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GOVERNOR CHARACTERISTICS FOR LARGE HYDRAULIC TURBINES

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Engineering and Research Center
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by

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February 1971

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CONTENTS

LIST OF FIGURES

APPENDIX

THE PROBLEM

In considering appropriate parameters for large hydropower generating units, of which those for Grand Coulee Third Powerplant are a prime example, control characteristics to satisfy power system needs strongly influence economics of the design. Parameters fundamental to control characteristics, such as the penstock water starting time and the mechanical inertia or flywheel effect, are subject to control by the designer but if abnormal values are required the increase is obtainable at appreciable incremental cost. To aid the designers with a basis for the most economical combination of the parameters to satisfy the requirements, an analysis of the requirements and their interrelation was undertaken. In this approach the starting point was a determination of the minimum or most economical control characteristics which could be accommodated by the power system into which the units are to operate. Following this, the governing characteristics were studied to determine a range of combinations of inertia and water starting time to yield the necessary control characteristics.

At this point the scope of the problem expanded considerably. When the relations for flywheel effect and water starting time were established, on the basis of a conventional temporary droop type of governor, to meet the required overall control characteristics, the amounts of flywheel effect and water starting time required were clearly excessive. Any workable combination of these parameters on this basis represented an excessive amount of added expense. Because of the important influence such large units would have on the power system the sacrifice of speed stability through bypassing of their governor dashpots to obtain workable responsiveness could not be considered acceptable. Relief to this constraint was found in special refinement of governing characteristics to better suit the requirements in this problem. Since this departed from conventional practice, the studies are recorded in this report.

SELECTION OF OVERALL CONTROL CHARACTERISTICS

Several important influences are placing stricter requirements upon the controllability of large hydropower generating units. Principal among these is the changing role of hydropower from baseload type of operation to daily peaking service. This is a result of integration of thermal and hydrosources through system interconnection. The large thermal sources require appreciable time to change boiler output. Efficiency is degraded by manipulation. The large steam turbines, for the benefit of slightly higher efficiency, develop a high percentage (roughly up to 80 percent) of their total output through the reheat cycle which introduces an appreciable delay, with a time constant in the order of 16 seconds, in this large component of their response. This yields somewhat sluggish governing characteristics. These factors combine to relegate the thermal sources, so far as practicable, to the role of baseload operation with a minimum of system regulating activity. Consequently, system needs are best served by exploiting the potential controllability of hydropower. This potential is high insofar as the energy of stored water is available with no more delay than that of opening the turbine gates and the associated time to accelerate the water column.

In more definitive terms, the controllability for such large hydropower units for daily peaking service should be such as to permit each unit to be loaded within 30 minutes while on regulation. Bypassing of the governing dashpot or its equivalent, as has often been done with smaller units to expedite response to the control signal, cannot be accepted without degradation of system performance. It has been adequately demonstrated by system experience in the Northwest, prior to 1964, that widespread bypassing of governor dashpots leads to instability of system speed regulation. Analyses of the tieline oscillation problem subsequent to that time have also shown that bypassing of governor dashpots would contribute substantially to the lower frequency tieline oscillations. It is therefore important that large hydropower units, such as the 600-mw units for the Coulee Third Powerplant which would strongly influence the power system, be capable of completely stable speed regulating capability under any operating condition. This is particularly important during the load pickup period of the day while generation schedules and loadings are changing rapidly.

A review of the control characteristics of the wide range of hydropower units of Bureau projects yielded the criteria that reasonable response to system control signals is represented by a response time constant of 30 seconds. Slow response is characterized by a response time constant of 50 seconds and a response time constant of 75 seconds is very sluggish. For the Coulee Third Powerplant units the response should be no slower than that of the existing units, in other words no more than 30 seconds. There is a corresponding need for sensitivity to speed deviations, rapidity of speed correction, and linearity of response maintained for small speed deviations. These characteristics will be discussed under "Refinement of Governor System" but they are sufficiently reflected in the gate response time constant for this to be a simple and useful criterion.

GOVERNOR TYPE AND CONTROL CHARACTERISTICS

Control characteristics of the conventional temporary droop-type governor may be obtained from the empirical formulae offered by Paynter[1], that is, for the optimum response:

$$\text{temporary droop, } \delta = \frac{T_w}{0.4\ T_m} = 2.5\frac{T_w}{T_m} \qquad (1)$$

and,

$$\text{recovery time, } T_r = \frac{T_w}{0.17} = 5.9\ T_w \qquad (2)$$

where

$$T_w = \text{water starting time} = \frac{\Sigma LV}{gh}$$

and

$$T_m = \text{mechanical starting time} = \frac{N^2WR^2}{1.6 \times hp \times 10^6}$$

The effective gate time constant T_g (actually a composite of two-time constants for this temporary droop type of governor) is approximated by the relation

$$T_g \cong \frac{\delta T_r}{\sigma} \qquad (3)$$

Substituting (1) and (2) into (3)

$$T_g \cong \frac{2.5\ T_w}{T_m} \times \frac{5.9\ T_w}{\sigma} \cong \frac{14.7}{\sigma}\ \frac{T_w^2}{T_m}$$

Supplying the usually employed numerical value of 0.05 for the permanent droop σ, the relation becomes

$$T_g \cong 300\ \frac{T_w^2}{T_m}$$

This relation, even though approximate, is nevertheless useful for preliminary appraisal of parameters. It identifies the relation between T_w and T_m for a given responsiveness and thus may facilitate preliminary design as will be illustrated later. Also, by inserting the preliminary values of T_w = 2 seconds and T_m = 8 seconds, a gate time constant, T_g = 150 seconds, is obtained for a temporary droop-type governor. This response would be exceedingly sluggish and unsatisfactory for regulating purposes. To meet the desired performance an excessive departure of T_w and T_m from normal would be required.

A more promising governing system for securing the desired response is a "double derivative" governor.[2] This system in preceding investigations had yielded a reduction of response time to about one-fourth that of the temporary droop type of governor. An approximate expression of response obtainable with it would thus be

$$T_g \cong 67\ \frac{TW^2}{T_m}$$

For the preliminary values of T_w = 2 and T_m = 8, T_g would be about 33 seconds. This was considered to yield satisfactory prospect for accomplishing the desired response, T_g = 30 seconds, by a minor adjustment of T_w or T_m either of which is within practical range for the designer. A plot of this relation between T_w and T_m for three degrees of responsiveness is shown in Figure 1. Although intended only as a preliminary guide it may be of interest to note that the final design data for the Grand Coulee Third Powerplant units meet the guide quite closely for the worst operating condition, full gate, at both the minimum and weighted average heads, as shown in the following table:

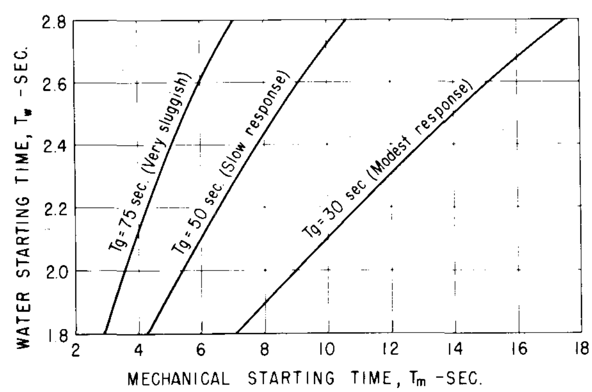


Figure 1. Relation of fundamental parameters T_w and T_m of hydrogenerating unit to governor control characteristics.

[1] Numbers refer to references at end of report.

Head	Gate	Horse-power	T_w	T_m
220 (minimum)	1.0	570,000	2.60 seconds	15.2 seconds
300 (average)	0.69	820,000 (rated)	1.83	10.5
302.5 (critical)	1.0	964,000 (maximum)	2.18	9.0
355 (maximum)	0.6	964,000	1.55	9.0

For usual operating conditions a slightly better degree of responsiveness can probably be safely utilized.

REFINEMENT OF GOVERNOR SYSTEM

General

Recognition of a need for refinement of speed governors for the hydraulic turbines of large power generating units had been brought about by a program of study and field tests for optimizing the adjustments of existing hydrogovernors. By helping to identify both the power systems' present needs for speed governing and the limitations of the heretofore conventional governing systems, that program of governor coordination provided a basis for the refinements to be discussed here.

The temporary droop type of governor has been so predominant for controlling the hydraulic turbines of power generating units in this country for the past several decades that it is used here as the basis for comparison and it is referred to as the "conventional" governor. A few electrohydraulic governors of various configurations and with improved characteristics are now beginning to appear.[3] However, they are still in the minority and none so far have incorporated all of the features this study has indicated to be both desirable and practical.

The temporary droop type of governor was a model of simplicity. It was well suited to the mechanical-hydraulic means of accomplishing its functions and it thereby established a predominantly satisfactory reputation of reliability which has served to excuse it from much critical study heretofore. Simultaneously, the results of growth and interconnection of power systems have tended to obscure the limitations of this type of governor in keeping up with the power systems present needs.

A decade or more ago when power systems were isolated and smaller, the band of speed deviations was wider and an appreciable part of the governor's capability was effective. The governor's sensitivity then produced reasonable activity to regulate the system speed. However, with the recent growth and interconnection of systems the band of speed deviations has been greatly reduced, a result of both statistical averaging of the random load variations and of the increased inertia of the interconnected system. Activity of the standard governors has reduced by an amount more than just in proportion to the speed deviations. Activity has subsided still further because of friction, backlash, and valve nonlinearity in amounts formerly unnoticeable but now an appreciable part of the narrower band of speed deviations. Thus the conventional governors now tend to be inactive without utilizing any perceptible part of their capability as speed governors to benefit the system. Some sort of speed governing function still results indirectly through the frequency bias action of the area load controls because of their presently higher sensitivity but this governing influence is rather crude because (1) the signal originates remote from the controlled units and merely reflects a sampling of the area speed, not necessarily representative of angular velocity, at the controlled generating unit, and (2) the intentional delays in this path, while quite appropriate and satisfactory for the area load control function, are too long to permit the most effective speed governing function.

Assessment of the current need for governing function as such is not complete until it is considered that the initial reduction of speed deviations which accompanies the interconnection of systems does not directly alleviate the need for governing. It tends mainly to transfer part of the evidences from local speed deviations to remote tieline load deviations. The imbalances of load and generation which were formerly reflected entirely by speed deviations, after extension of system interconnections become reflected partly by speed deviations and partly by deviations of load on the interconnections. The relatively wide deviations of load on the longer system interconnections is now a familiar phenomenon to system operators.[4]

Obviously the system control problem could be significantly relieved by not only (1) improving the governor's speed of response to the area load control signal as previously discussed, but by (2) improving the governor's sensitivity and speed of response to the local speed deviations so as to handle promptly and directly as much as possible of that component of the control function, thus freeing capability of the area load control for better handling of its share of control.

