CMR Institute of Technology, Bangalore
DEPARTMENT OF ELECTRICAL & ELECTRONICS ENGINEERING
II - INTERNAL ASSESSMENT

Semester: 8-CBCS 2017
Subject: POWER SYSTEM OPERATION & CONTROL (17EE81)
Faculty: Ms Sanitha

Date: 19 Jun 2021
Time: 09:00 AM - 10:30 AM
Max Marks: 50

Instructions to Students :

Answer any 5 questions.Each question carries 10 marks.

Answer any 5 question(s)

Q.No		Marks	CO	PO	BT/CL
1	Explain the functions of various components in a steam turbine governing system with necessary diagram.	10	CO3	PO2	L2
2	Derive the transfer function and the block diagram of complete ALFC loop.	10	CO3	PO3	L3
3	Draw the block diagram of two area system with necessary equations.	10	CO3	PO3	L3
4	What are tie line oscillations? What determines the frequency of these oscillations?	10	CO3	PO1	L1
5	Two control areas of capacity 500 MW and 10000 MW are connected through a tieline. The parameters of each area on its own capacity are R= 1 Hz/pu MW and D= 0.02 pu MW/Hz. There is an increase of 200 MW of load in area 2.Determine the steady state frequency deviation and change in tie line flow.	10	CO3	PO3	L3
6	Two control areas are connected via a tie line with the following characteristics . Area 1: R1=1 %, D1=0.8 pu , base MVA 1000 Area 2: R2=2 %, D2=1.0 pu , base MVA 1000 A load increase of 100 MW occurs in area 1.What is the new steady state frequency and the change in tie line flow if the nominal frequency is 50 Hz. Repeat if the load changes in area 2	10	CO3	PO3	L4

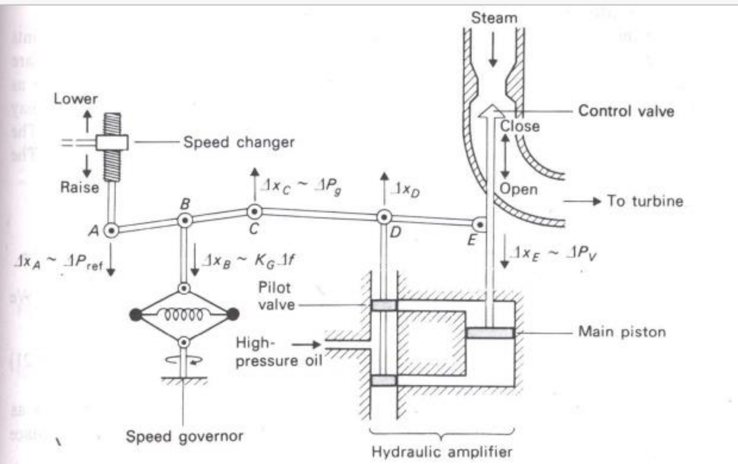

Figure 9-7 Simplified functional diagram of the primary ALFC loop.

megawatt increment ΔP_V. This flow increase translates into a turbine power increment ΔP_T in the turbine (not shown in the figure).

Very large mechanical forces are needed to position the main valve (or gate) against the high steam (or water) pressure, and these forces are obtained via several stages of hydraulic amplifiers. In our simplified version we show only one stage. The input to this amplifier is the position x_D of the *pilot valve*. The output is the position x_E of the *main piston*. Because the high-pressure hydraulic fluid exerts only a slight differential force on the pilot valve, the force amplification is very great.

The position of the pilot valve can be affected via the linkage system in three ways:

1. *Directly*, by the *speed changer*. A small downward movement of the linkage point A corresponds to an increase ΔP_{ref} in the reference power setting.
2. *Indirectly*, via feedback, due to position changes of the main piston.
3. *Indirectly*, via feedback, due to position changes of linkage point B resulting from speed changes.

