The proportions of the new governor can be chosen to cover adequately the ranges of the governor constants $K_a$, $T_a$, and $T_f$ as needed in practice [see equations (11)].

Typical experimental runs of the dashpot governor with equation (23) and the new governor for slow and fast prime movers are shown in Fig. 8. These are curves for sudden load changes. Both governors are adjusted for optimum performance. The slow engine case is for prime movers that accelerate initially at 10 percent per second on full load off, whereas the fast case is for 100 percent per second and the same disturbance.

Frequency response curves for the dashpot governor of equations (25) and (24) are shown in Fig. 9(a). The curve for the small needle valve opening is at the minimum value 0.003 of $c_0$, whereas the other is for the maximum, namely $c_f = 0.2$. Corresponding curves are shown in Fig. 9(b) for the new governor.

These are for a specific practical governor in which $a$, $\alpha$, and $\beta$ of equations (2), (16), and (17) are varied to give the extreme values of governor constants $K_a$, $T_a$, and $T_f$. The reader will note that there is a much greater choice of breakpoints on the curves in Fig. 9(b) than for those in Fig. 9(a), thus permitting a correspondingly greater choice of governor-engine performance.

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References


DISCUSSION

A. Frank D’Souza

As stated by Professor Oldenburger, the first feedback control system manufactured by industry was probably the flyball governor developed by Watt. Since that time many improvements have been made in its performance. Now the new governor described by Professor Oldenburger is a major improvement over the existing dashpot-type governor.

However, the author states that the mechanical ballhead is by far the best speed measuring means employed in industry and that it is superior to other devices. But the mechanical ballhead is an essentially nonlinear device. Besides the inertia, friction, and backlash of the ballhead, the centrifugal force acting on the flyweights is proportional to the square of the velocity. If the speed deviation is large, this nonlinearity may have adverse effect on the governor performance. An electrical error detecting unit, on the other hand, is a linear device when operating within its linear range. The writer would like to question the author whether the performance of the double servo governor of Fig. 7 could be improved further by replacing the ballhead with an electrical sensing unit, amplifier, and torque motor. If the performance can be improved in this manner, then the extra cost could perhaps be justified.

M. Leum

The author has written a paper of interest to users and manufacturers of speed governing systems. He has provoked the following questions and thoughts which will contribute to the general field of governing. The discussion to follow presents, first, a classification of isochronous governor types, secondly, a numerical example along with actual engine test data, and finally, comments on specific statements made by the author.

A multitude of ways has been found to construct droop and isochronous governors. In general the isochronous governor is derived by "de-drooping" the droop governor. The various methods used to obtain isochronous governing along with stability fall into four categories as discussed in the unpublished memoranda circulated within the Woodward Governor Company [12]. For the first category is the now familiar dashpot, or the secondary servo droop type as seen in Fig. 4. The dashpot consists of an actuating piston, linked to the governor servomotor, and a receiving piston linked to the speed sensing element. The two pistons are connected hydraulically. The entrapped fluid in the dashpot is connected to a sump through the needle valve, thus movements of the servomotor are transmitted to the speed sensing mechanism to cause droop which is allowed to bleed off through the needle valve in an exponential manner. This permits the governor servomotor to take a new position as required for the new engine load level without a corresponding permanent speed change.

The second category is the pressure compensated type. This type is similar to the dashpot governor but differs in that the servomotor position is converted to a pressure differential applied to the speed sensing element to cause droop. The pressure differential is allowed to dissipate through the needle valve. Fig. 10 is a schematic of this type governing system which has been in regular commercial production since 1947 and certainly is an advancement over the dashpot governor chosen by the author to represent the current state of the art.

Third on the list is what may be termed the power compensation scheme. Here a secondary pilot valve is displaced by the droop linkage, energizing a servo which is used to alter the fulcrum point of the droop linkage in such a way that the governor servomotor is forced to move an additional amount until the engine speed returns to the isochronous value. This scheme is similar to the subject of Professor Oldenburger's paper but differs in that the fuel is not directly under control of the secondary servomotor. Patents have been issued to Standerwick and to Crafts, et al., covering the power compensation scheme [13, 14].

The fourth type of governing scheme has been termed the "feed forward" governor and is basically a power compensation scheme except that the secondary servomotor is in direct control of the fuel. The author's governing system, Fig. 7, is of this type. Standerwick's patent, issued June, 1951, shows a scheme where two servomotors are connected to the fuel metering device; one servomotor is the final element of a droop governor and the other is under control of a secondary pilot valve and is used to reset the fuel such that isochronous operation is obtained [15]. The Muzzey patent, issued July, 1951, shows a method whereby a...
single servomotor is connected to the fuel metering device [16]. This servomotor is determined by the sum of the flow of fluid from a secondary servomotor and a pilot valve operated by this secondary servomotor. Thus, Muzzey established the principle of hydraulic addition shown in Fig. 6. Functionally the Muzzey scheme is the same as Fig. 7, and hence the basic principles of the "new" governor are known and have been previously applied.

