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IEEE Guide for the Application of Turbine Governing Systems for Hydroelectric Generating Units

IEEE Power Engineering Society

Sponsored by the
Energy Development and Power Generation Committee

IEEE

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IEEE Guide for the Application of Turbine Governing Systems for Hydroelectric Generating Units

Sponsor

Energy Development and Power Generation Committee
of the
IEEE Power Engineering Society

Approved 24 June 2004

IEEE-SA Standards Board

Abstract: This guide is intended to complement IEEE Std 125™-1988, providing application details and addressing the impact of plant and system features on hydroelectric unit governing performance. It provides guidance for the design and application of hydroelectric turbine governing systems. There is a heightened awareness within the electric utility industry of the importance in the effective application of governing systems for dynamic stability. The need exists to provide guidance in the effective governing system application for a better understanding among users.
Keywords: control, governor, governing system, hydroelectric, speed, stability

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Introduction

(This introduction is not part of IEEE Std 1207-2004, IEEE Guide for the Application of Turbine Governing Systems for Hydroelectric Generating Units.)

This document is a guide for the application of turbine governing systems for hydroelectric generating units. The Hydroelectric Power Subcommittee of the IEEE Energy Development and Power Generation Committee began to look into forming a working group to draft an application guide for hydroelectric units at the 1987 Winter Power Meeting. Subsequently, a PAR was issued and work began on the guide.

As progress was being made on the guide, governing technology was at the same time changing rapidly from mechanical to analog electronic to digital electronic controllers. Also, during this time period, new guides produced by working groups of the Hydroelectric Power Subcommittee addressed some portions of the original scope of this guide. Therefore, in 1998, the PAR for this Working Group was revised, and the Working Group's efforts were focused on producing a guide that acted as a companion document to IEEE Std 125-1988.

The final format of this guide contains four major clauses, which are directly related to the subject matter addressed in IEEE Std 125-1988. Clause 4 discusses the functions and characteristics of the turbine governing system and of the equipment related to the design of the turbine governing system. Clause 5 is somewhat tutorial in nature, discussing the major elements of the turbine governing system from a control theory perspective. Clause 6 provides some application insights to specifying a turbine governing system. Clause 7 provides a discussion of the issues related to the stability of the turbine governing system. Numerous bibliographic citations related to the subject matter are also provided, and examples are included to illustrate many of the systems and concepts discussed. Some more specialized information, dealing with the impact of turbine characteristics, system modeling and tuning, and performance auditing is presented within the informative annexes of the guide.

This guide is designed to be a reference document for practicing engineers in the hydroelectric industry. It is intended to offer application insight for applying turbine governing systems for hydroelectric units. IEEE Std 125-1988 offers guidance for what elements of a turbine governing system need to be specified, and this guide offers some experience-based guidance on the impact on system performance of these specifications.

Members of this Working Group represent a cross-section of the hydroelectric industry, including power plant owners, plant designers, equipment manufacturers, engineering consultants, and academic personnel.

The members of this Working Group wish to dedicate this guide to the memory of Bernard “Bud” Crittenden. Bud worked in the area of governing system design for 45 years. His numerous contributions to the industry involve many of the issues addressed by this guide. Perhaps Bud’s greatest contribution to the industry was his mentoring of a number of young engineers entering the field of governing system design and application. This guide can be viewed as a continuation of Bud’s work.

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IEEE Guide for the Application of Turbine Governing Systems for Hydroelectric Generating Units

1. Overview

1.1 Scope

This guide is intended to complement IEEE Std 125™-1988,[1] providing application details and addressing the impact of plant and system features on hydroelectric unit governing performance.

1.2 Purpose

The purpose of this guide is to provide guidance for the design and application of hydroelectric turbine governing systems. There is a heightened awareness within the electric utility industry of the importance in the effective application of governing systems for dynamic stability. The need exists to provide guidance in the effective governing system application for a better understanding among users. Present standards do not adequately address this need.

2. References

This guide shall be used in conjunction with the following publications. When the following specifications are superseded by an approved revision, the revision shall apply.

ANSI/ASME Std PTC29-1980, Speed-Governing Systems for Hydraulic Turbine-Generator Units.[2]

IEC 60308 (1970-01), International code for testing of speed governing systems for hydraulic turbines.[3]

IEC 61362 (1998-03), Guide to specification of hydraulic turbine control systems.

[1]Information on references can be found in Clause 2.

[2]ANSI publications are available from the Sales Department, American National Standards Institute, 25 West 43rd Street, 4th Floor, New York, NY 10036, USA (http://www.ansi.org/).

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IEEE Std 125-1988 (Reaff. 1996), IEEE Recommended Practice for Preparation of Equipment Specifications for Speed-Governing of Hydraulic Turbines Intended to Drive Electric Generators.[4]

ISO 4406:1999, Hydraulic fluid power—Fluids—Method for coding the level of contamination by solid particles.[5]

3. Definitions

For the purpose of this guide, the following terms and definitions apply. *The Authoritative Dictionary of IEEE Standards Terms*, Seventh Edition [B23][6] should be referenced for terms not defined in this clause.