It should prove a useful exercise for the reader to find, *qualitatively*, the workings of the mechanism. For example, give a "raise" command to the speed changer and prove that this indeed results in an increase in turbine output. Prove also that a speed drop will give the same effect.

measured in millimeters but in our analysis we shall rather express them a *power increments* expressed in megawatts or per-unit megawatts as the case ma be. The movements are assumed positive in the directions of the arrows. Th governor *output command* ΔP_g is measured by the position change Δx_C. Th governor has two inputs:

1. Changes ΔP_{ref} in the reference power setting
2. Changes Δf in the speed of frequency of the generator, as measured by Δx_B

An *increase* in ΔP_g results from an *increase* in ΔP_{ref} *and* a *decrease* in Δf. W thus can write for *small* increments

$$\Delta P_g = \Delta P_{\text{ref}} - \frac{1}{R}\,\Delta f \qquad \text{MW} \tag{9-21}$$

The constant R has dimension hertz per megawatt, and is referred to a *regulation* or *droop*. (For numerical values see Example 9-2 below.) Laplac transformation of Eq. (9-21) yields

$$\Delta P_g(s) = \Delta P_{\text{ref}}(s) - \frac{1}{R}\,\Delta f(s) \tag{9-22}$$

Using well-known block diagram symbols we have represented the governo as shown in Fig. 9-8.

9-3-2 Hydraulic Valve Actuator

The input position Δx_D of the valve actuator increases as a result of an increase command ΔP_g but decreases due to increased valve output, ΔP_V. Equal in creases in both ΔP_g and ΔP_V should result in $\Delta x_D = 0$. We can thus write

$$\Delta x_D = \Delta P_g - \Delta P_V \qquad \text{MW} \tag{9-23}$$

For small changes Δx_D the oil flow into the hydraulic motor is proportiona to position Δx_D of the pilot valve. Thus we obtain the following relationship fo the position of the main piston:

$$\Delta P_V = k_H \int \Delta x_D \, dt \tag{9-24}$$

The positive constant k_H depends upon orifice and cylinder geometries an fluid pressure.

Upon Laplace transformation of the last two equations and upon elimina tion of Δx_D we obtain the actuator transfer function

$$G_H(s) = \frac{\Delta P_V}{\Delta P_g} = \frac{1}{1 + sT_H} \tag{9-25}$$

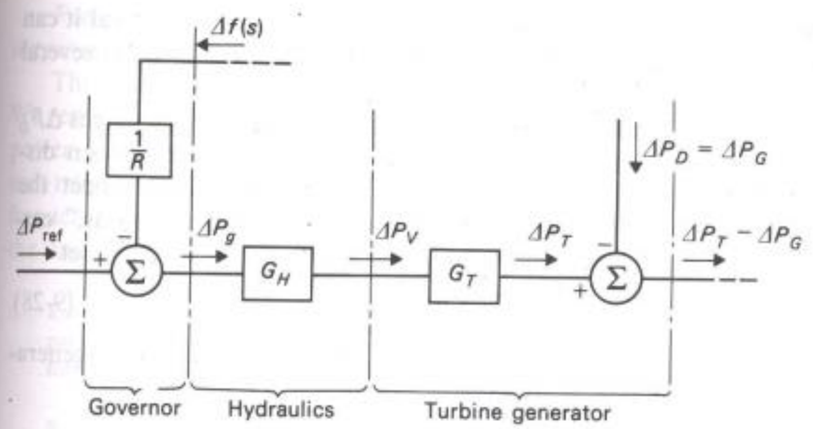

Figure 9-8 Linear model of the primary ALFC loop (minus the power system response).

where the *hydraulic time constant*

$$T_H = \frac{1}{k_H}$$

typically assumes values around 0.1 s.

The hydraulic valve actuator has been represented by the transfer function $G_H(s)$ in Fig. 9-8.

9-3-3 Turbine-Generator Response

In normal steady state and via the mechanism described in Sec. 4-9 the turbine power P_T keeps balance with the electromechanical air-gap power P_G resulting in zero acceleration and a constant speed or frequency. Perturbations ΔP_T and ΔP_G in these powers will upset the above balance. If the difference power, $\Delta P_T - \Delta P_G$, is positive the turbine generator unit will accelerate; if negative it will decelerate.

