CHAPTER 4

Governing Principles

Introduction

Turbine governors are equipment for the control and adjustment of the turbine power output and evening out deviations between power and the grid load as fast as possible.

The turbine governors[/3/,/4/,/5/,/6/] have to comply with two major purposes:

1. To keep the rotational speed stable and constant of the turbine-generator unit at any grid load and prevailing conditions in the water conduit.
2. At load rejections or emergency stops the turbine admission have to be closed down according to acceptable limits of the rotational speed rise of the unit and the pressure rise in the water conduit.

Alterations of the grid load cause deviations between turbine power output and the load. For a load decrease the excess power accelerates the rotating masses of the unit according to a higher rotational speed. The following governor reduction of the turbine admission means deceleration of the water masses in the conduit and a corresponding pressure rise.

To keep the rise of the rotational speed below a prescribed limit at load rejections, the admission-closing rate must be equal to or higher than a certain value. For the pressure rise in the water conduit the condition is opposite, e.g., the closing rate of the admission must be equal to or lower than a certain value to keep the pressure rise as low as prescribed.

For power plants where these two demands are not fulfilled by one single control, the governors are provided with dual control functions, one for controlling the rotational speed rise and the other for controlling the pressure rise. This is normal for governors of high head Pelton and Francis turbines.

For Pelton turbines the principle is:

- To set the closing rate of the needle control of the nozzles to a value which satisfies the prescribed pressure rise
- To bend the jet flow temporarily away from the runner by a deflector so the speed rise does not exceed the accepted level.

For Francis turbines the principle is:

- To set the closing rate of the guide vane opening to a value, which satisfies the rotational speed, rise limits

- To divert as much of the discharge through a controlled by-pass valve that the pressure rise in the conduit is kept below the prescribed level.

4.1 Feedback control system

The governor function for a turbine with water conduit is shown in the block diagram on Fig. 4.1.

The input reference signal is compared with the speed feedback signal. By a momentary change in the load a deviation between the generator power output and the load occurs. This deviation causes the unit inertia masses either to accelerate or to decelerate. The output of this process is the speed, which again is compared with the reference.