The authors have placed serious restrictions upon the dashpot governor which do not seem to be justified. The differential equation (1) does represent with good accuracy the dynamic characteristics of the dashpot governor manufactured by the Woodward Governor Company since 1911 for diesel engines and for hydraulic turbines. Common to various governors of this type is the adjustment known as compensation or temporary droop, which is evident in (10) by the constant \( c_2 \). This parameter is continuously adjustable externally and in effect moves the pivot point of the lever linking the final servomotor of Fig. 4 to the actuating dashpot piston having an area \( a_a \). This adjustment, along with the needle valve adjustment, \( c_4 \), permits greater freedom than Professor Oldenburger has allowed for the dashpot governor.

Parameters may be selected for a commercially available dashpot governor such that it will behave according to the equation:

\[
Z' = \frac{-0.34(D + 60c_2)N}{D + 20.8c_4 + 60c_2} \tag{25}
\]

Here

\[
T_s = \frac{1}{60c_4} \tag{26}
\]

\[
T_e = \frac{1}{20.8c_4 + 60c_2} \tag{27}
\]

With

\[
0.3 < c_2 < 1.0
\]

\[
0.003 < c_4 < 0.2
\]

The constant, \( c_2 \), known as the compensation, can take on any of the values in the range given by external adjustment. Similarly, \( c_4 \), an external adjustment, can be adjusted for any value in the range given. For an engine with an acceleration rate of 100 percent per second we set \( c_2 = 1.0 \) and \( c_4 = 0.07 \). For the engine with an acceleration rate of 10 percent per second we set \( c_2 = 0.31 \) and \( c_4 = 0.022 \) and predict the transients of Fig. 11 for full load rejection. These transients have a shorter duration than those shown in Fig. 8 and are certainly superior to what is considered optimum by the author for his "new" governor. The parameters chosen for the dashpot governor equation are for a standard production type. The author has apparently selected constants which will make the dashpot type appear to perform poorly. Further, why were not \( K_g \), \( T_a \), and \( T_p \) chosen by the author for his governor so as to yield transient performance as shown in Fig. 11? These constants are certainly within the range given by equation (11) and the frequency response curves of Fig. 9(b). To verify the theoretically computed transients of Fig. 11 a production governor was mounted on a diesel engine whose full load rejection acceleration rate is 55 percent per second with a rated speed of 1200 rpm. The differential equation describing this governor is written:

\[
Z' = \frac{-0.22(D + 40c_2)N}{D + 37c_2 + 40c_4} \tag{28}
\]

This equation combined with an engine equation (19) was used by the author and after proper selection of the servomotor stroke we compute the transient of Fig. 12. The actual engine transients for 100 percent load rejection, Fig. 13, show a speed error of 5.5 percent. The difference, 0.7 percent between the predicted peak of 4.8 percent, is attributed to the engine dead time of 0.013 seconds between the movement of the governor servomotor and the engine developed torque. This dead time is well defined and it is customary to take into account reference [17]. Optimum values of governor parameters may be quite simply determined while considering both the engine acceleration rate and the inherent dead time of internal combustion engines.

It is clear from the foregoing that the range of adjustment for the compensation, \( c_6 \), and of the needle valve, \( c_4 \), is adequate to handle engines whose acceleration rates vary from 10 percent per second to 100 percent per second. In fact this type of governor has controlled engines with acceleration rates as low as 3.0 percent per second and some with rates up to 1500 percent per second all with commercially available off the shelf parts.

Implied in Professor Oldenburger's paper is that the mechanical-hydraulic dashpot type of governor represents the latest state of the governor art prior to the proportional plus integral control described by him. This implication is unwarranted in view of the pressure compensated governor described earlier and the electric-
hydraulic governors, capable of performance far superior to that of any mechanical-hydraulic governors mentioned in the paper [18, 19, 20]. Electric governors have been in production for a number of years. Specifications of the Corps of Engineers, U.S. Army, for diesel engine generator sets require performance which cannot be met with a governor of the type advocated by the author. These specifications state, "For minimum regulation (isochronous) setting of the governor and for sudden increase or decrease in load up to and including rated load, the governing system shall re-establish stable engine operating conditions within one second. The maximum transient frequency change above or below the new steady-state frequency shall not be more than 11/2 percent of rated frequency [21]." Fig. 14 shows recorded performance of a typical production electric governor on the same engine used for tests recorded for Fig. 13. Fig. 14 was obtained without load sensing applied. Speed and load sensing governors have been developed successfully and have done a great deal to improve performance where required [22]. Electric governors have been found to be superior dynamically as well as having a resolution equal to or better than the standard flyweight governor.

Another misconception the author has is that he believes noise makes it necessary to adjust the needle valve, c4, thereby altering T4, to obtain required filtering. He further states that, "The lag and lead constants T9 and T4 go up together largely nullifying the increase in T9." Immediately following equation (24) the statement, "It follows that varying the needle valve, i.e., c4 does not change the lag T9, the product K9T9 but does affect the governor gain K9." These two statements are inconsistent and deserve explanation. It can be seen from equation (26) that, with c9 = 1.0 and c1 = 0.07, T9 will have a value of 0.04. Should c4 be halved, T9 only changes to 0.0436 while T4 is doubled. Therefore, the needle valve influence upon T4 is small, making it possible to alter the lead time constant. The lag time constant T9, is changed, independent of T4, through the compensation adjustment c9.

