GOVERNORS AND THE GOVERNING OF PRIME MOVERS

BY

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PREFACE

GOVERNORS have been the play toy of inventors for over a century, and have been the hobby of mathematicians for over thirty years. In spite of these facts, the knowledge of governors and of governing among both designing and operating engineers is very incomplete. There are several reasons for this condition. One of them is that governors do not cover a wide enough field to warrant a separate course in any engineering school. Instruction in governors is given in a scattered fashion. In courses in steam engineering, governors for steam engines and turbines are taken up. In courses on hydraulic motors, some time is spent on governors and governing; and the same is true of governors for internal combustion engines. Textbooks on prime movers reveal the same condition of affairs. Everywhere that which is apparent on the surface is reprinted, but nowhere (with very few exceptions) does the investigation go below the surface.

The present book aims to fill this gap. As far as I am aware, there exists to-day no other book of any consequence on governors and governing in the English language. There are books in French, German, and Swedish, but they are of little use to English-speaking engineers.

The present volume is a book of essentials and principles. Practice changes; there are fashions in engineering almost as changeable as those in women's clothes; but engineering principles do not change. For that reason, I have tried to dig out that which is essential, and to present it in such a manner that the reader is put in a position to judge existing and future types of governors as well as the properties of prime movers with regard to regulation. In consequence, the book contains not a single catalogue picture. Every drawing was especially prepared to show in a diagrammatic manner that which is important, with the intentional omission of everything else.

The book contains more than I give my students in the classroom. Students, as a rule, do not know what their lifework will be in later years. While in school, they can be given only the "meat of the essentials." They are, however, anxious to know of a source out of which they can fill the gaps which their necessarily brief instruction at school has left. Nevertheless, the book by no means covers the whole subject of governing of prime movers. Many extremely interesting subjects were omitted, for two reasons: First, it was necessary to keep the price of the book within such limits as not to make its purchase a burden. Second, it was desirable to restrict the mathematical side of the book to a level which undergraduates can master, so that the reader may be spared the troubles of intellectual indigestion.

It is, however, planned to follow this book, later on, with another one on the subjects of "dynamics and design of modern governors for prime movers," which will be primarily intended for those engineers who make governors and governing a life study. In that book many subjects can be taken up which are clearly beyond the limits of the present volume. The footnotes referring back to the preface indicate some of the subjects which are to be taken up in that volume. In addition, the other book will take up the effect of water inertia in hydraulic turbines and the governing of prime movers driving alternators in parallel. The present book was started in the spring of 1913. Lack of spare time and necessity for original work in the preparation of this book have delayed its completion until now. By the term "original work" I mean that practically all of the theories advanced by me were tried out practically. Some of the tests were made in the Mechanical Engineering Laboratory of the Carnegie Institute of Technology, but most of the trials were made in the field, in the attempt to help those who had trouble with regulation.

College graduates will have no trouble with the mathematical part. Operating engineers who have not had the good fortune of college training should likewise have no trouble, if they turn to the elementary derivations given in the appendix. By comparison, they will realize that the differential calculus is a wonderful short-cut.

Like other books dealing with a specialty of applied mechanics, the present book must contain, for some readers, much that is known, and must occasionally pass over the heads of others. It is impossible to pitch the scale right for everybody.

The bookmark, containing all symbols, will, I hope, be of assistance.

For those who wish to go more deeply into the subject of governing, and who have library facilities, the bibliography will be valuable. It was prepared by the Carnegie Library of Pittsburgh. The librarians worked faithfully on this task. I herewith thank them for their coöperation.

W. Trinks

Pittsburgh, May, 1919.

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LIST OF SYMBOLS

To accompany

"Governors and the Governing of Prime Movers."

By Prof. W. Trinks

A....Governor travel, or displacement.
B....Distance.
C....Centrifugal force.
D....Small difference.
E....Energy.
F....Force at collar to overcome governor friction.
G....Area.
H....Radius of gyration.
I....Moment of inertia } mass ×
J....Moment of inertia } (radius)²
K....Constant.
L....Lever arm.
M....Moment.
P....Strength of governor; *also* Restoring force per unit displacement.
Q....Weight or force.
R....Force on governor collar to overcome valve friction only.
S....Spring force.
T....Special time.
T_b....Brake-resistance traversing time.
T_f....Time required to fill volume of container. (In pressure governing.)
T_g....Traversing time of governor.
T_i....Starting time of inertia mass.
T_n....Time of one complete vibration of a governor. (Natural period of vibration.)
T_r....Relay traversing time.
T_s....Starting time of prime mover.
U....Relative speed deviation.
V....Volume; *also* Volume flowing in unit time.
W....Weight.
X....Relative governor deviation.
Z....Relative load change.
a....Angular acceleration.
b....Radial travel of centrifugal weights.
d....Differential.
e....Base of Naperian logarithms; *also* Efficiency.
f....Friction coefficient.
g....Gravity acceleration.
h....Height or space dimensions.
i....Angle.
j....Angle.
k....Angle.
l....Length.
m....Mass.
n....Revolutions per minute.
p....Static fluctuation; *also* Pressure.
q....Detention by governor friction.
r....Radius.
s....Space.
t....Time.
u....Angular velocity.
v....Linear velocity.
w....Angular velocity of auxiliary vector.
x....Abscissa; *also* Unknown quantity.
y....Ordinate.
z....Stability; *also* Root of an equation.

O.K. BY-A.S.

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GOVERNORS

AND THE

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INTRODUCTION

THE title of the present book, viz. "Governors, and the Governing of Prime Movers," clearly indicates that the study of governing comprises two distinct parts, one being a treatment of the governor as a mechanism, and the other being an investigation of the interaction between the governor and the prime mover.

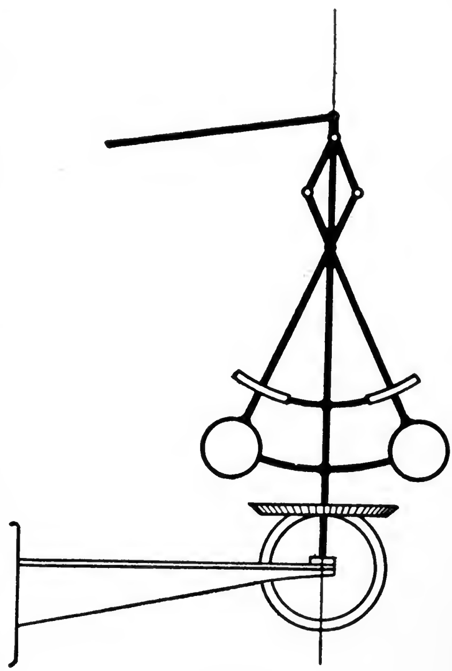


FIG. 1

That this division is not only logical but is also historical will be realized from a brief reciting of the history of governing for constant speed. In the historical sketch which now follows, many of the dates are approximate only; they must necessarily be so, because the development of correct engineering principles and the adoption of improved apparatus are very slow processes. Besides, the periods of development overlap in different countries.

Early mechanical engineers, that is to say the builders of water wheels, wind mills, and steam engines, studied governors as separate mechanisms and developed their theories with utter disregard of the reciprocal action of governor and prime mover. This condition existed, with very few exceptions, from the time of James Watt (who invented the centrifugal governor in the year 1784) to almost 1880. The Watt governor (Fig. 1) adjusts for higher and higher rotative speeds as the revolving balls move away from the axis of revolution. As might be expected, this property of the governor was considered a drawback, but not until the middle of the nineteenth century were any steps taken to remedy the defect. The search for the isochronous, or "astatic" governor, which would regulate for the same speed at all loads, began. Thus we find the "parabolic governor" which, when operated on a test block, is truly isochronous, introduced about 1850. For the same purpose governors with crossed arms were invented from 1860 to 1870 by different engineers (Kley, Farcot, Head) in different countries. With the same end in view, the high speed governor of Porter, Proell's inverted governor, and the oblong weight "Cosine" governor appeared in the two decades from 1860 to 1880. The same period gave birth to many other forms of governors, for instance the dynamometric (or load-) types (see Chapter X), which attempted to adjust the supply of energy directly by means of the demand for it.

All of these types of governor have disappeared for reasons explained in Chapter X. A new period dawned from about 1870 to about 1882, when governors were extensively applied to horizontal shafts (Porter's marine governor, Hartnell's crankshaft governor). Gravity could not, in these adaptations, furnish the controlling centripetal force, and springs had to be employed. The action of spring-loaded governors was so superior to that of weight-loaded governors that inventors placed many new types of spring-loaded governors on the market from 1880 to 1895 (about). The term "inventors" is used purposely instead of "engineers," because the vast majority of the designs of that period reveal inventive ability rather than good engineering judgment. From about 1890 on, the influence of electrical power generation helped to perfect mechanical governors in the same measure in which it has helped to perfect all types of mechanical equipment of power plants.

While inventors were busy bringing out new types of governors, usually without any knowledge of the mutual relations between prime mover and governor, these very relations were studied by two widely different types of men, namely the operating engineer, and the mathematician. The former found that the wonderful "isochronous" governors, such as the parabolic governor, the cosine governor, and many others, were worthless because they caused perpetual hunting and racing. Oil gag-pots were found to be necessary; and frequently stabilizing springs were concealed in them; but with all this practical progress, the theoretical foundation or proof for the exasperating behavior remained unknown to engineers, or else was not understood by them.

At the other end of the line, mathematicians studied the interaction between governor and prime mover. Starting about 1840 with the regulation of astronomical telescopes, British and French scientists gradually extended their investigations to the governing of prime movers. But their researches were of no benefit to the practical engineer, because they were published in the proceedings of astronomical or philosophic societies. More attention was paid to the very ingenious theoretical solution of the governing problem by the Russian engineer Wischnegradsky, which was published in 1876 and 1877 in Russian, French, and German. Although his publications are the foundation upon which modern theories of governing are built, they were scoffed at by engineers, partly because they failed to explain some of the plainly visible phenomena of regulation, and partly because they were clad in very mathematical form. Wischnegradsky's work, unfortunately, was given very little or no appreciation in English-speaking countries.

From about 1890 to the outbreak of the European war, French and German engineering literature (including Switzerland and Austria) have been very active on the mutual relations between governor and prime mover, while British and American engineers either took no interest in the theoretical development, or else were too busily engaged with practical experimenting to pay attention to theory. Of course there are exceptions, the most prominent of which is the large number of American contributions to the development of the theory of the action of long pipe lines on the governing of hydraulic turbines.

Even to-day, the subject of governing, that is to say, the interaction between the governor and prime mover, is very little understood by engineers. In the 1905 edition of the widely used English book on Dynamics by Routh, this statement is made: "The defect of a governor is, therefore, that it acts too quickly, and thus produces considerable oscillations of speed in the engine." A similar statement was made by Swinburne, the well-known English engineer, in "Industries," 1890. As a matter of fact, the principal progress in the improvement of regulation has consisted in making governors act more quickly. It is evident that there is need for clearing up this subject, if even the best of us can go astray on it. In consequence, considerable emphasis is placed on the problem of governing in the present book.

To-day the theories of governors as mechanisms and of governing have reached a certain stage of development beyond which it is scarcely necessary to go, for practical purposes. However, we must admit that present theory is by no means complete; on the contrary, it is far from it. But additional information can only be gained by a combination of experiment and mathematics which is at present beyond the reach of most practical engineers.

CHAPTER I

GENERAL STATEMENTS

1. Purpose of Governors. — Governors are used to automatically adapt the output of prime movers to the demand.

Governors are superfluous, whenever the torque resisting the motion of a prime mover increases considerably with the speed and depends solely upon the speed. This is the case in marine service, where the resistance grows approximately as the square of the speed, and in locomotive service. It is also the case in some classes of pumping machinery. If the resisting torque grows with the speed, and is not subject to variations except those caused by the speed of the prime mover, then the supply of energy can be adjusted by hand, because there can be only one speed for a given supply of energy. Hand adjustment is usually practiced in the cases mentioned, which means that under such circumstances the purpose of the governor is narrowed down to that of a safety speed-limit which prevents running away of the engine or turbine, if the resisting torque should be suddenly removed. If, however, the torque is repeatedly removed, as it occurs, when the propeller comes near the surface in a high sea, the governor approaches the more normal type mentioned below under (1).

Usually some factor in the operation of the prime mover is to be kept constant, or nearly constant, while other factors vary. Thus we find:

(1) Constant speed, variable torque.

(2) Constant (average per revolution) torque, variable speed.

(3) Constant quantity of one factor of the output, with variation of both speed and torque.

As examples of heading (1) may be cited engines or turbines operating electric generators, driving lineshafts, etc. In electrical power transmission, turning lights on or off, switching motors on and off, etc., varies the resisting torque, while the requirements of constant number of cycles and of constant voltage necessitate keeping the speed of the prime mover constant. Under (2) come pumping machines maintaining a constant pressure, but varying the output by adjustment of rotary speed. An example is found in air compressors supplying air to tools in mines. In spite of variable demand for air, the pressure must be kept constant. Under (3) come centrifugal pumping machines maintaining a constant flow of fluid against a variable pressure. The relation between pressure, speed, and torque is determined in this case by the characteristics of the prime mover and of the pump.

It is evident that purposes differing as widely as those mentioned under the three headings require different types of governing devices.

2. Forces Used in Governing. — Every governor performs two separate functions, namely:

(*a*) that of measuring the quantity which it is to keep practically constant,

(*b*) that of varying the supply of energy to the prime mover so that the just mentioned quantity is kept constant.

A governor, then, is both a measuring device and a motor. In conformity with the purposes mentioned in the preceding paragraph, governors measure

(1) angular velocity

(2) the intensity factor generated by the machine which is driven by the prime mover

(3) the extensity factor of this energy.

Under heading (2) come pressure and electromotive force. Under heading (3) come rate of flow and electric current.

In the evolution of the art of governing, many principles have been used, but one by one they were discarded (see Chapter X), until to-day practically one principle is left, namely this: A force is produced by the quantity to be measured; it is balanced by an external known force such as is derived from springs or weights. Any change in the quantity to be measured unbalances the system. As soon as the unbalanced force is strong enough to overcome resistance, it constitutes the motive force for adjusting the supply of energy. The principle in question has been pronounced defective and faulty, because, to cause the governor to act, it necessitates a change in the quantity to be kept constant. However, the change can be made exceedingly small; and the principle has the very great advantage that governing is accomplished regardless of the source of the disturbance. This important theorem will be fully explained later on.

If the just mentioned force acts directly on the mechanism which adjusts the supply of energy, the governor is said to be direct acting, or to be a "direct control" governor.

If the unbalanced force is too feeble to overcome the resistance of the energy controlling mechanism, it becomes necessary to call upon an auxiliary energy for motive power, which energy is released and controlled by the unbalanced governor force. If this is done, the governor is called a relay governor. Motor forces used in relay governing are most varied. The principal ones among them are:

(1) Mechanical force derived from the motion of the prime mover.

(2) Fluid pressure acting on a piston.

(3) Electromagnetic attraction.

All governors of the present period have certain general features in common, such as strength, work capacity, detention by friction, behavior in overcoming a passive resistance, promptness, resistibility and others. While, for that reason, it might be logical to discuss these properties in a general way, a different course will be followed for the sake of clearness. More than 90% of the total number of governors in operation are of the centrifugal type, so that it seems advisable to develop the general properties of governors with particular reference to the centrifugal governor. Transfer of the conclusions to other types of governors will not be difficult, once the theory of the centrifugal governor is clearly understood.

References to Bibliography at end of book: **35, 36, 53.**

CHAPTER II

THE DIRECT-CONTROL GOVERNOR AS A MOTOR

1. Strength of Centrifugal Governors. — When acting as a motor (that is, when shifting the power controlling mechanism to a new position), the governor does work. The latter

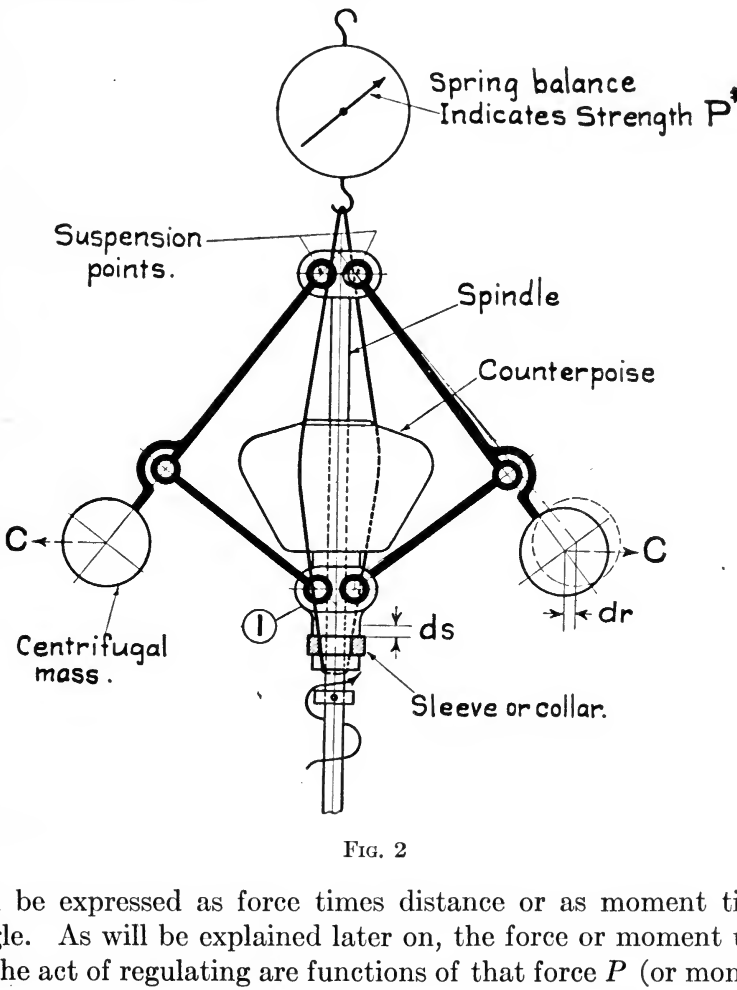


Fig. 2

can be expressed as force times distance or as moment times angle. As will be explained later on, the force or moment used in the act of regulating are functions of that force P (or moment

MP) which are required to move the governor, when not rotating, against centripetal force.

The place of action of this force (resp. moment) is shown in Figs. 2 and 4; the former represents a spindle type of governor which shifts the power controlling mechanism by up and down motion of the sleeve or collar (*1*); Fig. 4 represents a shaft governor which controls the supply of energy by shifting the center of the eccentric along the arc (*1*)–(*2*). The strength *P* can be determined experimentally by lifting up the sleeve of the governor and reading the force on a spring balance, as sketched in Fig. 2. The true strength is obtained, if friction is eliminated by vibration, for instance by rapping the governor with a wooden hammer. In German treatises on governors and in translations of such treatises the force *P* of the not-rotating governor is called the "energy" of the governor. This term is obviously misleading. It has been replaced in the present book by the term "strength."

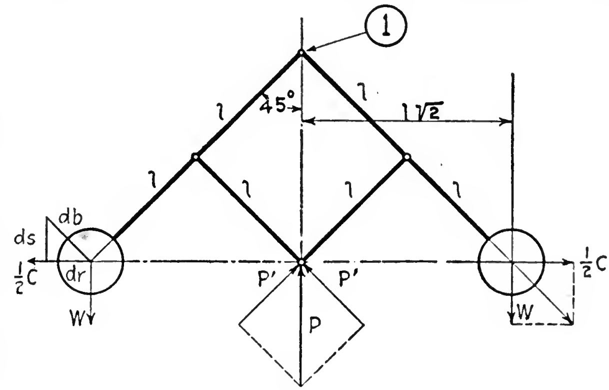


Fig. 3

The "strength" of the governor is closely related to that radial or centrifugal force 2*C*, Fig. 2, which is necessary to balance the frictionless governor, or to keep the governor sleeve floating. Let *ds* be a small displacement of the sleeve (say 3% of the total working travel of the sleeve), then the rigidity of the mechanism results in a radial displacement *dr* of the centrifugal weights. The graphical construction of *dr* for a given value of *ds* is shown in Fig. 2. From the theory of virtual displacements, we have

$$P\,ds = 2\,C\,dr \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

In equation (1), P is the strength of the governor, and C is that radial (and in Fig. 2 horizontal) force which, if applied at the mass center of each centrifugal weight, just balances P in the frictionless governor. If a spring balance were attached to each centrifugal weight, and if it were pulled radially outward, then it would indicate the force C (provided, of course, that the spring balance correctly indicates horizontal forces, and that friction is eliminated). In operation of the governor, the force C is furnished by radial inertia and is called centrifugal force. From equation (1) follows the important fact that the strength is proportional to the centrifugal force, so long as ds/dr remains constant.[1]

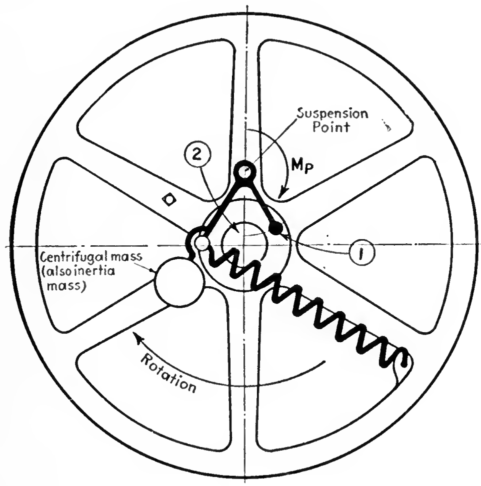


Fig. 4

In the case of the shaft governor, there is no well-defined, constant direction for the action of the regulating force, so that the regulating force is replaced by a regulating moment. But the "strength moment" coincides with the moment exerted by the spring (spring moment). Hence there exists no reason for the use of the term "strength moment." In shaft governors

[1] The relation between strength and centrifugal force can be illustrated by the aid of a simple example. In Fig. 3, let two weights W be suspended by massless rods as shown. Then the weights W can be kept from dropping either by two horizontal forces $\frac{1}{2}C$ or by a vertical force P. From a triangle of forces $\frac{1}{2}C = W$; we may also resolve P into two inclined forces $P' = \frac{P}{\sqrt{2}}$; take moments about the suspension point (*1*), then $\frac{P}{\sqrt{2}} l = W l \sqrt{2}$, or $P = 2W$. Hence $C = P$ in this case. This result can be had directly from the principle of virtual displacements. Move the weights through the small distance db, then radial travel dr of force C and axial travel ds of force P are alike; hence $P = C$, since $P\,ds = C\,dr$.

the centrifugal moment and the spring moment do not coincide, because there are many forces impressed upon the eccentric. This feature is dealt with explicitly in the chapter on shaft governors.

To exert its full strength, a centrifugal governor must slow down from its normal speed to a dead stop. In practice only very small speed fluctuations are allowed, so that only a small part of the strength can be exerted. To find this fraction, call u the normal angular velocity of the governor spindle, and du a small change of that angular velocity. From mechanics it is well known that centrifugal force $C = m\,r\,u^2$, where m = mass of the revolving body, and r = radius to mass center of that body. Corrections to this equation on account of oblong shape of revolving body are given in paragraph 4 of Chapter III. For symmetrical shapes, such as a sphere or a cylinder, no correction is needed. Let the governor be prevented from moving its sleeve, while the speed changes the amount du; then r remains constant; and since m naturally is constant, C and u are the only variables. Hence the change of centrifugal force is by differentiation.[1]

$$dC = 2\,m\,r\,u\,du = 2\,m\,r\,u^2\,du/u = 2\,C\,du/u \quad \ldots\ (2)$$

This equation is true for infinitesimally small changes of u only, but is in practice accurate enough even for $du = 10\%$ of u. The following equation is then approximately correct:

$$DC = 2\,C\,Du/u \quad \ldots\ (3)$$

where Du/u is the relative speed change. Du is not an infinitesimal quantity, but a small fraction of u, of the magnitude which is usually allowed in practice. The governor was in internal equilibrium before the speed change; it is not in internal equilibrium after the speed change. If the governor is frictionless, the radial force DC produces an axial force $DP = DC\,dr/ds$ at the sleeve; this force is known as the regulating force for the relative speed change Du/u. By substitution we obtain

$$DP = 4\,C\,\frac{Du}{u}\,\frac{dr}{ds} = 2\,P\,Du/u \quad \ldots\ (4)$$

(for 2 weights!)

[1] For an elementary derivation see Appendix, p. 209.

From this equation it follows that the regulating force is proportional to the strength of the governor and to the relative speed change. It equals the product of $4C$ (where C is the centrifugal force of each of the two weights), Du/u (which is the relative speed change) and dr/ds (which is the ratio of radial travel of centrifugal weight and of axial travel of sleeve for a small displacement of the governor).

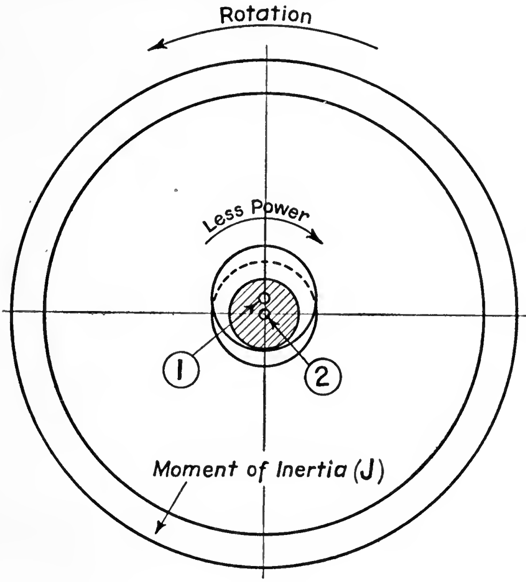


Fig. 5

To express this relation in figures, imagine a governor with a strength of 250 pounds; let the governor be in equilibrium at 300 revolutions per minute, and let the speed drop to 297 revolutions per minute. Then the governor will exert a force of $2 \times .01 \times 250 = 5$ pounds on its collar tending toward a new equilibrium position. In this case the regulating force equals five pounds.

It should be noted that the available regulating force becomes smaller as the governor approaches its new position. This feature will be explained in detail later on.

The corresponding moment equation for shaft governors is never written, because the pulsating moments continually impressed upon a governor of that type upset any elementary theory which might be founded upon a "regulating moment."

References to Bibliography at end of book: **1, 24, 35, 36, 37, 52, 53, 73.**

2. Regulating Force Due to Tangential Inertia. — In Fig. 5 is shown an eccentric with center (*1*), mounted loose on a shaft the center of which is (*2*). Let the valve gear be so arranged that clockwise relative rotation of the eccentric reduces the supply of power, and let the shaft rotate in the direction indicated by the arrow at the top of the illustration. The details of the valve gear are at present immaterial. A concrete example of variation of power by rotation of an eccentric is furnished by the well known "Buckeye" steam engine. If the demand for power is suddenly increased, an angular retardation, or negative angular acceleration a occurs, the magnitude of which depends upon the dimensions of the prime mover, upon the change of load and upon the moment of inertia "J" of the rotating mass connected to the eccentric. If the latter does not rotate relatively to the shaft, a moment ($a\,J$) is exerted upon the eccentric, tending to rotate it. Regulating forces or moments caused by tangential inertia differ from those produced by radial inertia (centrifugal force), inasmuch as the latter has to wait for the speed to change before it becomes effective (see equation 3 on page 7), whereas tangential inertia becomes effective at the instant when the change of load occurs.

From this fact it might appear at first thought as if tangential inertia should be universally used for speed governing purposes. Many such attempts have been made; but the survival of the fittest has eliminated most of them and has limited the application of tangential inertia almost entirely to shaft governors. The theoretical reasons underlying this decision of practice are partly given in this chapter, and will partly be explained in paragraph 6 of Chapter VI and paragraph 3 of Chapter IX, because they depend upon the dynamics of regulation.

The principle of using tangential inertia as a regulating agent differs from the principle enunciated in paragraph 2 of Chapter I and forms the principal exception to that principle. It is discussed in the present place to show the difficulties which even the slightest departure from the general principle introduces.

The governor shown in Fig. 5 is worthless, because it does not adjust any particular speed. It simply tries to keep constant the accidental speed at which the prime mover happens to be operating. A prime mover provided with such a governor could not be started, because the resulting acceleration would shut off the power, unless the inertia wheel were kept out of action by some other agency. Furthermore, the inertia wheel is inactive for very gradual changes of speed, because the acceleration "*a*" is then very small and usually insufficient to overcome friction resistance. With gradual change of load, the speed can thus rise or fall, until either equilibrium is attained by means other than the governing equipment, or else the prime mover is wrecked or stalled.

These reasons make it necessary to combine tangential inertia (if it is to be used for governing purposes) with centrifugal force and, speaking in very general terms, to make the latter so strong that it will compensate for the above mentioned shortcomings of tangential inertia. Such a combination may be effected either by the use of separate inertia and centrifugal masses, or by the use of a mass which serves both purposes.

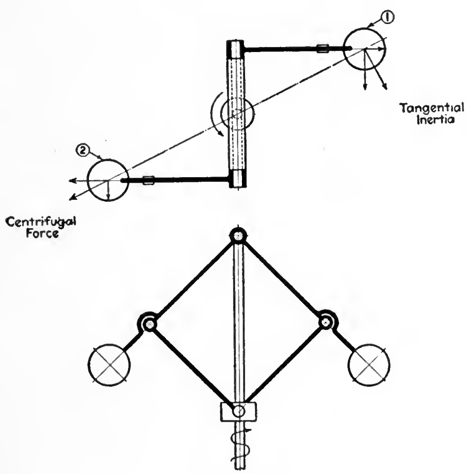


Fig. 6

In illustrations 6 to 9 both of these arrangements are shown. Figures 6 and 7 show spindle governors; 8 and 9 show shaft governors. The illustrations are purely diagrammatic and are intended to bring out principles only, with omission of all design details. In Figures 6 and 8 each mass is utilized for both radial and tangential inertia action; in Figures 7 and 9 separate masses are provided for the two effects.

On mass (*1*) of Fig. 6 the action of tangential inertia is indicated for a sudden increase of speed (reduction of load); on mass (*2*) the action of centrifugal force is indicated. Evidently the two forces act in the same sense for the direction of rotation marked in the illustration.

From Fig. 8 it is plain that the action of centrifugal force on mass center (*1*) turns clockwise about suspension point (*2*), so that an increase of speed likewise produces clockwise rotation. A study of the illustration also brings out the fact that for the indicated direction of rotation, tangential inertia acts

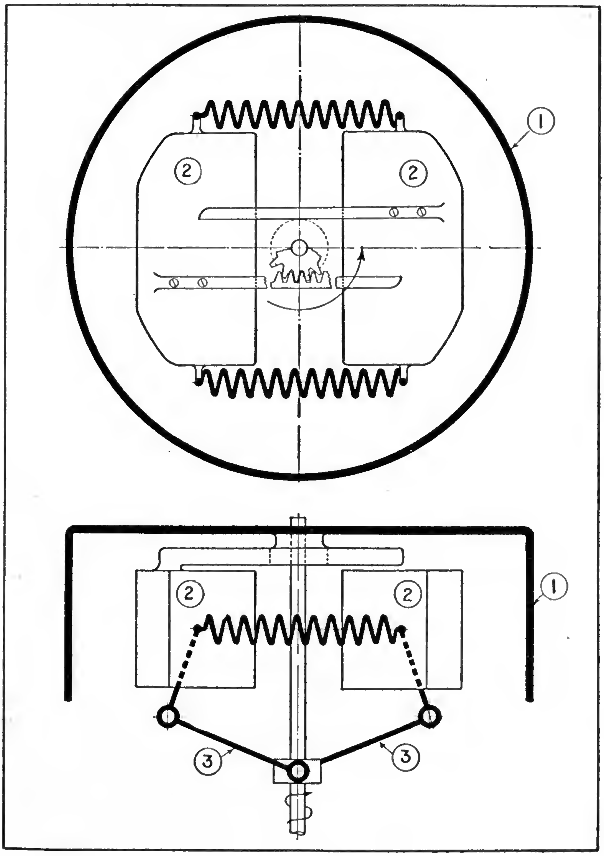


FIG. 7

in the same sense. Point (*3*) represents the center of the eccentric which operates the valve gear.

In Fig. 7, (*1*) is an inertia ring, loosely mounted on the revolving spindle. (*2*) and (*2*) are the centrifugal masses which are guided radially by bell cranks (*3*). The two mass systems are connected by a pinion and two racks. In this type of governor the complete separation of the two force systems is very distinct.

In Fig. 9, (*1*) is the inertia mass, and (*2*) is the centrifugal mass. They swing about the points (*4*) and (*5*) which are fixed in the revolving wheel. Point (*3*) is again the center of the eccentric. In this type the two force actions are not entirely separated, because each mass is partly subjected to the not intended force.

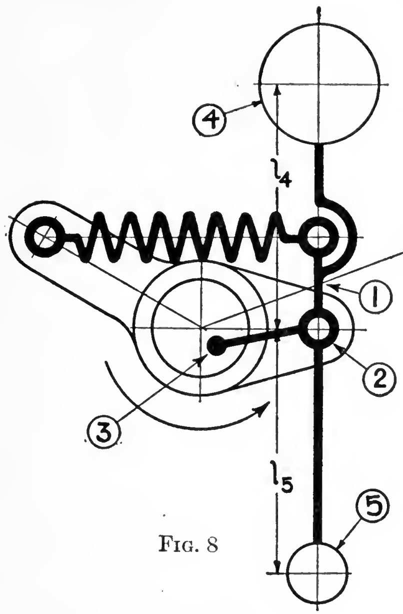


Fig. 8

From page 9 we remember that the regulating moment is ($a\,J$). This expression may be misconstrued into the belief that the moment could be increased to any desired amount by a corresponding increase of J. Such a conclusion would be wrong, because, as J grows, a becomes less for a given change of load. For, as long as the wheel does not yet shift the eccentric, it acts the same as if it were keyed to the shaft, or as if it were part of the flywheel.

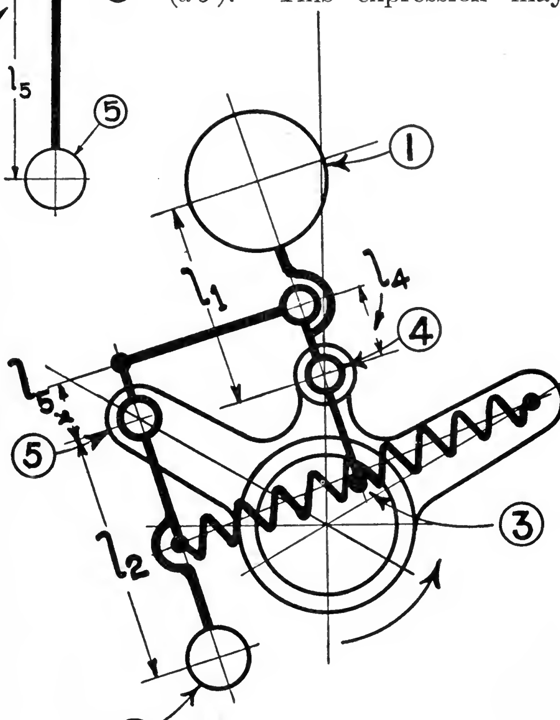


Fig. 9

And for a given change of load the rate of change of speed is the smaller, the greater the inertia of the flywheel.

The arrangement of Fig. 8 is frequently used; for this reason an expression will now be developed for the regulating moment produced by governors of this type.

In the diagram (Fig. 10), (*1*) is the center of the rotating shaft, (*2*) is the pivot about which the inertia weight with mass center (*3*) swings. Let the system start from rest with an angular acceleration a, and let joint (*2*) be inactive, that is rigid. For convenience of calculating, the motion of the mass may be resolved into a curvilinear translation of the mass center from (*3*) to (*4*), and into a rotation of the mass from (*5*) to (*6*) about its center (*4*). The total inertia moment must be the sum of the inertia moments due to translation and rotation. The moment about point (*2*) (which latter becomes point (*6*) at the end of the interval of time under consideration) then is

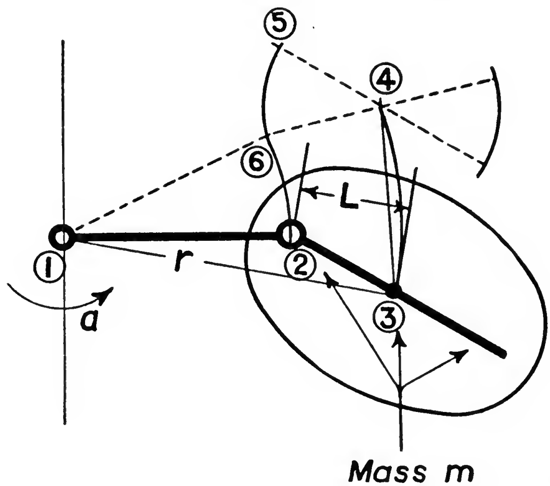


Fig. 10

$$M = m\,r\,a\,L + J\,a = a(m\,r\,L + J),$$

where J is the moment of inertia of the pendulum about its own mass center (*3*) and where L is the lever arm of the translatory inertia force $m\,r\,u$ about the pivot (*2*).

It is evident that the regulating force or moment is made smaller by the regulating motion of the governor; but that does not matter. For, as long as the governor moves toward its new configuration, its purpose is served, even though the available regulating force be lessened thereby. It should be noted that the regulating force of a centrifugal governor is likewise reduced as it approaches its correct position.

The regulating moment $m\,r\,a\,L + J\,a$ is the only moment which interests us. Other forces act, but they are not of vital influence. They will, however, be mentioned. Since, in practice, the whole system rotates, while the acceleration occurs, centrifugal forces must be added, and if the inertia mass swings about pivot (*2*), compound centrifugal forces must be added (the latter are known as Coriolis' forces, see paragraph 4 of Chapter IX). While the centrifugal forces have a moment, the compound centrifugal forces lie in the direction (*2*)–(*3*) and have no moment. However, they produce friction.

Study of Fig. 8 teaches that in this form of governor the moment due to $m\,r\,L\,a$ is quite small compared to the moment caused by $J\,a$. It should be understood that the moment $M = a(m\,r\,L + J)$ refers to an ideal frictionless governor only.

The theory of the action of tangential inertia in the interaction between governor and prime mover will be found in paragraph 3 of Chapter IX, which deals with the dynamics of regulation.

References to Bibliography at end of book: **2, 33, 36, 73.**

3. Work Capacity of Centrifugal Governors.— If the sleeve of a non-rotating centrifugal governor of the spindle type be raised through its working travel s, the work $P\,s$ is done, where P is the average strength of the governor. This work may also be expressed as $\int_0^s P\,ds$, where P is the variable strength of the governor.

Obviously this expression represents the greatest amount of work which can be done by the governor as a motor, unless speeds considerably in excess of the normal are considered. Hence, the expression $\int P\,ds$ is called the work capacity of the governor. Its numerical value is usually given in governor catalogues.

Again it is obvious that for purposes of regulation only a small part of the work capacity can be utilized, because a speed change of 100% cannot be allowed in practice.

Before the available part of the work capacity can be found, the influence of friction in governor and valve gear must be studied.

References to Bibliography at end of book: **36, 37, 73.**

4. Detention by Friction. — If a spindle governor be run at such a speed that it is floating, touching neither the upper nor the lower stop, the speed can be varied between the limits u_h and u_l, while the governor sleeve remains motionless. Any speed higher than u_h raises the governor, any speed lower than u_l lowers it. The ratio

$$q = \frac{u_h - u_l}{\frac{1}{2}(u_h + u_l)} \qquad (1)$$

has received various names, namely "detention by friction," "sluggishness," "time lag," "insensibility," or "degree of insensibility." In the present book the name "detention by friction" will be adopted. This quantity depends upon

(1) the design of the governor,

(2) its lubrication,

(3) the vibration to which it is subjected, while the speed changes,

(4) the forces transmitted to the governor through the sleeve.

Taking up these items in the order given, we find that in a governor of the type shown in Fig. 2 or in Fig. 6, centrifugal and centripetal forces are balanced by forces passing through numerous joints, each of which produces friction, whereas in the type shown in Fig. 7 these forces are balanced directly without the interposition of joints. Consequently the former type has, under otherwise equal conditions, more friction than the latter, and shows more detention due to friction. The friction can be somewhat reduced by the use of knife edges.

Item 2, lubrication, certainly affects friction. Governors may be so designed that the joints are always submerged in oil. Such a design furnishes the only case in which the degree of lubrication is definitely known. In all other cases it is uncertain.

In many governor catalogues the detention due to friction is given on the basis of a certain coefficient of friction and of absence of vibration. The calculations made for determining the detention are more or less arbitrary and of doubtful value, unless all are based on the same coefficient of friction. In that case the figures afford a comparison which can be used to advantage.

The immediate purpose of any such calculation is to find between what limits the force C in Fig. 2 can be varied without moving the mechanism. The solution of this problem is a matter of applied mechanics. It may be effected by graphic

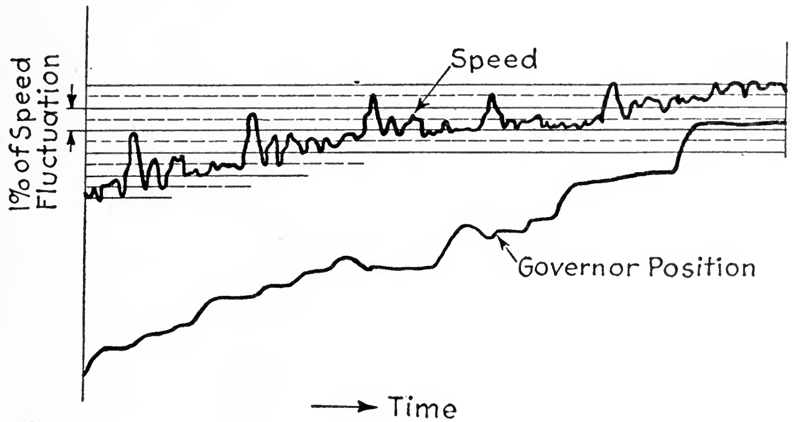


Fig. 11

statics or by taking moments, or by the principle of virtual displacements. In the appendix two numerical examples are given.

Under average conditions the detention varies for different types of governors and for various degrees of lubrication from $\frac{1}{2}\%$ up to 4%. The detention q usually varies for different positions of one and the same governor.

Item 3, vibration, considerably affects detention, as may be judged from the following fact which proves that vibrations always reduce and often eliminate friction. Objects lying quietly on an apparently horizontal table march off promptly, if the table is rapped by a vibrating instrument, such as a pneumatic hammer, riveter, etc. In the same manner a governor subject to vibrations loses the detention by friction in various degrees. To prove this fact, the author artificially produced vibrations in an ordinary Watt type spindle governor by atttaching an unbalanced revolving weight to its collar. Figs. 11 and 12 show speed and position of the governor plotted against time. Fig. 11 was taken with the weight at rest, and Fig. 12 was taken with the weight revolving at high speed.

It will be observed that Fig. 11 shows the peculiar jaggy motion of detention, whereas in Fig. 12 practically every trace of detention is eliminated. Note the practically straight line, which represents motion without detention. Incidentally

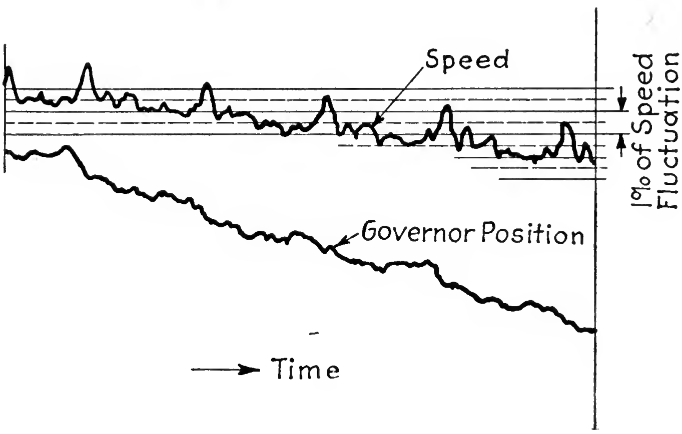


Fig. 12

it will be noticed that the speed of the governor varied cyclically during the test, whereas the speed appeared to the observer to be uniform. Circumstances of this sort make even the experimental determination of the detention by friction difficult.

Influence 4, namely that of forces transmitted through the governor sleeve, will be considered in Chapter VIII.

In spite of the fact that vibrations, variation of lubrication, and other circumstances make the detention by friction an uncertain quantity, we must admit that there are cases where the theory of the detention by friction is well applicable (see next paragraph). In such cases the following equations are useful:

$$\text{detention } q = \frac{u_h - u_l}{\frac{1}{2}(u_h + u_l)} = \frac{u_h^2 - u_l^2}{2\left(\frac{u_h + u_l}{2}\right)^2} = \frac{m\,r\,u_h^2 - m\,r\,u_l^2}{2\,m\,r\left(\frac{u_h + u_l}{2}\right)^2}$$

$$q = \frac{C_h - C_l}{2\,C} = \frac{C_h\frac{dr}{ds} - C_l\frac{dr}{ds}}{2\,C\frac{dr}{ds}} = \frac{P_h - P_l}{2\,P}$$

or approximately

$$q = \frac{2(P - P_l)}{2\,P} = \frac{P - P_l}{P} = \frac{F}{P}$$

In these equations C_h, C, and C_l are centrifugal forces for high, average, and low speed, covering the range over which the governor is detained by friction; P is the strength of the governor for the position under consideration. Evidently a governor can have only one value for its strength at a given position; hence the expressions $P_h - P$ and $P - P_l$ must mean additional axial forces at the sleeve, sufficient to overcome the internal friction of the governor. For this reason the said values have been denoted by the letter F.

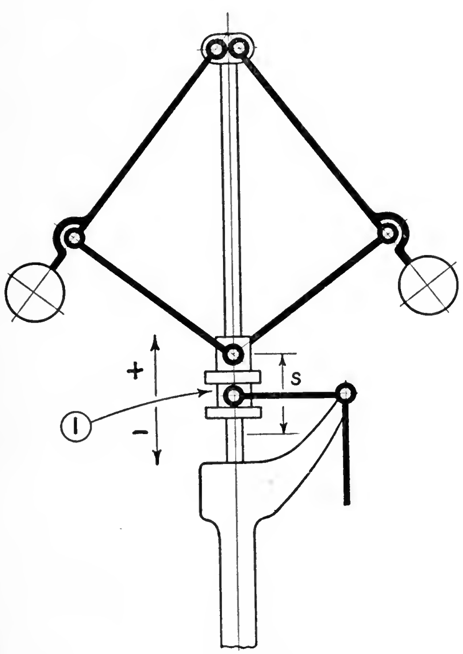


FIG. 13

If q has been determined by calculation, the equivalent friction force F can be found from the final equation $q = F/P$.

References to Bibliography at end of book: **36, 73.**

5. Overcoming a Passive Resistance by a Centrifugal Governor.—When moving the throttle valves of steam turbines, or well-balanced throttle valves of engines, governors overcome "passive resistance," that is to say a resistance which exerts no vibratory or shaking force upon the governor.

In order to determine the influence of a passive resistance upon a governor, imagine the latter to be replaced by a spring balance at point (*1*), Fig. 13. It should be understood that the linkage between the governor and the valve gear must be so arranged as to vary the energy supply to the prime mover from zero to its maximum value, while the governor sleeve passes through its range of travel. Let the forces which are measured in moving the spring balance up and down be X_u and $-X_d$ respectively.

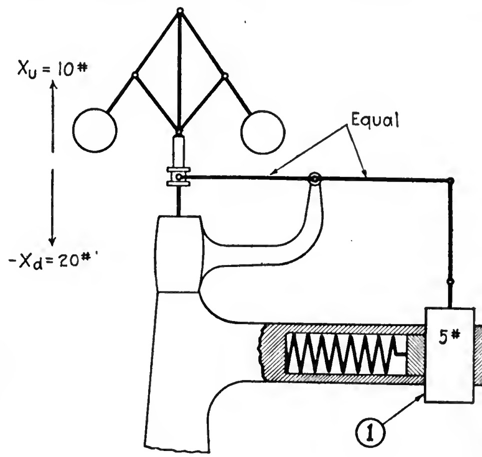


Fig. 14

Then $\frac{X_u + X_d}{2} = R$ is the passive resistance of the valve gear, that is to say a force which resists motion either way and affects the motor part of the governor. On the other hand, $\frac{X_u - X_d}{2}$ is a force which does not affect the motor part of the governor, but only the speed counter part.[1] Even if the passive resistance of the valve gear were entirely eliminated by

[1] To make these relations clear, imagine the valve resistance replaced by the frictional resistance of a spring load acting on a heavy block (*1*), Fig. 14. If the frictional resistance is 15 pounds, then the forces at the governor collar necessary to move the weight are 10 and −20 pounds as indicated. The resistance of 15 pounds concerns the motor part of the governor, and the constant force of 5 pounds reduces the equilibrium speed of the governor.

vibrations, $\frac{X_u - X_d}{2}$ would still act and would be noticed by the speed counting properties of the governor. See below in paragraph 1 of Chapter III.

R is then the force which the governor must overcome in order to shift the valve gear, provided that the governor itself is free from friction; but from the preceding paragraph it is known that most governors have friction and must, for this reason, overcome an additional force F (equivalent friction force). Hence the governor must exert a force $R + F$ to move the valve gear. This fact is mathematically expressed by the following equations:

$$(1)\quad R = \dot{D}\, P = 2\, P \frac{D_1 u}{u} \qquad \text{see (4) on page 7.}$$

$$(2)\quad F = q\, P = 2\, P \frac{D_2 u}{u} \qquad \text{see page 18.}$$

The 2 in the second equation comes from the definition for q; the value of q covers the whole detention above and below normal speed, so that $q = \frac{2\, D_2 u}{u}$ is equal to twice the relative speed change which is necessary to overcome the internal resistance F; $D_1 u$ is that speed change which is necessary to overcome the external resistance R. Addition of equations (1) and (2) furnishes

$$R + F = 2\, P \frac{D_1 u + D_2 u}{u}$$

From this follows the total speed change which is necessary to overcome both resistances:

$$(3)\quad D\, u = \frac{R + F}{2\, P} u = \frac{u}{2}\left(\frac{R}{P} + \frac{F}{P}\right)$$

Evidently the desirable effect of making the total speed change small can partly be obtained by the increase of the strength P; partly only, because F/P is a constant for a given type of governor and provides a lower limit for the speed change, no matter how strong the governor might be.

According to the theory just presented, it would be impossible to obtain close regulation, unless the internal friction of the governor were reduced to almost nothing. This viewpoint is strongly emphasized in catalogues advertising governors with small internal friction. On the other hand, everyday experience proves that in many cases close regulation is obtained even with governors having considerable internal friction. As may be surmised, vibratory forces from various sources partially or wholly eliminate friction in these cases. See also chapters on resistibility and on shaft governors.

In other cases, where vibratory forces are either absent, or quite small compared to the passive friction resistance, nothing remains but the use of strong governors with small internal friction (see chapter on relay governing). In that case it is advisable to investigate the required speed change for two positions of the governor and valve gear, because the strength of the governor and the resistance of the valve gear usually vary from position to position.

In practice the difficulty arises that neither the frictional passive resistance can be exactly predetermined nor the helping vibratory force is definitely known, so that judgment is required in the selection of a governor. In the regulation of gas engines working with dirty gas (producer gas, blast furnace gas, coke oven gas) exceptionally great strength of governors is needed, because the regulating valves which are operated by the governor gradually clog with dirt, and stick. Weak governors require too frequent cleaning of the regulating valves.

Insufficient strength of governors coupled with absence of vibratory forces can easily be recognized from the speed variations which occur even with the smallest and most gradual changes of load on the prime mover.

To make matters even clearer, an example will be given. Let the passive resistance of the governor rig be 10 pounds, measured at the governor collar. Let q, the detention by internal governor friction, be equal to $1\frac{1}{2}\%$. Let it be stipulated that the total speed variation in overcoming the valve resistance with gradual changes of load must not exceed 2%.

What strength P of the governor is necessary to meet these requirements? For a solution we will write equation 3 in this form:

$$2\frac{D\,u}{u} - q = \frac{R}{P}, \text{ from which } P = \frac{R}{2\frac{D\,u}{u} - q}$$

By substitution,

$$P = \frac{10}{.04 - .015} = \frac{10}{.025} = 400 \text{ pounds.}$$

References to Bibliography at end of book: **36, 73.**

CHAPTER III

THE CENTRIFUGAL GOVERNOR AS A MEASURING INSTRUMENT (SPEED COUNTER)

1. Equilibrium Speed, Static Fluctuation and Stability. — If, in a prime mover, the resisting torque is reduced, while the torque which is produced either remains constant or is not reduced as much as the resisting torque, the liberated excess energy is converted into kinetic energy, that is to say, the speed grows. For the sake of stability, the governor must be so arranged that the increase of speed causes it to move the torque-controlling mechanism toward the no-load position. The inverse series of events has to be gone through, if the load is increased.