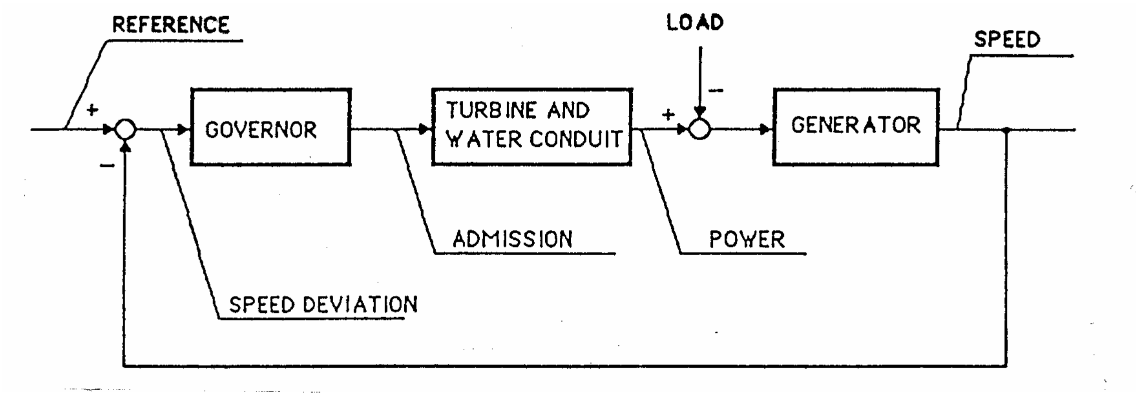

Fig. 4.1 Block diagram of a turbine closed loop system /6/

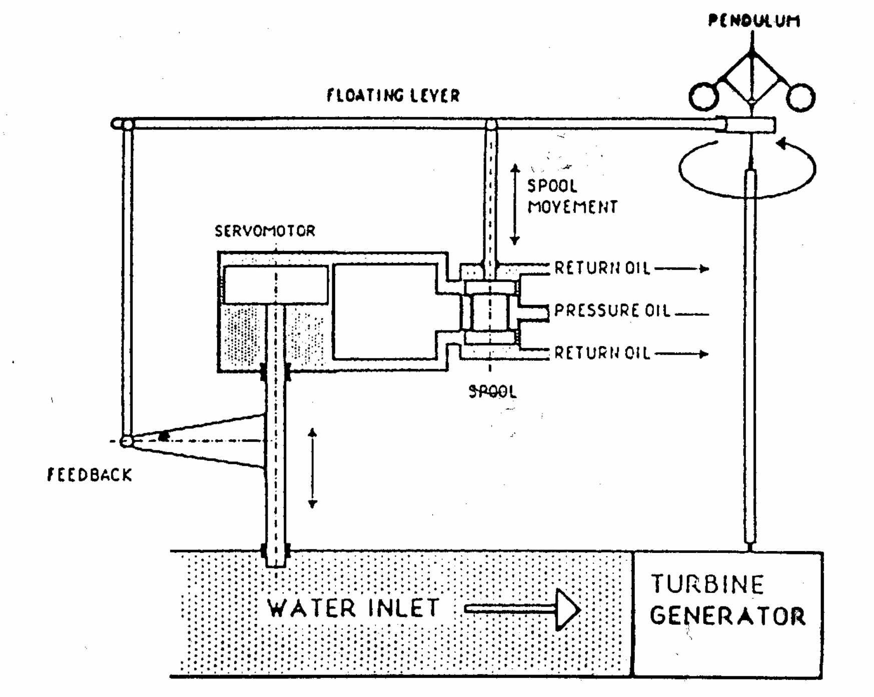

Fig. 4.2 A hydraulic governor with a direct acting pendulum /6/

A simple but classic example of a turbine governor is shown schematically in Fig. 4.2. This is a governor with a belt driven centrifugal pendulum. For explaining the governor actions it is chosen to start at a moment of stable equilibrium between power and load. In this condition the control valve is closed by the spool, which is in the neutral position.

When a decrease in the grid load occurs, the rotational speed starts increasing and the pendulum sleeve and the connected end of the floating lever moves upwards. The lever moves the spool accordingly upward out of the neutral position and opens the hydraulic conduits to the servomotor.

High-pressure oil flows to the piston topside. The piston moves downwards and reduces the gate opening and the turbine power. At the moment when the power is equal to the load, the rotational speed culminates as indicated on Fig. 4.3.

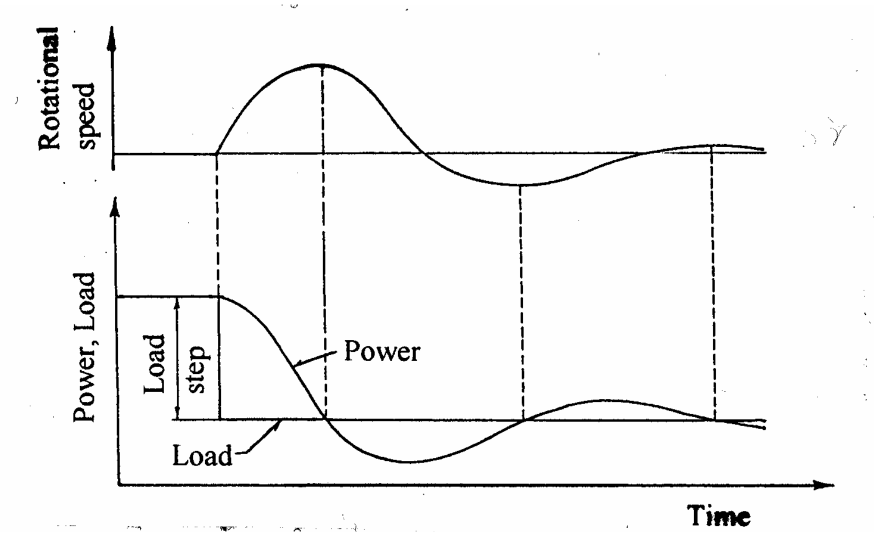

Fig.4.3 Time response of power output and rotational speed after a load reduction step

At this moment however, the spool valve is still open. The piston movement continues and the power output decreases even more. Consequently the speed decreases and the pendulum sleeve and the spool are moving downwards again.

During this movement the spool valve passes the neutral position and opens then for high-pressure oil flow to the opposite side of the piston. The piston movement is thereby returned and the power output increasing. Next time the rotational speed culminates the power again is equal to the load and therefore a succeeding swing in the speed and power output take place as previously described.

As Fig. 4.3 indicates, the swings are strongly damped because of the feedback system. This feed-back is arranged by a linkage connection between the servomotor rod and the end of the floating lever that is opposite to the sleeve as shown on the Fig. 4.2. When the piston moves in the closing direction, the floating lever moves the spool accordingly in the same direction towards the neutral position. In this way a stable control process is obtained.

A governor of a type as shown in Fig. 4.2 has a one-stage amplifier. Water turbine governors have normally several stages of amplifiers. The governors in use are of various designs, e.g., as mechanical-hydraulic and electro-hydraulic products. Design of governors and governor systems are described in Chapter 10.

4.2 Governor adjustment facilities

4.2.1 Proportional-integral-derivative functions (PID)

In governor systems as shown in Fig. 4.2 the rotational speed is dependent on the load for power/load

equilibrium conditions. That means higher rotational speed at zero loads than at max. load and this dependency are linear.