The respective shares of the control function between the speed governors and the area load control are established by the coordination of area frequency bias and speed droop. Hence, there is no conflict in restoring the governor's sensitivity to speed deviations to the same order as that of the area load control. Instead, the coordination is improved. However, this cooperative overlap in the hierarchy of system control should be accompanied by separation of the response times of the two layers of control. The governor should respond preferentially to speed deviation. Its response to the load control signal should not be so fast as to induce a speed deviation or to supersede correction of speed deviation.

Temporary Droop Governor

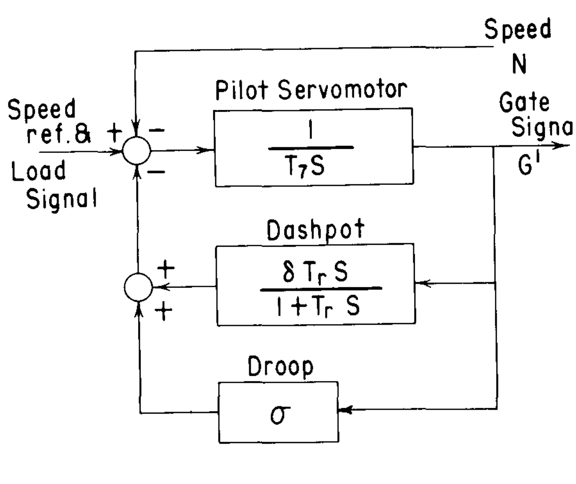
The temporary droop type of governor for hydraulic turbine control is shown schematically in Figure 2. It has the virtue of simplicity but that simplicity also limits its capability. As revealed by its transfer function, also shown in Figure 2, its dashpot provides only one term $(1 + T_r S)$, to compensate for the lags in the entire control loops, of which the water starting time is only one.

The servomotor system is not in reality a perfect integrator as implied by the simplified representation $1/T_7S$. It is in fact, besides an integrator, a cascade of delays including mainly two for the valve system and one for the main servomotor itself. These incidental delays may have as much degrading influence on overall performance as the water starting time.

An additional limitation of this form of governor is its inability to compensate for the incidental delays and nonlinearity of the servomotor system. With those delays in a simple cascade, performance of a very large unit such as for Grand Coulee Third Powerplant, would be exceedingly poor.

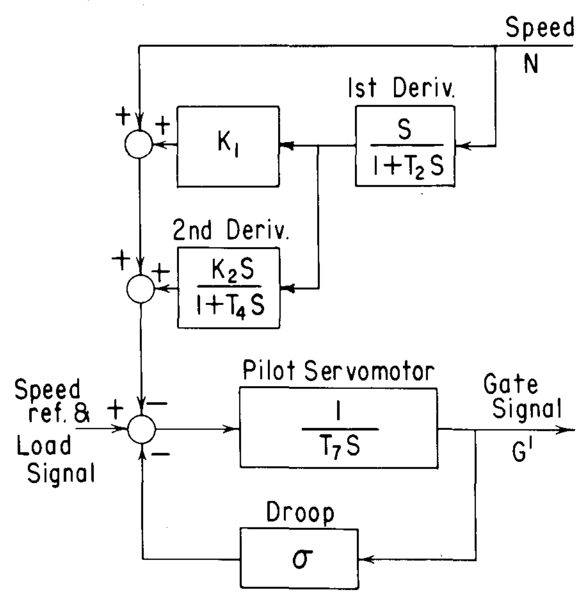
Double Derivative Governor

A superior form of governor developed in associated studies, particularly to accomplish better system load control without conflicting with stable speed control, is the "double derivative" governor shown in Figure 3. This is a special form of proportional-integral-derivative governor. Its superior performance is evident in Figure 4 (C and D) compared with performance of the temporary droop governor shown in Figure 4 (A and B).



$$\frac{\Delta G'}{\Delta N} = \frac{1+T_r S}{\sigma + (T_7 + \sigma T_r + \delta T_r) S + T_7 T_r S^2}$$

Figure 2. Block diagram and transfer function of temporary droop governor for hydraulic turbines. (See glossary for symbols.)



$$\frac{\Delta G'}{\Delta N} = \frac{1+(T_2+T_4+K_1)S+(T_2T_4+K_1T_4+K_2)S^2}{(1+T_2S)(1+T_4S)(\sigma+T_7S)}$$

Figure 3. Block diagram and transfer function of "double derivative" governor for hydraulic turbines.

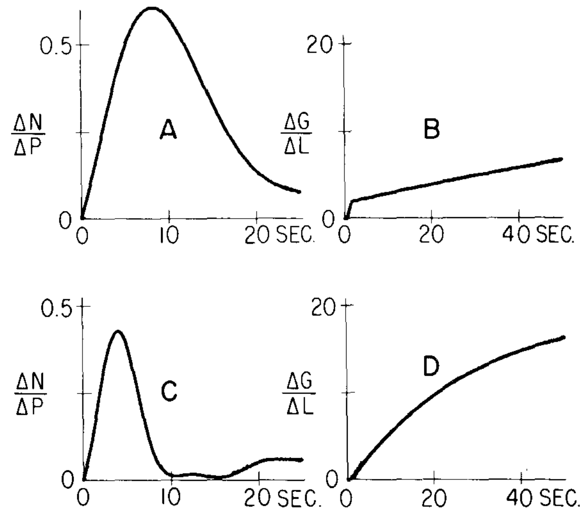


Figure 4. Performance of temporary droop governor of Figure 2 and derivative governor of Figure 3 with parameters for Grand Coulee Third Powerplant units (T_w = 1.83, T_m = 10.5, σ = 0.05).

A. Temporary droop governor: Speed transient after load increment, isolated operation (δ = 0.48, T_r = 15, T_7 = 0.1).

B. Temporary droop governor: Gate response to load signal, system operation without speed deviation (T_g = 147 seconds).

C. Derivative governor: Speed transient after load increment, isolated operation (K_1 = 5.2, K_2 = 7.4, T_7 = 1.5).

D. Derivative governor: Gate response to load signal, system operation without speed deviation (T_g = 30 seconds).

It is an interesting property of this and other high-order derivative governors to be described later that there is not just one optimum adjustment but a family of optimum adjustments. For each integration time chosen for the pilot servomotor there is a different optimum combination of derivative terms. Over a wide range of these optimum adjustments, control of the speed transient is essentially the same; however, movement of the turbine gates takes on different characteristics especially the response to a load control signal. It is this flexibility that makes this type of governor superior for reconciling the otherwise conflicting needs of (1) speed control without sacrifice of stability, and (2) rapid response to system load control without conflict with speed control. For illustration, performance of the governor with two widely differing sets of adjustment is shown in Figures 5A and 5B. In each case the speed deviation is limited to about the same magnitude but other characteristics are different.

In Figure 5A the servomotor integration time was quite short, 1.5 seconds. With this short integration time only modest amounts of the derivative terms are necessary for stability. Response of the gate system is lively with appreciable overtravel for a short period to quickly accelerate the inertia back to normal speed. Corresponding response of the gates to a load control signal is shown by the bottom curve in Figure 5A to be quite prompt, with a time constant of 30 seconds. This set of adjustments would be proper for normal interconnected power system where activity for speed control is modest but where good response to load control is needed.

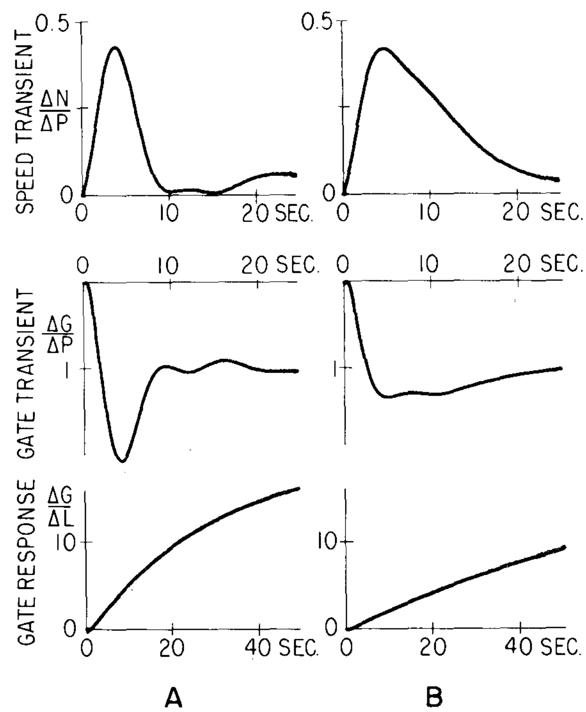


Figure 5. Performance of derivative governor of Figure 3 with short and long integration times.

A. Short integration time: T_7 = 1.5, K_1 = 5.3, K_2 = 7.4, σ = 0.05, $T_g = T_7/\sigma = 30$.

B. Long integration time: T_7 = 4, K_1 = 10, K_2 = 15, σ = 0.05, $T_g = T_7/\sigma = 80$.

In Figure 5B the servomotor integration time is much longer, 4 seconds. With this long integration time the derivatives must be much stronger to make the governor active enough for good stability. Response of the gate system is conservative. Note that overtravel of the gates is quite modest but this overtravel is sustained for a longer period while the inertia is gradually accelerated back to normal speed. The magnitude of speed deviation is not greater but the recovery takes longer. This set of adjustments was chosen to illustrate conservation of servomotor energy or activity. It would be applicable on a widely fluctuating load where there might be a need to minimize burden on the governor oil supply and where load control would not be needed. As shown by the bottom curve in Figure 5B, response to a load control signal would be exceedingly sluggish with a time constant of 80 seconds. Few applications exist anymore which would be benefited by the characteristics in Figure 5B but the comparison is revealing.

Advantages of this double derivative governor stem both from its higher order of compensation and from its particular form. The first derivative of speed is sufficient to accomplish the equivalent function of the temporary droop feedback in the more conventional governor, that is, each produces a single lead term of the form, $(1 + TS)$, in the numerator of the transfer function. This operates to compensate a dominant lag in the control loop, primarily that of the penstock water column. A second derivative of speed used in this configuration provides an additional lead term for compensating other lag in the control loop from the servosystem.