The turbine power increment ΔP_T depends entirely upon the valve power increment ΔP_V and the response characteristics of the turbine. Different turbine types vary widely in this regard. It is possible to express the turbine dynamics in terms of a turbine transfer function

$$G_T = \frac{\Delta P_T}{\Delta P_V} \tag{9-26}$$

In App. D we have derived G_T for the most common turbine types. A so-called *nonreheat steam turbine* has the simplest transfer function, consisting of a single time constant, i.e.,

$$G_T = \frac{1}{1 + sT_T} \tag{9-27}$$

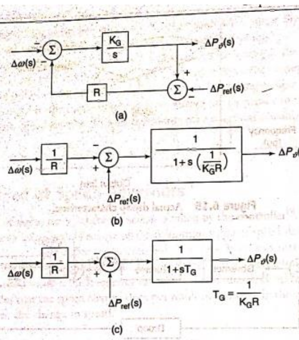

Figure 6.18 Model of governor.

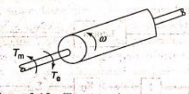

Figure 6.19 Torques acting on a generator.

Figure 6.18(b) and (c) shows how the block diagram can be reduced. T_G is the *governor time constant*. It can be seen that it depends on the speed regulation R and on the gain of the hydraulic amplifier, K_G.

6.6.2 Generator Model

There are two torques acting on a generator: the *shaft torque* (due to the prime mover) and the *electromagnetic torque*, neglecting losses. The shaft torque tends to accelerate the generator in the positive direction of rotation and the electromagnetic torque in the negative direction, as shown in Fig. 6.19.

The total accelerating torque is given by

$$T_a = T_m - T_e \tag{6.6}$$

From Newton's laws of motion, we have for rotatory motion

$$I\alpha = T \tag{6.7}$$

where I is the moment of inertia,
α is the angular acceleration,
T is the net torque.
Equation (6.7) can be written as

$$I\frac{d^2\theta_m}{dt^2} = T_m - T_e \tag{6.8}$$

where θ_m, the rotor angle, is now converted into an angle, measured with respect to a synchronously rotating reference axis such that

$$\delta_m = \theta_m - \omega_{sm}t \tag{6.9}$$

where ω_{sm} is the synchronous speed in rad/s,
δ_m is the angular displacement in rad.
From Eq. (6.9)

$$\frac{d^2\delta_m}{dt^2} = \frac{d^2\theta_m}{dt^2} \tag{6.10}$$

Substituting into Eq. (6.8), we get

$$I\frac{d^2\delta_m}{dt^2} = T_m - T_e\ N-m \tag{6.11}$$

Multiplying both sides by the angular velocity ω_m, we get

$$\omega_m I\frac{d^2\delta_m}{dt^2} = \omega_m(T_m - T_e)\ N-m \tag{6.12}$$

where
$\omega_m I = M$ = angular momentum or inertia constant
$\omega_m T_m = P_m$ = mechanical power input at the shaft minus rotational losses
$\omega_m T_e = P_e$ = electrical power output minus losses

We can write Eq. (6.12) as

$$M\frac{d^2\delta_m}{dt^2} = P_m - P_e = P_a\ \text{W} \tag{6.13}$$

M depends on the speed ω_m. However, since the deviation in speed is limited, M can be assumed to be a constant. The value of M varies over a wide range depending on the rating and type of the generator. Hence, another constant H is used to specify the energy stored in the machine.

$$\text{H} = \frac{\text{Stored kinetic energy in MJ at synchronous speed}}{\text{Machine rating in MVA}}\ \text{MJ/MVA} \tag{6.14}$$

H is also called *inertia constant*. It lies in a narrow range for different machines.
M and H are related as follows

$$M = \frac{2\text{GH}}{\omega_{sm}}\ \text{MJ-s/mech rad} \tag{6.15}$$

where G = MVA rating of machine. In pu, M = 2H.
Now, Eq. (6.13) can be written as

$$\frac{2H}{\omega_{sm}}\frac{d^2\delta_m}{dt^2} = \frac{P_m - P_e}{G} \tag{6.16}$$

We can express both δ_m and ω_{sm} in terms of electrical radians to get

$$\frac{2H}{\omega_s}\frac{d^2\delta_m}{dt^2} = P_m - P_e = P_a\ \text{pu} \tag{6.17}$$

Here, P_m = per unit mechanical power [P_m in MW/G].
ω_s = synchronous speed in electrical rad/s
$= \frac{P}{2}\omega_{sm}$
P_a = acceleration power