This type of governor function is designated as *proportional control* and denoted by the label P.

For units delivering the power to a grid system the frequency has to be constant at any load. Governors for these units therefore have adjustment means also for automatically recovering of the speed according to the grid frequency during regulation processes.

This type of governor function is designated as *integral control* and denoted by the label I.

Groups of governors are provided also with a derivative function in addition to the above functions. It may be utilised for improvement of the phase angle of the frequency response for the governor system.

This type of governor function is designated as *derivative control* and denoted by the label D.

Mechanical hydraulic governors are provided with PI-functions only while electro-hydraulic governors are designed with PID functions.

Governors with PID-functions have large ranges for adjustment of each of the PID function parameters.

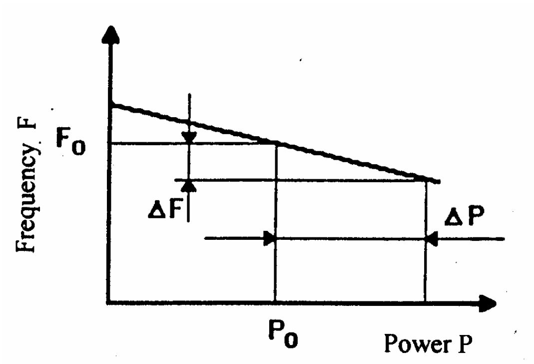

Fig.4.4 Permanent droop and strength of regulation

4.2.2 Permanent speed droop

A special and major property of a turbine governor is the permanent droop. It is defined as the percentage change in the frequency for a 100 % change of the power output from the unit.

For example a unit with a set value of the permanent droop at 5 % will respond with 40 % power increase for a change in frequency from 50 Hz to 49 Hz.

The permanent droop is adjustable and in practice it is chosen in the range 0 - 6 %.

In connection with the permanent droop another notion also plays a certain role. That is the *strength of regulation*[/4/]*, which* is defined as

$$\text{Strength of Regulation} = -\frac{\Delta P}{\Delta F}$$

where ΔP is power change

ΔF is frequency change

as illustrated in Fig. 4.4.

The load distribution between turbine-generator units connected to the same grid is dependent on the permanent droop setting of these units. In Fig. 4.5 two units with different permanent droop are shown where the load distribution for a given frequency change is indicated.

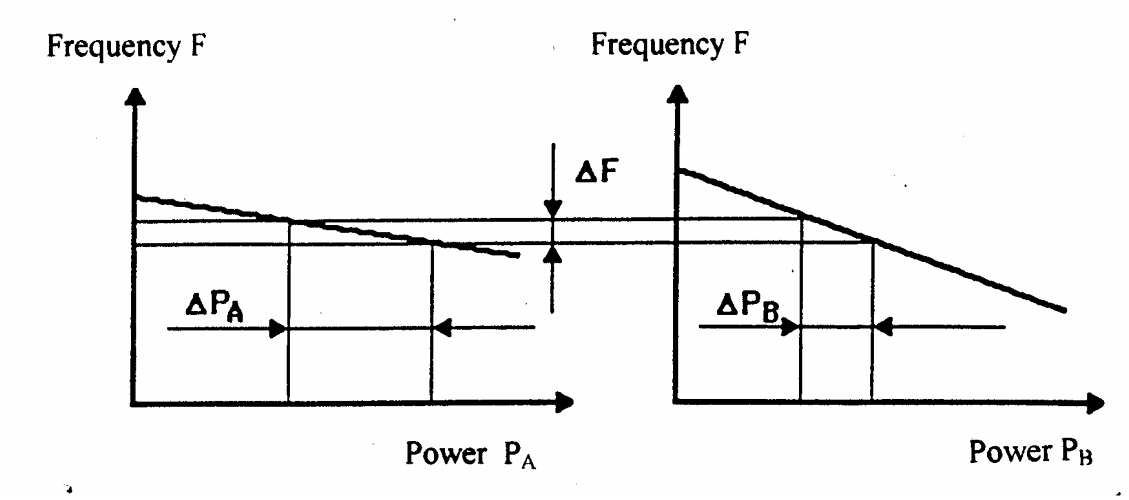

Fig. 4.5 Load distribution for different permanent droop of two units connected to the same grid /4/

4.3 Turbine governing demands

4.3.1 Frequency and load regulation

The governor shall be able to maintain stability of the generating unit when running on an isolated grid. Generally the units are designed for stable operation up to full load. In this mode of operation the governor shall keep the frequency within certain limits of deviation.

Load regulation on a rigid system is the most common operation mode. Each unit has little influence on the grid system frequency. The governor controls the load to the desired value. The variation of the load as function of the change in frequency is dependent on the permanent droop setting.