The tandem configuration for combining the speed signal, its first derivative and its second yields additional advantages, both technical and practical. This combination has flexibility such that the compensating terms may be complex, that is, the quadratic expression in the numerator of the transfer function may have complex roots. The significance of this is that still faster rise time of the control signal is possible so that performance can be approximately equivalent to third order compensation, but without the extra circuitry.

A practical advantage of this tandem configuration is that the derivative paths do not transmit any fundamental component. Therefore any drift of operational amplifiers used to perform these functions does not accumulate in a cascade product. Hence, drift is minimized.

Another peculiar flexiblity of this tandem arrangement is the ability to use separate speed sensors for the fundamental and for the derivative paths, if desired, to better utilize characteristics of the respective sensors. For the fundamental path, stability of the sensor is important as this controls the long-term stability of the governor, but response time for this path may be modest. For the derivative paths, a fast response signal is needed but since any long-term drift is not transmitted through these stages the drift characteristics of this sensor are not critical.

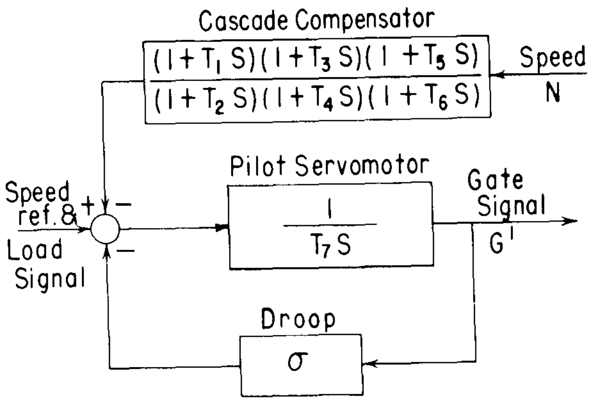
Finally, the derivative configuration allows freedom to insert the load control signal after the derivative stages. This retains for the speed control channel the compensation specifically tailored for best speed control but does not impose this compensation on the load control which should not be allowed to conflict.

Without this feature of detouring the governor compensation with the load control signal, conflict would come from the practice of proportioning the load control function by breaking up this signal into a series of intermittent, arbitrarily timed pulses to the governor speed level mechanisms. Worse still, it has been common practice to employ one such controller to generate control pulses for all units in a plant and sometimes more. Sampling rates for generation of the control pulses may range from 4 seconds to 5 minutes.[5] Augmentation of the fronts of these arbitrarily timed pulses by the speed control compensation not only accomplishes nothing beneficial for system control but actually contributes to random swings or system "noise." This influence is easily relieved by inserting the load control signal after the speed control compensation. This achieves smoother but nevertheless rapid control.

Equivalent High-Order Governors

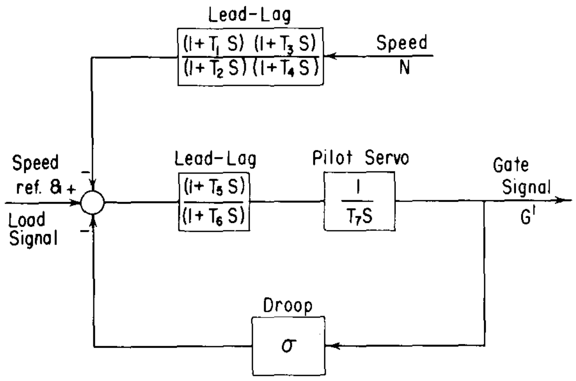
Extension of the study to identify some other governor configurations capable of the desired performance yielded the configuration shown in Figure 6. This governor employs a cascade of lead-lag stages for the compensation instead of the tandem path method of Figure 3. Three lead-lag stages are necessary to accomplish performance equivalent to the tandem path method of Figure 3 but technical performance is good and considerable flexibility of adjustment is available. This configuration would be confined to only one speed signal input for all functions but whether that constitutes a limitation depends upon the quality of speed sensor employed. The cascaded stages would tend to amplify any drift problem but this should be regarded merely as imposing stricter drift specifications within the individual stages.

While it might appear that only two lead-lag stages should be equivalent to the tandem path method with first and second derivatives, that is not the case. There is no cross coupling between successive lead-lag stages



$$\frac{\Delta G'}{\Delta N} = \frac{(1+T_1 S)(1+T_3 S)(1+T_5 S)}{(1+T_2 S)(1+T_4 S)(1+T_6 S)(\sigma+T_7 S)}$$

Figure 6. Block diagram and transfer function of cascade compensated governor for hydraulic turbines.



$$\frac{\Delta G'}{\Delta N} = \frac{(1+T_1 S)(1+T_3 S)(1+T_5 S)}{(1+T_2 S)(1+T_4 S)\left[\sigma+(\sigma T_5+T_7)S+T_6 T_7 S^2\right]}$$

Figure 7. Block diagram and transfer function for temporary droop governor upgraded with two lead-lag terms to yield performance equivalent to Figure 3 or 6.

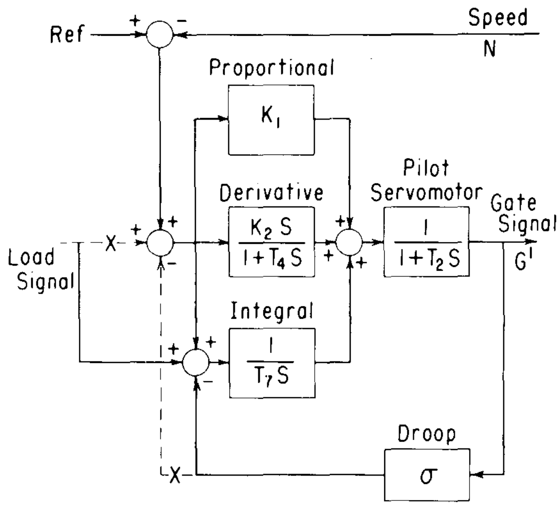
to generate the more rapid rise time reflected by complex roots in the tandem compensation. Only terms with real roots are produced by the lead-lag stages, consequently one more stage is required to generate equivalent performance in the signal shaping (electronic) circuitry.

The computer simulations show that by permitting underdamping of the succeeding servocontrol loop, seemingly equivalent performance would be obtained with only two lead-lag stages. However, this should not be accepted as truly equivalent. The servocontrol system actually contains many nonlinearities such as valve ports, friction in the gate mechanism, and compressibility of the unavoidably somewhat aerated servomotor oil system. It is subject to variables such as the oil pressure and hydraulic thrust upon the turbine gates at different gate openings. Thus, considering that the servocontrol loop is a high energy loop of massive moving parts with many nonlinearities and variables, it is clearly preferable that this loop of the overall control system be kept well damped and the required rise time be generated in the compensating network where computer-grade electronic circuitry can be employed.

Another governor configuration capable of performance equivalent to that of the tandem or "double derivative" system is shown in Figure 7. This is functionally a minor variation of Figure 6 in which one of the compensating terms is enclosed along with the pilot servomotor within the permanent droop feedback loop. The significance of this configuration is that it represents upgrading of the temporary droop-type governor system, with two additional lead-lag stages of compensation, to accomplish performance equivalent to the other high-order governor systems described. A temporary droop governor improved with one lead-lag function has been described by Schiott.[6] Although the block diagram of Figure 7 shows the third compensating term accomplished with a lead-lag, it could also be accomplished with the temporary droop feedback. Both methods reduce to the same mathematical expression.

This configuration somewhat reduces freedom to bypass the load control signal around the compensation designed for speed control. This confinement is minor if the last stage is used for one of the shorter time constants of the compensation. When this is done performance is essentially the same as Figure 4 (C and D) and it is therefore not shown separately.

A parallel configuration of proportional-integral-derivative governor, sometimes referred to as an industrial controller, is shown in Figure 8. This configuration of governor can yield the same transfer function as the "double derivative" governor of Figure 3 with certain modifications. For best performance the speed droop feedback should be returned to the integral stage only as shown by the solid line rather



$$\frac{\Delta G'}{\Delta N} = \frac{1 + (T_4 + K_1 T_7) S + (K_1 T_4 T_7 + K_2 T_7) S^2}{(1 + T_4 S)(\sigma + T_7 S + T_2 T_7 S^2)}$$

Figure 8. Parallel configuration of proportional-integral-derivative governor. Conventional arrangement is shown by dotted lines. Improved arrangement is shown by solid lines.

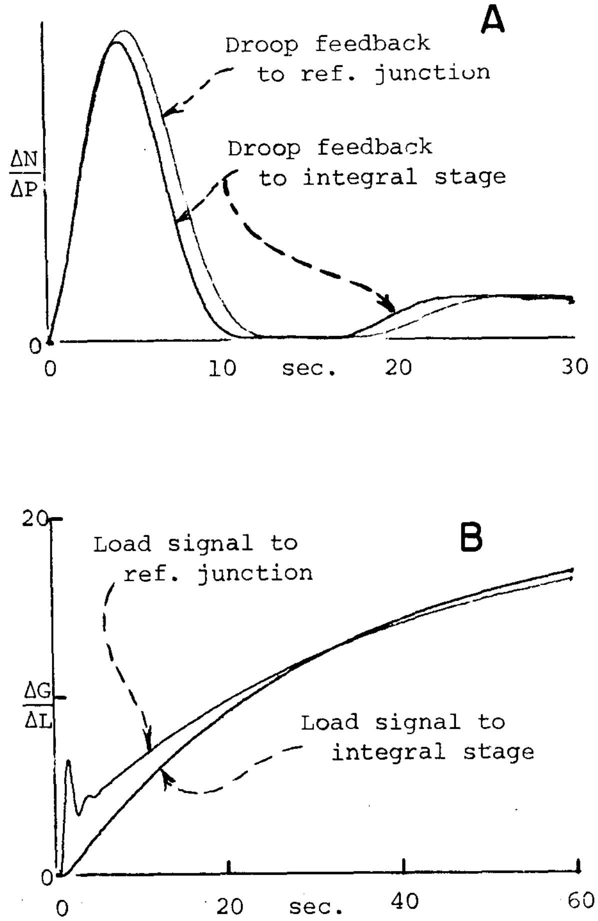


Figure 9. Performance of parallel configuration of proportional-integral-derivative governor shown in Figure 8 with conventional and improved arrangement.

A. Response of speed to load change.

B. Response of gates to load control signal.

than summed with the speed signal for all three terms as shown by the dotted line. The difference in performance is shown in Figure 9A. Similarly, the load control signal should also be delivered only to the integral term as shown by the solid line rather than to all three terms. This avoids initial augmentation of response to the load control signals through the proportional and derivative terms whose function is correct only for speed control. The difference in performance is shown in Figure 9B.

