APPENDIX -A1

MATHEMATICAL MODELING OF SINGLE AREA AGC

To understand the load frequency control problem, let us consider a single turbo-generator system supplying an isolated load.

1. TURBINE SPEED GOVERNING SYSTEM MODEL

Figure A.1 shows schematically the speed governing system of a steam turbine. The system consists of the following components:

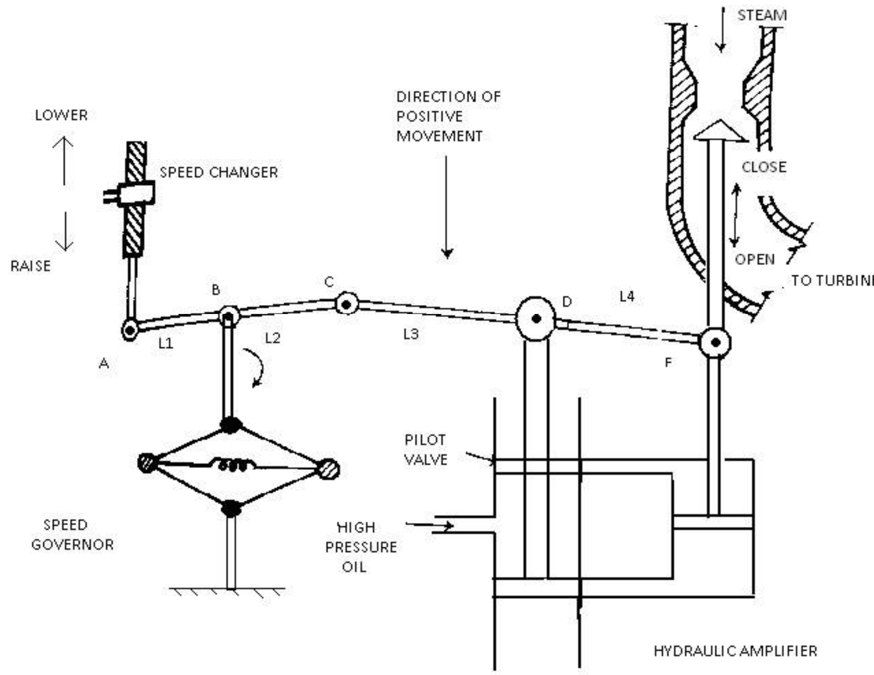
(1) Fly ball speed governor: This is the heart of the system, which senses the change in speed (frequency). As the speed increases the fly balls move outwards and the point B on linkage mechanism moves downwards. The reverse happens when the speed decreases.

(2) Hydraulic Amplifier: It comprises a pilot valve; movement is converted into high power level piston valve movement. This is necessary in order to open or close the steam valve against high-pressure steam.

(3) Linkage mechanism: ABC is a rigid link pivoted at B and CDE is another rigid link pivoted at D. This link mechanism provides a movement to the control valve in proportion to change in speed. It also provides a feedback from the steam valve movement (link 4).

(4) Speed changer: It provides a steady state power output setting for the turbine. Its downward movement opens the upper pilot valve so that more steam is admitted to the turbine under steady conditions (hence more

(5) steady power output). The reverse happens for upward movement of speed changer.



Assume that the system is initially operating under steady conditions, the linkage mechanism stationary and pilot valve closed, steam valve opened by a definite magnitude, turbine running at constant speed with turbine power output definite magnitude, turbine running at constant speed with turbine power output balancing the generator load. Let the operating conditions be characterized by balancing the generator load. Let the operating conditions be characterized by :

F^0 = system frequency (speed)

P^0g = generator output = turbine output

Y^0e = steam valve setting

We shall obtain a linear incremental model around these operating conditions.

Let the point A on the linkage mechanism be moved downwards by a small amount ΔYa. It is a command which causes the turbine power output to change and can, therefore, be written as

$$\Delta Ya = Kc\Delta Pc \qquad (A1.1)$$

Where ΔPc is the commanded increase in power:

The command signal ΔPc (i.e. ΔYe) sets into motion a sequence of events- the pilot valve moves upwards, high pressure oil flows on to the top of the main piston moving it downwards; the steam valve opening consequently increase, the turbine generator speed increase, i.e., the frequency goes up. Let us model these events mathematically.