A special mode of operation is the manual mode where the guide vane openings are controlled manually by means of a mechanical hydraulic load limiter. In this mode only the load can be controlled.

4.3.2 Start and stop sequence control

During the period of start the unit shall be run up to nominal speed as quickly and smoothly as possible. A start can be carried out both manually and automatically. The admission must be opened only when permitted by all overriding start conditions.

In shut down mode the admission shall be closed as quickly as possible but limited by the magnitude of the pressure rise in the tunnel and pressure shaft system. Due to safety reasons, the shut down signal will be given simultaneously to different stages in the governor, e.g. closing of the load limiter or the emergency operated shut down valve. The shut down valve is also functioning if the ordinary voltage supply has failed. The stop command can be given both manually and automatically.

4.3.3 Disconnection, load rejection

Disconnection means to open the generator main circuit switch. The generator is thereby separated from grid and the turbine power output results in a speed rise of the unit. The function of the governor is then to shut down the turbine not faster than the caused pressure rise is kept below the guaranteed level.

4.3.4 Load limiting

Load limiting must be possible according to external conditions. The load limiter device may be operated both manually and automatically.

4.4 Regulation requirements of water power plants

The governing of water turbines requires limitations of speed rise and pressure rise as well as fulfilment of regulating stability demands. To guarantee these requirements the adaptation of the dynamic behaviour of the conduit system and the generator inertia mass have to be examined carefully. Basic theory and principles for these examinations are surveyed in the following.

An example of the layout of a traditional high head power plant is shown in Fig. 4.6.

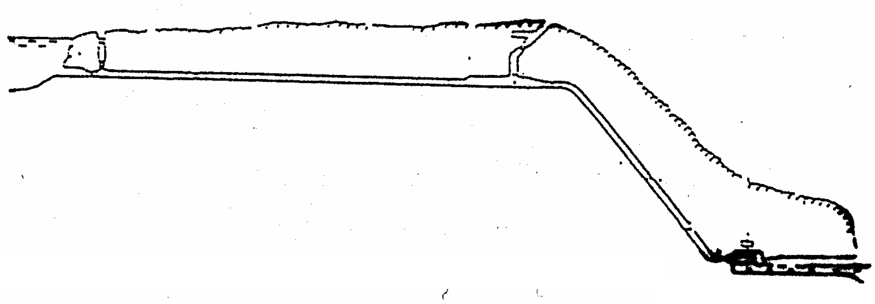

Fig. 4.6 The layout of a high head Hydro Electric Power Plant /6/

In such a power plant the pressure rise and regulating stability may be divided in two parts:

1. The mass oscillation problem for the tunnels and surge shafts both on the pressure side and the suction side.
2. The water hammer problem of the conduit between the turbine and the water level in the surge shafts both on pressure side and suction side.

4.4.1 Mass oscillations

The mass oscillation stability problem is treated according to the criterion for the minimum critical Thoma cross section area of the surge shaft, ref. /9/, for obtaining stable mass oscillations.

This criterion which is based on the equation of continuity, the equilibrium of forces and the assumption of an ideal turbine governor, is well described in the literature.

An ideal turbine governor is assumed to be able to keep the product of discharge and pressure as constant. However, this statement is not fulfilled if the tunnel is short with short surging time because the turbine governor then will be too slow to obtain constant output of the turbine. For short tunnels the margin to the critical Thoma cross section area of the surge shaft must be increased accordingly.

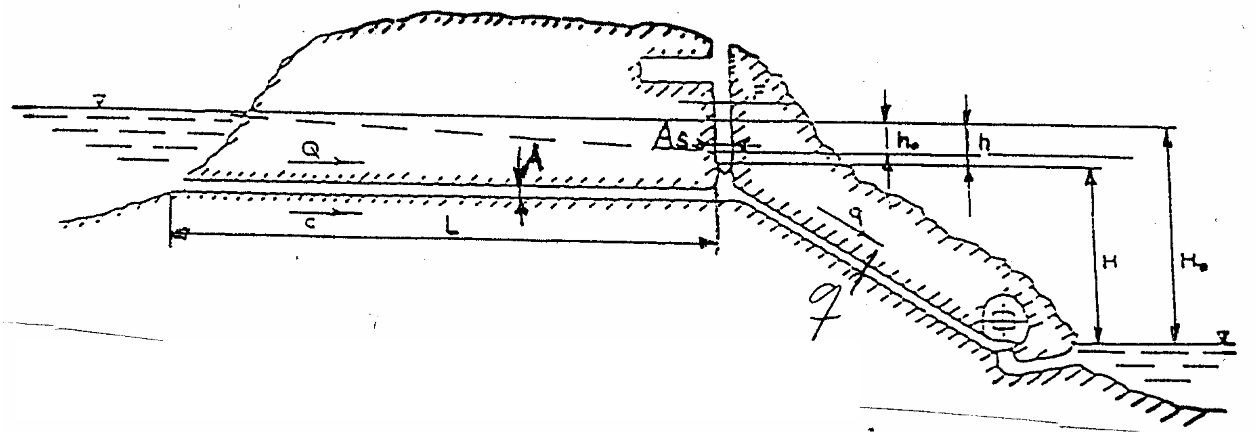

Fig. 4.7 Tunnel and surge shaft with values used in the simple Thoma criterion /6/