Relative advantages between the P-I-D configuration of Figure 8 and the "double derivative" configuration of Figure 3 depend strongly upon associated apparatus. With the P-I-D configuration of Figure 8 sufficiently prompt response of the pilot servomotor (with a lag time constant not greater than 0.1 second) is more difficult to obtain than with the configurations of Figures 3, 6, or 7 where the pilot servomotor itself can perform the function of integration.

Alinement

These higher-order governors offer three or four adjustments to be optimized. While alinement may seem somewhat complex compared with the temporary droop-type governor for which guides are available,[7 8] this is actually not a serious consideration. Some guidance has been offered in connection with Figure 5. Adjustments for loaded operation can be computed by any of several means: analog, digital, or by Bode plots. Although helpful, such computation is not essential. Field alinement by simulated isolation is quite simple and rapid.[9]

To fully exploit capability of the higher-order governors, a different adjustment should be available for synchronizing than for loaded operation. The adjustment for synchronizing will differ from the adjustment for loaded operation in that the pilot servomotor may be faster. Its integration time may be as little as one-tenth the value for loaded operation.

Refinement of Servomotor Control

The best servocontrol heretofore available has been accomplished by enclosing the pilot valve, main valve, and servomotor in a closed loop as shown in Figure 10

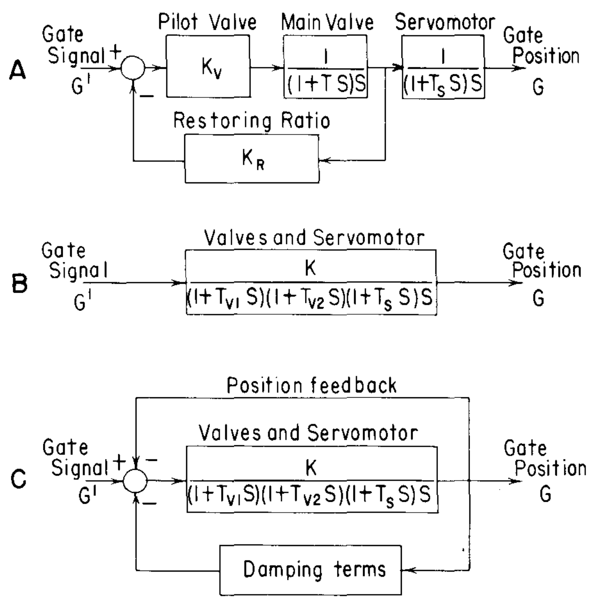


Figure 10. Servosystem representation.
A. Component functions.
B. Combined functions of components.
C. Closed loop.

and driving that loop by a pilot servomotor in which the governor's integration function is performed with higher accuracy. This simple feedback loop around the main servomotor system is quite helpful but performance is still limited because of the presence of three time constants in the loop which requires that gain be kept low to preserve stability. Because of the low gain, linearization and the reduction of response time by this technique are still modest.

Exploration of additional measures to improve response has included valve ports, oil viscosity, barriers to control oil aeration, and phase correction or damping terms in the feedback loop. The latter showed such promise as to supersede all the other relief measures considered. It permits increasing the servo loop gain by a factor approaching 10 with good stability. This accomplishes both linearization and reduction of response time.

For units as large as those for Grand Coulee Third Powerplant the valve delays might be held to the order of 0.2 second. Extrapolating from a servomotor lag term of 0.12 second measured on the existing 108-mw Coulee units, a servomotor lag time constant in the order of 0.5 second for the 600-mw units might be expected. With these values a simple feedback loop as shown in Figure 11A would be restricted to 0.75 per unit gain and response would require over 2 seconds.

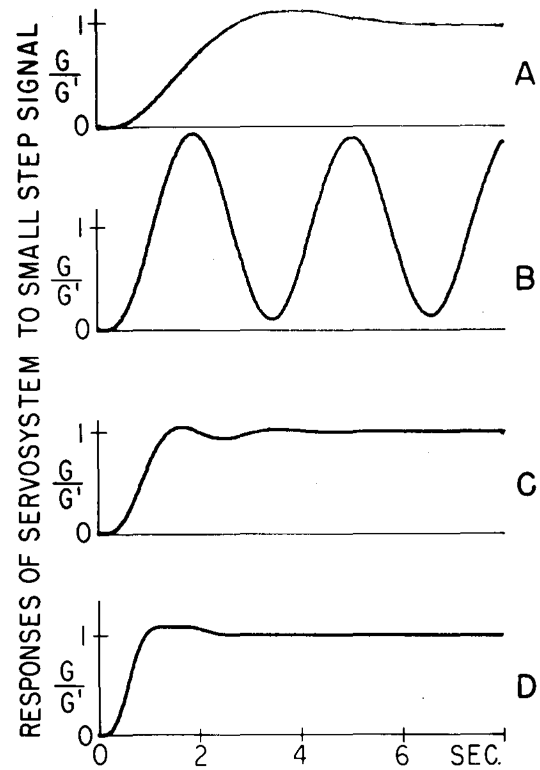


Figure 11. Responses of closed loop servosystem of Figure 10C to step signals within proportional range of valves, $T_{v1} = 0.2$, $T_{v2} = 0.2$, $T_s = 0.5$.
A. Gain K = 0.75, no damping.
B. Gain K = 3.3, no damping.
C. Gain K = 3.3, velocity damping = 0.57S.
D. Gain K = 10, velocity and acceleration damping = $0.41S + 0.085S^2/(1 + 0.1S)$.

If the gain is increased to 3.3 per unit, the response becomes underdamped and approaching instability as shown in Figure 11B. By the addition of velocity feedback the response becomes suitably damped, as shown in Figure 11C. Still further improvement of response and linearity would become possible through the addition of some acceleration feedback to the damping signal. Gain could then be increased to 10 for the response shown in Figure 11D. Ideal stability at high gain would result from the combined use of velocity and acceleration to supplement the displacement feedback, but some practical aspects warrant consideration.

It would be essential that the hydraulically operated main servosystem, the power "muscle" part of the governor, be sufficiently invulnerable to failure of any electrical or electronic component or power supply as to remain safely controllable. But mechanical-hydraulic means of generating a velocity signal are somewhat awkward, and for generation of an acceleration signal are still more difficult. The most accurate and convenient means of generating these signals is electrical or electronic, but these signals, from or through a separate power source, could conceivably be lost while the main hydraulic power source remains. If this were to happen, some degradation of performance could be accepted provided safe control could never be lost.

Thus, unless unusual independence or fail safe interlocking could be devised, it would be prudent to confine gain in the main servo loop to a value just below the point of instability without the stabilizing terms of the feedback. Then if the stabilizing terms were accidentally lost, servomotor response might become underdamped as in Figure 11B but oscillation would subside rather than increase.

This consideration would limit the gain to the order shown for Figure 11C and although with the addition of velocity feedback the damping is not yet ideal, that is not essential for this minor loop of the overall control system. Control of the speed transient becomes quite satisfactory as shown in Figure 12B. The improvement is substantial over that of Figure 12A where no velocity feedback has been employed and the servomotor response is that of Figure 11A.

The ideal combination of velocity and acceleration signals for stabilizing the servomotor loop, Figure 11D, might be considered usable if sufficient velocity signal were generated by mechanical-hydraulic or similarly independent means, so as to keep the loop from becoming unstable if the acceleration component were lost. Control of the speed transient would then be as shown by Figure 12C. While that modest further improvement may be attractive, it is at some risk and complication.

It is to be recognized that the substantial reduction of response time of the main servosystem made possible through use of velocity feedback and the corresponding improvement of control of the speed transient are only part of the improvement. Since the linearity is improved also, the response will remain effective at much smaller speed deviations.

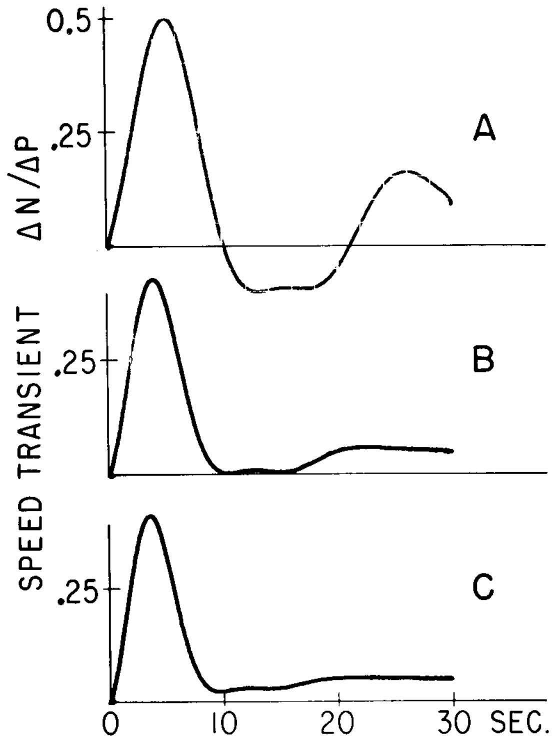


Figure 12. Influence of gain and damping in servosystem upon control of speed transients by derivative governor of Figure 3 with $\sigma = 0.05$, $T_2 = 0.1$, $T_4 = 0.1$, $T_7 = 1.5$, $T_w = 1.85$, $T_m = 10.5$.

A. Servo loop gain 0.75, no damping (Figure 11A); governor settings $K_1 = 5.83$, $K_2 = 2.66$.
B. Servo loop gain 3.3, velocity damping (Figure 11C); governor settings $K_1 = 4.85$, $K_2 = 3.54$.
C. Servo loop gain 10, velocity and acceleration damping (Figure 11D); governor settings $K_1 = 3.77$, $K_2 = 3.74$.

Verification by Field Tests

Practicality of the refinements indicated by these studies were verified by field tests on one of the largest units available, Grand Coulee Unit G–4. That unit was temporarily controlled according to the various schemes using a small analog computer and appropriate transducers.