Two factors contribute to the movement of C:

(1) ΔYa contributes-(L_2/L_1) ΔYa or $-k_1\Delta Ya$ (i.e. upwards) of $-k_1k_c$ ΔPc

(2) Increase in frequency Δf causes the fly balls to move outwards so that B moves downwards by a proportional amount $K_2\Delta f$. the consequent movement of C with A remaining fixed at

(3) ΔYa is + $(L_1+L_2/L_1)K_2\Delta f$ = + k'_2 Δf (i.e. downwards)

The net movement of C is therefore

$$\Delta Yc = - K_1 Kc \Delta Pc + K_2 \Delta f \qquad (A1.2)$$

The movement of D, ΔYd, is the amount by which the pilot valve opens. It is contributed by ΔYc and ΔYe and can be written as

$$\Delta Y_d = L_4/(L_3+L_4)\ \Delta Yc + L_3/(L_3+L_4)\ \Delta Ye \quad = K_3\Delta Yc + K_4 \Delta Ye \qquad (A1.3)$$

The movement ΔY_d depending upon its sign opens one of the ports of the pilot valve admitting high-pressure oil into the cylinder thereby moving the main piston and opening the steam valve by ΔYe. Certain justifiable simplifying assumptions, which can be made at this stage, are:

(1) Inertial reaction forces of main piston and steam valve are negligible compared to the forces exerted on the piston by high-pressure oil.

(2) Because of (i) above, the rate of oil admitted to the cylinder is proportional to port opening ΔY_d.

The volume of oil admitted to the cylinder is thus proportional to the time integral of ΔY_d.The movement ΔYe is obtained by dividing the oil volume by the area of the cross section of the piston.

Thus

$$\Delta Ye = k_5 \int_0^t (-\Delta y_d)\,dt \qquad (A1.4)$$

It can be verified from the schematic diagram that a positive movement ΔYd, causes negative (upward) movement ΔYe, accounting for the negative sign used in Eq. (A1.4).

Taking the laplace transform of equations (A1.2), (A1.3) & (A1.4). We get

$\Delta Yc(s) = -K_1Kc\Delta Pc(s) + K_2\Delta F(s)$

$\Delta Yd(s) = -K_3\Delta Yc(s) + K_4\Delta Ye\ (s)$

$\Delta Ye(s) = -K_5\ 1/S\ \Delta Y_d(s)$

Eliminating $\Delta Yc(s)$ and $\Delta Y_d(s)$, we can write

$\Delta Ye(s) = K_1K_3kc\Delta Pc(s)-K_2K_3\Delta F(s)$

$(K_4+(s/K_5))= [\Delta Pc(s) - 1/R\Delta F(s)] * (Ksg/(1+TsgS))$ (A1.5)

Where

$R = K_1K_c/K_2$ = speed regulation of the governor

$Ksg = K_1K_3K_c/K_4$ = gain of speed governor

$Tsg = 1/K_4K_5$ = time constant of speed governor

The speed governing system of a hydro-turbine is more involved. An additional feedback loop provides temporary droop compensation to prevent instability. This is necessitated by the large inertia of penstock gate, which regulates the rate of water input to the turbine.

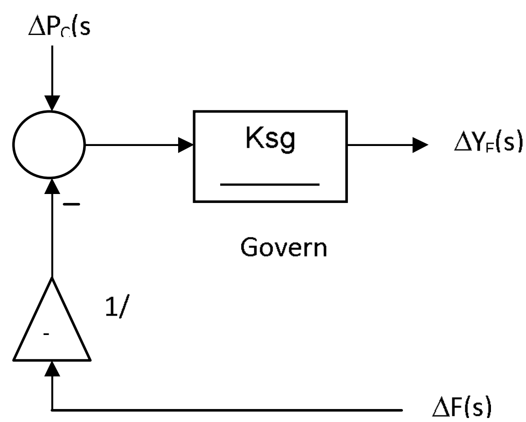


Fig. A1.1 Block diagram representation of speed governor system

2. TURBINE MODEL

Let us now relate the dynamic response of the steam turbine in terms of changes in power output to changes in steam valve opening ΔYe. Here two-stage steam turbine with reheat unit is used. The dynamic response is largely influenced by two factors, (i) entrained steam between the inlet steam valve and first stage of the turbine, (ii) the storage action in the reheated which causes the output if the low pressure stage to lag behind that of the high pressure stage. Thus, the turbine transfer function is characterized by two time constants. For ease of analysis, it will be assumed here that the turbine can be modeled to have ease of analysis it will be assumed here that the turbine can be modeled to have a single equivalent time constant.

