# THE CONTROLLED SYSTEM

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## THE CONTROLLED SYSTEM

Hydro Turbine Governing system requirements have increased greatly in recent years since variables other than turbine speed often have to be controlled by the Governor. The governor must be capable of accepting, in addition to speed, a variety of other signals and combine them in such a manner to achieve suitable operation in various possible modes. The governing system must be reliable for the safety of the turbine and for its ability to develop power at times when needed. Flexibility is necessary to be able to adapt to changing power plant demands.

Because of these demands now placed upon modern governing system it is necessary to be familiar with the various elements of the controlled system; to understand their influence upon both the dynamic and steady state behaviour of the plant. The controlled system can be split into three components. They are:

1. Water column
2. Turbine-generator
3. Load

The water column consists of the penstock, scroll case and draft tube. Intuitively it is possible to see that to change power developed by the turbine it is necessary to change the velocity of the water flowing in the penstock, consequently the rate at which power can be changed is determined partially by the ability to accelerate or decelerate the column of water. This ability to accelerate the water column that is, the inertia, is measured by the characteristic time known as the Water Time and demoted by $T_W$. The value of $T_W$ for the penstock is computed when the penstock length L, is known along with the water velocity, V, and net head, H from the formula

$$(1) \quad T_w = \frac{\sum LV}{gH}$$

The summation denotes addition of the various sections of penstock, having different water velocities. The same type of computation is performed for the scroll case and draft tube. Addition of the $T_W$ for the three portions (penstock, scroll case and draft tube) of the penstock yields the water time used for stability studies. The water time can be illustrated through use of Figure 2 which is the response of penstock flow, w per cent, to known per cent step change of head.

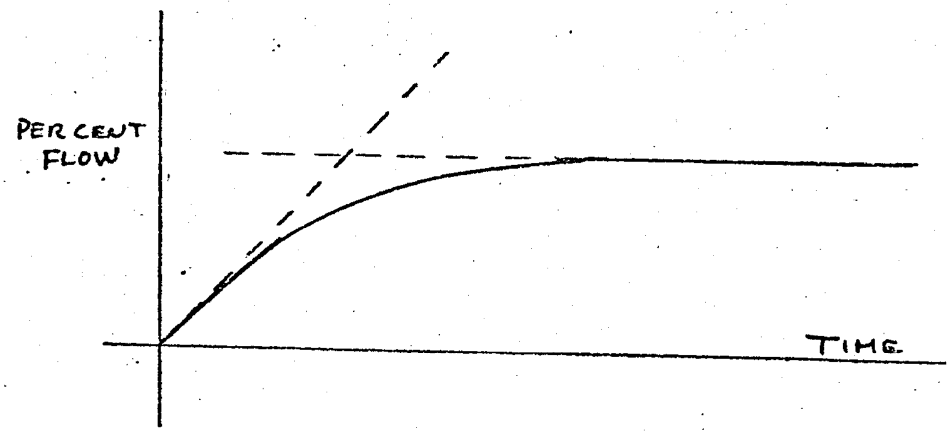


Figure 1

The water time $T_W$ is then the ratio of the percentage head step change to the initial slope of the flow-time curve of Figure 1.

The water starting time $T_W$ is normally between one and three seconds. Recently fairly large values have been encountered perhaps indicating an effort to reduce the cost of the penstock. These are minimum values of $T_W$, normally for closed casings.

$$(2) \qquad T_w \geq \frac{1}{3} + \frac{3D_o}{H}$$

and for open casings

$$(3) \qquad T_w \geq \frac{2D_o}{H}$$

Here $D_o$ is the runner outlet character in feet.

Pressure rise due to wicket gate closure can be related to the water time. The wicket gate closure time, $T_f$ is defined as the time in seconds required for the gates to traverse the entire stroke at the maximum velocity possible at any position. The maximum per unit pressure rise, $h_{max}$, may then be estimated from the relationship.

$$(4) \qquad \frac{h_{max}}{\sqrt{1+h_{max}}} = \frac{T_w}{T_f}$$

To a pressure rise computed in this manner, any affect upon pressure by speed rise would have to be added.

