(Paper No. 3117.)

“Experiments upon the Action of Engine-Governors.”

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ALTHOUGH the subject is admitted to be of great importance, comparatively few experiments have been made on engine-governing. The performances of different governors seem to vary so widely that it is by no means easy to choose the type best suited to an engine working under given conditions. With a view, therefore, to observe in closer mechanical detail the behaviour of different kinds of governor, experiments were made at Mason College, Birmingham, under the superintendence of Professor R. H. Smith, M. Inst. C.E., on nine governors of different types.

One method of obtaining a characteristic speed-curve from a Watt, or any ordinary form of vertical governor, is shown in *Fig. 1*.

A pencil attached to the sleeve S describes lines on the drum D, which is covered with white paper, and made to rotate. In this way it is easy to obtain the heights of the sleeve corresponding to the number of revolutions per minute of the governor. By revolving the governor slowly a straight line is described on the drum. As the speed is increased, the sleeve is gradually lifted, and attains its highest position at a certain

speed which is carefully noted. If the speed is now gradually decreased it is generally observed that the sleeve does not begin to descend until the speed has fallen by 5, 6, or more revolutions per minute, according to the character of the governor under test. By decreasing the speed still further the sleeve is brought to the same series of positions that it occupied when ascending, but in each position the number of revolutions of the governor is found to be less than during the ascent of the sleeve. This is due to the frictional resistances at the joints of all the connected pieces. These frictional resistances are now acting in the opposite direction during the descent, tending now to prevent the balls from closing, and consequently the speed must

be decreased sufficiently for the effect of gravity on the balls and sleeve to overcome these resistances. From such measurements a diagram, such as that shown in *Fig. 2*, can be constructed; the curve CA is for the rising, and the curve BD for the falling sleeve. The distance AB varies considerably with different governors, and is very small indeed (or even zero) with governors provided with a powerful spring, or with both a spring and a heavy weight, although the curves may diverge widely lower down. From these speed-curves, and other data connected with the particular governor considered, it is possible to obtain a practical measurement of those quantities which are mainly concerned in the efficient action of a governor, viz., its sensitiveness, its controlling force, and its controlling energy. These quantities must be expressed in terms of some unit of speed-variation, and in this Paper they will be referred to a change of speed, amounting to 1 per cent. of the mean speed for which the governor is designed.

The term "sensitiveness" will be used to indicate the distance (in inches) through which the sleeve would move in either direction from its mean position in response to a 1 per cent. variation of speed, if there were no frictional resistances to overcome, and when no controlling force is exerted.

The term "controlling force" will be used to indicate the force which would be exerted by the governor upon the controlling gear, if the given variation of speed were to take place without as yet producing any motion of the sleeve.

The term "controlling energy" will be used to indicate the inch-pounds of work that would be done by the sleeve, if, starting from its mean position, it were caused to travel through a distance equal to the sensitiveness, first upwards and back again, then downwards and back again; the speed during each ascent being always 1 per cent. greater, and during each descent 1 per cent. less, than the speed corresponding to the momentary position of a sleeve working without friction. Hence, the "controlling energy" is very nearly the same quantity as the "controlling force," multiplied by four times the distance which represents the "sensitiveness."

The Tangye (Soho), Acme, Lüde, Proell, Watt, Porter, Pickering, Turner-Hartnell, and Belliss governors were those actually tested, the last two being designed to run on the main shaft of the engine. The different governors were not placed on the engine, but those mounted in their own brackets were clamped firmly to a cast-iron table, and the other vertical governors were mounted on a testing-machine designed for the purpose, so that they could be driven at any speed desired. They were made to run steadily at different speeds and with various loads placed on their sleeves, either acting with the governor, *i.e.*, tending to open the balls, or

against it. The height of the governor sleeve was carefully measured, and the speed of rotation was taken by means of a speed-counter. The speed-curves shown are the mean of a very large number of readings, the trials being several times repeated.

Tangye Governor.—This governor, of which the chief working parts are shown in *Fig. 3*, controlled a throttle-valve $1\frac{1}{4}$ inch in diameter. The spring *S* can be lengthened or shortened by turning the cap *C* one way or the other. The top of the spindle *T* slides up and down in vertical parallel guides inside the

governor casting. The chief particulars relating to this governor are as follows:—

Mean resistance of spring	=	{ 79 lbs. per inch extension.
(Mean speed was supposed to be 400 revolutions per minute.)		
Distance between centres <i>O</i> and <i>P</i>	=	$2\frac{1}{8}$ inches.
Distance between <i>O</i> and the bearing point of arm on spindle-head, measured perpen- dicularly to the centre line of the arm . . }	=	$\frac{9}{16}$ inch.
Diameter of each ball	=	$1\frac{5}{8}$ inch.
Full movement of valve	=	0.56 inch.
Weight of governor ball and arm together	=	12 oz.

The speed-curves for this governor are shown in *Fig. 4*, where the full lines indicate the rise of the sleeve and the dotted lines the fall of the sleeve as the speed is increased and decreased. It will be observed that the curves are fairly regular in the vicinity of

the speed at which the governor was intended to run, viz., 400 revolutions per minute, the cap having eight turns. When the cap has only been turned round twice there is no tension on the spring S with the governor at rest. The tension on the spring as the balls open is shown at the different positions of the sleeve by the figures near the black curves drawn in full. It varies in this case from 0 with the governor at rest to 67.15 lbs. (curve 3) with the sleeve at the extreme limit of its movement. It is likewise shown for the other curves. The curves become slightly steeper when the cap is turned to apply a greater tension on the spring, i.e., the sleeve moves a greater distance for a given variation of speed, and therefore the governor becomes more sensitive. The crossing of the up and down curves on the left hand of the diagram is probably due to the erratic variations of friction in the working

Fig. 4.

parts of the governor, but it will be observed that this occurs at a much lower speed than that at which the governor is intended to work.