As mentioned in paragraph 1 of Chapter I, the purpose of speed governing is to keep the speed constant in spite of variations of torque. It will be shown in this paragraph and in paragraph 2 of Chapter IX that centrifugal governors must have a higher equilibrium speed (at least temporarily) at no load than at full load, in order to make use of the principle enunciated at the beginning of the present paragraph.

The smaller the speed variation, the better the governing. In consequence, several measures of the closeness of regulation have been introduced; most important among them is the term "fluctuation" or "static fluctuation," which means

$$p = \frac{u_u - u_d}{\frac{1}{2}(u_u + u_d)} \quad \text{(1)}$$

In this equation u_u means that speed at which the governor is in equilibrium in its uppermost (or equivalent) position, and u_d means that speed at which the governor is in equilibrium when in its lowest position. The term "is in equilibrium" means that the governor is floating between centrifugal and centripetal forces, with friction eliminated. Uppermost and lowermost position refer to no load and to full-load positions.

The terms "outermost" and "innermost" are more correct, but are seldom used.

If a prime mover could be made (which it cannot) to always run at a speed proportional to the equilibrium speed of the governor, a small value of p would mean close regulation, and $p = 0$ would represent the ideal case of constant speed regulation. A governor of this description ($p = 0$) is called isochronous.

In speaking of static fluctuation, governor manufacturers usually assume that the whole range of the travel of the governor collar is utilized for varying the torque produced by the prime mover, from zero to its maximum value. If that condition does not exist, that is to say, if a fraction of the travel of the collar varies the torque through its complete range, the static fluctuation of the governor, as given in the governor catalogues, remains the same, but the effect upon the prime mover is different and is equivalent to a reduction of static fluctuation, or to an increase of sensitiveness.

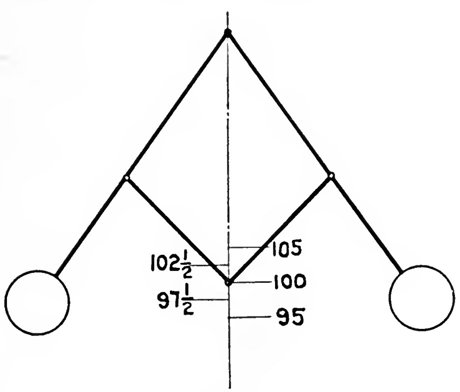


Fig. 15

A governor with a small static fluctuation is spoken of as a "sensitive" governor, so that the sensitiveness of a governor is closely related to the value $1/p$.

An example will illustrate these statements. In Fig. 15 the numbers represent the equilibrium speeds for certain positions of the governor collar. If the total available travel of the latter be utilized in governing, the static fluctuation is $\frac{105 - 95}{100} = 10\%$. If, on the other hand, the position marked $102\frac{1}{2}$ rev./min. is the no-load position, and the one marked $97\frac{1}{2}$ is the full-load position, then $\frac{102\frac{1}{2} - 97\frac{1}{2}}{100} = 5\%$ is the static fluctuation.

The mistake is often made to assume that an engine or turbine will work with close regulation, because it has been equipped with a sensitive governor, which, as before stated, means a governor with small static fluctuation. Such an assumption is wrong. The sensitiveness of a governor gives no guarantee for close regulation, and too great a sensitiveness will often make it impossible to obtain close regulation. For detail discussion of the causes underlying these facts see paragraph 2 of Chapter IX.

The static fluctuation of a governor is related to its stability, which means its ability to return to the position of equilibrium after it has been displaced from that position. For a judicious discussion of this point, knowledge of the "characteristic" of a governor is necessary. See the following paragraph.

References to Bibliography at end of book: **1, 11, 14, 24, 26, 36, 37, 53, 73.**

2. Characteristics of Governors (C-curves). — In the further treatment of the properties of governors as speed counters it is advisable to deal separately with spindle governors and with shaft governors. The latter have numerous moments impressed upon them, and it becomes necessary to express the equation of equilibrium in moments. In the spindle type of governor it is possible and advisable to deal with forces. The treatment of the spindle type is therefore simpler and will be taken up in the following paragraphs. For the shaft governor see Chapter VI.

The characteristic properties of a governor as a speed counter can be expressed and plotted in various ways; for instance, we may plot:

(1) equilibrium speed against position of collar,

(2) equilibrium speed squared against position of collar,

(3) equilibrium speed against radial displacement of centrifugal weight,

(4) equilibrium speed squared against radial displacement of centrifugal weight,

(5) centrifugal force against radial displacement of centrifugal weight.

At first thought, method 1 appears to be the most convenient and to offer the final criterion. Yet, closer study reveals the fact that method 1 shows nothing but the final result, whereas method 5 shows most of the hidden mechanics of the governor.

This latter method was, as far as the author knows, first used by Mr. W. Hartnell [1] and was later very much developed by Mr. M. Tolle.[2] The Hartnell-Tolle method will be used in this and the following paragraphs. The mass of the links will at first be neglected.

In the present discussion C represents that radial, outwardly directed force which balances (directly, or by means of a linkage) all other forces acting in the governor, such as

W = weight of centrifugal masses

Q = weight of counterpoise, see Fig. 16

S_1, S_2, etc. = forces exerted by springs

F = equivalent friction forces, see paragraph 4 of Chapter II.

The line of action of C passes through the mass center of each centrifugal weight.

In order to develop the properties of characteristics logically, we start with a simple case, namely with a modified type of Watt governor (see Fig. 16). In this type the two principal centripetal forces are due to the weights W and Q. It is advisable to let W represent the weight of all centrifugal masses, because Q can then represent the weight of the whole counterpoise. Since the horizontal force C balances the vertical forces W and Q, it may be considered as consisting of two components, C_w and C_q. C_w may be found from W by forming moments about fulcrum *(13)*, so that $C_w = W\frac{L_2}{L_1}$. C_q can likewise be found by the moment method;[3] but any method of mechanics, including graphic statics, may be employed. After the forces

[1] Proceedings of British Institute of Mechanical Engineers, 1882.

[2] Zeitschrift des Vereins Deutscher Ingenieure, 1895.

[3] The lever arm of Q about *(13)* is *(13)*–*(14)*, which will be seen from the resolution of Q into a horizontal and an inclined force. Shift the latter in its own direction to point *(14)*, and resolve it into a vertical and a horizontal force. All horizontal forces cancel, because there is an equal and opposite force on the other side of the governor. The vertical component equals Q.

C_w and C_q have been found, they are laid off directly under the center (*12*) of weight W from a horizontal base line (*1*) (*15*), Fig. 16. Thus (*15*) (*10*) represents C_w to some scale, (*15*) (*7*) represents C_q to the same scale, and (*15*) (*4*) represents

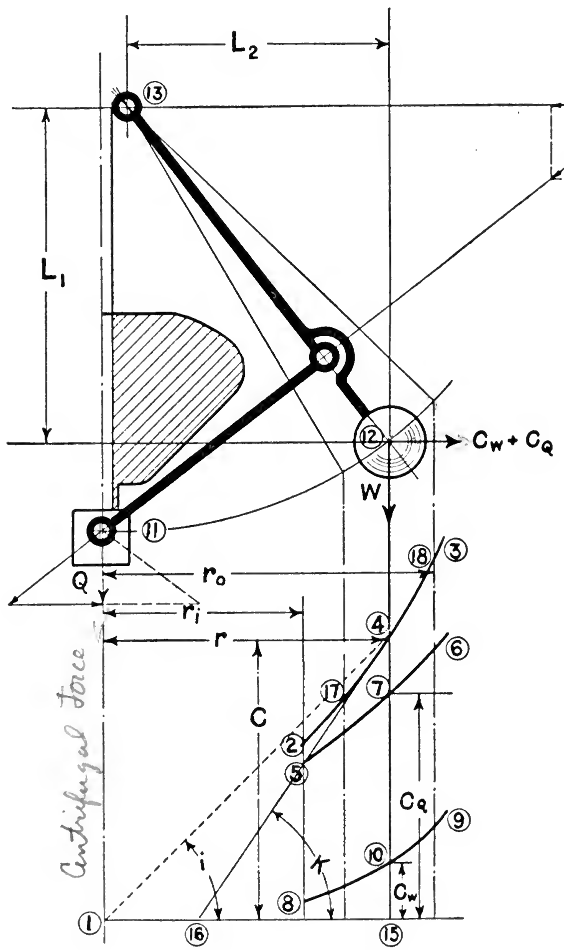


Fig. 16

$C = C_w + C_q$. By repetition of this process for several positions of the governor, the curves (*8*) (*10*) (*9*), (*5*) (*7*) (*6*) and (*2*) (*4*) (*3*) are obtained.

It should be noted that in the construction of these characteristic curves the governor has been treated solely as a mechanism and not as a speed counter. The mechanism in this case consists of a slider crank with fulcrum (*13*) and slide (*11*). The location of the governor spindle has not entered the calculation. Consequently, the shape of the characteristic curves of a weighted governor does not depend upon the location of the governor spindle with regard to the mechanism (*13*) (*12*) (*11*). On the other hand, we certainly must consider the location of the characteristic curve relative to the spindle if we wish to investigate the physical properties which it represents. To that end we must replace the auxiliary horizontal force C by its true function of centrifugal force of the revolving masses. Then

$$C = m\,r\,u^2 \qquad (1)$$

where

m = mass of the revolving centrifugal weights; $m = W/g$

r = radius from axis of rotation to mass center of W. The shape of the centrifugal weights is for the present limited to bodies of circular section. For other shapes see paragraph 4 of present chapter.

$u = 3.14\,\frac{n}{30}$ = that angular velocity at which equilibrium exists between centrifugal and centripetal forces.

n = revolutions of governor per minute

From (1) follows $u^2 = \frac{C/m}{r}$. In order to be able to tell at a glance the value of u for different positions of the governor, we introduce the following trigonometric relation: Calling "i" the angle (*4*) (*1*) (*15*) of Fig. 16, we have $\tan i = C/r = m\,u^2$, from equation (1); but, since m is constant, $\tan i$ is proportional to u^2, or u is proportional to $\sqrt{\tan i}$. Hence u grows, as long as i grows; it falls, whenever i decreases.

Observation of the angle i in Fig. 16 teaches that, in the type of governor there represented, the speed grows, as the weights move outward. The question is: Must this be so, or would it be just as good to have the speed drop, as the weights move outward? In order to decide this question refer to Fig. 17, in which (*2*) (*4*) (*3*) is a governor characteristic with rising speed. Imagine that the governor weights are displaced the small distance *dr* from position of equilibrium (*4*), while the speed remains constant. The centrifugal force grows from (*4*)(*7*) to (*5*)(*8*), but the centripetal force grows at the same time to (*6*)(*8*), so that a centripetal force (*6*)(*5*) results which urges the governor back to its position of equilibrium. The governor is stable.

If, on the other hand, the characteristic had followed the line (*4*)(*9*), the displacement *dr* (at constant speed) would have produced an excess centrifugal force (*5*)(*9*) which would

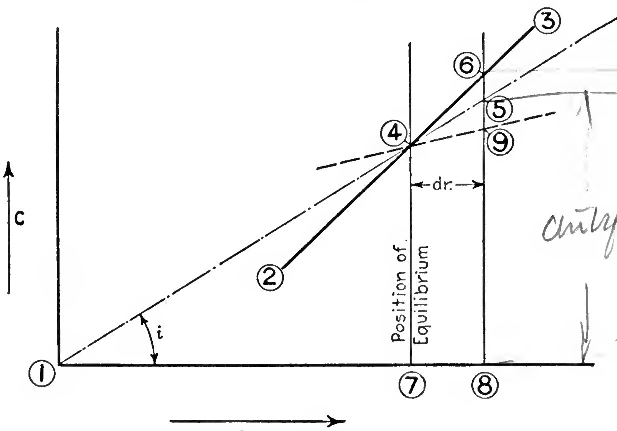


Fig. 17

urge the governor still farther away from its position of equilibrium. The governor would be unstable.

The rise and fall of the angle "*i*" of the characteristic in relation to the radius *r* is thus a certain criterion for the stability of a governor.

A quantitative measure of the stability is furnished by the well-known expression of mechanics: "Restoring force at unit displacement." Strictly speaking, this expression is correct only, if the restoring force is proportional to the displacement. In the absence of such proportionality we must use the ratio: restoring force at a given small displacement, divided by that same small displacement. In the case of a centrifugal governor, the value of this ratio is $z = \dfrac{d(C_p - C_f)}{d\,r}$, where C_p = centripetal force and C_f = centrifugal force. For the interpretation of this equation it should be remembered that the present investigation considers only the governor by itself, running on a test block, so that pushing the governor up and down, away from its position of equilibrium, does not alter the speed at which it runs, and the latter remains constant. For the displacement dr, Fig. 17, (6) (5) is the restoring force $d(C_p - C_f)$ which, as can be seen from the illustration, equals the differential (with regard to speed) of the centrifugal force for r = constant. But dC (for r = constant) equals $2\,m\,r\,u\,du$. Hence

$$\text{the stability} = \frac{(6)\,(5)}{dr} = \frac{2\,m\,r\,u\,du}{dr} = z$$

so that

$$z = 2\,m\,r\,u^2 \frac{\frac{du}{u}}{dr} = 2\,C \frac{\frac{du}{u}}{dr} \cdots\cdots \quad (2)$$

The stability of a governor, when running on a test block "by itself" and not governing a prime mover, is proportional to the total centrifugal force and to the value $\dfrac{\frac{du}{u}}{dr}$. By passing from differentials to finite differences we obtain

$$\frac{\frac{du}{u}}{dr} = \text{approx.} \frac{\frac{u_u - u_d}{\frac{1}{2}(u_u + u_d)}}{r_o - r_i} = \frac{p}{r_o - r_i}$$

and the stability $z = \dfrac{2\,Cp}{r_o - r_i}$. An elementary derivation of this equation is given in the appendix. This latter value is an approximation only, but is substantially correct for straight-line characteristics. If a characteristic is curved, the value $\dfrac{\frac{du}{u}}{dr}$ must be used, but since this derivative is not easily constructed in the diagram, it is advisable to fall back on the trigonometric function given on page 28. To this end we form the complete differential of C and obtain

$$dC = 2\,m\,r\,u\,du + m\,u^2\,dr = 2\,C\frac{du}{u} + \frac{C\,dr}{r}$$

or

$$2\,C\frac{du}{u} = dC - \frac{C\,dr}{r}$$

The left-hand member of this equation, if divided by dr, equals the stability z, hence

$$z = \frac{2\,C\frac{du}{u}}{dr} = \frac{dC}{dr} - \frac{C}{r} \quad \ldots\ldots\ldots\ldots\ldots (3)$$

But $dC/dr = \tan k$, and $C/r = \tan i$ (see Fig. 16).

The stability of a governor can, therefore, be very simply expressed as the difference of the tangents of two angles. Re-

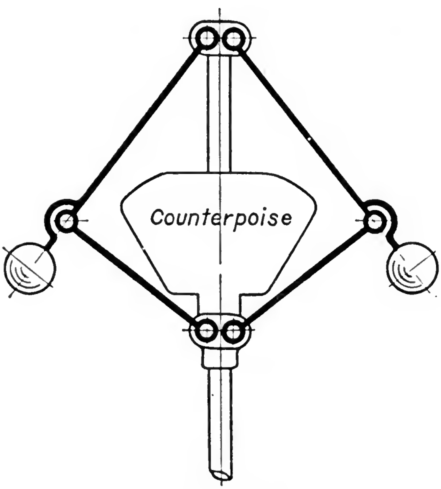


Fig. 18*a*

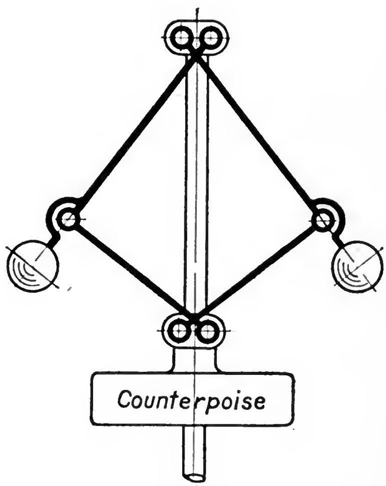


Fig. 18*b*

ferring again to the same illustration we note that the angle k is formed by the tangent to the characteristic at that point (*4*), where the stability is to be determined, and that the angle i is formed by the vector drawn from the origin (*1*) to the point (*4*) on the characteristic. From the fact that the stability depends upon the difference between the tangents of the angles k and i, it is evident that the governor is stable, whenever the point of intersection (*16*) lies to the right of point

(*1*), and that the governor is unstable, if the point (*16*) lies to the left of (*1*). "Right" and "left" must, of course, be reversed if the characteristic is drawn to the left of the spindle, instead of to the right as in Fig. 16.

From this influence of the intersection point (for instance: point (*16*), for point (*4*) of characteristic) follows the important conclusion that any weighted governor may be made more sensitive, or less stable, by moving its mechanism closer to the axis of rotation. The shifting of the relative positions of mechanism and of axis of rotation is illustrated in Figs. 18*a* and 18*b*. In both cases the governor mechanism and the shape of the characteristic are identical, but the stability in Fig. 18*a* is considerably greater than that in Fig. 18*b*.

It is, therefore, possible to design weight-loaded governors (which are based on a given mechanism) to have any desired stability by simply varying the location of the mechanism relative to the spindle. It will be shown later on that spring-loaded governors can, with equal or even greater facility, be designed for any desired stability. However, the question of what stability gives the best regulation cannot be decided by any study of the governor alone; it requires the study of the interaction of governor and prime mover. See Chapter IX.

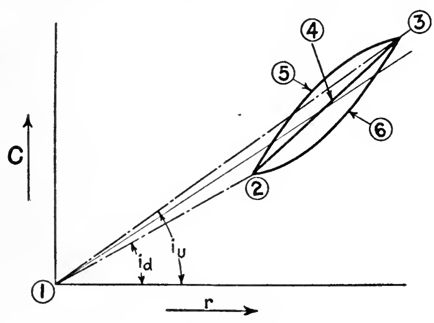


Fig. 19

Knowledge of the properties of the characteristic clears up the widespread confusion between the terms "stability" and "static fluctuation." In Fig. 19 three characteristics, (*2*) (*4*) (*3*), (*2*) (*5*) (*3*) and (*2*) (*6*) (*3*) are shown, all of which have the same static fluctuation, because the angles i_u and i_d are the same for all three. Tangents drawn to the curves at different points show, however, that the characteristic (*2*) (*5*) (*3*) is unstable near point (*3*), whereas the characteristic (*2*) (*6*) (*3*) is unstable near point (*2*). From this difference follows that knowledge of the static fluctuation alone without knowledge of the shape of the characteristic is not sufficient to judge the properties of a governor.

The difference between the two terms under consideration will be further emphasized by the following statement: As far as the governor alone is concerned — apart from its connection with a prime mover — the static fluctuation may be decreased without decrease of stability, by utilizing only a small part of the radial travel of the weights. Figure 16 shows that confinement of the weight travel between the vertical dash and dot lines reduces the static fluctuation. To see this more clearly, imagine straight lines drawn from point (*1*) to intersections (*17*) and (*18*) of the vertical dash and dot lines and the characteristic. The angles i made by these straight lines with the axis of abscissæ vary considerably less than those made by the straight lines drawn from (*1*) to (*2*) and (*3*). It will be remembered that angle i is, to a certain extent, a measure of the equilibrium speed.

While limitation of radial travel has been frequently employed for increasing the sensitiveness of governors, and while it does not change the stability of the governor proper, it does decrease the stability of regulation of a prime mover (see Chapter IX). It also reduces the work capacity of the governor and necessitates the use of a stronger governor, if a passive resistance is to be overcome. Finally, it reduces the ability of the governor to act as a shock absorber (see Chapter VIII).

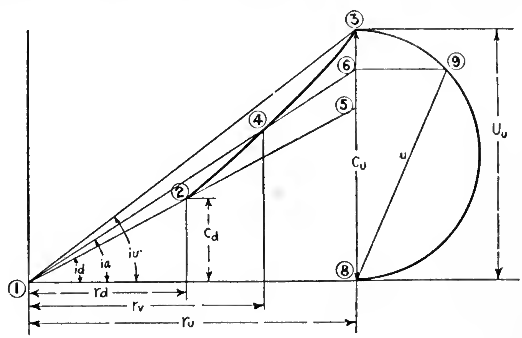


Fig. 20

Engineers who wish to go more deeply into the study of characteristics will find the two following geometrical relations of interest: The first of them expresses the static fluctuation by the angles i. To find this relation we transform equation (1) on p. 23 in the following manner (see Fig. 20):

$$p = \frac{u_u - u_d}{\frac{1}{2}(u_u + u_d)} = \frac{\frac{1}{2}(u_u^2 - u_d^2)}{\frac{(u_u + u_d)^2}{2}} = \frac{\frac{1}{2}\left(\frac{C_u}{mr_u} - \frac{C_d}{mr_d}\right)}{C_a/mr_a} = \frac{\tan i_u - \tan i_d}{2 \tan i_a}$$

In these equations the indices u, a, and d mean "up," "average," and "down."

In any regulation aiming at practically constant speed, u_a and u_u differ only by a few per cent, so that it is permissible to write

$$p = \frac{\tan i_u - \tan i_d}{2 \tan i_u} = \frac{\overline{(3)(5)}}{2 \times \overline{(3)(8)}}$$

The second relation solves the problem of quickly determining the equilibrium speed for any point of the characteristic, if the equilibrium speed for any other point is known. Referring again to Fig. 20, let u_u, point (3) of the characteristic be known, and let it be desired to find u at point (4). Draw ray (1) (4) to intersection (6) with vertical (3) (8), erect semicircle over (3) (8), draw horizontal through (6) to intersection (9) with semicircle, then $\overline{(8)(9)}$ equals u of point (4) to the same scale to which $\overline{(3)(8)}$ equals u_u. Proof: $\frac{u^2}{u_u^2} = \frac{\tan i_{(6)}}{\tan i_u} = \frac{\overline{(6)(8)}}{\overline{(3)(8)}}$, but by construction $u_u = \overline{(3)(8)}$, so that it cancels out, and $u^2 = u_u \times \overline{(6)(8)}$ remains, which means that u is the mean proportional of $\overline{(6)(8)}$ and $\overline{(3)(8)}$. The construction which is given above makes $\overline{(8)(9)}$ the mean proportional of these two distances.

References to Bibliography at end of book: **1, 11, 14, 25, 35, 36, 37, 52, 53, 72, 73.**

3. Constituent Parts of the Characteristic Curve.—While the analysis of the preceding paragraph teaches that the principal static properties of a governor depend upon its resultant characteristic, more detailed study proves that the individual character of the C_Q and C_W curves is of equally great importance.

In order to obtain sufficient material for this discussion, we shall investigate the characteristics of two governors quite different from the Watt type.

(1) Proell type, or inverted suspension type, Fig. 21. In this design of weight-loaded governor the centrifugal weight is

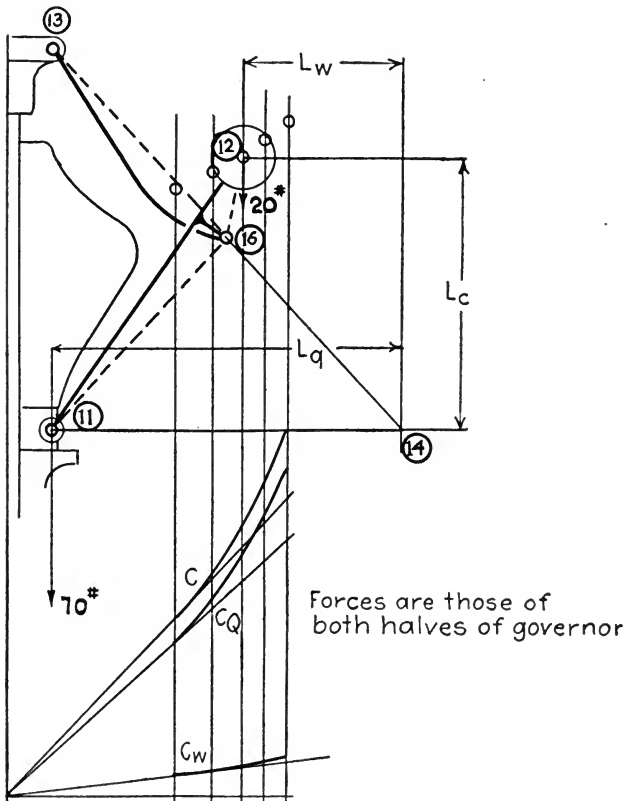


Fig. 21

not carried directly by the pendulum arm *(13) (16)*, but forms part of the connecting rod *(11) (16)*. The forces C_W and C_Q are found by forming moments about the instantaneous center *(14)*; the lever arms of the forces have been marked L_C, L_W, and L_Q.

In the illustration the forces have been entered to scale, and the C_W and C_Q curves have been drawn. Study of these curves shows that the C_W curve is slightly unstable in its inner part (at least for the selected location of the governor spindle) and that it is stable in its outer part. The C_Q curve shows excess stability. By variation of the ratio Q/W the stability of the governor may be varied. In the illustration, Q is so much greater than W that the governor has considerable stability, particularly in the outer position.

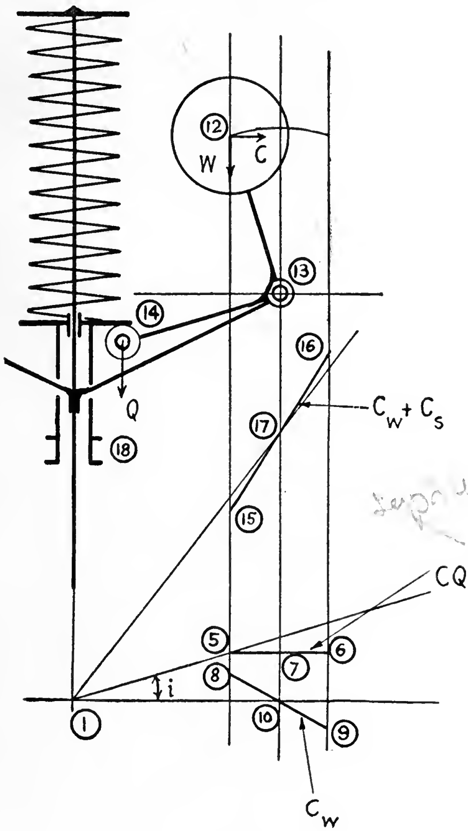


FIG. 22

(2) Hartnell type, Fig. 22. This governor consists of two (sometimes 3 or 4) centrifugal weights W with mass centers *(12)*; the weights are carried on bell cranks with fixed fulcrums *(13)*. The principal centripetal force is furnished by an axial spring. Again C_Q and C_W can be found by moments about the pivots, which are represented by points *(13)* in this case. Q is variable in this governor so that doubt may arise concerning the meaning of C_Q. For reasons which will appear later it is advisable to plot C_Q for a constant force Q. For the latter, we assume an arbitrary constant unit force applied at point *(14)*. Both the C_W curve *(8)(10)(9)* and the C_Q curve *(5)(7)(6)* show a decidedly negative static fluctuation; note that tan i drops. To render the governor useful, the central force Q must be made variable to such an extent that the resulting C curve furnishes a small positive fluctuation. Disregarding for

the present the limitations which may be imposed by the space available for the spring, we can determine the required spring force as follows; Draw the desired characteristic *(15)* *(17)* *(16)*, and the C_Q curve for those weights which move up and down with the sleeve *(18)*. These weights include that of the sleeve, of roller *(14)*, of the plate, and one half of the spring. Add the ordinates of this C_Q curve and of the C_W curve. Find the differences between this combined curve and the desired final C curve. These differences divided by the ordinates of that C_Q curve which belong to $Q = 1$ furnish the spring forces. The first attempt may result in impossible spring dimensions, or in improper utilization of the available space, but it will show in which direction the assumptions of the resulting C curve must be altered.

The description of these governors and of the construction of the characteristic curves is rather sketchy, but illustrations 16, 21 and 22 contain sufficient information for studying the individual influence of the C_W and C_Q curves.

When a governor governs a prime mover, it is subjected to forces acting upon its sleeve, that is to say forces which either add to or subtract from Q. Such forces are caused by

(1) Adjustment of equilibrium speed.
(2) Resistance to motion caused by friction of valve gear.
(3) Unbalanced weights or fluid pressure.
(4) Integrated effect of valve gear reaction.

Speed adjustment is usually accomplished by the addition or subtraction of a constant force, see Chapter V, for instance by the shifting of a weight along a lever or by the tightening of a helical spring. The addition of a constant force at the sleeve produces a proportionate increase or reduction of the ordinates of the C_Q curve. If the C_Q curve and the C_W curve are both stable and of the same character as the combined curve, such as, for instance, in Fig. 16, the addition or subtraction of a constant force does not affect the static character of the governor. But if the C_Q curve has excess stability in order to make up for insufficient stability of the C_W curve, see Fig. 21, any addition to Q increases the stability, and a reduction of Q reduces the stability of the governor. If, finally, the C_Q curve has negative stability (which is the case in most spring-loaded governors, such as in 22) an addition to Q makes the governor less stable, and if carried far enough, makes it utterly unfit for regulation. This effect of the addition of a constant force is always the same, no matter whether it comes from item 1, 2, 3 or 4 of the above given classification.

The forces entering the governor through the sleeve are a fruitful source of annoyance and of triangular litigation between the builder of the governor, the builder of the prime mover and the user of the latter. Thus, the excessive clogging of regulating valves with tar in gas engines, the excessive and onesided tightening of stuffing box glands of regulating spindles in engines and turbines furnish complaints for which neither the governor builder nor the builder of the prime mover is responsible, except that, perhaps, the latter did not install a large enough governor.

If these forces are constant throughout the working range of the governor, they will only alter the equilibrium speed, but will not upset stability of regulation, provided that the C_Q and C_W curves have like character and stability.

As a rule, the forces resulting from items 2, 3 and 4 of the list on page 47 vary with the position of the governor sleeve. In gas engines there may be (and there usually is) more tar deposit near one end of the swing of the regulating valve than there is near the other end. So-called balanced regulating valves are frequently so designed that they appear at first glance to be balanced, but are really not so, because near the closed position forces arise from impact or jet action of the fluid. These forces are transmitted to the governor and, by their action at one end of the travel only, often seriously upset regulation.

For the purpose of this book it will suffice to study the action of two types of these so-called balanced valves. Fig. 23 shows a double beat poppet valve, the bottom seat of which is slightly smaller than the upper one, for assembling purposes. Each valve is cup shaped for purposes of strength and rigidity. When the valve is wide open, there is an upward thrust, because there is then maximum flow, and because the concave side offers more resistance to flow than the convex side does. With an almost closed valve, that is to say with very low rate of discharge, there is a downward thrust. In this position of the valve, the upstream steam pressure is much greater than the downstream throttle pressure, and a static pressure effect due to the difference of areas of upper and lower valve is felt.

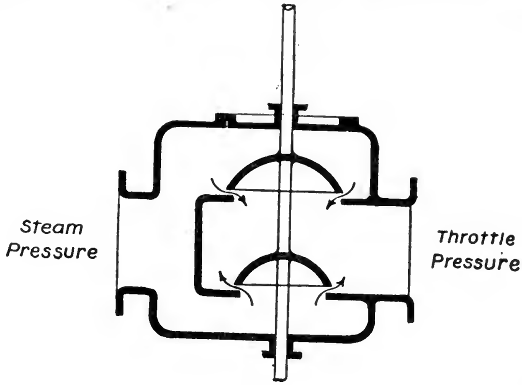


Fig. 23

So-called balanced butterfly valves are usually quite unbalanced, because the "nozzles" in which pressure is converted into velocity have very dissimilar shapes on the two sides of the valve. The lines (*1*) (*2*) (*3*) of Fig. 24 are intended to show respectively equal areas of flow and niveau surfaces of respectively equal pressure. It is evident that the pressure drop takes place with less travel at the bottom of the valve, so that a larger part of the bottom wing is exposed to high pressure than there is at the top. Consequently a force results in the direction of arrow (*4*). When the valve is opened, this force disappears.

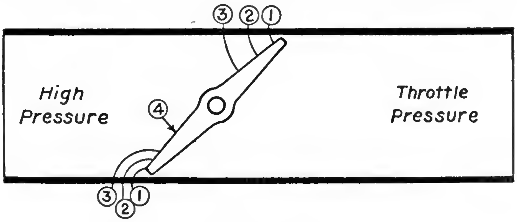


Fig. 24

The injurious influence which these unbalanced-in-certain-positions valves have upon satisfactory regulation has caused many engineers to keep the unbalanced forces away from the governor by interposing a relay, see Chapter XIII. But if a relay is not used, the forces in question should be made as small as possible by the builder of the prime mover; besides, the governor designer should be notified of the extent of these forces, so that he may provide for them in the design of the governor. In doing the latter he will again make use of the C_Q curve. If such a curve be drawn for $Q = 1$, the ordinate belonging to any given position of the governor sleeve can be multiplied by the force reacting upon the sleeve in the position under consideration. Repetition of this method for several positions of the sleeve furnishes a new C curve for the forces entering the governor from without. Combination of this curve with the other C curves must result in a curve with positive static fluctuation, as previously explained, if correct regulation is desired.

References to Bibliography at end of book: **11, 36, 53, 72, 73**

4. Influence of Shape of Centrifugal Weights on Characteristics. — The subject of the present paragraph is not of vital consequence for any centrifugal governor now on the market; neither is it likely to be of such consequence. It might, therefore, be passed in silence, if it were not for the human failing of reinventing designs which were relegated to the scrap heap long ago. And "Improving a governor" by changing the shape of the centrifugal weights is a popular subject for inventors.

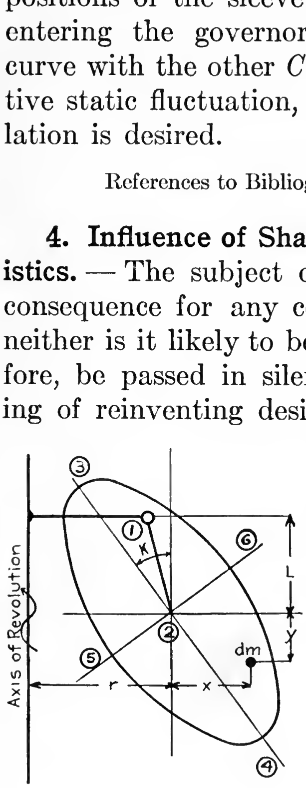


FIG. 25

In paragraphs 2 and 3 of the present chapter the theory of the characteristic curves of governors was developed for centrifugal weights of spherical or otherwise symmetrical shape, and the statement was made that for such shapes it was permissible to concentrate the whole mass of the weight in its mass center.

The correctness of this statement will now be analyzed.

In Fig. 25 the oval shape represents a centrifugal weight with mass center (*2*), and with suspension point (*1*). The moment of the centrifugal force of a small mass dm about the suspension point is $dM = dm\, u^2 (r + x)(y + L)$. For meaning of r and L see illustration. The moment of the whole mass is the integral

$M = u^2 \int dm\,(r + x)\,(y + L)$, or

$M = u^2\, r \int dm\, y + u^2\, r\, L \int dm + u^2 \int dm\, x\, y + u^2\, L \int dm\, x.$

By definition of the mass center, $\int dm\, y = 0$, and $\int dm\, x = 0$. It is also evident that $\int dm = m$, so that the centrifugal moment of the weight is reduced to $M = u^2\, r\, L\, m + u^2 \int dm\, x\, y$.

In the preceding paragraphs centrifugal force had been given as $m\, r\, u^2$ and centrifugal moment as $u^2\, r\, L\, m$. Evidently, the actual moment differs by the amount $u^2 \int dm\, x\, y$, and the equivalent centrifugal force which, when acting at the mass center, produces the correct moment is

$$C = r\, u^2 \left(m + \frac{1}{r\, L} \int dm\, x\, y\right) \ldots\ldots\ldots\ldots (1)$$

For all shapes which are symmetrical with regard to two axes at right angles to each other, $\int dm\, x\, y = 0$. Such shapes are the circle, the square, the octagon, etc. For centrifugal weights of any one of these forms the assumption that the mass may be concentrated in the mass center is, therefore, correct. For all other shapes the theories of paragraphs 2 and 3 must be modified.

To carry out this modification, it is desirable to narrow down somewhat the extent of the investigation, because the value of the correction $\frac{1}{rL} \int dm\, x\, y$ in equation (1) cannot be weighed or appreciated in its general form. If the investigation is limited to weights whose thickness h is constant at right angles to the plane in which they swing, the correction can be put into a form which presents a fairly clear view of the influence of the shape. To this end we use the principal axes *(3) (4)* and *(5) (6)* of the plane section of the swinging weight as a new system of coördinates.

Let m' = mass of unit volume in centrifugal weight; J_s and J_l = the smallest resp. largest moment of inertia of the plane section of the centrifugal weights; k = the angle between the axis of rotation and the axis to which the smallest moment of inertia is referred; then we have

$$\int dm\, x\, y = \tfrac{1}{2}\, m'\, h\,(J_l - J_s) \sin 2\, k \ldots\ldots\ldots (2)$$

For proof of this equation see Appendix, p. 214.

From equation (2) it follows that the difference of the two principal moments of inertia and the angle k of the inclination of the axis determine the extent of the correction. For $k = 0°$, and for $k = 90°$, the value of the correction is zero. The same is true for any value of k, if $J_s = J_l$, which bears out the statements made above concerning symmetrical shapes. The difference between J_s and J_l is the greater, for a given area of section, the more oblong the shape of the weight.

Let $C_p L = M_p$ be the centripetal moment — caused by spring forces, weights, etc. — which is to be balanced by the centrifugal moment M_c, and let C_p be the equivalent centripetal force referred to the center of each centrifugal mass; then

$$C_p L = m\,r\,u^2 L + \tfrac{1}{2}\,m'\,h\,u^2 (J_i - J_s) \sin 2\,k,$$

from which

$$u^2 = \frac{C_p}{m\,r + \dfrac{1}{2\,L} m'\,h\,(J_l - J_s) \sin 2\,k} = \frac{C_p}{m\,r}\,\frac{1}{1 + \dfrac{m'\,h\,(J_l - J_s) \sin 2k}{2\,L\,m\,r}} \quad (3)$$

It will be remembered that, for symmetrical centrifugal weights, the static properties of a governor were found from its characteristic by means of the equation $u^2 = \dfrac{C}{m\,r} = \dfrac{\tan i}{m}$.

Equation (3) teaches that this simple relation does not hold for governors with oblong weights. However, a characteristic may be so drawn with ordinates C' that $u^2 = \dfrac{C'}{m\,r}$, provided that we make

$$C' = \frac{C_p}{1 + \dfrac{m'\,h\,(J_l - J_s) \sin 2\,k}{2\,L\,m\,r}} \quad \ldots\ldots\ldots\ldots (4)$$

Evidently, the relations between $\sin 2\,k$, L and r in conjunction with the value of $(J_l - J_s)$ determine the difference between C_p and C'. In general, the difference is greatest for $k = 45°$. Whereas for symmetrical shapes of weights $\tan i = \dfrac{C}{m\,r} =$ constant, that is to say, a straight line characteristic passing through the origin, means an isochronous governor, it does not convey the same meaning for a governor with oblong weights. On the contrary, the characteristic for an isochronous governor with oblong weights is curved, the magnitude of the curvature depending principally upon $(J_l - J_s)$.

Equations (3) and (4) express the conditions quite clearly from a mathematical standpoint. A less comprehensive, but very much simpler explanation of the effect of the shape of the centrifugal weights upon the speed and upon the shape of the characteristic can be gained by studying the effect of splitting a concentrated centrifugal mass into two concentrated masses some distance apart and joined by a massless rod.

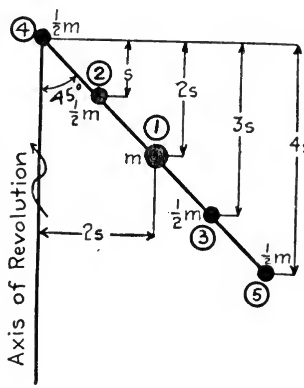


FIG. 26

In Fig. 26 the centrifugal moment of a mass m at point (*1*) is $M = m\,u_1^2\,4\,s^2$. If the mass m is split into two masses $\frac{1}{2}\,m$, located at points (*2*) and (*3*), the moment appears in the form $M = \frac{1}{2}\,m\,u_2^2(1^2 + 3^2)\,s^2 = m\,u_2^2\,5\,s^2$. If, finally, the masses $\frac{1}{2}\,m$ are located at points (*4*) and (*5*), the moment becomes $M = \frac{1}{2}\,m\,u_3^2(0^2 + 4^2)\,s^2 = m\,u_3^2\,8\,s^2$. The moment M remains the same in all three cases, because the centripetal (in this case gravity) moment is the same. The speeds are found from the foregoing equations by solving for u.

$$u_1 = \sqrt{\frac{1}{4}}\ \sqrt{\frac{M}{m\,s^2}}, \qquad u_2 = \sqrt{\frac{1}{5}}\ \sqrt{\frac{M}{m\,s^2}}, \qquad u_3 = \sqrt{\frac{1}{8}}\ \sqrt{\frac{M}{m\,s^2}}$$

The oblong shape of weight evidently lowers the equilibrium speed, if the weight is arranged as in Fig. 26.

The reverse action takes place if the long axis of the mass is placed as in Fig. 27. The moment equation for the mass at point (*1*) is $M = m\,u_1^2\,4\,s^2$. For two masses $\frac{1}{2}\,m$ at points (*2*) and (*3*) it is $M = \frac{m}{2}\,u_2^2(3 + 3)\,s^2$, and for two masses $\frac{1}{2}\,m$ at points (4) and (5) it is $M = \frac{m}{2} u_3^2 (0 + 0) s^2$, whence the speeds

$$u_1 = \sqrt{\frac{1}{4}} \sqrt{\frac{M}{m s^2}}, \quad u_2 = \sqrt{\frac{1}{3}} \sqrt{\frac{M}{m s^2}}, \quad \text{and} \quad u_3 = \sqrt{\frac{1}{0}} \sqrt{\frac{M}{m s^2}}.$$

In the arrangement of Fig. 27 the oblong shape of the weight raises the equilibrium speed.

The influence of the shape of oblong weights may be utilized by skillful designers for obtaining weight-loaded governors of small static fluctuation and constant stability. Governors based upon this principle were designed several decades ago. The best known example is the cosine governor. To-day, these governors can only be found in textbooks and mathematical treatises. For a given strength, oblong swinging weights increase the mass of a governor, and reduce its promptness (see Chapter IV). But governors whose mass is large compared to their strength produce great speed fluctuations whenever the load changes. This fact, the correctness of which will be proved in paragraph 2 of Chapter IX, by the introduction of the traversing time T_g, has practically eliminated the governor with oblong swinging weights.

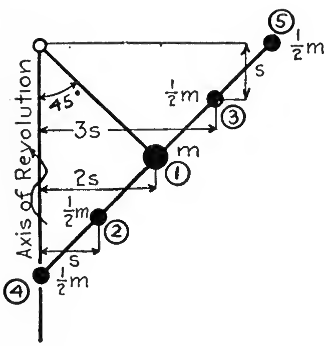


FIG. 27

In shaft governors the shape of the weight has no influence on the speed or on the characteristic (see Chapter VI).

References to Bibliography at end of book: **36, 73.**

CHAPTER IV

PROMPTNESS AND TRAVERSING TIME

In the chapter treating of the relation between governor and prime mover, proof will be furnished that closeness of regulation requires quick motions of the governor after a disturbance of equilibrium.

The property of a governor to move quickly to a new position has been termed "promptness" by French engineers. Unfortunately, promptness, like sensitiveness, is hard to define.

On the other hand there exists a convenient term which is closely related to promptness, can be correctly defined, and appears in all dynamic calculations on governors; this term is the "traversing time." Its meaning is exhibited most clearly by the following illustration: Let the centrifugal weights of a non-rotating, frictionless governor be moved to their extreme outward position and then be suddenly released. Starting with zero velocity, they traverse the path to their extreme inward position with increasing speed, under the influence of the centripetal forces or moments, in the "traversing time." From mechanics it is known that to make this time small, we must make the acting forces great, the inertia of the moving masses small and the distance-to-be-traversed short.

If either the centripetal moment M_p (about any one axis), or the moment of inertia of the governor parts (about the same axis) vary widely, determining the traversing time requires either a graphical integration, or else a dividing-up of the total travel into sections over each of which the variable quantities may be considered as constant. In all practical work, however, it is customary and permissible to replace these variables by their mean values. With this simplification the traversing time can easily be calculated from the equation which holds for constant acceleration, namely

$$T_g = \sqrt{\frac{2\,J\,i}{M_p}} = \sqrt{\frac{2\,m_r\,b}{C}} = \sqrt{\frac{2\,m_a\,s}{P}} \quad \ldots\ldots\ (1)$$

In this equation

T_g = traversing time

J = moment of inertia of governor parts, M_p = centripetal moment, i = total angle through which the governor parts swing — referred to the same axis of rotation

m_r = equivalent mass of governor parts referred to radial travel

m_a = equivalent mass of governor parts referred to axial travel

b, s, = total radial, resp. axial travel

C = centripetal force

P = strength of governor.

Smallness of the traversing time as a measure of promptness might be objected to, because under actual working conditions the moment M_p resp. the forces C or P are not available in full strength. Neither do the governor parts traverse their whole range of travel under ordinary changes of load. Any such objection overlooks the fact that both the available fraction of the force or moment and the fraction of the travel to be traversed are proportional to the change of load, so that these fractions cancel in the equation for the traversing time. Hence the time T_g is a true measure for the slowness of the frictionless governor, and $1/T_g$ is a measure for the quickness or promptness of the governor.

In a purely centrifugal governor the time T_g can be made small by increase of C and by reduction of m_r; fulfillment of both of these conditions excludes weight-loaded governors and necessitates spring-loaded governors. In the latter type, increase of C and reduction of m_r are accomplished by great angular velocity and by large orbit of centrifugal masses. Finally, smallness of path (angular path i, or linear path s, b) increases the promptness. Limitations are furnished by cost. Evidently larger and larger orbits and higher angular velocities cause mechanical difficulties and increased cost.

The reduction of mass which is so desirable for the sake of promptness is very undesirable, if the governor is subjected to displacing forces or reactions caused by the valve gear (see Chapter VIII). If a massive and yet prompt governor is desired, it must be made very powerful, or else tangential inertia must be utilized. Inertia masses increase J, m_r or m_a in equation (1) and are, therefore, injurious to promptness, as expressed by that equation. On the other hand, tangential

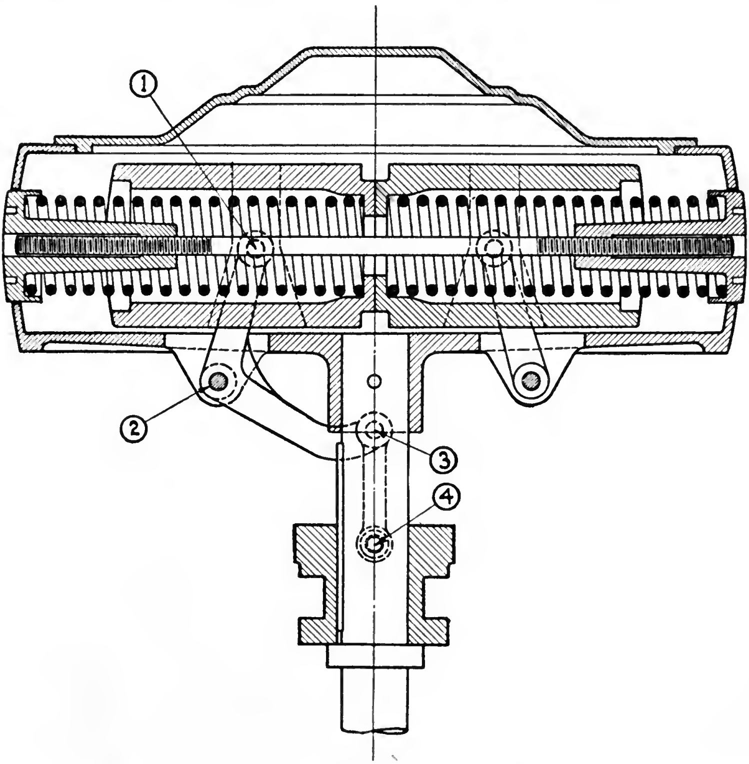


FIG. 28

inertia furnishes a regulating force whose value, as was shown in paragraph 2 of Chapter II, grows with the suddenness of the change of load. To what extent this feature increases the promptness cannot be treated without knowledge of the dynamics of regulation. See paragraph 3 of Chapter IX.

The following brief calculation of the traversing time of a spring-loaded governor will illustrate the use of equation (1):

In the governor shown in Fig. 28, the equivalent mass

$$m_r = \underset{(1)}{\frac{157.5}{32.2}} + \underset{(2)}{\frac{\frac{1}{3} \times 13}{32.2}} + \underset{(3)}{\frac{\frac{1}{2} \times 16.5}{32.2}} + \underset{(4)}{\frac{40}{32.2}} \times \left(\frac{3.25}{2.81}\right)^2 = 6.95$$

In this equation (1) is the mass of the centrifugal weights,
(2) is part of the mass of the bell cranks,
(3) is part of the mass of the springs,
(4) is the equivalent mass of the weight of the sleeve and of other parts moving with it. The factor $\left(\frac{3.25}{2.81}\right)^2$ reduces that mass to the radial travel of the weights.

Radial travel $b = 2.81''$

Mean centripetal force $C = 1200$ pounds.

With these values — which were found by measurement of a governor in the Mechanical Engineering Laboratory of the Carnegie Institute of Technology — we obtain the traversing time

$$T_g = \sqrt{\frac{2 \times 6.95 \times 2.81}{1200 \times 12}} = .052 \text{ second.}$$

For the greatest accuracy it would be necessary to take cognizance of the fact that the weights move on an arc, and that, for this reason, the equivalent mass should be multiplied by a factor greater than one; but this factor is in the present case so near one that it is permissible to neglect it.

The traversing time of a weight-loaded governor is, of course, much greater, unless exceedingly short radial travels of the centrifugal masses are used.

In the governor shown in Fig. 56 the average strength P is 500 pounds. The axial travel of the collar is $s = 3.25$ inches. The equivalent mass is that of all the parts moving with the collar plus that of the balls, the latter multiplied by the ratio $\left(\frac{\text{travel of balls}}{\text{travel of collar}}\right)^2$. The total equivalent mass is $\frac{542}{12 \times 32.2}$ in inch, pound, second units. Hence, the traversing time is

$$T_g = \sqrt{\frac{2 \times 542 \times 3.25}{12 \times 32.2 \times 500}} = .135 \text{ second.}$$

Reference to Bibliography at end of book: **36.**

CHAPTER V

ADJUSTMENT OF EQUILIBRIUM SPEED

(1) Slight errors in calculation or workmanship, and variation in the material of spring steel prevent governors from running at the intended speed. Unforeseen reaction from the valve gear has the same result.

Such conditions call for a single adjustment or a short series of adjustments.

(2) For synchronizing purposes or for adjustment of voltage the speed of engines or turbines must be varied up to 3 or 4% either way from mean. Regulation must be equally exact with any one of the adjusted mean speeds.

(3) For the regulation of pumping machinery, whether of the displacement or of the velocity type, the equilibrium speed must be adjusted within wide limits; the ratio $\frac{\text{highest equilibrium speed}}{\text{lowest equilibrium speed}}$ occasionally reaches values of 3 or 4. Fortunately the requirements for exactness of regulation are not so strict in this case.

The adjustment or adjustments (1) can be made while the prime mover and the governor stand still, whereas the adjustments (2) and (3) must be made with the governor in motion.

For the adjustments (1) a number of methods exist, such as change of mass of centrifugal weights (drilling of holes, filling with lead), change of spring tension, variation of spring scale (grinding off the outer-diameter fiber of helical springs, cutting off end coils, etc.), change of lever arm of spring, and others.

For the adjustments under (2) and (3) the following means are available :

(1) Change of speed ratio between prime mover and governor (equilibrium speed of governor remains constant) attained by

(a) change gears,

(b) friction discs or conical pulleys;

either of these subdivisions can be constructed in different forms mechanically.

(2) Change of equilibrium speed by variation of axial force Q, acting on collar. Shaft governors need no collar while governing prime movers (see Chapter VI), but an element, equivalent to a collar, is occasionally provided solely for the purpose of speed adjustment.