Equation (6.17) is called the *swing equation*. We can linearize Eq. (6.17) to get

$$\frac{2H}{\omega_s}\frac{d^2\Delta\delta}{dt^2} = \Delta P_m - \Delta P_e \tag{6.18}$$

We express the speed deviation also in pu to get

$$\frac{d\Delta\omega}{dt} = \frac{1}{2H}(\Delta P_m - \Delta P_e) \tag{6.19}$$

Taking the Laplace transform we get

$$\Delta\omega(s) = \frac{1}{2Hs}(\Delta P_m(s) - \Delta P_e(s)) \tag{6.20}$$

Eq.(6.20) can be written in a block diagram form, as shown in Fig. 6.20.

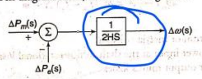

Figure 6.20 Model of generator.

6.6.3 Load Model

We have seen the load models in detail in Chapter 2. Some loads exhibit variation in active power drawn with respect to frequency variations. This relationship is given by

$$\Delta P_{Lfreq} = D\Delta\omega$$

or D = load damping constant

$$D = \frac{\Delta P_{Lfreq}}{\Delta\omega} \tag{6.21}$$

where ΔP_{Lfreq} = frequency-dependent load.

$$D_{New} = D\times\left(\frac{1000}{800}\right) = 1.5\times\frac{1000}{800} = 1.875$$

Neglecting losses, the change in electrical output of the generator, ΔP_e, is equal to the load. Therefore,

$$\Delta P_e = \Delta P_L + D\Delta\omega \tag{6.22}$$

where ΔP_L = non-frequency sensitive load change
$D\Delta\omega$ = frequency sensitive load change

Or

$$\Delta P_e(s) = \Delta P_L(s) + D\Delta\omega(s) \tag{6.23}$$

Equation (6.22) can be introduced into Fig. 6.20 to obtain Fig. 6.21.

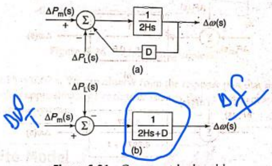

Figure 6.21 Generator + load model.

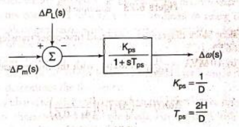

Figure 6.22 Standard first-order model for generator + load.

In Fig. 6.21, $2H$ can be replaced by M, where both are in pu. The transfer function of Fig. 6.21 can be written in the form of a standard first-order transfer function $\frac{K_{ps}}{1+T_{ps}s}$, as shown in Fig. 6.22,

where

K_{ps} is the *power system gain* and
T_{ps} is the *power system time constant*.

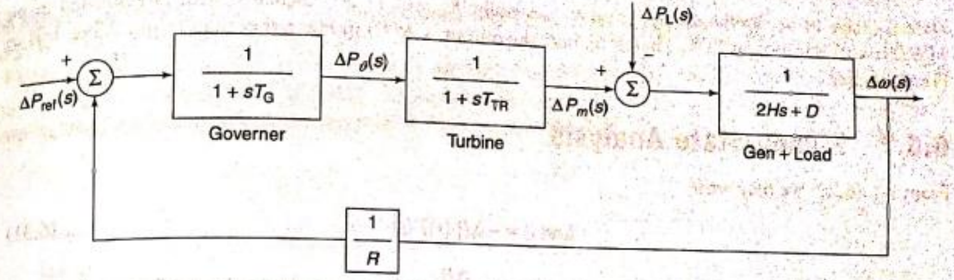

Figure 6.29 Block diagram of complete ALFC.

We are now interested in deriving the effect of change in load on the frequency without change in the reference set point. By changing the reference set point, we can set the system to give specified frequency at any load point as explained in Fig. 6.7(b). This is a secondary control to be discussed later. Here, we assume P_{ref} is kept at a constant value so that $\Delta P_{ref} = 0$. We now find the transfer function $\frac{\Delta\omega(s)}{-\Delta P_L(s)}$. From the block diagram of Fig. 6.29,

$$\Delta\omega(s) = -\Delta P_L(s)\left[\frac{\frac{1}{2Hs+D}}{1+\frac{1}{R}\left(\frac{1}{2Hs+D}\right)\left(\frac{1}{1+sT_G}\right)\left(\frac{1}{1+sT_{TR}}\right)}\right] \tag{6.29a}$$

$$= -\Delta P_L(s)\left[\frac{(1+sT_G)(1+sT_{TR})}{(2Hs+D)(1+sT_G)(1+sT_{TR})+\frac{1}{R}}\right] \tag{6.29b}$$