Two severe criticisms made by the author of the dashpot and pressure compensated governors are that they are subject to sticking of the receiving piston or buffer piston and the clogging of the needle valve. The dashpot is designed such that flow through the needle valve is in both directions, making it self-cleaning. A clogged needle valve has never been experienced by the Woodward Governor Company in over a half million governors. Likewise, sticking of the receiving piston has not been experienced with modern dashpot governors.

Professor Oldenburger has suggested that his "new" governor is based upon different mathematical principles from those used to analyze the dashpot governor. He also stated that pneumatic controllers are based upon completely different mathematical principles. Would the author explain what these principles encompass?

Examination of the frequency response curves of Fig. 9 indicate that the author's governor has constants with the following range:

\[ 0.0085 \leq K4 \leq 0.356 \]
\[ 0.068 \leq T4 \leq 88.0 \]
\[ 0.019 \leq T9 \leq 0.9 \]

It does not seem practical to obtain a range of 1120 to 1 for the constant T9 with a mechanical linkage as shown in Fig. 7. Furthermore, it is not practical to drive linkage of this complexity from such a low power device as the ballhead. Friction at this point will introduce considerable deadband to the governor. It very likely is necessary to insert a power amplifier between the flyweights and the mechanical linkage to minimize the friction effects. A study would have to be made of the detrimental effects of such an amplifier as has been pointed out by the author in connection with the "acceleration" governor.

Four conclusions made in this discussion are worth emphasizing. The first is that the principle of the "feed forward" governor and of hydraulic addition date back to Standerwick, Muzzey, and Rexford patents. Therefore it is believed that the governing system presented by Professor Oldenburger is not novel. The second point is that the author has omitted the compensation adjustment common to dashpot governors which does enable a single governor to be suitable for 10 percent per second and 100 percent per second engines and yield better transients than shown by the author in Fig. 8. This suggests an error in what the author considers optimum. Third, the arrangement shown in Fig. 7 is not considered practical since reasonable flyweights are incapable of delivering sufficient power to drive linkage requiring such large range of adjustment without sacrificing accuracy of speed error measurement. Finally, the "new" governor advanced by the author is not capable of performing in accordance with the more stringent transient specification written for diesel engine generator sets.
of prime movers where production of any given type or model is quite limited and applications vary, the need is great for a single type of control capable, through simple adjustment, of providing stable high performance operation for several prime mover types and applications. The author's approach through his "new governor" provides simple and independent adjustments of governor gain, lag, and lead time constants. It is indeed a clever solution.

I hope the author can, as he has with several previous papers, supplement his formal paper with a report of his actual experience with operating hardware. In discussing this paper with my colleagues, the following specific questions were raised:

(a) What ratio of diameters of rate servomotor, $a$, to main servo, $c$, have been found practical?

(b) Have relative rotation pilot valves been found necessary?

(c) Have similar or identical values of pilot valve $k_p$ and rate valve $K_r$ been found practical?

(d) Have leakage problems been found critical in the rate servomotor?

The author shows, Fig. 8(a, b), that the "new governor," Fig. 7, is superior in performance to the "dashpot governor," Fig. 4, when both have "optimum adjustment." The "new governor" is demonstrated to have the same basic equation (1) as the dashpot governor. Granting the desirability of independent adjustments if adjustments are required, will the author discuss differences in performance that might be expected if both types of control had optimum constants selected for a single specific application.

Reference [8], by the author, describes a dashpot governor, and in reference [8], Fig. 3 shows frequency response curves that up to 10 cps are stated as a simple matter to modify by changing constants $c_1$ and $k_1$. In the present paper, Fig. 9(a, b), couldn't similar changes in crossover be established by changing $c_1$ and $k_1$ of the dashpot governor Fig. 4 and thereby make Fig. 9(a, b) more nearly alike?

The author in his Introduction states greatly increased re-

 Additional References

L. J. Moulton*

The author presents in this paper an interesting variation to the basic type of proportional-plus-integral control, in this case, as an isochronous prime-mover governor. The discussor has had experience with successful application of several variations of this basic type of control. These systems deliver a high level of isochronous control performance that is quite insensitive to temperature. Mechanization has proved to have a number of desirable features as the author points out.

If one considers failures which could cause uncontrolled application of maximum input energy to the prime mover, the "new governor," Fig. 7, with the two pilot valves appears less reliable than the "dashpot governor," Fig. 4, with only one pilot valve. The Fig. 4 valve has the added advantage of flyweights acting to break its valve loose in the event it sticks. Fig. 7 pilot valves are shown with follower springs. Compared with the reference [11] governor with two pilot valves, little difference is apparent.