The result of high gain in a servocontrol loop without damping is shown in Figure 13A. This result compares well with the study shown in Figure 11B. The

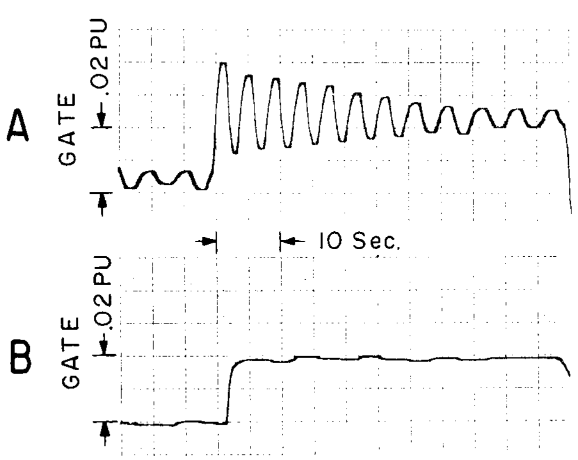


Figure 13. Field test of closed loop servosystem on Unit G-4 in existing Grand Coulee Powerplant. Response to 0.02 per unit step signal with loop gain of 7.

A. Position feedback only, no damping.
B. With velocity feedback damping.

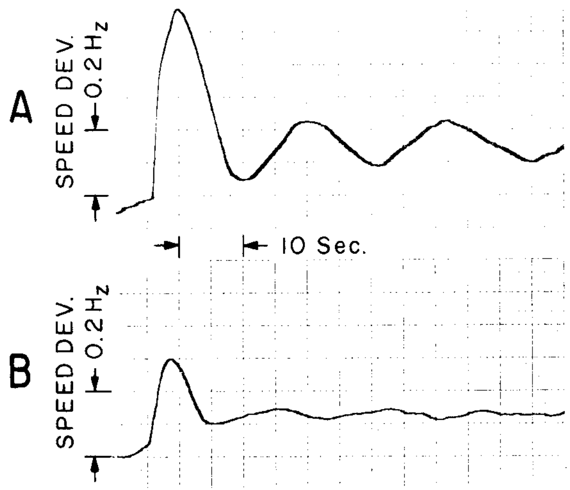


Figure 14. Field test of governor systems on Unit G-4 in existing Grand Coulee Powerplant. Speed transient response to 0.03 per unit load increment by simulated isolation.

A. Mechanical temporary droop-type governor.
B. Electronic derivative governor of Figure 3 with closed loop servosystem response as in Figure 13B.

difference in usable gain is because the time constants in the existing units are not as long as those expected in future larger units. When velocity feedback damping is added, the response becomes nearly ideal as shown in Figure 13B. The result compares well with the study of Figure 11C. While the improvement in response time is evident, the reduction of nonlinearity can be inferred from the reciprocal of gain improvement; that is, to one-seventh.

Performance of the conventional mechanical temporary droop type of governor is shown in the field test result of Figure 14A while Figure 14B shows the field test performance of the more refined type of governor of Figure 3 and with a high gain, damped servo loop with characteristics shown in Figure 13B. These field test results compare well with the study results of Figure 4. A minor difference is that the mechanical temporary droop governor performance shown in Figure 14A does not have benefit of a closed-loop servosystem, while benefit of that feature is represented in the study shown in Figure 4A. With allowance for that difference the field test results are considered a close verification of the improved performance to be expected from the refinements indicated by these studies.

SEVERE CONDITION PERFORMANCE

The conditions under which a governor must function encompass a rather wide range. Rapid, linear, and stable response to the small deviations of system frequency and to system load control signals in normal operation is only part of the governing requirement. Of considerable importance also is the performance under the extreme conditions of startup, and various degrees of load rejection ranging up to rejection of full load. Performance under these conditions can be classified as large signal performance. It is characterized by certain unavoidable nonlinearities such as gate limits and valve limits. It was considered appropriate to examine performance under these conditions to establish dynamic ranges required for the respective signals and to choose simple but adequate auxiliary control features such as for automatic startup.

Startup

Well controlled startup performance is especially important for peaking plants where the units may be started and stopped daily. Computed performance during startup in which the gates are allowed to open to 40 percent is shown in Figure 13. This was computed for a derivative governor of the type shown in Figure 3, servocontrol system of Figure 11C, and basic parameters as for Figure 4 (C and D). Performance is seen to be quite satisfactory without auxiliary control, overshoot of speed being only 2 percent and reverting to stable speed for synchronizing in about 50 seconds.

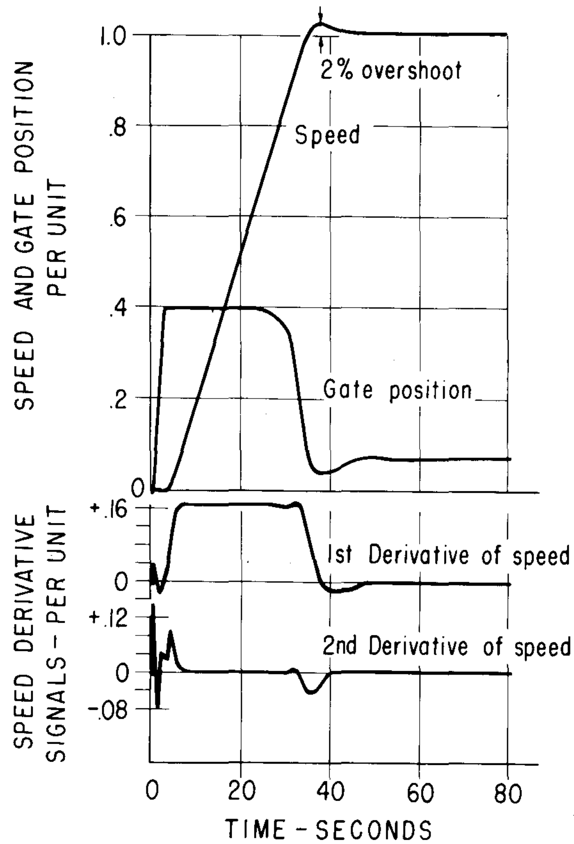


Figure 15. Startup with gate limit at 0.4 per unit.

The startup in this study was initiated by allowing the gates to open at their maximum rate until held by the gate limit at 40 percent of full opening. The maximum rate of gate movement is 8 seconds for full travel or 3.2 seconds for travel to the 40 percent position. The rate is limited by the valve stops. After breakaway, speed rises smoothly until at 84 percent normal speed the speed signal plus acceleration terms start the gates rapidly toward the closed position. Finally, the gates reopen to about 8 percent to maintain normal speed.

This computed sequence allows a number of interesting observations. A speed switch set at 90 percent of normal speed could safely be used to release the 40 percent gate limit used for the starting sequence. At that point the gates are already under control of the governor speed terms. The first derivative of speed, which is acceleration, becomes uniform at about 0.16 per unit soon after breakaway and remains uniform corresponding to the uniform slope of the speed curve until at 32 seconds the sum of speed (0.84 per unit) and its first derivative (0.16 per unit) exceeds the reference (1.0 per unit) and the gates start closing rapidly. The first derivative of speed becomes slightly negative for a few seconds after maximum speed corresponding to the slight downward slope of the speed curve until rated value is reached.

The second derivative signal corresponds to curvature of the speed trace and it shows a modest positive value at breakaway as the speed bends upward. A modest negative value is displayed between 32 and 40 seconds as the speed bends downward. For each derivative signal a dynamic range of 0.2 per unit of the rated speed signal is more than enough for this set of conditions.

A more cautious startup operation is shown in Figure 16 where the gates are allowed to open only to 20 percent of full opening. As would be expected, acceleration is more gradual and the overshoot of speed of 1 percent is half that developed when starting with 40 percent of full gate opening.

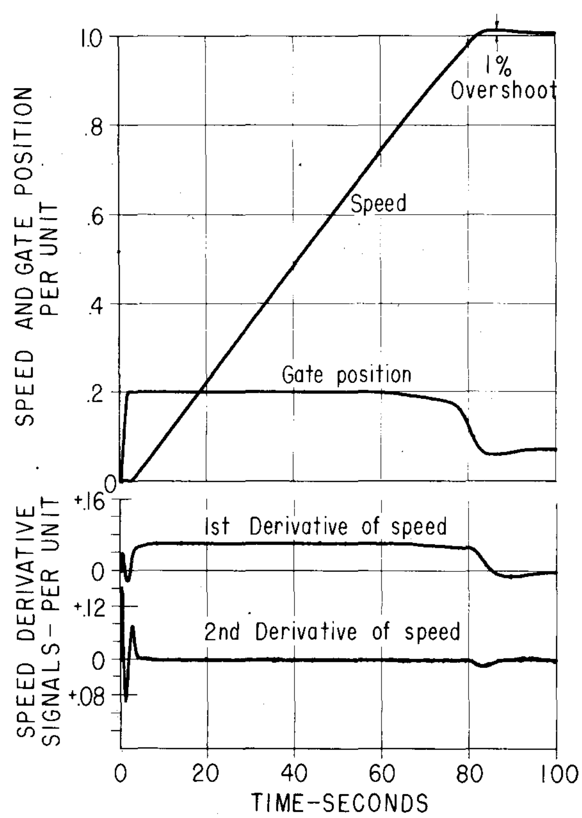


Figure 16. Startup with gate limit at 0.2 per unit.

From a governing standpoint, this cautious startup procedure may be seen to yield more constraints than benefits. The time for speed to become stable at set value is 100 seconds or twice as long as the time required when starting with 40 percent gate, yet the reduction of overshoot is of small consequence since a larger value could easily be tolerated. Furthermore, the speed signals do not assume control of the gates until the speed reaches 94 percent of normal. If the 20 percent gate limit for starting is to be released by a speed switch, its setting must not be less than 95 percent of normal, a more exacting setting than that required when starting with 40 percent of full gate opening.

From these computations it is evident that governing needs impose no restriction against the more rapid starting procedure. Hence, final choice of procedure can be made on the basis of mechanical, structural, or hydraulic conditions.

A few comments concerning parameters represented in these computations are in order here. The governor characteristics, water hammer, and mechanical inertia were all adequately represented in these runs by analog computer. The turbine torque versus speed characteristic was represented as constant whereas in reality the torque of a Francis-type turbine is higher at reduced speed. More rigorous representation would show the speed to rise a little more steeply at first, diminishing to the slopes shown near rated speed. However, since the correct value is represented for rated speed, the dynamic behavior near rated speed is accurate and this is the region of prinicpal interest here.