The transfer function is given by

$$T(s) = \left[\frac{(1+sT_G)(1+sT_{TR})}{(2Hs+D)(1+sT_G)(1+sT_{TR})+\frac{1}{R}}\right]$$

An alternate expression for the transfer function commonly used is

$$T(s) = \left[\frac{\frac{K_{ps}}{1+sT_{ps}}}{1+\left(\frac{K_{ps}}{1+sT_{ps}}\right)\left(\frac{1}{1+sT_G}\right)\left(\frac{1}{1+sT_{TR}}\right)\frac{1}{R}}\right]$$

$$= \left[\frac{K_{ps}(1+sT_G)(1+sT_{TR})}{(1+sT_{ps})(1+sT_G)(1+sT_{TR})+\frac{K_{ps}}{R}}\right] \tag{6.30}$$

7.2 Tie-Line Control with Primary Speed Control

Let us consider a two-area system as shown in Fig. 7.1.

Let us take the positive power flow to be P_{12}, to be the power flow from area 1 to area 2. The power flow on the tie-line from area 1 to area 2 is

$$P_{12} = \frac{E_1 E_2}{X_{12}} \sin(\delta_1 - \delta_2) \tag{7.1}$$

where

$$X_{12} = X_1 + X_{tie} + X_2$$

Equation (7.1) can be linearized about an initial operating point $\delta_1 = \delta_{10}$ and $\delta_2 = \delta_{20}$ as

$$\Delta P_{12} = \frac{E_1 E_2}{X_{12}} \cos(\delta_{10} - \delta_{20}) \Delta\delta_{12} \tag{7.2}$$

$$\Delta\delta_{12} = \Delta\delta_1 - \Delta\delta_2 \tag{7.3}$$

$$\text{Let } T = \frac{E_1 E_2}{X_{12}} \cos(\delta_{10} - \delta_{20}) = P_{max} \cos(\delta_{10} - \delta_{20}) \tag{7.4}$$

where T is called the *synchronizing torque coefficient* (often designated as P_s).

Substituting Eq. (7.4) into Eq. (7.2), we get

$$\Delta P_{12} = T(\Delta\delta_1 - \Delta\delta_2) \tag{7.5}$$

The block diagram representation of the two-area system with only primary control is shown in Fig. 7.2. A positive ΔP_{12} means an increase in power flow from area 1 to 2. This is equivalent to a load increase in area 1 and/or decreasing load in area 2. Therefore, the feedback from ΔP_{12} has a *negative sign* for area 1 and positive sign for area 2. We will now see how the system behaves for a change in the load.

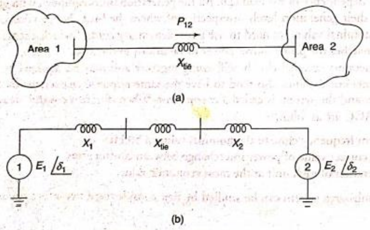

Figure 7.1 (a) Two-area system and (b) Electrical equivalent.

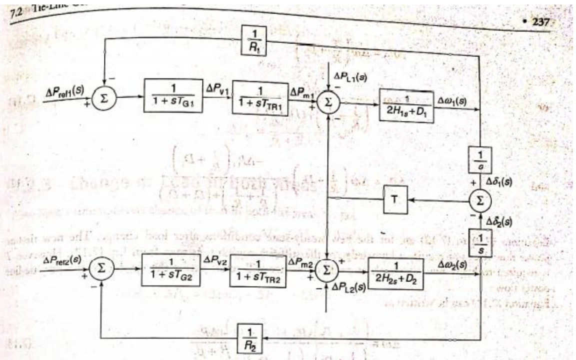

Figure 7.2 Two-area system with primary loop.

