SYSTEM ANALYTICAL MODELS

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System Analytical Models

2.1 Fossil - Fuel Boilers and Associated Controllers

The control of fossil-fuel fired steam boilers poses a number of problems. In general these problems become more pressing as the size of generator units increase.

Descriptions of the physical features of particular boiler designs are given in Reference 1.

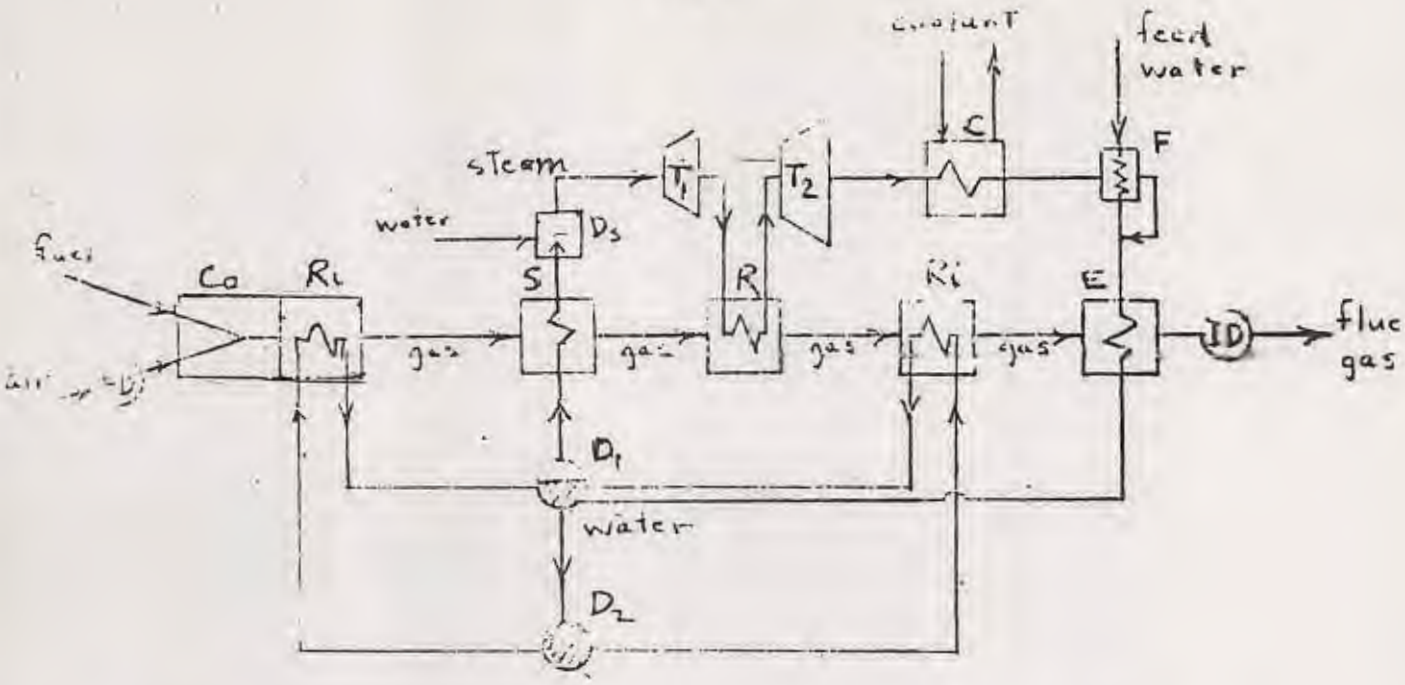
The various system parameters in a high performance steam generator are highly interactive, however, for the purposes of description, the usual boiler control systems are discussed here separately under the headings of combustion control, feedwater control and steam temperature control. These apply to natural circulation units most commonly applied in Canada at the present time. These controls are applied to the basic heat transfer system such as that illustrated in Figure 2.1 The actual dynamic relationships which govern the heat transfer process itself are quite complex. The developments of these equations are outlined in Reference 3.

Combustion Control

The function of the combustion control system is to maintain a firing rate sufficient to meet the load demands on the boiler. It must: maintain the outlet steam pressure as near constant as possible; keep the fuel/air ratio at the optimum, a deficiency of air leads to the collection of explosive gases in the furnace and an excess of air leads to inefficient operation due to the heat the excess air carries out of the furnace; finally it must maintain the interior of the furnace at a slightly negative pressure (draft) to prevent the escape of poisonous gases into the furnace room. The inputs to the combustion control systems are usually the outlet steam pressure, the fuel/air ratio, furnace pressure (draft). In some instances it is necessary to use generator megawatt load or boiler steam flow information to provide predictive signals to the control systems to prevent excessive deviations from desired values due to long inherent delays in the process.

Feedwater Control

The function of the feedwater control system is to regulate the water supply to the boiler (from the condensor and feedwater make-up supply) such that a safe level is maintained in the boiler. Too low a water level in the boiler results in damage due to overheating of the tubes exposed to hot gases; too high a level results in water droplets passing out of the boiler to the turbine leading to mechanical damage to it. The water level in the steam drum cannot be used as a primary measurement alone because of surging due to bubble formation during periods of increasing load demand. In modern electric utility boilers the practice is generally to measure steam outlet flow, water inlet flow and water level in the boiler and to control the inlet flow to maintain a balance between the mass flows of water in and steam out. The water level information is used to provide a slow acting reset signal so that a desired average level is maintained over longer periods of time.



Co - combustion chamber
R_1 - steam riser
S - superheater
R - reheater
E - economizer
F - feedwater heater
D_s - desuperheater

C - steam condensor
T_1 - high pressure turbine
T_2 - low pressure turbine
D_1 - steam drum
D_2 - water drum
FD, ID - forced and induced draft fans.

Figure 2.1 Simplified mass-energy flow diagram of typical fossil fuel steam generator

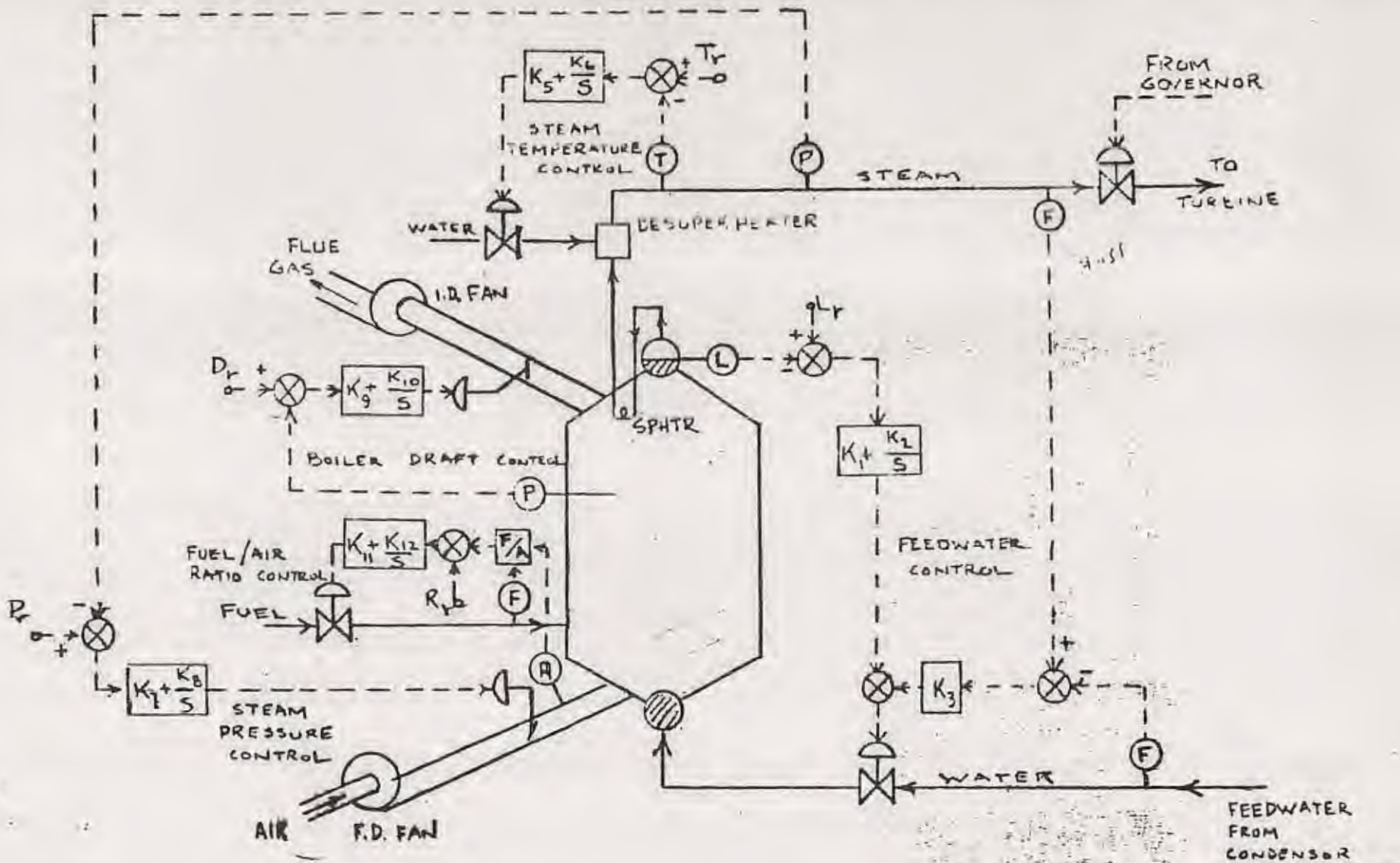


Figure 2.2 Typical control system for a natural circulation steam boiler for electric utility service.

Steam Temperature Control

Precise regulation of superheated steam is essential to prevent thermal cycling with resultant mechanical fatique damage to the turbine. Steam outlet temperature tends to fluctuate due to a number of effects such as changes in the amount of excess air, changes in inlet water temperature or type of fuel, contamination of the heat absorbing surfaces, the fluctuation of load demand on the boiler, etc.

A number of methods of outlet steam temperature control are available and various combinations of these basic ideas are often employed. The most common principles applied are as follows:

- change the angles of injection and depth of insertion of fuel supply nozzles in the furnace.
- control gas flow patterns in the furnace using movable control surfaces.
- steam attemperation using pure water sprays directly or by deviating part of the steam through heat exchangers.