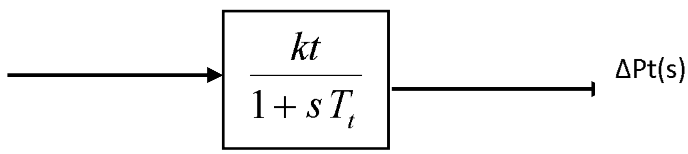


Fig. A1.2 Turbine transfer function model

3. GENERATOR LOAD MODEL

The increment in power input to the generator-load system is ΔPg-ΔPd

Where $\Delta Pg = \Delta Pt$, incremental turbine power output (assuming generator incremental loss to be negligible) and ΔPd is the load increment.

This increment in power input to the system is accounted for in two ways:

(i) Rate of increase of stored kinetic energy in the generator rotor at scheduled frequency (f^o), the stored energy is

$$W^o ke = H * Pr\ KW = sec\ (kilojoules) \quad (A1.6)$$

Where Pr is the KW rating of the turbo-generator and H is defined as its inertia constant.

the kinetic energy being proportional to square of speed (frequency), the kinetic energy at a frequency of ($f^o + f^o\ \Delta f$) is given by

$$Wke = W^o ke\ ((f^o + \Delta f)/f^o)^2$$

$$\approx H\ Pr\ (1 + (2\Delta f/ f^o)) \quad (A1.7)$$

Rate of change of kinetic energy is, therefore

$$d / dt\ (Wke) = 2HPr/ f^o * d/dt\ (\Delta f) \quad (A1.8)$$

(ii) As the frequency changes, the motor load changes being sensitive to speed, the rate of change of load with respect to frequency, i.e. $\partial Pd/\partial f$ can be regarded as nearly constant for small changes in frequency Δf and can be expressed as $(\partial Pd/\partial f)\ \Delta f = B\ \Delta f$

There the constant B can be determined empirically. B is positive for a predominantly motor load.

Writing the power balance equation, we have

$$\Delta Pg - \Delta Pd = (2HPr/ f^o) * d/dt\ (\Delta f) + B\Delta f$$

Dividing throughout by Pr and rearranging, we get

$$\Delta Pg(pu) - \Delta Pd(pu) = (2H/ f^o) * d/dt\ (\Delta f) + B(pu)\ \Delta f$$

Taking the Laplace transform, we can write $\Delta F(s)$ as

$$\Delta F(s) = \frac{\Delta Pg(s) - \Delta Pd(s)}{B + (2H/ f^o)s} = [\Delta Pg(s) - \Delta Pd(s)] * (Kps/(1+ sTps)) \qquad (A1.9)$$

Where

$Tps = 2H/B\ f^o$ = power system time constant

$Kps = 1/B$ = power system gain

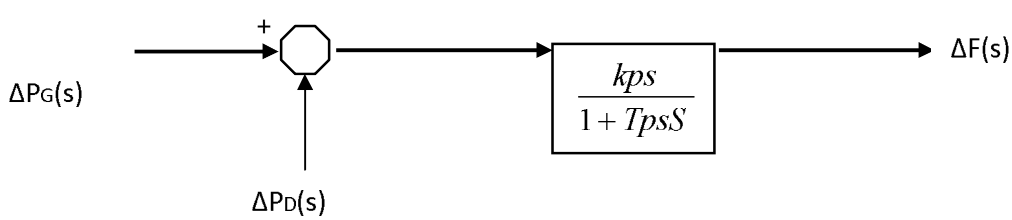


Fig.A1.3 Block diagram representation of generator-load model