Variation of Q may be obtained by

(a) addition or subtraction of mass (water, etc.) or of fluid pressure

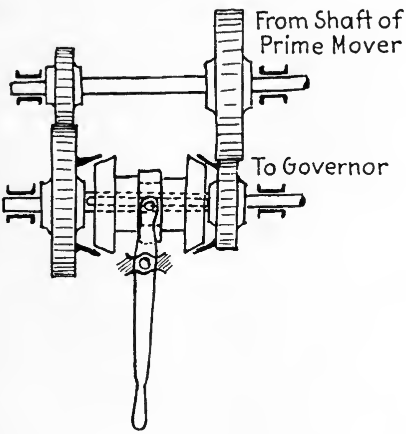


FIG. 29

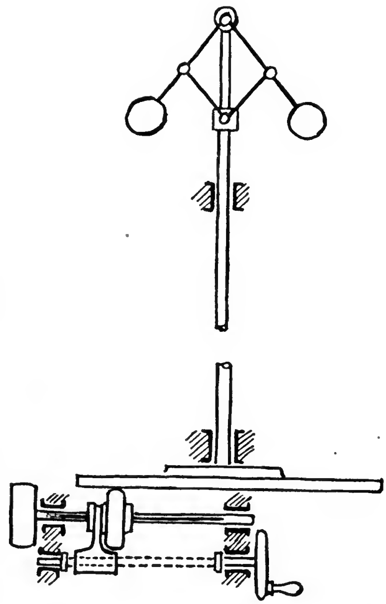


FIG. 30

(b) variation of the lever arm of a force

(c) variation of spring tension

(3) Variation of governor mechanism, for instance, displacement of movable fulcrum (for illustration see Fig. 39).

(4) Change of relative position between power controlling mechanism and position of governor (for illustration see Fig. 40).

Method (1) makes possible a wide range of speed adjustment. Change gears, one design of which is diagrammatically shown in Fig 29, act in steps and must be used in connection with other adjusting means to make gradual adjustment over the whole range possible. Even then there is a jump whenever the change in the gears is made, so that change gears have their limitations.

Frictional devices (for diagrammatic illustration see Fig. 30) are very satisfactory, provided that they are large enough to prevent serious slipping. The latter is caused particularly by the frictional resistance of the governor and by cyclical speed fluctuations.

Frictional devices have come into disrepute for large prime movers, because designers have hesitated to use the unsightly large friction discs which are necessary for success and have put "more decent looking" smaller sizes on the engines. The result in such cases invariably is slippage, wear, unsatisfactory regulation and thoughtless condemnation of the entire principle.

Detail description of these devices and calculation of necessary sizes is prohibitive on account of the endless variety of possible designs and arrangements.

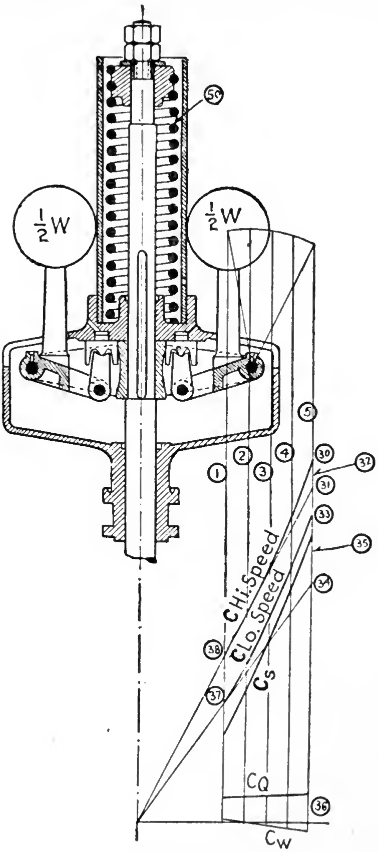


FIG. 31

Means No. (2), that is to say variation of axial force Q, is by far the most common. In paragaph 3 of Chapter III the use of the C curves for an investigation of the stability of the governor with varying values of Q was indicated. It will now be further illustrated by means of an example which is based on the type shown in Fig. 22; this type has been widely adopted in steam turbine practice.

The governor is shown in Fig. 31. For the numerical calculation the following values were used:

Weight of both centrifugal masses $W = 15$ pounds
Weight of axially moving parts (including $\frac{1}{2}$ weight of spring) = 45 pounds

Travel of collar $s = 2$ inches

Initial tension of spring = 150 pounds and 150 + 85 pounds

Scale of spring = 140 pounds per inch.

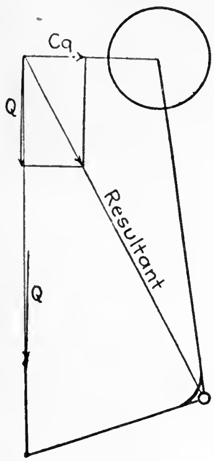


Fig. 32

The characteristics C_Q are easily drawn if it is remembered that the direction of Q, C_Q and of their resultant are given (see Fig. 32) because the latter must pass through the fixed fulcrum. The detail construction of the characteristics is given in the Appendix, p. 215. The characteristics proper are shown in Fig. 31.

From $u = \sqrt{\frac{C}{m\,r}}$ it follows for position (3) $n_1 = 268$ r.p.m. (initial tension = 130 lb.), and $n_2 =$ 300 r.p.m. (initial tension = 215 lb.) so that the tightening of the spring has produced 12 % increase of speed. The static fluctuation (see page 24) is found for the small initial tension from $p_1 = \frac{\overline{(33)(34)}}{2 \times (35)(36)}$ which equals 15.1 %; in the same way we find $p_2 = \frac{(30)(31)}{2 \times (32)(36)}$ which equals 5.8 %. Evidently the static fluctuation is considerably reduced by the tightening of the spring. The stability which was positive over the whole range for the small initial tension disappears near the inner positions for the great initial tension (see points (*37*) and (*38*)), so that the governor is worthless between positions (*1*) and (*2*) for the great initial tension, that is for the high speed.

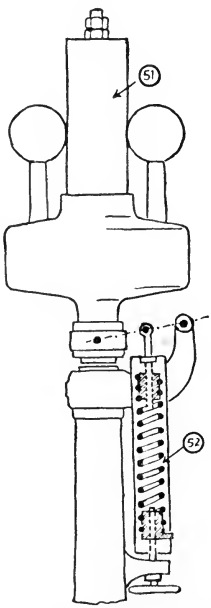


Fig. 33

In this calculation it was assumed that the variation of Q is produced by the tightening of the main spring (*50*) (Fig. 31), but the diagram, and with it the resulting variation of p, remain just the same, if the tightening of the main spring is replaced by the addition of a weight to the axial forces, or if the main spring (*50*) of Fig. 31 is replaced by two springs (*51*) and (*52*) (see Fig. 33), one of which is at rest, accessible, and is tightened for the purpose of speed adjustment. In Chapter IX proof will be given that too small a value of p (static fluctuation) interferes with proper regulation. In Fig. 31 the static fluctuation (as measured by the tangent

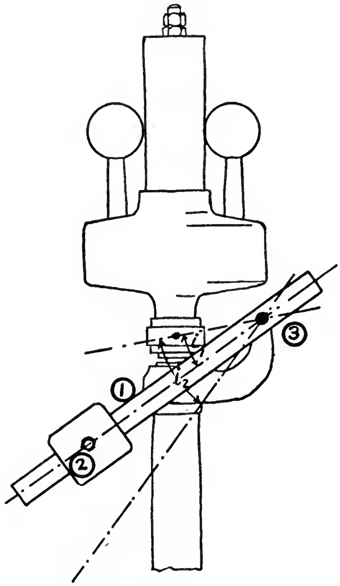


Fig. 34

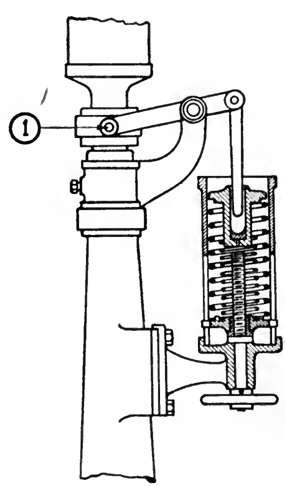


Fig. 35

to the characteristic at point (*38*)) has become zero or even negative. This condition, which arises in all similar types of governors, whenever a constant force is added at the collar, is serious enough to call for some different method of adjustment.

Conditions will be better if the added force Q at the collars varies with the movement of the latter in some ratio to the spring force.

If addition of mass is permissible, an arm (*1*), carrying a movable weight (*2*), may be keyed on the rocker shaft (*3*) making angles i_1 or i_2, etc. (see Fig. 34), with the governor lever. In this manner, the added effective force is greater in the top position of the collar than it is in the bottom position, and the ratio of $\frac{\text{force added in top position}}{\text{force added in bottom position}}$ remains constant, no matter what the added force may be. The angle i may be so selected that this ratio fits the scale of the spring in the governor quite closely.

In spite of these static advantages the just described method is seldom used, because the inertia of the sliding weight is detrimental to the promptness of the governor.

A similar static advantage (without the drawback of inertia) is attained by the use of a "speeder" with several springs of different free lengths, as shown in Fig. 35. Such a device furnishes a load which grows faster as point (*1*) rises. Tightening up the speeder does not add a constant force, but a variable force, as may be seen by comparing the working ranges (*3*), (*4*) and (*5*) of Fig. 36 with each other. The same result can be obtained by replacing the two or three springs with a single volute (cone-shaped) spring; it too, as is generally known, has the property of a scale growing with the deflection, because the coils of larger diameter deflect first and come up solid against the stop, leaving the additional deformation to smaller, and consequently stiffer coils. The number of schemes which may be employed for adjusting the speed of this type of governor is by no means exhausted herewith. It is possible to change lever arm and initial tension of the speeder spring in such a way that the static fluctuation remains practically constant. Figure 37 illustrates diagrammatically such an arrangement. The parts drawn in heavy black lines move with the governor and swing about fixed point (*4*). Spring (*5*) which is fastened to fixed point (*6*) takes hold at a point which

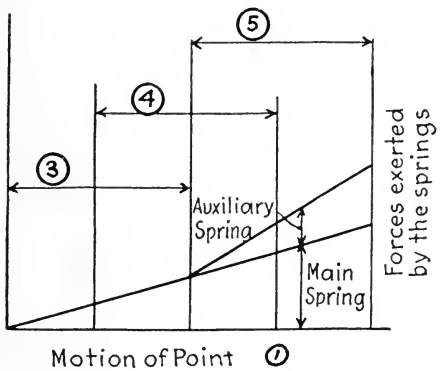


FIG. 36

is movable relative to the levers. A few positions (*1*), (*2*), (*3*) have been indicated by dotted lines. The determination of the proper location of point (*6*) and of points (*1*) and (*7*) is not difficult, but rather tedious.

The speed adjustment of governors of the Hartnell type has been treated in detail, not because the Hartnell type deserves special attention, but because it furnishes

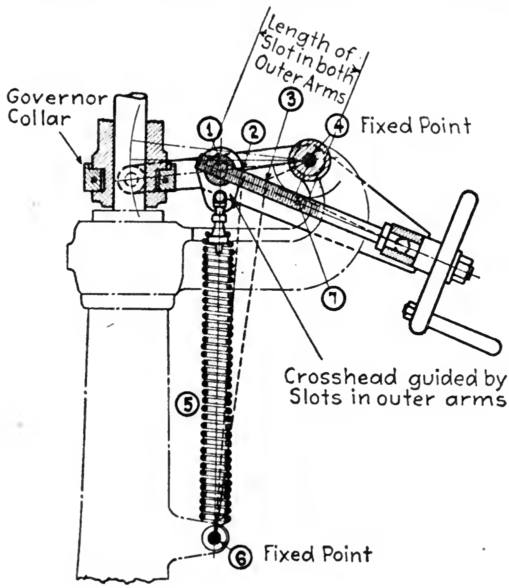


FIG. 37

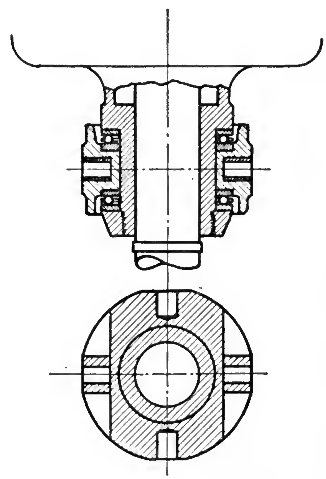


FIG. 38

an example of how the statics of speed adjustment should be treated in any type of governor.

In connection with speed adjustment by variation of axial force Q three additional features deserve mention, namely:

(1) friction of governor collar,

(2) detention by governor friction due to forces at collar,

(3) variations of governor strength due to forces at collar.

(1) Whenever forces are transmitted through a governor collar, the latter becomes a step bearing, the location of which makes lubrication somewhat difficult. If lubrication is neglected, heating and wear result. Ball and roller bearings have given better service in this place than sliding bearings. Figure 38 shows a successful design.

From the standpoint of avoiding trouble with the collar bearings it is desirable to push up on the collar, if great speed changes are desired, because then the heaviest thrust is coupled with the slowest speed.

(2) Many of the modern governors are so designed that the joints are free from forces caused by centrifugal action. If, however, 5% speed adjustment is desired, approximately 10% of the centrifugal force must be transmitted through joints due to the added force at the collar. In addition, some force must be transmitted through the collar even at normal speed, if a speeder spring is used, so that on an average 12% to 15% of the centrifugal force must be transmitted through the joints if 5% speed adjustment is desired. This fact should not be overlooked with regard to governor friction and wear of joints.

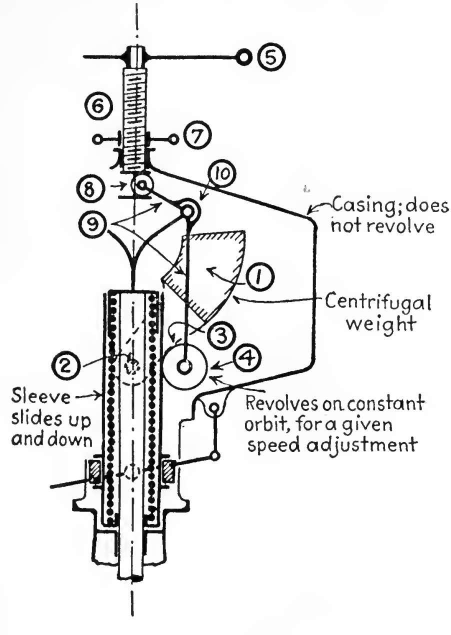


Fig. 39

(3) Forces at the collar affect the strength of the governor. This is usually of small consequence, except when the axial force pushes up and considerably reduces the speed. In this case the strength of the governor is very much reduced, and the governor does not handle valve gears offering great friction resistance.

The means mentioned under No. 3, namely, variation of governor mechanism, is seldom practiced. A brief description of one example will illustrate the principle.

In Fig. 39, (*1*) is the mass center of a centrifugal weight. The weight is guided by the vertical path of point (*2*) and by the gliding of surface (*3*) on roller (*4*). This latter path can be varied by the turning of handwheel (*5*) which, by means of screw (*6*) with lock nut (*7*), raises or lowers point (*8*). This motion, by means of revolving lever (*9*) with fulcrum (*10*), moves roller (*4*) in or out. Radial displacement of weight (*1*), and lever arm of centrifugal force are varied by this adjustment, resulting in a wide range of speed adjustment; the maximum mean speed is more than twice the minimum mean speed. The governor just described was invented by Strnad (Germany).

Although the outward appearance of the governor is simple,

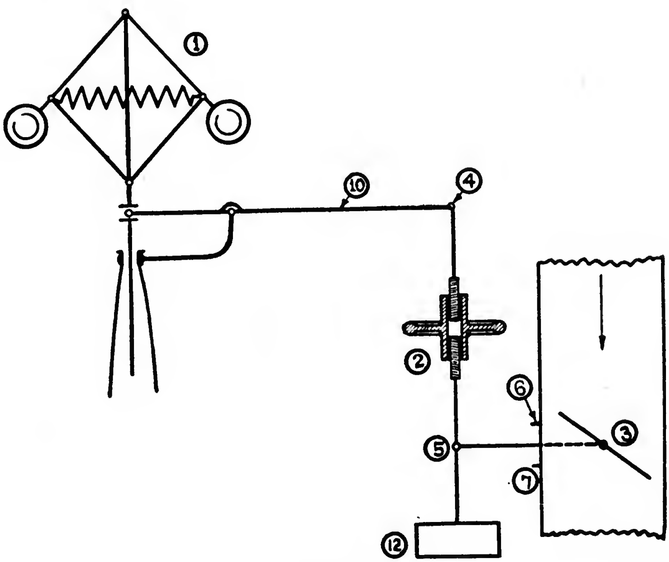


Fig. 40

the interior is complicated and expensive. The joints transmit heavy forces, which means that they are subject to wear.

Means No. 4, namely, change of relative position of governor and power-controlling mechanism, is often used. Figure 40 illustrates this method diagrammatically. By means of the turnbuckle (*2*) the distance (*4*) (*5*) may be adjusted. During this adjustment the governor (*1*) remains at first stationary, because it is held in equilibrium between centrifugal and centripetal forces. The resulting change in the position of the power-controlling device (*3*) unbalances the equilibrium between power generation and power absorption. Hence the speed varies, and the governor, in moving towards a new position of speed equilibrium, shifts (*3*) until power equilibrium is again attained. If the difference in speed does not require a difference in the position of controlling device (*3*), the latter will return to its original position; in all other cases it will not quite return to that position. In any event the position of governor (*1*) has been changed, and with it the speed of the prime mover.

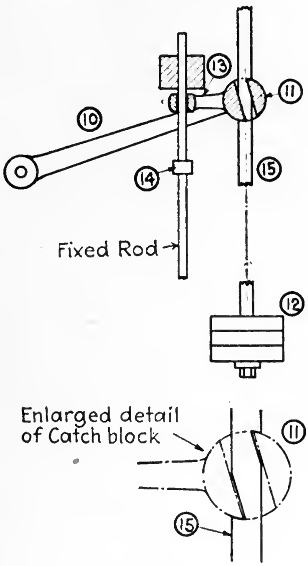


FIG. 41

It is necessary to limit the motion of the controlling device (*3*) by stops such as are diagrammatically indicated in (*6*) and (*7*) (unless the design of said device makes the use of additional stops superfluous), because the position of the governor is not fixed relative to the power-controlling device.

The magnitude of the speed adjustment which can be attained by this method depends upon the static fluctuation of the governor. If it be small, the limits of speed adjustment must be narrow. If the greatest speed of the governor is several times the smallest speed, very wide limits of speed adjustment can be obtained, but with high normal speed of the prime mover a removal of the load produces a dangerously high no-load speed. To insure safety under these conditions, automatic disconnecting devices have been designed which disconnect the governor from the power-controlling device, as soon as a predetermined maximum speed has been reached, and release a force which shuts off the power. Figure 41 shows a frequently used device of this

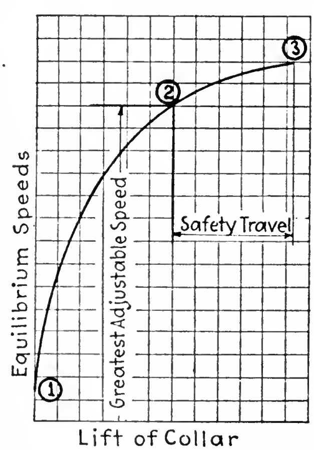


FIG. 42

type;[1] the illustration represents a detail of point *(4)* of Fig. 40. As soon as rod *(10)*, Figs. 40 and 41, reaches a nearly horizontal position corresponding to the highest safe speed of the governor, the rod *(13)* of catch block *(11)* strikes collar *(14)* and releases the rod *(15)* carrying weights *(12)*. These weights, in dropping, close the power-controlling valve. Before the engine can be started again, the rod *(15)* must be lifted to again engage catch block *(11)*.

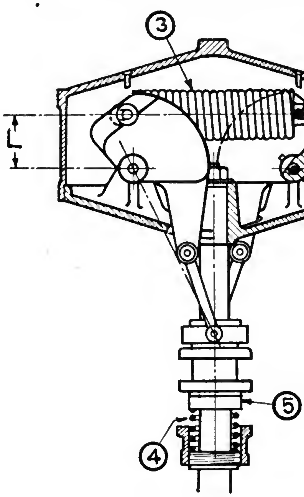


Fig. 43

The necessity of a disconnecting device can be avoided by a governor whose speed curve has the shape indicated in Fig. 42. Part *(1)* *(2)* of this curve is used for speed adjustment, as described. Part *(2)* *(3)* is used to bring the power-controlling device to no-load position in case of accident. Such a governor was designed by Professor J. Stumpf (Germany). It is shown in Fig. 43. In this governor skillful application is made of variation of leverage of main spring *(3)*, and of lost motion of spring *(3)* and spring *(4)* in this manner: In the low positions of collar *(5)* the spring *(4)* pushes upward, allowing a low speed to raise the governor. Spring *(4)* is very short and soon gets out of action with rising speed. On account of slots *(6)*, the main spring *(3)* is ineffective in the bottom part of the governor travel. With rising collar it takes hold and vastly increases the equilibrium speed. In the higher position of collar *(5)* the force of spring *(3)* increases slowly, whereas its lever arm L decreases rapidly, which causes the speed to change very little for the upper part of the travel of collar *(5)*.

Another governor with a speed curve similar to that shown in Fig. 42 is described in Chapter XII.

[1] This device, as well as the whole method No. 4, was originated and developed by Mr. K. Weiss (Switzerland). The method was first used in the United States by Mr. B. Nordberg, of Milwaukee, Wis.

Governors of this type are useful for pumping machinery only. The modern tendency in such machinery is to adapt the supply to the demand and to regulate either for constant pressure or for constant rate of delivery. Forces other than centrifugal are used for this purpose. Speed-changing governors of that type are, therefore, not considered in the present chapter, but are discussed in Chapters XI and XII.

References to Bibliography at end of book: **30, 36, 51, 73.**

CHAPTER VI

SHAFT GOVERNORS

1. Forces Acting in Shaft Governors.— The term "shaft governor" commonly denotes a governor the centrifugal weights of which move in a plane at right angles to the axis of the shaft or spindle, and which adjusts the position of an eccentric or crank relative to other revolving parts. Figures 4, 5, 8, 9, 44 illustrate shaft governors.

The arrangement of centrifugal and inertia weights in shaft governors, their method of fastening, the linkage between weights and eccentric, the arrangement of springs, and the transmission of motion to the valve gear belong in treatises on machine design and engine design. They do not form subjects of the present book.

Spindle governors could be studied apart from the active forces which the valve gear might impress upon them, because most spindle governors have to overcome mainly passive resistance, compared to which the small existing reaction is usually of little or no importance. Shaft governors, on the other hand, have to move valves directly against inertia forces, against varying friction forces and against varying unbalanced steam pressure. These forces react upon the governor and tend to disturb the equilibrium between centrifugal forces and spring forces. Not only does this reaction induce vibration, but it has an "integrated effect," that is to say, a definite average force or moment which is added to the spring moment and changes the speed from what it would be, if that effect were zero (see also pages 38 and following). The average effect of the reacting force or moment is, in practically every shaft governor, large enough to upset static equilibrium, and must therefore be considered either by theoretical study, or else by experiment.

In view of these disturbing influences, a complete static calculation of a shaft governor involves the following steps:

Select a static fluctuation p, about the permissible value of which see Chapter IX, and compute the various moments tending to displace the centrifugal weights, viz. centrifugal moments of all masses, moments due to inertia of reciprocating masses, due to valve friction, eccentric strap friction, and that due to unbalanced steam pressure, for three, or better, four positions of the weights. Then locate and dimension a spring so that its moment (negatively) coincides with the sum of all other moments. This latter problem is not difficult, if the

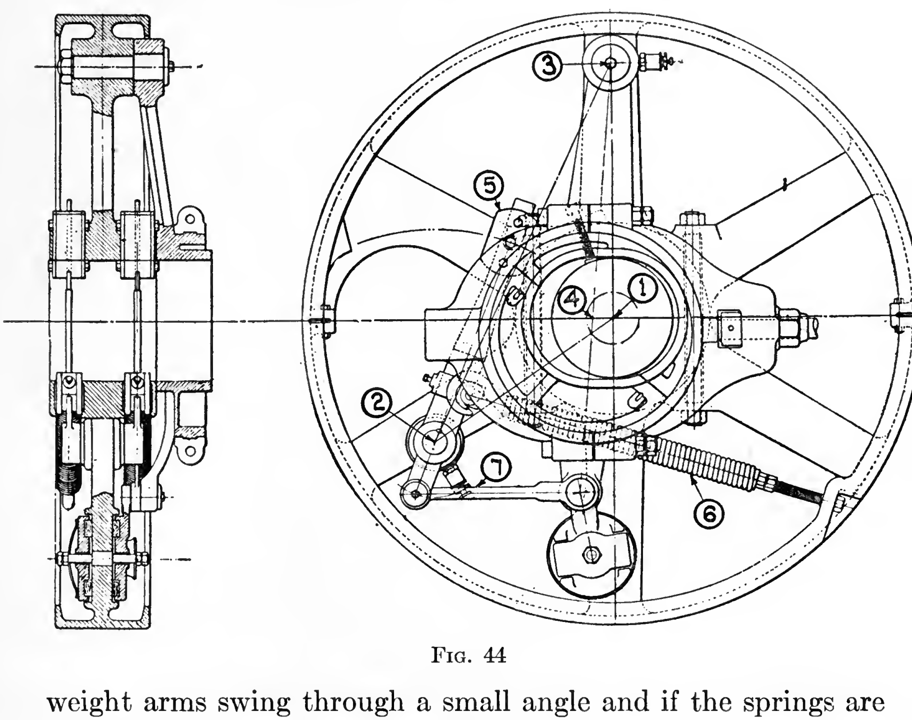


Fig. 44

weight arms swing through a small angle and if the springs are protected against disturbing deformations due to centrifugal force. If, however, the weight arms swing through a large angle, say more than 40°, or if the springs are allowed to bow out under the action of centrifugal force, no satisfactory coincidence of moments can be attained with one set of springs, and unusual expedients must be resorted to.

On account of the great variety of shaft governors on the market, and on account of the still greater number of valve gears which they control, it is impossible to follow out this method in all its details within the limitations of the present book. It is, therefore, advisable to study the underlying principles with the aid of one particular example. The careful study of one example will show the methods which are to be followed, with slight modifications, in all cases.

References to Bibliography at end of book: **1, 13, 24, 30, 36, 40, 50, 51, 58, 59, 68, 73, 74.**

2. Centrifugal Moment of Rotating Masses. — Figure 44 shows a simple type of governor suitable for study. This governor is used for controlling slide valves, including piston valves. The triangle *(1) (2) (3)* is fixed relative to the shaft — the center of which is *(1)* — and rotates with it. *(4)* is the center of the eccentric which is shifted across the shaft by the regulating forces, maintaining an approximately constant lead of the valve. The centrifugal weight *(5)* tends to fly out; it is restrained principally by the spring *(6)*, but to some extent also by the centrifugal force of the eccentric *(4)* and of its strap. *(5)* and *(4)* are joined mechanically by link *(7)*.

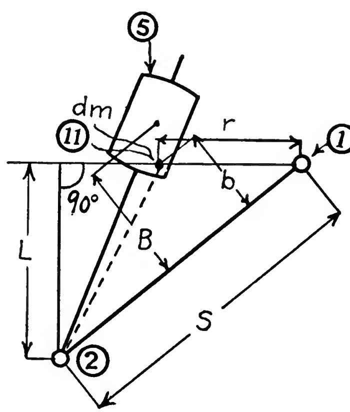


Fig. 45

For the sake of clearness some parts of the governor have been drawn diagrammatically in Fig. 45. With the notations of this latter illustration, the centrifugal moment of the small mass *dm* about the suspension point *(2)* is $d\,M_c = dm\,r\,u^2\,L$. But the area of the triangle *(1) (2) (11)* is $\frac{1}{2}\,r\,L = \frac{1}{2}\,s\,b$ which makes the element of the centrifugal moment

$$d\,M_c = dm\,s\,b\,u^2.$$

The moment of the whole centrifugal weight is

$$M_c = s\,u^2 \int dm\,b = m\,B\,s\,u^2 \quad \ldots\ldots\ldots\ldots (1)$$

where B is the perpendicular distance from mass center of weight *(5)* (including supporting lever) to line *(1) (2)*, and m

is the mass of the centrifugal weight (*5*), including lever. From this equation follows the fact that in shaft governors the distribution of the mass around the mass center of the centrifugal weight has no influence upon the speed. In this respect shaft governors differ from spindle governors (see paragraph 4 of Chapter III).

Equation (1) applies to any revolving mass in a shaft governor and furnishes the moment of the mass about its own suspension point. If applied to the eccentric, it furnishes the moment of the centrifugal force of its mass about point (*3*), of Fig. 44, which, however, can easily be converted into a moment

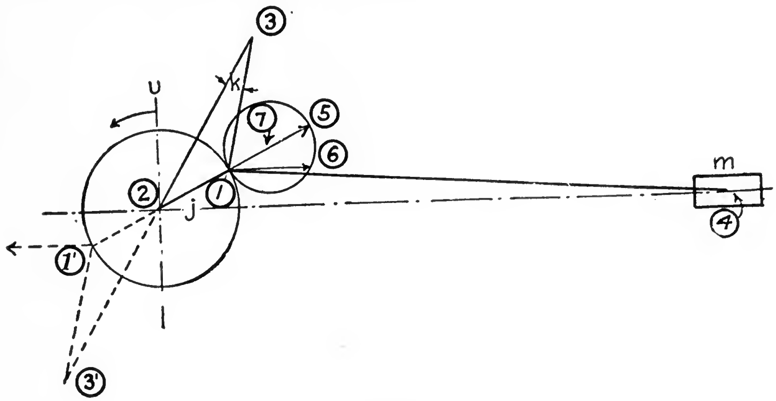


Fig. 46

about point (*2*) by means of the leverages of the connecting linkage.

The eccentric strap does not revolve, but each particle describes a path similar to an ellipse and exerts individual inertia forces, the integrated effect of all of which is transmitted to the eccentric proper. Generally speaking, the integration can only be accomplished by a tedious point by point process. Fortunately, the distribution of material in the eccentric strap is symmetrical about the center of the eccentric in the vast majority of cases. The whole mass may then be considered as concentrated in the center of the eccentric and be added to the mass of the latter.

3. Moment due to Inertia of Reciprocating Parts.—If the valves moved by the governor are driven by a cam, the integrated effect of the moment cause by their inertia can be ascertained only by a graphical integration, but if the valves are operated directly by the eccentric with practically sine harmonic motion, the integral may easily be found thus:

In Fig. 46, (*1*) is the center of the eccentric and (*3*) is the suspension point of the eccentric. (*3*) revolves about (*2*), which is the center of the shaft. The eccentric reciprocates a mass m_v (mass of the valve) by means of long eccentric rod (*1*)(*4*). The following derivation is based upon the supposition that mass m_v is not large enough to displace point (*1*) perceptibly from its circular path.

When (*2*)(*1*) coincides in direction with (*2*)(*4*), the inertia force is $m_v\, r\, u^2$. In the position shown in the illustration it is smaller. Usually the eccentric rod (*1*)(*4*) is very long compared to the eccentric radius (*1*)(*2*). The direction of the inertia force is then practically constant, and its magnitude varies with the cosine of the angle j. If (*1*)(*5*) represents $m_v\, r\, u^2$, then (*1*)(*6*) = (*1*)(*5*) $\cos j$ is the inertia force in the position shown in the illustration, and the circle with diameter (*1*)(*5*) is the locus of the terminals of the force vectors, referred to the rotating system.

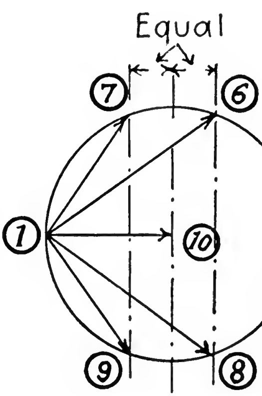


Fig. 47

In one revolution of the shaft the just mentioned circle is passed over twice. This will readily be seen by a study of the forces existing when the shaft has turned through 180°, see dotted position, because the forces in the two positions have the same magnitude and direction relative to the rotating system.

The average effect of a force vector describing a circle with constant angular velocity may be found by the following reasoning (see Fig. 47). Any four vectors (*1*)(*7*), (*1*)(*6*), (*1*)(*9*), (*1*)(*8*), located symmetrically as shown, have the average horizontal component (*1*)(*10*), whereas all the vertical components cancel. Since the same is true for any four symmetrically located vectors, (*1*)(*10*) is the average of all the force vectors, both with regard to magnitude and direction.

Application of this result to Fig. 46 means that one half of (*1*)(*5*), or (*1*)(*7*), or, in symbols, $\frac{1}{2}\, m_v\, r\, u^2$, represents the average of the inertia forces, both with regard to magnitude and direction. The same effect is produced by a mass $\frac{1}{2}\, m_v$, located in the center of the eccentric, and revolving with the latter. The effect of the inertia of the reciprocating valve is, therefore, easily taken care of by adding one half of its mass to the rotating mass concentrated at the center of the eccentric.

If mass m_v is big compared to the masses swinging in the governor, the path of point (*1*) is no longer circular. The equations of motion of the point (*1*) then lead to an elliptic integral which is quite difficult.

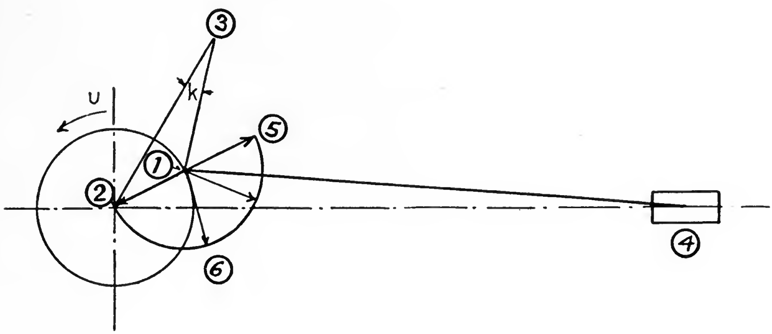


FIG. 48

References to Bibliography at end of book: **36, 40, 50, 58, 59, 74.**

4. Moment Due to Friction of Valve Gear.— The average reaction of valves having a sine harmonic motion may be found by a similar analysis. Figure 48 is a repetition of Fig. 46 with the exception that inertia forces have been replaced by friction forces. If, as before, we neglect the small angularity of the eccentric rod, the direction of the friction forces remains constant; hence this direction revolves uniformly with regard to the rotating system. In Fig. 48 the friction vectors have been entered. For a constant value of the friction force, the vector describes a semicircle (*5*)(*6*)(*2*), jumps back to (*5*), again describes the semicircle, etc. If the friction force is not constant, the semicircle is distorted.

The integrated effect of the friction force is represented by the average vector. In any case, including that of variable friction force, it can be found by graphical construction. In the case of constant friction force the terminal of the average vector is the mass center of the semicircular line (*5*)(*6*)(*2*). The magnitude of the average vector is therefore $2/\pi$ times the friction force. It acts at right angle to the diameter (*2*)(*1*).

The integrated effect of friction and of inertia must be determined for several values of the angle k, that is to say for several configurations of the governor. This is necessary for the purpose of equating the spring moment to the sum of all other moments over the whole range of the swing of the governor.

References to Bibliography at end of book: **36, 40, 50, 58, 59, 74.**

5. Moment due to Friction between Eccentric and Strap. — Friction between eccentric and strap is produced by the weight and the centrifugal force of the strap, by the forces transmitted through the eccentric rod (valve inertia, valve friction, unbalanced steam pressure), and by the pressure of the assembling fit between eccentric and strap. The combination of these forces for various angular positions of the whole governor and for a few positions of the parts in the governor can always be made graphically. The friction moment resulting from these forces is quite uncertain, because the coefficient of friction between eccentric and strap varies within wide limits with quantity and viscosity of oil. It is frequently observed that shaft governor engines have a different speed at starting up (when the oil is cold and thick) from that which finally establishes itself, after the oil has been heated by friction. Another uncertainty is introduced by the deformation of the eccentric straps. The latter are too often designed so weak that they cause an amount of binding and gripping sufficient to frustrate any calculation. Nevertheless, a brief account will be given of the distribution of frictional forces around the eccentric, because the seemingly erratic behavior of many shaft governors cannot be understood without the knowledge of these forces.

Figure 49 shows a governor eccentric and strap with the principal forces acting upon them, and Fig. 50 illustrates the distribution of forces by means of vectors having their origins at point (*1*).

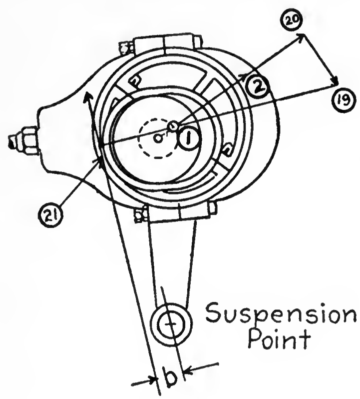
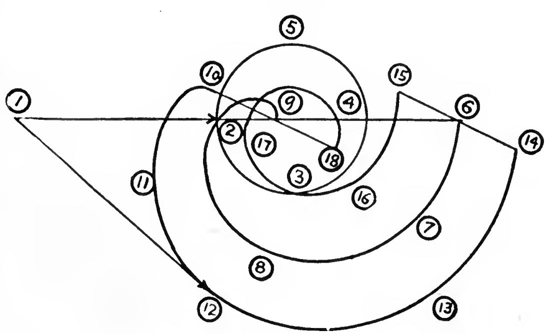


FIG. 49

For a given load, the centrifugal force of the eccentric strap has a constant magnitude and direction (*1*) (*2*) relative to the revolving eccentric. Valve inertia is, in most cases, a sine function and adds the circle (*2*) (*3*) (*4*) (*5*), compare also Fig. 46. The force due to valve friction, if the latter be constant, is also a circular function and acts in the same direction as valve inertia; but the function is discontinuous and changes its sign abruptly at the ends of the valve travel, see also Fig. 48. Addition of this force to those previously enumerated furnishes the vector terminal curve (*6*) (*7*) (*8*) (*9*) (*2*). The weight of the eccentric strap is constant in magnitude and in absolute direction, and, therefore, rotates continuously around the revolving eccentric. The same is true of unbalanced steam pressure, unless the latter be variable. The resultant of strap weight and of steam pressure changes the final locus of the terminus of the force vector to the curve (*14*) (*13*) (*12*) (*11*) (*10*) (*15*) (*16*) (*17*) (*18*) (*14*), etc. Thus (*1*) (*12*) is a vector picked out at random.



FG. 50

The correct method of finding the average friction moment from these forces would consist in determining the friction moment for each force vector and then averaging the friction moments thus found. This method would be tedious and is entirely out of place in view of the uncertainties with regard to the coefficient of friction. Since the vectors sweep over a small part of the circumference of the eccentric only, the friction moment caused by the average vector coincides nearly enough with the true average friction moment for all practical purposes, which leads to the following approximative construction.

In Fig. 49 lay off *(1)(2)* equal to centrifugal force of strap in the direction from center of shaft to center of eccentric. In the same direction lay off *(2)(20)* equal to maximum inertia force of reciprocating valve parts, reduced to the center of the eccentric. At right angle to this direction, and opposite to the direction in which point *(1)* rotates about the center of the shaft, lay off *(20)(19)* equal to 64% of the constant valve friction force ($2/\pi = .64$). Then *(1)(19)* is the average vector and acts upon the eccentric at point *(21)*. The product of this average force, the friction coefficient and the lever arm b furnishes the average friction moment for the position under consideration.

References to Bibliography at end of book: **36, 40, 50, 58, 59, 74.**

6. Spring Moment. — The spring in a shaft governor must be of such size, and must be so located, that the equilibrium speed of the governor (when governing the engine) rises gradually and steadily from full-load position to no-load position under any conditions of careful operation, with the exception of the first minute or two after starting. A few additional exceptions due to tangential inertia or to compensating oil pots will be dealt with later on.

For the purpose of expressing this rather general statement in a more tangible form, let us equate the spring moment to the rest of the moments.

Let M_s = spring moment

M_c = any one of the speed moments (centrifugal, valve inertia)

M_f = any one of the moments due to friction.

For equilibrium, the spring moment must balance all other moments, so that

$$M_s = \Sigma M_c + \Sigma M_f$$

where Σ is the mathematical sign for "the sum of." ΣM_c means the sum of all speed moments.

The speed moments are all of the form $m\ r\ L\ u^2$, see paragraph 2 of present chapter. Those friction moments which are caused by the friction produced by centrifugal forces are of the form $f\,m'\,r'\,L'\,u^2$, where f is a coefficient of friction. All other friction moments are of the form $F\ l$, where F is a friction force which is practically independent of the angular velocity u. By substitution of these values we obtain

$$M_s = u^2[\Sigma\ m\,r\,L + \Sigma\ f\ m'\ r'\ L'] + \Sigma\ F\ l \ \ldots\ldots (1)$$

In order to see clearly the conditions which the spring moment must meet in order to make u rise steadily, let us solve the equation (1) for u.

$$u = \sqrt{\frac{M_s - \Sigma\ F\ l}{\Sigma\ m r L + \Sigma f m' r' L'}} \ldots\ldots\ldots\ldots (2)$$

It is immediately seen that the greatest obstacle to securing a gradual and steady rise of u is the variability of the friction coefficient, which affects both $\Sigma\ F\ l$ and $\Sigma\ f\ m'\ r'\ L'$. In practice, lubrication varies; friction of valves and stuffing boxes varies not only with lubrication, but also with steam pressure, steam temperature, tightening of packing bolts by engineers, and with other, but minor, features. The variation of friction moment affects u the more, the smaller M_s and $\Sigma\ m\ r\ L$. Vice versa, the greater M_s and $\Sigma\ m\ r\ L$ for a given engine, the smaller the disturbances caused by changes in friction forces. The latter thought, namely to make the spring moment and the centrifugal moment quite great, is carried into practice by all builders of small engines using shaft governors. For small and medium sized engines the magnitude of the spring moment required for faultless operation under all conditions is seldom, if ever, determined by calculation, but by practical tests.

Both calculation and experience prove that with those shaft governors in which the spring forces are transmitted to the centrifugal weights through pins, there exists for each engine a certain point, above which an increase of spring moment does more harm than good, because the friction in the governor pins becomes so great that the reacting or impressed forces coming from the valve gear cannot keep the governor vibrating or "limbered up" (see also Chapter VIII). The governor becomes sluggish and does not respond quickly to changes of load. This undesirable feature can be avoided by placing the centrifugal mass directly on the spring (see Fig. 54), or by the proper arrangement of inertia masses.

The doubly beneficial influence of tangential inertia in shaft governors was recognized at a comparatively early time in the history of such governors. First, tangential inertia steadies the governor against impressed forces reacting from the valve gear (see Chapter VIII); and second, it prevents racing and hunting no matter whether the speed drops off as the load comes on, or whether it rises (see paragraph 3 of Chapter IX). The latter effect of tangential inertia was exploited to obtain reversed speed curves (higher speed at full load than at no load), but the use of "we-don't-care-what-the-speed-curve-is" governors on engines driving alternators brought stern realization of the fact that, at least for the purpose of driving alternators in parallel, a gradual and steady rise of the speed is required as the load drops off, and that every other arrangement is a positive failure. One firm after another had this (often quite expensive) experience, between the years 1900 and 1905. The result is that the statement made above concerning a sufficiently great spring moment to maintain a steady speed rise in spite of reasonable variations of friction holds good for all present-day shaft governors.

In view of this explanation, it will pay to make the following comparatively simple, although rather tedious, calculation, whenever a new size or type of shaft governor is to be built. For all friction forces assume maximum and minimum values within reasonable limits. For guidance in this assumption the coefficient of friction may be varied from the low value of $\frac{1}{2}\%$ to the high value of 12%. Compute the friction moments with both of these values for three or four positions (configurations) of the governor, as indicated in paragraphs 4 and 5 of the present chapter, and substitute in equation (2). In either case u must rise gradually from full load to no load. If it does not, either the static fluctuation p, or the spring moment M_s, must be increased, or else the location and arrangement of the spring must be changed, as explained below.

The rapid growth of the steam turbine in central stations limits the use of shaft governors for close-regulation alternating-current work more and more. For that reason a detailed example of the above outlined calculation would benefit only very few readers. It is, therefore, omitted.

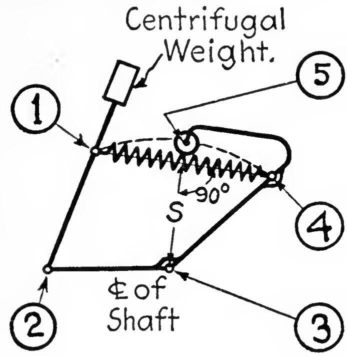


FIG. 51

The following brief remarks will, however, be of interest. Each speed moment, if plotted against angle of swing of governor weight, is part of a slightly distorted sine curve. The distortion is caused by the fact that the governor is not isochronous. The spring moment can never be made to be a sine curve, because both spring force and its lever arm vary, as the governor moves in or out. Consequently, it is quite difficult to make the spring moment coincide with the sum of the other moments as required by a gradually and steadily rising speed, if either the main centrifugal mass or the eccentric turns

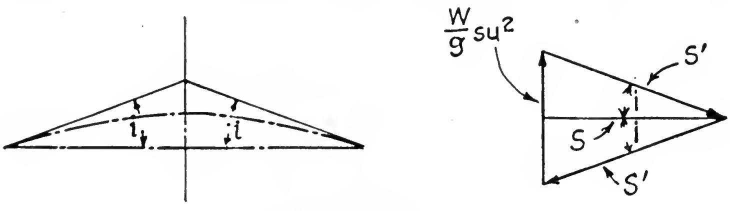


FIG. 52

through a large angle. Large in this case means more than 50°. If the main centrifugal mass is large compared to the influence of other moments, and swings through an angle not exceeding 30°, the problem of securing a steadily rising speed is very easily solved. This feature deserves consideration in the design of new shaft governors.

As an illustration of the complications introduced by the swinging of weights or eccentrics through large angles, the well known Thompson governor of the Buckeye Engine Co. may be cited. The eccentric is turned through about 110° on the shaft, so that the spring must balance a moment which is roughly of the form $K \sin i + K' \sin (3\, i + i')$, where i is the angle through which the lever arm (carrying the centrifugal weight) swings. For many years after its introduction, this governor was not fit for close regulation. It was made fit by the addition of auxiliary springs which oppose the main springs in the inner position of the governor and go out of action somewhere near the central position of the weights. Even with this ingenious expedient, the governor is sensitive to changes of friction in eccentric and piston valves.

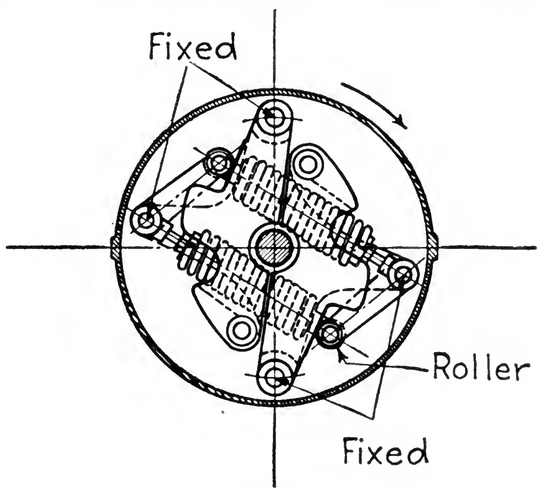


Fig. 53

In the detail calculation of springs of shaft governors many influences which, at first thought, seem too trivial to have much effect, must be considered. Among them is the influence of the mass of the governor springs. Not only does the spring exert a centrifugal moment which must be taken care of in the balancing of the various moments, but frequently the centrifugal force distorts the spring so that both its force and its line of action are altered. This condition is illustrated in Fig. 51. The system *(2) (3) (4) (5)* rotates rigidly about point *(3)*. Roller *(5)* keeps the spring from bowing out. If this roller is removed, the center line of the spring assumes a parabolic shape as indicated in an exaggerated manner by the dotted line *(1) (4)*. The centrifugal force tending to bow the spring out is $\frac{W}{g} s\, u^2$, where W is the weight of the spring. The meaning of s is clear from Fig. 51.

If S is the elastic force and l the length of the spring before, and S' and l' the corresponding values after bowing out, then $\frac{S'}{S} = \frac{l'}{l}$; but l' is found from the relation that the center line of the spring becomes an arc of a composite curve which is so shallow that it can be replaced by a parabola. Hence $l' = l(1 + \frac{1}{6} i^2)$, where i follows (see Fig. 52) from the relation $i = \frac{W s u^2}{g 2 S}$. From these relations both change of spring force and change of lever arm can be computed.

In modern shaft governors for poppet valve engines with lay shafts it is customary to place the springs just as close to the shaft as possible (see Fig. 53). The centrifugal force of the spring in this case is so small as to be without noticeable effect.

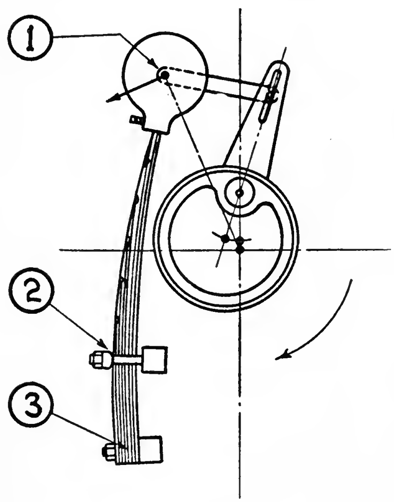


Fig. 54

Centrifugal force of the spring also has a component in the direction of the spring. However, the influence is small and can be taken care of by adjustments described below.

In some of the best shaft governors, leaf springs are used. If the spring is held as indicated in Fig. 54, the nature of the fastening is somewhat uncertain. It is common to figure the spring as if it were a cantilever, built in at point (*2*) and loaded at point (*1*). Actually it is held at points (*2*) and (*3*) and can deflect between these points. This feature should be taken care of in the calculation. The spring is a beam which is semi-constrained at (*3*), supported at point (*2*) and loaded at point (*1*). While an exact mathematical solution may be difficult, careful judgment allows a very close approximation.

The exact predetermination of all forces in a shaft governor places an enormous amount of work on the engineering department. For that reason it is often preferable not to be quite so particular in the calculations, but to allow for all kinds of adjustments; so that the man in the field can work out his own salvation. Although the foregoing explanations contain the principles upon which all adjustments should be based, the effects of the more important adjustments will be briefly considered.

(1) Change of spring tension. A change of initial spring tension is equivalent to adding or subtracting a constant spring force throughout the range of governor motion (with the exception of the disturbance caused by the centrifugal force of helical springs). An increase of spring tension increases the speed, but decreases stability and static fluctuation; a decrease of spring tension decreases speed but increases static fluctuation and stability.

(2) Change of spring leverage. If the movable end of a helical spring is shifted along the weight arm in such a way that its tension in mid-position of governor is not altered, the spring moment changes directly as the spring leverage. The change in spring moment necessitates a corresponding change of centrifugal moment, which means a change of equilibrium speed. A greater spring leverage works the spring through a wider range of deflection, which means a greater variation of spring moment between full-load position and no-load position, or briefly an increase of static fluctuation and of stability.

It is possible to locate a slot in the weight arm in such a way that while the spring leverage is increased, its tension is decreased, with the result that the speed of the engine is not varied by the adjustment. In this case only the stability and static fluctuation are varied, the speed remaining constant.

In practice a variation of spring tension is a good deal easier than variation of the spring leverage, because it calls simply for the adjustment of the tension screw, whereas it is necessary to slacken the spring almost all the way in order to shift it along the lever. Slight variations in speed are therefore mostly accomplished by variation of spring tension. Engineers know that practice bears out theory, because by increasing spring tension farther and farther they "strike a racing point," which necessitates an increase of spring leverage.

(3) Adjustment of fixed end of spring in a circular slot, concentric with point of attachment of movable end of spring in mid-position of governor (see Fig. 55). This has the effect of varying the lever arm of the spring for a given position of the governor and of varying the ratio of lever arms in the full-load and no-load positions. This adjustment offers very good means for making the spring moment balance all other moments with a steady speed curve throughout the range of governor motion, unless the angle of swing be too great.

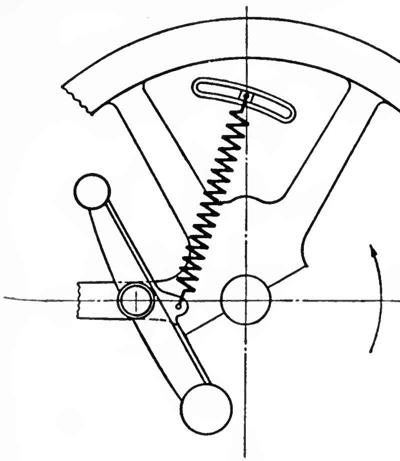


Fig. 55

(4) Adjustment of quantity of centrifugal mass. From equation (2) of this paragraph it follows that addition of centrifugal weight (increase of m) reduces the speed, and, vice versa, that reduction of centrifugal mass means an increase of speed. From the same equation it follows that the stability and static fluctuation remain unchanged only on condition that each and every centrifugal mass is changed in the same ratio (not only the so-called centrifugal weight).