The approach used for the control of once-through and supercritical boilers is considerably different. For an introductory discussion of these units refer to Reference 1.

In Figure 2.1 is illustrated a typical arrangement of control devices for an natural circulation steam boiler. In practice these control schemes can be implemented using any combination of electrical, electronics, pneumatic or hydraulic systems. The practice at the present time is to use analog systems although there is a tendency to move toward digital systems as on-line digital data logging is used. Modern steam units are now usually started up, controlled and shut-down under direct digital control.

Reference 6 gives a brief review of the control characteristics of steam boilers for large power swings.

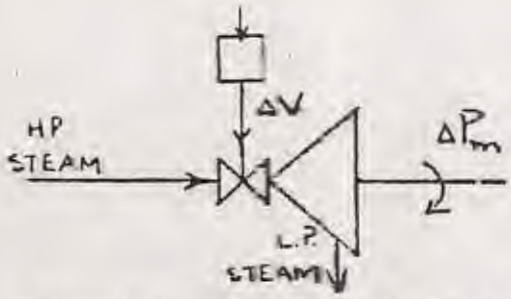
2.2 Nuclear Reactors and Associated Controllers

The control of nuclear power reactors is a rapidly changing field due to the rapid changes in control philosophy as related to reactors and also due to the rapid advances in nuclear engineering itself. There are many types of power reactors now under consideration for utility service and all have engine control features. References 7 and 8 provide a brief review of the pertinent engineering features of present day reactors. For a study of the control of such units the reader is referred to specialist literature on the subject.

2.3 Non-Reheat Steam Turbine

The power output of a steam turbine does not respond instantaneously to changes in the steam inlet valve position due to the effect of entrained steam in the turbine itself. The transfer function for small changes in power level relating the turbine power output to changes in steam valve opening is

$$\frac{\Delta P_m}{\Delta V} = \frac{K}{T_e S + 1} \qquad (2.1)$$



where

ΔP_m is the turbine power output (p.u.)

ΔV is the steam valve opening (p.u.)

T_e is the steam starting time determined by the volume of entrained steam

$T_e \doteq \frac{1}{3}$ second

Figure 2.4 - Non-reheat steam turbine

K is a constant determined by the slope of the mass flow-valve opening characteristic. This is roughly equal to unity near full load.

2.4 Reheat Steam Turbine

If there is a stage of reheat in the steam cycle, the transfer function of the turbine will be altered. This situation is indicated approximately in Figure 2.3.

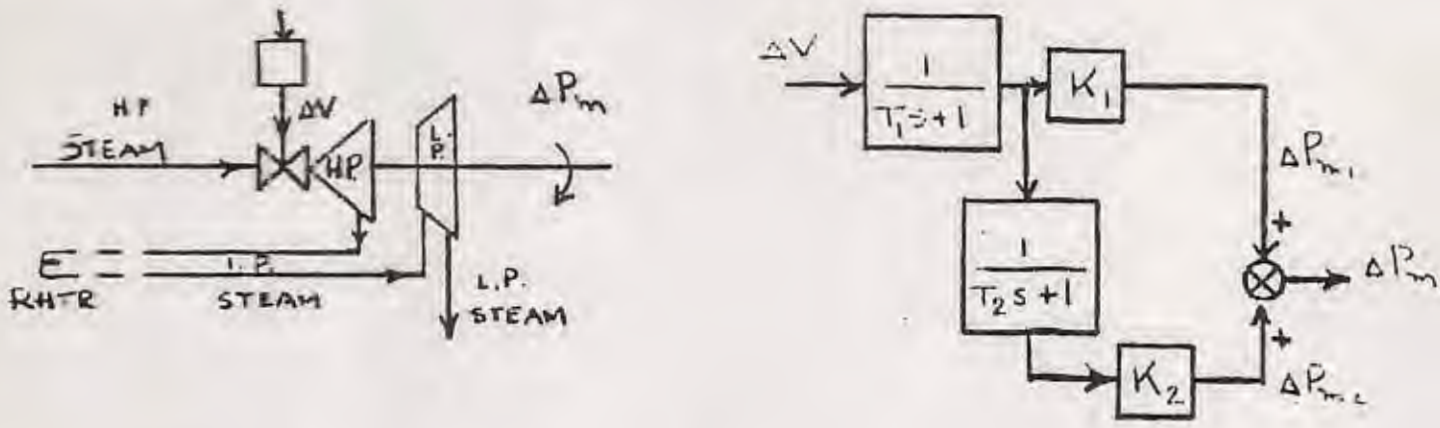


Figure 2.5 - Steam turbine with one stage of reheat

For this case $\Delta P_m = \Delta P_{m1} + \Delta P_{m2} = \Delta V \left\{ \frac{K_1}{(T_1 S + 1)} + \frac{K_2}{(T_1 S+1)(T_2 S+1)} \right\}$

$$\frac{\Delta P_m}{\Delta V} = \frac{K_1 (T_2 S+1) + K_2}{(T_1 S + 1)(T_2 S + 1)} \qquad (2.2)$$

Where ΔP_m and ΔV are in per unit

$T_1 = \frac{1}{3}$ second

$T_2 = 3 \rightarrow 5$ seconds

$K_2/K_1 = .25 \rightarrow .5$

$K_2 + K_1 = 1$

References 9, 10 and 11 describe the mechanical features of typical steam turbines used for utility service.

2.5 Hydraulic Turbine With Penstock - No Surge Tank

This development is based on the following assumptions:

- the conduit and water are not compressible
- the water velocity through the turbine is proportional to the product of gate opening and the square root of the net head.
- the turbine power output is proportional to product of net head and water velocity.
- head losses in the conduit and gate are negligible.

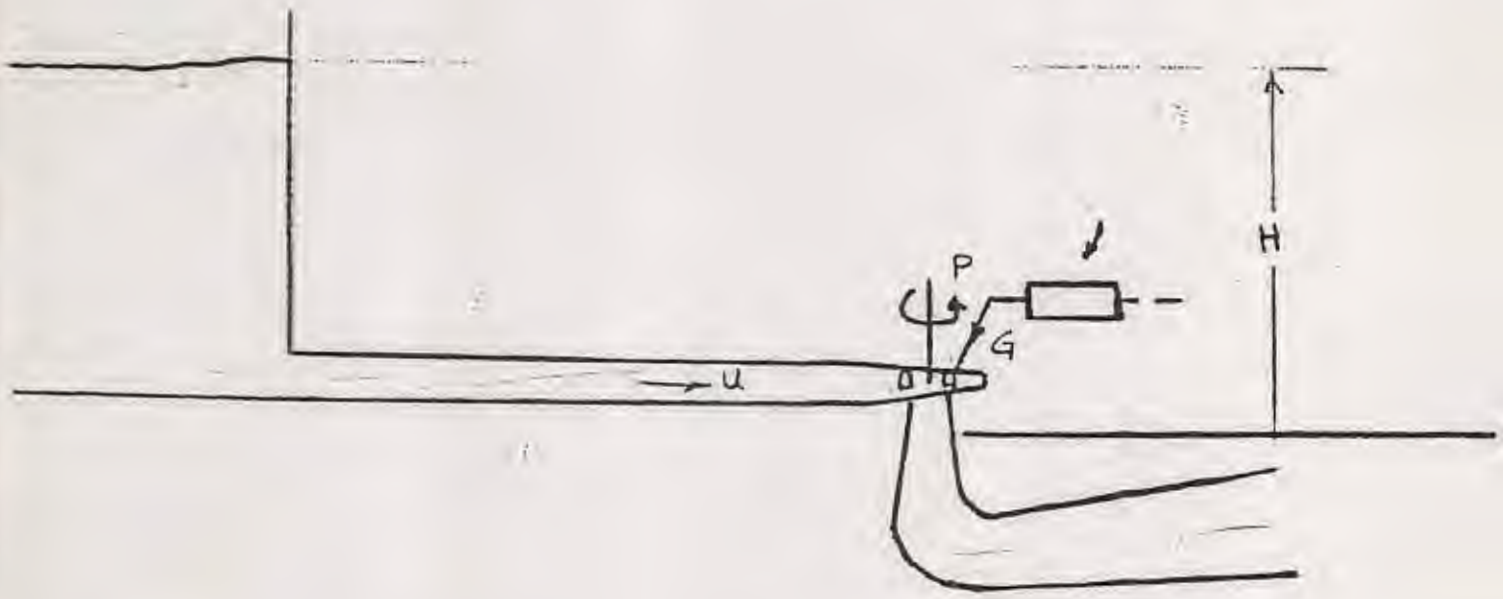


Figure 2.5 -Hydraulic turbine installation without super tank

Power Expression

$$P = K_1 Hu$$

where P is the turbine power output

H is the net head operating

u is the turbine flow

K_1 is a constant

for small changes

$$\Delta P = K_1 H \Delta u + K_1 u \Delta H$$

normalizing by dividing both sides by $P_o = K_1 H_o u_o$, the steady state power level, gives

$$\frac{\Delta P}{P_o} = \frac{\Delta u}{u_o} + \frac{\Delta H}{H_o} \tag{2.3}$$

Turbine Flow

$$u = K_2 \sqrt{H}\ G$$

where K_2 is a constant and G is the wicket gate opening. Then for small changes

$$\Delta u = K_2 \sqrt{H}\ \Delta G + \frac{K_2 G}{2\sqrt{H}}\ \Delta H$$

Normalizing by dividing through by $u_o = K_2 \sqrt{H_o}\ G_o$ (the steady state flow) gives

$$\frac{\Delta u}{u_o} = \frac{\Delta G}{G_o} + \frac{\Delta H}{2H_o} \tag{2.4}$$

Inertial Effect in the Penstock

$$\frac{\zeta L A}{g}\ \frac{d}{dt}\ (\Delta u) = -A\zeta\ \Delta H$$

Normalization by multiplying both sides of the equation by $\frac{u_o}{u_o H_o}$ and Laplace transformation gives

$$\frac{L\ u_o}{g\ H_o}\ s\ \frac{(\Delta u)}{u_o} = \frac{-\Delta H}{H_o}$$

If $\frac{L}{g}\ \frac{u_o}{H_o} = T_w$ is defined as the "water starting time" then

$$T_w s\ \frac{(\Delta u)}{u_o} = \frac{-\Delta H}{H_o} \tag{2.5}$$

Equations (2.3) to (2.5) can be expressed in matrix form.