3.1 beta ratio: A measure of the efficiency of a hydraulic oil filter, defined for a specific particle size as the ratio of the number of particles of the specified particle size that are trapped by the filter to the number of particles that pass through the filter.

3.2 damping ratio: This ratio of a second-order closed-loop control system is defined by:

$$\zeta = \sqrt{1 - (\omega_d / \omega_n)^2} \tag{1}$$

where

ζ is the damping ratio,
ω_d is the damped natural frequency of the system response after a step change,
ω_n is the undamped natural frequency of the system response after a step change.

This ratio is a measure of how oscillatory a control system is in responding to a step change. A control system with a damping ratio of 1.0 is critically damped, with no oscillatory action and no overshoot on the initial transient after a step change. A control system with a damping ratio of 0.0 is undamped, resulting in continuous oscillatory action.

3.3 electrohydraulic governor (sometimes called an electric-hydraulic governor): A turbine governing system that uses either analog electronic or digital electronic circuitry to develop the setpoint signal that is used to position the control actuators on the hydroelectric turbine. An electrohydraulic interface is used to convert the electronic setpoint signal into a hydraulic oil flow from a hydraulic servo valve system. The hydraulic servo valve system determines the position of the turbine control actuators.

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[6]The numbers in brackets correspond to those of the bibliography in Annex A.

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3.4 governor control system: A feedback control system that controls the speed and power output of a prime mover, such as a hydroelectric turbine. The governor control system comprises a "setpoint" or reference input, a feedback from the speed of the prime mover, optional feedbacks from other parameters as appropriate for the application, a controller function, and one or more control actuators.

NOTE—Figure 1 illustrates a basic governor control system.[7]

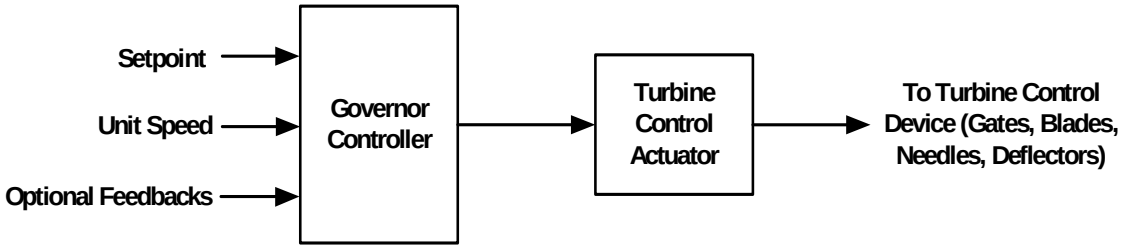


Figure 1—Basic governor control system

3.5 islanded operation: Operation of a generating unit that is interconnected with a relatively small number of other generating units, such as may occur after inadvertent tripping of circuit breakers that interconnect the island with a large interconnected power system. An island of generating capability may feed local loads connected to its electrical distribution system.

3.6 isolated operation: Operation of a generating unit without being interconnected with other generating units. An isolated unit may feed electrical loads connected to its electrical distribution system, such as the equipment within the plant that is powered by the station service system.

3.7 kidney loop filter: A hydraulic oil filtration system in which oil from the system's sump tank is continuously circulated through a filtration element to maintain the cleanliness level of the hydraulic oil.

3.8 mechanical-hydraulic governor: A turbine governing system that typically uses rotating weights to measure the rotating speed of the hydroelectric turbine. The setpoint signal used to position the turbine control servomotors is developed by mechanically linking the speed sensing device, the mechanical setpoint, and compensation devices to a hydraulic servo valve system. The hydraulic servo valve system determines the hydraulic oil flow to set the position of the turbine control actuators.

3.9 turbine governing system: A control system that controls the operation of a prime mover. The turbine governing system may include functions that are not directly related to the governor control of turbine speed or power output.

4. Functions and characteristics

This clause describes some of the functions and characteristics of system elements that are commonly involved in the specification and design of a turbine governing system.

4.1 Servomotor position feedback

Typically, the servomotor position feedback to the turbine governing system is calibrated at the ends of the servomotor travel (if possible), since these two points are the most repeatable positions achievable. For wicket gates, there is some "squeeze" near zero percent gate. The wicket gates contact each other before the

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wicket gate servomotor reaches the end of its travel. The closing force of the gate servomotor deflects the wicket gates and their connecting linkages, in effect "squeezing" them and improving the sealing of the wicket gates against water leakage. The governor system hydraulic pressure may affect the amount of wicket gate squeeze that can be achieved. Therefore, the most repeatable results can be achieved by calibrating the zero percent wicket gate position feedback position when the hydraulic pressure supply system is at its nominal pressure and the maximum achievable gate squeeze is applied.