In this governor the sleeve is at mid-position when 0.28 inch from its starting point, and the mid-position is indicated by a horizontal line drawn across the curves serving also as the base-line of a force-diagram. At the points where this base-line is intersected by any pair of curves, the corresponding load or spring-pull is plotted off as a pair of vertical ordinates, while the horizontal abscissas give the speeds at which the sleeve is balanced under that load at mid-position. By joining the series of points so obtained two almost parallel straight lines, shown in Fig. 4, are obtained, the upper one corresponding to the "down" curve, and the lower one to the "up" curve. The mean horizontal dis-

tance of these straight lines apart is represented by 2.7 horizontal divisions, equal to 13.5 revolutions per minute. Thus the up or down friction needs 6.75 revolutions per minute variation of speed, or 1.7 per cent. of the normal speed of 400 revolutions per minute to overcome it. The mean inclination of these lines, as measured upon the squared paper, is given by a height of 7.24 divisions upon a length of 37.6 divisions. The horizontal distance between them, multiplied by this ratio, will give the vertical distance between them; and one-half of this vertical distance will represent (upon the force-scale) the force that is required to overcome the frictional resistance. Hence the friction

$= \frac{1}{2} \times \frac{7.24}{37.6} \times 2.7 \times 10 \text{ lbs.} = 2.6 \text{ lbs.}$, which is plotted downwards, vertically below the load points corresponding to the

ACME GOVERNOR.

dotted curves, and is also plotted upwards, vertically above those load points which correspond to the curves drawn in full. The controlling-force curve is then drawn through the points thus obtained. As the governor is intended to run at a speed of 400 revolutions per minute, 1 per cent. of this is 4 revolutions,

and this is represented by $\frac{4}{5}$ of one horizontal division. Therefore,

$$\text{controlling force of governor} = \left(\frac{\frac{4}{5}}{37.6} \times 7.24 \times 10 \right) = 1.54 \text{ lb.};$$

that is, a pull of 1.54 lb. on the spindle for 1 per cent. variation of speed without any motion of the valve.

As regards sensitiveness, taking curves 3, 3, it will be seen that an alteration in speed of 170 revolutions per minute is required to move the spindle through a distance of 0.39 inch. Therefore,

$$\text{governor sensitiveness} = \frac{4}{170} \times 0.39 = 0.0091 \text{ inch,}$$

that is, 0.0091-inch movement of spindle per 1 per cent. variation of speed. The

$$\text{controlling energy} = (4 \times 1.54 \times 0.0091) = 0.056 \text{ inch-lb.}$$

Acme Governor.—This governor was of the usual type, controlling a 3-inch throttle-valve, and was made to run at 400 revolutions per minute. *Fig. 5* shows the arrangement provided for varying

the load on the valve spindle, and therefore for varying the speed of the engine. The following are some of the particulars relating to this governor :—

Weight of each hemispherical flyer . . .	=	6 lbs. (approximate).
Full movement of valve	=	0·79 inch.
Mean resistance of the two side springs of the governor, together	=	{ 15·3 lbs. per inch extension.
Mean resistance of the regulating spring S .	=	{ 25·6 lbs. per inch extension.
Unstrained length of spring S	=	5·81 inches.
Length of spring S when on the governor and with slide-block B at mid-position .	=	6·9 inches.
Therefore tension on it when thus placed .	=	27·9 lbs.
<i>With slide-block B at top position—</i>		
Length of spring S with the balls closed .	=	7·19 inches.
Tension on it	=	35·3 lbs.
Tension on spring S with the balls full out .	=	25·6 lbs.
<i>With slide-block B at bottom position—</i>		
Length of spring S with the balls closed .	=	6·82 inches.
Tension on it	=	25·85 lbs.
Tension on it with balls full out	=	36·35 lbs.

Therefore, by means of the screw A and the slide-block B a pressure can be placed on the sleeve, acting upwards when B is at the bottom of the slide, and downwards when B is at the top. With B at mid-position, there is no load or pressure on the valve-spindle due to the spring S. The lever L and the slide are both keyed on to the same pivot-shaft, so that they move together.

The curves taken by running this governor at different speeds are shown in *Fig. 6*, and the difference between the characteristic curves given here and those obtained for the Tangye governor is very marked. The middle curve, that for no load besides friction, is very regular in its working portion, although there is no striking difference between the three pairs of curves after the valve-spindle has moved more than 0·2 inch from the position it occupies with the governor at rest. The hemispherical flyers appear to have a little difficulty in opening properly at first, but work properly afterwards. The loads placed on the valve-spindle, due to the regulating spring, are shown near the two side black curves. The approximate frictional forces are also shown at the various positions of the valve-spindle as before. This governor does not appear to vary greatly as regards sensitiveness, whether a load be placed so as to act with it or against it. To find the friction, the pressures on the sleeve and the speed are plotted off for the sleeve at mid-position, or 0·395 inch from the lowest position. In the one case the pressure acts downwards, and in the

