 degree elbow into a tee at the boiler end of the 10″ main. In both these diagrams the admission of steam is accompanied by a drop of pressure in the pipe, as would be expected, and a corresponding rise of pressure at the point of cut-off. In diagram 96*a* the pressure falls again very quickly after cut-off; and a succession of wavy lines occur until the middle of the stroke, and then the pressure is nearly constant to the end. In diagram 96*b*, on the contrary, the fall of pressure just after the cut-off is much less marked, and there is considerable more rise in pressure as the end of the stroke is approached. The only difference in the conditions under which these diagrams were obtained was in the lengthened pipe through which the steam passed. It would seem, therefore, that the arrangement of the pipe has much to do with the character of the fluctuations. It will be noticed also in these diagrams that the fluctuations resemble in some respects those which occur on previous diagrams taken from a pair of engines with both cylinders running. In this case, however, only one cylinder was in operation. Here is another indication that the arrangement of the pipe has much more effect upon the character of the fluctuations than would at first be supposed.

ENGINE No. 97.

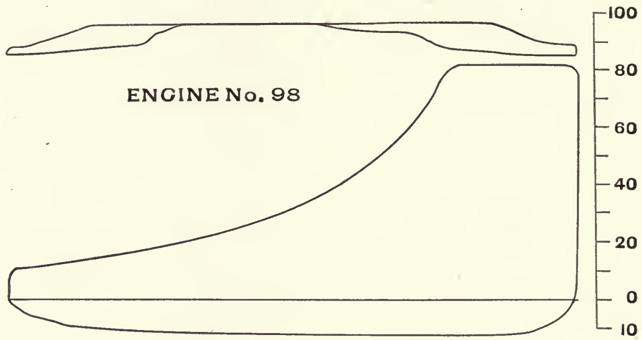
Diagram No. 97 is from a 10″ steam pipe supplying a 30″ x 72″ Corliss engine, making 60 revolutions per minute.



The steam pipe is 108 feet in length from the main header in the boiler-room, and it contains five short right-angle elbows. The fluctuations of pressure here are of much less extent than in any of the preceding diagrams, due in part probably to the relatively light load on the engine. In view of what the preceding diagrams have shown, the real cause of so little variation may be some peculiar arrangement of the pipes which acted favorably.

ENGINE No. 98.

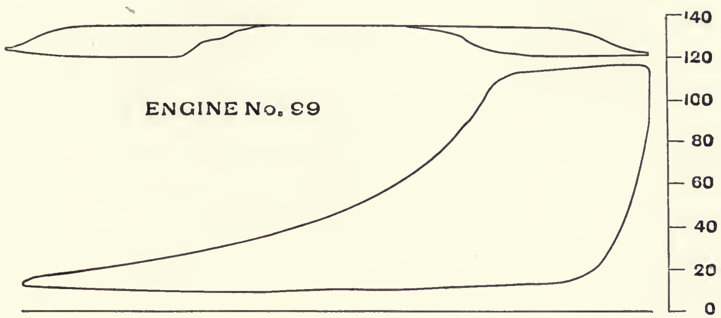
Diagram No. 98 is taken from an 8″ steam pipe supplying a 24″ x 48″ Corliss engine running at a speed of 62 revolutions per minute.



The pipe is 135 feet in length, and contains 5 short right-angle elbows and two 45 degree elbows. The lines in this diagram are very clearly marked. There is a sudden drop in the pressure just at the beginning of the stroke, and there is a marked rise of pressure at the point of cut-off. There seems to be little variation of pressure after this time until nearly the end of the stroke. During the very last part of the stroke, however, the pressure drops the same as noticed in many of the preceding diagrams, although there appears to be no action in the working of the steam in the cylinder that should cause it. This is one of the things that makes the reasons for the particular conformation of steam-pipe diagrams obscure.

ENGINE No. 99.

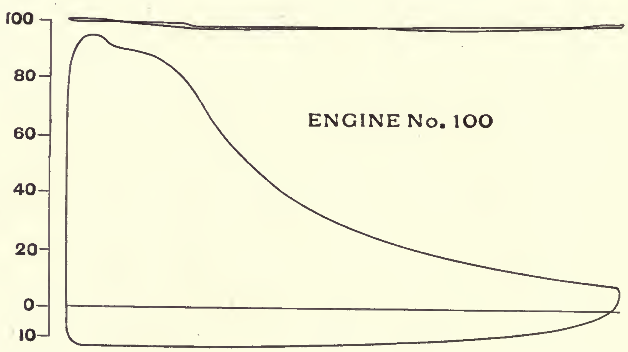
Diagram No. 99 was taken from a 6″ pipe supplying a 14″ and 26″ x 42″ compound engine running at a speed of 100 revolutions per minute.



The length of the pipe was 75 feet, and it contained two short right-angle elbows. Here is another case where the lines of the diagram are clearly marked, and there can be no misconception in regard to the action going on in the pipe. This diagram is similar to the one which precedes it, and has the same general features. There is this peculiarity, however, that there is practically no drop of pressure during the period of admission. The drop occurs toward the very end of the previous stroke. Subsequent to the cut-off and prior to this drop, the pressure is well nigh constant. The indicator diagram here given is from the head end, and refers simply to the high-pressure cylinder.

ENGINE No. 100.

Diagram No. 100 refers to a case where a reservoir was installed in the steam pipe close to the cylinder, and the diagram was taken from this reservoir. The engine is a Corliss 30″ x 48″, running at a speed of 80 revolutions per minute. The receiver is supplied from an 8″ pipe 223 feet in length, which contained six short right-angle bends, while the engine is supplied from the reservoir through a 10″ pipe 12′ long, containing two short right-angle elbows. The size of the reservoir is 42″ in diameter and 8′ in height.



In this diagram the fluctuations of pressure do not seem to follow the admission and cutting off of the steam to any great extent, and at the worst they are confined within narrow limits. The extreme change of pressure from the highest to the lowest is three pounds. Comparing this with the previous instance, No. 99, the difference is exceedingly marked. There the change of pressure was some thirteen pounds. If we investigate these two cases carefully it will be found that the rate of flow of steam from the boiler to the reservoir is forty-three feet per second, and in the other case (No. 99) the rate of flow close to the throttle valve was twenty-eight feet per second. The conditions as to the speed of the steam and the quantity withdrawn per stroke with reference to the size of the pipe was therefore much more severe in the case where the reservoir was used. It thus appears that with the same conditions of service, the favorable effect produced by the reservoir would have been even greater than that here indicated.