If field problems are used as a measure of reliability, the discussor's experience indicates adjustments and particularly moveable pivot or fulcrum adjustments as major sources of problems. The "new governor," Fig. 7, appears to have twice as many adjustments as the dashpot governor, Fig. 4, and many more pivots than Fig. 4.

If the author "by greatly increased reliability" means there is no dashpot with its floating piston, certainly this is true. However, a study of problem history covering the many hours of dashpot governor operation may not show this area to be the major source of field troubles. It is agreed that from a design and manufacturing viewpoint floating pistons and needle valves require a great deal of attention.

In conclusion, may I express my appreciation for the opportunity to review this well organized and written paper which builds logically on past papers by the author. It has been interesting and exciting to find such a simple straightforward solution. One can't help saying: "Why didn't I try to do it that way?"

Author's Closure

Dr. A. F. D'Souza's points with regard to rotational speed measurement are well taken and show real insight into the problems involved. The voltage output of a generator with a fixed load varies linearly with speed. For the normal governing of prime movers it would be better if the centrifugal force of the flyweights also varied linearly with speed instead of as the square. Even so there are mechanical governors in successful operation over a 12:1 speed range. Since the governor gain is then very small at the minimum prime mover speed the response is slow, but may still be stable. In the case of overspeed trips, i.e., devices that will shut the engine down when excessive speed is attained, the fact that the centrifugal force varies as the square of the speed is a real advantage in giving sharp cutoff. The ballhead scale $k_b$ is proportional to the square of the speed. When $k_b$ is large enough the net scale $k_b$ becomes zero, resulting in infinite ballhead gain and a snap action of the ballhead to cut off fuel.

Professor D'Souza states that a ballhead is subject to inertia, friction, and backlash. These factors are present but good modern design has reduced their effects to remarkably small proportions. Backlash is negligible in current hydraulic governors, in fact in properly designed units it is not detectable. Some friction is present, but this is needed to damp the ballhead. If there were no friction the ballhead would oscillate as a mass-spring system. The mass of the flyweights is required to generate the ballhead force. Inertia is associated with this mass. However, the flyweights are spring loaded. With a sufficiently large spring scale the natural frequency of the ballhead can be made arbitrarily high. In practice this natural frequency is so high, that the ballhead lag is negligible insofar as it affects governor-system transients. On the other hand, it is usually chosen low enough so that excessive "noise" in the governor drive, corrupting the speed signal, is usually filtered at the ballhead. A remarkable part of the ballhead is the element used to set the reference speed. This is the speeder spring. Under constant ambient conditions this mechanical component, when compressed through a given dis-
tance to attain a desired loading force, opposing the centrifugal force of the flyweights, maintains this force constant to very great accuracy. It is the ability of the spring to hold a constant reference that makes the ballhead remarkable. Tests have been made on ballheads in which the ballhead speed was changed by 0.002 percent with the ballhead output varying accordingly. This same ballhead yields a usable corresponding output for a speed change range of at least 0.002 to 10 percent or more, depending on the location of mechanical stops on the output. Thus a good ballhead is capable of giving an output motion as a function of speed error over at least a 500:1 range. The nonlinearity of the ballhead output can be partially compensated for by the use of a nonlinear spring where the scale of the spring increases as the spring is compressed. What is also remarkable about the ballhead is that relatively large amounts of power may be drawn from it without deteriorating the measurement. Thus a ballhead combines the advantages of resolution, sensitivity, range, and power.

A major weakness in the measurement of speed by electrical means lies in the element used to set the reference speed. Instead of the ballhead speeder spring producing a constant fixed reference force one must have an electrical equivalent. This must yield a fixed reference voltage (or current). Such a voltage can be obtained from a source provided that the load on the source is very small and constant. Each speed measuring element has a lag. Where no power need be extracted from the element and only a measurement of speed is required the electrical measuring element is dynamically linear and mechanical in that the lag can be made much less. For governor applications it is desirable to draw some power from the electrical speed measuring element. Otherwise substantial power amplification must follow this element to make the measurement usable. A power supply is needed where the power is drawn from a line. Line voltages fluctuate continuously. To generate a constant voltage, electrical elements must be used to change the varying line voltage into a sufficiently constant reference. To attain by electrical means the constancy and power available in the ballhead is costly at the present time. Tuned bridge circuits are employed to obtain a voltage proportional to speed. These are only effective over a small range, such as ± 5 percent of rated speed. The ballhead is not subject to this limitation. Present direct current generators create excessive noise in the Alternating current generator is in common use on electrical governors. The outputs are rectified and filtered to obtain a d-c voltage proportional to speed. Unless the generator frequency is very high the useful speed range of such generators is limited since at low speed the filters are no longer effective. Such generators must be relatively large if substantial amounts of power are to be drawn from them. Electrical measurement of speed for governors is practical and yields satisfactory performance over a wide range of applications. However, by comparison the mechanical ballhead has advantages.