$$\begin{bmatrix} \Delta P/P_o \\ 0 \\ 0 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 1 \\ 1 & -1 & \frac{1}{2} \\ 0 & T_w S & 1 \end{bmatrix} \begin{bmatrix} \Delta G/G_o \\ \Delta u/u_o \\ \Delta H/H_o \end{bmatrix}$$

These equations can be solved to get the required transfer function

$$\boxed{\frac{\Delta P/P_o}{\Delta G/G_o} = \frac{(1 - T_w S)}{(1 + \frac{T_w S}{2})}}$$

2.6 Hydraulic Turbine With Surge Tank at the Turbine

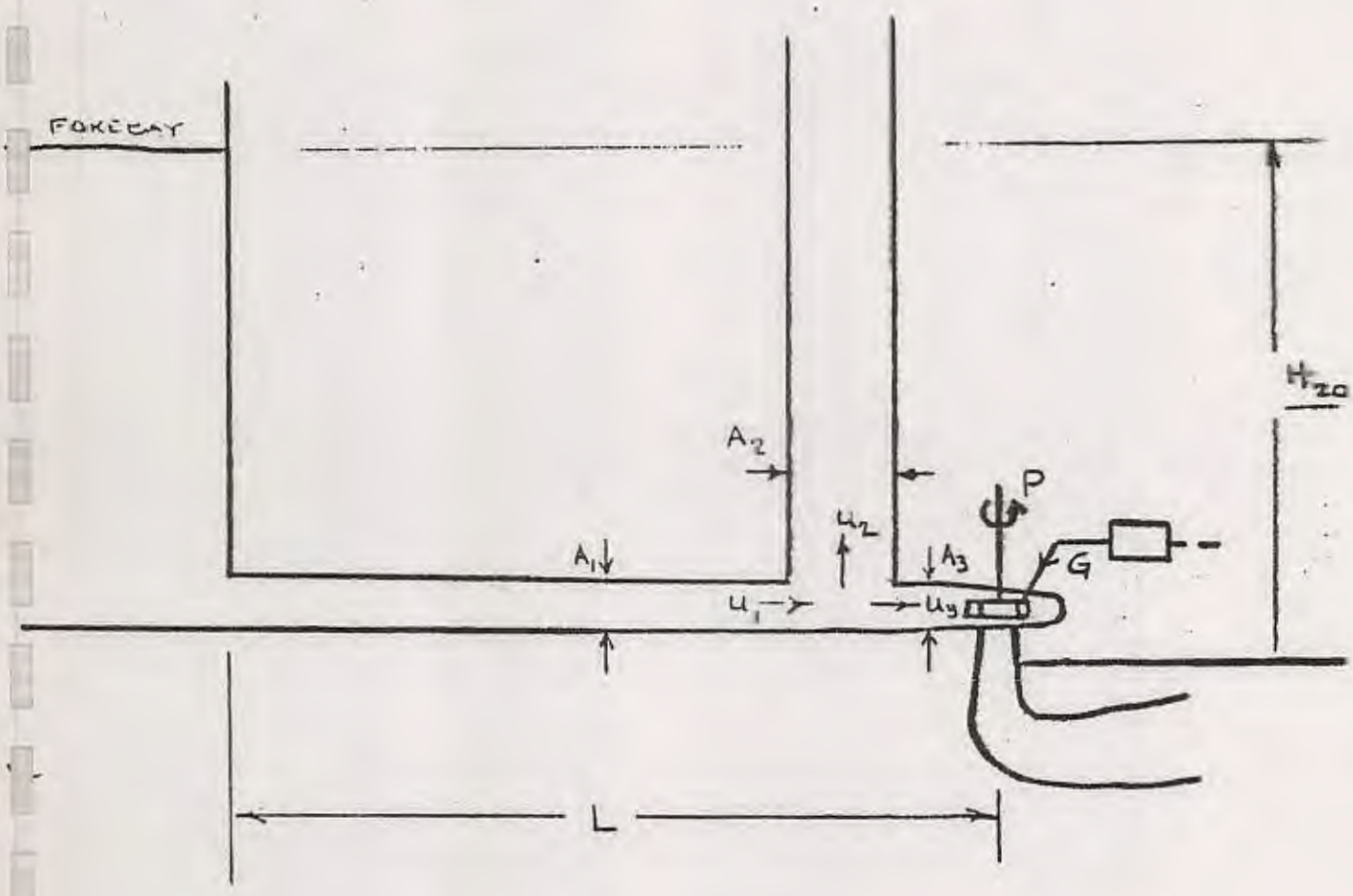


Figure 2.5 - Hydraulic turbine installation with a surge tank at the turbine

Power Expression

$$P = K_1 H_2 u_3$$

where P is the turbine power output

H_2 is the available head

u_3 is the water mass flow rate through the turbine

for small changes

$$\Delta P = K_1 H_2 \Delta u_3 + K_1 u_3 \Delta H_2$$

normalizing by dividing by $P_o = K_1 H_{20} u_{30}$

$$\frac{\Delta P}{P_o} = \frac{\Delta u_3}{u_{30}} + \frac{\Delta H_2}{H_{20}} \tag{2.6}$$

Turbine Flow

$$u_3 = K_2 \sqrt{H_2}\ G$$

where G is the wicket gate opening

for small changes

$$\Delta u_3 = K_2 \sqrt{H_2}\ \Delta G + \frac{K_2 G \Delta H_2}{2\sqrt{H_2}}$$

normalization gives

$$\frac{\Delta u_3}{u_{30}} = \frac{\Delta G}{G_o} + \frac{\Delta H_2}{2H_{20}} \tag{2.7}$$

Inertial Effect in the Penstock

$$\frac{\zeta L A_1}{g} s\ (\Delta u_1) = -A_1 \zeta \Delta H_2$$

normalize by multiplying both sides by $\frac{u_{10}}{u_{10} H_{20}}$, then

$$\frac{L u_{10}}{g H_{20}} s\ \left(\frac{\Delta u_1}{u_{10}}\right) = \frac{-\Delta H_2}{H_{20}}$$

$$\text{or} \quad \frac{\Delta H_2}{H_{20}} = -sT_w \frac{\Delta u_1}{u_{10}} \tag{2.8}$$

where T_w, the water starting time is given by

$$T_w = \frac{L u_{10}}{g H_{20}}$$

Continuity at the Junction

$$A_1 u_1 = A_2 u_2 + A_3 u_3$$

$$u_1 = \frac{A_2}{A_1} u_2 + \frac{A_3}{A_1} u_3$$

for small changes

$$\Delta u_1 = \frac{A_2}{A_1}\,\Delta u_2 + \frac{A_3}{A_1}\,\Delta u_3$$

normalization gives

$$\frac{\Delta u_1}{u_{10}} = \frac{A_2}{A_1}\,\frac{\Delta u_2}{u_{10}} + \frac{A_3}{A_1}\,\frac{\Delta u_3}{u_{10}}$$

but $u_{10}A_1 = u_{30}A_3$ and $u_{20} = 0$ so

$$\frac{\Delta u_1}{u_{10}} = \frac{A_2}{A_1}\,\frac{\Delta u_2}{u_{10}} + \frac{\Delta u_3}{u_{30}} \tag{2.9}$$

Pressure at the Bottom of the Tank

$$P_2A_2 = \rho\, H_2A_2 = \rho\left(\frac{u_2}{s}\,A_2\right)$$

so $sH_2 = u_2$ or $s(H_{20} + \Delta H_2) = u_{20} + \Delta u_2$

now $u_{20} = 0$ so normalization gives

$$s\,\frac{\Delta H_2}{H_{20}} = \frac{u_{10}}{H_{20}}\,\frac{\Delta u_2}{u_{10}} \tag{2.10}$$

Equations (2.5) to (2.10) can be summarized in matrix form as follows:

$$\begin{bmatrix} \Delta P/P_o \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 1 & 1 \\ 1 & 0 & 0 & -1 & \frac{1}{2} \\ 0 & sT_w & 0 & 0 & 1 \\ 0 & 1 & -A_2/A_1 & -1 & 0 \\ 0 & 0 & -u_{10}/H_{20} & 0 & s \end{bmatrix} \begin{bmatrix} \Delta G/G_o \\ \Delta u_1/u_{10} \\ \Delta u_2/u_{10} \\ \Delta u_3/u_{30} \\ \Delta H_2/H_{20} \end{bmatrix}$$

These equations can be solved to give the required transfer function.

$$\frac{\Delta P/P_o}{\Delta G/G_o} = \frac{\left(\frac{L}{g}\,\frac{A_2}{A_1}\,s^2 - T_w s + 1\right)}{\left(\frac{L}{g}\,\frac{A_2}{A_1}\,s^2 + \frac{T_w}{2}s + 1\right)} = \frac{\left(s^2 - \frac{u_{10}}{H_{20}}\,\frac{A_1}{A_2}\,s + \frac{gA_1}{LA_2}\right)}{\left(s^2 + \frac{u_{10}}{2H_{20}}\,\frac{A_1}{A_2}\,s + \frac{gA_1}{LA_2}\right)}$$

2.8 Speed Governors

2.8.1 The principles of governor operation

The speed governors used on generator prime movers may be considered either as speed controllers or power controllers depending on the mode of generator operation. If the electric generator is supplying an isolated load, the governor tends to operate as a speed controller; however, if the generator is connected to a large system where the total generating capacity is much larger than that of the unit itself, the speed (frequency) cannot be appreciably altered by the individual unit and therefore the governor operates as a power controller. The general arrangement of the functional elements of the governor is shown in Figure 2.14. Both feedback paths are normally in operation. They combine to give the governor drop characteristics required for stable parallel operation of generating units.