There exist additional adjustments which the man in the field can make. Among them are: shifting the mass center of one or more of the centrifugal weights; changing the number of active leaves or coils in a spring; bringing into action an auxiliary spring for part of the travel; and others. However, any one of these adjustments should not be made blindly, without full knowledge of the theoretical effect which they have. They should be tried only after thorough discussion with the designing engineer. Adjustments of oil gag pots and friction brakes are, of course, frequently made. For such adjustments see the chapter on cyclical vibrations of governors.

References to Bibliography at end of book: **36, 40, 50, 51, 58, 59, 74.**

CHAPTER VII

NATURAL PERIOD OF VIBRATION OF A CENTRIFUGAL GOVERNCR

In paragraph 2 of Chapter III that force was discussed which returns a stable governor to its position of equilibrium, after the governor has been displaced from that position. From Fig. 17 it follows that the restoring force is proportional to the displacement, so that the frictionless governor makes sine harmonic vibrations about the position of equilibrium. In reality these vibrations are more or less damped, the damping depending upon the internal friction of the governor. From mechanics the fact is known that solid friction does not change the period of vibration.

In general, the natural vibrations of a governor are of no importance. They become of importance if their period coincides, or nearly coincides, with that of the forces impressed upon the governor by the valve gear. Knowledge of the natural period is also necessary for determining the stability of regulation. In order to prepare for these interactions between governor and prime mover, a method will now be given for computing the period of natural vibrations.

From mechanics the time of a complete vibration is known to be

$$T_n = 2\pi\sqrt{\frac{\text{vibrating mass}}{\text{restoring force per unit displacement}}}$$

In this equation both the vibrating mass and the restoring force must be referred to the same representative point. If, for instance, the restoring force is measured in the direction of centrifugal force at the mass center of the revolving weights, the motion of all governor masses must be referred to that direction and to that point. If, on the other hand, the mass motion is referred to the governor sleeve, the restoring force must also be referred to the sleeve. For the method of such reduction see below.

From paragraph 2 of Chapter III the restoring force per unit displacement is known to be $\frac{2\ C p}{r_o - r_i}$ for a straight-line characteristic. If the characteristic is curved, p is taken from the tangent to the characteristic, as previously explained, at the position of equilibrium. The vibration takes place about the latter position. Substitution of the restoring force furnishes

$$T_n = \pi \sqrt{\frac{2\ m_e (r_o - r_i)}{p\ C}} \quad \text{.............} (1)$$

In this equation m_e is the equivalent mass. The method of finding the equivalent mass is illustrated by Fig. 56, in which two closely adjacent positions of a governor have been drawn. If for the present only the weights Q[1] and W are considered, the equivalent mass is

$$m_e = \frac{2}{g} \frac{W (ds)^2 + Q (dh)^2}{(dr)^2} \quad \text{............} (2)$$

The square of the travel results from the double effect that first the greater travel requires greater accelerating force and that second this force acts with a longer lever arm.

If all forces and mass motions of a governor are referred to radial direction, the equivalent mass m_e for any system of masses is given by the equation $m_e = \frac{\Sigma\ m (ds)^2}{(dr)^2}$, where m is any mass in the system and ds is the space through which it travels, while the centrifugal mass travels dr radially. In any governor the value of m_e may be expressed by the angles, lengths of arms, etc., but it is usually easier to use the method indicated above, that is to say, take the travels directly from two positions of the governor.

If a governor is connected to a valve gear, its period of vibration changes, because the mass of the valve gear parts has to swing with the governor and enters into the sum of all masses times travel squared.

As an example, the time of vibration of the loaded Watt governor shown in Fig. 56 will now be computed.

[1] Q = weight of ½ counterpoise.

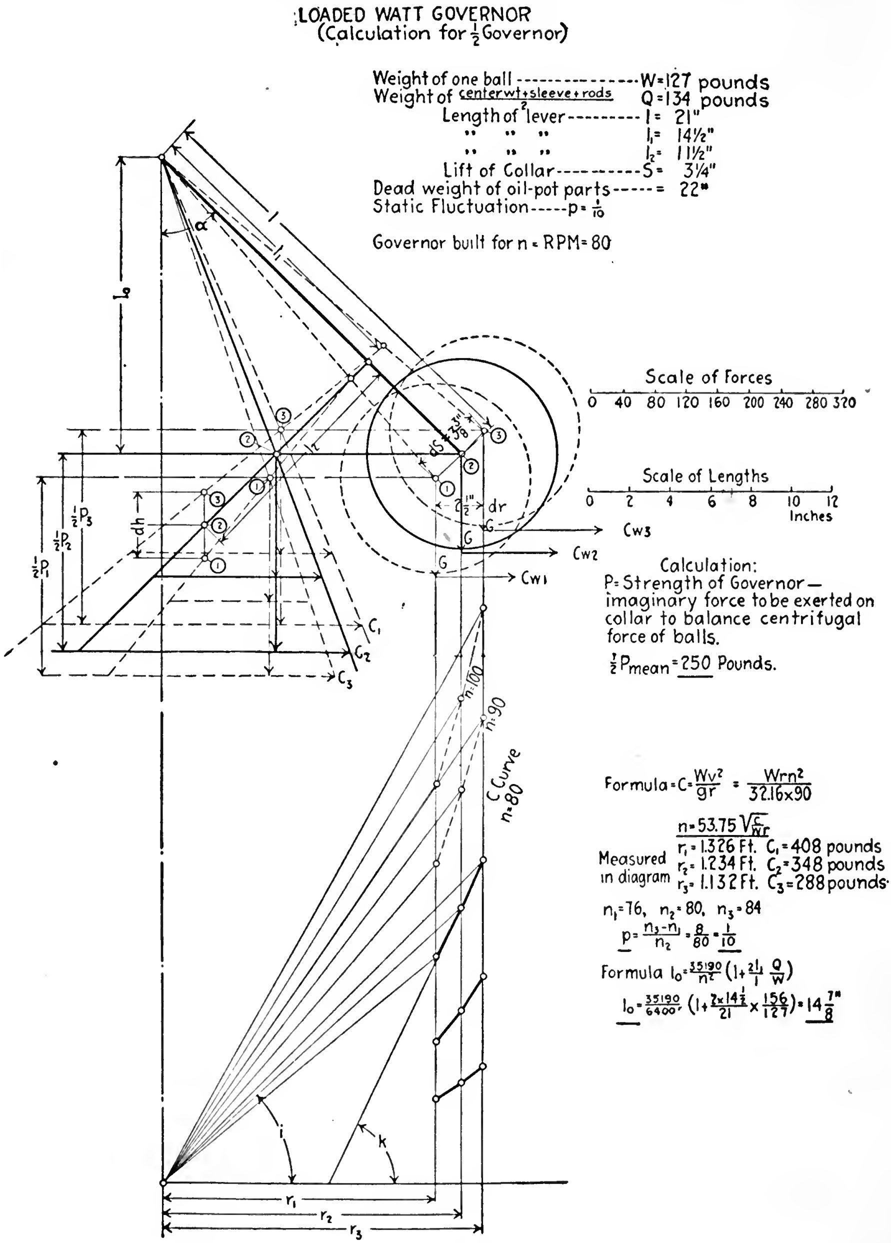

LOADED WATT GOVERNOR
(Calculation for ½ Governor)
Weight of one ball ---------------W=127 pounds
Weight of centerwt+sleeve+rods / 2 Q=134 pounds
Length of lever---------- l= 21"
" " " l₁= 14½"
" " " l₂= 11½"
Lift of Collar------------S= 3¼"
Dead weight of oil-pot parts------= 22#
Static Fluctuation-----p= 1/10
Governor built for n = RPM= 80
Scale of Forces
0 40 80 120 160 200 240 280 320
Scale of Lengths
0 2 4 6 8 10 12
Inches
Cw3
Cw2
Cw1
Calculation:
P= Strength of Governor—imaginary force to be exerted on collar to balance centrifugal force of balls.
½ Pmean = 250 Pounds.
Formula = C = Wv²/gr = Wrn²/32.16x90
n = 53.75 √(C/Wr)
Measured in diagram
r₁ = 1.326 Ft. C₁ = 408 pounds
r₂ = 1.234 Ft. C₂ = 348 pounds
r₃ = 1.132 Ft. C₃ = 288 pounds
n₁=76, n₂=80, n₃=84
p = (n₃-n₁)/n₂ = 8/80 = 1/10
Formula l₀ = 35190/n² (1 + 2l₁/l · Q/W)
l₀ = 35190/6400 (1 + 2x14½/21 x 156/127) = 14 7/8"
n=100
n=90
C Curve n=80
½P₁
½P₂
½P₃
dh
dS
dr
C₁
C₂
C₃
r₁
r₂
r₃

FIG. 56

For this governor, the equivalent mass reduced to radial travel (from p. 78, equation 2) equals

$$m_e = \frac{254 \times 3.38^2 + 268 \times 3.25^2}{32.2 \times 2.5^2} = 28.6 \frac{\text{pounds sec}^2.}{\text{ft.}}$$

This expression gives a mean value, because finite paths Dr, Ds and Dh have been used instead of the corresponding differentials dr, ds and dh.

From p. 31 the restoring force per unit displacement is known to be $z = \frac{dC}{dr} - \frac{C}{r}$, or $z = \tan k - \tan i$ (Fig. 16), where k is the angle which the tangent to the characteristic curve makes with the base line, and i the angle which the straight line drawn from the same point of tangency to the origin makes with the base line. The tangents are obtained by forming the ratios of the drawing lengths of the two short sides in the right angle triangles and by multiplying each length by its scale. Thus we obtain:

$$z = \frac{340}{.55} - \frac{340}{1.23} = 340 \text{ pounds per ft. of radial displacement.}$$

By substitution we obtain the period of vibration

$$T_n = 2\pi \sqrt{\frac{m_e}{z}} = 6.28 \sqrt{\frac{28.6}{340}} = 1.82 \text{ seconds.}$$

References to Bibliography at end of book: **14, 36, 37, 83.**

CHAPTER VIII

EFFECTS OF OUTSIDE FORCES IMPRESSED UPON GOVERNORS

1. Resistibility.— The power-controlling mechanisms which are adjusted by governors may be broadly divided into two classes, namely

(1) Devices which offer passive or friction resistance only to the governor,

(2) Devices which react upon the governor and tend to cyclically displace it from its position of equilibrium.

The features of the governor which are valuable in dealing with the first class of mechanisms are strength, work capacity, and, for certain cases, tangential inertia, see Chapter II.

The feature which is valuable in dealing with the second class of mechanisms has not received much attention from writers and has, therefore, no recognized name in any language. Since the feature in question is the ability to resist, I have adopted the term "resistibility." Dr. Proell uses the term "moment of resistance" for shaft governors.

The necessity for this property in governors subjected to great reaction will easily be realized. Heavy intermittent forces from the valve gear throw a light governor back and forth, causing unequal and irregular power distribution, and, consequently, great speed fluctuations. And yet the light, but, in this case, useless governor may be very strong, because the strength, in the sense of paragraph 1 of Chapter II depends upon $m\,r\,u^2$; and $r\,u^2$ may be very great, offsetting the smallness of m. It will presently be shown that resistibility is indeed closely related to the mass of a governor.

Returning to the statement that light governors are thrown back and forth under the influence of vibratory forces, we are forced to admit that the actual motion of such a governor is not amenable to mathematical treatment, except through repeated semigraphical point-by-point integration, which is, of course, too slow for any practical purpose. The difficulty is partly caused by the interaction of governor and prime mover. Any motion of the governor affects the speed of the prime mover, and variation of the speed moves the governor. This motion is superposed over the cyclical vibration impressed upon the governor by the valve gear. The latter motion is modified by the natural vibration of the governor (see Chapter VII) and by the resistance of solid and liquid friction. From these facts it is evident that an analytical solution of the problem of finding the motion of the governor is out of question.

Fortunately, a complete solution of this problem is not needed for practical purposes. Since it is not permissible to have governor motions of such a magnitude that they disturb power distribution and vary the speed of the prime mover, governors must be made resistant enough to prevent excessive vibrations, and the influence of such speed variations may, in consequence, be omitted from the study of cyclical governor vibrations. The problem then is to make governors resistant enough to limit vibrations (caused by reacting forces) to a small fraction of the swing or travel of the governor parts. For the solution of this problem the knowledge of certain facts concerning vibrations is required. For that reason they will be briefly reviewed.

If a mass is subjected to a cyclically fluctuating force, and is not affected by any other force, it performs a vibration which can easily be computed in every detail. It is called the impressed vibration. Mass furnishes the only opposition limiting the amplitude of the impressed vibration. The impressed vibration is modified by friction, both liquid and solid, and by the natural vibration. The latter may magnify or diminish the amplitude of the impressed vibration, depending upon the ratio of their frequencies. If that ratio is near one, and if there is very little friction, very large amplitudes result. As a matter of fact, governors with very small friction, when used in connection with reacting valve gears, have occasionally struck that coincidence of frequencies which in mechanics is called resonance. Under such conditions a frictionless governor is useless and friction must be added to make it useful, usually much to the dismay of the designer who tried to produce correct regulation by the use of a frictionless governor. Fortunately, a relatively small amount of friction suffices to prevent undue magnification of the impressed amplitude.

In paragraph 2 of Chapter IX proof is furnished that a certain amount of friction is needed in any governor to insure stability of regulation. And such an amount of friction is needed that resonance (magnification of impressed amplitude) cannot occur, if regulation is to be stable. For this reason resonance may also be dismissed from the discussion.

There remains then only the amplitude of the impressed vibration, damped by solid and liquid friction. Friction reduces the amplitude of any vibration and thus increases resistibility. This brings up the question to what extent friction should be depended upon to furnish resistibility. No generally applicable answer can be given, for the following reasons: The amount of friction which is necessary in a governor to produce stability of regulation varies with the promptness of the governor and with the kinetic energy stored up in the rotating masses of the prime mover per horse power of capacity (see paragraph 2 of Chapter IX). A considerable increase of friction over the amount which is necessary for stability of regulation does harm, because it delays the governor, when the latter is adjusting the power-controlling mechanism after a change of load, and thereby causes undue and excessive speed fluctuation. This uncertainty of the permissible amount of frictional damping in governors makes it advisable to depend mainly upon mass for keeping down the amplitude of cyclical vibrations and to depend upon friction to a limited extent only. If a rule is wanted, the following will serve as a rough guide. Use a governor with sufficient mass to reduce the amplitude of the impressed vibration to $1\frac{1}{2}$ times the permissible amplitude, and obtain further reduction of amplitude by solid or liquid friction. Just what amplitude is permissible will be considered in the next paragraph.

Solid friction is very effective as a damping agent; however, several precautionary measures must be observed to make its use successful. In governors, such as shown in Figs. 2 and 4, great centrifugal forces are transmitted through the pin joints. The diameters and lengths of these pins are usually made very small for the purpose of reducing friction, and the joints wear considerably, if the governor vibrates. It is, therefore, much better to make the pressure between the rubbing surfaces so small that wear is very slow. Furthermore, it is advisable to make the friction adjustable and to arrange the parts so that slight wear does not noticeably vary the friction. Finally, it is desirable to keep oil away from the rubbing surfaces, so that a varying amount of oil may not disturb the proper frictional damping. Figure 57 shows an arrangement which embodies these features and which has given satisfaction.

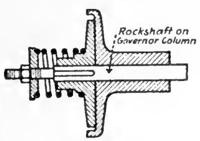


FIG. 57

While resistibility is easily defined as ability to resist impressed forces, a mathematical definition is impossible, because both mass and friction share in the resistance. If friction is omitted from the discussion, resistibility becomes (in spindle governors) simply equivalent mass of governor at the sleeve or collar of the latter, and, in shaft governors, equivalent moment of inertia of governor parts about the suspension point of the eccentric. For this reason, makers of direct-acting governors should give the equivalent mass of each size of governor in their catalogues.

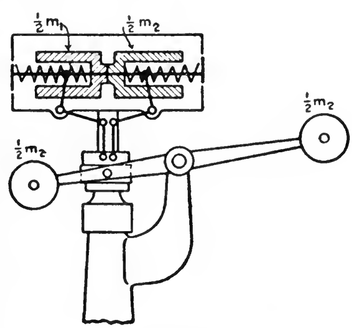


FIG. 58

While it is evident that valve gears which impose great alternating or vibratory forces upon the governors require the latter to be quite heavy and massy, it is less evident how that mass should be distributed in the governors for best advantage. For the purpose of reducing amplitude of impressed vibration, the governor mass need not be so arranged that all of it produces centrifugal force. For reduction of cyclical vibration, the mass is very effective, if arranged as shown in Fig. 58, where both m_1 and m_2 contribute to the resistibility, whereas

m_1 only produces centrifugal force. However, the expedient of placing a large part of the governor mass outside of the governor proper considerably reduces the promptness of the latter and causes great fluctuations of the speed, whenever the load changes suddenly (see Chapter IX). It must not be used where close regulation is essential. In the latter case it is much better either to use a larger-size governor, or else to arrange the additional mass in the governor in such a manner that it furnishes tangential inertia acting in the right direction (see paragraph 2 of Chapter II). In that paragraph proof was furnished that the regulating force due to tangential inertia is proportional to the change of load of the engine. Paragraph 3 of Chapter IX contains proof that tangential inertia increases the stability of regulation. As before stated, it increases resistibility without increasing centrifugal force. Since in a vibrating system a comparatively small centrifugal force suffices to adjust the position of equilibrium, tangential inertia furnishes the ideal vibration-resisting force. Of all types of governors, shaft governors are subjected to greatest fluctuating forces. The use of tangential inertia in shaft governors, which was introduced by American engineers shortly after the year 1890, is, in consequence, well justified and is correct engineering.

References to Bibliography at end of book: **23, 28, 50.**

2. Cyclical Vibrations of Governors.— As explained in the preceding paragraph, the necessary resistibility of a governor is determined by the impressed force (resp'y moment) and by the permissible amplitude of the vibration of the governor. The latter is determined chiefly by the consideration that, for a constant load, the supply of energy must be the same, stroke after stroke of the engine. A governor vibrating too much will strike the stops, if the load is either very light or very heavy. The result is irregular motion of governor and of engine. Besides, vibrations of large amplitude wear out the governor in short time, unless it was specially designed for such service.

Both requirements, namely that of clearing the stops, and that of avoiding wear, refer particularly to spindle governors. The latter are commonly so adjusted that their total available travel just varies the energy supply between the limits of no load and full load, which means that a vibrating governor strikes the stops near the extreme ends of its travel. In spindle governors a third feature enters, namely the visibility of the vibration. Engineers seem to have an instinctive feeling that a vibrating governor is wrong, and take steps to quiet it.

Matters are very different with shaft governors. The vibrations are not visible and, for that reason, do not interfere with any preconceived notion of operating engineers. Experience with vibrating shaft governors has taught designers to make the supporting pins of liberal proportions so that wear is slight. Furthermore, the valve gear is frequently of such nature that the governor stops can be placed considerably outside the limits of no power and of maximum power. In that case the governor can (and usually does) vibrate with a considerable amplitude without producing any harmful effects upon regulation.

Evidently, relatively greater vibrations can be permitted in shaft governors than in spindle governors, unless the latter are designed for cyclical vibrations. It is also evident that a large number of circumstances influence the greatest permissible amplitude so that no universal figure can be given for that quantity. The following values may, however, be used as a guide for average conditions:

For spindle governors, maximum permissible amplitude equals $\frac{1}{20}$ of governor travel, so that maximum displacement equals $\frac{1}{10}$ of governor travel.

For shaft governors, maximum permissible angular amplitude equals $\frac{1}{12}$ of total angle, so that maximum angular displacement equals $\frac{1}{6}$ of total angle.

These are limiting values, and it will pay in practice to keep the amplitude of cyclical vibrations somewhat below these values. It should be noted that the tendency of some designers to keep any and all impressed forces away from governors for the purpose of entirely eliminating cyclical vibrations is fundamentally wrong. In paragraph 4 of Chapter II, mention was made of the fact that vibrations reduce or eliminate friction and the time lag (detention) caused by it. Consequently, moderate cyclical vibrations of governors are beneficial. Their action in eliminating friction may be analyzed as follows: Let Q be the regulating force which tends to move the governor collar at a given instant. Let F be the resisting frictional force, and let F be greater than Q, so that the governor would not move, if it were not for the effect of the impressed cyclical vibration. Let $\frac{1}{2}s$ be the amplitude of the latter. While the governor collar moves in the direction of the force Q, the latter acts into the space s and puts into the governor the work Qs in the shape of kinetic energy, so that the governor moves its center of vibration the distance Qs/F in the direction of Q. During the return travel against the force Q, the latter reduces the kinetic energy of the governor the amount Qs so that the center of vibration is again shifted the distance Qs/F in the direction of the force Q. With each complete vibration, the collar moves the distance $2\,Qs/F$, which means that any force, no matter how small, will affect a vibrating governor and will move it. Skillful and experienced designers make use of this fact.

The problem of so selecting the resistibility of the governor that the amplitude is kept within the desired moderate limits consists of two separate problems, namely

(1) Ascertaining the impressed force or moment as a function of time,

(2) Double integration of the resulting linear or angular acceleration.

The first part, namely the determination of the impressed force or moment, is really a problem of engine or valve gear design. The great diversity of valve gears prohibits a detailed discussion, so that the latter must be limited to a few remarks concerning general guiding principles.

Forces reacting upon the governor are caused by friction of valves and valve gear parts, by inertia of valves and valve gear parts, by unbalanced steam pressure and by unbalanced weights (in shaft governors). In the case of releasing gears the principal force is the friction between the catchblocks which, in turn, is determined by the force required to drive the valves.

This latter force is found as below given for automatic valve gears.

In the case of non-releasing valve gears all inertia forces are computed on the basis of the valves being moved by a non-vibrating governor. To this end the displacements of all moving valves and valve gear parts from their respective midpositions are plotted against time. The second derivatives of these curves furnish the accelerations which later, upon multiplication by the respective moving masses, furnish the inertia forces. Forces caused by friction of valves and by unbalanced steam pressure on valve stems are computed from the well known laws of mechanics.

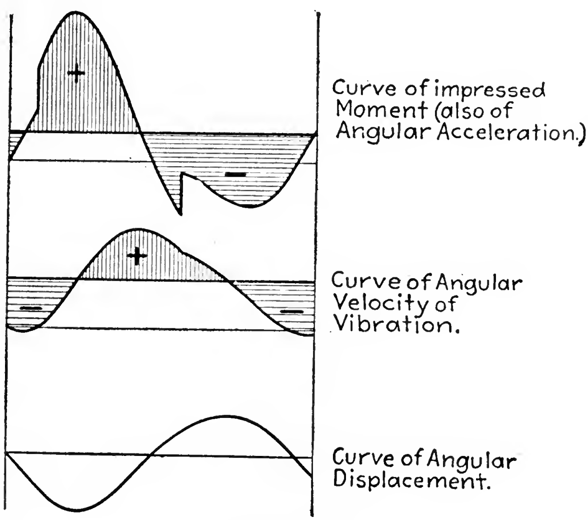


FIG. 59

In Fig. 59 the upper curve is an example of an impressed moment plotted against time (one revolution of the engine). The curve should not be taken as representing all possible cases. On the contrary, curves of impressed moments are most varied in shape and depend very much upon the type of valve gear.

From the moment curve the amplitude is found by a double integration. If the impressed moments are divided by the moment of inertia of the governor parts,[1] the moment curve

[1] For the governor, Fig. 8, the moment of inertia is with sufficient accuracy mass (*4*) times l_4^2 plus mass (*5*) times l_5^2 plus moment of inertia of connecting link. For a governor of the type of Fig. 9 both impressed moment and moment of inertia must be referred to either point (*4*) or else to point (*5*). If (*4*) is made the point of reference, the moment of inertia is roughly $J = m_1\, l_1^2 + m_2\, l_2^2 \left(\frac{l_4}{l_5}\right)^2$

becomes a curve of angular accelerations which, upon being integrated once furnishes angular velocity of vibration. The second integration furnishes angular displacement. Each integration can be carried out either by the addition of mean ordinates, or by the use of a planimeter. Or else the double integration may be accomplished in one operation by the drawing of an equilibrium polygon. It is evident that in a cyclical vibration the mean value of the angular velocity of vibration must be zero and that likewise the mean value of the impressed moment must vanish. This fact must be observed in the integration. In the foregoing the influence of friction has been neglected. Reasons for the advisability of neglecting it were given in paragraph 1 of the present chapter.

For a fairly complete solution the whole calculation should be made for at least three positions or configurations of the governor. While the work is not difficult, it is certainly long and tedious so that a healthy guess is usually considered the lesser evil.

A great deal of trouble has been caused by cyclical vibrations due to unbalanced weights in shaft governors. Although this type of governor has not the importance to-day which it had at the end of the nineteenth century, a brief discussion of the underlying principles will even at this date be helpful to a large number of engineers.

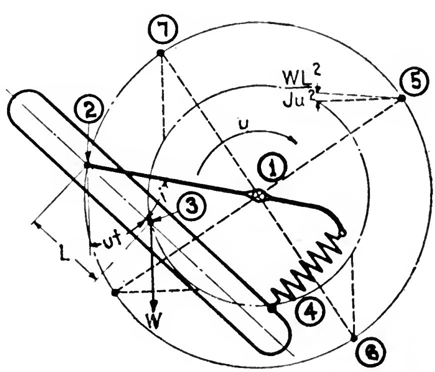


Fig. 60

Figure 60 diagrammatically represents a shaft governor with an unbalanced weight such as is used for instance in the Rites governor. (*1*) is the center of the shaft. (*2*) is the suspension point of weight W whose mass center is (*3*). The system rotates about (*1*) with uniform angular velocity u. Spring (*4*) is so dimensioned that it balances the centrifugal force of W in any position.

The motion of weight W can be found under the following simplifying assumptions:

(1) The angular displacement of the weight is so small that it can be neglected compared to the average value of $u\,t$ which, of course, is 45°.

(2) Effects of friction and of resonance are neglected. Justification for these simplifications is given in paragraph 1.

The angular acceleration of the weight W (the moment of inertia of which about point (*2*) is J) is

$$\frac{d^2i}{dt^2} = \frac{du}{dt} = a = \frac{W\,L \sin u\,t}{J}$$

By integrating [1] twice we obtain first

$$Du = -\frac{W\,L}{u\,J} \cos u\,t + K$$

and then

$$Di = -\frac{W\,L}{u^2\,J} \sin u\,t + K\,t + K',$$

where K and K' are constants the values of which depend upon the location of the zero line as shown in Fig. 59. Du is the change in angular velocity, or the difference of angular velocity between weight W and shaft (*1*), or, in other words, the relative angular velocity of weight W. For a cyclical vibration the average value of Du must vanish, and to have that occur, K must equal zero. The same reasoning holds true for Di and K', so that the angular displacement of vibration becomes

$$Di = -\frac{W\,L}{u^2\,J} \sin u\,t \text{ with angular amplitude of } \pm \frac{W\,L}{u^2\,J} \quad \ldots\ldots (1)$$

Equation (1) teaches that Di reaches a maximum (negatively), when $\sin u\,t$ becomes a maximum which occurs in point (*5*) of Fig. 60. In points (*6*) and (*7*) $\sin u\,t = 0$ so that there is no displacement. The effect of gravity in this simple case then consists in lifting the center of the (slightly distorted) circle, which the weight describes, the distance $\frac{W\,L^2}{u^2\,J}$. If H is the radius of gyration of the weight W, the distance which the

[1] For the benefit of those who are not versed in integration, it has been done graphically in the Appendix, Fig. 139.

weight is raised becomes $\frac{m\,g\,L^2}{u^2\,m\,H^2} = \frac{g\,L^2}{u^2\,H^2}$. The center of the circle described by the eccentric which is linked to the weight W is likewise shifted. The direction of the displacement depends upon the kinematic connection. Early experimenters were surprised to find the raising of the path of the vibrating weight; but this fact appears quite simple and natural, if comparison is made with the ordinary pendulum, in which the

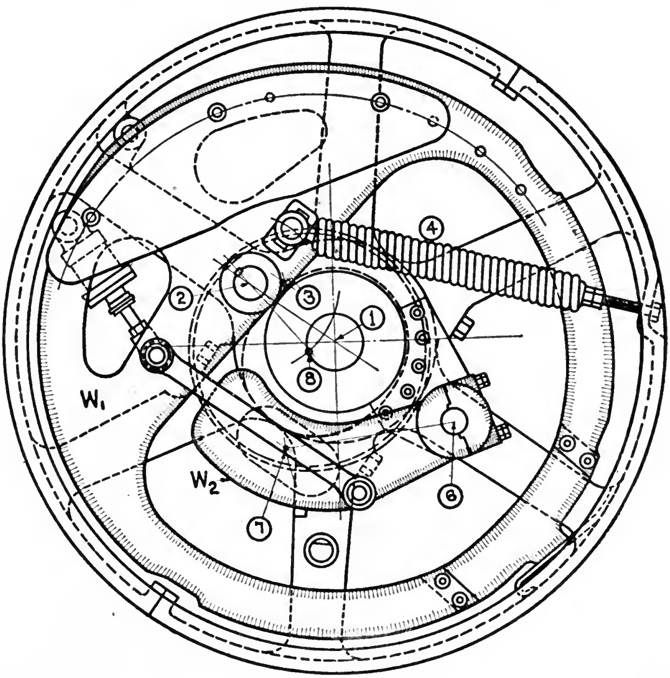


FIG. 61

average location of the mass center is higher during vibration than it is when the pendulum is at rest.

Other experimenters doubted the raising of the center of the path, because they had found a shifting to one side. Again the explanation is not difficult. From the theory of vibrations it is known that friction causes a phase lag which, in the case of resonance, reaches the value of 90°. Evidently, the actual direction of the displacement varies with the friction so that it is advisable not to count upon any given direction, but rather to keep the amplitude of vibration within safe limits as previously given in this paragraph. For a given angular velocity u of the governor shaft the displacement may be kept small by making radius of gyration H large compared to mass radius L. For small values of u, say for less than 150 revolutions per minute, the requirement of small amplitude means (with a single centrifugal weight, as in Fig. 60) so small a value of L that the centrifugal moment becomes insufficient for speed counting. The use of a single weight is, therefore, inadvisable for slow speeds, and double weights, partly or wholly balancing each other, must be used.

FIG. 62

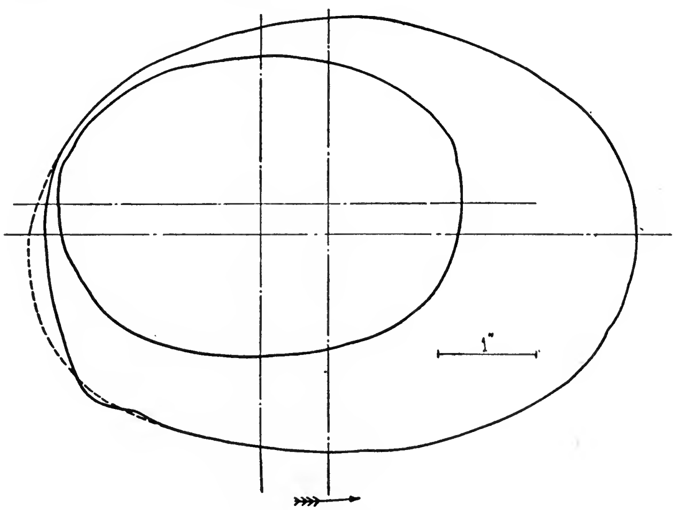


FIG. 63

An actual case from the author's practice will serve to illustrate these theories. The Rites shaft governor shown in Fig. 61 was originally built with a single annular weight W_1. The suspension point of the latter is at (*2*); its mass center is at (*3*). The center of the eccentric swinging rigidly with weight W_1 is at (*8*). The governor was noisy, making a hammering sound.

Indicator cards taken from the engine had the shape given in Fig. 62. A pencil point fastened to the eccentric rod close to

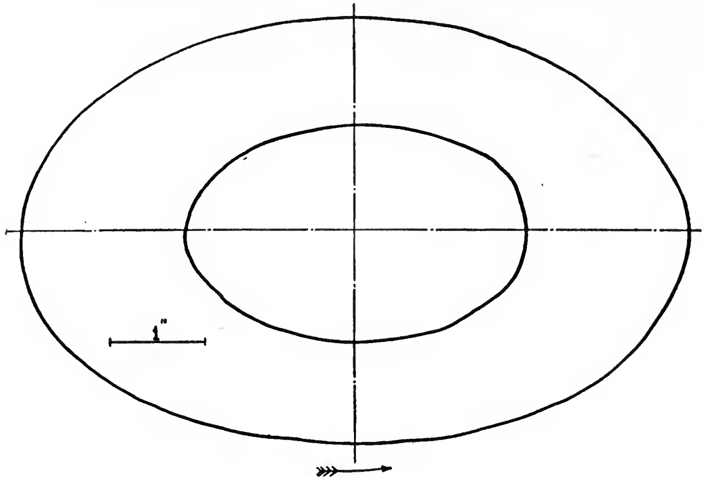


FIG. 64

the governor described the egg-shaped curves reproduced in Fig. 63. The outer curve is described with full load on the engine. The spring holds the governor against the stop most of the time so that the motion of the eccentric is almost symmetrical. The inner curve is described with a partial load on the engine. It is shifted both upward and sideways. To remedy matters, the weight W_2 with suspension point (*6*) and mass center (*7*) was added, and was connected to W_1 by a link. The two weights now almost balance each other against gravity. The results were very satisfactory. Figure 64 shows the curves described by the same point on the eccentric rod after the change, and Fig. 65 shows the corresponding indicator cards. The hammering of the weights in the

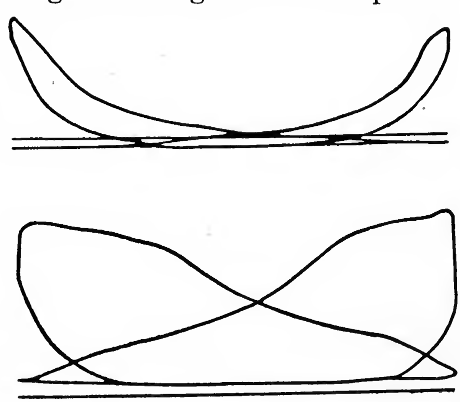


FIG. 65

governor had entirely disappeared. Of course, a new and stronger spring (*4*) was installed, because the addition of the second weight W_2 increased the centrifugal moment for the desired speed.

Cyclical vibrations of governors are also caused by the interaction of governor and prime mover. Their period differs from that of the vibrations studied in the present paragraph. They are dealt with in Chapter IX.

References to Bibliography at end of book: **28, 40, 50, 59, 63, 70.**

CHAPTER IX

INTERACTION BETWEEN DIRECT-CONTROL GOVERNOR AND PRIME MOVER

1. Action of Governors Regulating Prime Movers. — In paragraph 1 of Chapter I the statement was made that governors are used to keep some one quantity practically constant while the output of a prime mover varies. The purpose of the following investigations is to find out to what extent the aim of keeping angular velocity practically constant has been realized.

In Fig. 66 load or output has been plotted against time, and two changes of load have been indicated, one, *(1)* *(2)*, sudden, and the other, *(3)* *(4)*, gradual. In an ideal regulation the governor would follow the change of load without time lag. In that ideal case the broken line *(1)* *(2)* *(3)* *(4)* would also represent governor position.

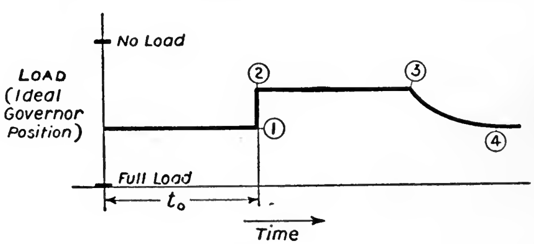


FIG. 66

In reality, governors have mass which requires time for its acceleration, so that a time lag exists between load and governor position. During the interval of lack of coincidence between load (power consumed) and power generated, the difference of work causes a rise or fall of speed which, in turn, affects the motion of the governor.

In the language of mechanics, the governor and the prime mover form a system with two degrees of freedom. Translated into everyday language it means that the position of the governor determines the rate at which the speed of the prime mover varies, and that the speed of the prime mover determines the force which tends to change the position of the governor. To every load on the prime mover there belongs a certain position of the governor, and to every position of the governor belongs a certain speed, but these two variables — (1) governor position, (2) speed of prime mover — are free to vibrate about their equilibrium values.

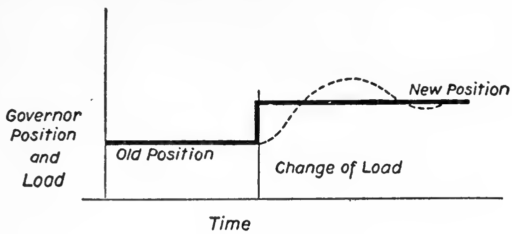


Fig. 67

In the motion resulting from a change of load a broad distinction must be made between the two following cases:

(1) Following a disturbance of equilibrium the governor reaches its new position of equilibrium after a finite number of vibrations the amplitude of which continually decreases (see Fig. 67).

(2) The governor performs vibrations with ever increasing amplitude about its new position of equilibrium and never comes to rest (Fig. 68).

Case No. 2 is, of course, worthless; case No. 1 is the only allowable one in practice. Guarantees for closeness of regulation usually state that for a given and sudden change of load the maximum speed variation shall not exceed a certain amount. Any such guarantee really involves two promises, namely:

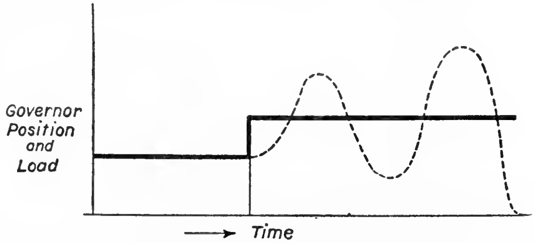


Fig. 68

(a) that the regulation shall be stable,

(b) that the amplitude of the first wave shall be small.

In accordance herewith the problem of regulation divides itself into two problems, to wit

(a) the investigation of the stability of regulation,

(b) computation of the greatest speed variation.

These two problems form the subject of the following paragraphs.

Between the conditions of stability (Fig. 67) and of instability (Fig. 68) lies the limiting case (Fig. 69). In this limiting case any change of load releases a vibration which continually maintains its original amplitude. The regulation is neither stable nor unstable. The limiting case is, naturally, never met with in practice, except accidentally, but it marks the division line between useful and useless, and is therefore of importance. In addition, it has the advantage of being amenable to mathematical investigation by comparatively simple means. The condition of stability will turn out to be independent of the relative load change. Hence, stability at any change of load includes the case of constant load. If the regulation is stable, the governor will not hunt at constant load.

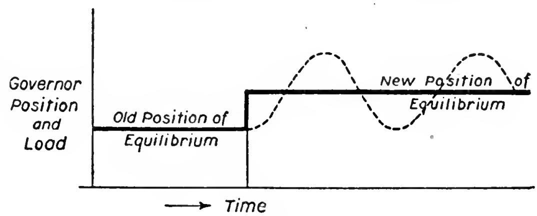


Fig. 69

To make an analytical solution possible, simplifying assumptions must be made. The various existing theories differ mainly in the nature of these assumptions. Among the theories thus offered, the one put forth by Professor Stodola in 1899 appears to the author to contain the smallest departure from actual conditions. Besides, all the assumptions made by Stodola are of such nature that the effect of the neglect can be estimated. The following principal assumptions will be made:

(1) There is no time lag between the position of the governor (including torque-controlling mechanism) and the torque corresponding to the governor position. In the language of Mr. Hartnell (Inst. of Mech. Engineers, 1882), there is no detention due to stored-up energy between the regulating device and the prime mover.[1]

(2) The action of the governor is continuous. While this is no assumption with steam and water turbines, it is a simplifying assumption for steam and gas engines, where the governor

[1] This excludes from the following derivations the case of the compound engine with large receivers or the case of compound steam turbines with large receivers.

action is intermittent from stroke to stroke. This simplification is well permissible, because the time in which a governor finally assumes its new position after a change of load always extends over a great number of engine revolutions (see Fig. 70).

For minor assumptions see the following paragraphs.

References to Bibliography at end of book: **36, 73.**

2. Limiting Case. — In order that we may proceed from the simple to the more complex, the following minor assumptions will be made in this paragraph:

(1) The governor is free from friction.

(2) The governor is so large that the frictional resistance of the valve gear can be neglected.

(3) The radial displacement of the centrifugal weights (motion of representative point) is so small that the centrifugal force may be considered constant over the range of motion.

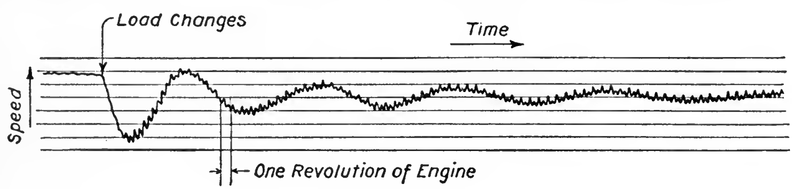


FIG. 70

(4) The torque exerted by the prime mover is proportional to the displacement of the governor from no-load position.

The actions which take place after a change of load has occurred are illustrated by Fig. 71. The abscissæ show "position of representative point" which in this paragraph means "radial distance of mass center of centrifugal weights from center of rotation." All forces and accelerations are to be referred to this point and to radial motion, as was explained in earlier paragraphs. Instead of this motion, that of the sleeve or collar may be used with spindle governors, but the selected motion is applicable to spindle governors and to shaft governors, whereas the apparently simpler and more easily observed motion of the sleeve cannot be used with shaft governors. The relation of the diagram (Fig. 71) to the governor proper is illustrated by Fig. 72. The meaning of the various ordinates of Fig. 71 will become clear from the following reasoning:

Let a prime mover develop the balanced torque (*1*) (*2*) and let the load be suddenly reduced to a torque (*4*) (*5*),

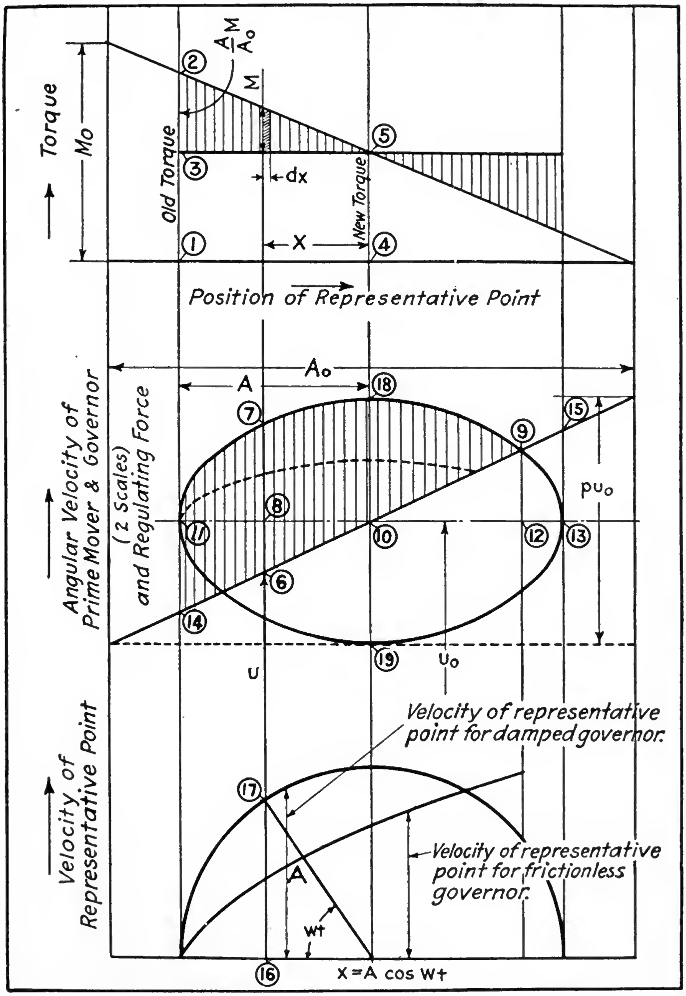


FIG. 71

then the torque (*2*) (*3*) becomes unbalanced and produces an angular acceleration equal to

$$a_i = \frac{\text{unbalanced torque}}{\text{moment of inertia of rotating masses of prime mover}} = \frac{M_0}{I}\frac{A}{A_0},$$

where A/A_0 = relative change of load. The change of velocity resulting from this acceleration sets the governor in motion, slowly at first, because the governor mass must be accelerated from rest. This motion of the governor and of the torque-controlling mechanism reduces the unbalanced torque and the acceleration. Let at a certain time the governor point be x

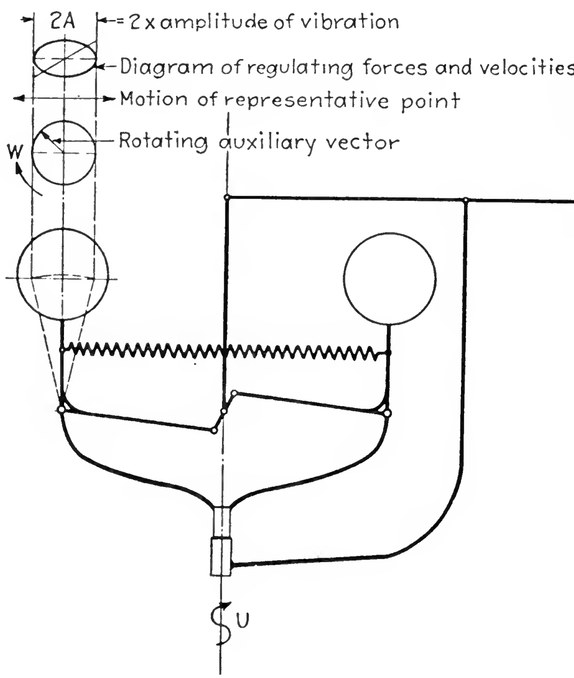


FIG. 72

distant from its new position of equilibrium; then the angular acceleration of the prime mover is

$$a = \frac{M}{I} = \frac{M_0}{I}\,\frac{x}{A} \qquad \text{(1);}$$

the angular acceleration of the governor is proportional to a and depends upon the ratio of gearing between prime mover and governor. While the governor moves towards (4), its new position of equilibrium, a is positive and the angular velocity of the prime mover increases, until point (4) has been passed.

The accelerating forces acting upon the governor are disclosed by the middle diagram of Fig. 71. Line *(14)* *(10)* *(9)* represents the curve of equilibrium speeds (angular velocities) of the governor, or, to another scale, the corresponding speeds of the prime mover. The base line for this curve lies down beyond the limits of the page. As the governor moves from point *(1)*, its old position of equilibrium, towards *(4)*, the speed of the system grows. Compare also Fig. 87. Let the rise of speed be indicated by the curve *(11)* *(7)* *(18)*. Since for position x, *(6)* indicates the equilibrium speed, and *(7)* indicates the actual speed, the excess speed *(6)* *(7)* causes an unbalanced centrifugal force which is approximately equal to $2C\frac{\text{excess } (6)(7)}{u_0}$ (see equation 3 of paragraph 1, Chapter II). Unless this force is counteracted by some resistance (and it is at present assumed that there is no resisting friction or oil gag pot), it will accelerate the governor (representative point) until point *(9)* has been reached, that is when actual speed and equilibrium speed coincide. But the limiting case — now under discussion — of never ending vibrations of constant amplitude requires that the amplitudes on both sides of the position of equilibrium — *(10)* or *(4)* *(5)* — must be equal. Since the accelerating unbalanced forces are approximately proportional to the excess speeds, the vertically section lined area *(14)* *(11)* *(7)* *(18)* *(9)* *(10)* represents to some scale the kinetic energy stored up in the governor, when it reaches point *(9)*; with conditions as depicted here the governor cannot possibly come to rest at *(13)*, but will come to a temporary stop farther away from the position of equilibrium than the starting point is. This fact is evident from a study of the curve of velocity of the representative point at the foot of Fig. 71.[1] Regulation by a governor devoid of friction is therefore impossible. This theorem was first deduced by Wischnegradsky in 1877. Whether the ordinates of the curve *(11)* *(7)* *(18)* *(9)* are right or not does not matter; for if the actual curve were not *(11)* *(7)* *(18)* *(9)*,

[1] The overtraveling of the governor is caused solely by the inertia of its masses. If centrifugal force could be produced without mass, the present paragraph would be superfluous. The harmful effects of mass can be practically eliminated in relay governors. See the paragraph on that subject.

but the dotted curve lying below it, the governor would shoot past the mark just the same, and the amplitude of the vibrations would increase without limits.

In order to prevent this undesirable action, and in order to bring about the constant amplitude vibrations of the limiting case, we must modify the forces acting upon the governor by other forces in such a manner that the kinetic energy stored up on the way from *(11)* to *(10)*, or from *(1)* to *(4)*, is reconverted into potential energy at point *(13)*. Just what must be done to this end is easily seen, if the excess speed *(6)(7)* is resolved into two component parts *(6)(8)* and *(7)(8)*. The two component parts are quite different in their action. The part *(6)(8)* becomes a retarding force after the new position of equilibrium — point *(10)* — has been passed, whereas the component *(7)(8)* remains an accelerating force all the time. The forces resulting from these excess speeds have received various names. The force resulting from *(6)(8)* has been named "static regulating force" or "regulating force due to wrong position," while the force due to *(7)(8)* has been called "dynamic regulating force" or "regulating force due to wrong speed." In this book the terms "static" and "dynamic regulating force" are used.

The areas *(11)(14)(10)* and *(10)(15)(13)* are equal, which means that any kinetic energy which has been stored up in the governor parts while traveling from *(11)* to *(10)* is reconverted into potential energy on the way from *(10)* to *(13)*, provided that the static regulating force alone acts, which means that the dynamic regulating force must not be allowed to act.

To this end the latter force must always be balanced by some external resistance, for instance solid or liquid friction; and the question arises: Can a resisting force be found of such a nature that it will always just equal the dynamic regulating force, and oppose it? In attempting to answer this question we must remember that a force diagram as represented by the straight line *(14)(15)* results in a harmonic vibration. From mechanics it is known that in harmonic vibration the diagram of velocities of the vibrating body, plotted against its position, is either a circle or an ellipse, depending upon the scale of the ordinates. Such a diagram of velocities is shown at the bottom of Fig. 71. But if the motion of the governor point is harmonic, the curve *(11)* *(7)* *(18)* *(9)* is also an ellipse, as will be proved further down. The velocity of the governor point is, therefore, always proportional to the instantaneous value of the dynamic excess speed; and since the force caused by the latter is to be counteracted, the counteracting force must be proportional to the linear speed of vibration of the governor point.

An oil gag pot comes very close to furnishing such a resisting force. Its resistance grows with some power of the velocity, and for the slow motion of a governor the resistance is very nearly proportional to the velocity itself. As a rule, the resistance offered by a gag pot is adjustable. Let it be so adjusted that the resistance just equals the dynamic regulating forces caused by the excess speeds, such as *(7)* *(8)*. Then all conditions are fulfilled for the occurrence of the "limiting case." Discussion of the importance of this case is profitably postponed, until the greatest speed fluctuation *(18)* *(19)* $= 2\,u_e$ has been calculated, which will now be done.

With the dynamic regulating forces always balanced by the oil pot, the forces represented by line *(14)* *(15)* produce harmonic vibrations absolutely identical with those treated in Chapter VII. Their equation is $x = A \cos w\,t$, where A is the amplitude (see Fig. 71, bottom), and w is the angular velocity of the auxiliary vector (see Fig. 72). The latter (viz. w) and the time of natural vibration of the governor are interconnected by the equation $w = \frac{2\,\pi}{T_n}$; substituting for T_n from equation (1) Chapter VII, we obtain

$$w = \frac{2\pi}{\pi}\sqrt{\frac{p\,C}{2\,m_e(r_0 - r_i)}} = \sqrt{\frac{2\,p\,C}{m_e\,A_0}} \quad \text{. . . . . (2)}$$

In this equation the letters have the same meaning as before, namely

p = static fluctuation, found from tangent to characteristic,
C = average centrifugal force,

A_0 = radial travel of weights (see Fig. 71),
m_e = equivalent mass, including all valve gear parts moving with the governor (see Chapter VII).

T_n (resp. w) determines the time during which the unbalanced moments M, top of Fig. 71, can act to vary the velocity u of the prime mover. The law which this variation of u follows is found in the following manner: The angular acceleration of the prime mover is (equation No. 1, page 100)

$$a = \frac{M_0}{I} \frac{x}{A_0} = \frac{A}{A_0} \frac{M_0}{I} \cos wt.$$

But the change of angular velocity in the time t is

$$Du = \int a\, dt = \frac{A}{A_0} \frac{M_0}{I\, w} \int \cos wt\, d(wt)$$

$$= \frac{A}{A_0} \frac{M_0}{I\, w} \text{ (difference of } \sin wt) \quad \ldots\ldots\ldots\ldots (3)$$

If we start with $t = 0$ at the vertical *(2) (3) (1) (11)*, the initial value of $\sin w\, t$ is 0 and we have

$$Du = \frac{A}{A_0} \frac{M_0}{I\, w} \sin wt.$$

Forming the expression $\sin^2 wt + \cos^2 wt = 1$, we obtain

$$\frac{(Du)^2}{\left(\frac{A}{A_0} \frac{M_0}{I\, w}\right)^2} + \frac{x^2}{A^2} = 1,$$

which is the equation of an ellipse. This is the proof for the elliptic shape of the curve *(11) (7) (18) (9)*, promised on page 103.