With reference to Fig. 4.7 the equation of continuity is

(4.1)

where A_s is the cross section area of the surge shaft
$h = H_o - H$ is the head difference between reservoir level and surge shaft level
q is the turbine flow
Q is the flow in the tunnel

The equilibrium of forces

$$\frac{L}{A}\frac{dQ}{dt} = g\left[h - h_o\left(\frac{Q}{Q_o}\right)^2\right] \tag{4.2}$$

where A is the cross section area of the tunnel
g is the acceleration of gravity
$h_o = (f/R_h)LQ_o^2/(2g\,A^2)$ is the head loss in the tunnel
f is the Darcy-Weissback friction factor
L is the length of the tunnel
Q_o is the steady state flow before disturbance or mean flow
R_h is the hydraulic radius of the tunnel cross section area

The equation of discharge and pressure control of the turbine (with an idealised turbine governor)

$$q(H_o - h) = Q_o(H_o - h_o) \tag{4.3}$$

According to ref./8/ the three equations give the following critical cross section area

$$A_{scr} = \frac{LQ_o^2}{2gh_o(H_o - h_o)A} \qquad \text{for } h_o < \frac{1}{3}H_o \tag{4.4}$$

When substituting for head loss h_o and $H_o - h_o = H_n$ which is the net head,

$$A_{scr} = \frac{AR_h}{fH_n} \tag{4.5}$$

For a tunnel with an air cushion surge chamber the cross section area of the surge shaft A_s could be substituted for an equivalent cross section area $A_{eq.}$ By the equation of polytrophic changes of the air volume, the area A_{eq} is calculated for the compression of the air and surging of the level. According to ref./2/ the equation for A_{eq} is

$$A_{eq} = \frac{1}{\frac{1}{A_w} + \kappa \frac{H_o'}{V_o}} \tag{4.6}$$

where A_W is the water level area in the air cushion surge chamber

$H_o' = H_o + 10$ is the absolute air pressure head [m] in the air cushion surge chamber

V_o is the air volume in the air cushion surge chamber

κ is the polytropic exponent normally $\kappa \approx 1.3$

Normally $\frac{1}{A_w} << \kappa \frac{H_o'}{V_o}$ so that the formula may be simplified to

$$A_{eq} = \frac{V_o}{\kappa H_o'} \tag{4.7}$$

Then the critical air volume can be found by the simplified equation

$$V_{ocr} \approx \kappa H_o' A_{scr} \tag{4.8}$$

Normally a safety factor to the critical cross section area should be 1.3 - 1.5. According to ref./2/ however, it has been proven that for smooth full profile driven tunnels a larger margin should be used because of a smaller difference between steady state flow friction and friction of flow oscillations. For short tunnels an even larger margin is required than for long tunnels because a real turbine governor cannot satisfy Equation (4.3) for short tunnels on account of the shorter surging time of the pressure.

4.4.2 Water hammer pressure rise versus closure time and speed

The water hammer problems may be divided in to main areas:

1. The water hammer transient problems causing pressure rises that affects the stresses in penstocks and stress carrying parts of the turbine.
2. The governing stability problems caused by pressure oscillations in the conduit system.

The treatment of water hammer problems are based on ref. /1/. General solutions of governing stability problems and transients are based on ref. /8/. Special stability theory for power plants with influence of the frictional damping of oscillations in rough and smooth tunnels as well as the influence from the turbine characteristics is presented in ref. /2/.