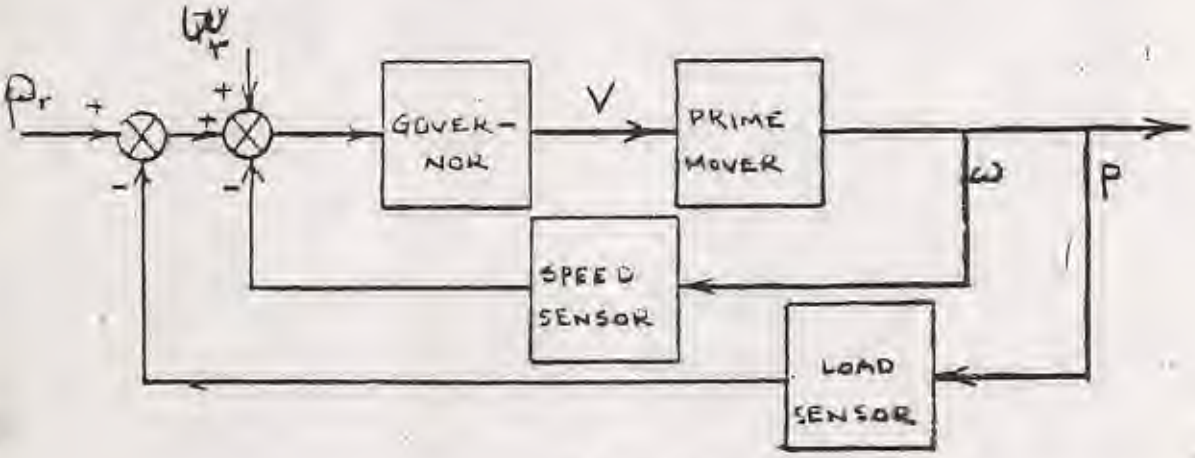


Figure 2.14

For speed and power control study purposes, both the steady-state gain and the transient performance of the governor are of interest. In order to force parallel connected generators to share loads, the governors must be set with a drooping characteristic. Most of the early analytical work on governors was concerned with this steady-state droop characteristic so that the governor terminology which has developed to describe the transient performance of governors is an extension of steady-state characteristics into the dynamic range of operation.

Before developing the transfer functions of typical speed governors, the steady state droop characteristics will be considered.

The steady-state performance of a governor depends on the value of the steady-state speed droop R. That is, under steady-state conditions

$$\Delta V = \frac{1}{R} \Delta\omega$$

or in other words the amount of turbine inlet valve movement which results from a given frequency error is equal to the frequency error divided by R, the "governor droop". All quantities are expressed in per unit with full rated values as a base. These relationships are illustrated in Figure 2.15. The value $\frac{1}{R}$ is sometimes referred to as the "steady-state gain of the governor".

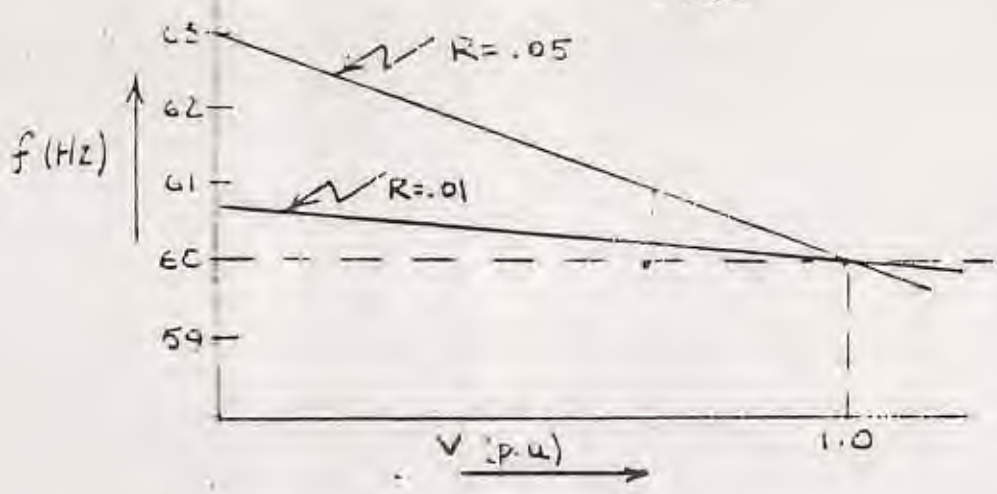


Figure 2.15

If the steam valve were to move over 1.0 p.u. of its travel (full travel) then

$$R = \frac{\Delta\omega}{\omega_r} \times 100$$ is the droop in per cent, ω_r is rated frequency

$$\text{or} \quad R = \frac{\Delta f}{60} \times 100$$

and $\boxed{\Delta f = \frac{6}{10} R}$ for a 1 p.u. change in valve opening or 1 p.u. change in power output of the turbine

An alternative method exists for the expression of the same information, that is the per cent variation in turbine output power per $\frac{1}{10}$ cycle frequency disturbance.

If we define

$$B = \frac{\left(\frac{\Delta P}{P}\right) 100}{\Delta f} = \frac{\text{\% power change}}{\text{cycle frequency change}}$$

$$\text{or} \quad B' = \frac{\left(\frac{\Delta P}{P}\right) 100}{10 \ \Delta f} = \frac{\text{\% power change}}{\frac{1}{10}\text{ cycle frequency change}}$$

then $B' = \frac{10}{\Delta f}$ for a 1 p.u. change in power

or $\boxed{\Delta f = \frac{10}{B'}}$

If one now equates the two alternative statements for Δf.

$$\Delta f = \frac{10}{B'} = \frac{6}{10} R$$

$$B'R = \frac{100}{6} = 16\ 2/3$$

This is a useful relationship to remember.

2.8.2 Mechanical-hydraulic governor for a steam turbine

Figure 2.16 illustrates schematically the basic mechanical and hydraulic features of a conventional mechanical-hydraulic governor used for steam turbine speed control. In some governors, especially those of more recent design the speed sensing and computing parts of the governor are electrical rather than mechanical and an electric-hydraulic transducer is used to couple the electrical signals into the final power gain stages which, in nearly all cases, are hydraulic. The overall transfer function of the more modern units are still very similar to those given here.

Figure 2.17 is an operational block diagram representation of the governor. G_s is the transfer function of the speed sensor. From this diagram, the overall transfer function of the governor can be calculated. Consider that the X variables represent small deviations away from steady state positions and that $X_8 = 0$. Then the inner loop reduces to

$$\frac{X_4}{X_3}(s) = \frac{K_1}{(s + \frac{K_1 e}{f})}$$

and
$$\frac{X_5}{X_1}(s) = \left(\frac{a+b}{a}\right)\left(\frac{c+d}{c}\right)\left\{\frac{K_1K_2}{s^2 + \frac{K_1 es}{f} + \frac{hjdK_1K_2}{c(g+h)(i+j)}}\right\}$$

or more simply
$$\frac{X_5}{X_1}(s) = \frac{A}{(s+m)(s+n)} = \frac{A'}{(T_g s+1)(T_R s+1)}$$

note $X_1 = G_s(\Delta\omega)$

In practice the operation of the speed sensor is normally fast compared with the rest of the governor and also T_R is usually much smaller than T_g so that the transfer function relating steam valve movement X_5 to speed error $\Delta\omega$ is given by

$$\frac{X_5}{\Delta\omega}(s) = \frac{1/R}{(T_g s + 1)}$$ p.u

where X_5 and $\Delta\omega$ are in per unit

R is the steady state speed droop, normally $.01 < R < 0.1$ per unit

T_g normally is in the order of .1 - .25 seconds.

Figure 2.18 and 2.19 illustrate the signal flow graph representation of the governor.

2.8.3 A mechanical hydraulic governor for water turbines (Woodward Governor Co. Type)

Figure 2.20 illustrates schematically the mechanical and hydraulic systems of a conventional temporary droop governor used for hydraulic turbine speed control. The actual physical arrangement of linkages may vary from one unit to another; however, the basic scheme shown will apply to most cases.