The maximum value for Du, namely u_e = *(10) (18)* follows by substitution of $\frac{\pi}{2}$ for $w\, t$. Hence

$$u_e = \frac{A}{A_0} \frac{M_0}{I\, w} = \frac{A}{A_0} \frac{M_0}{I} \sqrt{\frac{m_e\, A_0}{2\, p\, C}} \quad \ldots\ldots\ldots (4)$$

Equation (4) furnishes the magnitude of the never-ending speed fluctuation of the limiting case. In the discussion of this case the following statements will be taken up:

(1) Stability of regulation can always be enforced by an oil pot, provided that p is positive.

(2) The speed fluctuation given by equation (4) is the smallest fluctuation possible for a given sudden change of load.

(3) Stability of regulation and smallest possible speed fluctuation depend in part only upon the properties of the governor. A large part of the responsibility for correct regulation rests with the prime mover.

The correctness of statement No. 1 is practically self-evident from the reasoning followed in the present chapter, and becomes very clear from the fact that the dynamic regulating force *(7)(8)* of Fig. 71 and 73, no matter how great, can always be balanced by an oil gag pot. However, it may also be deduced mathematically from equation (4) by the relation that regulation is stable as long as u_e is not imaginary. Now, the only quantity under the radical which can become negative is the static fluctuation p. All other quantities are necessarily positive. Accordingly, the radical can never become imaginary as long as p is positive.

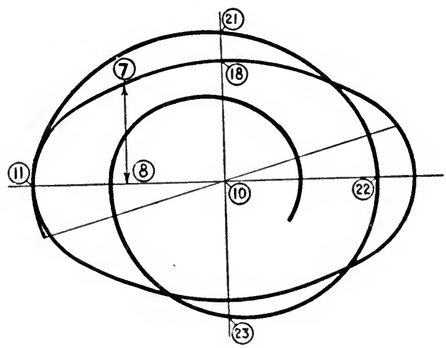


FIG. 73

It should, however, be thoroughly understood that an oil pot, while producing stability of regulation, does not prevent changes of speed; when the load changes, it can only keep them within limits, the extent of which is given in equation (4).

For proof of statement 2 refer to Fig. 73. It will be remembered that the ellipse *(11)(7)(18)* represents not only the speed changes, but also the oil pot forces of the limiting case; from that it follows that damping the vibrations out of existence requires a tightening of the oil brake. But the latter action retards the motion of the governor still more. The speed must vary correspondingly more for the first wave, so that *(10)(21)* is greater than *(10)* *(18)*. If, on the other hand, the oil pot forces are reduced below the value required for the limiting case, the speed fluctuation of the first wave is reduced, true enough, but the amplitudes of the governor vibration and with them the speed fluctuations continually increase, so that after a few waves they exceed u_e of the limiting case.

While the speed fluctuation of a practical case of stable regulation (for instance *(10)* *(21)*) can be computed numerically, the general solution for this fluctuation presents almost insurmountable difficulties. And since even the numerical calculation of an individual case involves exponential functions, the use of equation (4), giving the minimum possible fluctuation, recommends itself on account of its simplicity.

For proof of statement No. 3, equation (4) will be transformed as follows:

$$\frac{u_e}{u} = \frac{1}{2}\frac{A}{A_0}\frac{M_0}{I\,u}\sqrt{\frac{2\,A_0\,m_e}{C}}\sqrt{\frac{1}{p}} \quad \text{(4a)}$$

The fraction $\frac{u_e}{u}$ equals the relative speed fluctuation.

It can easily be shown that the factors $\frac{1}{2}\frac{A}{A_0}\frac{M_0}{I\,u}$ depend upon circumstances over which the governor has no control. For $\frac{A}{A_0}$ is that fraction of the total capacity of the prime mover which is suddenly put on or removed. For the developing of complete sine harmonic vibrations the upper limit of this fraction is $\frac{1}{2}$. If values of $\frac{A}{A_0}$ between $\frac{1}{2}$ and 1 are considered, the total speed fluctuation *(18)* *(19)* of Fig. 71 cannot develop, but equation (4) may still be used as an approximation for finding the fluctuation *(10)* *(18)* of the first wave. Evidently the speed fluctuation is proportional to the fraction of total load suddenly removed or applied.

The factor $\frac{M_0}{I\,u}$ is of particular interest. It has a physical meaning which is revealed by a test for dimensions. Substitution of dimensions for moment, moment of inertia and of angular velocity furnishes

$$\frac{\text{length} \times \text{force}}{\frac{\text{force}}{\text{length}} \times \text{time}^2 \times \text{length}^2 \times \frac{1}{\text{time}}} = \frac{1}{\text{time}}$$

Hence $\frac{I\,u}{M_o}$ is a special time. From the equation

unbalanced moment $\times$ *time* =

moment of inertia $\times$ *change of angular velocity,*

it follows that $\frac{I\,u}{M_0}$ is that time in which the maximum torque[1] M_0 of the prime mover accelerates the rotating masses of the latter from rest to their regular velocity u. Professor Stodola gave it the name "starting time." Hereafter it will be denoted by T_s. The additional relation $\frac{1}{2}T_s = \frac{1}{2}\frac{I\,u}{M_0} = \frac{\frac{1}{2}\,I\,u^2}{M_0\,u}$ shows that $\frac{1}{2}\,T_s$ is the ratio of kinetic energy stored up in the revolving masses at normal speed to the maximum power of the prime mover. $T_s = \frac{I\,u}{M_0}$ then depends solely upon the kinetic energy stored up in the revolving masses (flywheels, turbine disks, etc.) per unit of capacity of the prime mover, and the smallest possible speed fluctuation depends directly upon this quantity. This very important fact has been too often overlooked — and is still being overlooked — by builders of engines and turbines who design or buy governors solely upon the basis of static fluctuation and who later on wonder why the actually occurring speed fluctuation is "so much greater than it should be theoretically." As a matter of fact, theory teaches in full agreement with practice that with insufficient kinetic energy of the rotating parts regulation is very poor. Either the governor hunts continually, or, if the governor has been quieted down by the never failing remedy of an oil pot, excessive speed fluctuations appear at the time of a change of load.

[1] Average during one revolution.

The remaining factors of equation (4a) depend mainly upon properties of the governor. The radical $\sqrt{\frac{2A_0m_e}{C}}$ will easily be recognized by comparison with equation (1) of Chapter IV to be the traversing time of the governor, or rather that traversing time which exists when the centripetal force of the governor has to accelerate not only the masses of the governor proper, but also those of the valve gear parts which are mechanically connected to the governor. This fact should remind designers of engines and turbines that the promptest governor can be handicapped by being required to move massive valve gear parts.

With these simplifications the relative speed fluctuation — to either side from mean — of the limiting case boils down to

$$\frac{u_e}{u} = \frac{1}{2}\frac{A}{A_0}\frac{T_g}{T_s}\sqrt{\frac{1}{p}},$$

so that the total speed fluctuation is

$$\frac{2u_e}{u} = \frac{A}{A_0}\frac{T_g}{T_s}\sqrt{\frac{1}{p}} \quad \dots\dots\dots (5)$$

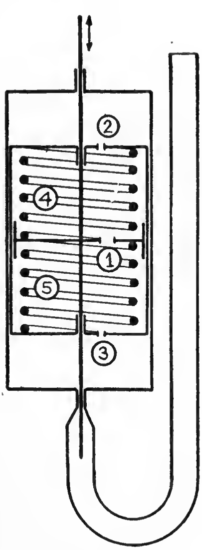


FIG. 74

The only quantity in this equation which remains to be discussed is the influence of p, the static fluctuation. Evidently, the speed fluctuation grows, as p is reduced, which proves conclusively that governors with small static fluctuation — that is to say with great sensitiveness — either produce hunting or else require large oil pots. The rather unexpected result that governors designed for small speed fluctuation produce large speed fluctuations proves that the much and long sought isochronous governor ($p = 0$) is impracticable, because the slightest change of load produces infinitely great speed fluctuations.

It is true that isochronous governors have been successfully used for speed regulation, but in every such case the success is due either to the presence of tangential inertia in the governor (see paragraph 3 of present chapter), or else to the use of floating springs in an oil pot. The latter device temporarily increases the stability of the governor, as will be seen from the following description. Illustration 74 represents diagrammatically — with intentional omission of design features— an oil pot with floating springs. Hole (*1*) in the central piston is large compared to holes (*2*) and (*3*) in the surrounding box piston. Hence any quick motion of the piston rod deforms springs (*4*) and (*5*) so that p is temporarily increased by the difference of the spring forces. If the force upon the piston rod is not vibratory, but steady in one direction, the box piston adjusts itself quite gradually to the new load, or rather new position of the governor, eliminating the forces of the springs (*4*) and (*5*) for static calculations (isochronism) or for final effects of change of load. Air under the piston of an ordinary oil pot acts in a similar way, but it is not dependable on account of leakage.

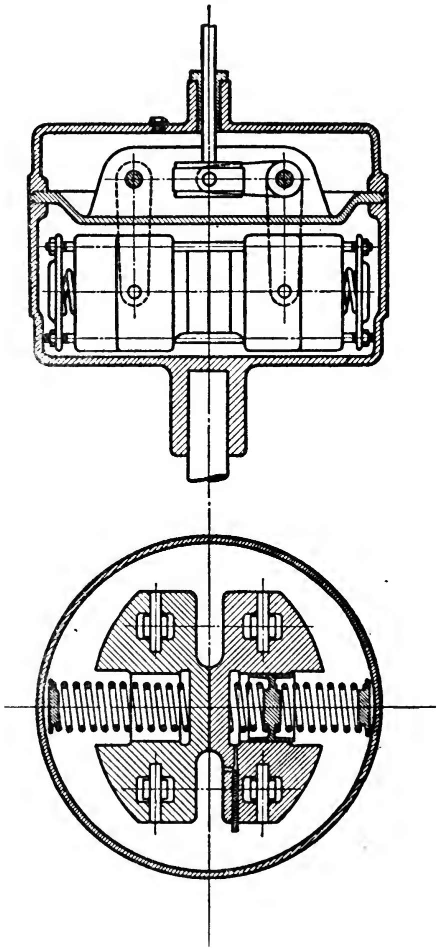


Fig. 75

The yielding or compensating oil pot of Fig. 74 may be made an integral part of the governor proper. Two notable examples of such combination designs are the Chorlton-Whitehead governor and the Bee governor, both of British design. The principle of both governors is the same, and it will suffice to describe one, for instance the Chorlton-Whitehead governor. It is shown in Fig. 75. In the manufacturer's own words it acts as follows:

"The spring on one side is divided into two unequal portions, a plate forming the piston of an oil gag pot being interposed between the two portions. The oil gag pot itself is formed in the weight, and is provided with a small adjustable valve, by means of which free communication can be made from the gag pot to the body of the governor. The bottom casing being entirely filled, and the bottom portion about four fifths full of oil, the governor is ready for work, and the action will be as follows:

"If the gag-pot valve is wide open, so that the oil has a free passage to and from the interior, the gag-pot piston will be quite free to move, and the divided spring will behave as one complete spring, the piston merely acting as a washer.

"The compound spring in this condition is exactly equivalent to the single spring in the other weight, and both are designed to counteract exactly the centrifugal force of the weights in all positions when running at constant speed, — in other words, the governor is isochronous, and to run when in this condition would cause it and the engine to hunt violently, as the governor would have no stability.

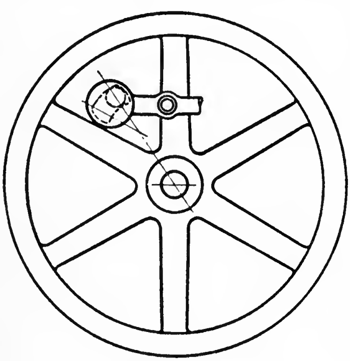


Fig. 76

"If, on the other hand, the valve be closed, the gag-pot piston becomes immovable, and puts the shorter spring out of action. We have thus decreased the effectual length of the spring, and the governor will run with a certain amount, say 8 %, of the variation between inner and outer positions, making the governor very stable — too much so, in fact, for close governing.

"If, now, the gag-pot valve be partly opened, a point can be found where hunting just does not take place, and still the governor will be in equilibrium in any position after a certain time has elapsed, allowing the gag-pot piston to move to its new place; in other words, the governor will be isochronous subject to a certain time lag, which in practice amounts to two or three seconds."

A very ingenious application of the compensating oil pot to shaft governors was made by Armstrong. In his governor, which is illustrated in Fig. 76, the centrifugal mass is not solid, but contains a separate section in the shape of a roller, which fits snugly in an oil-filled cavity of the main centrifugal weight. If the roller were fixed in the weight, the governor would have a considerable static fluctuation; but the cavity for the roller is so located that centrifugal force acting on the roller slowly moves it to new position after a change of load and of governor position, shifting the mass center of the weight and bringing the speed back almost to its original value.

Equation (5) may be used in connection with the relations for T_s for computing the weight of flywheel required per horsepower of an engine, as follows:

$$T_s = \frac{I\,u^2}{M_o\,u} = \frac{W}{g}\,\frac{v^2}{M_o\,u}\,\frac{550}{550} = \frac{W\,v^2}{HP\,g\,550}$$

From these equations follows the weight per horsepower

$$\frac{W}{HP} = \frac{550\,g\,T_s}{v^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

where v = wheel velocity at radius of gyration. In a numerical calculation, compute T_s from (5) and substitute in (6).

To test the applicability of equation (5), an example will now be figured. Given: Maximum dynamic fluctuation = .08, change of load 50%, equivalent to $\frac{A}{A_o} = \frac{1}{2}$, static fluctuation $p = .03$, $T_g = .07$. This is the traversing time of the governor treated on pages 45 and 46, but corrected for mass of the valve gear. To insure a sufficiently rapid dying-out of the vibrations after a change of load, the dynamic fluctuation of the limiting case (never-ending vibrations) must be taken smaller than the given maximum permissible fluctuation. For this reason $2\frac{u_e}{u}$ will be taken to be .06.

With these values, equation (5) furnishes

$$T_s = \frac{.5 \times .07}{.06 \times \sqrt{.03}} = 3.37 \text{ sec.}$$

For cast iron flywheels an average value of v is 80 ft./sec. Then

$$\frac{W}{HP} = \frac{550 \times 32}{6400} = 2.75\ T_s = \text{appr. } 9\tfrac{1}{2} \text{ pounds.}$$

In practice, weights varying from 10 to 70 pounds are installed per horsepower (for v = 80 ft./sec.). While our result agrees quite well with the lower limit of the values used in practice, it is far away from the higher limit. However, there are good reasons for the use of heavier flywheels than indicated by our arithmetical calculation. The principal ones among these reasons are:

(1) The static stability of a governor is seldom constant over the whole range of its travel, so that the inclination of line *(14)* *(15)* in Fig. 71 varies. Similarly the torque curve *(2)* *(5)* in the same illustration is seldom a straight line. But for the use of equation (5) that value of p counts which is derived from the worst instantaneous conditions in these two curves. Hence the value of p to be used in equation (5) is usually less than the value of p which is derived from maximum and minimum speeds of the governor. In poorly designed valve gears of engines and turbines the value of p in equation (5) may be only $\frac{1}{10}$ of the value derived from the governor.

(2) Equation (5) presupposes that the governor is infinitely strong. Actually it must overcome resistances which at intervals block the travel of the governor and change its motion. This problem is dealt with in paragraph 4 of the present chapter.

(3) Equation (5) is based upon the assumption that the action of the governor is continuous and that the effect of such action is instantaneous. In reality, the governor can, in engines, act only once in every stroke; besides, volume of steam or of explosive mixture beyond control of the governor delays governor action in both engines and turbines, so that heavier wheels are

required for stability of regulation (see paragraph 5 of present chapter).

At the conclusion of this paragraph, a hint on the size of oil gag pots may be valuable. With prime movers which are subjected to frequent and heavy load changes, and with valve gears which impress great vibratory forces upon the governor, the oil gag pot of the latter must dissipate a considerable amount of energy. If the oil pot be small, the heat is not radiated away fast enough, the oil heats up, becomes thin, and the gag pot loses its grip. It must then be tightened by closing the oil valve. If the oil pot is slightly too small, the only harm consists in the necessity of frequent adjustments of the oil pot, but if the latter be much too small, the thinning of the oil by the extreme heat will prevent the gag pot's furnishing enough damping power even if the needle valve is entirely closed, because the thin oil leaks between the piston and the cylinder.

Additional statements on proper size of gag pot are given in paragraph 1 of Chapter VIII and in paragraph 4 of the present chapter.

References to Bibliography at end of book: **3, 7, 17, 28, 32, 35, 36, 64, 65, 79.**

3. Influence of Tangential Inertia in Governors upon Stability of Regulation.— Study of the limiting case affords a good insight into the effects which the presence of weights subject to tangential inertia has upon the stability of regulation (see also paragraph 2 of Chapter II).

It was shown in paragraph 2 that, in the limiting case, a governor performs endless vibrations of constant amplitude, with corresponding speed fluctuations of the prime mover. Referring back to illustration 71, let the representative point of the governor be X distant from the position of power equilibrium, or in position *(6) (8) (7)*. Then the prime mover is, on account of the sine harmonic nature of the vibrations, being accelerated by a torque which equals $\frac{X}{A_o}M_o$. Since A_o, M_o and the rotating masses are constant, the acceleration is proportional to X. From paragraph 2 of Chapter II it is known that the regulating force or torque of an inertia governor is proportional to the angular acceleration, as long as the governor does not regulate, or else regulates slowly. In all practical cases the governor motion in question is so slow compared to the rotative speed that its influence upon the regulating force can be neglected. Since in the limiting case the acceleration is proportional to X, the regulating force or torque, caused by tangential inertia, is also proportional to X. Static regulating force in the limiting case, as was shown in paragraph 2 of the present chapter, is likewise proportional to X, so that tangential inertia in this case simply produces a proportional increase of the static regulating force (*6*) (*8*), as indicated in Fig. 77 by the line (*20*) (*21*). This latter line takes the place of line (*14*) (*15*) of Fig. 71. In the limiting case the inclination of this line measures the force (resp. torque) per unit displacement accelerating the regulating motion of the governor masses. Its influence is measured in the equations by the value of the static fluctuation "p." The effect of tangential inertia then is to replace p in equation (5) of paragraph 2 by an "equivalent static fluctuation" p', involving both p (the

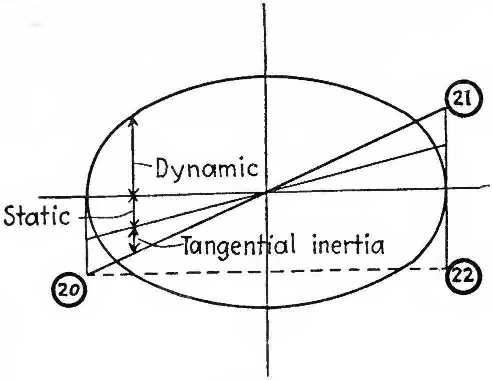


Fig. 77

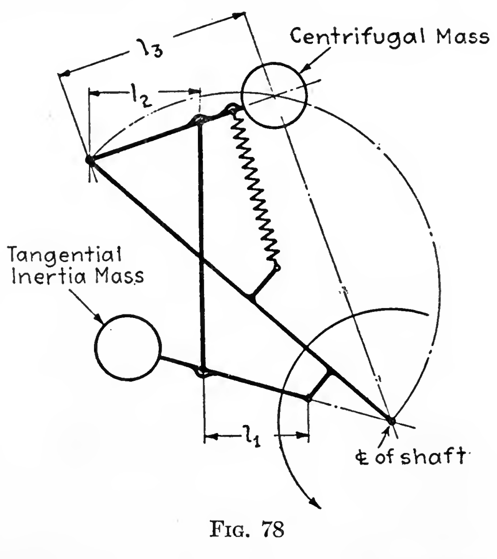


Fig. 78

static fluctuation proper) and a function of the inertia masses of the governor. The latter function is given by the relation that the quantity which is to be added to p in the shape of an increase of static fluctuation must produce the same regulating moment of the governor masses which the inertia moment produces. Call p'' the added quantity, then the regulating moment for removal of $\frac{1}{2}$ of the total load is $2\,C\,l_3\,\frac{\frac{1}{2}\,p''\,u}{u} = p''\,C\,l_3$ —(compare equation (3) of paragraph 1, Chapter II), — where l_3 is the lever arm of the centrifugal mass about its suspension point (see Fig. 78). Turning to the inertia mass, we find that removal of one half of the load produces an angular acceleration of $\frac{1}{2}\frac{M_o}{I}$, (see paragraph 2 of Chapter II), and a regulating moment equal to $\frac{1}{2}\,\frac{M_o\,J}{I}$ about the suspension point of the inertia weight. As before, I is the moment of inertia of the rotating parts of the prime mover, while J is the total moment of inertia of the inertia mass about its suspension point. To refer the regulating moment of the inertia mass to the pivot of the centrifugal mass, multiply it by $\frac{l_2}{l_1}$ (see Fig. 78). We then have

$$p''\,C\,l_3 = \frac{1}{2}\,\frac{M_o}{I}\,J\,\frac{l_2}{l_1}$$

from which follows

$$p'' = \frac{1}{2}\,\frac{M_o}{I}\,\frac{J\,l_2}{l_3\,l_1}$$

Finally, we obtain the equivalent static fluctuation

$$p' = p + \frac{1}{2}\,\frac{M_o}{I}\,\frac{J\,l_2}{l_3\,l_1} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

If the rotative speed of the governor spindle differs from that of the engine or turbine shaft, M_o/I must be multiplied by a factor involving the speed ratio.

Since the stability of regulation depends solely upon the inclination of line *(20)* *(21)*, it is quite immaterial how p' is constituted, and we obtain the most interesting result that p

may be zero, or even negative, provided that p'' is large enough to make p' positive. In other words, if a governor is built with a sufficient amount of tangential inertia, the static fluctuation may be zero (isochronous governor), or even negative (reversed speed curve), and yet the stability of regulation will be preserved.

Many inertia governors can — and this circumstance is well known to operating engineers — be so adjusted that the engine runs at a higher speed with full load than it does at no load. Between the years 1892 and 1905 this fact was widely advertised, so much so that some engines appeared to consist principally of a wonderful governor and a few other, relatively unimportant parts. The total disappearance of these advertisements suggests the inference that regulation with a reversed speed curve is of no lasting commercial value. Reasons for the latter statement are apparent.

For many purposes, particularly for parallel operation of alternating current generators, a uniform, positive static fluctuation is necessary. A positive speed drop furnishes the only means of distributing the load between the different generating units in the power plant. For this reason, prime movers for alternating current generators are seldom, if ever, equipped with inertia governors. The rapid growth of central station power plants generating alternating currents has deprived the inertia governor of its importance. The passing of the inertia governor from the central station should not be misconstrued into the belief that inertia governors are not suited to alternating current work; they are so suited, but they offer no advantages whatsoever except as shaft governors. For mathematical proof of this statement refer to equation (4) of paragraph 2 of present chapter. This equation, it will be remembered, gives the speed fluctuation of the limiting case. Under the radical stands the expression $\frac{m_e}{2\,p\,C}$ which, in words, equals the equivalent governor mass divided by static regulating force. To make the speed fluctuation small (that is to say: to make regulation good), we must make $\frac{m_e}{2\,p\,C}$ very small. But in the case of the inertia governor, equivalent governor mass includes not only centrifugal masses, but also the mass of the inertia weights, so that m_e is increased, which is undesirable. If tangential inertia of centrifugal masses is utilized, instead of using separate centrifugal and tangential masses, the equivalent mass is increased just the same, because, for a given product of centrifugal force times radial path of this force, the product of total mass times total travel of mass must be increased over what it would be if the weights traveled radially, in which latter case they are subjected to centrifugal force only, but not to tangential inertia. On the other hand, the regulating force has also been increased by the addition of inertia force. Thus both numerator and denominator of what may be termed the promptness factor have been increased. Evidently, addition of tangential inertia is useful, where reduction of p to a very small value is demanded, because it keeps the denominator of the promptness factor from becoming almost zero. But where p must have a definite, positive value, say from .02 to .04, the fraction $\frac{\text{total equivalent mass}}{\text{total regulating force}}$ cannot be reduced by the addition of tangential inertia. Attention is again called to the fact that this fraction represents the "quality factor" of the governor in equations (4) and (4a) of paragraph 2. Nothing can be gained by application of tangential inertia to the spring-loaded, high-speed governors of our modern steam turbines, hydraulic turbines, and gas engines.

The logical conclusion to be drawn from this reasoning is that tangential inertia should be limited to shaft governors which have to handle heavy valve gears. See also paragraph 1 of Chapter VIII.

References to Bibliography at end of book: **2, 7, 32, 33, 36, 43, 65.**

4. Solid Friction as a Damping Agent. — Wischnegradsky's theorem that stable governing is impossible without the use of an oil gag pot was not taken seriously by engineers at the time of its publication, because of the great number of governors working satisfactorily and giving excellent regulation without the use of an oil brake. At the same time, governors in those days (1877) were not free from friction. The thought, therefore, suggests itself that solid friction may, under certain conditions, satisfactorily take the place of liquid friction.

Unfortunately, the exact calculations, as soon as solid friction is substituted for liquid friction, become so complicated, even in the comparatively simple limiting case, that we must be satisfied with approximations. Three cases will be considered, — first, that of friction caused by compound centrifugal force; second, the case of constant friction; and third, the case of friction which is greater at rest than it is during motion.

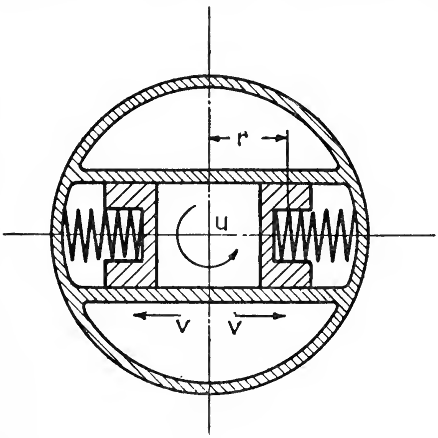


Fig. 79

For the purpose of studying the action of compound centrifugal force (sometimes called Coriolis' force), suppose that in the governor of which Fig. 79 is a plan view, two weights, each of mass m, are rotating with an angular velocity u. The springs are so dimensioned that they balance the centrifugal forces of the weights in any position. Let the weights move outward with a linear velocity v. Then a force is exerted between each weight and its guiding wall by reason of two separate actions. First, the weight moves into a larger orbit so that its absolute velocity must be increased; and second, the absolute velocity of each weight changes its direction. In time dt the radius of the orbit of the weight is changed from r to $r + v\,dt$ which means that the peripheral velocity of the weight is increased from $u\,r$ to $u\,(r + v\,dt)$ so that change of velocity $dv = u\,v\,dt$. But *force* $\times$ *time* = *mass* $\times$ *change of velocity*, or in symbols $Q_1\,dt = m\,u\,v\,dt$, and $Q_1 = m\,u\,v$ for each weight. In the same time dt the velocity v has been turned through the angle $u\,dt$. Figure 80 shows that the change of velocity is $u\,v\,dt$ from which follows the acceleration $u\, v\, dt/dt = u\, v$, and the force $Q_2 = m\, u\, v$ for each weight. The total force exerted by each weight against its guiding surface equals the sum $Q_1 + Q_2 = 2\, m\, u\, v$. The force which each moving weight exerts against the constraining side wall is directly proportional to the velocity v of the vibrating motion. If we now assume that the coefficient of friction is constant, then we find that the damping friction is also directly proportional to the velocity of vibration of the governor. If, therefore, all other friction be eliminated and only that friction which is caused by compound centrifugal force be allowed to act, then and in that case the damping action is identical with that furnished by liquid friction. If, in addition, the coefficient of friction is of such magnitude that $2\, f\, m\, u\, v$ just equals the dynamic regulating force, all conditions which are necessary for the limiting case are fulfilled, and all the calculations of paragraph 2 of the present chapter apply without change.

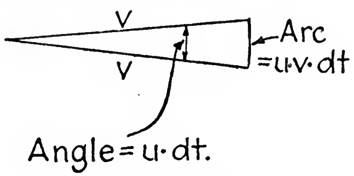


FIG. 80

Friction caused by compound centrifugal force is present in every governor with practically no exception. If, for instance, the governor illustrated in Fig. 2 is considered, it will be seen that compound centrifugal force tends to bind the joints at the top of the spindle across the corners to such an extent that a very effective damping action is secured — often, however, at the expense of rapid wear in case of repeated sudden changes of load.

In practice, friction due to compound centrifugal force can never entirely take the place of the oil gag pot, because it is not adjustable. Since the value of the friction force is $2\, f\, m\, u\, v$, adjustment could be secured only by variation of the friction coefficient f; and such variation at the will of the operator is difficult, if not impossible. Nevertheless, the force under discussion is helpful, and any complications in governor design for the purpose of eliminating this friction are not only superfluous, but entirely out of place. Friction by compound centrifugal force is probably responsible for the correct action of many governors without oil pots and, at least partly so, for the lack of confidence placed in Wischnegradsky's theories at the time of their publication.

In the majority of governors and governing devices, many forces in addition to compound centrifugal force produce solid friction. For the purpose of studying the effect of the latter, let us assume for a while that the resistance caused by friction is constant over the whole travel of the governor.

Referring back to Fig. 71, we remember that the limiting case consists of continued vibrations of constant amplitude about the position of equilibrium. We also remember that this condition is brought about by dissipating (by means of some damping force) the work done by the dynamic regulating force. Figure 81, which illustrates the limiting case with constant solid friction as a damping agent, is very similar to Fig. 71. The principal addition is the line *(9)(15)*, which has been drawn in such a way that the work areas shaded horizontally and vertically are equal, which makes the rectangle *(11)(9)(15)(13)* equal to the half ellipse *(11)(18)(13)*. At this point the approximation begins. With solid friction, curve *(11)(18)(13)* cannot be an exact ellipse, because the latter is based upon the equality (at *any* time) of dynamic regulating force and of damping force. However, the difference between the ellipse and the curve replacing it will, except under special conditions explained below, be so small that we may well use the ellipse as an approximation; the true curve most certainly must have a horizontal tangent at point *(18)*.

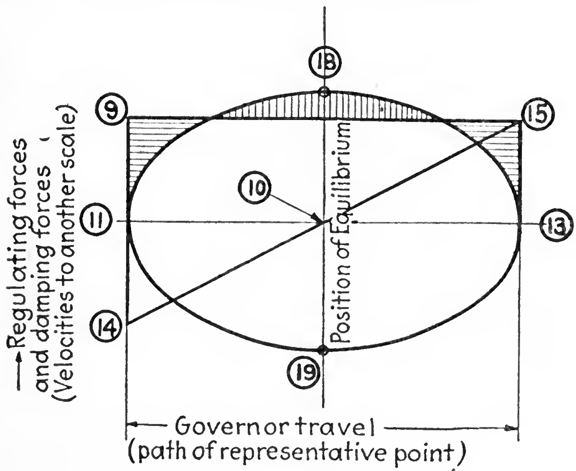


FIG. 81

With the work of the dynamic regulating forces dissipated by friction, the governor moves under the influence of the static regulating force only. The vibrations are very nearly sine harmonic, with constant amplitude. It is, therefore, apparent that solid friction, on general principles, can serve for damping governor vibrations. This statement must be qualified by the restricting explanation that there are two vital differences between liquid friction (including the equivalent action of friction by compound centrifugal force) and constant frictional resistance. The first difference is that, while with an oil pot stability can be enforced, even if the static fluctuation is infinitely small, there exists with solid friction a minimum, below which the static fluctuation cannot be reduced, if stability is to be preserved. The second difference is that, with solid friction, there occur continuous speed fluctuations no matter whether the load varies or remains constant.

For proof of the assertion that, with solid friction, stability requires a minimum value of the static fluctuation, again study Fig. 81. As before, *(11)* *(18)* *(13)* is the near-ellipse of dynamic regulating forces, *(11)* *(9)* = *(13)* *(15)* is the constant damping force of friction so that area *(11)* *(9)* *(15)* *(13)* equals area *(11)* *(18)* *(13)* which (from the well-known formula for the area of an ellipse) means that $(11)\,(9) = \frac{\pi}{4}\,(10)\,(18)$. Let *(11)* *(13)* equal the whole travel of the representative point of the governor. Then line *(14)* *(15)* indicates the regulating forces caused by the smallest allowable static fluctuation p. The motion of the governor could not be continuous, if p were smaller than here shown; for, if *(11)* *(14)* were smaller than *(11)* *(9)*, there would not be enough regulating force at either end of the governor travel to start it. The two halves *(11)* *(18)* *(13)* and *(13)* *(19)* *(11)* of the near-ellipse would be pulled apart up and down in the illustration. Area *(11)* *(9)* *(15)* *(13)* of the damping forces then would no longer equal the area of the dynamic regulating forces, and the speed would fluctuate between wider and wider limits.

The mathematical determination of the smallest static fluctuation which is allowable with solid friction as a damping force must evidently start from the relation that the static regulating force (*11*)(*14*) equals damping force (*11*)(*9*), which latter equals $\frac{\pi}{4}$ times regulating force (*10*)(*18*). But (*10*)(*18*) is caused by u_e of equation (5) of paragraph 2. The cited equation is $\frac{u_e}{u} = \frac{1}{2}\frac{A}{A_o}\frac{T_g}{T_s}\sqrt{\frac{1}{p}}$. For the conditions indicated in Fig. 81 (governor swinging from stop to stop) $\frac{A}{A_o} = \frac{1}{2}$. Since the damping force (*11*)(*9*) only serves the purpose of relating speed difference (*14*)(*11*) to speed difference (*10*)(*18*), we conclude that $\frac{(14)(11)}{u} = \frac{p}{2} = \frac{\pi}{4}\frac{1}{2}\frac{1}{2}\frac{T_g}{T_s}\sqrt{\frac{1}{p}} = \frac{\pi}{16}\frac{T_g}{T_s}\sqrt{\frac{1}{p}}$,

and
$$p\sqrt{p} = \frac{\pi}{8}\frac{T_g}{T_s}$$

or finally
$$p = .54\sqrt[3]{\frac{T_g^2}{T_s^2}} \quad \text{. . . . . (1)}$$

This, as before stated, is the smallest permissible static fluctuation, if solid friction of constant magnitude (referred to the representative point) is employed, and if the governor has constant stability over its whole travel. This latter condition is seldom fulfilled so that actually the smallest permissible value of p must exceed that found from equation (1). It is interesting to note that p can be the smaller, the prompter the governor (T_g small), and the greater the moment of inertia of the rotating masses of the prime mover compared to the capacity of the latter (T_s long).

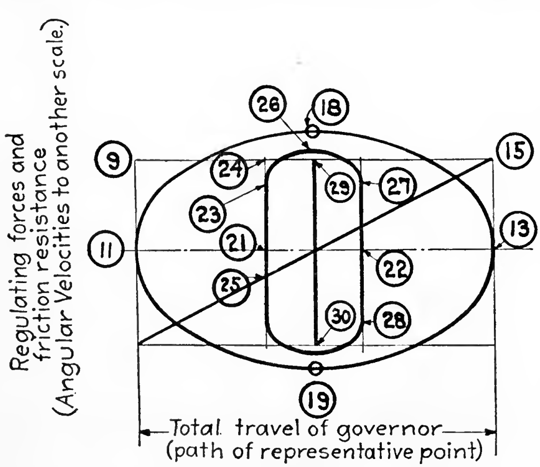


FIG. 82

For proof of the second assertion, namely that constant friction as a damping agent results in never-ending vibrations, even at constant load, refer to Fig. 82, which is similar to the preceding illustrations, but is more complete. The outer curve *(11)(18)(13)(19)* is, as before, the near ellipse of the forces caused by the speed changes resulting from a sudden change of 50% of the total load. *(9)(15)* is the line of constant friction resistance. Now let the engine or turbine operate at a little above half of maximum load (governor position *(21)*), and let the load be suddenly reduced to exactly one half of maximum load. Then the governor will, in the limiting case, vibrate in the range *(21)(22)*. However, the static regulating force now is small compared to the resisting friction, and, before motion of the governor can begin, the speed must rise, until a dynamic regulating force *(21)(23)* has been produced which is determined by the equation *(23)(24)* = *(21)(25)*. The relation between governor motion and speed of prime mover is given by the near ellipse *(23)(26)(27)*. At this last point the governor is detained by friction; the speed must drop through a range corresponding to the distance *(27)(28)* before the return stroke of the governor can take place. The motion is intermittent, discontinuous. Now let the change of load become smaller and smaller. Then the curve of motion will finally shrink into the line *(29)(30)*, which means that even at constant load there will be a continued speed fluctuation of sufficient magnitude to overcome frictional resistance.

Whenever the just described conditions are realized in practice, the speed fluctuation is even greater than is indicated by the distance *(29)(30)*, because the latter equals twice the minimum friction necessary for stability with endless vibrations for 50% change of load, and because in practice a somewhat greater friction will be provided for safety.

From the foregoing explanations it is evident that solid constant friction as a damping agent for governors is rather undesirable under the assumed conditions. However, it will do, after a fashion. On the other hand, friction which is greater between surfaces-at-rest than it is between surfaces-in-relative-motion will not do at all, as may be seen from Fig. 83. If the governor is suddenly displaced the distance (*2*) (*3*) from position of equilibrium and is then released, friction of rest holds it. As before, the speed must rise to a value indicated by point (*4*), determined by the relation (*4*) (*5*) = (*1*) (*2*), before motion can begin. As soon as motion is under way, the dynamic regulating force grows — see curve (*4*) (*6*) — while the damping force of friction drops rapidly. The work of the dynamic regulating force cannot be dissipated by the damping force of friction, and the vibrations increase in amplitude, until one of the limiting stops is struck.

If solid friction really acted as indicated by Figs. 82 and 83, Wischnegradsky's theorem of the necessity of an oil gag pot would not have been doubted at the time of its publication. But as a matter of fact it is extremely difficult to make solid friction behave in the above assumed manner. It can be done in a governor regulating a very well-balanced steam or hydraulic turbine, said governor being driven by well-hobbed gears running in oil, and said governor being entirely free from impressed vibrations. But these ideal conditions did not exist and could not be produced in 1877. Steam and gas engines with considerable cyclical speed fluctuation were the principal prime movers. Governors were driven by belts or by inaccurately cut gears; and most governors were subjected to impressed vibratory forces. We shall presently see that under such conditions solid friction is indeed quite frequently equivalent to an oil gag pot.

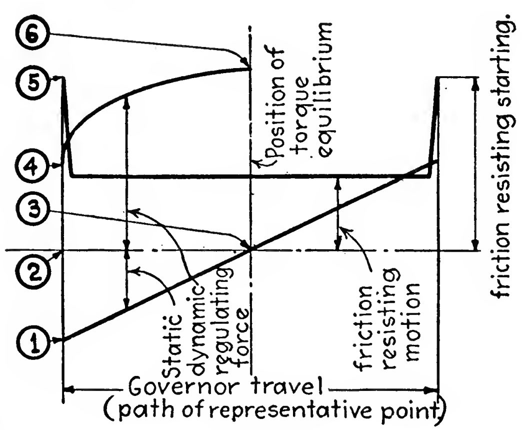


Fig. 83

For the sake of simplicity consider instead of a governor a particle of mass m which is moved back and forth by a force Q reversing its direction every t seconds. Let the motion be resisted by a frictional force F which is quite small compared to Q. Then total change of velocity between opposite directions of motion equals $\frac{Q\,t}{m}$; maximum velocity in each direction equals $\frac{1}{2}\frac{Q\,t}{m}$ and average velocity in each direction equals $\frac{1}{4}\frac{Q\,t}{m}$. Hence, the total displacement of the vibrating particle equals $\frac{1}{4}\frac{Qt^2}{m}$. Now let a small, constant force Q_1 which is smaller than the friction force F, act upon the particle m in the direction of its vibratory motion. Then the center of vibration is shifted the distance $\frac{Q_1 \times \frac{1}{4}\frac{Q\,t^2}{m}}{F}$ every half vibration; for proof, see paragraph 2 of Chapter VIII. The particle, therefore, moves forward with an average velocity $v = \frac{Q_1\,Q\,t}{4\,m\,F}$ which may be written $v = K\,Q_1$, where K is a proportionality factor. The "average" forward velocity of the jerky motion is proportional to the moving force, from which we conclude that the equivalent resistance is also proportional to the velocity, for, if it were not, the velocity could not remain constant for a given Q_1, but would change.

Under the described conditions solid friction is absolutely equivalent to an oil gag pot. While the conditions are somewhat idealized, the action of the forces impressed upon the governor by many valve gears is sufficiently close to the ideal case to make the latter dependable for practical conclusions. The diagram of one complete vibration will, in the language of the other illustrations of this paragraph, look somewhat like Fig. 84, the number of zigzags depending upon the relative frequencies of impressed vibration and natural vibration. Since it is somewhat difficult to visualize the diagram 84, a tachometer record taken from a steam engine, and showing the same action, is reprinted in Fig. 85. The upper curve represents the motion of the governor, and the lower curve illustrates the speed fluctuation of prime mover and governor. The governor was on purpose subjected to greater impressed forces than will ever occur in practice. These forces were not the same at all loads, which accounts for the difference in the governor motion before and after the change of load.

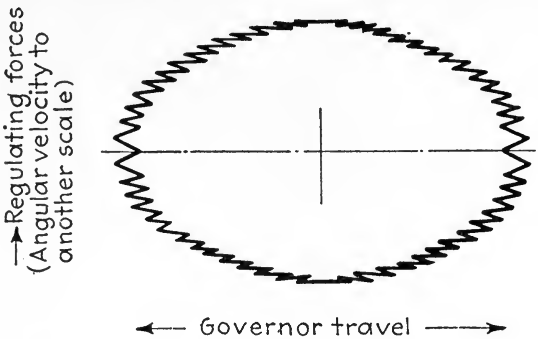


FIG. 84

The mathematically treated ideal case does not hold, if the regulating force Q_1 is greater than the resisting friction F, because the average velocity then grows continually, and the equivalence of solid friction and liquid friction is lost. To

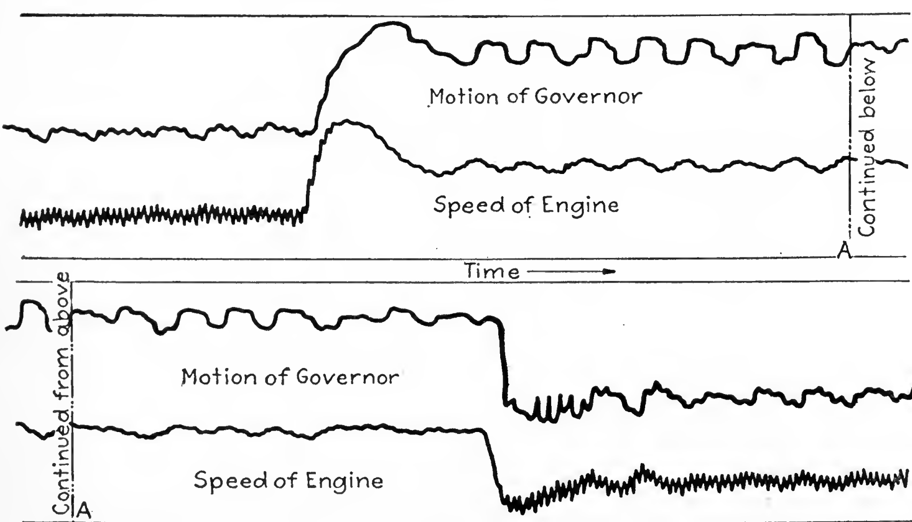


FIG. 85

again make F exceed Q_1 in value, the former may be increased, but there are limits, because the impressed force Q must stay great compared to the friction force F, to preserve proportionality. Fortunately, friction due to compound centrifugal force steps in helpfully just about at the place where solid friction of the vibrating system leaves off.

In addition to the case of a vibrating governor, there is another set of circumstances under which solid friction acts very much like liquid friction, viz. the case of a governor in which solid friction of considerable magnitude is alternately applied and removed, for instance by jar or vibration. To treat this case mathematically assume that the governor is absolutely free from friction during t seconds and is held by quite great friction during the next t seconds, and so on. During each free period the governor — which we will replace by a particle of mass m — reaches a final velocity $\frac{Q_1 t}{m}$, where Q_1, as before, is the regulating force. This velocity is then almost instantly dissipated, and the governor remains at rest during the next t seconds. The average velocity is, therefore, one quarter of the above value, or average $v = \frac{1}{4}\frac{Q_1 t}{m}$. Again, the average velocity is proportional to the regulating force so that the action of heavy friction applied at intervals and relieved at intervals closely resembles the action of an oil brake. The regulation diagram of the limiting case will look somewhat like Fig. 86, the number of steps depending upon the ratio of the frequencies of friction releases and of natural vibration of governor.

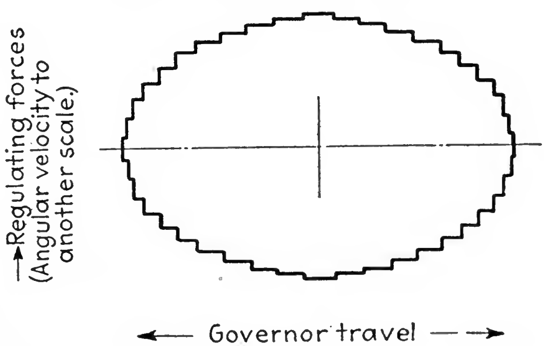


Fig. 86

While the just explained method of damping is effective, it is not as efficient as an oil brake, because it produces greater speed fluctuations. It very much increases the time of natural vibration of the governor, because the latter has to start from rest after each stoppage. However, satisfactory regulation can be obtained in spite of this drawback, if the ratio $\frac{T_g}{T_s}$ is made quite small.

It is now evident that the practical engineers who objected to Wischnegradsky's theorem on the ground that so many governors worked well without gag pots were right in their practical observations. But they did not realize, first, that the governors to which they pointed as examples of correct regulation without damping by liquid friction were damped by other agencies, particularly by solid friction and, second, that solid friction works well only on condition that its action is made equivalent to that of an oil gag pot.

We may add that, the more uniform the angular velocity of the prime mover, the freer the latter from vibration, the smoother the drive of the governor, and the freer the latter from impressed vibrations on the one hand, the smaller is, on the other hand, the value of solid friction as a damping agent, and the more must we approach a frictionless governor with liquid friction for damping. And such is indeed the trend of evolution in the governing of modern steam turbines and hydraulic turbines.

References to Bibliography at end of book: **28, 36, 65.**

5. Greatest Speed Fluctuation with Direct-Control Governing. — In the sales specifications of engines and turbines the following clause is frequently used with regard to regulation, "If *A* per cent of the rated load are suddenly removed or put on, the speed will not vary more than *B* per cent either way from mean speed." No standard values for *A* and *B* have ever been adopted by engine and turbine builders, but the common use of this clause is proof that there exists a definite practical demand for computing the speed fluctuation of at least the first wave after a disturbance. Unfortunately the analytical calculation is extremely unsatisfactory from a practical standpoint. To get an analytical expression, many assumptions must be made, and even then the calculation is far from simple; so far from it, in fact, that it deters all engineers with the possible exception of a few governor specialists. In spite of this condition the elements of the analytical calculation will be given in the present chapter, because they afford a very good insight into the mechanism of regulation.

The assumptions are the same as those made in the preceding paragraphs of this chapter, namely that the action of the governor is continuous, that there is no time lag due to stored-up fluid energy beyond control of the governor, and that all resisting friction is equivalent in its action to an oil gag pot.

Figure 87 shows the relations between the factors entering

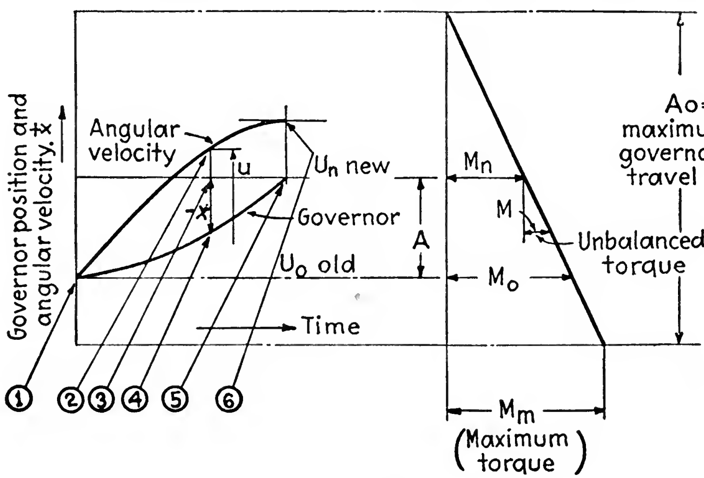


Fig. 87

into the equations. Let both prime mover and governor work in equilibrium with initial or "old" torque M_o, and let the resisting or counter torque suddenly be reduced to a "new" value M_n. Then the torque $(M_o - M_n)$ is unbalanced, and the speed rises as indicated by curve (*1*)(*2*). Under the influence of unbalanced centrifugal force and of tangential inertia the governor moves, as indicated by curve (*1*)(*4*). The lag of the governor motion (*1*)(*4*) behind the speed change (*1*)(*2*) is, of course, due to its mass.

When the governor has reached position (*4*), the unbalanced torque is $M = \frac{-x}{A_o} M_m$. The ordinate x is measured from the new position as a basis, because, as will be seen later, the governor has a tendency to perform vibrations about the new position. The minus sign is used, because in the chosen position M is positive, and x is negative, so that the negative sign is required to make M positive. From the fundamental relation that rate of change of angular velocity equals unbalanced torque divided by moment of inertia, we have $\frac{du}{dt} = \frac{-x}{A_o} \frac{M_m}{I}$. To make this equation applicable to all cases, we introduce the following notations: $\frac{x}{A_o} = X =$ relative change of governor position, and $\frac{u - u_n}{u_n} = U =$ relative speed change; then $du = d(u - u_n) = u_n \frac{d(u - u_n)}{u_n} = u_n \, dU$, from which we derive $u_n \frac{dU}{dt} = -X \frac{M_m}{I}$; but as formerly (in paragraph 2) we have the starting time $T_s = \frac{u \, I}{M_m}$. Throughout this calculation we neglect the (comparatively) very small difference between u and u_n, and finally obtain the equation for the "rate of change of speed deviation" in the following simplified form:

$$\frac{dU}{dt} = -\frac{X}{T_s} \qquad \text{(1)}$$

On the governor act (1) unbalanced centrifugal force, (2) tangential inertia, (3) damping resistances. Turning first to the centrifugal forces, we note that in the position under consideration the speed *(2) (4)* is unbalanced. In the illustration, the scale for velocities has been so related to the scale of governor positions that the curve showing the latter also represents the curve of equilibrium speeds of the governor. At point *(5)* the governor has reached its correct position, but the actual speed having reached point *(6)*, there is still a force urging the governor on. Going back to position *(4)*, we recognize that the distance *(2) (3)* represents the speed difference producing the dynamic regulating force (due to "wrong speed") and that (3) (4) represents the speed difference producing the static regulating force (due to "wrong position"). The speed difference driving the governor is thus $(u-u_n)-\frac{x}{A_o}pu_n$. The second term is positive, because x itself is negative. The force driving the governor then is $2C\left(\frac{u-u_n}{u_n}-\frac{x}{A_o}p\right)=2C(U-Xp)$, if we refer the motion to radial travel of the governor. The motion may be referred to the sleeve, in which case C is replaced by the strength P, or it may be referred to angular swing of weight arm, in which case C is replaced by spring moment of governor. It is evident that the speed relation between prime mover shaft and governor spindle drops out of the equation, because we deal only with relative speed changes.

For the sake of completeness let us consider a combined centrifugal and inertia governor of any one of the types illustrated in paragraph 2 of Chapter II. The diagrammatic drawing of such a governor (Fig. 88) will assist in deriving the equations. From paragraph 2 of Chapter II it is known that the inertia regulating moment is $\frac{du}{dt}(m\,r\,L+J)$. The governor shown in Fig. 88 is of a simple type, in which $m\,r\,L = 0$, so that the regulating moment is reduced to $\frac{du}{dt}J = \frac{du}{dt}m_i L_1^2$. Note that the moment for positive $\frac{du}{dt}$ produces a positive change of x, that is to say it moves the centrifugal weight away from the axis of rotation. If all equations are referred to radial motion, the force exerted by tangential inertia is $\frac{du}{dt}m_i\frac{L_1^2}{L_1}$ at point (1), and $\frac{u_n}{u_n}\frac{du}{dt}m_i\frac{L_1 L_2}{L_3}$ at the mass center of the centrifugal weight.