other vertically upwards. Two straight lines are obtained, as for the Tangye governor, but they are not quite parallel. The horizontal distances between them is equivalent to 11 revolutions, 13 revolutions, and 14 revolutions per minute respectively in the three pairs of curves. Their mean inclination is measured by a height of 20·12 divisions upon a length of 16·2 horizontal divisions. For the mid-position of sleeve, the frictional resistance has the following values—

$$\begin{aligned} \text{Block at centre of slide, friction} &= 2 \cdot 6 \times \left(\frac{1}{2} \cdot \frac{20 \cdot 12}{16 \cdot 2} \text{ lbs.} \right) = 3 \cdot 22 \\ \text{,, bottom ,, ,,} &= 2 \cdot 2 \times \left(\frac{1}{2} \cdot \frac{20 \cdot 12}{16 \cdot 2} \text{ lbs.} \right) = 2 \cdot 73 \\ \text{,, top ,, ,,} &= 2 \cdot 8 \times \left(\frac{1}{2} \cdot \frac{20 \cdot 12}{16 \cdot 2} \text{ lbs.} \right) = 3 \cdot 48 \end{aligned}$$

As the normal speed of the governor is supposed to be 400 revolu-

Fig. 6.

The movement, as measured by the other curves, is—

$$\text{By the right-hand curves, } \frac{4}{102.5} \times 0.4 = 0.0156 \text{ inch.}$$

$$\text{,, left-hand ,, } \frac{4}{127.5} \times 0.4 = 0.0125 \text{ inch.}$$

For the normal speed of 400 revolutions per minute, the middle curves approach the nearest, and the sensitiveness may be taken as 0.016 inch. The

$$\text{controlling energy} = (4 \times 1.98 \times 0.016) = 0.13 \text{ inch-lb.}$$

The points for the force curve are obtained by setting off, upon the ordinates of the speed curves, the positive or negative heights representing the frictional resistance as already given. The force curve is found to be practically a straight line.

Lüde Governor.—This type of governor is shown in *Fig. 7*. It is provided with four pendulums, two of which are shown, a heavy muff, W, and a light spring, E. The throttle-valve is of the rotating type, and is worked by the rod L and a small lever attached to the spindle of the valve. In the governor tested, the throttle-valve was $3\frac{1}{2}$ inches in diameter. The governor was supposed to run at 200 revolutions per minute, and the sleeve had a movement of 1.5 inch. The internal spring E was tested, and its resistance at the different positions of the sleeve is shown in *Fig. 8*, which gives the speed curves obtained, both with no load and also when several different loads are placed on the sleeve. The following are details relating to this governor:—

LÜDE GOVERNOR.

- Diameter of each of the small balls . . . = $2\frac{3}{16}$ inches.
- Mean resistance of spring = 15.2 lbs. per inch compression.
- Weight of muff, together with the } = 33.5 lbs.
- four pendulums }
- Mean speed with no load on sleeve . . . = 200 revolutions per minute.

It will be observed that in this case the curves are nearly straight for the greater part of their length, and nearly approach to the vertical, showing great sensitiveness. This becomes more marked as the load on the governor sleeve is increased. With a 50-lb. load, a variation in speed of 30 revolutions per minute sends the governor sleeve from its lowest position to its highest. One peculiarity is that some of the curves join at the top, the sleeve commencing to fall immediately the speed of rotation is slightly decreased. This governor appeared to possess great stability, and worked splendidly within reasonable limits of load and speed. Its power to shift a valve-gear, however, did not appear to be large, considering the size and weight of the governor. As the rotating throttle-valve with which it was provided required very little

Fig. 8.

effort on the part of the governor to move it, this drawback would not appear to be of great importance. The sleeve is at mid-position when 0.75 inch from its lowest position. The various loads and speeds are plotted off above the mid-position line, and two curves are obtained which diverge more and more as the load on the governor is increased. These variable loads are exclusive of the weight of the muff and the compression of the spring, which are the same in all cases with the sleeve at mid-position. As this governor is not expected to work with nearly such heavy loads on it, the three right-hand pairs of curves will not be considered in estimating the controlling force and the sensitiveness. The horizontal distances between the ascending and descending curves represent variations of speed amounting to 11, 4.4, 3.9, 4.5, and 8.75 revolutions per minute respectively.

The frictional resistances, as measured from each pair of curves, are as follows—

With no load on the sleeve, friction	=	2.62	lbs.
" 10 lbs.	"	1.04	"
" 20 "	"	1.90	"
" 30 "	"	2.21	"
" 50 "	"	4.29	"

The normal speed of the governor is supposed to be 200 revolutions per minute, and 1 per cent. of this is 2 revolutions, equivalent to $\frac{2}{3}$ of one horizontal division. Accordingly the controlling force is 0.953 lb. when the load on the sleeve is between nothing and 10 lbs., but amounts to 1.96 lbs. when the load is 20 lbs. Under normal conditions, with no load on the sleeve, the sensitiveness is found to be 0.056 inch, while the controlling energy is 0.213 inch-lb. The sensitiveness increases greatly as the extra loads on the governor are increased.

PROELL GOVERNOR.

Proell Governor.—From tests of this governor, *Fig. 9*, the curves shown in *Fig. 10* were obtained.

Fig. 10.

The following are details of this type:—

Weight of each ball	=	4.8 lbs.
Unstrained length of large spring of the governor	}	7 $\frac{1}{2}$ inches.
Compressive strength		120 lbs. per inch extension.
Pitch of the coils	=	$\frac{5}{16}$ inch.
Outside diameter of coils	=	2 $\frac{3}{8}$ inches.
Diameter of steel of which the spring was made as given by the micrometer gauge.	=	0.313 inch.