If the penstock is long, the time required for a pressure wave to travel the length of the penstock and return must be considered. This time is 2L/a where a is the speed of sound in water.

This time, 2L/a, is known as the reflection time $T_r$, and has considerable influence upon pressure rise and stability of the turbine governor combination.

The water time for the scroll case is obtained by using the effective length of the scroll case and the average velocity of the water. Thus for the scroll case the water time can be computed from the formula LV/2gH. The draft tube inertia is computed in the same manner as for the penstock, that is, summing the LV products for various sections of the draft tube.

The motion of the turbine-generator shaft is determined by the basic equation;

$$(5) \quad \text{(Inertia)} \begin{array}{c}\text{Shaft}\\ \text{Acceleration}\\ \text{Rate}\end{array} = \begin{array}{c}\text{driving}\\ \text{torque}\end{array} - \begin{array}{c}\text{Load}\\ \text{torque}\end{array}$$

When torque is expressed ft-lb and speed in revolutions per second the inertia factor in the above equation is $\pi WK^2/60$ where $WK^2$ is the total shaft inertia, generator plus turbine. A more common way to represent analytically the torque unbalance equation is to use per unit values of torque and speed. Then the equation is written

$$(6) \quad T_a \frac{dn}{dt} = m_t - m_g$$

where

$$T_a = \frac{0.62}{10^6} \frac{(WK^2) N_r^2}{P_r}$$

with $N_r$ = rated speed, rpm
$P_r$ = rated horsepower

Per unit variables are formed by referring their deviations to rated conditions, thus n = $\Delta N/N_r$. The term, $T_a$, known as the unit acceleration constant usually has values between five and twelve seconds. Another term more familiar to some is regulating constant, C and is equivalent to $1.62 \times 10^6 T_a$. An increase of acceleration time tends to reduce transient speed deviations.

The driving torque $m_t$ developed by the turbine is a function of speed, $n$ head $h$, and the gate opening as follows (next page)

$$(7) \quad m_t = (1+h)^{3/2} f_m(n, z)$$

Water flow through the turbine is dependent upon the same three variables, thus

$$(8) \quad q = (1+h)^{1/2} f_q(n, z)$$

Syrioptic Curves, $f_m(n, z)$ and $f_q(n, z)$ describing turbine operation throughout the full range of each variable are desireable, particularly if a detailed study of the system is required. It is common to linearize equations (5) and (6) to obtain

$$(9) \quad m_t = \frac{\partial m_t}{\partial n} n + \frac{\partial m_t}{\partial z} z + \frac{\partial m_t}{\partial h} h$$

$$(10) \quad q = \frac{\partial q}{\partial n} n + \frac{\partial q}{\partial z} z + \frac{\partial q}{\partial h} h$$

The six slopes can be read from the turbine synoptic curves at any particular operating point. Normal values of these slopes for an ideal turbine near the point of best efficiency are

$$\frac{\partial m}{\partial n} = -1.0 \qquad \frac{\partial q}{\partial n} = 0$$

$$\frac{\partial m}{\partial h} = 1.5 \qquad \frac{\partial q}{\partial h} = 0.5$$

$$\frac{\partial m}{\partial z} = 1.0 \qquad \frac{\partial q}{\partial z} = 1.0$$

These values are used when turbine performance curves are not available. For large load changes, 10% and above a non-linear representation of the turbine is required. For small changes the turbine may be linearized by using the six slopes at the operating point of interest. The slope, $\partial m / \partial n$, is more commonly known as the turbine self regulation factor and has been denoted by the letter $e_t$ by the hydraulic turbine technical committee of the IEC.