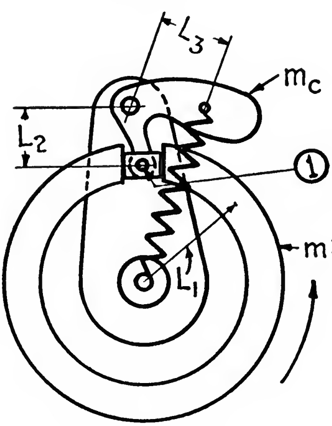


FIG. 88

This expression for the regulating force due to tangential inertia may be written $\frac{dU}{dt} u_n m_i \frac{L_1 L_2}{L_3}$.

The force exerted by an oil gag pot or by equivalent solid friction appears in the form $-K\frac{dx}{dt}$ or $-K\frac{dX}{dt}A_o$. It has the negative sign because it opposes increase of x. Adding up we obtain the sum of all unbalanced forces acting upon the governor, namely

$$2\,C(U - X\,p) + u_n m_i \frac{L_1 L_2}{L_3}\frac{dU}{dt} - K\,A_o\frac{dX}{dt}.$$

This sum of forces equals equivalent mass (m_e) [1] times acceleration so that we obtain

$$m_e\frac{d^2x}{dt^2} = 2\,C(U - X\,p) + u_n m_i \frac{L_1 L_2}{L_3}\frac{dU}{dt} - K\,A_o\frac{dX}{dt}.$$

Further developments will show that the character of the regulation depends upon the relation between certain time elements. With two of these we are familiar, namely the starting time T_s, and the governor traversing time T_g. To introduce the latter, divide the equation by $2\,C$; then the left-hand member becomes $\frac{m_e A_o}{2\,C}\frac{d^2X}{dt^2}$; but $A_o = \frac{1}{2}\frac{C}{m_e}T_g^2$ (see Chapter IV) so that the left-hand member can be written $\left(\frac{T_g}{2}\right)^2\frac{d^2X}{dt^2}$. Similar time values can be introduced for $\frac{u_n m_i L_1 L_2}{2\,C\,L_3}$ and for $\frac{K A_o}{2C}$; studying the latter expression, we realize that $\frac{C}{K}$ is that velocity which an unbalanced force C would finally produce when opposed by the oil-brake. And $\frac{K A_o}{C}$ is the time which is required to traverse the whole travel A_o of the governor

[1] m_e in this case equals $m_c + m_i\left(\frac{L_2}{L_3}\right)^2$.

with said velocity $\frac{C}{K}$. This time may be called the brake-resistance traversing time. It will be denoted by T_b, so that $\frac{KA_o}{2C} = \frac{1}{2}T_b$. Similar reasoning can be applied to the expression $\frac{u_n m_i L_1 L_2}{2CL_3}$. $u_n L_1$ is the linear velocity of the inertia mass m_1. A centrifugal force C would exert the force $\frac{CL_3}{L_2}$ on this mass, so that $\frac{CL_3}{L_2 m_i}$ is a linear acceleration. $C\frac{L_3}{L_2 m_i}T_i = u_n L_1$ where T_i might be called the starting time of the inertia mass. Hence $\frac{u_n m_i L_1 L_2}{2CL_3} = \frac{T_i}{2}$. With these notations the equation of motion of the governor masses is

$$\left(\frac{T_g}{2}\right)^2 \frac{d^2X}{dt_2} - U + Xp - \frac{T_i}{2}\frac{dU}{dt} + \frac{T_b}{2}\frac{dX}{dt} = 0 \ldots . (2)$$

Equations (1) and (2) contain U and X. We can eliminate one at will and, by doing so, obtain the final differential equation either for X (governor deviation) or for U (speed deviation). By differentiation of (2) with regard to t and by substitution from (1) we obtain

$$\left(\frac{T_g}{2}\right)^2 \frac{d^3X}{dt^3} + \frac{X}{T_s} + p\frac{dX}{dt} + \frac{T_i}{2}\frac{1}{T_s}\frac{dX}{dt} + \frac{T_b}{2}\frac{d^2X}{dt^2} = 0 \ldots . (3)$$

The introduction of the various time elements in the theory of governors is due to Professor Stodola (Schweizerische Bauzeitung, 1893 and 1894).

To eliminate X and obtain an equation for U, form the derivatives of X from (1) and substitute in (2). We obtain

$$-\left(\frac{T_g}{2}\right)^2 T_s\frac{d^3U}{dt^3} - U - T_s p\frac{dU}{dt} - \frac{T_i}{2}\frac{dU}{dt} - \frac{T_b T_s}{2}\frac{d^2U}{dt^2} = 0 \ldots (4)$$

By arranging both (3) and (4) with regard to the order of the derivatives, we obtain

$$\frac{d^3X}{dt^3}+\frac{T_b}{2}\left(\frac{2}{T_g}\right)^2\frac{d^2X}{dt^2}+\frac{p+\frac{T_i}{2T_s}}{(\frac{1}{2}T_g)^2}\frac{dX}{dt}+\frac{1}{T_s\left(\frac{T_g}{2}\right)^2}X=0\ldots(5)$$

and

$$\frac{d^3U}{dt^3}+\frac{T_b}{2}\left(\frac{2}{T_g}\right)^2\frac{d^2U}{dt^2}+\frac{p+\frac{T_i}{2T_s}}{(\frac{1}{2}T_g)^2}\frac{dU}{dt}+\frac{1}{T_s\left(\frac{T_g}{2}\right)^2}U=0\ldots(6)$$

The equations for U and X are identical. At first thought this may appear strange, because we know from previous explanation that the curve of governor motion and the curve of speed of prime mover do not coincide. However, this seeming discrepancy will disappear shortly.

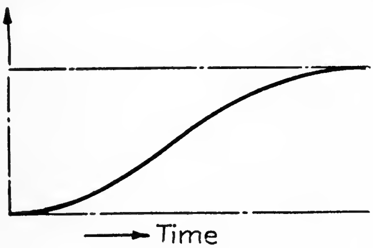


FIG. 89

Equations of the type of (5) and (6) are known in mathematics as linear differential equations of the third order with right-hand member equal to zero. Their integral equations are well known. Let equation (5) be written

$$\frac{d^3X}{dt_3}+K_1\frac{d^2X}{dt^2}+K_2\frac{dX}{dt}+K_3X=0;$$

then the character of the motion represented by the equation depends entirely upon the relation between the three coefficients K_1, K_2, and K_3, all of which are built up of time elements, with the exception of p. Correct governing is a game of getting there first with the lowest terminal velocity. The theory of vibrations teaches that, if the product $K_1 K_2$ is greater than K_3, the governor comes to rest after a change of load. The motion is then represented either by Fig. 89 or Fig. 90. If $K_1 K_2 = K_3$ then the motion is kept up indefinitely with constant amplitude (limiting case, see Fig. 91). If $K_1 K_2$ is less than K_3, the amplitude increases continually.

The case of Fig. 90 is the one most usually occurring in practice. It is represented by the equation

$$X \text{ (or } U) = K_4 e^{z_1 t} + (K_5 \sin z_3 t + K_6 \cos z_3 t) e^{z_2 t} \ldots\ldots (7)$$

where e is the basis of natural logarithms, (2.71828), and where z_1, $z_2 + z_3\sqrt{-1}$, $z_2 - z_3\sqrt{-1}$ are the three roots of the equa-

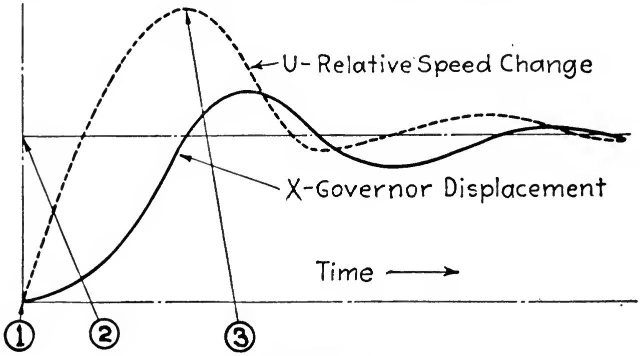


FIG. 90

tion $z^3 + K_1 z^2 + K_2 z + K_3 = 0$. The coefficients K_4, K_5 and K_6 are to be found from the state of motion at the instant when the change of load occurred. They differ in the solution for X, from those obtained in the solution for U, as will easily be under-

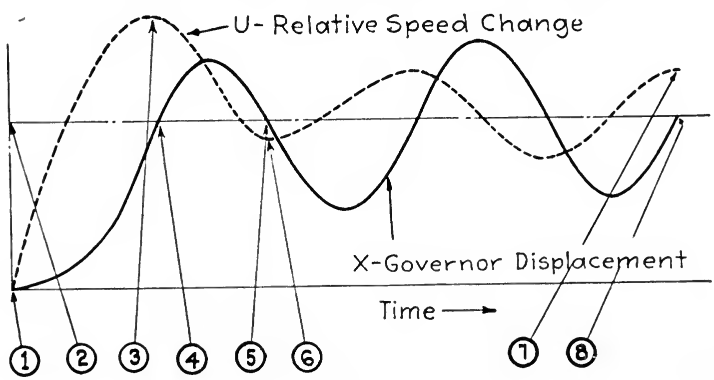


FIG. 91

stood from a study of Fig. 91. At the beginning of a disturbance the velocity curve starts with a finite angle, whereas the governor motion curve must start tangent to the horizontal axis. This explains how the two curves of U (velocity) and X

(governor displacement) can be quite different in spite of having the same differential equations. The coincidence of the latter simply expresses that both vibrations are made stable or unstable by the same combination of time elements, that their period of vibration is identical, that they have the same rate of damping, etc. Although the proof cannot be given here, it may be mentioned that (1) the period (or length of each wave) is always the same, independent of the load change, (2) that both the speed fluctuation and the governor displacement (referred for instance to the first wave) are directly proportional to the load change.

To find this speed variation of the first wave, the derivative of equation (7) for U must be equated to zero, that is to say, we must form $\frac{dU}{dt} = 0$; we must then solve the resulting equation for t, which can be done by trial only, because t appears both as an exponent and as a factor of an angle. The resulting value of t is substituted in equation (7), and U maximum is found.

Creative practice pays no attention to these equations, because they are too complicated. It takes much longer to master and apply them than it takes to build a governor and try it. Besides, the practically important features can be ascertained much more easily from the limiting case. In spite of this condition, no apology is offered for the introduction of this brief sketch of the theory, because no one can ever appreciate the great usefulness of the simple equations of the limiting case, unless he has wrestled for days, or even for weeks, with the solution of the complete equation.

In the advanced volume a few examples of the complete calculation will be given for the benefit of those who wish to make the governing of prime movers a life study.

In estimating the probable maximum fluctuation from the limiting case, we must take care of the assumptions upon which it is based. Referring to Fig. 91, note that the vibration is not sine harmonic at the outset, but soon becomes so. Consequently, the speed changes *(1)* *(2)*, *(3)* *(4)*, *(5)* *(6)*, etc., are not equal to one another. Definitely known from theory are the speed change (*1*) (*2*) $= \frac{A}{A_o} p$, and the amplitude (*7*) (*8*) of the sine harmonic vibration which finally establishes itself, and which equals $\frac{1}{2} \frac{A}{A_o} \frac{T_g}{T_s} \sqrt{\frac{1}{p}}$. But what we are desirous of knowing is the fluctuation (*1*) (*3*) of Fig. 90. As in Fig. 91, (*1*) (*2*) equals $\frac{A}{A_o} p$ in Fig. 90 also; but (*2*) (*3*) is the trouble maker. We may replace (*1*) (*3*) by $\frac{A}{A_o} p + \frac{1}{2} K \frac{A}{A_o} \frac{T_g}{T_s} \sqrt{\frac{1}{p}}$, where K is a stop-gap coefficient, the lowest value of which is approximately 1.2, and the average value of which is 1.6. K may reach 2 or 3, if much energy is stored up beyond control of the governor, as for instance in compound or triple-expansion engines, or in four-cycle gas engines. In the latter case it is better to use the average coefficient 1.6 and to add separately the speed change caused by the stored-up energy. Then we have

$$\frac{du}{u} = \text{(approximately) } U = \frac{A}{A_o} p + .8 \frac{A}{A_o} \frac{T_g}{T_s} \sqrt{\frac{1}{p}} + \frac{E}{I u^2} \ldots (8)$$

where E is the available stored-up energy, I the moment of inertia of the rotating parts, and u the average angular velocity. In a tandem double-acting four-cycle gas engine, E, for a sudden change from full load to no load equals twice the maximum work done by each cylinder per power stroke, because one cylinder carries a compressed rich charge and another cylinder has taken in a full charge of rich mixture. Both of these charges will do maximum work, no matter what the governor does. If there is a large amount of explosive mixture stored between the governor valve and the cylinder, matters are worse.

Since the steam turbine is to-day the principal prime mover, a brief sketch of a method of computing the speed rise due to energy beyond control of the governor will be appropriate. In a steam turbine both the steam in the steam chest, W_s (pounds), and the steam in the turbine, W_t, will do work after the closing of the governor valve. This work is passed on to the turbine with an efficiency e. Let E_1 be the Rankine cycle work of a pound of steam between inital and exhaust conditions (which work can be found from a total heat-entropy chart), then the work done by the steam not under governor control equals

$$E = (W_s + \tfrac{1}{2} W_t)e\, E_1.$$

Not all of W_s does work, because a small amount remains at the end of the expansion; however, it can be neglected. Only one half of W_t is put into the equation, because the steam in the turbine has only about half as much available energy per pound as that in the chest. e is .4 for very small turbines, .65 for the largest turbines.

The relative speed rise is $\dfrac{Du}{u} = U = \dfrac{(W_s + \frac{1}{2} W_t)\, e\, E_1}{I\, u^2}$.

The numerical addition of the three items in speed rise, namely static speed rise, dynamic speed rise under control of the governor, and dynamic speed rise beyond control of the governor is an approximation only. The true calculation is hopeless, as before mentioned. If, for any reason, the exact knowledge of the greatest speed fluctuation should be desired, a point by point method may be used. (See **83,** Ruelf, Der Reguliervorgang bei Dampfmaschinen, and **34,** Koob, Das Regulierproblem in vorwiegend graphischer Behandlung.)

References to Bibliography at end of book: **15, 18, 32, 34, 36, 57, 65, 73, 83.**

CHAPTER X

DISCARDED TYPES OF SPEED GOVERNORS

THE speed governors which were discussed and the theory of which was developed up to this point are based on "Watt's principle," which means that the centrifugal force of revolving masses is opposed by a centripetal force, and that any difference between the two constitutes the motive force of the governor. The inertia governors treated in paragraphs 2 of Chapter II and 3 of Chapter IX are seemingly an exception to this statement; in reality they are not, because they are useful only on condition that tangential inertia is coupled with centrifugal force.

At first thought, Watt's principle seems to be very unsatisfactory, because a change of speed (which the governor is intended to prevent) must occur before the governor can act. It is, therefore, quite natural that inventors of many countries should have worked hard to design governors on the basis of apparently more promising principles. However, the governors based on Watt's principle have survived, while governors based on other principles have disappeared.

Nevertheless, knowledge of the discarded principles is valuable, particularly if it is coupled with knowledge of the reasons why they failed to meet the requirements of practice. For only by such knowledge can people with inventive minds be prevented from reinventing, with the courage of ignorance, mechanisms which were relegated to the scrap heap long ago.

Tangential Inertia

The principle of using tangential inertia for governing was investigated in paragraph 2 of Chapter II. It suffices here to refer to that paragraph. The general idea of using tangential inertia was first published by Werner and William Siemens in

1845. Hence it is known as "Siemens' principle." It is clearly illustrated by Fig. 92. The inertia mass (*1*) tends to maintain a constant speed, no matter what the latter may be. To keep

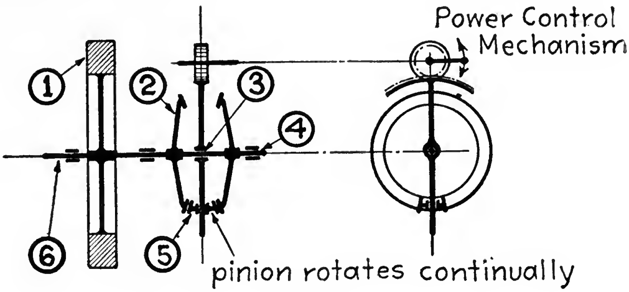


Fig. 92

wheels (*1*) and (*2*) from losing speed and running down, shaft (*4*) must drive shaft (*6*) by means of sufficient friction. Any acceleration or retardation of shaft (*4*) causes the supporting

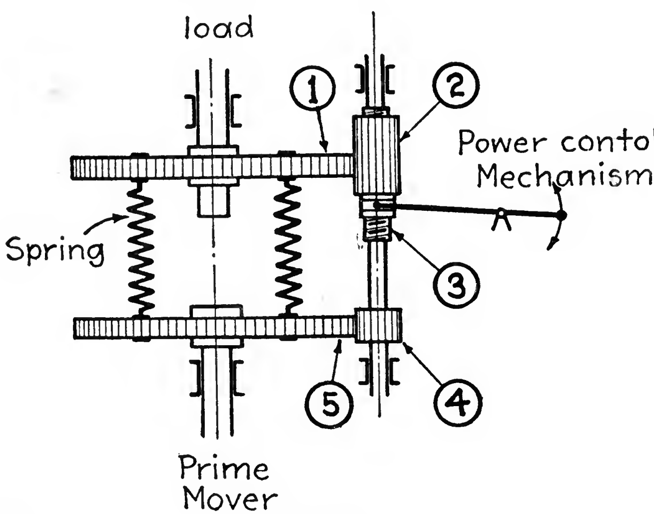


Fig. 93

axle of planetary gear (*5*) to rotate about point (*3*), which motion is utilized to adjust the energy control mechanism.

The reasons why Siemens' principle, by itself, cannot be used for governing were explained in paragraph 2 of Chapter II.

Dynamometric (or Load-) Governors

Poncelet, in 1829, proposed a governing principle which is illustrated by Fig. 93. The torque from prime mover to load passes through a flexible spring coupling. The twist causes gears (*1*) and (*5*) to be displaced with regard to each other. Hence gears (*2*) and (*4*) (in mesh with (*1*) and (*5*)) are also displaced, which causes gear (*2*) to travel along screw (*3*). Axial displacement of (*2*) shifts power control mechanism.

Seeming advantage: The action is practically instantaneous as soon as load changes.

Drawback: The feature which makes this governor impossible is its failure to adjust the power-control mechanism, if intensity of available energy varies (for instance steam pressure, back pressure, head of water in hydraulic turbines, etc.). It has other drawbacks, namely vibrations due to flexible couplings, and never-ending vibrations in case the load changes suddenly (no damping). It does not maintain a constant speed.

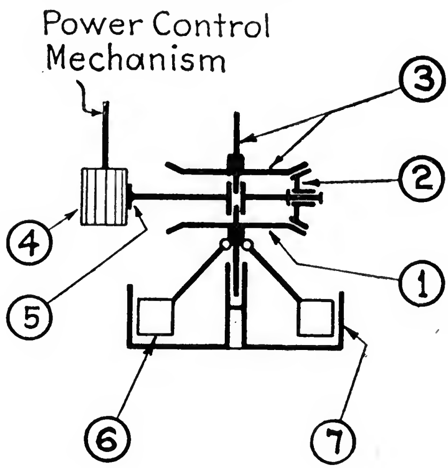


Fig. 94

Chronometric Governors

If the flywheel of Fig. 92 is replaced by a time-piece, which maintains a uniform speed of bevel gear (*2*), then the governor is called a "chronometric governor." This type was invented by Sir C. W. Siemens in 1843. One of the many possible forms is shown in Fig. 94. Bevel gear and spindle (*3*) are driven by the prime mover. Planetary gear (*2*) drives the time-piece (*6*) through bevel gear (*1*), and operates the power-control mechanism through segment (*5*) and gear (*4*). The latter pair does not revolve continually, but moves only to adjust the position of the power-control mechanism.

In the mechanism shown in the illustration, the uniform speed of (*6*) and (*1*) is obtained by the friction brake (*7*), which, if desired, may be replaced by an oil brake. If the weights (*6*) go faster than desired, the friction force is greater, so that their speed is reduced. If they go slower than desired, the friction is reduced, so that the driving mechanism can speed them up.

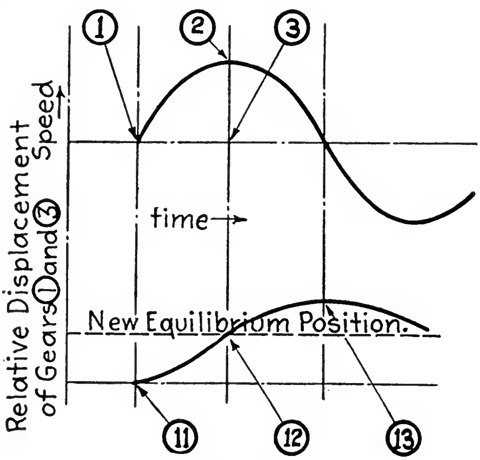


Fig. 95

Compared to the purely centrifugal governor, the chronometric governor possesses several disadvantages. First, the friction brake consumes power, or, if an independent source of energy is employed for the time-piece, its energy is wasted, in addition to the disadvantage of requiring attention.

Second, a "jack in the box" gear is an unwelcome complication.

Third, sudden changes of load produce never-ending speed fluctuations. Let part of the load suddenly be removed at points (*1*) and (*11*), Fig. 95; then gear wheel (*3*) of Fig. 94 will go ahead of wheel (*1*). At points (*2*) and (*12*) of Fig. 95, torque equilibrium exists, and $\frac{du}{dt} = 0$; but wheel (*3*), Fig. 94, still runs ahead (see excess speed (*2*) (*3*), Fig. 95), so that the governor travels too far, namely to point (*13*) instead of point (*12*). The governor oscillates about its new position, and the speed of the prime mover oscillates about the "synchronous speed." The three disadvantages taken together have made it impossible for the chronometric governor to compete with the high-grade centrifugal governor.

Cataract Governors

(Also known as pump governors, hydraulic governors, atmospheric governors or pneumatic governors)

The principle of this method of regulation is clear from Fig. 96. A pump (*1*) driven by the prime mover delivers fluid to a receptacle (*3*), from which it returns to the pump through an orifice (*2*), which is usually made adjustable. The depth of the liquid in the upper receptacle (respectively the pressure, if the fluid be a gas), regulates the supply of energy to the

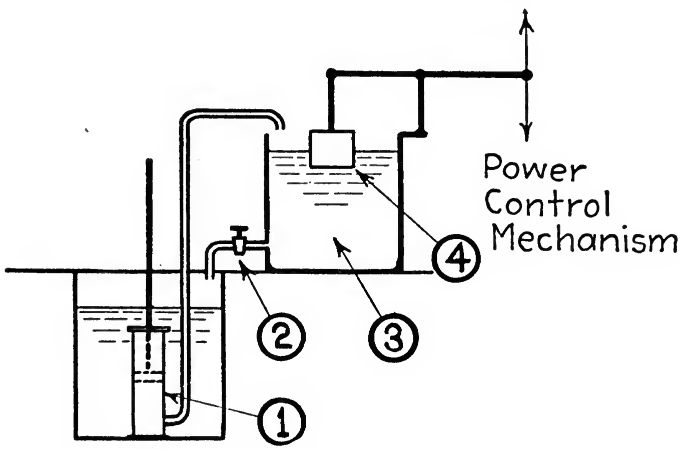


FIG. 96

prime mover. In the illustration, the float (*4*) operates the power-control mechanism.

The cataract principle antedates the centrifugal governor of James Watt. It was used on the Cornish pumping engines. It can be used to-day to govern the number of strokes per minute of direct-acting pumps and compressors (which, as is well known, have neither crank nor flywheel).

For turbines, and for engines with flywheels, the cataract principle has been very little used. The principal drawback is variation of viscosity (and of density) of the working fluid in the governor. If the type in question is to be used for engines or turbines, centrifugal pumps or rotary pumps are preferable to reciprocating pumps.

With proper design, the principle is capable of development, for special purposes of regulation, particularly in connection with direct-acting pumps.

Vane Governors

The resistance which a fluid offers to revolving vanes has been used as a principle upon which to design governors for prime movers. Figure 97 illustrates this type. Vanes (*1*) in box (*2*) are driven by the prime mover. The liquid which partly fills the box is taken along by the vanes with a fraction of their angular velocity. The moving liquid exerts a drag on the

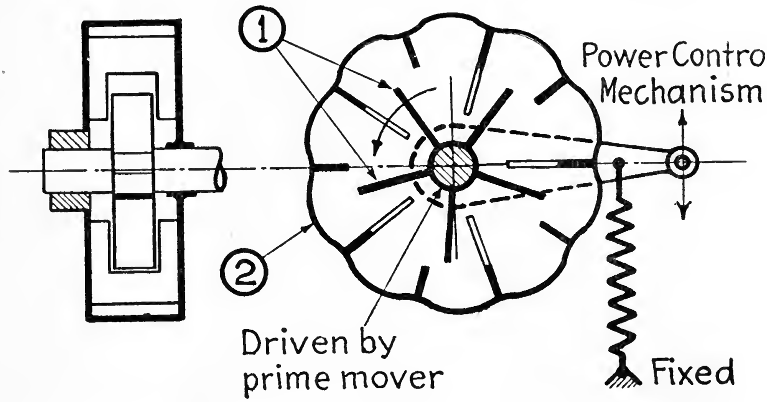


Fig. 97

casing (which is provided with many projections for that purpose). The drag is resisted either by a spring or by a weight.

Governors designed on this principle bear different names (derived from the names of their inventors) in different countries. They always involve a loss of energy. Besides, the dissipated energy is converted into heat, which changes the density and viscosity of the oil (or other fluid) and with them the equilibrium speed of the governor. The principle has been abandoned and is not likely to be introduced again.

Centrifugal Compensator

Recognition of the reasons why an isochronous governor fails to produce isochronous governing led Knowles in 1884 to the invention of a practical solution, the principle of which is illustrated in Fig. 98. Knowles' principle has been discarded, not because it is impracticable, but because the same end can be attained by much simpler means (see paragraph 2 of Chapter IX). In the illustration, (*1*) is a centrifugal governor with sufficient static fluctuation to insure stability of regulation; (*6*) is a similar governor, the regulating travel of which is kept within narrow limits. (*1*) is the main governor, (*6*) is the auxiliary governor. From previous explanations it is known that different speeds belong to the positions (*2*) (*3*) (*4*) of governor (*1*), and would likewise belong to positions (*10*) (*9*) (*8*) of the power-control mechanism, if the length of the link (*3*) (*9*) remained constant. If, however, after a change of load, governor (*1*), and with it the upper end of link (*3*) (*9*), is returned to one and the same position, then one and the same speed corresponds to all positions of the power-control mechanism. This desirable object is gained by making link (*3*) (*9*) adjustable in length and by automatically returning governor (*1*) to the same position. The auxiliary governor (*6*), by means of friction wheel (*5*), pulleys (*7*), and turnbuckle (*11*), slowly adjusts the length of link (*3*) (*9*). Equilibrium obtains if the speed of the prime mover keeps the auxiliary governor free from the friction wheel. The equilibrium speed of governor (*6*) should coincide with the mid-position speed of governor (*1*).

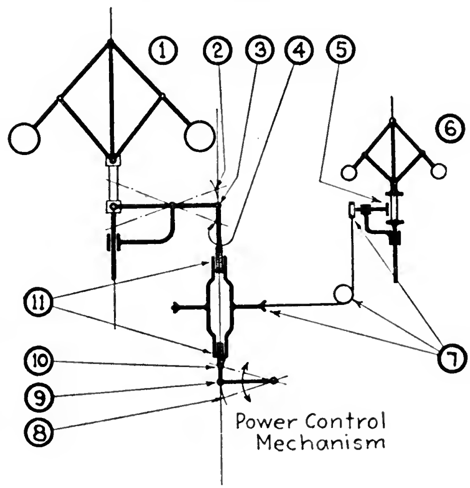


Fig. 98

Governor (*6*) takes the place of the compensating oil pots explained in Chapters IX and XIII; so that it may be called a centrifugal compensator.

While Knowles' principle has been discarded in direct-control governing, a modified form, with one governor only, is extensively used in relay governing (see Fig. 123).

Electrical Governors

At the present time power is transmitted long distances almost exclusively by electrical means, so that prime movers for power generation and transmission (as distinguished from prime movers for pumping, blowing, etc.) are used more and more for driving electric generators to the gradual exclusion of everything else.

Electrical energy is quick acting and is free from mechanical friction. It lends itself beautifully to the design of regulators and governors. In consequence, many attempts have been made to use electricity directly for governing, and to circumvent the comparatively slow mechanical governor.

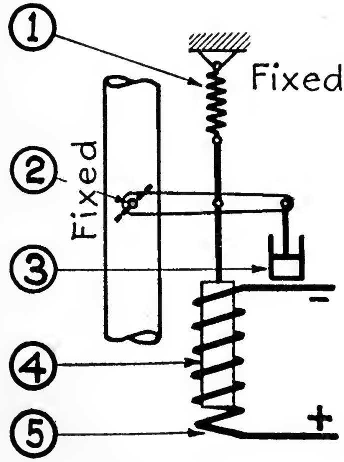


Fig. 99

The principle of an electrical type of governor which was in use for more than a decade will be understood from Fig. 99. Core (*4*) and coil (*5*) form a solenoid, the attraction of which is counteracted by a spring (*1*). If the current in (*5*) is increased, core (*4*) is pulled down, and the throttle valve (*2*) moves toward closed position. In practice, the travel of the core must be kept very small, to avoid hunting. The use of an oil gag pot (*3*) reduces hunting, but does not eliminate it. To keep the travel of (*4*) small, a relay was interposed in practice (see Chapter XIII).

The governor in question was used mainly on constant (direct) current arc lighting machines. Maintaining constant current meant great variation in speed, depending upon the number of lamps in the circuit, and for that purpose the governor was well suited. Direct current arc lighting for streets is to-day a matter of past history. When the direct current arc lighting machine went out of usage, the governor, Fig. 99, went with it.

In alternating current work, it is desirable to maintain a given number of cycles, and to govern voltage by electrical means. But a constant number of cycles means a constant speed of the prime mover, and for that purpose the revolution counter, or, in other words, the centrifugal governor, is the best solution. If maintaining a constant voltage, without any regard to the number of cycles, were the prime consideration, then and in that case it is highly probable that governing would be accomplished to-day by electrical means.

References to Bibliography at end of book: **4, 5, 6, 22, 31, 36, 39, 42, 61, 62, 80.**

CHAPTER XI

GOVERNING FOR CONSTANT RATE OF FLOW

In connection with prime movers driving pumps, cases are encountered in which it is necessary to keep the rate of delivery of the pump constant. The type of governor employed for this purpose varies with the type of pump.

In positive displacement pumps (including reciprocating and rotary blowers), the delivery is, within reasonable limits, proportional to the angular velocity, and independent of the resistance to flow on the discharge side. Consequently, constant delivery is, within the same limits, identical with constant speed, and prime movers driving constant displacement pumps or blowers are, for the purpose in question, equipped with constant speed governors. If the resistance to flow varies widely, constant speed does not mean constant delivery, because the volumetric or delivery efficiency drops in all positive displacement machines, as the resistance is increased. The reduction of volumetric efficiency is due to several causes such as leakage and slip, reëxpansion in clearance volume, heating of incoming gas, reëvaporation (if vapor is pumped) etc., which need not be discussed here. Since increased resistance means increased pressure on the delivery side, the reduction of volumetric efficiency can automatically be compensated for by a slowly applied increase of speed under the influence of increased pressure. (In the case of vacuum pumps, substitute "suction side" and "decrease of pressure" for "discharge side" and "increase of pressure.") There should be no difficulty in accomplishing this result, although no applications of this thought appear in practice.

Matters are quite different with velocity pumps, of which type the centrifugal pump and the turbo-blower are representatives. The rate of delivery is a function of both angular velocity and resistance to flow, so that, for a given speed, the flow may vary within wide limits. Speed governors, therefore, will not do for maintaining constant rate of flow, but a force must be derived from the flow itself. Here a wide field is open to the ingenuity of inventors, because there exist so many methods for deriving a force from the rate of flow. If we glance over the variety of centrifugal governors and remember that they are all based upon one single force action, namely that of centrifugal force, then we begin to realize what a number of "constant volume" governors is possible using either the impact disk, or the Venturimeter, or the Pitotmeter or nozzles and orifices, or constant pressure difference with variable orifice area, or temperature rise methods, etc. With all this variety of possibilities there is a certain similarity between volume governing and speed governing, as will be apparent from the following:

In centrifugal pumps and low pressure turbo-blowers the relation between angular velocity u, pressure p and rate of delivery V is (near enough for all practical purposes) given by the equation

$$p = K_1 u^2 - K_2 V^2,$$

where K_1 and K_2 are design constants peculiar to each machine,[1] see also Fig. 100. If now the pump or blower operates solely against a friction resistance (centrifugal pumps in closed systems, blowers for furnaces, etc.), the pressure generated by the pump is used for overcoming friction resistance only, which may be expressed by the equation $p = K_3 V^2$. (In high lift centrifugal pumps this relation does not hold. For such pumps, however, volume governing is practically never used.)

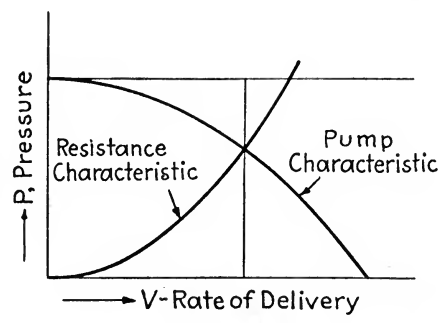


FIG. 100

[1] This equation does not apply to turbo-compressors, on account of the change of density with pressure. But turbo-compressors are governed for constant pressure, and not for constant volume. Therefore, the discrepancy does not matter.

Combining the two equations furnishes $(K_3 + K_2)V^2 = K_1u^2$ or finally

$$V = u\sqrt{\frac{K_1}{K_2 + K_3}} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

Evidently the rate of delivery is directly proportional to the speed, as long as the external resistance K_3 remains constant. But the latter is usually the case. If, for instance, a turbo-blower delivers air to a blast furnace, the changes of resistance in the furnace, although of considerable magnitude, occur so slowly that during the period of one process of regulation (caused, for instance, by a sudden change of steam pressure), the resistance may be considered constant. But if relation (1) holds, all the theories applying to speed governing must also apply to volume governing. There must be a static fluctuation (difference of pressure between great and small resistance), there exists a limiting case of never-ending vibrations, a gag pot is needed to enforce stability, etc. For this reason it is unnecessary to go further into the theory of volume governing, because the theory of speed governing can be applied with small changes.

The two types of "volume" governors which are at the present time in use in the United States are diagrammatically illustrated in Figs. 101 and 102. In both illustrations the force derived from the rate of flow is shown as operating the energy-controlling device directly, whereas, in practice, a relay is used. The latter part of the governors will be found discussed in Chapter XIII.

In Fig. 101 (General Electric) an impact disk is used. The disk produces a pressure difference due to throttling and, at the same time, acts as piston for converting the pressure difference into a regulating force. In Fig. 102 (Rateau, Ingersoll-Rand) a partial vacuum is produced by a Rateau multiplier (two or three Venturi tubes in a set). The suction is transmitted to a separate piston (*4*).

Many of the definitions from the statics of centrifugal governors can be applied here, if properly modified. In both types the "strength" of the governor is found by the force which is required to move the governor when it is not in operation, which in the present case means without flow. The whole strength cannot be utilized in regulation, because that would require cessation of flow. The governor has a work capacity depending upon the governor piston displacement and upon the pressure difference due to the allowable difference in the rate of flow. Both governors have stability and a static fluctuation. In Fig. 101 stability is obtained by cone angle *i* and the spring (*1*). In Fig. 102 it is obtained by the spring (*1*). If the steam pressure varies, or the resistance of the furnace varies, the control valve (*2*) must change its position, which

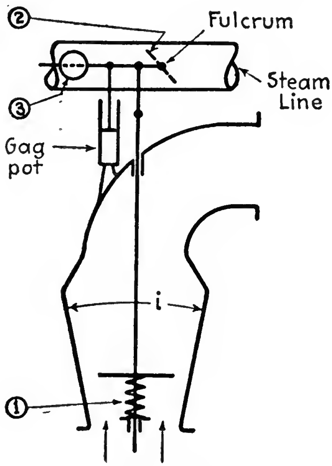


Fig. 101

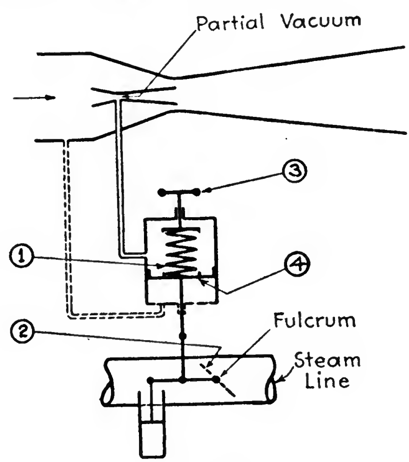


Fig. 102

(on account of the just mentioned stability) means that the rate of flow must change. If the stability is too small, that is to say if it is attempted to regulate too closely, never-ending vibrations will result; the governor will hunt, unless special gag pots, etc., are used (see paragraph 2 of Chapter IX).

Just as in speed governors adjustment of speed is obtained by variation of strength, so in volume governors adjustment of rate of flow is obtained by the same means. In Fig. 101 the strength is varied by the shifting of the weight (*3*) along its supporting lever, and in Fig. 102 the same is accomplished by turning hand wheel (*3*), which varies the initial compression of spring (*1*).

The complete theories of these two governors, including the measuring apparatus for the rate of flow, are not difficult. But they will not be given here, because the governors are patented, which means that the theory would benefit only a very limited circle of engineers.

In connection with volume governors it should be remarked that frequently more is expected of them than they can accomplish. This refers particularly to the application of such governors to blast furnace blowing equipment. In blast furnace practice the furnace, the boilers and the blowers are interdependent. If the furnace hangs (resistance becomes very great), more steam is used, while the gas supply from the furnace to the boilers remains constant. The steam pressure drops, so that still more steam is used. Soon a limit is reached, and the rate of flow of air to the furnace cannot be maintained, unless the heat input into the boilers is increased by other means, such as coal fire. If the quantity of blast falls off, the supply of gas also falls off, the steam pressure is lowered still more, etc. Most obviously neither a constant volume governor on a turbo-blower nor a constant speed governor on a reciprocating blower can obviate or cure such a condition, and should never be expected to do so.

Reference to Bibliography at end of book: **81.**

CHAPTER XII

GOVERNING FOR CONSTANT PRESSURE

In the governing of pumping machinery, the most widespread requirement of regulation is to keep constant the pressure (or suction) produced by the pump, no matter how much the demand or rate of flow of fluid pumped may vary.

Pressure governing differs from speed governing in many respects. Although the reasons for the differences will not appear until later in the chapter, the principal differences will be mentioned here for the sake of greater clearness. First, the influence of pressure governing upon the prime mover is felt more slowly than that of speed governing, because change of pressure is the accumulated effect of speed deviation. Second, different pumps have widely varying characteristics (relations between pressure, rate of flow and speed), so that a governing device which is successful with one type of pump may be a failure with another type of pump. Third, while in speed governing the two variables, namely torque and angular velocity, are, within reasonable limits, independent of each other, the variables in pressure governing are not independent of each other; the resisting torque (average per revolution) of a pump is a function both of the pressure and of the rate of flow. Finally, the word "regulation," when applied to governing for constant pressure, is very elastic. While in air compressor plants for mines and factories variations in pressure of 15% to 20% or even more are allowed, and are considered "close enough," pressure fluctuations on the suction side of gas exhausters for by-product coke ovens must be kept down to 1/10% of the prevailing pressure, and even that small variation is called "not close enough."

The understanding of the phenomena of pressure regulation presupposes familiarity with the characteristics of different types of pumps. For that reason, a brief discussion of the most important features of pump characteristics will now be given.

The two types of pumps on which pressure regulation is practiced are the displacement type and the velocity type. Figures 103 and 104 are indicator cards of a water pump and of an air compressor, both of which are displacement pumps. The section lined areas indicate losses due to friction through valves and ports. These losses increase with the speed, so that displacement pumps have an automatic stability. If the prime mover delivers a given amount of work per revolution, and if the demand upon the pump remains constant, the speed will not increase, for two reasons: First (if the demand remains constant), the pressure produced by the pump would rise with speed, and second, the friction losses would increase with speed. But both of these mean more work per revolution, which excess work the prime mover, under the assumed conditions, is unable to give. In some air and gas compressors the just mentioned stability is very small or even zero. By study of the dotted line in Fig. 104, it is easily seen that, while the friction work below the atmosphere increases, the work of compression above the atmosphere falls off, on account of the reduced delivery, which latter is occasioned by the pressure drop through the inlet valve and ports. The stability is negative in certain compression ranges of vacuum pumps and compressors with fixed discharge pressure, but variable intake pressure. It is well known that such pumps require maximum work per stroke, if the intake pressure equals one third of the discharge pressure (see Fig. 105). Below that intake pressure, more speed means reduced intake pressure, less work per stroke; this in turn means still more speed, etc. The system is naturally unstable. Fortunately, close governing for

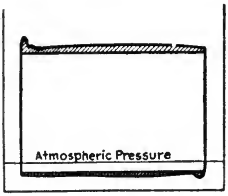


Fig. 103

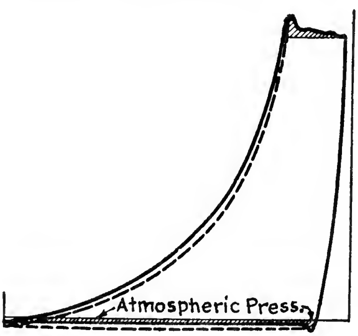


Fig. 104

constant discharge pressure is never required with this wide range of intake pressure.

The prime mover also contributes to stability or lack of stability. Although study of this problem rightfully belongs in the chapter on the self-regulating properties of prime movers, it will be taken up here, as far as it concerns pumping machinery. The prime movers driving displacement pumps are steam engines and internal combustion engines. Only very occasionally steam or hydraulic turbines are used. In steam engines, control is effected either by throttling or by variation of cut-off. In throttle control, there belongs a certain position of the throttle to a given speed and pressure drop. If the engine speeds up with a fixed position of the throttle, the pressure drop through the valve increases, the work done per revolution is decreased, the deficiency of work is taken out of the flywheel, and the speed is reduced again. Hence throttle control results in inherent stability. It is quite different with cut-off control, because one and the same position answers for all speeds and rates of delivery as long as steam- and back-pressure remain constant. In practice, this holds true within certain limits only, because there is always some throttling in the ports, which makes itself felt more and more at higher and higher speeds. But the effect of speed is small, so small as to give very little inherent stability. The latter must be enforced by the proper regulating mechanisms which are described in the second paragraph of this chapter.

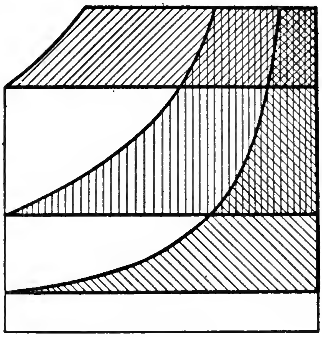


Fig. 105

The characteristic properties of pumps are expressed in characteristic curves. For displacement pumps such "characteristics" are not in common use, because the ideal characteristic of a displacement machine is a vertical straight line, see Fig. 106, which means that the rate of delivery does not depend upon the pressure. (Fig. 106 is drawn for variable discharge pressure and for constant suction pressure.) Actually, several circumstances, depending upon the type of pump and upon the fluid pumped, change the shape of the characteristic. Such circumstances are slip, reëxpansion in clearance space, heating of incoming air or gas, etc. But even with consideration of these features, the characteristics of displacement pumps are almost vertical within the practical range. Three characteristics are shown, namely for unit or normal speed, for $\frac{1}{2}$ speed and for double speed. For a given discharge pressure, the delivery is almost proportional to the speed. As a rule, the delivery does not grow as rapidly as the speed, particularly in air and gas compressors (see Fig. 104), unless an intake pipe is provided of sufficient length to use the inertia of the fluid in it for shoving an extra quantity into the pump at the end of the retardation period (end of the stroke in reciprocating machines).

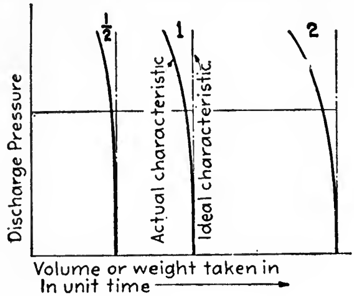


FIG. 106

Turbo-pumps and turbo-compressors have no "indicator cards," so that the features affecting regulation must be studied directly from the pump characteristics. While the characteristics of centrifugal (or turbo-) pumps for liquids differ from those of turbo-compressors, they will not be studied separately in this sketch, because governing for constant pressure is not common in connection with centrifugal pumps, whereas it is quite common with turbo-compressors. In this latter type of turbo-machine speed, torque, pressure, and rate of delivery cannot be related by any rational algebraic formula, because the characteristics vary greatly with the design of the blades, of the diffusor, with the rate of cooling between the stages, with the quantity of air slipping back per stage, and with other details. But in spite of this variety there is sufficient similarity between the characteristics to allow the drawing of conclusions for pressure regulation. In Fig. 107 the characteristics of a turbo-compressor are shown drawn to scale. Arbitrarily, a certain rate of delivery has been called unity, and just as arbitrarily a certain speed has been called unity.

Let p_c be the discharge pressure which is to be maintained constant, on the assumption that intake pressure is naturally constant. (If both intake and discharge pressures are subject to change, matters are more complicated.) The characteristics show that $2\frac{1}{2}$ % of change of speed changes the rate of delivery from .81 to 1.06, or roughly 26 %. This behavior of turbo-compressors has been cited as a great advantage, because, with constant speed governing, the pressure changes so little over quite a wide range of delivery. This property is indeed an advantage in cases where the demand (for air, for instance) stays within narrow limits, because then pressure governing is reduced to the common and well-worked-out problem of speed governing. It may be noted in passing that this part of a turbo-compressor characteristic differs radically from the displacement pump characteristic (Fig. 106). In one case the curve is almost horizontal; in the other case it is almost vertical. Returning to the case in question, it must be stated that cases of limited range of delivery almost invariably turn into cases of wide range of delivery in the course of time, so that the advantage is more apparent than real. The relation between speed variation and delivery variation for a given constant discharge pressure is apparent from the nest of characteristics. If the demand drops below a certain critical value V', which equals about .75 in the diagram Fig. 107, the pressure can no longer be kept constant, but drops rapidly to p_o, rises again to p_c under the influence of the governor, drops back to p_o, and so forth. The turbo-machine surges or pumps, unless special governing devices are provided which either throttle the inlet, or bleed the discharge.

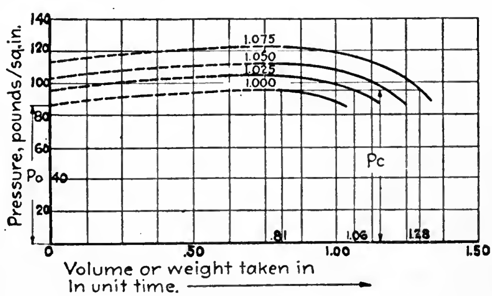


FIG. 107

From a comparison of displacement and velocity characteristics it is evident that maintaining constant pressure against a variable demand requires much greater change of speed in displacement machinery than it does in turbo-machinery.

Turning now to the regulating devices used for maintaining constant pressure against a variable demand, we see immediately that we have two possibilities for regulation, namely (1) to keep the speed of the pump and prime mover constant and to vary the output of the pump proper, and (2) to vary the speed of the prime mover. Method No. 1 is obviously independent of the type of prime mover, and may be applied just as well if the pump is driven by an electric motor or by belt from a line shaft. Devices for varying the output of the pump in spite of constant angular velocity were indeed originally developed for belt-driven and for motor-driven pumps. Although they are now, for the sake of uniformity of pump design, frequently used on engine- or turbine-driven pumps, they do not belong in a book on the governing of prime movers, and will not be described here.[1] In the application of the second method the most natural step to be taken consists in deriving a force from the pressure produced by the pump and in opposing it by a known force, as indicated in paragraph 1 of Chapter I. Any difference between the two forces is used to move the energy-controlling mechanism. This method leads to devices which are diagrammatically illustrated in Figs. 108, 109 and 110. In these illustrations no attempt has been made to impart features of machine design. In these devices the pressure, acting upon some form of piston, tends to lift a weight (*1*) or to deform a spring (*2*), or to do both. If the weights (*1*) only are used, the tendency is to maintain a constant pressure irrespective of the position of the float, bellows, or piston. If the force of a spring such as indicated by (*2*), Fig. 108, is added to the weight, the tendency is to adjust a different pressure for each position of the float, bellows, or piston.

In this description, the phrase "the tendency is to" was used advisedly, because these devices will, under certain conditions, govern satisfactorily, and will result in abominably

[1] The principal means used are by-passes, throttling of inlet, keeping inlet or outlet valves open, varying clearance space, starting and stopping; many of them are known as unloading devices.

poor regulation under different conditions. The safest way to determine which of the two will happen would be to derive and discuss the complete differential equations of motion. But pressure regulation, as before stated, has one more variable than speed regulation, which makes the differential equations of pressure regulation more complicated than those of speed regulation. For that reason an attempt will be made to construct a limiting case between stability and instability in a manner similar to that followed in speed governing. This

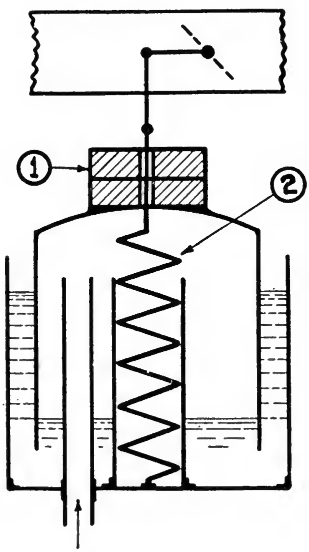


FIG. 108

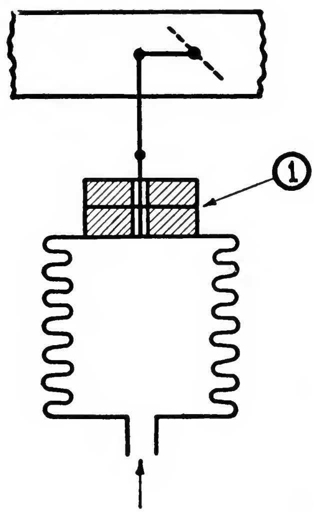


FIG. 109

limiting case will be derived with the aid of Fig. 111, which represents a diagram of an air or gas pump (*2*) delivering into a container of volume V_2 from which the fluid is discharged in a steady stream through a nozzle (*1*). Let the pump deliver $V'u$ cubic feet of fluid per second, where u is the average angular velocity of the pump. Valve (*4*) controls the flow of steam to the prime mover (not shown) operating the pump. Now let the governor be set in motion up and down. Under what conditions will governor, speed and pressure perform sine harmonic vibrations about the values of equilibrium?

Let x be the displacement of the governor from equilibrium position, then

$$\frac{du}{dt} = -\frac{x}{A_o}\frac{M_m}{I} \quad \text{.................} (1)$$

which is true upon the assumption that x is small, which means that the pressure variations are so small as not appreciably to affect the unbalanced moment. A_o, as before, is the whole governor travel, M_m is the maximum torque, and I is the moment of inertia of all rotating parts.

Let Du be the angular velocity in excess of the average; then we find the

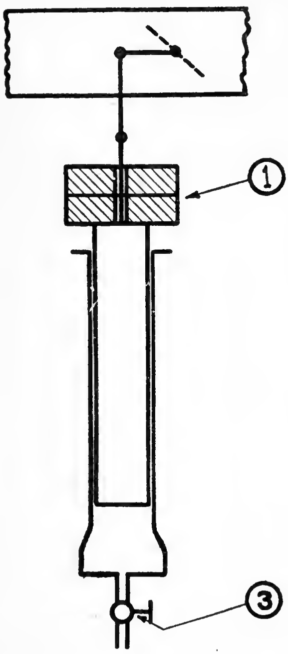


Fig. 110

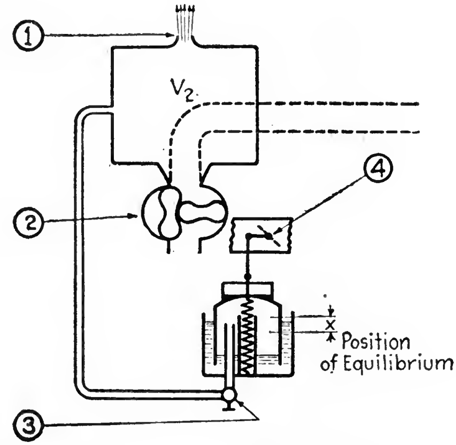


Fig. 111

pressure rise in time dt from $(DuV'dt + V_2)p = V_2(p + dp)$ from which we have

$$dp = Du\frac{V'}{V_2}p\,dt \quad \text{.................} (2)$$

Turning to Fig. 112, assume that the governor performs sine harmonic vibrations of the amplitude $\frac{1}{2}A$. Then the speed will also fluctuate up and down with a phase lag of 90° behind the governor motion. But with a container as shown in Fig.