7.2.1 Change of Load in Area 1

Consider a load change of ΔP_{L1} in area 1. When the system has reached a steady state, both areas will have same steady-state frequency deviations. Therefore,

$$\Delta\omega = \Delta\omega_1 = \Delta\omega_2 \tag{7.6}$$

(or $\Delta f - \Delta f_1 = \Delta f_2$. Remember that *in pu both Δf and $\Delta\omega$ are the same*). If we assume that the mechanical powers are constant (which means ΔP_{ref} is constant), the tie-line and rotating masses exhibit damped oscillations called *synchronizing oscillations*. For area 1, we can write

$$\Delta P_{m1} - \Delta P_{12} - \Delta P_{L1} = \Delta\omega D_1 \tag{7.7}$$

For area 2, we have

$$\Delta P_{m2} + \Delta P_{12} = \Delta\omega D_2 \tag{7.8}$$

We have further

$$\Delta P_{m1} = \frac{-\Delta\omega}{R_1} \tag{7.9}$$

$$\Delta P_{m2} = \frac{-\Delta\omega}{R_2} \tag{7.10}$$

Substituting Eqs. (7.9) and (7.10) in Eqs. (7.7) and (7.8), respectively, we get

$$-\Delta P_{12} - \Delta P_{L1} = \Delta\omega\left(\frac{1}{R_1} + D_1\right)$$

4

7.6 Tie-Line Oscillations

We saw in the previous section that the system state matrix is a 9 × 9 matrix for a two-area system. With some simplification, we can get a fairly good idea of the effect of the system parameters on the dynamic response. Let us make the following assumptions:

1. Neglect turbine and governor time constants.
2. Neglect damping constants D_1 and D_2.
3. Both areas are identical.

With these assumptions, the two area equations reduce to

$$\Delta P_{m1}(s) = \frac{-\Delta\omega_1(s)}{R} \tag{7.70}$$

$$\Delta P_{m2}(s) = \frac{-\Delta\omega_2(s)}{R} \tag{7.71}$$

$$\Delta\omega_1(s) = \frac{1}{2Hs}[\Delta P_{m1}(s) - \Delta P_{L1}(s) - \Delta P_{12}(s)] \tag{7.72}$$

Substituting for $\Delta P_{m1}(s)$, we get

$$\Delta\omega_1(s) = \frac{1}{1+\frac{1}{2RHs}}\left[\frac{-\Delta P_{L1}(s) - \Delta P_{12}(s)}{2Hs}\right] \tag{7.73}$$

Similarly we get

$$\Delta\omega_2(s) = \frac{1}{1+\frac{1}{2RHs}}\left[\frac{-\Delta P_{L2}(s) + \Delta P_{12}(s)}{2Hs}\right] \tag{7.74}$$

$$\Delta P_{12}(s) = \frac{T}{s}[\Delta\omega_1(s) - \Delta\omega_2(s)]$$

$$= \frac{T}{s}\frac{1}{\left(1+\frac{1}{2RHs}\right)}\left[\frac{1}{2Hs}(\Delta P_{L2}(s) - \Delta P_{L1}(s) - 2\,\Delta P_{12}(s))\right]$$

or

$$\Delta P_{12}(s)\left[1+\frac{2T}{2Hs^2+\frac{s}{R}}\right] = \frac{T}{2Hs^2+\frac{s}{R}}[\Delta P_{L2}(s) - \Delta P_{L1}(s)]$$

or

$$\Delta P_{12}(s) = \frac{\frac{T}{2H}}{s^2+\frac{s}{2RH}+\frac{T}{H}}[\Delta P_{L2}(s) - \Delta P_{L1}(s)] \tag{7.75}$$

The poles of the denominator determine the oscillations in ΔP_{12}. We compare the denominator with the standard second-order characteristic equation.

$$s^2 + 2\alpha s + \omega_n^2 = s^2 + 2\xi\omega_n s + \omega_n^2 \tag{7.76}$$

We can see that

$$\alpha = \frac{1}{4RH} \tag{7.77}$$

And

$$\omega_n = \sqrt{\frac{T}{H}}\ \text{pu or}\ \sqrt{\frac{2\Pi f_0 T}{H}}\ \text{rad/s} \tag{7.78}$$