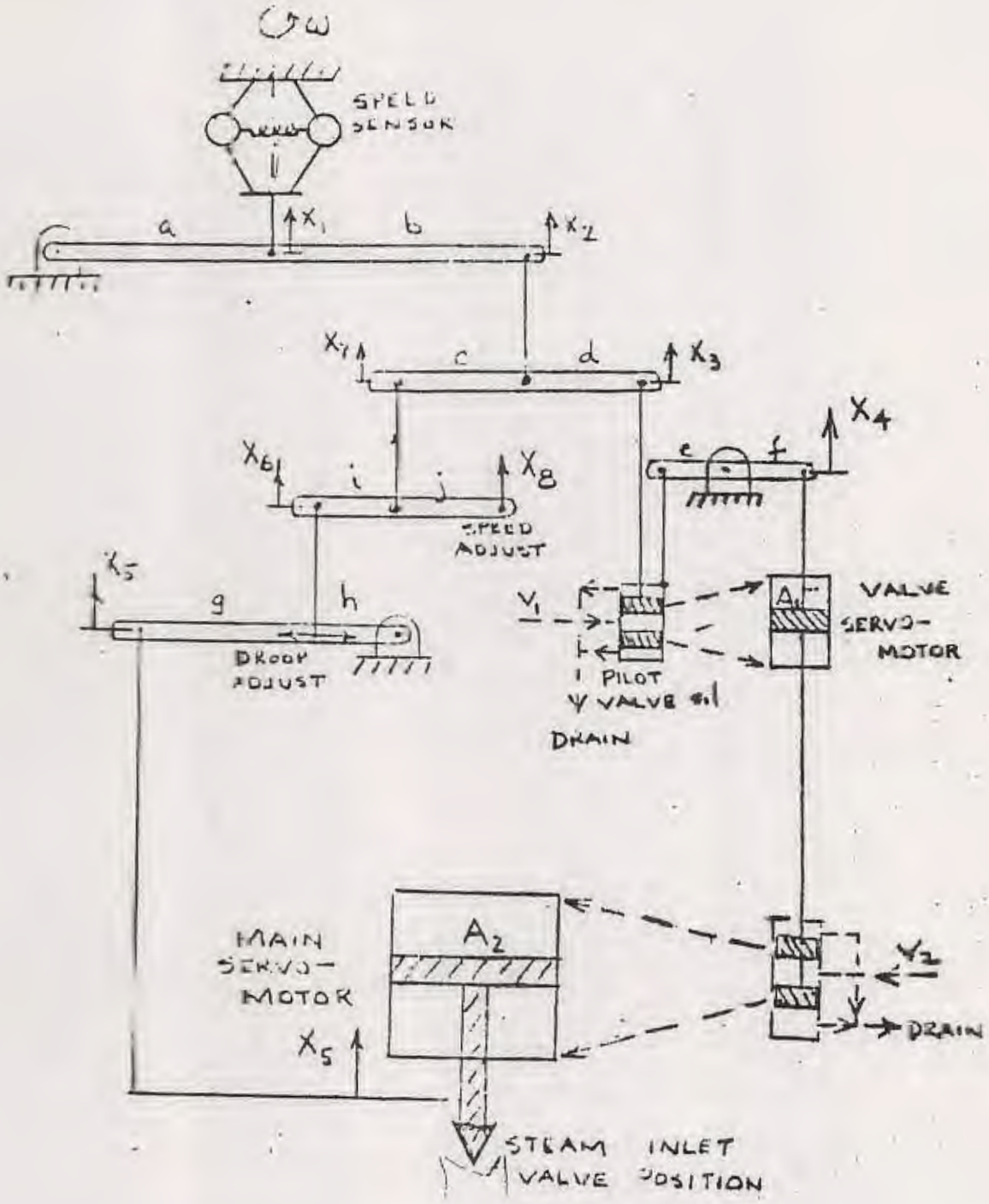


Figure 2.16 Mechanical-hydraulic steam turbine governor

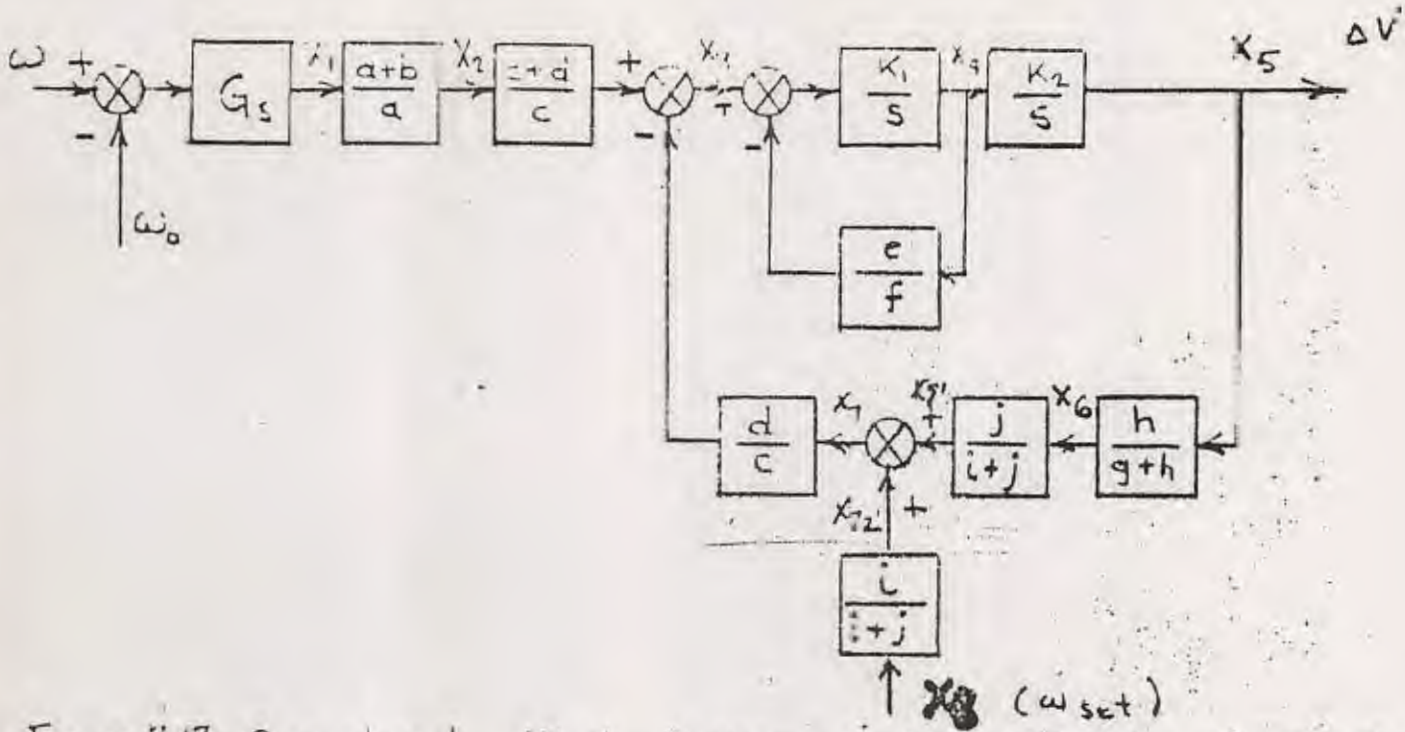


Figure 2.17 Operational Block Diagram - Steam Turbine Governor

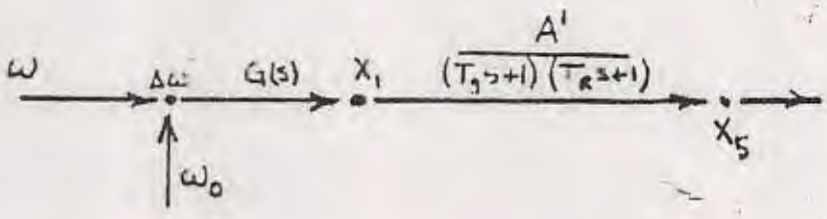


Figure 2.18

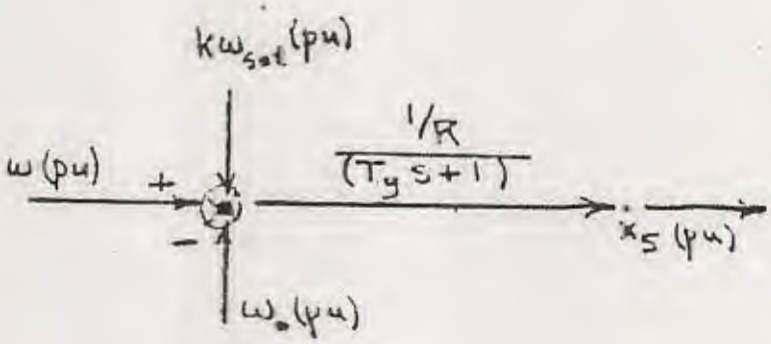


Figure 2.19

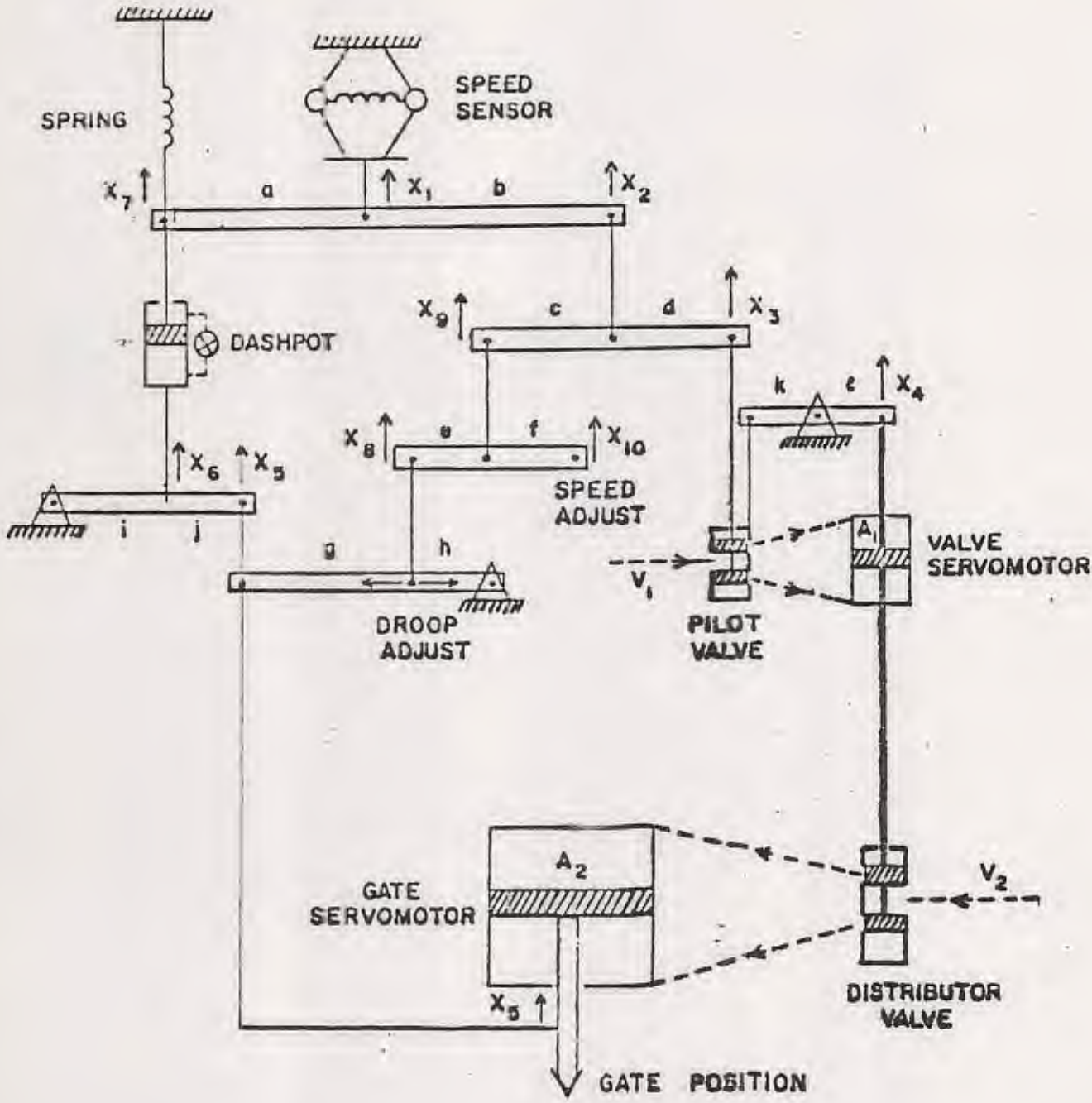


Figure 2·20 - Schematic of a temporary droop governor for Hydraulic turbines

The dotted lines in Figure 2.20 represent oil flow lines and the solid lines represent mechanical linkages such as shafts, cables or rods.

Figure 2.21 is a signal flow graph representation of the governor shown in Figure 2.20. All branches in the flow graph are defined by linkage ratios except for the following:

(i) branch $\frac{\Delta\omega}{\omega_o}(S) \rightarrow X_1(S) = G_c(S)$ which is the transfer function of the flyball speed sensor head. For small speed variations

$$G_c(S) = \frac{G_c'}{s^2 + 2\xi\omega_n S + \omega_n^2}$$

Where ω_n is the natural frequency of the speed sensor dependant on the spring constant and flyball inertias. ξ is the damping ratio and G_c' is an appropriate constant. This transfer function is usually considered to be a constant equal to G_c'/ω_n^2 since ω_n is normally considerably higher than any of the other components in the frequency control loop.