[THE INST. C.E. VOL. CXXXVII.]

2 c

When placed in the governor, the length of the spring was $5\frac{1}{8}$ inches, which was equivalent to a pressure of 232·5 lbs. on the cap with the governor at rest.

Natural length of small regulating spring	=	$7\frac{3}{4}$ inches.
Pitch of coils	=	$\frac{1}{2}$ inch.
Outside diameter	=	$1\frac{1}{4}$ inch.
Diameter of its steel	=	0·177 inch.
Mean strength	=	16·7 lbs. per inch.

With the slide-block at the bottom of the slide, the regulating spring was compressed $\frac{3}{4}$ inch, when the governor was at rest; the pressure then exerted is 12·52 lbs., which is equivalent to 8·24 lbs. acting vertically upwards at the centre of the governor spindle. With the slide-block at the centre of the slide, the small spring was stretched $\frac{5}{32}$ inch with the governor at rest; the downward

pull then exerted is 2·6 lbs., equivalent to 1·33 lb. at the centre of the governor spindle. With the slide-block at the top of the slide, the small spring is extended $\frac{5}{8}$ inch, the governor being at rest; the downward pull then exerted is 10·43 lbs., equivalent to 3·97 lbs. at centre of governor spindle.

The downward force exerted at the centre line of the governor spindle by the small spring has the following values when the balls are fully extended—

	lbs.
Block in lowest position	1·87
„ central „	5·2
„ highest „	6·56

The pressure exerted by the large spring with the balls fully extended was 326·25 lbs.

Fig. 11 shows the distances from the centre line of the governor-spindle to the slide-block, in its different positions.

Diameter of each ball	=	$3\frac{1}{4}$ inches.
Weight of ball and arm	=	6 lbs. $4\frac{1}{2}$ ozs.
Length of governor arm from ball centre to } pin centre	=	8 inches.
Full movement of governor-sleeve	=	1·56 inch.

Fig. 12 shows the leverage exerted by the bell-crank levers which are attached to the governor arms. As shown by the curves in Fig. 10, the governor was exceedingly sensitive, a

difference of only 9 revolutions per minute being sufficient to send the governor-sleeve from its lowest to its highest position, the slide-block being at the centre of the slide.

The governor-sleeve is at mid-position when 0·78 inch high. The loads placed on the sleeve in this case are those due to the weight of the lever, &c., and the tension or compression of the regulating spring. As it is difficult to estimate the pressure on the governor-sleeve due to the weight of the lever, this was measured by a spring-balance attached to the lever at the centre line of the governor-spindle, and found to be 3 lbs. There was also a change in the pressure on the sleeve due to the shifting of the slide-block and its attachments.

The pressures exerted at centre line of governor-spindle with the block in the three positions were—

2·01 lbs.	when the slide-block was in its lowest position
1·56 " " " "	middle "
and 1·16 " " " "	highest "

The loads due to the regulating spring, with the sleeve at mid-position, were—

3·28 lbs.	acting vertically upwards with slide-block in lowest position
2·91 " " " "	downwards " middle "
and 5·24 " " " "	highest "

The total loads for the three positions are therefore—

(3 + 2·01 - 3·28) lbs.	= 1·73 lb. acting downwards for 1st position
(3 + 1·56 + 2·91) lbs.	= 6·47 lbs. " 2nd "
and (3 + 1·16 + 5·24) lbs.	= 9·4 lbs. " 3rd "

These loads have been plotted upwards from the mid-position line, as shown in the *Fig.*, and to the given scale. Two straight lines are obtained, as shown; and, measuring their inclination by the divisions of the squared paper as before, the following results are obtained—

With slide-block in lowest position, friction	. . .	=	lbs. 0·43
central "	"	=	0·708
highest "	"	=	1·04

The normal speed of the governor was supposed to be 190 revolutions per minute, and 1 per cent. of this is 1·9 revolutions, equivalent to 1·9 horizontal divisions. Hence controlling force = 0·50 lb.

Considering the middle pair of curves, *i.e.*, with the slide-block at mid-position, it requires an increase in speed of from 187.5 revolutions to 194.3 revolutions per minute to lift the sleeve from a height of 0.25 inch to 1.5 inch. Therefore,

$$\text{sensitiveness} = \frac{1.9}{6.8} \times 1.25 = 0.349.$$

$$\text{Controlling energy} = (4 \times 0.5 \times 0.35) = 0.70 \text{ inch-lb.}$$

All the governors referred to were driven by belting from a

counter-shaft, which was in its turn driven by means of a vertical steam-engine. The other vertical governors tested were driven by means of an electric motor, the speed of which could be regulated exactly by means of suitable resistance frames. The testing arrangements are shown in *Figs. 13 and 14.*

Watt Governor.—The Watt governor, *Fig. 15*, was of an old pattern, with large balls and short arms, as shown in *Fig. 16*. The curves, *Fig. 17*, given can hardly be taken as representing results given by Watt governors, as in this case there was a considerable amount of sticking in the governor, partly due, no doubt, to the peculiar manner in which the arms were suspended. The curves also show a large decrease in sensitiveness as the valve becomes nearly closed, as shown by the right-hand curve from A to B. There was also a large amount of friction be-

TESTING ARRANGEMENT—SIDE VIEW.

TESTING ARRANGEMENT—PLAN.

tween the valve-spindle V and the stuffing-box as the governor-sleeve approached its full height. This governor ran at a low speed, the mean being about 95 revolutions per minute. The

sensitiveness was small. The full movement of the sleeve was 1·6 inch.