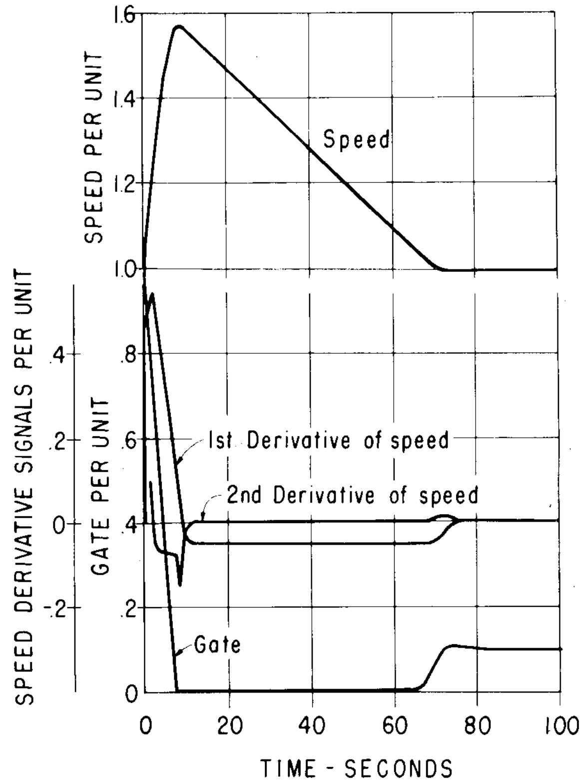


Figure 17. Computed dynamic performance for rejection of rated load.

Full Load Rejection

The most severe condition for which a governor must respond is that of full load rejection. Performance computed for this condition is shown in Figure 17. There it is seen that the gates are started in the close direction at maximum velocity almost instantly. This is accomplished by the two derivative terms, both of which become strongly positive at the instant of rejection. Although a range of 0.5 per unit for the derivative signals is sufficient to show the acceleration that results, only 0.187 per unit is sufficient to drive the valve system for maximum velocity of gate travel.

Since this type of governor accomplishes on the basis of its own speed signal as much control of the gates as their velocity limits will permit, an overspeed shutdown device would accomplish nothing additionally useful under normal circumstances. However, if an overspeed shutdown device is desired for backup purposes, there appears to be adequate range for a speed switch to distinguish between normal operation during a rejection when overspeed should not exceed 160 percent and runaway speed which ordinarily would exceed 180 percent. A speed switch setting in the vicinity of 165 percent of normal speed should serve the backup function satisfactorily without producing unnecessary inconvenience.

The maximum overspeed which will be developed by the real apparatus should be slightly less for this full load rejection than that yielded by the computer simulation results of Figure 17 since the decrease of turbine torque at high speed has not been rigorously represented. However, the torque is correctly represented for rated speed and the dynamic behavior shown should be accurate in that vicinity where the recovery is shown to be smooth. The gates reopen to a position to maintain speed at no-load and the speed presently becomes stable at the set value.

Partial Load Rejection

A partial load rejection can be considered a more severe condition from a system load standpoint than a full load rejection since some of the load could be carried into overspeed, depending upon how the power system may be separated. At any rate, the performance for partial load rejections shown in Figures 16A and 16B are informative as to dynamic behavior at and beyond the governing system's limit of linearity.

Figure 18A shows performance to be expected following rejection of the maximum amount of load for which the governor performance is reasonably linear. This is a rejection of 30 percent of rated load. It may be noted that the distributing valve has just been driven to its limit. However, by comparison with Figure 4C, it may be seen that the speed transient has not been appreciably distorted. The first derivative signal developed was sufficient to drive the valve to its limit setting before speed had risen appreciably. The second derivative signal also contributes to prompt action of the valve although its initial spike was of too short duration to show.

Dynamic performance following rejection of 50 percent load (from an initial load of 80 to 30 percent) is shown in Figure 18B. Here it may be noted that the valve is held at its limit stop for 7 seconds. The turbine gates are driven at maximum velocity to the fully closed position where they remain for about 5 seconds until the speed and its derivatives call for reopening of the gates to a position to sustain normal speed. Because of limiting of velocity and of gate position, the maximum speed developed is more than a proportional amount higher than that of Figure 18A, and recovery is somewhat slower. Yet behavior of speed is stable through the entire transient and it becomes optimum as soon as the governor can operate in its linear range. Derivative signals exceeding 0.25 per unit are developed. A dynamic range of 0.4 per unit for the derivative terms should accommodate the practical range of parameter adjustments.

SYNCHRONIZING PERFORMANCE

A governor is seldom called upon to operate isolated and at no-load for extensive periods, nevertheless, performance under this condition is important insofar

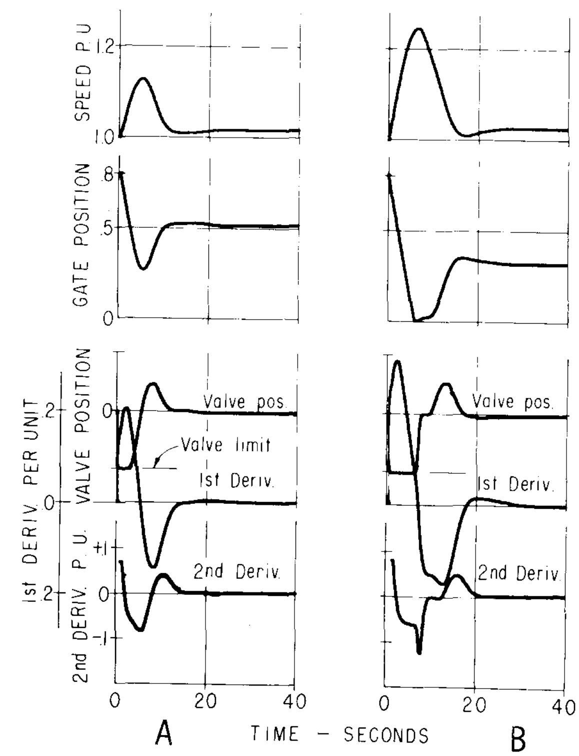


Figure 18. Computed performance for partial load rejection.

A. Rejection of 30 percent load, the largest increment for linear response.

B. Rejection of 50 percent load, showing nonlinear performance.

as it may influence the time required to synchronize a generating unit to the system. In a powerplant for peaking purposes, the units may be started and stopped daily and the rapidity with which they can be synchronized may become rather important. Following emergencies such as load rejection from temporary loss of transmission lines, the rapidity with which synchronizing can be accomplished may become still more important.

The conditions for governing at no-load for synchronizing differ appreciably from the conditions discussed in preceding sections of this study. The signals for speed adjustment are small and there should be no other appreciable perturbations. No influence of valve limits should be encountered as during severe conditions of load rejection or startup. However, the effect of water starting time T_W is only about one-tenth that effect at full load. Consequently, adjustments which yield best performance under loaded conditions would yield performance which could be appreciably improved upon for synchronizing.

Performance computed for the Coulee Third Powerplant units under typical synchronizing conditions is shown in Figure 19. With governor parameters set for loaded operation, the response of speed to a speed change signal is shown by Curve A to be quite stable but to require about 30 seconds to reach the final value. Appreciably better performance for sychronizing can be achieved by the readjustments yielding Curve B which reaches a steady value in 9 seconds. Still more rapid arrival at ultimate speed is potentially possible with the more extensive readjustments yielding Curve C which arrives at the final steady value in 5 seconds. For comparison, the adjustments for loaded operation and for better synchronizing, Curves B and C are tabulated following.

	K_1 1st deriv. coefficient	K_2 2nd deriv. coefficient	T_7 pilot servo integration time
Loaded operation (A)	5.2	7.4	1.5 seconds
Synchronizing (B)	3.5	1.5	0.5
Synchronizing (C)	2.5	1.0	0.25

In practice, some influence from gate friction and minute amounts of backlash in the gate mechanism may somewhat limit ability to accomplish ideal control at the maximum rate. Adjustments in the vicinity of Curve B are considered to be reasonable from a practical standpoint but the potentially useful range of adjustment is nevertheless indicated by the tabulation.

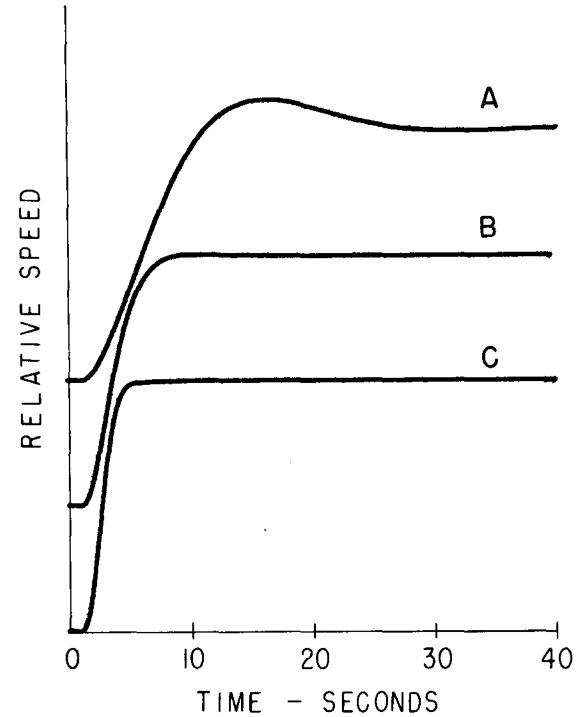


Figure 19. Response of speed to speed change signal under conditions for synchronizing.

A. With governor adjustments for loaded operation, $K_1 = 5.2$, $K_2 = 7.4$, $T_7 = 1.5$ sec.

B. With governor parameters readjusted for synchronizing, $K_1 = 3.5$, $K_2 = 1.5$, $T_7 = 0.5$ sec.

C. With governor parameters readjusted for most rapid synchronizing, $K_1 = 2.5$, $K_2 = 1.0$, $T_7 = 0.25$ sec.

CONCLUSIONS

1. Refinements of governing systems detailed herein will benefit speed control, area load control, and the coordination of these two functions.

2. The refined governing systems can accomplish the desired control with less confinement from the basic parameters of water starting time and mechanical inertia that would be imposed by the heretofore conventional governing systems. Some guide for proportioning of parameters has been offered.

3. Governing of large hydraulic turbines can be substantially improved by the addition of damping to the servomotor control loop so that its gain can be increased for faster response and better linearity.

APPLICABILITY

Results of this investigation are considered applicable to hydrogenerating installations in general. Although the study was undertaken for benefit of the larger installations, expense of incorporating the refinements would be small.