(ii) branch $(y(S) \rightarrow x_4(S)) = k_1/S$ which is the transfer function relating the pilot valve opening (y) to the valve servometer displacement (x_4). The constant k_1 depends directly on the oil flow rate V_1 and inversely on the cross-sectional area of the valve servometer piston.

(iii) branch $(x_4(S) \rightarrow x_5(S)) = k_2/S$ which is the transfer function relating the distributor valve opening (x_4) to the displacement of the gate servomotor (x_5). The constant k_2 depends directly on the oil flow rate V_2 and inversely on the cross-sectional area of the gate servomotor piston.

(iv) branch $(x_6(S) \rightarrow x_7(S)) = G_d(S)$ which is the transfer function of the reset dashpot mechanism. To evaluate $G_d(s)$ refer to Figure 2.20. At the junction of the spring and dashpot there will be a balance of forces thus

$$D_d S(x_6(S) - x_7(S)) = K_s x_7(S)$$

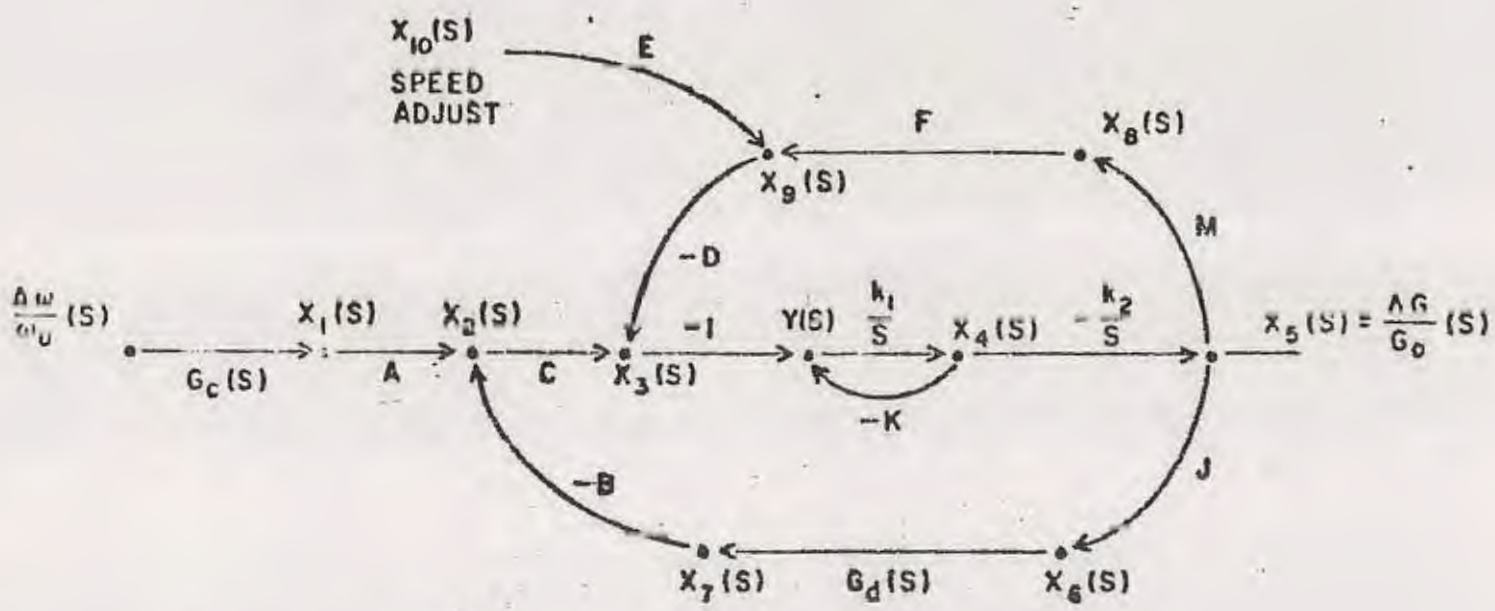
$$G_d(S) = \frac{x_7(S)}{x_6(S)} = \frac{SD_d}{K_s + D_d S} = \frac{SD_d/K_s}{D_d/K_s + 1} = \frac{T_r S}{T_r S + 1}$$

where D_d is the damping coefficient of the dashpot,

K_s is the spring coefficient,

and $D_d/K_s = T_r$ is the relaxation time constant.

The signal flow graph shown in Figure 2.21 can be reduced to that of Figure 2.22. A comparison of Figure 2.22 indicates that the permanent droop (σ) of the governor is given by



$A = (a+b)/a$

$B = b/a$

$C = (e+d)c$

$D = d/c$

$E = e/(e+f)$

$F = f/(e+f)$

$M = h/(g+h)$

$J = 1/(1+j)$

$K = k/\ell$

$G_d(S) = T_r S/(T_r S+1)$

$G_c(S) \approx G_c'/\omega_n^2$

Figure 2.21 - Signal flow graph of a temporary droop governor

$$\sigma = \frac{DFM}{ACG_c(S)} = \frac{d\ f\ h\ a}{(e+f)\ (g+h)\ (a+b)\ (c+d)\ G_c(S)}$$

and the temporary droop feedback path is given by

$$\frac{\delta T_r S}{(T_r S+1)} = \frac{S\ G_c(S)\ J\ T_r\ B/A}{(T_r S + 1)}$$

where

$$\delta = G_c(S)\ JB/A = \frac{bi}{(a+c)(i+j)\ G_c(S)}$$

To determine the transfer function of the forward path of the temporary droop governor refer to Figures 2.21 and 2.22. If the pilot valve-valve servomotor combination is fast compared with the distributor valve-gate servomotor combination; i.e. $k_1 >> K_2$, then the transfer function for $\frac{x_5}{x_3}$ (S) is given by

$$\frac{x_5}{x_3}(S) = \frac{ACG_c(S)k_2/K}{S(S/(k_1K) + 1)} = \frac{ACG_c(S)/K}{S(S/Kk_1k_2) + 1/k_2)} \quad \frac{k_2ACG_c(S)/K}{S}$$

$$= \frac{1}{T_s S}$$

where

$$T_s = \frac{K}{k_2ACG_c(S)}$$

Figure 2.23 illustrates the governor transfer function used in these notes; it is expressed by the relationship

$$\frac{\Delta G/G}{\Delta\omega/\omega_o} = \frac{(1 + T_R S)}{T_R T_S S^2 + (T_S + \delta T_R + \sigma T_R)S + \sigma}$$

2.8.4 Electro-hydarulic governor for water turbines (ASEA type)

The basic measuring and controlling operations of a governor can be carried out using electrical methods. One method (used by ASEA) makes use of power frequency a-c servomechanism techniques for the computing part of the governor and hydraulic techniques for power gain.

A simplified drawing of such a governor appears in Figure 2.24. The electrical part of the governor initiates action in the hydraulic part when the input to the operational amplifier A is unbalanced. Input X_1 is the temporary droop input and input X_2 is the input point for the various quantities considered in the "summary loop". Input X_2 to the operational amplifier is zero if the voltage across T_3 primary is zero. In the summing loop the voltage drop across C_1 is proportional to the frequency deviation away from reference frequency (i.e. 60 Hz), the drop across T_1 is proportional to the offset frequency desired, the drop across