The load on the turbine at generator is normally very complex and difficult to represent exactly, as can be seen intuitively when we consider that a unit supplying power to an interconnection is influenced by the tie lines, the other generation sources and their governors, the type and location of load centers, and the individual voltage regulators. The result of this complication has been to take a pesimistic view, that is, if the worst case can be handled, then all other operation will be acceptable. A severe case encountered is that of a single unit supplying an isolated, resistive load. The load torque, $m_g$, is usually a function of speed and can be expressed by

$$(11) \quad m_g = m_o + e_g n$$

The coefficient, $e_g$, is commonly known as the load self regulation factor and will depend upon the type of demand upon the generator. For the resistive load and assuming the voltage regulation is instantaneous relative to the governor-turbine speed transients, $e_g$ has a value of $-1$. If a purely, inductive load is felt by the generator the coefficient is zero. This corresponds to constant torque motors. Figure 2 is a plot of load torque plotted against speed.

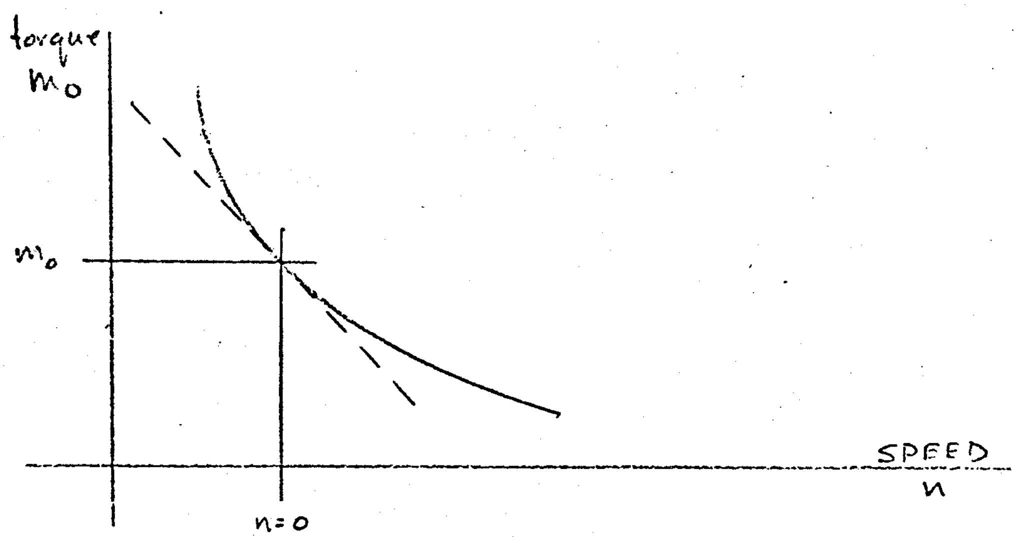


Figure 2

Hence for an increase of speed in the case of the resistive load, the load torque on the unit decreases which is in a direction to impair stability of the turbine. The turbine self regulation factor is normally of the same magnitude as the load self regulation factor, however they are in opposite directions and can cancel each other out.

An operating point of considerable interest is speed-no-load. From the standpoint of synchronizing it is important that the unit be stable and responsive to speed setting changes. At this operating point both self regulating coefficients near or at zero, again be by allowing $e_t$ to be zero we take a pessimistic, but safe, stand. If an exact value of $e_t$ were known, of course this value would be used to evaluate stability while synchronizing.

When the unit under study is supplying power for an infinite bus the expression for the load torque, $m_g$ , is

$$(12) \quad m_g = \frac{k_s}{p}(n - n_s) + k_d(n - n_s) + m_o$$

Where $k_s$ represents the relative synchronous torque and is equivalent to $3500/\theta$ where $\theta$ is the electrical phase shift in degrees between the unit and the system when generating rated power. The constant $k_d$ corresponds to damping torque related to fully connected amortisseur writings.

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