The general equations

With reference to Fig. 4.8 parameters additional to those being used previously are:

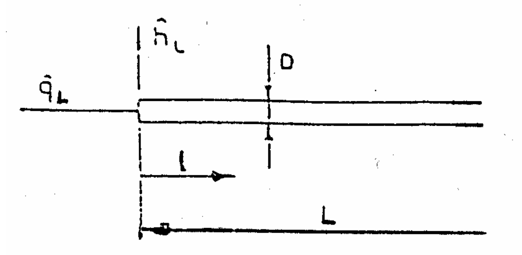

Fig. 4.8 Serction of pipe with diameter D

A is the cross section area of the pipe
D is the diameter of the pipe
l is the parameter of the pipe length from the pipe outlet
L is the length of the pipe

The most commonly used dynamic equations of water hammer problems yields:

The equation of continuity

$$\frac{\partial q_L}{\partial l} = \frac{AH_o g}{a^2 Q_o} \frac{\partial h_L}{\partial t} \tag{4.9}$$

Equilibrium of forces

(4.10)

where $h_L = \dfrac{\Delta H}{H_o}$ is relative pressure variation

$\Delta H = H - H_n$	is the pressure head variation
H	is the instantaneous pressure at the turbine inlet
H_n	is the net head
$K = \dfrac{\pi D \tau}{\rho Q_o q_L}$	is the head loss friction factor, ref. /2/ (4.11)
$q_L = \dfrac{\Delta Q}{Q_o}$	is the relative flow variation
ΔQ	is the change of the flow
ρ	is the density of the fluid
τ	is the friction shear force per unit length of the pipe

These basic Equations (4.9), (4.10) and (4.11) may be solved according to methods described in refs. /2/ and /7/.

Important parameters

The stability of the governing process of a hydro turbine-generator unit is dependent on the velocity c of the water in the conduit, the inertia of the rotating masses of the unit and the length L of the water conduit. The length L is defined as the distance between the turbine and the surge chamber or the distance between the turbine and the reservoir if there is no surge chamber.

According to the dependency of these parameters, some important compound parameters are created:

$T_a = \dfrac{\pi^2 n^2 I}{(30)^2 P_n}$ is the acceleration time of the rotating masses of the unit [sec]

$$T_w = \frac{\sum Lc}{gH_n} \quad \text{is the inertia time constant of the water masses in the conduit [sec]} \qquad (4.12)$$

$$h_w = \frac{ac_m}{2gH_n} = \frac{T_w}{T_a} \quad \text{is the water hammer number} \qquad (4.13)$$

where a is the propagation speed of the water hammer wave
g is the acceleration of gravity
H_n is the nominal head
$c = \frac{Q_n}{A}$ is the water velocity in the conduit
A is the cross section area of the water conduit
Q_n is the nominal flow
$c_m = \frac{\sum Lc}{\sum L}$ is the mean water velocity
n is the rotational speed
I is the inertia of the rotating masses
P_n is the nominal power of the unit
$T_r = \frac{2\sum L}{a}$ is the penstock reflection time

Simplified analysis of pressure rise from water hammer

During shut down of the unit a fast closure of the guide vanes (or needles of Pelton turbines) is required to avoid high speed rise. A fast closure however, causes a high pressure rise in the penstock and the speed rise has influence on the flow through reaction turbines.

For Pelton turbines the speed rise problem is normally solved by jet deflectors which deflect the jets quickly away from the runner and thus allowing for a slow closure of the needles. The corresponding solution of the speed rise problem of Francis turbines is to bypass the discharge through an energy dissipater with a valve controlled by the governor, see Chapt. 10.

For keeping speed rise and pressure rise within specified limits, some simple formulas are used for determination of closing and opening times of a gate at the downstream end of a penstock. These formulas[/3/] which are presented in the following, are valid for linear closure with constant speed. Frictional damping is neglected and the relative pressure $z = H/H_n$ is introduced.

- Closing time T_{CL}:

  For elastic water and a long elastic penstock $2h_w < (1 + \sqrt{z})$, then the closing time for linear closing from 100% opening

$$T_{CL} = T_w \frac{2}{1 - z} \qquad (4.14)$$

  Note: T_{CL} is negative for $\Delta H > 0$

- Opening time T_O:

For elastic water and elastic penstock

$$T_O = T_w \frac{2\sqrt{z}}{1-z} \tag{4.15}$$

Note: T_O is positive and $|\Delta H| < 1.0$

- For inelastic water and inelastic penstock:

Inelastic water and inelastic penstock means a short penstock $2h_w < 1 + \sqrt{z}$.

Closing time

$$T_{CL} = T_w \frac{\sqrt{z}}{1-z}$$

Opening time (4.16)

$$T_O = T_w \frac{\sqrt{z}}{1-z}$$

In this case maximum pressure occurs from large openings.

Note

The maximum value z is obtained with the maximum closing speed; that means 100% closure during the time T_{CL}. If the closure from a certain opening to zero takes the time 2L/a, then the reduction in velocity will be $\Delta c = c_o(2L/a)/T_{CL}$. By substituting for $z = 1 + \Delta H/H_o$, $T_w = L\Delta c/(gH_o)$ and $T_{CL} = 2L/a$ in formula (9.14), the maximum pressure rise becomes

$$\Delta H = \frac{a\Delta c}{g} \tag{4.17}$$

The simplified formulas may be used also for complex high pressure systems. In such cases the length of the penstock should be regarded as the length from the turbine to the nearest water level in the surge shaft. If an air cushion surge chamber is used, the nearest water level will be in the air cushion chamber where full reflection of pressure waves occurs. The air cushion surge chamber will be located within a relatively short distance from the turbine, and thereby excellent regulating conditions will be obtained.