Mr. Leum's discussion is essentially an effort to defend the dashpot governor and point out similarities between the author's new governor and governors invented by others, covered in the patent literature. The dashpot isochronous governor is an excellent product. It is relatively inexpensive and compact, has few parts, is reliable, and may be tailored to yield optimum performance for both slow and fast engines. It has been around for a long time and has been brought to a very high stage of development. Mr. Leum classified isochronous governors into four categories. The author does not agree with this classification since the governors in his second category belong in his first. His first category comprises the dashpot type governors. One of the many versions is shown in the author's Fig. 4. This example was chosen because the author found it the simplest for purposes of exposition. Mr. Leum's second category comprises the pressure compensated type governor shown in his Fig. 10. The governor of this figure should be included in his first category; it is a dashpot governor. The servo piston serves simultaneously as the transmitting piston. The buffer piston of his Fig. 10 is the receiving or proportioner piston of the author's Fig. 4, with an area a. Both governors have a dashpot needle valve as shown. Instead of bringing a mechanical feedback from the buffer piston to the ballhead as in Fig. 4, dashpot pressure is brought to the ballhead in Mr. Leum's Fig. 10. This pressure is proportional to the buffer piston position. Thus pressure (or force) feedback is equivalent to position feedback. Reference was made in the paper to a dashpot feedback. The governor of his Fig. 10 is a version of this type.

Marquette and other governor manufacturers went to force feedback because of sticking of the receiving or buffer piston. Friction at a floating piston cannot be eliminated completely. As a result the receiving piston can settle in equilibrium in different positions even though the dashpot pressure p becomes zero. Since the pilot valve is closed in equilibrium this forces the flyweights to take different equilibrium positions. The governor is thus not isochronous, but will settle at various equilibrium speeds depending on the position taken by the receiving piston. On the other hand, the dashpot governors with force feedback come to the same equilibrium speed regardless of the position of the receiving piston. The oil bleeds through the needle valve in equilibrium to relieve the dashpot pressure. Since this pressure is the variable feedback from the dashpot to the ballhead, removal of it makes the governor in equilibrium speed independent of the receiving piston.

Mr. Leum's remaining two categories may be thought of as forming a class of nondashpot governors. He states that governors of this class, prior to the new governor of the author, involve two servos and the same principles as the author's new governor. Mr. Leum ignores the main purpose and unique features of the new governor. The configuration of the new governor of the author's Fig. 7 was developed to solve the problem of the direct and independent adjustability of the three governor parameters, namely, gain k, lead Td, and lag Tr. The governors cited in Mr. Leum's patent references do not have this capability. A major difference making this possible is that the pilot and rate valves in the new governor are in parallel rather than in series. Although nondashpot isochronous governors with two servos have been employed before it required a certain combination of components to yield a unit with the independent adjustability of parameters and the correct ranges of these parameters. Apparently, the mechanical and hydraulic circuitry needed is unique. One reason for the built-in adjustability of the new governor is to cut down on the field service now required to match a governor to an engine.

The patents referred to by Mr. Leum are by no means equivalent to the author's new governor. As he points out, in the first Standerwick patent cited the secondary servo is in a feedback path from the main servo. The secondary valve is in series with the first. The second Standerwick patent he mentions is a three servo governor. The last and main servo is used for power amplification only. The first two servos are not the result of adding droop and isochronous governors as in the new governor. The ballhead strokes the pilot valve spool. This meter's oil to a first servo. The servo mechanically positions the relay valve which in turn meters oil to the main servo. This servo also positions a bushing on the pilot valve through a feedback linkage. It simultaneously strokes a secondary valve spool which meters oil to the second servo. Thus the pilot and secondary valves are in series. The second servo also mechanically positions the pilot valve bushing providing a second feedback path to the pilot valve. Thus this second Standerwick governor also has the pilot and secondary valves in series, with feedback loops from both first and second servos to the pilot valve. The motions of the first and second servo pistons are added to stroke the relay valve spool.

The Crafts patent cited by Mr. Leum covers a governor where the ballhead output motion is first hydraulically amplified. This is followed by a pilot valve which meters oil to the main servo. The main servo strokes the secondary valve spool, which is again in series with the pilot valve. The secondary valve meters oil to
the secondary servo. This servo, however, is not connected to the throttle or rack, but through mechanical feedback strokes the secondary valve spool.

The Muzzey patent referred to by Mr. Leum involves hydraulic addition similar to that shown in the author's Fig. 6. However, it has a sleeve at the pilot valve for feedback from the secondary servo to this valve. The pilot valve meters oil to the secondary servo. The secondary valve spool is stroked by the secondary servo, whence the pilot and secondary valves are in series and not parallel. The secondary valve meters oil to the main servo. In his conclusions, Mr. Leum also mentions a patent by Rexford [23]. This covers a nonlinear dashpot governor.

The dashpot character may not be immediately apparent since there is no needle valve. This unit belongs in Mr. Leum's first category rather than the third or fourth, where he has placed it. Here the dashpot needle valve is replaced by a variable orifice. The main servo piston is at the same time the transmitting piston of the dashpot. The secondary piston is in the same cylinder as the main piston, but this secondary piston is spring loaded, and is merely the receiving piston of the dashpot. The area of the variable orifice is mechanically changed by the position of the secondary servo piston.