111, the pressure lags 90° behind the speed, the pressure being a maximum when the excess speed disappears. This means that the pressure lags 180° behind the governor. Hence, the pressure change is (negatively) proportional to the governor displacement, and the pressure curve in Fig. 112 is a straight line *(1)(4)*. Comparing Fig. 112 with Fig. 111, we note that excess pressure, such as that at point *(4)*, urges the governor still farther away from mid-position. But to get sine harmonic motion, we must subject the governor to a force which pulls

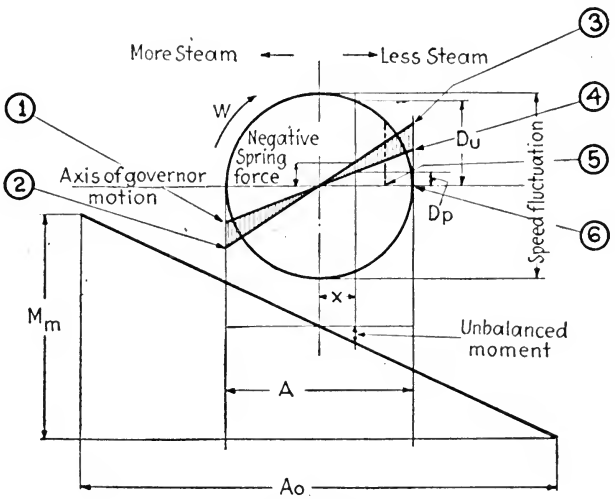


FIG. 112

it back to equilibrium position and which is proportional to the displacement x. A stability spring must therefore be added to return the governor. If line *(2)(3)* in Fig. 112 represents (negatively) the spring forces, then only the difference between spring force and excess pressure force is available for returning the governor, as indicated by vertical section-lining.

The angular velocity w of the auxiliary vector of the sine harmonic vibration is determined by the section-lined ordinates, because $w = \sqrt{\frac{P_1}{m}}$, where P_1 is the restoring force at unit displacement. At the end of the governor swing the restoring force is (3) (4) = (3) (6) − (4) (6) or $\frac{A}{2}S_1 - G\,D_o p$, where S_1 is the spring force for unit deformation, G = cross-sectional area of governor float, $D_o p$ = excess pressure at the end of governor swing. Now suppose the stiffness S_1 of the spring to be adjustable. Make the spring more flexible. Then w will become smaller, the excess speed will grow, and the excess pressure will grow. The section-lined strip becomes smaller, first, because point (3) comes down, and second, because point (4) goes up. A limit is soon reached. There must be a minimum scale $S_{1\,min}$ to the spring to make return of the governor possible. If, on the other hand, the spring is made stiffer, the motion of the governor is quickened, and both excess speed and excess pressure tend to become smaller. It should, however, be understood that during the process of changing from the steady vibration with large amplitude to another steady vibration with smaller amplitude, the three deviations of governor displacement, speed and pressure will fall out of step for a while and will cause quite heavy fluctuations.

The minimum spring scale necessary for stability is easily computed. For sine harmonic vibration with angular velocity w of auxiliary vector, equation (1) becomes

$$\frac{du}{dt} = \frac{1}{2}\frac{A}{A_o}\cos(wt)\frac{M_m}{I},$$

From which

$$Du = \frac{1}{2}\frac{A}{A_o}\sin(wt)\frac{M_m}{Iw},$$

Substitute this in equation (2) then

$$dp = \frac{V'}{V_2}p.\frac{1}{2}\frac{A}{A_o}\frac{M_m}{Iw}\sin(wt)\,dt;$$

p here represents the average value of the pressure in V_2. By integration we obtain

$$Dp = -\frac{V'}{V_2}p.\frac{1}{2}\frac{A}{A_o}\frac{M_m}{Iw^2}\cos(wt),$$

with a maximum value at end of swing of

$$D_o p = \frac{1}{2} \frac{A}{A_o} \frac{V'p}{V_2} \frac{M_m}{I\, w^2},$$

But we also have $$w = \sqrt{\frac{S_1}{m} - \frac{D_o p}{\frac{A}{2}} \frac{G}{m}}.$$

The two expressions for w must be equal, hence

$$\frac{S_1}{m} - \frac{D_o p}{\frac{1}{2}A} \cdot \frac{G}{m} = \frac{1}{2} \frac{A}{A_o} \frac{V'p}{V_2} \frac{M_m}{I} \frac{1}{D_o p}.$$

This equation may be solved for S_1 or for $D_o p$. It is more instructive to do the latter. By multiplication with $\frac{D_o p\, A\, m}{2G}$, we get an equation of the form $(D_o p)^2 - K_1 . D_o p = - K_2$, from which $D_o p = \frac{1}{2}K_1 \pm \sqrt{(\frac{1}{2}K_1)^2 - K_2}$, or, by substitution of the values of the constants,

$$D_o p = \frac{1}{4} \frac{S_1 A}{G} \pm \sqrt{\left(\frac{1}{4} \frac{S_1 A}{G}\right)^2 - \frac{1}{2} \frac{A}{A_o} \frac{V'p}{V_2} \frac{M_m}{I} \frac{A}{2} \frac{m}{G}} \quad \ldots\ldots (3)$$

From this equation it follows that there may be a great or a small pressure rise for a given spring force, as long as the value of the radical is real. If the latter becomes zero, then we work with the minimum spring scale. Hence we have

$$\frac{1}{16} \frac{S_1^2 A^2}{G^2} = \frac{1}{2} \frac{A}{A_o} \frac{V'p}{V_2} \frac{M_m}{I} \frac{A}{2} \cdot \frac{m}{G}$$

or $$S_1 = 2 \sqrt{\frac{V'p}{V_2} \frac{M_m}{I} \frac{m \cdot G}{A_o}} \quad \ldots\ldots\ldots\ldots\ldots (4)$$

By the introduction of time elements, this equation may be simplified. We know that $\frac{M_m}{I} = \frac{u}{T_s}$, also that $\frac{pG}{mA_o} = \frac{2}{(T_g)^2}$, where T_g is governor traversing time under the influence of pressure; $V'uT_f = V_2$, where T_f is the time required to fill the volume of the container. Then we obtain

$$S_1 = \frac{2m}{T_g}\sqrt{\frac{2}{T_s T_f}} \qquad \text{(5)}$$

for the smallest permissible scale of the spring.

From this equation the static pressure fluctuation can be found, because

pressure difference (between extreme positions) × *governor area* = *spring scale* × *governor travel, hence when*

$$p' = \frac{\textit{static pressure difference between extreme positions}}{\textit{average pressure}},$$

$$p' = \frac{S_1 A_o}{G p}.$$

The conclusion from this reasoning is that, for stability of regulation, the governor must have a minimum static pressure fluctuation, just as in speed governing the governor had to be designed for a minimum static speed fluctuation. For many classes of service such a fluctuation is admissible, but occasionally very close pressure regulation is desired; a static fluctuation is not permissible in such a case, and temporary stability must be secured by other means. Among such means are compensating dashpots, similar to those used in speed governing (Chorlton-Whitehead governor, Bee governor, etc., see paragraph 2, Chapter IX), or throttling of the inlet to the governor such as at point (*3*), Fig. 111, or inclosure of the governor in an almost airtight box.

The question of stability of regulation being settled, the problem arises, what must be done to damp the vibrations out of existence? The device used in speed governing, namely damping by an oil gag pot, does not work in this case, as will readily be understood from Fig. 112. The period of a damped vibration is longer than that of an undamped vibration, hence the speed rise is greater, and the excess pressure becomes greater (instead of smaller, as desired); the governor does not return promptly. Another method of arriving at the same conclusion consists in arresting the governor at point (*5*), Fig. 112, until the speed has dropped to normal. During that time the pressure rises considerably. As soon as the governor is released there is no force to return it, and it crawls farther away from the equilibrium position under the influence of accumulated pressure.

While governor displacement damping fails, "speed damping" does the trick. If in Fig. 113, which is a reproduction of part of Fig. 112, the speed should not rise quite as high on the travel to the right, and should not fall quite as low on the travel to the left, then the pressure fluctuation would be reduced, more restoring force would be effective, the governor would quicken in its movements, the pressure fluctuation would become still less, etc., and the vibration would cease. This desirable effect is produced by throttling both in the ports and valves of the prime mover, and in those of the pump. For a given position of the power-controlling mechanism. throttling makes more of the engine or turbine torque available at low speed, and reduces it at high speed. The speed is reduced during the excess speed period, and is increased during the speed deficiency period; it will follow a line somewhat like the dotted spiral of Fig. 113, and the vibrations will rapidly die out.

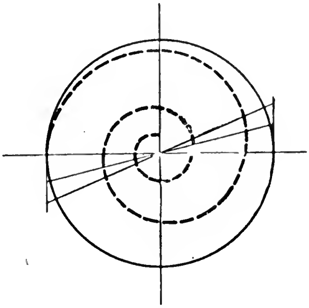


Fig. 113

Summing up, the devices shown in Figs. 108, 109 and 110 may work satisfactorily, and may not. Satisfactory regulation is favored by long starting time T_s (large flywheels), long filling time T_f (large container), and small governor traversing time. Stability of regulation requires static pressure fluctuation or, where such fluctuation is not permissible, temporary stability. Throttling in valves and ports of prime mover and pump damps vibrations out of existence.

Pure pressure governing of the character here described occurs seldom. As a rule, the container is a piping system of small cross-sectional area compared to its length, see dotted lines in Fig. 111. In that case any increase of speed immediately causes an increase of pressure at the mouth of the pump, on account of the friction in the pipe. The pressure does not lag 90° behind the speed. It has a component which is in phase with the speed and a component which is 90° behind the speed. In a vibration diagram with abscissæ representing governor displacement, the component in phase with the speed is represented by an ellipse with the governor displacement as an axis, while the component lagging 90° behind the speed is represented by an inclined straight line, as before. The resultant phase depends upon the relative magnitude of the two components (see Fig. 114, where the inclined ellipse resulting from a combination of the two phases has been shown.) We then face the peculiar situation that the speed component requires speed regulation, and the pressure component requires pure pressure regulation. For the sake of the speed component an adjustable oil gag pot is usually necessary, unless displacement damping can be secured by other means.

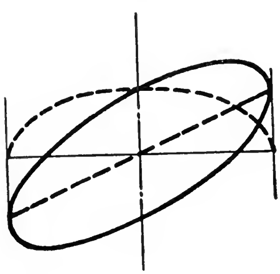


FIG. 114

Friction in pressure governors is just as harmful as friction in speed governors, if close regulation is desired. It produces a "detention by friction" and endless fluctuations of pressure, unless it is eliminated. Just as before, impressed cyclical vibrations of pressure (caused, for instance, by the pulsating delivery of displacement machines), jarring and vibration transmitted mechanically to the governor, and impressed cyclical forces eliminate friction. In the absence of these agents, a frictionless governor must be used; this means knife edges, roller or ball bearings, and floats or bellows (Figs. 108 and 109) in preference to pistons (Fig. 110). If the latter design must be used on account of high pressure, good lubrication and plenty of oil grooves for distributing the pressure uniformly around the piston are a great help; because absence of such grooves, coupled with local deformation due to high pressure, forces the plunger to one side and causes it to stick.

All prime movers regulating for constant pressure must also be equipped with a speed limit centrifugal governor so that excessive demands upon pump or compressor will not wreck the outfit. Such runaway governors may be (and usually are) of very simple design, because they enter into action only occasionally, and because all pretense to close regulation is abandoned when they do enter into action. Of course they must be strong enough to handle the valve gearing.

While pressure governing by the devices shown in Figs. 108, 109, and 110 is successful with proper design when a single prime mover and pump work against the demand for air, water, or gas, the assistance of a centrifugal governor cannot be spared as soon as two or more units work in parallel (deliver to the same container). For proof of this statement study the action of two steam-driven compressors jointly furnishing compressed air to a mine. Let these compressors be of a high class type, with steam cut-off governed by pressure governors, such as shown in Fig. 110. Let one governor have a little more friction than the other. Now let the steam pressure drop gradually. Then the governor with less friction will adjust its engine and will maintain the pressure. The governor with more friction will not move, because the pressure is maintained constant by the other governor; its engine will not receive enough steam, and the compressor will slow down considerably, causing the other machine to speed up an equal amount. As a makeshift, one of the machines is sometimes operated at constant speed by a centrifugal governor, all the regulating being done by the second machine. This will do in the case of two machines, but when three or more run in parallel, the fluctuations are too severe to be taken care of by one pressure governor and compressor only.

In order to make all machines share evenly (or nearly so) in the delivery, a combination centrifugal and pressure governor is frequently used. The principle of such a combination governor is shown in Fig. 115. A centrifugal governor (*1*) of great static fluctuation, such as described in Chapter V, controls one end of a floating lever (*2*). The other end of the latter is controlled by a pressure governor (*4*) of the type shown in Fig. 110. Point (*3*) of the floating lever controls the supply of energy to the prime mover. Variable demand for air (or other fluid pumped) at practically constant pressure means speed of prime mover fluctuating with the demand. Point (*3*) will, in high class reciprocating units, maintain practically the same position for any speed, except that it will move temporarily for the purpose of accelerating or retarding the pump. The result is that, as a first approximation, the floating lever always passes through the same point (*3*), while the two ends move. It is to obtain great speed variation over this range of movement that the centrifugal governor is arranged with excessive speed fluctuation. In principle this governor is identical with the governor described in Chapter V. Naturally a safety device must be used to prevent overspeeding in case of the pressure main or the pipe to the pressure governor breaking. Any of the devices mentioned in Chapter V may be used. For the sake of completeness an additional and different arrangement is illustrated in Fig. 115. The necessary great static fluctuation is obtained by a combination of static properties of the governor proper (*1*) and of a central spring (*7*); the maximum tension of the latter is determined by a weight (*5*), which normally rests on a fixed stop (*6*) and which, in the just mentioned case of overspeeding, is carried by the spring (*7*) acting upon a lever with fulcrum (*8*).

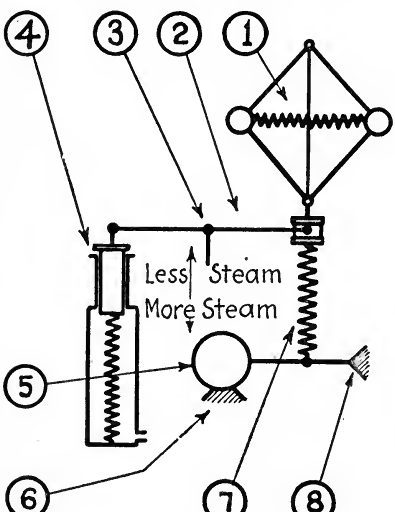


FIG. 115

The combined speed and pressure governor distributes the work evenly, provided that the pressure governor is set with a sufficient static pressure fluctuation. To see that the governor fills the bill, imagine that one compressor has lagged behind the others; the speed governor end of lever (*2*) will drop and point (*3*) will move down and give more steam, unless point (*4*) on the pressure side goes up. But the latter cannot occur, because it would unbalance the pressure governor. Hence more steam is given, and the slow machine is speeded up.

References to Bibliography at end of book: **54, 75, 81.**

CHAPTER XIII

RELAY GOVERNING

1. Reasons for the Use of Relay Governors and Forces Acting upon Them. — In the preceding chapters the governor was studied in its two capacities of measuring instrument and of motor for adjusting the valve gear. In the present chapter, we will investigate the properties of governors the motive power of which need only be large enough to release and control another source of energy which, in turn, adjusts the power-controlling mechanism of the prime mover.

The use of relay governors, as this type is called, dates back to about 1870, but their development remained slow until the electric generator demanded precise regulation of large hydraulic turbines. In that type of prime mover, frictional resistance to motion of power-controlling mechanism is especially great, so that direct-control governors were out of the question. In the development of this type of governor, American engineers played a leading part. Relay governing for hydraulic turbines was so successful that the same method of governing was transferred to steam turbines and to large gas engines. While in steam turbines the resistance to adjustment of steam-controlling mechanisms can be kept comparatively small, the dynamic action of steam flow causes widely varying forces upon the governor in different parts of its travel (compare paragraph 3 of Chapter III), unless the controlling mechanism is designed with great care and knowledge. In the latter case direct-control governors are quite feasible and are indeed used with steam turbines passing up to 30,000 pounds of steam per hour. In larger machines direct-control governors, even with careful design, become so large and expensive that it pays to use relay governors instead. Frequently, relay governors are adopted for smaller sizes also in order to obtain uniformity of design.

Many large gas engines are equipped with relay governors, because the resistance of the gas-controlling valves is uncertain on account of dust and tar in the gas. Tarry matter in the gas will easily increase the resistance of the valves to 50 (or more) times the value of the resistance of the valves with clean gas.

Last, not least, a relay frees the governor (measuring instrument) from the troublesome integrated effect of valve-gear reactions, which effect is so difficult to predetermine correctly.

The very nature of relay governing involves certain differences between it and direct-control governing, and these differences should be clearly understood. The purpose of relay governing is to keep all valve gear forces away from the governor. In consequence, the friction-eliminating influence of vibratory impressed forces is absent. In relay governors there are no cyclical vibrations due to vibratory reactions, and detention by friction (see paragraph 4 of Chapter II) becomes of great importance. The absence of vibration forces has become more complete with the perfection in the production and lubrication of toothed gears used for driving the governors. The old-time jarring of the governor due to rough gears is not found in steam turbines. And finally, turbines have no cyclical variation of angular velocity such as steam and gas engines have; this does away with another friction-reducing agent.

These three facts have necessarily resulted in the use of relay governors with smallest possible internal friction. Knife-edge joints have become universal in the type of governors under discussion. Even small friction in centrifugal governors of this type produces never-ending vibrations. The latter are especially noticeable when the load is constant.

Mr. C. A. Parsons, one of the foremost pioneers in steam turbine design, recognized this situation at a very early date, and overcame governor friction by means of impressing a vibration upon the governor. This was accomplished by an eccentric in connection with an oil gag pot. His system of governing became known as "puff governing," because steam was admitted in puffs. Several turbine builders make use of this principle at the present time; they have, however, reduced the governor vibration to a very small amount, just sufficient to eliminate friction.

A second difference, and an important one, results from the addition of a variable. The governor releases a force, which action takes time. Then the force adjusts the energy-controlling mechanism, which, again, takes time. Now it may be that the sum of these two time elements is less than the time which a direct-control governor would require for the same work, but a new variable has been added nevertheless. Instead of dealing only with the promptness or traversing time of the governor, we now deal with the latter plus the promptness or traversing time of the relay. In future calculations this time will be used. It is defined as that time which is required by the relay mechanism (hydraulic cylinder, electric motor, gearing from prime mover) to move the power-controlling mechanism (valve gear, throttle valve, hydraulic gate) from full-load position to no-load position. The symbol T_r will be used for the relay traversing time, while T_g will be retained for governor traversing time.

What has just been stated with regard to introducing an additional time element applies with equal force to additional detention by friction. The regulating or relay mechanism may have considerable friction which, in some forms of relay governors, is just as harmful as, or even more harmful than friction in the governor proper.

A third, and extremely vital, difference between direct-control governing and relay governing is that the latter enables the designer practically to get rid of the troublesome influence of mass of governor. This is accomplished by methods which use only a very short governor travel and which will be described in detail later on.

A fourth difference lies in the fact that relay governors are easily adapted to obtaining reversed speed curves. This latter property of relay governors was brought out in the governing of hydraulic turbines with long supply lines. In such governors, gate movements must be slow (unless stand-pipes or surge tanks are provided), to protect the line from water hammer. This requirement necessitates a governor of great static fluctuation for the sake of stability of regulation. But great static fluctuation means a great speed difference between no load and full load, which is not permissible in electric lighting. This predicament led to the invention of automatic speed-restoring devices which gradually bring the speed back to normal after every change of load. These devices will be found described later on.

The sources of energy which are released by the governor are either fluid pressure, or else electrical energy, or finally mechanical energy derived from the prime mover. Fluid pressure may be derived from steam, oil, or water under pressure, or from compressed air. Steam and compressed air are elastic, or expansive, so that relay pistons operated by these fluids tend to cause overtravel and, in consequence of that, racing and hunting. Frictional damping will stop this, but introduces speed fluctuations from another source. Water and oil are superior in this respect. Water often carries impurities which wear out the pilot valve. Oil under pressure is doubtless the best fluid for hydraulic relay governors.

Electrical energy has occasionally been used for relay governing of prime movers, but has been given up on account of the greater convenience of steam and pressure oil. It is used to a much greater extent for regulation purposes outside of power plants where the transmission of steam and pressure oil is inconvenient and for regulating prime movers which keep constant some quantity at a considerable distance from the power house.

The use of the mechanical energy of the prime mover is very inviting at first thought, but in practice many difficulties must be overcome. Heavy clutches must be thrown in, held in place, and pulled out. Wear occurs in these clutches, and must be taken care of automatically. So difficult is a satisfactory solution that in modern mechanical relay governors the clutch is operated by a hydraulic cylinder, which latter is controlled by a relay valve. Such a governor may be called a combination hydraulic and mechanical relay governor.

References to Bibliography at end of book: **8, 9, 10, 12, 14, 16, 19, 20, 21, 24, 27, 29, 36, 38, 41, 44, 45, 46, 47, 48, 53, 55, 56, 60, 66, 67, 69, 71, 73, 75, 76, 77, 78, 81, 82, 83.**

2. Relay Mechanisms. — While relay mechanisms necessarily vary with the forces which are brought into action by the governor (measuring instrument), there are certain principles which must be observed in any type of relay governor; and since the hydraulic governor is the one most commonly used, the principles in question will be demonstrated with reference to that type of governor. Application of the same principles to mechanical or electrical relays will not be difficult.

In the following illustrations the governor (measuring instrument) is represented by the symbol *G'r*. The type shown is a centrifugal governor which measures angular velocity. But the operation of the different mechanisms would remain exactly the same if the centrifugal governor were replaced by a constant-volume governor as shown in Chapter XI, or by a constant-pressure governor as shown in Chapter XII.

Figure 116 illustrates a very simple arrangement, which, unfortunately, is even now used by those who have not taken time to study governors. The arrangement is faulty, as will appear from the following reasoning: Consider a governor and prime mover in which the governor and the relay have the right position for equilibrium; then imagine the governor suddenly depressed from equilibrium position, point (*2*), Fig. 117, to point (*1*), and then released. Under the influence of its own stability (due to static fluctuation) and of an increase of speed, see curve (*9*) (*12*,) it will return toward mid-position (*3*). While it does so, the power piston has traveled downward, from point (*6*) to point (*8*). It reaches its greatest displacement when the governor passes through mid-position at point (*3*). During all this time the speed grows. The rate of increase of speed is greatest at point (*12*). Beyond the line (*3*) (*8*) (*12*) the governor travels upward under the influence of the continually growing speed. The latter reaches a maximum in point (*10*), when the power piston has returned to mid-position, point (*7*). The governor is still driven upward

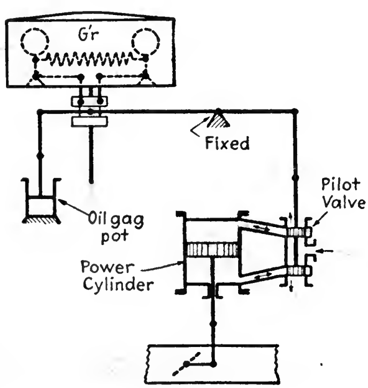


FIG. 116

by a regulating force due to the excess speed. The latter does not come back to normal at point (*11*) until the power piston has traveled upward a considerable distance. But at that time the governor is not only away a much greater distance — (*4*) (*5*) — from the position of equilibrium than it was at the beginning of the disturbance — (*1*) (*2*) — but it also has acquired considerable kinetic energy, which carries it still farther away from correct position. The amplitude of the vibration grows, until the governor strikes a stop. The mechanism is useless.

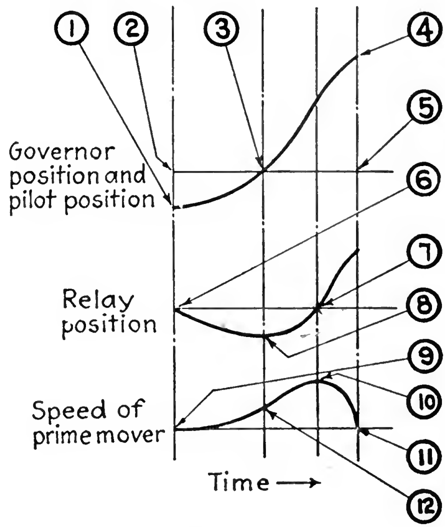


FIG. 117

It is true that the arrangement under discussion can be made to regulate after a fashion by giving the governor great stability and much damping, by making the pilot valve very small and by using a prime mover and torque absorber with decidedly self-regulating properties. Even then the device will work only on condition that the changes of load are very, very gradual. The prime mover works in that case with long, slow vibrations at constant load and with speed changes which are severe, even if the load varies only a small fraction of the total.

Rigid Return of Pilot Valve

The shortcomings of the arrangement shown in Fig. 116 were recognized at an early date. The movement of the pilot valve was "compensated" for by the addition of rod (*5*) with fulcrum (*4*) for lever (*3*), Fig. 118. The latter is now a floating lever. If the governor rises, (*4*) is temporarily a fixed point. The pilot valve is raised, the power piston moves down, turning lever (*3*) about fulcrum (*1*) and returning the freely movable pilot valve to mid- or closed position. The device is known as an "anti-racing" device, or as a "compensating return," or simply as a "return," meaning, of course, a return of the pilot valve to mid-position. It will presently be shown that other quantities are compensated for in relay governors so that the name of "compensator" for the return should be abandoned.

Since the pilot valve must return to mid-position for power equilibrium, there belongs a definite position of the governor to each position of the power piston (see dotted positions of lever (*3*)). And since a given angular velocity corresponds to each position of the governor, it follows that a definite angular velocity of the governor (and of the prime mover) belongs to each position of the power piston. The arrangement is fully equivalent to the direct-control governor, except that the resistance to governor motion is minimized and that the traversing time of the governor is replaced by it plus relay traversing time. Evidently, it is desirable to make the latter as short as possible, but there are limits, because a short traversing time can be obtained only by means of a large pilot valve, and that, in turn, means greater resistance to governor motion.

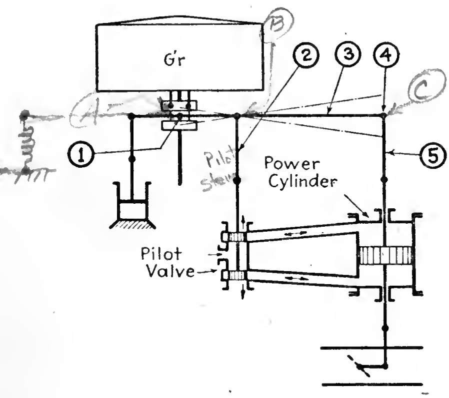


Fig. 118

The arrangement of Fig. 118 is not the only method of returning the pilot valve. The stem (*2*) of the pilot valve may be split and may be provided with right and left hand threads and a turnbuckle for changing the length. Many arrangements for returning pilot valves have been made by inventors, and the patent records are full of return mechanisms in disguise.

The arrangement which has just been described should be called a "relay with rigid return" in contradistinction to those with slowly yielding return which will be described below.

The simple rigid return has been almost universally adopted for steam turbine regulation. In order to get along with a very small governor travel, designers, as a rule, place the pilot valve between the governor and the power cylinder.

Stabilizing Relay with Limited Governor Travel

Figure 119 illustrates a relay mechanism which is commonly ascribed to Dr. Proell (Zeitschrift des Vereines Deutscher Ingenieure, 1884), and which forms a bridge between the mechanisms used for steam turbines and those used in hydraulic turbines. In this mechanism the lever (*3*) turns about a fixed point (*1*). The travel of the lever is limited by two lugs (*4*), and is just sufficient to open wide the ports in the pilot valve. The power piston is joined to the lever (*3*) by a weak spring (*2*), the force of which restores the governor to mid-position after a disturbance. When the governor has again reached mid-position, the power piston has assumed a different position and the length of the spring has been changed. By studying either a decrease of load or else an increase of load one sees readily that for a decrease of load the governor is loaded, which means an increase of speed, and that for an increase of load the governor is unloaded, which means a reduction in speed. The increase or decrease is proportional to the spring force and the latter, in turn, is proportional to the change of load. The regulating mechanism has, in consequence, a "static speed fluctuation," which is not, however, produced by the properties of the governor, but by those of the relay mechanism. If the travel of the governor is made small enough, the stability or instability of the latter is of no consequence. While, with this exception, the action of this relay mechanism is just the

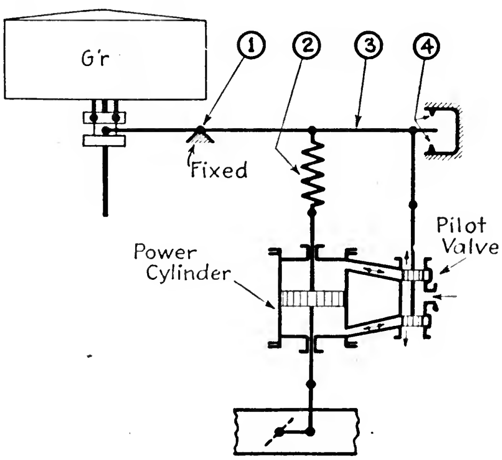


FIG. 119

same as the one of the mechanism shown in Fig. 118, it should be noted that the travel of the governor can be made very small, so that the troublesome effect of governor mass can be practically eliminated.

Relay Mechanism with Slowly Yielding Return

As before mentioned, the governing of hydraulic turbines with long penstocks requires, on one hand, a governor of great static fluctuation (in order to save the water lines), and on the other hand a governor of small static fluctuation in order to maintain a constant voltage or frequency. For the purpose of attaining both of the apparently conflicting requirements, the same principle is used which was described in paragraph 2, Chapter IX, in connection with the Chorlton-Whitehead governor. This principle, applied to a relay mechanism, is illustrated in Fig. 120.[1] It will be remembered that the principle consists in first imparting temporary stability to the governor mechanism, and then gradually, by means of an oil pot, removing the stability. Figure 120 is very similar to Fig. 119. The difference consists in the connection between the power piston and the spring (*2*). In Fig. 119 the connection was rigid, whereas in Fig. 120 an oil gag pot (*5*) is interposed. If the by-pass on the gag pot is closed (and if we disregard the clearance between the gag pot piston and its cylinder), the action of the mechanism of Fig. 120 is identical with that

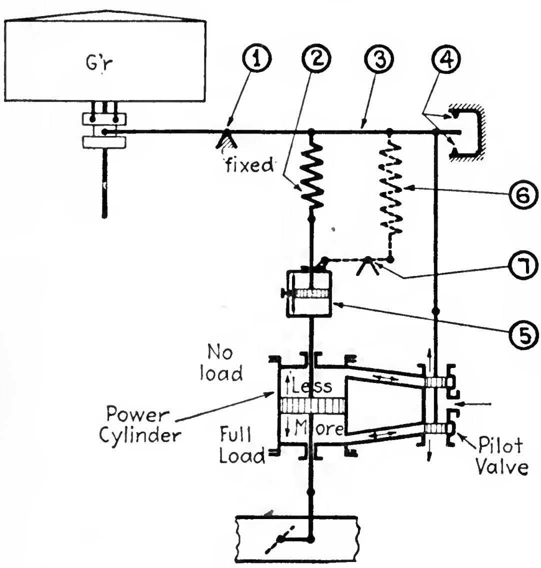


FIG. 120

[1] For the present, parts (*6*) and (*7*) should be disregarded.

of Fig. 119. As a matter of fact, we can best understand the action of the mechanism shown in Fig. 120 if we assume the by-pass closed during a change of load and suppose it to be opened after the regulating process with rigid connection has been completed. After the by-pass has been opened, spring (*2*) can assume its free length, the loading or stability-producing influence of the spring on the governor disappears, and the latter assumes the original speed. The action is just the same as if the tension of a loading spring in any of the devices in Chapter V were changed while the governor is in operation. Centrifugal force and centripetal force are no longer equal, and a regulating process is started. In the case in question the change of speed (or the change of load) becomes more and more gradual as the spring tension disappears.

The result is a perfectly isochronous governor, with stability during the process of regulation. Oil pot (*5*) is commonly called a "compensator," because it compensates for the change of speed which would otherwise be caused by the different position of the power piston. It might be called a "speed restorer," but the term compensator is so generally used that it should be retained. It should be thoroughly understood that illustration 120 is purely diagrammatic. In practice the compensator is usually located some distance away from the power cylinder and is operated by the latter through a system of levers.

Slowly-yielding Return with Adjustable Static Fluctuation

Isochronous regulation, while apparently ideal, is not always desirable. For parallel operation of synchronous (alternating current) generators, a small, positive static fluctuation is necessary for proper distribution of work among the several generators operating in parallel. On the other hand, asynchronous generators require a negative static fluctuation (reversed speed curve). The mechanism of Fig. 120 can be adapted to either one of these requirements by the addition of a second spring (*6*) stretched between governor lever (*3*) and a second lever with fulcrum (*7*). The bottom lever is attached to the gag pot (not to its piston). Then spring (*2*), in conjunction with the gag pot, determines the temporary stability of the governor, whereas the spring (*6*) determines the static fluctuation; for, while spring (*2*) loses its tension after every process of regulation, spring (*6*) does not. Observe that spring (*6*), in the location shown in Fig. 120, is compressed at no load, and is extended at full load. This means that the speed is reduced at no load and is boosted up at full load. The static fluctuation is negative. If, on the other hand, the spring (*6*) were fastened to the bottom lever between fulcrum (*7*) and the gag pot, then the static fluctuation would be positive. Obviously, this arrangement lends itself to an adjustment of the static fluctuation while the prime mover is in operation. To that end the auxiliary spring must be made movable right and left.

Adjustment of static fluctuation may be obtained with other mechanisms, for instance with the one shown in Fig. 121, which illustrates a design with only one spring, (*1*). The latter is fastened to a lever (*4*) with fulcrum (*2*). The extreme end of the lever is operated by the gag pot. Just as before, we split the process of regulation, assuming gag pot and gag piston to be solid during the first part of the process. If the load drops, the governor rises, the pilot valve is lifted, the power piston rises, returns the pilot valve and compresses the spring (*1*). Up to this point the mechanism acts exactly like a rigid return, such as was shown in Fig. 118, because the force of the deformed spring is taken up by the power cylinder and has no action upon the governor (measuring instrument). The speed has risen on account of the static fluctuation of the centrifugal governor. If now the by-pass in the gag pot is opened,

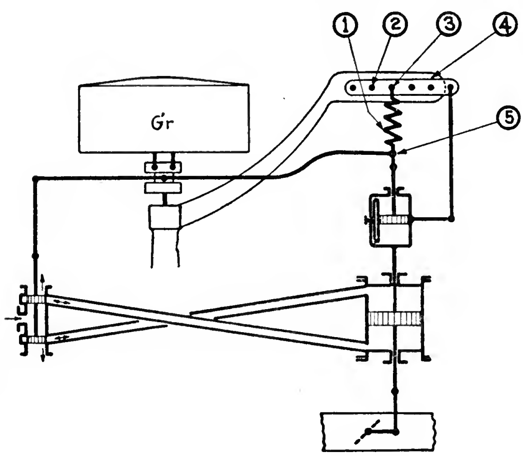


Fig. 121

the spring (*1*) is free to act slowly on point (*5*) of the floating lever. The pilot valve is lifted, the power piston rises, returns the pilot valve, and reduces the supply of energy; the speed drops, the governor drops, raising the pilot valve; the power piston drops, bringing the supply of energy back to the right amount. If point (*3*) were fixed, the spring would be free from stress when point (*5*) had returned to the initial position. But, since point (*3*) has been moved up a small distance on account of lever (*4*) being linked to the gag pot, point (*5*) will not quite return to its original position; it will assume a slightly higher position than before. The pilot valve, of course, is finally in mid-position just as before the disturbance. With the pilot valve again in mid-position and point (*5*) slightly raised, the governor *G'r* must assume a slightly higher position, so that a small positive static fluctuation is left. In this design, the static fluctuation of the governor is used to furnish stability of regulation, whereas relative displacements of points (*3*) and (*5*) determine speed difference which finally establishes itself after a disturbance. For purposes of adjustment, lever (*4*) and the bracket supporting it are provided with numerous holes. The connecting pin can be put into any one of these holes so that the fulcrum can be shifted. With the fulcrum to the left of (*3*), such as at (*2*), the speed curve rises from full load to no load. With the fulcrum to the right of (*3*), the speed curve becomes reversed. With the fulcrum at (*3*), the governor is isochronous.

Brief mention should be made of the fact that isochronism or, if desired, a reversed speed curve, can also be obtained by letting the slowly-yielding gag pot vary the ratio of speeds between the prime mover and the governor.

The Double Relay

For close regulation, the governor traversing time and the relay traversing time must be small. To secure a short traversing time for a powerful relay, the pilot valve (or equivalent releasing mechanism) must be of considerable size. And the governor proper must be of sufficient size to handle the pilot valve quickly. The result is that the commonplace statements about relay governing permitting the use of very small governors are untrue for powerful relays, as many an engineer has found to his sorrow.

This difficulty was overcome by the introduction of a relay for the relay. Of course, each relay must have its return, so that the double relay becomes very complicated and "grasshoppery," if both the relays are returned by levers. External complication is avoided by the floating relay valve shown in Fig. 122. If the small central stem moves, ports in the floating relay valve are uncovered, oil or water pressure is admitted to one end, and the other end is open to exhaust. The ports are so arranged that the floating (outer) valve promptly follows any movement of the central stem. The floating relay valve indeed removes practically all resistance from the governor, if everything is in working order. New complications are introduced, because there are more parts that can stick or bind. The ports of the central stem are so small that scale or lint interferes with the action. Water should never be used with the floating relay valve; nothing but pure oil is satisfactory. In the theory of the double relay, an additional time element enters, namely the traversing time of the floating valve. The floating relay is commonly used on very large steam turbines and hydraulic turbines. In very large water turbines, a double floating relay valve has recently been introduced.

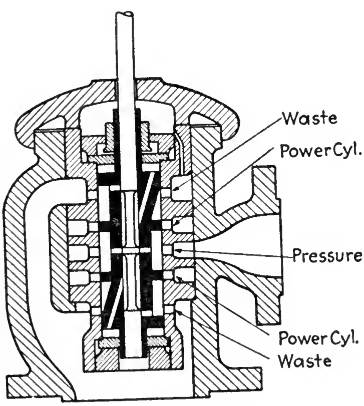


Fig. 122

A very good diagrammatic illustration of all the parts necessary for a complete relay governor was given by Thoma in Zeitschrift des Vereines Deutscher Ingenieure, 1912. This illustration is here reproduced in Fig. 123. The diagram is so clear that very little explanation is needed. The following notes may be helpful. The centrifugal governor (*2*) has no collar or sleeve, but a spherical end bearing stem. The floating lever is held in place by a spring (*1*), so that lost motion is eliminated even if wear takes place. This arrangement is also useful in avoiding strain of the mechanism in case the power-control mechanism reaches a stop before the governor mechanism does. Instead of oil-pot compensation a pair of friction wheels (*3*) is used which, after a change of load has occurred, gradually returns the centrifugal governor to the same position

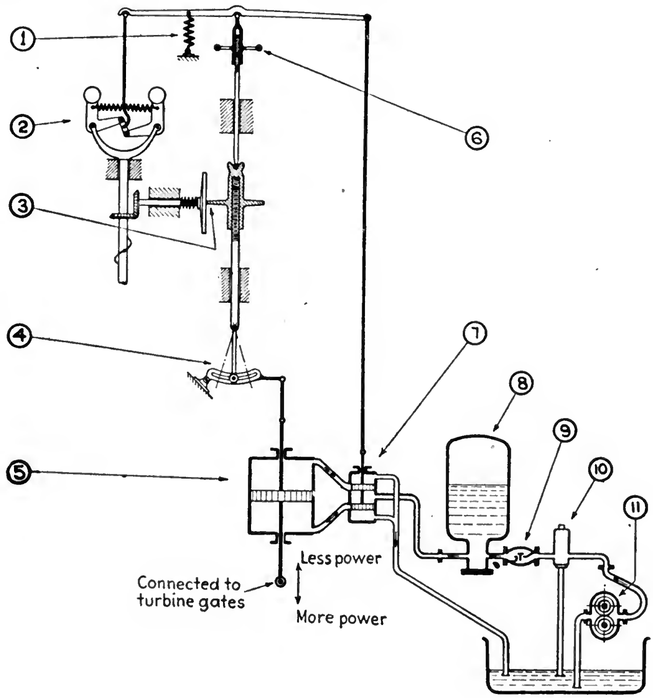


FIG. 123

and restores the original speed. The friction wheel compensator is an American invention and was introduced by the Woodward Governor Co. of Rockford, Ill., long ago. By means of hand wheel (*6*) that position of the governor which maintains equilibrium can be adjusted, which means that the speed is adjusted by making use of the fact that the governor has a static fluctuation. Link (*4*) serves the purpose of adjusting the speed and effectiveness of the return of the pilot valve (*7*).

Oil for the power cylinder (*5*) is furnished by the gear pump (*11*), which delivers oil through the overflow valve (*10*) and the check valve (*9*) to air chamber (*8*). The overflow valve (*10*) is so designed that the pump is relieved of load as soon as the oil in the air chamber has risen to a predetermined highest level.

References to Bibliography at end of book: **8, 9, 12, 21, 24, 36, 38, 41, 44, 45, 46, 47, 48, 53, 55, 56, 66, 67, 69, 71, 73, 75, 77, 81.**

3. Work Capacity of Relay Governors. — The multiplicity of designs of relay governors precludes a simple set of rules for the work capacity of such governors. Only a few guide posts can be given here.

The largest direct-control governor built has a work capacity of about 30 ft. pounds (for 2% speed variation). Relay governors have been built with 20,000 ft. pounds (or even greater) work capacity. Right here an important difference between the two types of governors should be mentioned; namely that in direct-control governing more work may be done by the governor, if we allow a greater speed variation, whereas in a properly designed relay governor the speed change has no influence on the work capacity.

Properly considered, the term "work capacity" is incomplete, because it omits the element of time. This fact becomes manifest if we consider mechanical relay governors in which the governor (measuring instrument) throws in a clutch and makes the whole energy of the prime mover available for moving the energy-controlling device. In such a governor the torque which can be transmitted to the control is limited by the mechanical strength of the mechanism. Shafts, gears, keys, etc., must not break under the strain But work transmitted in unit time equals torque times angular velocity, and total work transmitted equals torque times angle passed over. Hence, the same mechanism, running at a given rotary speed, will transmit more work, if it is given more time. But in the governing of hydraulic turbines a comparatively long time is required whenever the water is supplied through a long pipe. This condition is favorable to the use of a mechanical governor, which explains the fact that this type of governor was developed in connection with hydraulic turbines and is to-day limited to such use.

In hydraulic governors the work capacity is the product of displacement of power cylinder and pressure of fluid. If water or steam is used the pressure is usually limited to that of the available supply, but if oil is used, the pressure is determined solely by questions of leakage and strength; it often equals 200 pounds per square inch. In this definition of work capacity, three features are not included, namely, (1) detention by friction, (2) capacity of oil pump, and (3) size of pilot valve and ports. (1) Packing against high pressure may produce great friction, particularly if cup-leathers are used in the piston. By improper design, 25% to 30% of the total force may be absorbed by friction. (2) The work capacity of an oil-pressure governor in a given period of time is limited by the capacity of the oil pump. If frequent and heavy changes of load occur, a given oil pump can serve a small governor only, and, vice versa, a given governor requires a large pump. But in the design of standard governors, or even of a given governor for a given purpose, the frequency and amount of load changes are unknown. Hence it has become customary to make the displacement per minute of the oil pump equal to 5 to 10 times the displacement of the power cylinder per stroke. The pump operates all the time. The excess passes off through a spring-loaded relief valve and is returned to the pump inlet. In a Swiss design, the pilot valve has negative lap, which means that the valve is open in mid-position. The oil can flow through the open valve. In centrifugal governors, the oil pump is usually a gear pump and is arranged at the bottom of the governor spindle (see Fig. 124). (3) The size of the pilot valves and of the ports in a hydraulic governor has no influence on the total work capacity, but has a very marked influence on the work capacity in unit time and on the relay traversing time. The larger the pilot valves and ports, the shorter the relay traversing time, T_r. It will be proved that, for closest regulation, T_r must be small, which means large pilot valves. But large pilot valves offer considerable resistance to the governor. The operating fluid exerts a dynamic thrust on the edges of the pilot valve, and, besides, the danger from sticking due to one-sided pressure and unequal expansion grows with the size of the pilot valve. For this reason the diameter of the latter never exceeds one-fourth of the diameter of the power piston, and occasionally goes down to one-tenth of that diameter. The diameter of a floating relay valve may go up to one-third of that of the power cylinder, or even more.

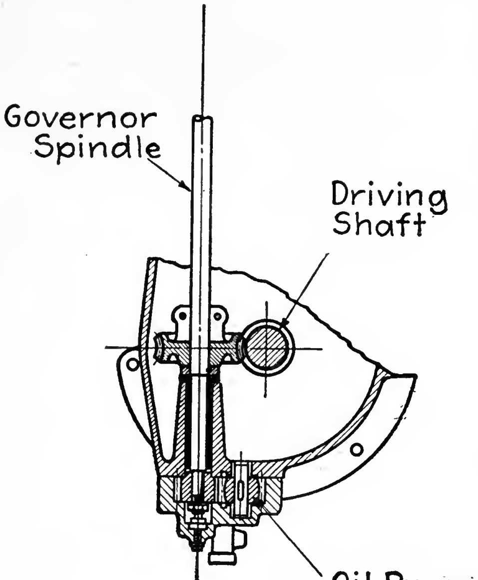


FIG. 124

It should be remembered that the resistance to motion offered by the pilot valve (or clutch, or other force-releasing mechanism) is a load on the governor (measuring instrument), which requires a change of speed, before motion of the pilot valve can begin (see paragraph 5 of Chapter II, on overcoming passive resistance). In relay governing, the pilot valve is part of the governor, and its frictional resistance becomes part of the internal friction of the governor.

References to Bibliography at end of book: **8, 9, 12, 36, 38, 41, 56, 60, 66, 67, 69, 73.**

4. Speed Fluctuations.—A complete theory of relay governing is very complex. The most ambitious attempt at a solution was made by Professor Stodola and was published in the Schweizerische Bauzeitung, 1893 and 1894. Professor Stodola considers the action of the water in long-pipe lines, and even introduces the action of surge tanks with compressed air. He gets a linear differential equation of the 7th order, and, by means of a determinant, derives stability relations. While his work deserves to be better known, it exceeds the limits of this volume.

However, a fair knowledge of the influence of the fundamental quantities on the process of regulation may be gained from the following simplified treatment.

The statement was made in the description of types of relay governors that, with proper design, the utilized travel of the governor could be made very small, so small that the mass of the governor becomes almost negligible. The simplified theory in question neglects the mass of the governor entirely, or, if the governor has mass, assumes that the natural period of

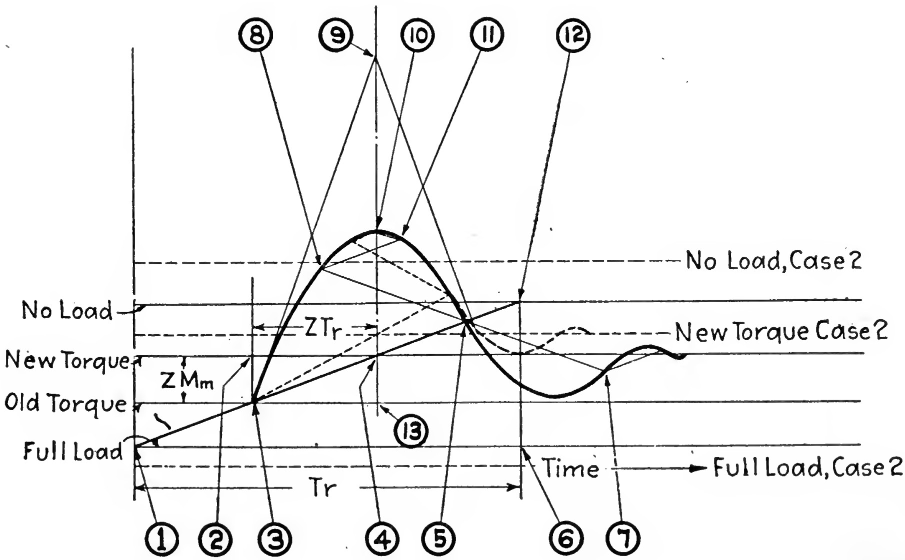


Fig. 125

vibration of the governor is very small compared to the relay traversing time.

The second assumption is that the speed of the relay mechanism or power mechanism is constant. This is very nearly so in mechanical relay governors, and is true in hydraulic governors whenever the pilot valve is wide open.

The usual assumptions about linear relations between speed and governor position, and between governor position and torque are likewise made here.

In Fig. 125, the abscissæ represent time. The ordinates represent two different quantities, namely (1) relay position (which also indicates position of power control), and (2) speed of governor or of prime mover. The speed curve can be used for both governor and prime mover if we let it represent the ratio of actual speed to average speed; for this ratio the letter U has been used. At time *(3)* let the torque suddenly be changed from "old torque" to "new torque." Then the torque *(2)* *(3)* is unbalanced and causes a change of speed, the rate of which is given by the equation

$$\frac{du}{dt} = \frac{M}{I} \text{ or } \frac{du}{u\,dt} = \frac{M}{I\,u} \text{ or } \frac{d\,U}{dt} = \frac{1}{T_s},$$

where U is the relative speed change, and where T_s is the starting time of the prime mover (compare paragraph 2 of Chapter IX). In the illustration, the straight line *(3)* *(9)* represents this rate of change of speed. The latter sets the governor and relay in motion. According to the assumptions of the present elementary theory, the rate of motion of the relay is constant. It traverses the whole range from full load to no load in the "relay traversing time" T_r. Hence the straight line *(1)* *(12)* indicates the rate of change of position of the power-controlling mechanism. Evidently the unbalanced torque decreases on the way from point *(2)* to point *(4)*, and vanishes at the latter place. But unbalanced torque causes change of speed and is proportional to the rate of change. Hence, the curve of speed must have a horizontal tangent vertically above point *(4)*. From mathematics it is known that a curve the tangent of which changes proportionally to the change of abscissæ is a parabola; but that is the case with the speed curve *(3)* *(8)* *(10)*, in question, which may, therefore, be called the parabola of speed deviation. To the right of *(10)*, the speed drops, because the unbalanced torque is negative. For the sake of a lucid graphical representation, the scale for the speed curve and the scale for the change of torque should be so coördinated that the distance *(6)* *(12)* represents the static fluctuation if measured with the speed scale. If the scales are chosen in this manner, the straight line *(1)* *(12)* represents not only position of relay, but also equilibrium speed of governor. The governor passes through an equilibrium position at the intersection of the straight line (*1*) (*12*), with the parabola of speed deviation (see point (*5*)). The governor and relay reverse themselves, and a new branch of the parabola is started. No matter how many branches of the parabola there are, they are all parts of the same parabola, because the latter is determined solely by T_s and T_r, and both of these quantities are constant. Straight line (*5*) (*7*) has negatively the same slope as line (*1*) (*12*). If it be extended to the left to point (*8*) (intersection with parabola), the two parabola sections (*8*) (*11*) (*5*) and (*5*) (*7*) are identical. The same is true for the next branch of the parabola, so that the total number of waves can easily be found by drawing the zigzag line (*3*) (*5*) (*8*) (*11*), etc., in the first parabola.