Speed rise of turbine and generator

The speed rise of the turbine may also be calculated in a simplified way. According to the 2. law of Newton the maximum rotational speed n is found by the formula:

$$\frac{n}{n_n} d\left(\frac{n}{n_n}\right) = \frac{P - P_G}{T_a} dt$$

where n is rotational speed of the turbine

n_n is the nominal rotational speed of the turbine

P_G is the generator load

P is the turbine power output

P_n is the nominal turbine power output

$T_a = \frac{\pi n I}{30 P_n}$ is the acceleration time of the rotating masses of the unit

I is the moment of inertia of the rotating masses of the unit

t is the time

By introducing the speed rise parameter ε‘ and integrating one get the area A_n as indicated in Fig. 4.9.

$$\varepsilon' = \frac{1}{2}\left(\frac{n^2 - n_n^2}{n_n^2}\right) = \int_0^{T_{CL}} \left(\frac{P - P_G}{T_a}\right) dt = \frac{A_n}{T_a} \tag{4.18}$$

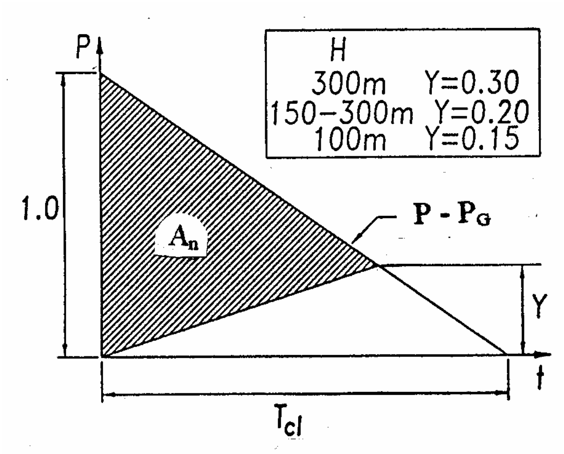

Fig. 4.9 Illustration of the integration of the accelerating torque versus time /3/

The presentation of the integrated torque as like the shaded area A_n in fig. 4.9, is based on a simplified consideration of the effect of the zero efficiency point. The zero efficiency point occurs just before closure of the guide vane cascade. The effect of this may be accounted for by simply to evaluate ε‘ in equation (4.18) satisfactorily by the following equation

$$\varepsilon' = \frac{1}{2} T_{CL} \frac{(1 - Y)(P - P_G)|_{t=0}}{T_a} = \frac{A_n}{T_a} \tag{4.19}$$

where

T_{CL} is the closing time

$(P - P_G)|_{t=0}$ is the difference between turbine power and generator power at t = 0

Y is a numerical value dependent on the net head H_n

Values of Y are:

High head turbines $H_n > 300$ m	Y = 0.30
Medium head turbines 150 m $< H_n <$ 300 m	Y = 0.20
Low head turbines $H_n <$ 150 m	Y = 0.15

The pressure rise may also be accounted for by multiplying the expression of ε‘ by $z^{3/2}$, where $z = H/H_n$ is the relative pressure. Then the speed rise becomes

$$\varepsilon' = \frac{A_n}{T_a} z^{\frac{3}{2}} \tag{4.20}$$

Further the maximum speed can be calculated according to Equation (4.20)

$$2\varepsilon' = \frac{n^2 - n_n^2}{n_n^2} = \left(\frac{n}{n_n}\right)^2 - 1$$

From this

$$\frac{n}{n_n} = \sqrt{1 + 2\varepsilon'}, \quad \text{and} \quad n = n_n + \Delta n$$

then

$$\frac{\Delta n}{n} = \sqrt{1 + 2\varepsilon'} - 1 \qquad (4.21$$

4.5 Governing stability

4.5.1 Modes of operation

Regarding governing conditions there are three different modes of operation on isolated loads:

1. Steady state operations when the unit is operating at constant load, head and command input.
2. The total system is subject to small changes caused by fluctuations in load or command input. In this mode none of the governor elements will reach the limit of closing or opening speed. The stability guarantees are always referred to this mode.
3. The total system is subject to changes, which is resulting in speed limits, closing and opening movements of parts of the governor system. This is the situation during load rejections when the main servomotors are operating at maximum closing speed.

When the generator is connected to a large grid with constant frequency, the function of the governor is only to change the generator output. If the generator is being disconnected, a load rejection will occur, which means that the system is in mode 3.