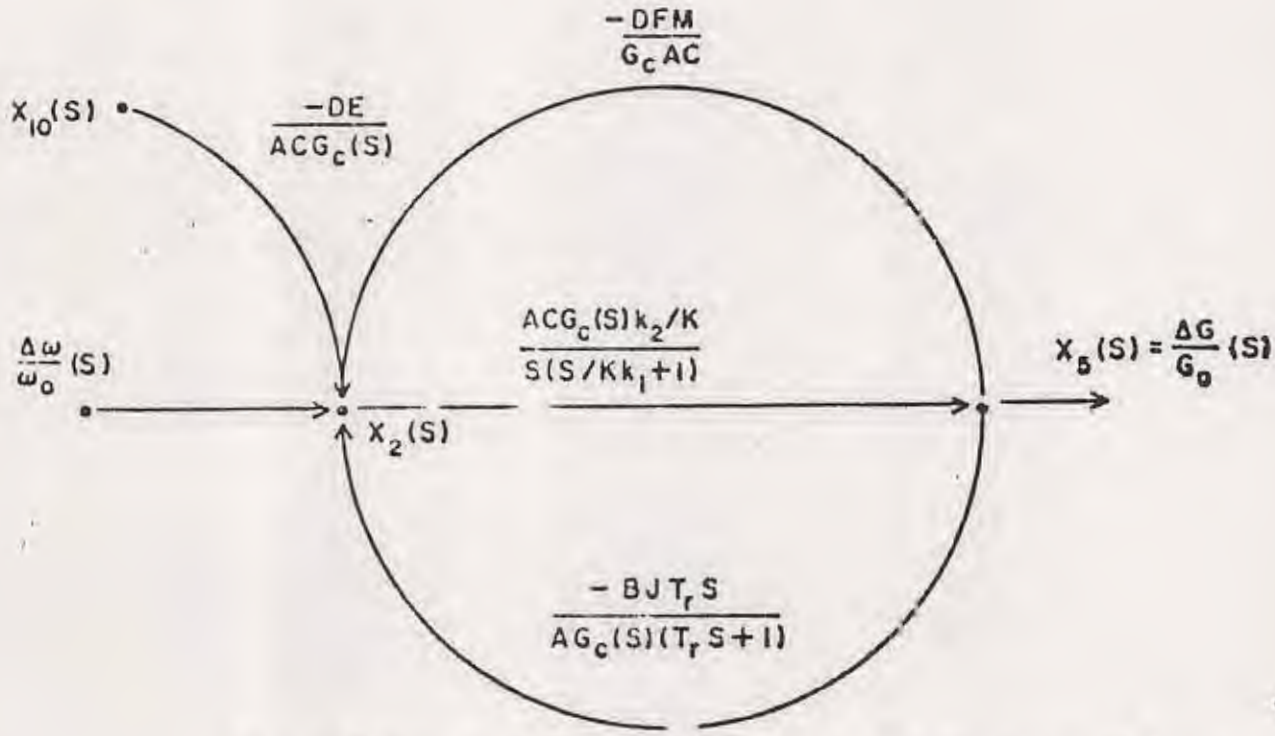


Figure 2.22 - Reduced signal flow graph of a temporary droop governor

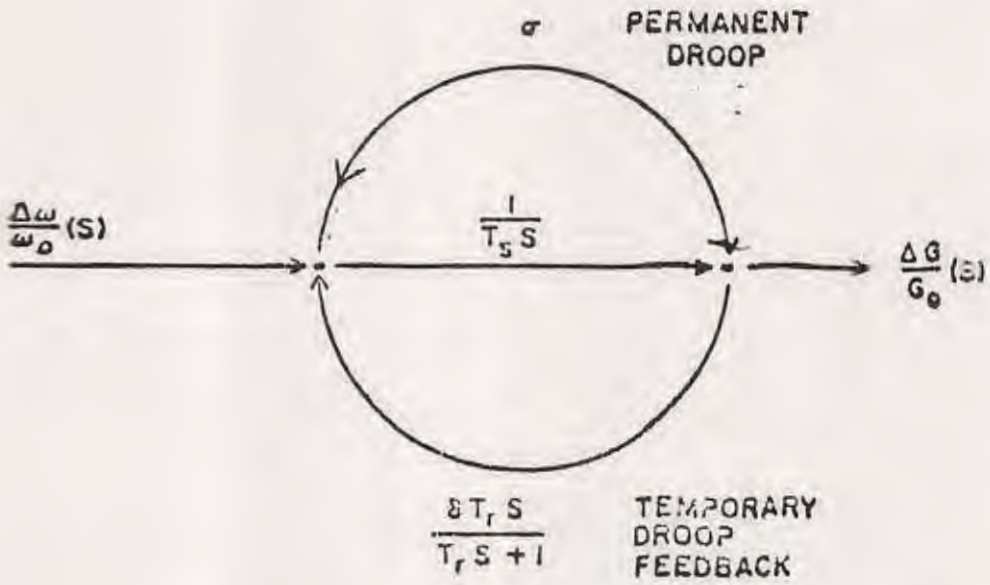


Figure 2.23 - Conventional signal flow representation of a temporary droop governor

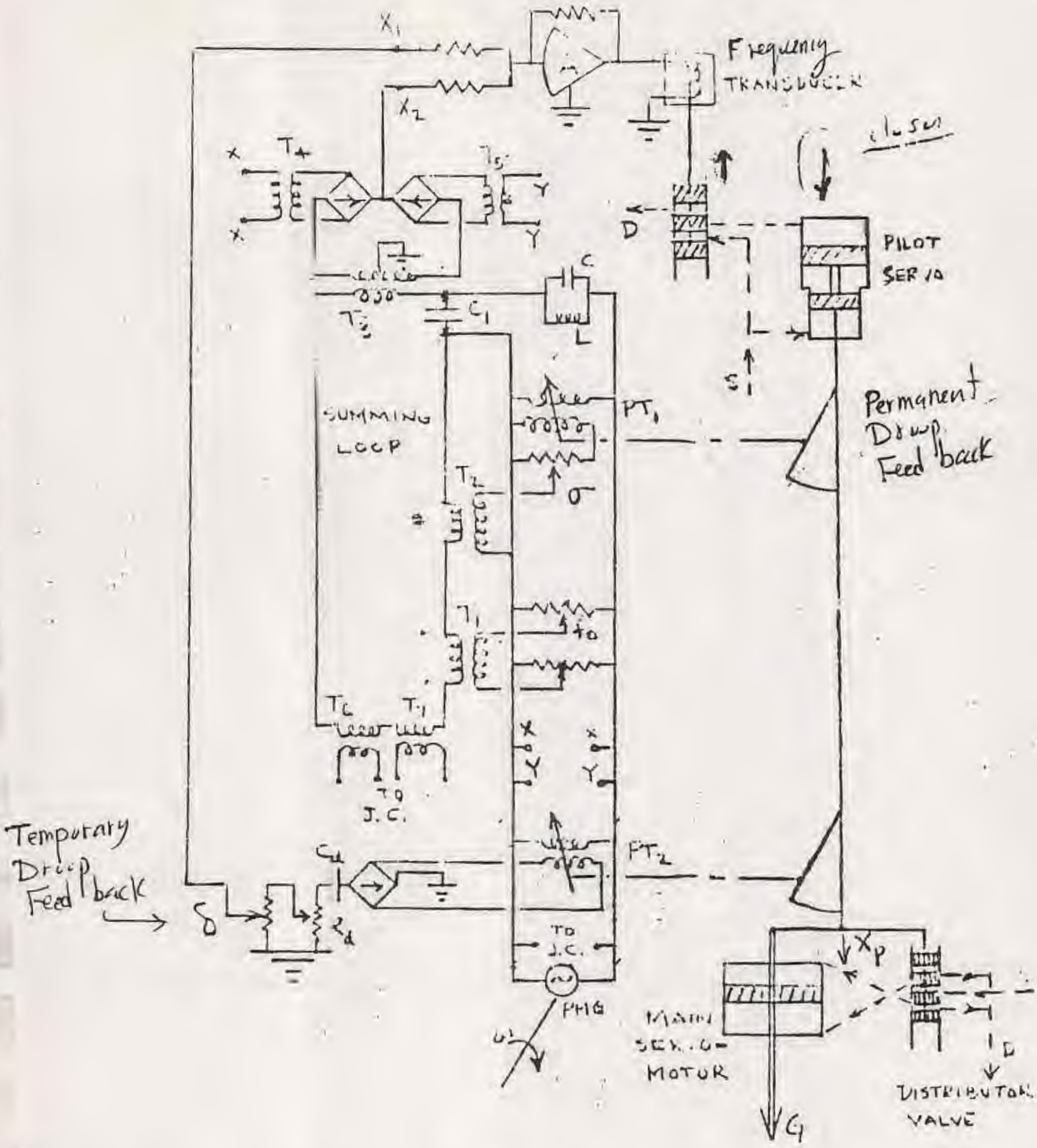


Figure 1.24 A.C. Servo type electrohydraulic governor

T_7 is proportional to the power output of the turbine (i.e. permanent droop input s) and the voltage drops across T_6 and T_7 are determined by the joint control circuit which acts to maintain a specified power division amongst the total units running in a plant. The permanent magnet generator PMG supplies a frequency proportional to turbine shaft angular velocity. Normal power frequency is used so that normal power supply can be used for static test purposes.

A simplified sketch of the joint control circuitry for an ac servo governor is shown in Figure 2.25 for a three machine plant. The potentimeters R_1, R_2, R_3, R_4, R_5 and R_6 are linked mechanically to the gate servo shafts of the three generators R_1 and R_2 to unit number 1, R_3 and R_4 to unit number 2 and R_5 and R_6 to unit number 3. The voltages on the secondaries of transformers T_a, T_b, T_c, T_e, T_f and T_g will therefore be proportional to the power outputs of the units with which they are associated. The potentiometer R_o is set to give an output across T_d proportional to the desired total power output from the station and potentiometer R_7 is set to give an overall drooping characteristic for the entire plant. The lower summation loop in the schematic diagram provides an output from transformer T_i if the actual summation of power from all three units is not equal to the demanded output from the station biased for the desired steady state speed droop. The outputs from T_i go to the T_7 inputs of the three unit governors (see Figure 2.24).

The primaries of T_a, T_b, T_c are connected in parallel so that their outputs will be zero provided all three units share the load changes proportionately. The outputs of T_a, T_b and T_c go to the T_6 inputs of the three unit governors (see Figure 2.24).

The ac servo type electrohydraulic governor can be reduced to the block diagram of Figure 2.26. The Joint Control Circuitry provides corrective action in the event that an individual unit does not pick up its share of a load demand on the station. This action is not strong however, and may be neglected in assessing the actual dynamic control performance of a governor. If the joint control path is omitted, the transfer function of the governor can be obtained from Figure 2.26. The result is

$$\frac{X_p}{\Delta f}(s) = \frac{(1 + sT_R)}{(1+sT_f)\ \{T_s T_R s^2 + (T_s + \delta T_R + \sigma T_R)s + \sigma\}}$$

The relationship between the pilot servo shaft postion X_p and the gate servo shaft position G is given by

$$\frac{G}{X_p}(s) = \frac{1}{1 + sT_g}$$

It is to be noted that if the T_g and T_f time constants are small compared with T_R then the transfer function

$$\frac{G}{\Delta f}(s) = \frac{(1 + sT_R)}{T_R T_S s^2 + (T_S + \delta T_R + \sigma T_R)s + \sigma}$$

which is the same as that of the mechanical-hydraulic governor.