The curves 1, 1, are for the governor with no valve attached to the valve-lever; and curves 2, 2, are for the governor with the valve attached. Two straight lines are obtained, as for the other governors, and, measuring their mean inclination as before, the results given by the calculation are as follows—

For the curves 1, 1,	friction =	3·36	lbs.
„ 2, 2,	„	=	4·3

The normal speed of the governor is given by the makers as 90 revolutions per minute, and 1 per cent. of this is 0·9 revolution, which is equivalent to 1·8 horizontal divisions. Hence the controlling force = 4·18 lbs.

Sensitiveness (taking the right-hand curve) = 0·113 inch.

Controlling energy = $4 \times 0·113$
 $\times 4·18 = 0·189$ inch-lb.

WATT GOVERNOR.

Porter Governor.—Speed-curves for a Porter governor, which was loaded with a central weight of 33 lbs., the small balls each weighing 2·58 lbs., are given in Fig. 18. The throttle-valve spindle was connected to a lever in a similar manner to that shown for the Watt governor. The valve itself was of the double equilibrium type, and weighed 2·75 lbs. It will be observed that the working curves given by this governor are much more regular than those given by the Watt. Three pairs of curves were taken under the following conditions:—(1) Without the valve being attached to the governor lever; (2) With the valve attached, and the nut of the stuffing-box screwed moderately, *i.e.*, under working conditions; (3) With the stuffing-box screwed tightly.

Even with the working curves there appears to be a considerable amount of frictional resistance in the governor and at the valve-

spindle, especially when the sleeve nearly reaches its highest point. The heavy central weight, however, overcomes this, and the

Fig. 17.

SPEED-CURVES FOR WATT GOVERNOR.

sensitiveness of this governor was considerably greater than that of the Watt, being 0.082 as compared with 0.011 for a 1 per cent. variation of balanced speed.

Fig. 18.

SPEED-CURVES FOR PORTER GOVERNOR.

The sleeve in this case is at mid-position when at a distance of 1.015 inch from its lowest position. The loads are due to the weight

of the throttle-valve and the frictional resistance at the stuffing-box. The load for curve 1, 1, is 0, because in this case the throttle-valve was not attached to the lever. For the load-points of curves 2, 2, 2·43 lbs. must be plotted upwards for the up curve, because the forces opposing the upward motion of the sleeve, reckoned at the centre of the governor spindle, amount to 2·43 lbs., and act vertically downwards. For the down curve (1·02 - 1·41) lb. = 0·39 lb. must be plotted downwards, because the forces tending to oppose the governor sleeve are now acting vertically upwards; - 0·39 lb. downwards = + 0·39 lb. plotted upwards, and, therefore, in the diagram 0·39 lb. is plotted upwards from the mid-position line. The force required to lift the throttle-valve at mid-position for curves 2, 2, is 5 lbs., equivalent to 2·43 lbs. at the centre of the governor spindle. For the down curve the weight of the throttle-valve must be subtracted from the frictional resistance. The weight of throttle-valve = 2·75 lbs. = 1·41 lb. at centre of spindle. For the load-points of curves 3, 3, similarly 8·75 is plotted upwards for the up-moving curve, and 7·34 - 1·41 lb. = 5·93 lbs. plotted downwards for the down-moving curve, the friction forces in this case acting vertically upwards. (For the up-moving curve, friction = 8·75 lbs. to 1·41 lb. = 7·34 lbs.; for down-moving curve 1·41 lb. must be subtracted from 7·34 lbs.) Joining the load-points thus obtained, the two straight lines of the *Fig.* are traced, and the calculations, as before, give the following results:—

	lbs.
For the curves 1, 1, friction	= 1·71
„ 2, 2, „	= 2·87
„ 3, 3, „	= 9·12

The governor is intended to run at a mean speed of 260 revolutions per minute, and 1 per cent. of this = 2·6 revolutions. The controlling force = 1·04 lb.; the sensitiveness (for curve 2) = 0·082; and the controlling energy = $4 \times 1·04 \times 0·08$ inch-lb. = 0·3328 inch-lb.

Pickering Governor.—Four pairs of curves obtained from a Pickering governor which controlled a 2-inch throttle-valve are given in *Fig. 19*. Curves 2, 2, represent most accurately the working conditions. There was considerable frictional resistance at the stuffing-box of the valve-spindle, although this was not screwed tighter than when actually working on an engine. The controlling force of this governor was not very large for its size, and its sensitiveness was not as great as that of an ordinary Porter governor. A load could be placed on the

valve-spindle by means of a worm-wheel and a spiral spring, and the four curves were traced with the worm-wheel turned through different angles, and producing upon the spindle pressures of 8 lbs., 13.33 lbs., and 22.16 lbs. in its successive positions.

Fig. 19.

SPEED-CURVES FOR PICKERING GOVERNOR.

In this case the sleeve is at mid-position when 0.28 inch high. The curves are traced by plotting off the loads upwards from the mid-position line as before, and the following results are obtained:—

	lbs.
For the curves 1, 1, friction	. . = 4.28
„ 2, 2, „	. . = 3.85
„ 3, 3 and 4, 4, friction	= 3.58

The normal speed of the governor is supposed to be 400 revolutions per minute, and 1 per cent. of this is 4 revolutions, equivalent to $\frac{1}{25}$ of one horizontal division. The controlling force = 0.424 lb.; the sensitive-ness (taking curves 2, 2) = 0.018; and the controlling energy = $(0.424 \times 0.018 \times 4) = 0.0305$ inch-lb.