Several other nondashpot governors with at least two valves and two servos have been patented [24–27]. The author derived and studied the equations and transfer functions of all of the governors cited. He concluded that it was not feasible to choose the physical dimensions and other properties of these governors so as to obtain a governor with the transfer function (1) where the three governor parameters can be adjusted directly and independently with the ranges (11) as needed in practice.

Mr. Leum points out that there is another constant that may be adjusted to vary the governor parameters. This is the ratio \( c_2 \) of transmitting piston displacement to servo piston displacement. It is important that the author stated in his text that the needle valve parameter \( c_4 \) is the only quantity that is "readily" adjustable in dashpot governors. In addition to \( c_2 \), the quantity \( c_1 \) is adjustable on Marquette and other governors. However, as pointed out by Mr. Leum the range of adjustment (27) of \( c_2 \) is small, namely 0.3 to 1. Generally, varying \( c_2 \) over this range has a minor effect on performance, and this performance will not deviate much from that obtained for the mean value of \( c_2 \), namely, \( c_2 = 0.65 \). Reduction of \( c_2 \) tends to make system response more oscillatory. In the usual application, system performance will be best if \( c_2 \) is chosen at the maximum value \( c_2 = 1 \). However, there is an area where reduction of \( c_2 \) helps. If an engine has a lag of about 0.1 sec and maximum acceleration of about 10 percent per sec a governor of type (25) will yield better response with \( c_2 = 0.3 \) than \( c_2 = 1 \). For this special case, adjustment of \( c_2 \) is helpful. The range of values of \( c_2 \) is dictated by the following considerations. If \( c_2 \) is large the transmitting piston will place an excessive load on the servo, whereas if \( c_2 \) is small the response will be too oscillatory. The reader will note that in the force feedback dashpot governor shown in Mr. Leum's Fig. 10 there is no separate transmitting piston, and hence no adjustment \( c_2 \). In this governor the needle valve coefficient \( c_4 \) is the only adjustment available for varying governor parameters. Where governor constants are adjustable a range for each constant of about 40:1 is usually desirable. For \( c_2 \) it is only 3.3:1.

We remark that when \( c_4 \) is small, as is usually the case, the \( c_4 \) term may be dropped from the denominators of \( K_t \) and \( T_e \) in equations (10). The author's dashpot governor equation (23) was derived with this simplification. The author and associates have found from extensive industry experience that the dashpot governor equation (23) applies to the standard off the shelf unit in general use. To show the difference in performance between a dashpot governor and the new governor the author could have combined the governor parameters over continuous ranges to tune a governor to a given engine. Where a governor is not tuned to a given engine poorer control is obtained with the resultant waste of fuel consumed in constantly accelerating and decelerating cycles of the engine.

The major factor limiting governor-engine performance is the lag in the engine and load. In first approximations this lag may usually be treated as first order with a time constant \( T_e \). Thus for sudden load changes equation (19) should be replaced by the equation

\[
N' = \frac{K_e}{T_e D + 1}
\]

If great accuracy is needed this must be further complicated, but experience shows that the equation generally suffices. Combining equations (1) and (29) yields the differential equation

\[
N'''' + \left( \frac{1}{T_e} + \frac{1}{T_e} \right) N'''' + \frac{K_e K_d T_e}{T_e T_e} N'' + \frac{K_e K_d}{T_e T_e} N = 0
\]

In first approximation studies one may lump the lags \( T_e \) and \( T_e \) and replace equation (30) by

\[
N'''' + \frac{K_e K_d T_e}{T} N'' + \frac{K_e K_d}{T} N = 0
\]

The coefficient \( T^{-1} \) of \( N'''' \) in equation (31) is minus the sum of the characteristic roots of the equation. Optimum or nearly optimum performance is obtained by dividing this sum equally among the roots so that roots of the form

\[
\text{Numbers in brackets designate Additional References.}
\]
are obtained where
\[ \nu = \frac{1}{3T} \]  
\[ j = \sqrt{-1} \]  
(34)

The solution of equation (31) for a step change in load is then of the form
\[ N = e^{\frac{1}{3T} (C_1 + C_2 \cos \nu t + C_3 \sin \nu t)} \]  
(35)

for constants \( C_1, C_2, \) and \( C_3. \) The values of these constants depend on the amount of the load change. From experience it is known that the transient (35) appears to the observer to have died out in about 14\( T \) sec. Cases where \( T = 1/4 \) occur often. Using this as an example the duration of the transient is about 1/4 sec. This time will occur no matter how the total lag is divided between the governor and the controlled system.