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GLOSSARY OF SYMBOLS

δ	Temporary droop
σ	Permanent droop
G	Gate position
G′	Pilot servomotor position
g	Gravitational constant
hp	Horsepower
h	Head
K	Composite gain
K_1	Coefficient of first derivative
K_2	Coefficient of second derivative
K_R	Restoring ratio
K_v	Valve constant
L	Length of penstock section
ΔL	Load control signal
N	Speed
ΔN	Speed deviation
ΔP	Load increment
S	Laplace operator
T	Valve delay
T_1 T_3 T_5	Lead time constants
T_2 T_4 T_6	Lag time constants
T_7	Pilot servomotor integration time
T_g	Gate response time constant to load signal
T_m	Mechanical starting time = $N^2WR^2/1.6 \times hp \times 10^6$
T_r	Recovery time of temporary droop
T_s	Main servomotor lag time constant
T_{v1}, T_{v2}	Resultant valve system delays
T_w	Water starting time = $\Sigma LV/gh$
V	Velocity in penstock section
WR^2	Inertia

For speed no load Tw = 1/10 normal = .183

* For velocity damping only

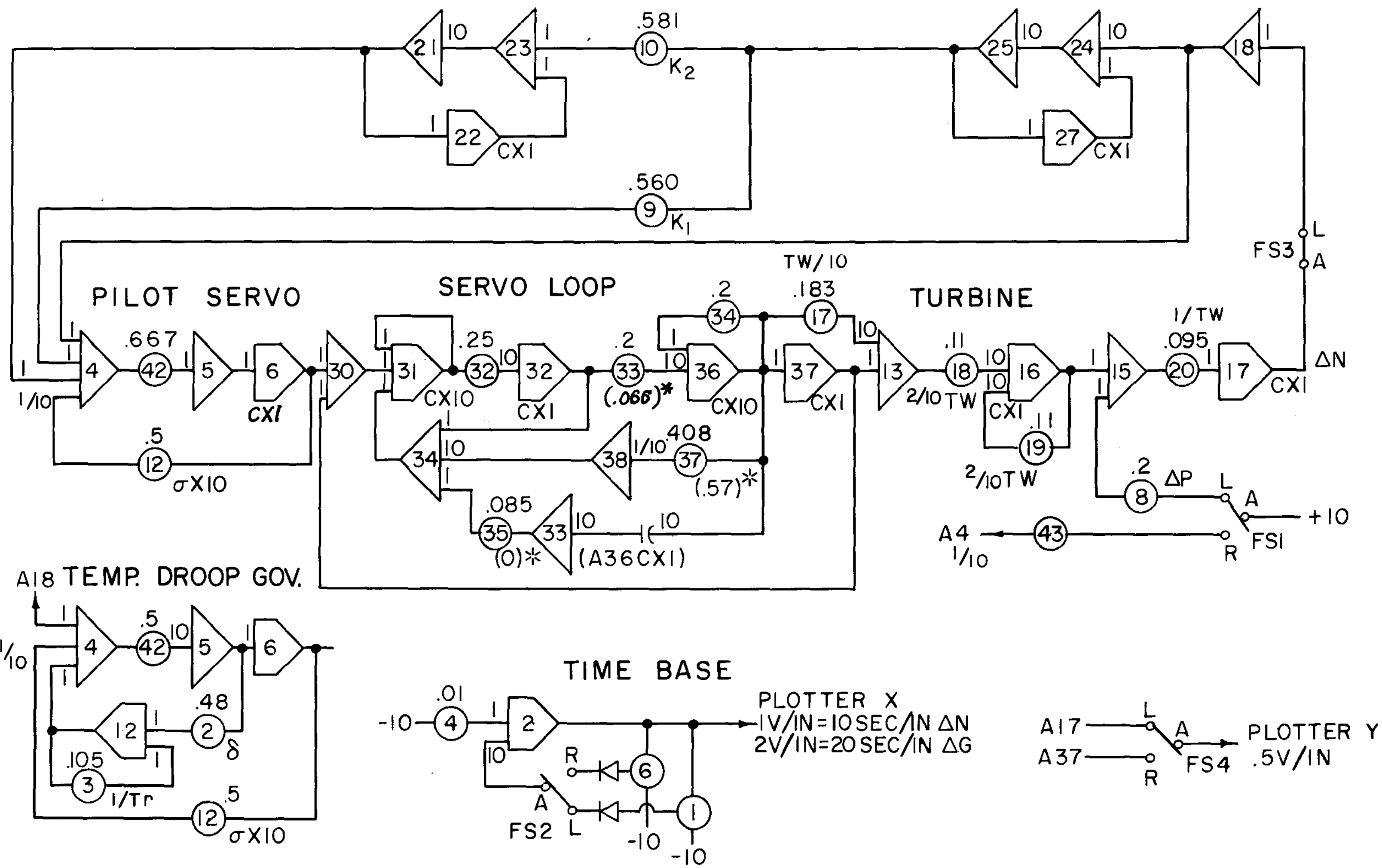

FIGURE A – ANALOG DIAGRAM FOR DOUBLE DERIVATIVE GOVERNOR WITH DAMPED SERVO LOOP

APPENDIX I

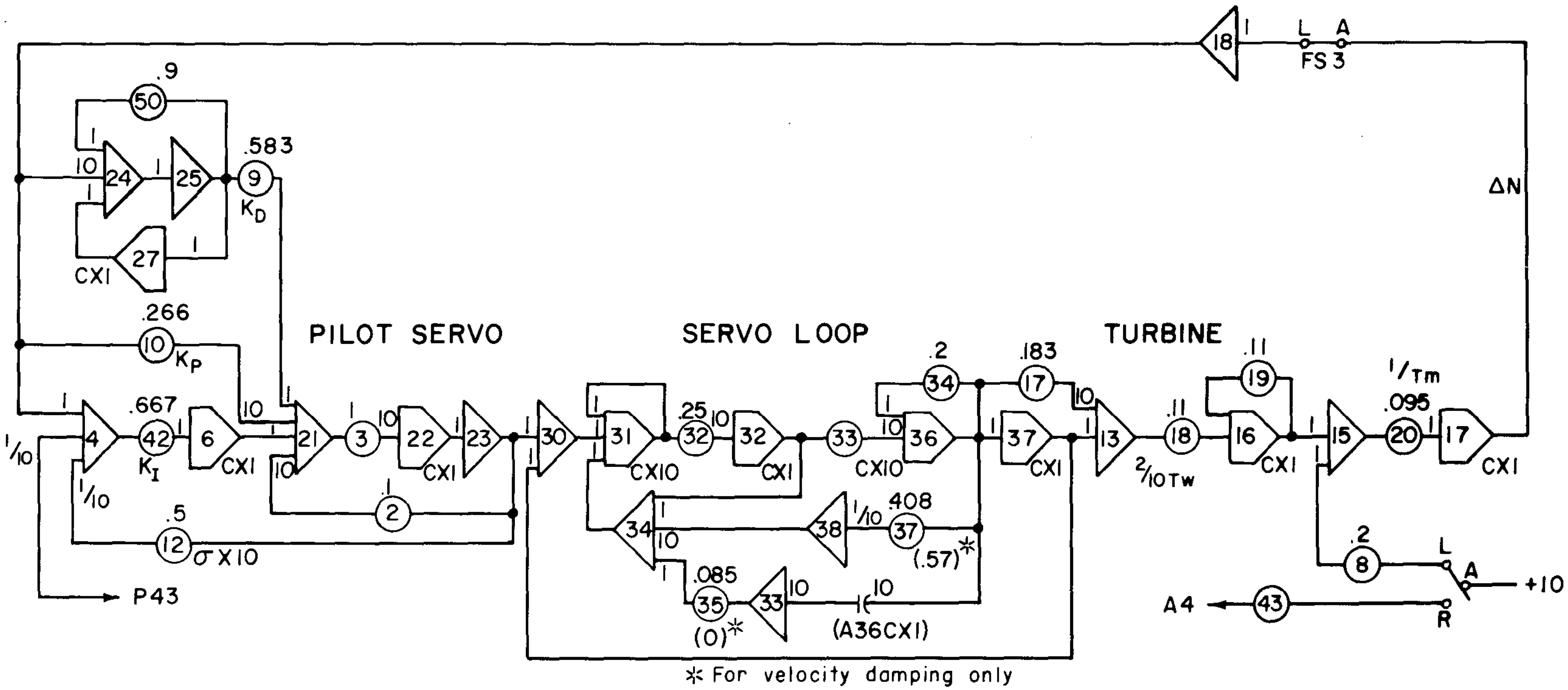

FIGURE B - ANALOG DIAGRAM OF MODIFIED PARALLEL PATH PROPORTIONAL-INTEGRAL-DERIVATIVE GOVERNOR WITH DAMPED SERVO LOOP

APPENDIX I

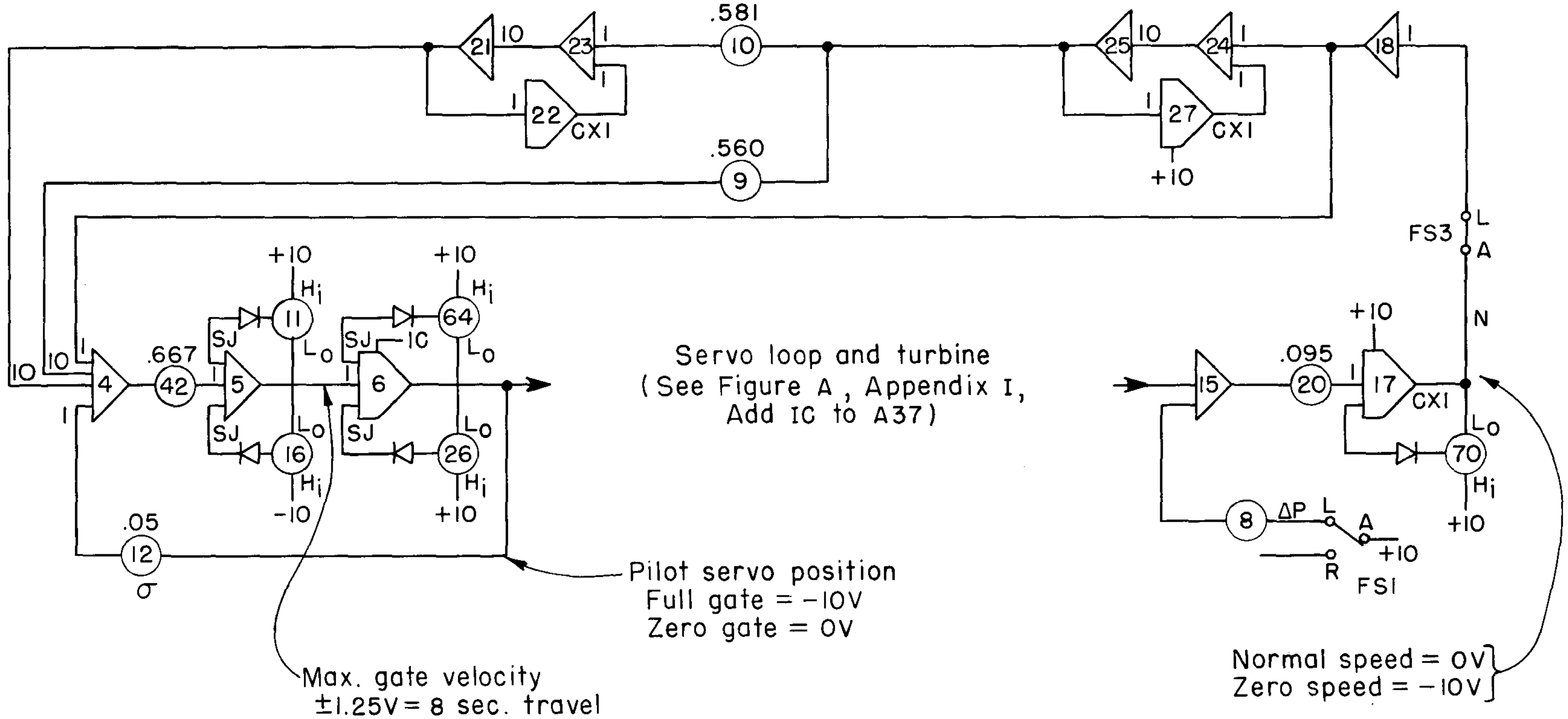

FIGURE C—ANALOG DIAGRAM FOR STUDY OF LARGE SIGNAL PERFORMANCE OF DOUBLE DERIVATIVE GOVERNOR-START UP, LOAD REJECTION--ETC.

APPENDIX I

CONVERSION FACTORS--BRITISH TO METRIC UNITS OF MEASUREMENT

The following conversion factors adopted by the Bureau of Reclamation are those published by the American Society for Testing and Materials (ASTM Metric Practice Guide, E 380-68) except that additional factors (*) commonly used in the Bureau have been added. Further discussion of definitions of quantities and units is given in the ASTM Metric Practice Guide.

The metric units and conversion factors adopted by the ASTM are based on the "International System of Units" (designated SI for Systeme International d'Unites), fixed by the International Committee for Weights and Measures; this system is also known as the Giorgi or MKSA (meter-kilogram (mass)-second-ampere) system. This system has been adopted by the International Organization for Standardization in ISO Recommendation R-31.

The metric technical unit of force is the kilogram-force; this is the force which, when applied to a body having a mass of 1 kg, gives it an acceleration of 9.80665 m/sec/sec, the standard acceleration of free fall toward the earth's center for sea level at 45 deg latitude. The metric unit of force in SI units is the newton (N), which is defined as that force which, when applied to a body having a mass of 1 kg, gives it an acceleration of 1 m/sec/sec. These units must be distinguished from the (inconstant) local weight of a body having a mass of 1 kg; that is, the weight of a body is that force with which a body is attracted to the earth and is equal to the mass of a body multiplied by the acceleration due to gravity. However, because it is general practice to use "pound" rather than the technically correct term "pound-force," the term "kilogram" (or derived mass unit) has been used in this guide instead of "kilogram-force" in expressing the conversion factors for forces. The newton unit of force will find increasing use, and is essential in SI units.

Where approximate or nominal English units are used to express a value or range of values, the converted metric units in parentheses are also approximate or nominal. Where precise English units are used, the converted metric units are expressed as equally significant values.

Table I

QUANTITIES AND UNITS OF SPACE

Multiply	By	To obtain
	LENGTH	
Mil	25.4 (exactly)	Micron
Inches	25.4 (exactly)	Millimeters
	2.54 (exactly)*	Centimeters
Feet	30.48 (exactly)	Centimeters
	0.3048 (exactly)*	Meters
	0.0003048 (exactly)*	Kilometers
Yards	0.9144 (exactly)	Meters
Miles (statute)	1,609.344 (exactly)*	Meters
	1.609344 (exactly)	Kilometers
	AREA	
Square inches	6.4516 (exactly)	Square centimeters
Square feet	929.03*	Square centimeters
	0.092903	Square meters
Square yards	0.836127	Square meters
Acres	0.40469*	Hectares
	4,046.9*	Square meters
	0.0040469*	Square kilometers
Square miles	2.58999	Square kilometers
	VOLUME	
Cubic inches	16.3871	Cubic centimeters
Cubic feet	0.0283168	Cubic meters
Cubic yards	0.764555	Cubic meters
	CAPACITY	
Fluid ounces (U.S.)	29.5737	Cubic centimeters
	29.5729	Milliliters
Liquid pints (U.S.)	0.473179	Cubic decimeters
	0.473166	Liters
Quarts (U.S.)	946.358*	Cubic centimeters
	0.946331*	Liters
Gallons (U.S.)	3,785.43*	Cubic centimeters
	3.78543	Cubic decimeters