Turner-Hartnell Shaft Governor.—This governor was tested by mounting it on a mandrel and running it steadily at different speeds in the lathe, and with the springs at different compressions, the eccentric-strap being taken off. When these observations had been made, the eccentric-strap was put on, and an arrangement was made as shown in Fig. 20 for applying a load. The wire W was so fixed as to prevent the eccentric-strap from

swinging round. The load was applied by means of the strong spring-balance S and the strong thick india-rubber band A. The vertical stalk P was fixed on a bracket bolted to the lathe-bed. The pointer R was arranged to touch the same point of the eccentric-strap every revolution, and as the throw of the eccentric became less, due to the governor weights opening, it had to be moved inwards until it again touched the eccentric-strap. The alteration in the eccentricity was measured directly between two fine parallel lines, one being drawn on the fixed socket carrying the pointer, thus being fixed, and the other on the pointer itself.

Fig. 21.

TURNER-HARTNELL GOVERNOR.

The following are details relating to this governor :—

Diameter of governor disks	=	15½ inches.
Diameter of shaft	=	1½ inch.
Average resistance of the first spring . . .	=	{ 135·3 lbs. per inch compression.
Average resistance of the second	=	{ 99 lbs. per inch compression.

The springs were compressed when the governor weights opened, as shown in *Fig. 21*. The lengths of the springs without

Fig. 22.

SPEED-CURVES FOR TURNER-HARTNELL GOVERNOR.

any compression were $4\frac{1}{2}$ inches (strong spring) and $4\frac{1}{2}$ inches respectively. The weight of each flyer was 9·8 lbs. The speed-curves, taken with the eccentric-strap removed, are shown in *Fig. 22*. The springs were screwed until the length of each was $4\frac{1}{8}$ inches in curve 1, $3\frac{7}{8}$ inches in curve 2, and 3·71 inches in curve 3. When the weights were fully extended their length was $3\frac{1}{16}$ inches. These curves possess remarkable features, the chief being the great decrease of speed required to cause

the weights to move inwards after they have reached their outermost position.

The governor was constructed to run at 200 revolutions per minute, and it will be observed at this speed curve 2 is a little irregular when the weights are opening and also when they are closing.

In Fig. 23 is shown the effect on curve 2 of applying different loads on the eccentric-strap by means of the spring-balance and india-rubber strap. With the greatest load the pull on the strap varied between 34.5 lbs. and 37.5 lbs., averaging 36 lbs. With the medium load the pull on strap varied between 24 lbs. and 26.5 lbs., averaging 25.25 lbs. With the minimum load the pull on strap varied between 15.75 lbs. and 18.25 lbs., averaging 17 lbs. The forces plotted on the diagram are the average forces here given. As will be seen, the effects on the governor were irregular. With the medium tension acting the governor the sensitiveness was altered

Fig. 23.

from 0.023 to 0.016, and the speed for keeping the weights at mid-position from 197.7 to 211.8 revolutions per minute. For curves 1, 1, taking the strength of the stiffer spring at 94.8 lbs. per inch of compression, the spring-pressure calculated from its variations of length

will be:—35.55 lbs. when the weights are closed, 136.3 lbs. when the weights are fully extended, and 85.92 lbs. at mid-position. In the same way, taking the weaker spring as having a strength of 91.1 lbs. per inch, the spring-pressures will be:—37.0 lbs. when the weights are closed, 133.8 lbs. when the weights are fully extended, and 85.4 lbs. at mid-position. At mid-position, therefore, the spring-pressure may be taken at the mean value of 85.66 lbs. Repeating the same calculation for the curves 2, 2, the spring-pressure at mid-position is found to be 108.86 lbs., and for curves 3, 3, it amounts to 123.73 lbs. At mid-position of the gear, the balancing speed in the case of curves 1, 1, is 173.5 revolutions per minute, while for curves 2, 2, it is 198.7 revolutions per minute, and for curves 3, 3, it is 216.7 revolutions per minute. Comparing curves 1 and 2, therefore, for a unit incre-

ment of speed (1 revolution per minute), the increment of spring-pressure, is—

$$\frac{108.86 - 85.66}{198.7 - 173.5} = \frac{23.2}{25.2} = 0.92 \text{ lb.}$$

And comparing curves 2 and 3, for the same increment of speed, the increment of spring-pressure amounts to—

$$\frac{123.73 - 108.86}{216.7 - 198.7} = 0.82 \text{ lb.}$$

Therefore the average increase of pull of one spring for an increase of 1 revolution per minute

$$= \frac{0.92 + 0.82}{2} = 0.87 \text{ lb.}$$

But the mean speed for curve 2 = 198.7 revolutions per minute, and 1 per cent. of this = 1.987 revolution per minute, and therefore increase of pull of spring for an increase in speed of 1 per cent. = $(0.87 \times 1.987) = 1.728 \text{ lb.}$

The distance from the spring centre to the centre of the trunnion is $2\frac{5}{16}$ inches, while the distance of the centre of the pin, which carries and guides the eccentric, from the centre of the

trunnion is 1 inch; so that, apart from centrifugal force, a pull of 1 lb. on each of the two springs would pull the eccentric over with a force of $(2 \times 2\frac{5}{16} \times 1) = 4\frac{5}{8} \text{ lbs.}$ In other words, if the momentary difference between the centrifugal force of each weight and the actual pull of the spring amounts to 1 lb., it will require a force of $4\frac{5}{8} \text{ lbs.}$, applied by the eccentric, to balance that difference. Hence, the controlling force of this governor as applied to the eccentric will be $(1.728 \times 4\frac{5}{8}) = 7.992 \text{ lbs.}$