For the GMC-671 engine of Mr. Leum's figures the equation of the engine is
\[ N' = \frac{630Z}{e^{T_0d}} \]  
(36)

when the load is zero. The no load to full load servo stroke \( Z_0 \) is taken to be 1.0 in. Here the dead time \( T_0 \) varies in the range
\[ \frac{1}{80} \leq T_0 \leq \frac{1}{50} \]  
(37)

depending on when the rack is moved relative to the firing cycle of the engine. In rough approximation studies \( e^{T_0d} \) may be replaced by \( (T_0D - 1) \), so that equation (36) takes on the form of equation (29). When the engine is driving a self-excited generator supplying a load the equation
\[ N = \frac{630 Z}{e^{T_0d}(D + \delta)} \left( \frac{D}{11} + 1 \right) \]  
(38)

applies, where typically \( \delta = 1.2. \) Here \( \delta \) corresponds to the windage and friction associated with the rotor and connected parts and to electrical damping of the load. The dead time \( T_0 \) is the same as for equation (36). The first order lag with time constant 1/11 sec is due to the presence of the load. In equation (38) there is actually an expression \( (D + \delta) \) as for equation (38), but for zero load the quantity \( \delta \) is negligible. In first approximation studies the \( \delta \)-term in equation (38) may also be neglected. The \( T_0 \) lag may be lumped with the 1/11 sec time constant to yield a lumped time constant of about 1/9 sec. Thus equation (38) becomes equation (29) with \( K_1 = 630, \ T_0 = 1/9, \) valid when the generator supplies a load.

In the experience of the author and his associates the engine-load lag has varied from 0.01 to 1 sec. A slow engine is by definition one with a large lag and low acceleration rate. For the slow engine of Fig. 8(a) the author took a case where the maximum prime mover acceleration is 10 percent/sec and the total system lag \( T \) is 1 sec. In deriving his curves Mr. Leum neglected the engine-load lag. For the fast engine of Fig. 8(b) the author took an engine with 100 percent maximum acceleration and a total system lag of 0.1 sec. The curves of Figs. 8(a and b) are for partial step load changes, such as from 100 to 0 percent load, where the complete system is linear and one does not run into discontinuities. Better curves are obtained for full load reductions where discontinuities come into the picture. The curves of Figs. 11 and 12 are not nearly as good when engine-load lag is included in the calculations.

In practice the governor time constant \( T_g \) should be chosen as small as possible. If it is too small there may be excessive jiggles, not sufficiently filtered by the ballhead, at the servo connected to the rack, throttle, or equivalent element of the prime mover. Thus in properly matching a governor to an installation the time constant \( T_g \) should be chosen so as to yield an acceptable jiggles at the servo piston. Sometimes the noise in the speed signal is so small filtering is not required. Experience shows that the acceptable jiggles is about 1 percent of the no-load to full load stroke at rated speed. Governor drives should be chosen for minimum jiggles. For normal jiggles frequencies, doubling the governor time constant cuts the jiggles amplitude in half. Since increasing \( T_g \) to cut jiggles tends to deteriorate governor-engine response, a compromise between jiggles and system performance must be made. Similarly, ballhead lag should be kept to a minimum. Proper matching of the engine to the installation requires that the governor gain \( K_1 \) and load time \( T_g \) be selected to give best transient response with \( T_g \) as small as possible, and jiggles acceptable. This involves minimum speed error and transient duration without excessive oscillation, i.e., roots of the form (33) and (34) if possible. With \( T \) and hence \( T \) in equation (31) fixed, the roots (33) with
\[ \frac{1}{3T} \leq \nu \leq \frac{2}{3T} \]  
(35)
results in definite and observable improvement in transient response. Thus the load and speed sensing governor is a unit for special application and is outside of the class of speed governors discussed in the paper, which may be used with good and economically justified results on any installation. The U. S. Army Corps of Engineers deserves much credit for stimulating the development of speed and load sensing governors for the class of fast diesel applications where their use is indicated.

Mr. Leum’s Figs. 13 and 14 are not comparable to the author's Fig. 8. Figs. 13 and 14 are for full load changes. In this case the governor experiences built-in discontinuities to make the servo move with maximum speed to the no fuel position. As pointed out above, Fig. 8 applies to partial step load changes where nonlinearities, such as stops, are avoided. Fig. 15 shows the response of the same engine as that used for Mr. Leum’s Fig. 13 but with the Marquette L. E. governor. The servo of this governor moves to the no fuel position faster than the governor of Fig. 13 and has a smaller lag. In Fig. 14 Mr. Leum has carried the process even a step further by cutting the governor lag even more. The author has made such runs with fast hydraulic governors on the same engine with similar results. For such runs the rack jiggles high. That this should be so is easily seen from the theory. At 1200 rpm the speed signal drawn from the GM-671 engine shaft contains a noise component 0.5 sin 22°.