The speed change (*13*) (*10*) can be expressed by a simple formula. Let Z be the relative load change, or the ratio $\frac{\text{load change}}{\text{maximum load}}$, which means $Z = \frac{(2)\,(3)}{(6)\,(12)}$, then time (*2*) (*4*) $= ZT_r$, and unbalanced torque at beginning of disturbance equals ZM_m; if this latter torque acted during the whole time T_r, the speed change would be $Du = \frac{Z\,M_m}{I} ZT_r$ and the relative speed change would be $U = Z^2 T_r \frac{M_m}{I\,u} = Z^2 \frac{T_r}{T_s}$. But the unbalanced torque drops off, and the speed curve follows parabola (*3*) (*8*) (*10*) instead of straight line (*3*) (*9*). From a well-known property of the parabola, (*13*) (*10*) equals one half of (*13*) (*9*), so that the actual relative speed change is found from

$$U = \frac{Z^2}{2} \frac{T_r}{T_s} \dots\dots\dots\dots\dots\dots (1)$$

If, for instance, the relay time is two seconds and the starting time equals 10 seconds, then for 50 % load change we have

$$U = \frac{(.5)^2}{2} \times \frac{2}{10} = .025, \text{ or } 2\tfrac{1}{2}\,\%.$$

It is interesting to note that the static fluctuation of the governor and relay mechanism does not appear in this equation (1). In this respect, relay governing with constant speed of relay mechanism differs radically from direct-control governing.

Yet it stands to reason that the static fluctuation which, to a certain extent, is a measure of the stability of the system, will have some influence on the process of regulation. In the diagram (Fig. 125) this influence appears as follows: The scales for torque and for speed were so selected that *(6)* *(12)* represents both full load torque, and p, where p is the static fluctuation. Therefore, if p is changed, one of the two scales must be changed, and since a straight line is more easily changed than a parabola, the scale for speed is left unchanged, which means that we must change the scale for torque. In consequence, inclination of line *(1)* *(12)* will vary with the variation of static fluctuation p. If p is very small, then a very small change of speed will move the governor through the whole range of adjustment. In Fig. 125 a regulation diagram for a greater static fluctuation has been sketched in dotted lines. Evidently, the first wave remains just as it was before, but the regulation is completed in less time. For governors working with a constant speed of relay (mechanical governors), a great static fluctuation is advantageous.

The regulation diagram, Fig. 125, is particularly useful for impressing upon engineers the very harmful influence of friction in the governor and of lost motion in the relay mechanism. The effects of governor friction are illustrated in the diagram Fig. 126, which is of the same type as Fig. 125. Since it is rather difficult to see before the mind's eye the connection between the diagram and the governor mechanism, a constant speed relay has been shown in Fig. 127 for the purpose of supplementing Fig. 126. If in Fig. 127 the speed grows, the governor rises, making electric contact with the upper one of two contacts. Wires and control switches are provided for starting an electric motor in the proper direction to reduce the supply of energy. By the movement of the motor the electric contact is broken, unless the governor keeps on going with a speed as great as (or greater than) that of the relay. The moving of the contacts by the motor constitutes the return.

During the process of regulation, the governor will go ahead of the relay, and we will assume here that it can do so by virtue of the contacts being slip contacts or mercury dip contacts.

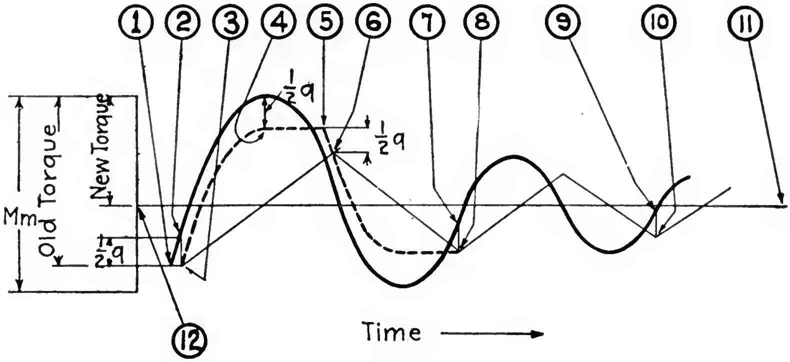


Fig. 126

By making this assumption we make the present study applicable to the hydraulic relay in which the pilot valve can likewise overtravel and can go ahead of the relay. Attention is again called to the assumption that the governor is practically massless, which means that its natural period of vibration is very small compared to the relay traversing time.

If there is friction in the governor, the angular velocity must grow by $Du = \frac{1}{2}\ qu$ (see paragraph 4 of Chapter II) before the governor begins to move. This phase of the regulating process is represented by the straight line (*1*) (*2*), because during the time (*1*) (*3*) the unbalanced torque is constant and equal to the difference (*old torque minus new torque*). From point (*2*) on, the speed curve is parabolic, just as before.

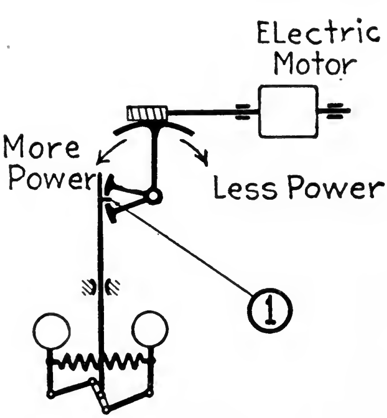


Fig. 127

The governor displacement lags behind the motion of a frictionless governor, as indicated by the difference between the speed curve (which also represents position of frictionless governor) and the dotted parabola *(3)* *(4)*, which represents the actual motion of the governor. At point *(4)* the governor stops; it remains motionless until the speed has dropped $\frac{1}{2}qu$ below the equilibrium speed belonging to the position of point *(4)*, which occurs at point *(5)*. The governor changes contacts as soon as the upward motion of the contacts and the downward motion of the governor intersect, which occurs at point *(6)*. The relay (in this case the electric motor) reverses and continues in its new direction until, in the diagram, the straight line representing its motion drops $\frac{1}{2}q$ below the speed parabola, see points *(7)* and *(8)*. Again, the governor is reversed, and so forth. It is evident that the governor never comes to rest. The tangent points of two successive parabola branches move closer and closer to the line *(12)* *(11)* representing the new torque. As soon as the tangent points fall on this line, the reverse of the relay occurs simultaneously with the inflexion points of the parabola. All following parabola branches are alike.

From the diagram, Fig. 126, it is clear that the remaining speed fluctuation is greater than q, the detention by friction. Distance *(1)* *(2)* = *(9)* *(10)* represents not only $\frac{1}{2}q$ but also that fraction of the total torque which is unbalanced when the parabola intersects line *(12)* *(11)*, but since the whole torque corresponds to a speed deviation p (static fluctuation), the unbalanced torque ratio is $Z = \frac{\frac{1}{2}q}{p}$. From equation (1) we find the corresponding speed deviation to be $U = \frac{1}{2}\left(\frac{\frac{1}{2}q}{p}\right)^2 \frac{T_r}{T_s}$ to either side, which means a total speed fluctuation of U due to friction of

$$U_f = \left(\frac{q}{2p}\right)^2 \frac{T_r}{T_s} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

Let $T_s = 8$ sec., $T_r = 1.5$ sec., $p = 2\%$, $q = 1\%$, then $U_f = \left(\frac{1}{4}\right)^2 \frac{1.5}{8} = 1.17\%$; for $q = 2\%$, U_f would be almost $4\frac{3}{4}\%$. The influence of friction grows with the square in this type of regulation.

The harmful effect of friction is very much reduced, if the governor is prevented from going ahead of the relay. This favorable condition can be obtained in Fig. 127 by letting contact tongue (*1*) fit between the relay contacts so that it abuts and presses against one or the other.

The same harmful effect which is caused by friction in the governor is also produced by several other resistances. Evidently, any force which must be exerted by the governor to release the action of the relay mechanism requires a speed difference of the sort indicated in Fig. 126. Such a force must be exerted, if the pilot valve in hydraulic relay governors moves hard (for instance by being forced to one side through imperfections in workmanship). Such a force must also be

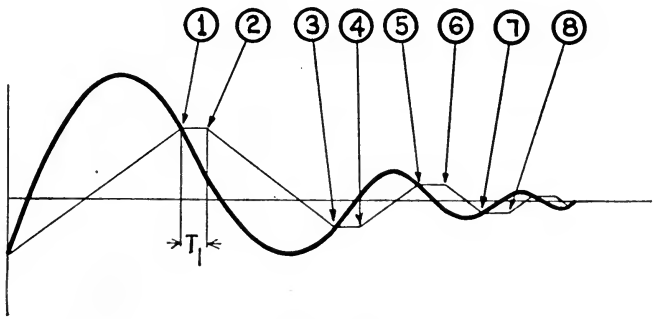


Fig. 128

exerted in mechanical relays to keep the clutch engaged, etc. A very similar, although not strictly identical, effect is produced in hydraulic relays by lap of the pilot valve and by excessive clearance of the pilot valve in its bushing. Lost motion in the joints of the relay mechanism has an effect which is similar in one respect and different in another. Inasmuch as the governor has to travel a certain distance to take up lost motion, the problem resembles that covered by Fig. 126; but inasmuch as time is lost in the return by the power part of the relay, the problem differs. The effect of time lag in the mechanism is commonly explained by the method shown in Fig. 128. The diagram is so similar to Fig. 126 that no explanation is needed except this: At point (*1*) the governor does not reverse the relay; the reversal occurs after the lost time T_1, that is to say at point (*2*). The same action occurs at points (*3*) and (*4*), (*5*) and (*6*), (*7*) and (*8*), etc. The process is drawn out longer than it would be without time loss, but it finally ends without lasting vibration. For practical application it should be remembered, however, that lost motion involves the features of both Fig. 126 and Fig. 128, so that lost motion, as a rule, results in never-ending speed fluctuations of small amplitude.

In many relay governors, particularly in those of the hydraulic type, the velocity of "opening" and "closing" is not constant, but grows with the relative displacement of governor and piston in power cylinder. The regulation diagram then looks somewhat like Fig. 129. The slope of the power-piston velocity curve varies with the lengths of the intercepts between the two curves. For those who are mathematically inclined, it may be remarked that the assumption of proportionality between slope and intercept leads to a very interesting theory of relay governing. It will be explained in the author's book on advanced theory and practice of prime-mover governing. For the purposes of the present rather elementary theory we replace the curve (*1*) (*3*) (*5*) of Fig. 129 by the broken line (*1*) (*2*) (*3*) (*4*) (*5*). The speed curve then is resolved into a succession of straight lines, such as (*1*) (*6*) and (*7*) (*5*), and parabolas, such as (*6*) (*7*). The diagram is identical with Fig. 128, and the greatest speed fluctuation is easily found by graphical construction.

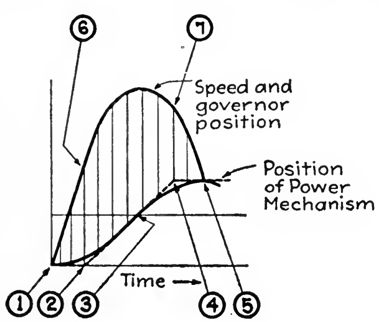


Fig. 129

References to Bibliography at end of book: **8, 9, 10, 12, 16, 20, 21, 36, 38, 41, 47, 56, 66, 67, 69, 73, 78, 82, 83.**

CHAPTER XIV

GOVERNOR TROUBLES AND THEIR REMEDIES

While the principles which underlie all cures for governor troubles are clearly indicated in the foregoing chapters, the principal troubles, their causes and remedies, are dealt with here for the benefit of the operating engineer.

1. Regulation is not Close Enough. — There is too much speed difference (pressure difference, rate of flow difference) between full load and no load.

Several methods are open to make regulation closer. A smaller portion of the travel of the governor may be utilized for the whole range from full load to no load, or the governor proper may be altered. If it is of the spring-loaded type, regulation is made closer by using a softer (more flexible) spring. In a helical spring use more coils, or thinner wire, or grind off the outer layers of the spring (reduce its diameter). In leaf springs grind some off the width of the spring. If the spring acts on a lever, reduce the length of the latter and tighten spring.

All of these changes reduce the static fluctuation of the governor. The latter (for a given prime mover and governor) must not be reduced beyond a certain limit which is set partly by the design of the governor and partly by that of the prime mover (see paragraph 2 of Chapter IX). If further reduction of the static fluctuation is desired, the flywheel effect must be increased, or a prompter governor must be used, or else tangential inertia must be used, or finally a compensating oil pot may be applied. For all of these, see paragraph 2 of Chapter IX.

2. Racing.— Two broad classes must be distinguished :

(a) Racing which appears from the very beginning of the operation, that is, with a new machine ;

(b) Racing which appears after the machine has been in operation several months or years.

Racing which appears after the machine has been in operation several months or years

Discussing this latter case first, we find that racing may gradually develop from several causes:

First, the oil may leak out of the gag pot, or may be pumped out of it by quick motion of the governor. Refill oil pot and make such changes in its design that emptying during regular operation becomes impossible (see paragraph 2 of Chapter IX). The adjustable passage for oil may be too large, or the oil may be too light. In governors with compensating oil pot (paragraph 2 of Chapter IX, paragraph 2 of Chapter XIII) too free an opening of the oil pot will likewise cause racing. The same result is produced by air entering the gag pot with the oil.

A *second* reason for racing (or running away akin to racing) results from such changes in the transmitting linkage between governor and power-control mechanism [1] as do not sufficiently shut off the supply of energy near or at the no-load position. Racing due to this cause occurs only if the load is very light. Obviously, proper change in the linkage will cure this condition.

A *third* reason for racing (and a very prolific one) is friction, either in the governor or in the valve gear (paragraph 4 of Chapter II, 5 of Chapter II, 3 of Chapter XIII). To ascertain whether or not friction is responsible for the trouble, jiggle or pump the governor up and down by hand, taking great care that the upward and downward thrust exerted on the governor are alike. This eliminates friction (paragraph 4 of Chapter II, 2 of Chapter VIII, 4 of Chapter IX) and stops racing, if the latter be due to friction in the governor or power-control mechanism (in the case of direct-control governing) or due to friction in the governor proper, in the case of relay governing.

Pumping the governor will not always eliminate frictional speed fluctuation (or pressure fluctuation, or delivery fluctuation in other forms of governors), in the case of relay governors, because there may be friction in the relay or in the power-con-

[1] Misadjustments of this sort frequently occur in the wake of valve adjustments and repairs.

trol mechanism. To discover such friction, try to move the latter mechanism by hand or by a lever in the direction of least resistance (care being taken not to overstrain the mechanism). If doing so stops the speed fluctuation, the frictional resistance is too great for the available relay force, or the available force has become less (see also under leakage). In this case the frictional resisting force should be measured. The remedy consists, of course, in removing the undue friction. The causes for a gradual or for a sudden rise of friction are so manifold that they cannot all be enumerated here. Among them are: gumming of valves or valve stems due to improper oil; sticking of valve stems due to solid matter in steam; lack of lubrication; deposit of tar (in regulating valves of gas engines); wearing of knife edges; binding due to lack of alignment; and others. It occasionally requires very patient study to hunt down and locate the spot where the undue friction occurs.

A *fourth* reason is leakage in the pilot valve in hydraulic relay governors. Placing of pressure gauges at both ends of the power cylinder is a good means for judging whether or not leakage is responsible for the trouble. If it is, a new pilot valve and bushing are needed. In oil-pressure governors the oil pump may be worn out or deranged.

A *fifth* reason is lost motion in the relay mechanism (paragraph 4 of Chapter XIII). A weak spring or a weight keeping the parts always in contact in the same direction will eliminate the influence of lost motion (see (*1*), Fig. 123), and will, incidentally, indicate whether or not lost motion is responsible for the racing.

A *sixth* reason is the dulling of knife edges in the releasing gear of steam engines. At the right cut-off the dulled edges slip so that alternately too much and not enough steam is admitted. The remedy is plain.

A *seventh* reason is found likewise in releasing gears and more particularly in those which depend upon a partial vacuum in a dashpot for closing the valve. If the dashpots are worn or improperly adjusted, anything from scarcely perceptible racing up to disastrous running away of the engine may occur.

To tell whether racing or overspeeding is caused by dashpot trouble, tie a very long, flexible spring to the dashpot arms so that the spring assists the dashpot in closing the valve. Make the force of the spring in pounds approximately equal to 5 to 7 times the area of the vacuum pot in square inches. If the irregular speed was due to vacuum pot trouble, the spring will cure it. Of course, the spring looks like an afterthought, so that the vacuum pot should be renewed.

An *eighth* reason is frequently found when the governor is belt driven. It consists in a belt which slips at intervals. Application of a belt tightener will soon tell whether or not the belt is responsible for the trouble.

A *ninth* reason appears whenever springs are tightened or weights are added for the purpose of increasing the speed of the prime mover, unless the governor is so designed that the speed can be increased without variation of stability (see Chapter V). Increase the stiffness of the adjusting spring (by using fewer coils) or increase its lever arm, or (in the case of shifting a weight) put a spring into the oil pot. Every one of these means restores the stability (and static fluctuation) which was lost by the previous adjustment.

Racing which appears immediately after the installation of the prime mover

Any of the reasons which produce racing in an old prime mover will, of course, have the same action in a new prime mover. But there exist other causes which will produce fluctuations at once, and which, once removed, will not return. They will now be enumerated.

In order to tell the source of the trouble, observe whether the racing occurs at all loads or whether it occurs only at certain loads (for instance, near the no-load position).

If the racing occurs at all loads, and if none of the causes before given exist, then the static fluctuation is too small (paragraph 2 of Chapter IX), or there is not enough damping (by friction or oil gag pot). If permissible, the static fluctuation should be increased, but this measure, of course, increases the speed (respectively pressure, or delivery) difference between full load and no load. If permissible, the oil pot should be made more effective, either by adjustment, or by a larger pot, if close adjustment overheats the oil. Again, more effective damping by oil pot increases speed fluctuations whenever the load changes. In that case racing can only be cured by more inertia in the prime mover (heavier flywheel) or by a prompter governor (smaller traversing time, Chapter IV, paragraph 2 of Chapter IX). If temporary speed fluctuation after a change of load is permissible, racing can be cured by a compensating dashpot, or similar arrangement (see paragraph 2 of Chapter IX — Bee governor, Chorlton-Whitehead governor, Armstrong governor, also figures 120, 121, 123). This type of governor and oil pot deserves to be more widely used than it now is. Use of tangential inertia in governors also cures racing, but can be recommended for shaft governors only (see paragraph 3 of Chapter IX) ; and even there it is sometimes of doubtful value.

In relay governors racing is caused by absence of a return in the governor mechanism (see paragraph 2 of Chapter XIII). Study of the governor will immediately tell whether a return (compensator, anti-racing device) has been provided. The pilot valve, or other relay, must be returned to make the governor satisfactory.

If racing or speed fluctuations occur in one certain position of the governor and valve gear, one of two causes is present (in the case of speed governing). Either the power-control mechanism does not vary the supply of energy uniformly with the travel of the governor, reducing stability to an unpermissibly small amount in one spot (paragraph 2 of Chapter IX) or else the power-control mechanism reacts in certain positions upon the governor in such a way as to reduce the stability or to remove it altogether (see paragraph 3 of Chapter III).

The first case is very common in governors operating a throttle, either for steam in steam engines or turbines, or for air and gas in internal combustion engines. When a throttle is just cracked, a small increase of area changes the flow through the valve very much more than when it is almost wide open (see Fig. 130). The curve in this illustration shows the general trend only, because the curves differ a little for wet steam, dry steam, air, and for different gases. However, the curve is given solely for the purpose of pointing out that there must be a variable-rate linkage or a cam interposed between governor and throttle if there is to be constant stability over the whole range from full load to no load, unless the governor is designed with variable stability in such a way as just to counteract the action of the throttle. The variability of the rate of flow through a throttle in different stages of opening is one of the most fruitful sources from which poor governing springs. And the source seemingly never dries up.

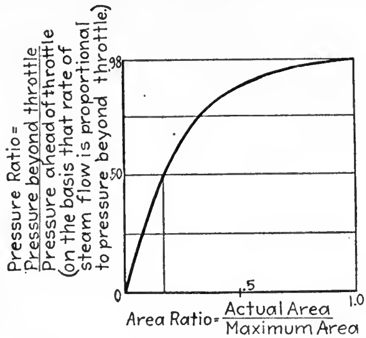


FIG. 130

The second case, namely reaction by the valve gear in certain positions (paragraph 3 of Chapter III) should really not exist, because it is usually due to lack of foresight or lack of knowledge on the part of the designer. If it does exist either near the full-load or near the no-load position, it can be cured by an auxiliary spring which enters into action at one end of the governor travel only. Such a spring is diagrammatically shown in Fig. 131, where it enters into action near the no-load position. The spring can, of course, be so placed as to enter into action at the full-load position. It can also be placed out of sight in the oil gag pot. To do so is a favorite trick of skilled erecting engineers, because it gets away from the criticism that the spring is an afterthought.

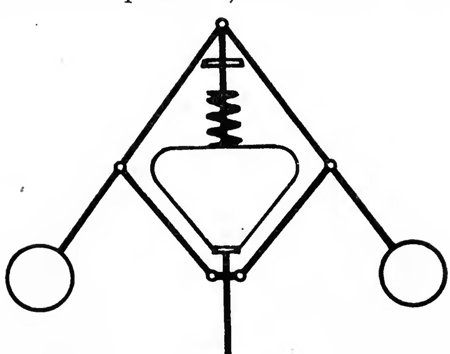


FIG. 131

Racing at light loads occurs in most centrifugal pumps and centrifugal blowers. The surest remedy is to increase the load, for instance by opening a by-pass from discharge to suction. Properly designed governors for turbo-pumps and turbo-blowers automatically take care of this feature (Chapters XI and XII).

3. Speed Fluctuation is too great, when load changes suddenly. — This may be due to any one of many causes, the principal ones of which are given below.

First, there is too much energy stored up in prime mover beyond control of the governor. No adjustment in existence will improve this condition. It means a rebuilding of the prime mover, or an additional power-control mechanism, for instance on the exhaust, and so designed that it enters into action temporarily for great and sudden changes of load. This condition is almost invariably a sign of poor and thoughtless design (see paragraph 5 of Chapter IX). Final judgment as to whether or not stored up energy beyond control of the governor is responsible for excessive speed fluctuation requires study of the engine or turbine and calculation of the volumes in which steam under pressure, or explosive mixture, etc., is stored.

Second, there is not enough flywheel effect in the prime mover. The decision whether or not defective regulation is due to this cause is usually quite difficult, because addition of flywheel effect always improves speed regulation. The best method for arriving at a decision is to figure the necessary *mass* × *radius squared,* from equation (8) of paragraph 5, Chapter IX, great care being taken really to include in the equation every bit of the energy beyond control of the governor. If the latter energy is very small, the problem can be solved by the use of a very prompt governor (Chapter IV, paragraph 2 of Chapter IX), except in the case of hydraulic turbines with long pipe lines.

Third, the oil pot is adjusted too tight. This can readily be investigated by changing the adjustment of the gag pot. The remedy is obvious, but loosening the oil pot may bring never-ending vibrations (paragraph 2 of Chapter IX), in which case the trouble is traced back to the cases of either insufficient flywheel, or too small static fluctuation, or too great a traversing time.

Fourth, the traversing time of the governor is too great (governor is not prompt enough). This condition is, likewise, best detected by calculation. The governor must be replaced by a higher grade governor (Chapter IV) with high rotative speed, large orbit of weights and spring loading.

Fifth, the governor is too small and takes too much time to move the power-control mechanism to its new position. Insufficient size or capacity of the governor has either one or both of two effects. First, the governor cannot overcome the frictional resistance of the power-control mechanism (paragraph 5 of Chapter II), and second, the governor cannot move the mass of that mechanism fast enough (Chapter IV). In either case the governor must be replaced by a larger one, or else the resisting force and mass must be reduced or a relay governor must be installed.

Sixth, the static fluctuation is too small. This reduces the period of vibration and either requires much damping, or else causes never-ending speed fluctuations, unless tangential inertia is used. See paragraph 2 of Chapter IX. Static fluctuation can be measured by the following method: Operate the engine or turbine with absolutely constant load conditions at no load, 50 % of maximum load and 95 % to 98 % of maximum load (the latter should be 100 %, but it is better to stay below the maximum to avoid getting beyond the range of the governor), with the oil gag pot set so tight that all vibrations are eliminated. Measure the speeds and determine from them the static fluctuation (paragraph 1 of Chapter III). From equation (5), paragraph 2 of Chapter IX, find whether this value is too small or not. The static fluctuation must be increased, either permanently by a change in the governor, or temporarily by a compensating oil pot (paragraph 2 of Chapter IX, and paragraph 2 of Chapter XIII).

4. The Governor Vibrates — (Jerks, Dances). — In engines, the flywheel may be too light so that cyclical speed fluctuation keeps the governor alive, but in the great majority of cases the governor is too light for the vibratory forces impressed upon it by the valve gear. The valve gear can, in a few cases, be so altered that the reacting force is kept away from the governor. In the majority of cases the resistibility (Chapter VIII) must be increased. This is done by adding a friction brake, or by making the gag pot more effective. If either one of these two remedies results in too great a speed variation with sudden changes of load, a heavier and more massive governor is needed.

5. Joints or Knife Edges in the Governor Wear too Fast. — The trouble is due to one of two causes: Either the governor is too small, or it is not adapted to the type of valve gear which it has to handle. Knife edges and joints with small diameter pins are very good in governors which are subjected either to no shaking (vibratory) forces, or to vibratory forces which are so small in comparison to the size of the governor that very little, if any, cyclical vibration results. Particularly objectionable are knife edges which are vibrated while heavily loaded.

The trouble can be remedied in several ways. A new governor may be installed transmitting forces directly from centrifugal weights to spring, without joints. Or a larger governor may be installed so that the vibratory force becomes small in comparison with the size of the knife edges, or else the governor may be redesigned and may be provided with large pins with light unit pressure. Friction is not harmful in such a case, because the vibratory force eliminates it (paragraph 4 of Chapter IX) as long as the impressed force is great enough to maintain cyclical vibration.

6. Machine Design Troubles. — Governors are subject to a number of troubles which are in no manner attributable to their governor-features, but are solely machine design or workmanship difficulties. Among them are: heating and wear of collar in spindle governors; wobbling of governor due to unbalanced centrifugal masses; buckling of compression springs; stripping of governor gears due to non-uniform rotational speed of driving shaft; wear of clutches in mechanical relay governors; vibration of water pipes in hydraulic governors due to water hammer; and many more.

These difficulties are so bound up with general machine design and shop practice that their discussion would by far exceed the limits of the present volume.

CHAPTER XV

THE SELF-REGULATING FEATURES OF PRIME MOVERS AND OF MACHINERY OPERATED BY THEM

In Chapter I mention was made of the superfluity of governors under certain working conditions of engines or turbines. If, for instance, the resisting torque grows either directly with the speed, line (*1*) (*2*) of Fig. 132, or with a higher power of the speed, curve (*1*) (*3*), and if the torque is not subject to any variables independent of the speed, then a governor is rather superfluous, because the combination of prime mover and resistance is perfectly self-governing.

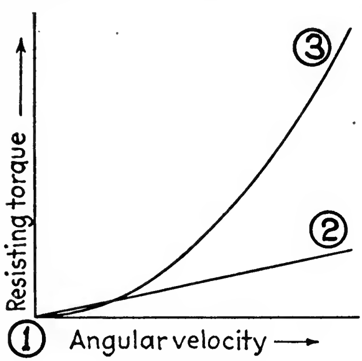


Fig. 132

Cases of this description need not be considered here; but even where governors are needed, similar self-regulating features are frequently met with. We will now investigate to what extent such features may be depended upon to assist close governing.

The problem comprises two subdivisions, namely:

(1) the self-regulating features of prime movers proper,

(2) the self-regulating features of their load.

In studying the prime mover proper, we recognize that friction of the working fluid furnishes the principal means of self-regulation. In a steam engine, the average torque per revolution, for a given position of the power-control mechanism, may be expressed by the formula $M = M_o - ku^2$, where M_o is the torque which would be obtained if the working fluid were devoid of friction, and where k expresses frictional resistance per unit of angular velocity. k depends upon the relative size of the steam pipe, its length from the boiler to the engine, the size of the steam and exhaust ports, conditions of exhaust pipe, capacity of condenser, etc. In engines with throttle control, k is very much greater than in engines with cut-off control.

A quantitative estimate of the regulating properties of throttling may be gained from the following simple calculation: The influence of the throttle is greatest if the down-stream pressure (throttle pressure) is less than 53% of the up-stream pressure. In that case the weight flowing is proportional to the product of initial steam pressure multiplied by throttle area. As long as these two quantities are constant, — and they are constant for a given position of the power-control mechanism — the rate of flow remains constant. But the weight flowing in unit time is proportional to cylinder displacement in unit time multiplied by density of steam after passing the throttle, and the density is roughly proportional to the mean effective pressure. Also, we have the relation that cylinder displacement in unit time is proportional to angular velocity. Taking all these facts together, we find that roughly $m.e.p. \times u$ = constant, or in words, mean effective pressure times angular velocity is constant. Let, for instance, the angular velocity grow by one per cent, then the mean effective pressure, and with it the torque, drops one per cent. And vice versa, if the torque drops one per cent, the speed grows one per cent.

Now compare this to regulation by a governor. If the speed grows one per cent, the torque drops from 25% to 50%, depending upon the design of the governor. It is evident that the self-regulating properties of steam engines cannot be counted upon whenever close regulation is essential.

It must be admitted that in a few exceptional cases the influence of throttling is greater than that which is apparent from the above calculation, particularly in noncondensing engines with expansion below the back pressure. But these exceptions are so few that their scarcity confirms the rule.

A study of the self-regulating properties of steam turbines and hydraulic turbines yields exactly the same result which was found from the above simple calculation (namely that for one per cent growth of speed, the torque drops one per cent) although the mechanics of self-regulation of turbines is very different from that of the corresponding property of engines. From the theory of turbines it is known that, for a given throttle opening, the torque M exerted at angular velocity u is given by the equation

$$M = 2\,M_o\left(1 - \frac{u}{2u_o}\right)$$

where M_o is the torque developed at speed u_o, and where both M_o and u_o refer to the "best" speed of the turbine, that is to say, to the speed at which it develops its greatest power. By differentiation we obtain $dM = -\frac{M_o}{u_o}du$; or $\frac{dM}{M_o} = -\frac{du}{u_o}$ which, in plain English, means that for one per cent drop of angular velocity, the torque grows one per cent. We can, therefore, only repeat the statement that the self-regulating tendencies cannot and must not be depended upon for close regulation.

In internal combustion engines conditions are even less favorable to self-regulation. While throttling of gases through the ports exerts influence similar to that described before under steam engines, the effect of ignition may be either favorable or unfavorable to the point under discussion, as will presently be understood. In Fig. 133 let (*1*) be a normal indicator card. If the ignition be advanced (so that the spark occurs earlier), card, (*2*) will result. If the spark, on the contrary, be retarded, card (*3*) will be obtained. If an engine operates with a very early spark, even a slight slowing down will cause the combustion to be completed in a smaller fraction of the stroke of the engine; a condition of premature ignition is approached, the area of the indicator card (the work per stroke) falls off, the engine slows down still more,

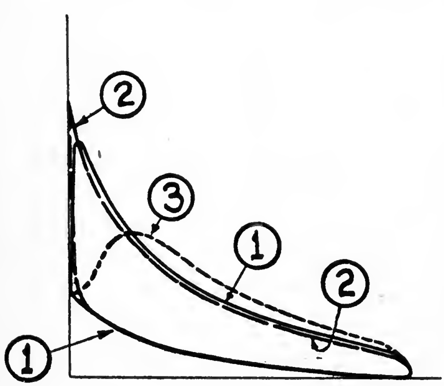


Fig. 133

and so forth with cumulative effect. In that case the engine has absolutely no self-regulating features; it is positively unstable, and may even thwart the influence of the governor, unless the latter be very quick in its action. In the opposite case of a late spark there is greater stability, but the fuel consumption is so high as to preclude the operation of engines with that adjustment, except very occasionally under exceptional circumstances.

Summarizing, we find that the self-regulating features of prime movers cannot be depended upon as a help to close regulation, but that they are of assistance, if crude or coarse regulation only is aimed at.

We find a very similar situation if we investigate the second part of the problem, namely the governing influence of the load or resistance against which the prime mover works. No matter whether a prime mover drives machinery directly, or whether it does so by means of electrical transmission of power, there is almost invariably an increase of torque required if the speed grows. This condition is frequently due to increased air resistance, and frequently to other causes. Consider, for example, an electric lighting system, a diagram of which is shown in Fig. 134.

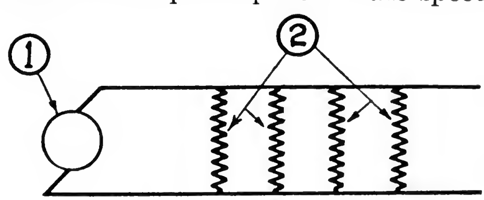


FIG. 134

(*1*) is the generator, and (*2*) are the lights. Unless counteracted by a voltage regulator, an increase of speed produces a corresponding increase of electromotive force, and the latter results in a proportional increase of current. Hence, a one per cent growth of speed produces 2 % increase in power; but, since power is speed times torque, the 2 % increase of power is evenly divided between the assumed 1 % rise of speed and the necessarily resulting 1 % increase of torque.

It should be noted that this increase of resisting torque occurs only on condition that the external resistance of the circuit remains constant. Switching lights on and off, of course, varies the torque independently.

In practice, very few cases with independently variable external resistance are encountered in which for a given condition of such resistance (whether electrical, hydraulic or mechanical) the resisting torque grows faster than the square of the speed, which means that for one per cent increase of speed the resisting torque practically never grows more than two per cent.

Couple with this result the conclusion reached before, namely that for one per cent increase of speed the driving torque never drops more than one per cent, and we have the final conclusion that one per cent growth of speed never produces self-regulating features amounting to more than three per cent of the existing torque. This conclusion allows nothing but a reiteration of the statement previously made, that the self-regulating properties of prime movers and of their loads cannot be counted upon, if close regulation is desired, but that they are of considerable assistance if coarse regulation is permissible.

Reference to Bibliography at end of book: **28.**

APPENDIX

ELEMENTARY DERIVATION OF EQUATION (3) OF PARAGRAPH 1, CHAPTER II

Let C be the centrifugal force caused by angular velocity u, mass m, and by radius r, and let C_1 be the centrifugal force caused by angular velocity u'; m and r remaining the same as before; then

$$C = m\,r\,u^2$$

$$C_1 = m\,r\,(u')^2$$

$$C_1 - C = m\,r\,(u'^2 - u^2) = m\,r\,(u' - u)\,(u' + u)$$

$$C_1 - C = m\,r\left(\frac{u' + u}{2}\right)^2 \cdot \frac{2(u' - u)}{\frac{1}{2}(u' + u)}$$

but $\dfrac{u' + u}{2} = u_a =$ average between u' and u,

hence $$C_1 - C = C_a\frac{2(u' - u)}{u_a},$$

where $C_a =$ average value of centrifugal force.

Let $C_1 - C = DC$, and $u_1 - u = Du$,

then $$DC = \frac{2\,Du}{u_a}\,C_a;$$

but for small changes of u, C_a and u_a differ very little from C and u, so that approximately

$$DC = 2\,C\,\frac{Du}{u}.$$

DETENTION BY FRICTION

The detention which will be calculated is that of friction due to centrifugal forces and weight or spring forces; but it does not include that friction due to compound centrifugal forces (Coriolis' force). It is, therefore, only correct for *slow* changes of position of the governor.

The force acting on a pin causes a moment, due to friction, which may be expressed thus:

$$M = Q_1\,f\,r,$$

where Q_1 is the force on the pin bearing, f is the coefficient of friction, and r is the pin radius.

In any particular linkage, for instance that shown in Fig. 135, the moments produced by friction on the pins (*1*) and (*2*) may be combined into the moment of a single force, as Q, in any direction, having a moment arm L, which is the perpendicular dropped from (*3*), the instantaneous center of rotation of the two pins.

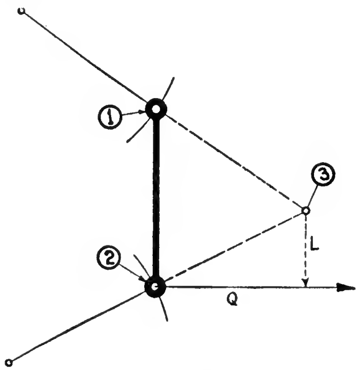


FIG. 135

$$Q\,L = f\,Q_1\,r_1 + f\,Q_2\,r_2 = M$$

$$Q = \frac{M}{L} = f\,\frac{Q_1\,r_1 + Q_2\,r_2}{L}.$$

The force Q may have its point of application transferred along its line of action, and may then be combined with other forces, until all are combined into a single force.

In the Watt type of governor, shown in Fig. 136, for pins (*1*) and (*2*)

$$Q_a\,L_a = M_1 = f\,Q_1\,r_1 + f\,Q_2\,r_2$$

taking Q_a, in this case, in the line of direction of the next link, to which (*1*) (*2*) is connected.

Q_a is the force in the direction (*2*) (*3*).

$$Q_a = \frac{M_1}{L_a} = f\,\frac{Q_1\,r_1 + Q_2\,r_2}{L_a}$$

$$Q_r = Q_a \cos i$$

Q_r is a force acting through (*3*), vertically.

$$Q_r = f\,\frac{Q_1\,r_1 + Q_2\,r_2}{L_a/\cos i} = f\,\frac{Q_2\,r_2 + Q_1\,r_1}{l_1}$$

For pins (*2*) and (*3*), $Q_b\,L_2 = M_2 = f\,Q_2\,r_2 + f\,Q_3\,r_3$

$$Q_b = \frac{M_2}{L_2} = f\,\frac{Q_2\,r_2 + Q_3\,r_3}{L_2}$$

Q_b is also a vertical force through (*3*).

Combining the two forces Q_r and Q_b,

$$\frac{F}{2} = Q_a + Q_b = f\,\frac{Q_1\,r_1 + Q_2\,r_2}{l_1} + f\,\frac{Q_2\,r_2 + Q_3\,r_3}{L_2}$$

and since in this case all of the pins have the same diameter,

$$\frac{F}{2} = f\,r\left(\frac{Q_1 + Q_2}{l_1} + \frac{Q_2 + Q_3}{L_2}\right).$$

In the governor shown in Fig. 136, when in mid-position,

$$L_2 = 13\tfrac{1}{4}'' \qquad r = .25''$$
$$l_1 = 17'' \qquad \text{Assume} \quad f = .10$$

The forces may be found graphically, as shown in Fig. 137.

The total centrifugal force, C, of one weight is considered as divided

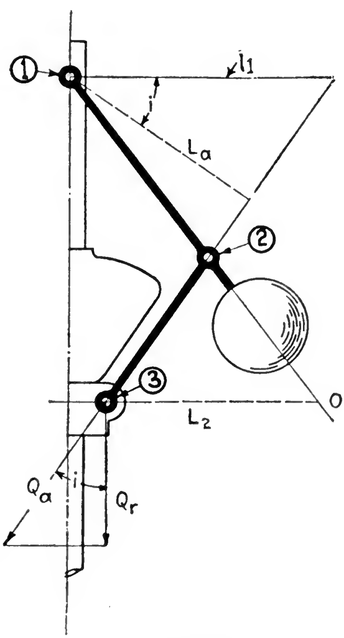


Fig. 136

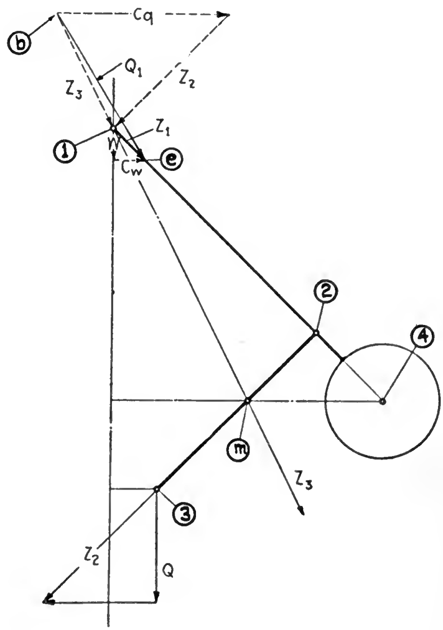


Fig. 137

into two parts, one of which, C_W, holds the fly-weight in equilibrium while the other, C_q, holds the counterpoise in equilibrium. Since neither C_W nor W acts on pin (*2*), their resultant acting on (*1*) must lie in the direction (*1*)(*4*) (Fig. 137); and by drawing the force triangle at (*1*), one of the forces, Z_1, acting on pin (*1*) is found.

The weight Q ($\frac{1}{2}$ weight of counterpoise) is held in equilibrium by two forces, one of which (horizontal) is balanced by the corresponding force from the other side acting through the collar. The other component Z_2 acts in the direction of the link (*2*)(*3*), and as this is regarded as a massless rod, it is the only force acting on (*2*) and (*3*), hence

$$Q_2 = Q_3 = Z_2.$$

The direction of the force (Z_3) on (*1*) produced by Z_2 and C_Q is found from the consideration that all forces on a link must (for equilibrium) pass through a common point, which in this case is (m). The magnitude of the force Z_3 is found by drawing the triangle of forces, shown above (*1*) in the figure.

The total force acting on (1) is the resultant of Z_1 and Z_3, or force (b)(e), Fig. 137.

$$\text{Force } (b)\ (e) = Q_1$$

$$W = 16 \text{ pounds} \qquad Q = 59 \text{ pounds},$$

and for mid-position, $\quad Q_1 = 90 \qquad Q_2 = Q_3 = 84.$

$$\frac{F}{2} = .25 \times .10 \times \left(\frac{174}{17} + \frac{168}{13\frac{1}{4}}\right) = .573, \text{ friction in one half.}$$

Total friction, both sides of governor $= F = 1.146$ pounds.
Strength of governor $= 140$ pounds $= P$.

$$q = \frac{F}{P} = \frac{1.146}{140} = .0082.$$

The detention due to governor friction in the particular Watt-type governor shown in Fig. 56 is—

$$F = f\,r\left(\frac{Q_1 + Q_2}{l} + \frac{Q_2 + Q_3}{L_2}\right)$$

$$L_2 = 16\tfrac{1}{4}''. \qquad r = .25''$$
$$l = 20\tfrac{1}{4}''. \qquad f = .10$$
$$Q_1 = 315. \qquad Q_2 = Q_3 = 193.$$

$$\frac{F}{2} = .025\left(\frac{508}{20.25} + \frac{386}{16.25}\right) = 1.221$$

$$F = 2.442$$

$$q = \frac{F}{P} = \frac{2.442}{500} = .0049$$

In the Hartung governor, shown in Fig. 28, no centrifugal forces are transmitted to the pins, except that necessary to balance the weight of the collar $= 40$ pounds

P_1 = strength of governor = 1100 pounds
r = radius of pins = .25″
L_1 = lever arm (1) (2) = 5.″
L_2 = lever arm (2) (3) = 5.71″
Weight of one fly-weight = 79 pounds.

Figure for mid-position:

Horizontal force on (*1*) to counterbalance $\frac{1}{2}$ of collar weight $= \frac{20 \times 5.71}{5} = 22.84$ pounds. This force also acts on (*2*), in opposite direction.

Pin (*1*) . . . . $Q_1 = \sqrt{(22.84)^2 + (79)^2} = 82.1.$

If the collar moves a small distance Ds, then the rubbing path of friction force on pin (1) is $Ds \cdot \frac{5}{5.71} \cdot \frac{.25}{5} = .0437\ Ds.$

Assume friction coefficient = .10
Friction work on (1) = $.10 \times 82.1 \times .0437\ Ds = .3595\ Ds.$

Pin (*2*) . . . . $Q_2 = \sqrt{(22.84)^2 + (79 + 20)^2} = 101.9.$

Rubbing path $= Ds\ \frac{.25}{5.71}$

Friction work $= .10 \times 101.9 \times .0437\ Ds = .446\ Ds.$

Pin (*3*) . . . . $Q_3 = 20.$

Friction work $= .10 \times 20 \times 0.437\ Ds = .0874\ Ds.$

Pin (*4*) . . . . $Q_4 = 20.$

In mid-position, rubbing path = 0, Friction work = 0.

Total friction work $= (.4460 + .3595 + .0874 + 0)\ Ds = .8929\ Ds.$

Equivalent force at collar, $\frac{F}{2} = \frac{.8929\ Ds}{Ds} = .8929$

$F = 1.7858$ pounds.

$q = \frac{F}{P} = \frac{1.7858}{1100} = .001623$ for mid-position.

CENTRIFUGAL MOMENT OF OBLONG WEIGHTS

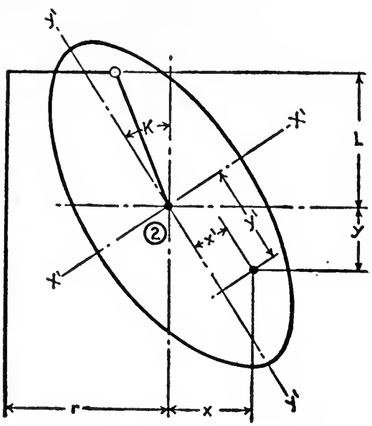


FIG. 140

M = Centrifugal Moment = $\int dm\, u^2(r + x)\,(y + L) = \int dm\, u^2 ry + \int dm\, u^2\, rL + \int dm\, u^2\, x\, y + \int dm\, u^2 \times L$ but $\int dm\, y = 0$ and $\int dm\, x = 0$, because (*2*) is mass center. $\int dm\, u^2\, r\, L = m\, u^2\, r\, L$. See Fig. 140.

To find a simpler expression for $\int dm\, x\, y$, refer it to the principal axes $X'Y'$, and call the coördinates x', y'. For these axes $\int dm\, x'y' = 0$, because the moments of inertia are a maximum and a minimum. By analytical geometry: $x = x' \cos k + y' \sin k$, and $y = -x' \sin k + y' \cos k$. Substitute in $\int dm\, x\, y$ and carry out multiplication

$$\int dm\, x\, y = -\int dm\, x'^2 \sin k \cos k + \int d\, m\, y'^2 \sin k \cos k.$$

The two terms containing the product $x'y'$ vanish, as above explained. Substitute $h'\, dA\, m'$ for dm, where h' = thickness at right angles to paper, dA = differential of area in plane of paper, m' = mass per unit volume. Then

$$\int dm\, x'^2 \sin k \cos k = h'\, m'\, \tfrac{1}{2} \sin 2\, k\, J(y') \quad \text{and}$$
$$\int dm\, y'^2 \sin k \cos k = h'\, m'\, \tfrac{1}{2} \sin 2\, k\, J(x'),$$

where $J(x')$ means the moment of inertia of the plane section of the centrifugal weight about the X' axis, and $J(y')$ the corresponding moment about the Y' axis.

With these notations the centrifugal moment is

$$M = m\, r\, u^2\, L + \tfrac{1}{2}\, h'\, m'\, u^2 \sin 2\, k\, (J_{(x')} - J_{(y')})$$

In the text J_{long} and J_{short} were used in place of $J_{x'}$ and $J_{y'}$. This is correct for the illustration, but must be replaced by $J_s - J_l$, if the X' direction is longer than the Y' axis.

DETAIL CONSTRUCTION OF CHARACTERISTIC FOR FIGURE 31

In the illustration (Fig. 138), (*1*), (*2*), (*3*), (*4*), (*5*) are different positions of the mass center of the centrifugal weight; (*11*), (*12*),

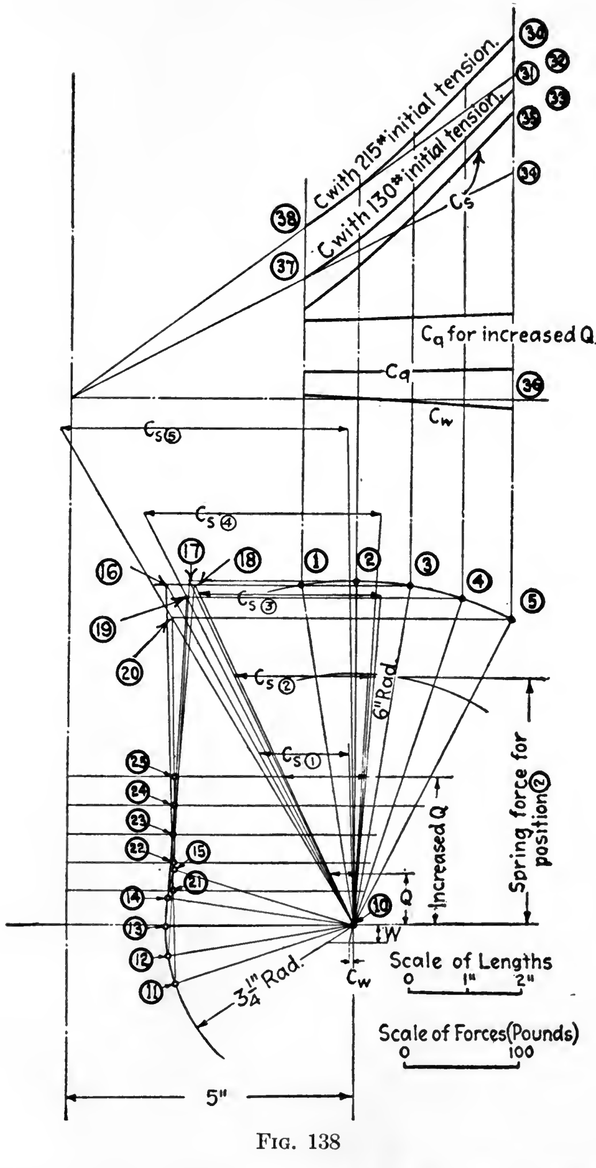


FIG. 138

(*13*), (*14*), (*15*) are the corresponding positions of the inner end of the bell crank; (*21*) (*22*), (*23*), (*24*), (*25*) are the corresponding positions of the upper end of the connecting link; point (*16*) is the intersection of the horizontal through (*1*) and the straight line (*11*) (*21*); points (*17*), (*18*), (*19*), (*20*) are found by intersection of corresponding lines for the other positions. The values of $W = 15$ pounds, $Q = 45$ pounds and increased $Q = 45 + 85$ pounds have been laid off near point (*10*), (right-hand bottom Fig. 138) to the scale of one inch representing 100 pounds. The triangle of forces which is shown in Fig. 32 has been reproduced in convenient form for construction on Fig. 138, as shown in this sketch. The same construction has been carried out with increased Q, and with the various spring forces, each spring force being computed from the "initial tension plus scale times deflection." The spring force for position (*2*) has been marked.

The centrifugal forces necessary to balance the weight of the centrifugal masses are found by a series of triangles of forces below point (*10*). The various centrifugal forces have been laid off and combined above the respective positions of the centrifugal mass. For position (*3*),

$$\text{Angular velocity} = \sqrt{\frac{32.2 \times 12}{15 \times 5.9}}\ \sqrt{C} = 2.09\sqrt{C}$$

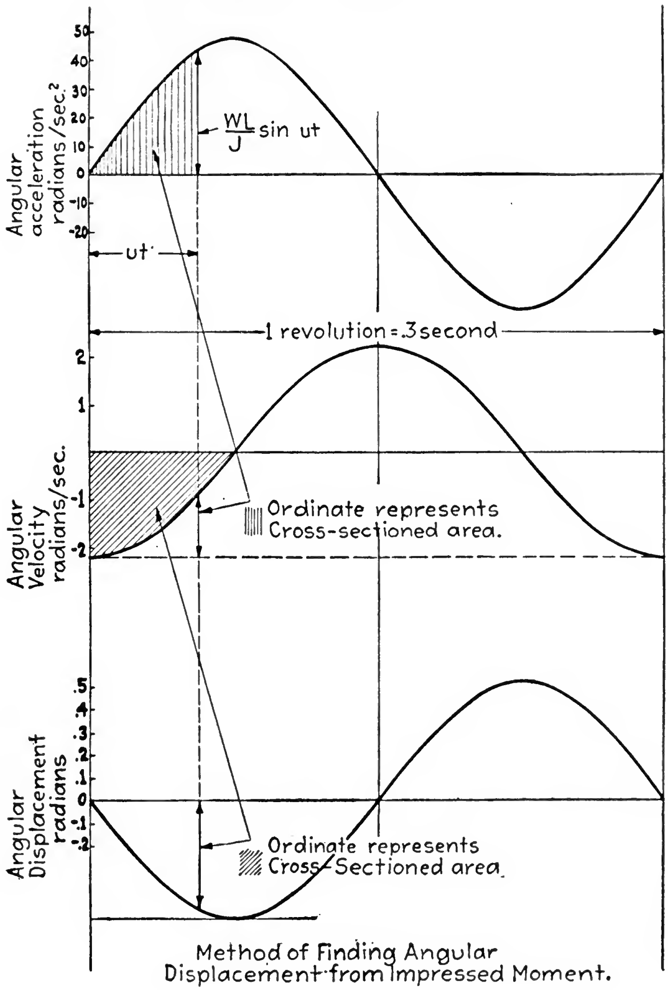


FIG. 139

The illustration, Fig. 139, shows a method of finding angular displacement from a curve of impressed moments, as exemplified by the moment caused by gravity. The illustration is so clear that it needs no comment, with the exception that the units and scales must be carefully watched.

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The following is offered not as a complete and comprehensive bibliography, but as a list of the most important and most readily accessible contributions to governor science. While some of the articles referred to contain theories and statements of doubtful value, they have been included for the purpose of illustrating the historical development of the theory of governing.

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