	3.78533	Liters
	0.00378543*	Cubic meters
Gallons (U.K.)	4.54609	Cubic decimeters
	4.54596	Liters
Cubic feet	28.3160	Liters
Cubic yards	764.55*	Liters
Acre-feet	1,233.5*	Cubic meters
	1,233,500*	Liters

Table II

QUANTITIES AND UNITS OF MECHANICS

Multiply	By	To obtain
	MASS	
Grains (1/7,000 lb)	64.79891 (exactly)	Milligrams
Troy ounces (480 grains)	31.1035	Grams
Ounces (avdp)	28.3495	Grams
Pounds (avdp)	0.45359237 (exactly)	Kilograms
Short tons (2,000 lb)	907.185	Kilograms
	0.907185	Metric tons
Long tons (2,240 lb)	1,016.05	Kilograms
	FORCE/AREA	
Pounds per square inch	0.070307	Kilograms per square centimeter
	0.689476	Newtons per square centimeter
Pounds per square foot	4.88243	Kilograms per square meter
	47.8803	Newtons per square meter
	MASS/VOLUME (DENSITY)	
Ounces per cubic inch	1.72999	Grams per cubic centimeter
Pounds per cubic foot	16.0185	Kilograms per cubic meter
	0.0160185	Grams per cubic centimeter
Tons (long) per cubic yard	1.32894	Grams per cubic centimeter
	MASS/CAPACITY	
Ounces per gallon (U.S.)	7.4893	Grams per liter
Ounces per gallon (U.K.)	6.2362	Grams per liter
Pounds per gallon (U.S.)	119.829	Grams per liter
Pounds per gallon (U.K.)	99.779	Grams per liter
	BENDING MOMENT OR TORQUE	
Inch-pounds	0.011521	Meter-kilograms
	1.12985 x 10^6	Centimeter-dynes
Foot-pounds	0.138255	Meter-kilograms
	1.35582 x 10^7	Centimeter-dynes
Foot-pounds per inch	5.4431	Centimeter-kilograms per centimeter
Ounce-inches	72.008	Gram-centimeters
	VELOCITY	
Feet per second	30.48 (exactly)	Centimeters per second
	0.3048 (exactly)*	Meters per second
Feet per year	0.965873 x 10^{-6}*	Centimeters per second
Miles per hour	1.609344 (exactly)	Kilometers per hour
	0.44704 (exactly)	Meters per second
	ACCELERATION*	
Feet per second^2	0.3048*	Meters per second^2
	FLOW	
Cubic feet per second (second-feet)	0.028317*	Cubic meters per second
Cubic feet per minute	0.4719	Liters per second
Gallons (U.S.) per minute	0.06309	Liters per second
	FORCE*	
Pounds	0.453592*	Kilograms
	4.4482*	Newtons
	4.4482 x 10^{-5}*	Dynes
	WORK AND ENERGY*	
British thermal units (Btu)	0.252*	Kilogram calories
	1,055.06	Joules
Btu per pound	2.326 (exactly)	Joules per gram
Foot-pounds	1.35582*	Joules
	POWER	
Horsepower	745.700	Watts
Btu per hour	0.293071	Watts
Foot-pounds per second	1.35582	Watts
	HEAT TRANSFER	
Btu in./hr ft^2 deg F (k, thermal conductivity)	1.442	Milliwatts/cm deg C
	0.1240	Kg cal/hr m deg C
Btu ft/hr ft^2 deg F	1.4880*	Kg cal m/hr m^2 deg C
Btu/hr ft^2 deg F (C, thermal conductance)	0.568	Milliwatts/cm^2 deg C
	4.882	Kg cal/hr m^2 deg C
Deg F hr ft^2/Btu (R, thermal resistance)	1.761	Deg C cm^2/milliwatt
Btu/lb deg F (c, heat capacity)	4.1868	J/g deg C
Btu/lb deg F	1.000*	Cal/gram deg C
Ft^2/hr (thermal diffusivity)	0.2581	Cm^2/sec
	0.09290*	M^2/hr
	WATER VAPOR TRANSMISSION	
Grains/hr ft^2 (water vapor transmission)	16.7	Grams/24 hr m^2
Perms (permeance)	0.659	Metric perms
Perm-inches (permeability)	1.67	Metric perm-centimeters

Table III

OTHER QUANTITIES AND UNITS

Multiply	By	To obtain
Cubic feet per square foot per day (seepage)	304.8*	Liters per square meter per day
Pound-seconds per square foot (viscosity)	4.8824*	Kilogram second per square meter
Square feet per second (viscosity)	0.092903*	Square meters per second
Fahrenheit degrees (change)*	5/9 exactly	Celsius or Kelvin degrees (change)*
Volts per mil	0.03937	Kilovolts per millimeter
Lumens per square foot (foot-candles)	10.764	Lumens per square meter
Ohm-circular mils per foot	0.001662	Ohm-square millimeters per meter
Millicuries per cubic foot	35.3147*	Millicuries per cubic meter
Milliamps per square foot	10.7639*	Milliamps per square meter
Gallons per square yard	4.527219*	Liters per square meter
Pounds per inch	0.17858*	Kilograms per centimeter

GPO 834-578

ABSTRACT

In considering appropriate parameters for large hydropower generating units, the choice of control characteristics to satisfy power system needs strongly influences the economics of design. Parameters fundamental to control characteristics which can be determined by the designers are mechanical inertia or flywheel effect and penstock time constant. To aid the designer with a basis for the most economical combinations of parameters to satisfy the power system needs, a study analyzing the requirements and their interrelation was made. Results of the study are given. The influence of the governing system and refinements with several new features for better speed control were investigated toward improved coordination between unit speed control and area load control. A guide for proportioning of parameters is proposed.

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