For stability analysis of operation on isolated grid it will often be sufficient to consider the system on the basis of the three compound parameters T_a, T_w and h_w[35]. However, some power plants have a rather complex tunnel and conduit system which makes it necessary to work out stability analysis of the complete water way system.

As an example a conventional high or medium high head system with inlet tunnel, surge chamber, penstock with a 45° slope and a tailrace tunnel, is sketched in Fig. 4.10. Concerning stability analysis the constants T_w and h_w are calculated from the length of the penstock between the water level in the surge chamber and the nearest free surface level in the tail water.

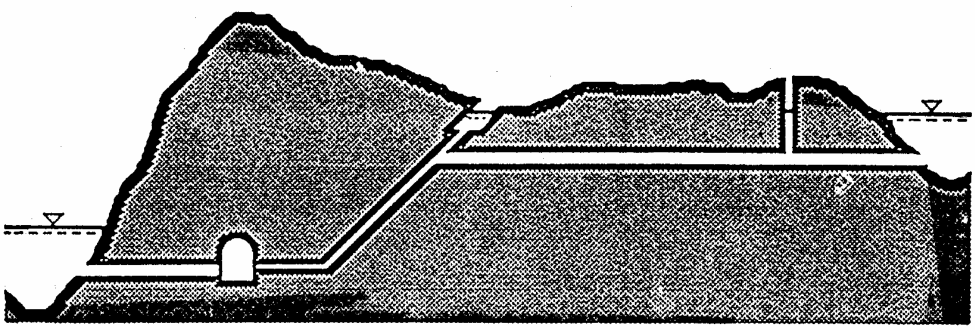

Fig.4.10 System with surge chamber /6/

In Norway high pressure tunnels have been excavated for increasing pressure levels. As a consequence the surge shaft length has increased. Therefore a relatively large part of travelling water hammer waves in the conduit passes the surge shaft because of the increased inertia mass of water in the surge shaft. These waves are more or less reflected from every free water level in all branches of the tunnel and conduit system. In turn this leads to unstable turbine governing.

An improvement of the conditions has been made by the introduction of an air cushion surge chamber schematically shown in Fig.4.11. By means of such surge chambers the distance from the turbine to the nearest water level becomes short as well, and an excellent governing stability is thereby obtained.

In addition a short closing time of the turbine governor can be obtained with low speed rise without a high-pressure rise.

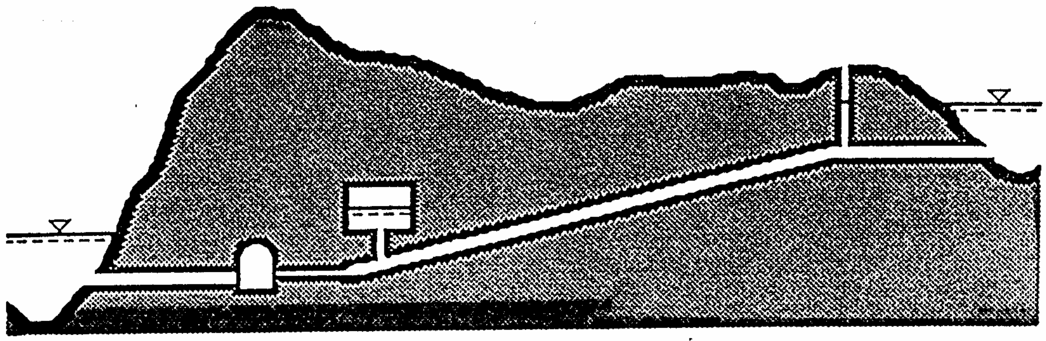

Fig.4.11 System with air cushion surge chamber /6/

For the stability analysis the Equations (4.9) and (4.10) have been Laplace transformed and the frictional shear force has been found to be a function of the flow oscillations and the frequency of the oscillations, ref. /2/. It should be emphasised that the influence from the turbine characteristics must be included to find the correct stability margins.

4.5.2 Rules of thumb

Some rules of thumb[/3/] may be established to make a brief judgement of the governing stability based on the water hammer number h_w and the relation between the time constants T_a and T_w. These rules are:

$2 < h_w$	$T_a/Tw \geq 2.5$
$1 < h_w < 2$	$T_a/T_w \geq 3.0$
$0.7 < h_w < 1$	$T_a/T_w \geq 3.5$
$0.5 < h_w < 0.7$	$T_a/T_w \geq 4.0$
$0.3 < h_w < 0.5$	$T_a/T_w \geq 4.5$

An efficient test for proving the governing stability is the frequency response test. Because the frictional damping of oscillatory flow is of great importance to know, it is most correctly estimated by this test.

Note

Modern governors with pressure feed back signal systems have additionally improved the stability of the governing systems.

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