The damping is determined by the relative values of α and ω_n and the roots of Eq. (7.75).
The roots of Eq. (7.75) are

$$s_1, s_2 = -\xi\omega_n \pm j\,\omega_n\sqrt{1-\xi^2} \tag{7.79}$$

$$= -\alpha \pm j\,\omega_d$$

and $$\omega_d = \omega_n\sqrt{1-\xi^2} \tag{7.80}$$

α is called the *damping factor* or *damping constant*. ω_n is the *natural undamped frequency* of oscillations, ω_d is called the *damped or conditional frequency* and ξ is called the *damping ratio*. We now have the following cases:

1. When $\xi = 1$ or $\alpha = \omega_n$. This condition is called "critical damping."
2. When $\xi = 0$ or $\alpha = 0$, the roots are purely imaginary and we get purely sinusoidal oscillations for a step change in input. In Eq. (7.74), the input is $\Delta P_{L2}(s) - \Delta P_{L1}(s)$. Therefore, for a step change in the load, we would get sustained sinusoidal oscillations of tie-line power at a frequency of ω_n. From Eq. (7.76), we can see that $\alpha = 0$ when $R = \infty$. This means that there is no governor speed control.
3. When $\alpha < \omega_n$, $\xi < 1$, we get a pair of complex conjugate roots. The system is "under damped" and we have oscillations in tie-line power flow which have a frequency ω_d as in Eq. (7.79). The time constant of the system is $1/\alpha$.
4. When $\alpha > \omega_n$, we have an "over damped" system. The roots are both real.

The above analysis is only approximate, but is helpful in knowing the effect of the choice of parameters on the stability of the system. If we consider the damping constants of the load, then α is modified as

$$\alpha = \frac{1}{4H}\left[D + \frac{1}{R}\right] \tag{7.81}$$

5

5)

$$R_1 = \frac{10000}{500} = 20 \text{ Hz/pu m}$$

$$D_1 = 0.001 = 0.001 \text{ pu mw/Hz}$$

$$\beta_1 = \frac{1}{R_1} + D_1 = 0.051 \text{ pu mw/Hz}$$

$$\beta_2 = \frac{1}{R_2} + D_2 = 1.02 \text{ pu mw/Hz}$$

$$\Delta P_{L_2} = \frac{200}{10000} = 0.02 \text{ pu}$$

$$\Delta f = \frac{-\Delta P_{L2}}{\beta_1 + \beta_2} = \frac{-0.02}{0.051 + 1.02} = -0.0187 \text{ Hz}$$

Steady state freq = 50 − 0.0187 = 49.9813 Hz

6

6) Load change in area 1

$$\Delta f = \frac{-\Delta P_{L1}}{\beta_1 + \beta_2}$$

$$\Delta P_{L1} = \frac{100}{1000} = 0.1 \text{ pu}$$

$$\beta_1 = \frac{1}{R_1} + D_1 = \frac{1}{0.01} + 0.8 = 100.8$$

$$\beta_2 = \frac{1}{R_2} + D_2 = \frac{1}{0.02} + 1 = 51$$

$$\Delta f = \frac{-0.1}{100.8 + 51} = -6.58 \times 10^{-4} \text{ pu}$$

$$\Delta f = -6.5 \times 10^{-4} \times 50 = -0.032 \text{ Hz}$$

$$\text{New freq} = 50 - 0.032 = 49.968 \text{ Hz}$$

$$\Delta P_{12} = \frac{-\Delta P_{L1} \beta_2}{\beta_1 + \beta_2} = \frac{-0.1 \times 51}{100.8 + 51} = 0.0335 \text{ pu}$$

$$= 33.59 \text{ MW}$$

When load changes in area 2

$$\Delta P_{L2} = 100 \text{ MW} = \frac{100}{1000} = 0.1 \text{ pu}$$

$$\Delta f = \frac{-\Delta P_{L2}}{\beta_1 + \beta_2} = \frac{-0.1}{100.8 + 51} = -6.58 \times 10^{-4} \text{ pu} = -0.032 \text{ Hz}$$

$$\text{New freq} = 50 - 0.032 = 49.968 \text{ Hz}$$

$$\Delta P_{L12} = \frac{\Delta P_{L2} \beta_1}{\beta_1 + \beta_2} = \frac{0.1 \times 100.8}{100.8 + 51} = 0.0664 \text{ pu}$$

$$= 66.4 \text{ MW}$$