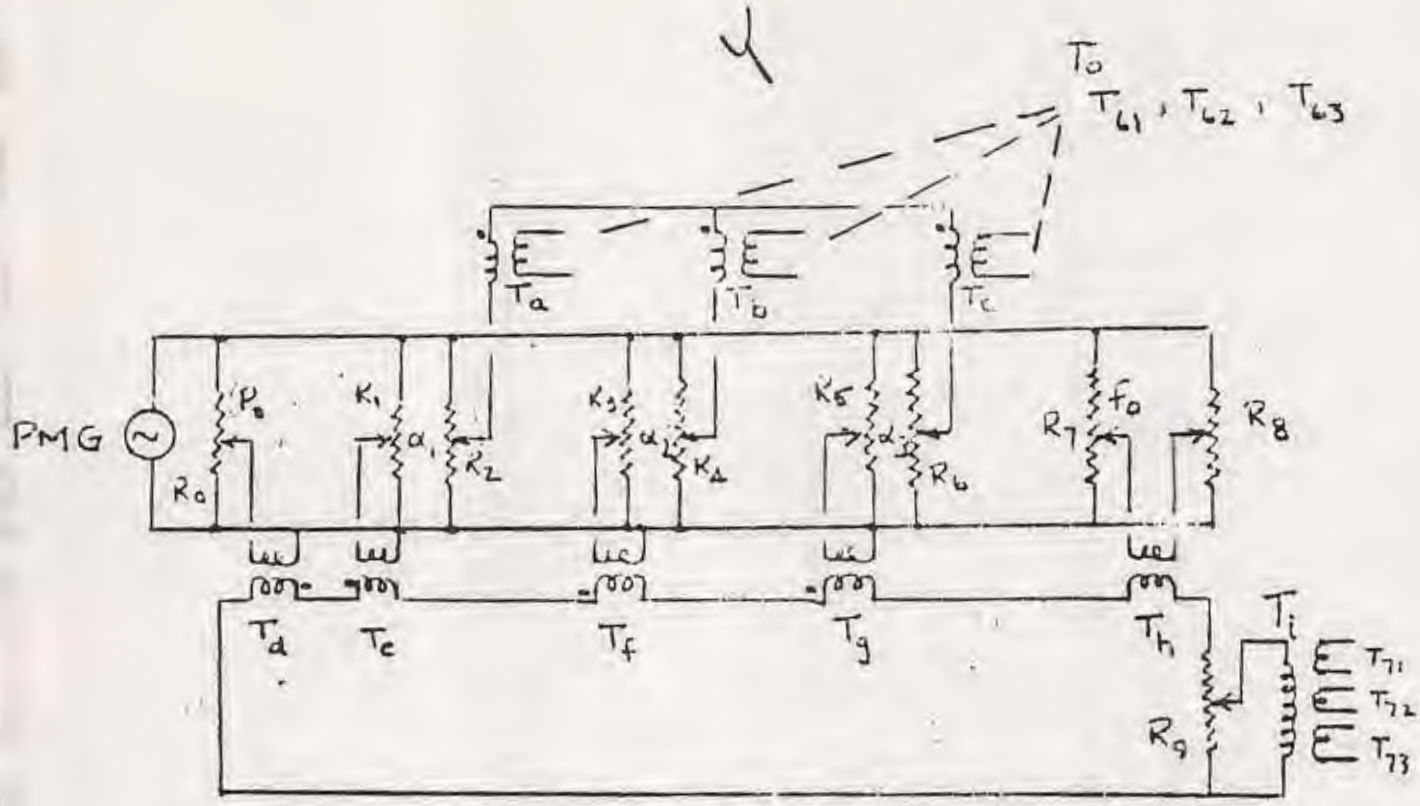
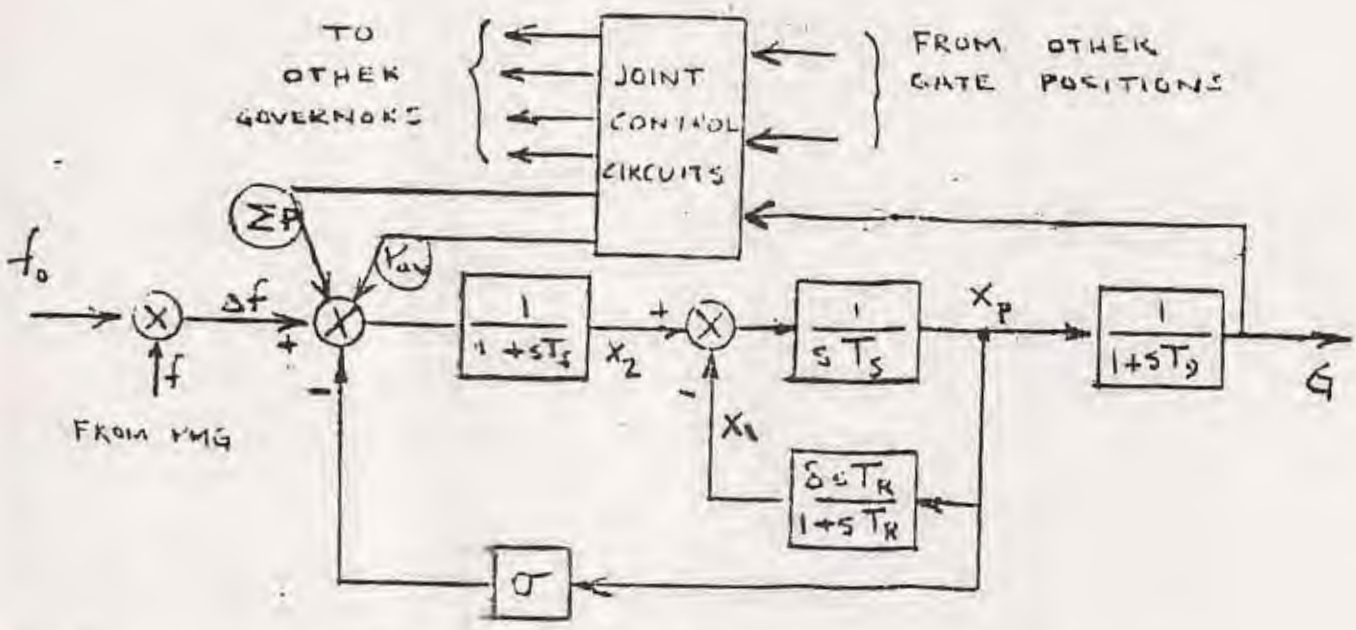


Figure 2.25 Joint Control Scheme For A.C. Servo Governor



2.26 Operational Block Diagram of A.C. Servo Governor

2.8.5 Electronic-hydraulic Governor for Water Turbines (Woodward Governor Co. Type)

Electronic circuits using semiconductor devices are very reliable and can be used to advantage in governor design. They offer a number of significant advantages in that supervisory control signals and protection signals can be very easily introduced into the basic electronic circuits. The Woodward Transistorized Electronic Governor which is described briefly hereunder is an example of this type of governor.

In this governor (see Figure 2.27) the speed is sensed by a magnetic pick-up which generates a series of pulses at a rate proportional to the angular velocity of the turbine shaft. The speed sensing circuit compares the actual speed ω with the desired value and sends a dc error signal proportional to $\Delta\omega$ to the electronic amplifier. Proportional plus derivative feedback is applied around the amplifier and these may be adjusted to alter the dynamic performance of the governor. The output of the amplifier feeds an electro-hydraulic transducer. The output of the transducer is amplified hydraulically to the power level required by the wicket gate servo motor. The governor droop characteristic is obtained from the measured electric power output of the generator.

Leum(*) has shown that the output voltage of the electronic amplifier is given by

$$e = \frac{1}{k}\left(\frac{s}{\beta} + 1\right)[\Delta\omega - RP]$$

where e is the electronic amplifier output

k is the gain adjustment of the electronic amplifier

β is the time constant adjustment of the electronic amplifier

$\Delta\omega$ the frequency error

R the speed regulation

P the electric power output and

S the Laplace operator

normally $0.02 \leq k \leq .2$

$1.0 \leq \beta \leq \infty$

$0 \leq R \leq 0.10$

The electro hydraulic transducer performance can be represented by the relationship

$$s\Omega = 0.057\,(s + \alpha)e$$

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* M. Leum "The Development and Field Experience of a Transistor Electric Governor for Hydro Turbines", Trans Power App & Syst, Apr/66, I.E.E.E. TP 65-666

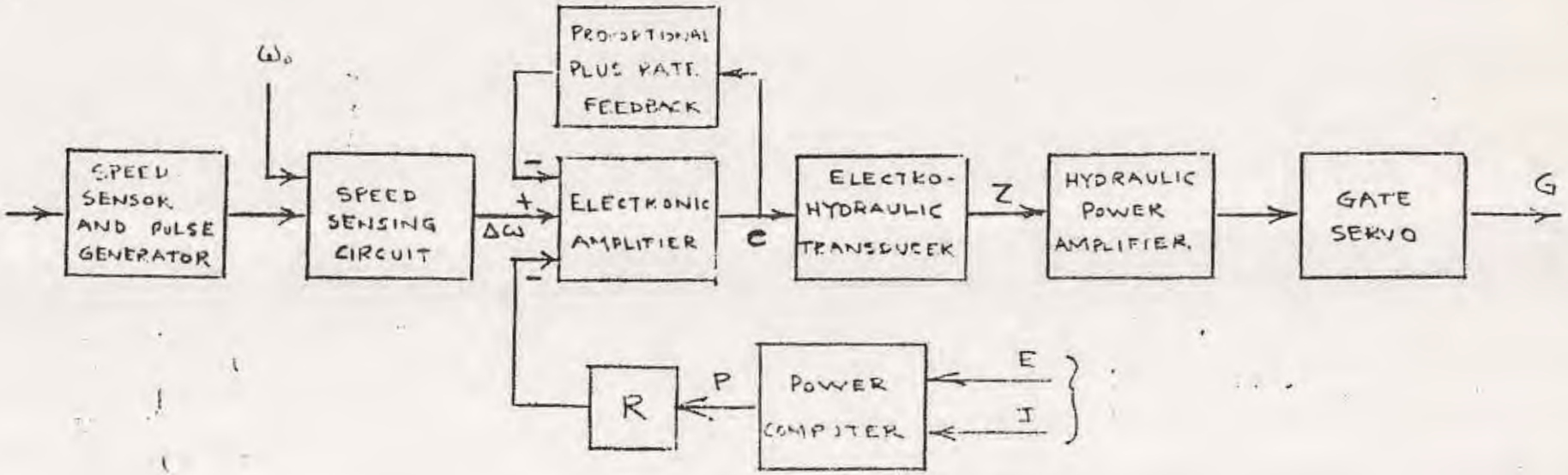

Figure 2.27

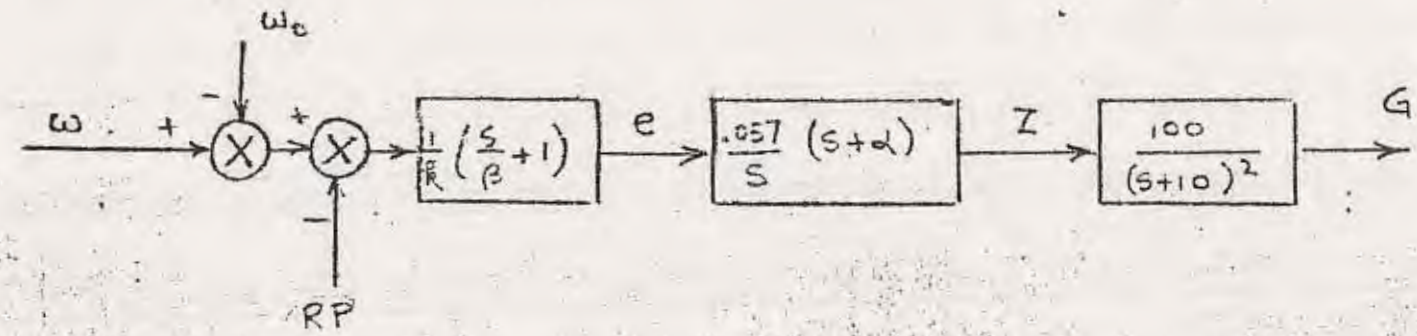

Figure 2.28