Mr. Leum states that the author is subject to a misconception in claiming that it is necessary to adjust \( c_1 \) and thereby alter the governor lag \( T_p \) to obtain required filtering of rack jiggles. The author disagrees. It is true that most filtering is done at the ball-head which behaves like a second order lag. In current governors this lag is not adjustable. Additional filtering may be required and accomplished by increasing \( T_p \). Since \( T_p \) corresponds to a first order lag this filtering is less effective than that of the ball-head. The author was correct in stating that as \( c_1 \) is decreased both governor lead \( T_p \) and lag \( T_d \) go up together. The effect is appreciable if \( c_1 \) is very large at the start. Where \( c_1 \) is initially small as is normally true in practice, the \( c_1 \)-term in the denominator of \( T_p \) is negligible, and further reduction of \( c_1 \) increases \( T_p \) very little. On the other hand, if \( c_1 \) is taken large to make \( T_p \) very small, as required for fast prime movers with small jiggles, then the \( c_1 \)-term in the denominator of \( T_p \) dominates so that \( T_p \) and \( T_d \) go down together, nullifying the effect desired, namely, to make \( T_p \) very small without altering \( T_e \).

The author pointed out above that the governor industry went from position to pressure feedback to avoid having sticking of the buffer or receiving piston affect steady state speed. Even where pressure feedback is employed, sticking of the buffer or receiving piston may occur. Tests and theory indicate that the force due to pressure on the piston must reach some magnitude \( e \) (for a positive \( e \)) before the piston starts moving from rest. The dashpot is a computer. When for small disturbances the force on the piston are too small and the piston does not move, the computation is not correct. When this occurs the character of the system response for very small disturbances will be different than for larger, and may be more oscillatory. With good design this effect is minimized, but there is always a chance that it will be noticeable. When the force on the piston is large enough it will move, and computations for which it is intended usually will be accomplished. The governor will regulate in any case, and in this sense sticking is a minor factor in governor performance. Although the new governor was not invented specifically to solve this sticking problem, avoidance of a floating piston is a clear asset. To the author’s knowledge no governor company has known experience with clogging of needle valves, but this could easily occur without having this specifically called to their attention. Since in practice needle valves are usually almost closed there is always a danger that even small particles of dirt will clog the valve in spite of the fact that flow occurs in both directions through the valve. Thus a good oil filter should be and is used in dashpot governors. It is likely that if a needle valve did clog up this might go unobserved because the governor would probably still regulate, but the transients would not be as good.

The mathematical principles of the new governor are explained in the paper in connection with formulas (18). The use of hydraulic addition to generate a linear combination of \( x_1 \) and \( x_2 \) for given variables, \( x_1 \) and \( x_2 \), is one of the keys to the success of the governor. On the other hand, the dashpot governor is based on the mathematical principle of obtaining a lead-lag unit by having a lag in a feedback path from the servo. The principles of pneumatic controllers are explained in reference [9]. The key mathematical principle there involved is that an integral plus proportional plus rate controller is produced by feeding back to the input a quantity of the form

\[
\frac{1}{T_1 D + 1} \frac{1}{T_2 D + 1} p_0
\]

where the times \( T_1 \) and \( T_2 \) are unequal and \( p_0 \) is the controller output pressure. The feedback operator is thus a difference of two first order lags. In the case of a dashpot governor there is only one such lag in the feedback path.

Mr. Leum concludes from the frequency response curves of Fig. 9 that \( T_e \) varies over a range of 1120:1, and that it is not feasible to drive a linkage of this complexity. Actually, \( T_e \) depends on \( p \), \( \alpha \), and \( \beta \) which are adjusted. These constants are varied over much smaller ranges, such as 40:1. The linkage system draws negligible power from the ballhead, and does not affect the accuracy of speed measurement.

The author’s governor is capable of meeting the stringent specifications of diesel-generator sets. The means by which the governor is converted into a speed and load sensing unit to meet these specifications are explained in the paper.

Mr. L. J. Moulton’s discussion is concise and to the point. His remarks, based on extensive experience, are well taken. Answers to his questions follow. A practical ratio of the diameter of the rate servo to that of the main servo is about 2:1. Relative rotation pilot valves have not been found necessary. Identical values of the pilot and rate valve flow coefficients \( k_2 \) and \( K_2 \) are practical. Leakage problems at the rate servomotor have not been critical. As noted above the proportions of the dashpot governor may be selected to yield the same optimum performance for a given installation as with the new governor. Mr. Moulton is correct when he surmises that the frequency response curves of Fig. 9(a) for a dashpot governor can be made more nearly like the curves of Fig. 9(b) if both the needle valve coefficient \( c_1 \) and the dashpot spring scale \( k_1 \) are changed. In practice one must open the governor to change \( k_1 \). Here one spring must be replaced by another.

In mentioning reliability the author referred to the matter of having system response agree with design. With the governor less sensitive to temperature changes, system response varies less. The author was particularly concerned with accurate functioning
of the computer part of the governor. Thus the needle valve “constant” $c_t$ may vary over a 10:1 or 20:1 range as the oil temperature goes from cold to hot. This changes the computation performed by the dashpot.

No trouble was experienced with the springs loading the valve spools of Fig. 7. These have a preload of at least one pound which is found to be adequate to prevent sticking. These springs keep the system tight at all times. In the actual governor the lever mechanism is simpler than shown in Fig. 7.

The new governor was developed at the Marquette Division of the Curtiss-Wright Corporation.

Additional References