Taking the curve for the medium tension in Fig. 23, a change of speed from 190 to 226 revolutions per minute alters the position of the eccentric by $(1.1 - 0.8) = 0.3 \text{ inch, i.e., a change of 36 revolutions per minute corresponds to a movement of 0.3 inch. Therefore the sensitiveness} = 0.016. \text{ (In Fig. 22 the sensitiveness} = 0.023.) \text{ The controlling energy} = (7.99 \times 0.016 \times 4) \text{ inch-lbs.} = 0.511 \text{ inch-lb.}$

Belliss Shaft Governor.—The governor of this type, Fig. 24,

was supposed to run at about 300 revolutions per minute, and to control an engine of 60 HP. The brass sleeve S is moved by the levers KK along the shaft P. A lever is fixed on to the sleeve at

Fig. 25.

W which moves the valve-gear. In order to test the governor it was placed between the centres of a lathe, and driven by the electric motor. The regulating arrangement was attached as shown in Fig. 25.

Fig. 26.

Three pairs of curves, Fig. 26, were obtained as follows:—
Curves 1, 1, with no regulating spring attached to the lever.
Curves 2, 2, with regulating spring attached, but with no tension at starting. Curves 3, 3, with the regulating spring stretched 1.28 inch (= 34.12 lbs. pull) at the start.

The following are details relating to this governor:—

Mean strength of governor springs = 5·6 lbs. per inch extension.
 Mean strength of regulating spring = 26·66 lbs. per inch extension.
 Weight of throttle-valve and vertical gear = 4·75 lbs.
 Weight of one ball, with arm, &c. = 7·42 lbs.

This governor appeared to be very powerful, and at the same time fairly sensitive. Curves 3, 3 give the nearest approach to the actual working curves.¹ The movements of the sleeve S were in this case measured directly, which could easily be done as the sleeve kept very steady, and the speed of rotation could be regulated exactly. The speed was recorded by a speed counter, driven by the cone pulley of the lathe.

The sleeve is at mid-position when about 0·44 inch from its starting point. The loads calculated are zero for curves 1, 1, owing to the fact that no regulating spring is on the governor lever; 17·07 lbs. for curves 2, 2; and 37·76 lbs. for curves 3, 3. These are plotted off above the mid-position line, according to scale; and, tracing the two lines shown in *Fig. 26*, the following results are obtained by the method already described:—

	lbs.
For the curves 1, 1, friction	= 8·8
„ 2, 2, „	= 2·34
„ 3, 3, „	= 2·21

The mean speed of the governor is given as 300 revolutions per minute, and 1 per cent. of this is 3 revolutions, equivalent to 1·5 horizontal divisions. Hence, the controlling force of the governor = 3·89 lbs.

Considering curve 3, which is the nearest approach to the working curve, if the speed is altered from 238 to 305 revolutions per minute, the sleeve is moved from 0·1 inch to 0·8 inch; therefore, a change in speed of 67 revolutions per minute moves the sleeve 0·7 inch; therefore, the sensitiveness = 0·031. The controlling energy = $(3·89 \times 0·031 \times 4) = 0·482$ inch-lb.

Comparison of Different Governors.—The frictional resistance was reduced to a minimum in each governor by well cleaning and oiling the working parts, but even then it varied considerably in amount, as is shown by the forms taken by several of the curves,

¹ It was intended to obtain curves such that the mean speed of rotation was 300 revolutions per minute, the speed given by the makers for this particular governor, but this was impossible, owing to the shortness of time during which the governor could be spared for the experiments.

and also by the calculations obtained from them. It will be observed that the Proell governor possessed very little frictional resistance in its working parts as compared with some of the others, and was closely followed in this respect by the Tangye and four-pendulum governors. It should be remembered, however, that the Proell governor had no valve attached, and therefore the frictional resistance of the valve-spindle at the stuffing-box is an element not included in the experiments. The Pickering governor curves exhibit a frictional resistance much greater than might have been expected, considering that it possesses no pin joints. It is, therefore, certain that the greater part of the frictional resistance in many governors is produced at the stuffing-box of the valve-spindle, although, in the governors tested, this was not screwed tighter than is usual in practice. It is also probable that in a Pickering governor there is a considerable amount of frictional resistance at the points where the lever of the regulating spring presses on the governor valve-spindle, and also inside the cap at the top of the governor.

The following Table shows the chief results obtained from the nine governors tested :—

Type of Governor.	Mean Speed Revolutions.	Sensitive- ness.	Con- trolling Force.	Con- trolling Energy.	Approximate Mean Frictional Resistance in Lbs. for Working Curves.
	Per Minute.		Inch-lbs.	lbs.	
Tangye (Soho)	400·0	0·009	1·54	0·056	2·6
Acme	400·0	0·016	1·98	0·13	3·22
Lüde (4-pendulum) governor	Load = 0 200·0	0·056	0·95	0·213	2·62
	Load = 20 lbs. 229·25	0·102	1·96	0·799	1·90
Proell	190·0	0·35	0·50	0·70	0·71
Watt	90·0	0·113	4·18	0·189	4·3
Porter	260·0	0·08	1·04	0·33	2·87
Pickering	400·0	0·018	0·42	0·03	3·85
Turner - Hartnell } (shaft-governor)	200·0	0·016	7·99	0·511	..
Belliss (shaft-go- vernors)	300·0	0·031	3·89	0·482	2·21

Comparing the values of the controlling force, it will be noticed that the small Tangye governor exhibits a high value; it is, however, lacking in sensitiveness. Another notable feature is the large controlling force of the Turner-Hartnell governor, and the large amount of frictional resistance in its working parts. It is evident that the controlling force depends largely on the

weight of the balls, and the central weight or muff of the governor, for the highest values are exhibited by the Lûde, Watt, Porter, Turner-Hartnell, and Belliss governors, which all had either heavy balls, weights, or a heavy central load. As regards sensitiveness, the Proell governor appears to be far superior to the rest. It is striking how favourably the Porter governor compares with some of the more modern types, possessing, as it does, both sensitiveness and great controlling force. The Watt governor also shows large controlling force, due no doubt to its very heavy balls.

Results.—The following may be regarded as a summary of the results arrived at:—

1. In testing governors by running them steadily at different speeds, great difficulties have to be overcome to obtain the natural characteristics of the governors considered, chiefly owing to frictional resistances, which appear to vary erratically.

2. If the different governors were placed on the engine and tested by suddenly altering the load, there would still be uncertainty in the results, owing to the variations of the frictional resistance in the valve-gear which is connected with the sleeve (or the eccentric) of the governor.

3. The controlling force of a governor is increased by increasing the weight of the balls or the central load.

4. The controlling force and sensitiveness are increased by providing the governor with powerful springs.

5. Increasing the masses of the revolving balls or weights also increases the amount of frictional resistance.

6. Shaft-governors can be made much more powerful, bulk for bulk, than vertical governors.

7. The advantages of using the more modern types of governor over using the older types, such as well-designed Watt and Porter governors, are not so great as perhaps might be expected.

In conclusion, the Author desires to express his appreciation of the advice and assistance of Professor R. H. Smith, M. Inst. C.E., under whose superintendence most of the experiments were carried out.

The Paper is accompanied by twenty-eight drawings, from which the Figures in the text have been prepared.

APPENDIX.

TO OBTAIN THE CONTROLLING FORCE GRAPHICALLY.

A method of obtaining graphically the controlling force of a vertical governor is shown in *Figs. 27*.

Let C = centripetal force in lbs. acting at the centre of the governor ball in order to maintain equilibrium at the normal speed.

$W \Rightarrow$ weight in lbs. of each of the governor balls.

T, t = tensions in lbs. on the upper and lower arms respectively.

N = number of revolutions of the governor per minute at the normal speed.

F = the total load on the sleeve divided by the number of balls.

f = the controlling force exerted at the sleeve for each ball in the governor, or the whole controlling force divided by the number of balls.

θ and ϕ = the angles made by the centre lines of the upper and lower arms with the vertical centre line of the spindle when the governor is running at normal speed.

R = radius, in feet, of the circle described by the centres of the governor balls.

The force diagram is constructed as follows:—

Draw a vertical line and mark off BV, VD , representing, to the same scale, W and F^1 respectively in magnitude and direction.

Draw VL parallel to the upper arm OS , and VG parallel to the lower arm OY .

Draw VM and DX both perpendicular to BD . Through X draw XA parallel to VL , and through B draw BA parallel to VM and DX .

It is evident that the line BA represents the force C in magnitude and direction.

Now the force C is partly due to the effect of W and partly to the effect of F . Calling the former part C_w , and the latter C_f ,

$$C = C_w + C_f.$$

In the diagram, the line BL represents C_w , and LA represents C_f .

Now

$$C = \frac{4\pi^2 N^2 R}{g} \cdot W,$$

¹ It will be noticed that only half the central weight is here taken into consideration, the other half being supposed to act on the other side of the centre line of the governor spindle.

so that C is proportional to N^2 , and 1 per cent. increase of N is equivalent to an increase of $0.02 C$ in C .

Produce BA to E , making AE equal to $0.02 C$.

Draw EG parallel to AX , produce VX to G , and draw GK parallel to XD .

Then the line DK represents half the force f to the same scale as F and W .

To obtain its magnitude from the similar figures $VDXM$, $VKG P$:—

$$\frac{f}{F} = \frac{0.02 C}{C_f} = 0.02 \left\{ \frac{C_w + C_f}{C_f} \right\},$$

$$i.e. \quad f = 0.02 F \frac{C_w + C_f}{C_f} = 0.02 \left\{ 1 + \frac{C_w}{C_f} \right\} F.$$

f is the force for 1 ball only. The controlling force for 2 balls is $2f$, and for 4 balls it is $4f$.

This expression for the magnitude of the controlling forces is applicable to any vertical governor, no matter what may be the train of mechanism between the balls and sleeve.

For the special kind of governor shown in *Figs. 27*, f can be obtained as follows :—

$$\begin{aligned} 0.02 C &= \frac{F}{F(\tan \theta + \tan \phi)} = \frac{1}{\tan \theta + \tan \phi}, \\ i.e. \quad f &= \frac{0.02 C}{\tan \theta + \tan \phi} = 0.02 \frac{W \tan \theta + F(\tan \theta + \tan \phi)}{\tan \theta + \tan \phi} \\ &= 0.02 \left\{ F + \frac{W}{1 + \frac{\tan \phi}{\tan \theta}} \right\} \\ &= 0.02 \left\{ F + W \frac{H - h}{H} \right\}. \end{aligned}$$