4.2 Servomotor position

There is always some mechanical nonlinearity between the servomotor position and the controlled parameter, such as the wicket gate angle, wicket gate opening, or blade angle. The percent servomotor stroke is usually displayed and used by the turbine governing system. Some manufacturers, however, use linearizing calibration devices or algorithms to convert the servomotor position feedback to actual gate angle, gate opening, or blade angle. One benefit of feedback linearization is to make the governor parameters agree more closely with the turbine model data. For example, if blade control algorithms are based on gate opening rather than on servomotor stroke, using gate feedback linearization can help correlate the blade control cam or data map more closely with the actual turbine characteristics, improving the efficiency of operation. There is generally little or no impact on the achievable performance of the turbine governing system by calibrating the servomotor position feedback in this manner.

4.3 Servomotor time

The servomotor full-stroke travel time for hydraulic servomotors is affected by several factors. The mechanical loading of the servomotor affects its rate of travel. This mechanical loading consists of both frictional loading (of the turbine control device and its connecting linkages) and dynamic loading of the water passing through the turbine control device. The dynamic loading of the water passing through the wicket gates is a function of the wicket gate angle, and this effect is greater near the closed position of the wicket gates. The frictional loading can change as a function of the lubrication of the moving parts, the cumulative wear of these moving parts, and the angle of the wicket gates or runner blades. The dynamic loading of the water can be affected by the magnitude of the flow, the head across the unit, and the rate of travel of the turbine control device. Typically, the servomotor time is initially set with the unit unwatered. The servomotor time may be readjusted, if necessary, with the unit running.

Servomotor timing devices restrict the flow of hydraulic oil to limit the maximum travel rate of the hydraulic servomotor. Some examples of servomotor timing devices include orifice plates, timing valves, and adjustable mechanical stops ("stop nuts") on the distributing valve spool.

4.3.1 Maximum transient overspeed

The servomotor closing time of the primary turbine control device generally affects the maximum transient overspeed achieved by the turbine as a result of a full load rejection. In general, there is a maximum allowable transient overspeed that a unit can experience without resulting in an unacceptable level of degradation of the rotating components. After a load rejection, the primary turbine control servomotor should reduce the water flow to the turbine quickly enough to limit the maximum turbine speed to a value at or below this maximum overspeed specification. During certain portions of the transient overspeed condition experienced after a load rejection, the water flow through the turbine may actually be limited by the flow cutoff characteristics of the turbine rather than directly by the operation of the primary turbine control device.

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4.3.2 Maximum water passage pressure

The servomotor closing time of the turbine control device that controls the water flow through the water passage generally affects the pressure rise in the water passage due to the water hammer effect. In general, there is a maximum pressure that the water passage to the turbine can safely withstand without risk of the catastrophic rupture of some component of the water passage. The maximum closing rate of the turbine control servomotor that controls the water flow through the water passage should not cause the water passage pressure to exceed this maximum pressure specification.

4.3.3 Minimum water passage pressure

The servomotor opening time of the turbine control device that controls the water flow through the water passage generally affects the pressure drop in the water passage due to the water hammer effect. In general, there is a minimum absolute pressure that the water passage to the turbine can safely withstand risk of catastrophic collapse of some component of the water passage. The maximum opening rate of the turbine control servomotor that controls the water flow through the water passage should not cause the water passage pressure to drop below this minimum pressure specification.

4.4 Cushioning time

The cushioning time is the time of travel from the point at which the cushioning, or slow closure, feature takes effect until the turbine control device is fully closed. Cushioning is a feature that is built into turbine control servomotors, such as wicket gate servomotors, to soften both the mechanical impact and the water hammer effect when the turbine control device reaches the end of its travel. If two or more servomotors are used on a turbine control device, the connections from the servomotors to the turbine control device generally need to be adjusted so the cushioning effect becomes effective on all servomotors at the same time in the closing direction of the servomotor stroke. The cushioning time is generally adjusted with some type of flow control valve to set the cushioning time of the servomotors. Generally, each servomotor has a cushioning time control valve, and these valves should be adjusted so the servomotors equally share the mechanical loading of the turbine control device during the cushioned portion of the servomotor stroke. In some designs, there is a single cushioning control device for two servomotors.

4.5 Permanent speed droop and speed regulation

Speed droop (also known as “permanent speed droop”) and speed regulation (also known as “power droop”) are used to coordinate the responses of interconnected units to changes in the system frequency. Permanent speed droop can be developed either by using feedback from the wicket gate position (or, sometimes, from the governor’s position setpoint to the wicket gate actuator) or by using feedback from the unit power generation. If unit power generation is used to develop the permanent speed droop characteristic, the permanent speed droop term is usually called “speed regulation” or “power droop.”

4.5.1 Permanent speed droop

Permanent speed droop determines the amount of change in gate servomotor position a unit produces in response to a change in unit speed. Permanent speed droop (in per-unit terms) is defined as the change in unit speed (in % rated speed) divided by the change in governor output (% gate position). Permanent speed droop is usually expressed in terms of a percentage, which is 100 times the per-unit value. Figure 2 is a block diagram representation of a typical governor controller using permanent speed droop.

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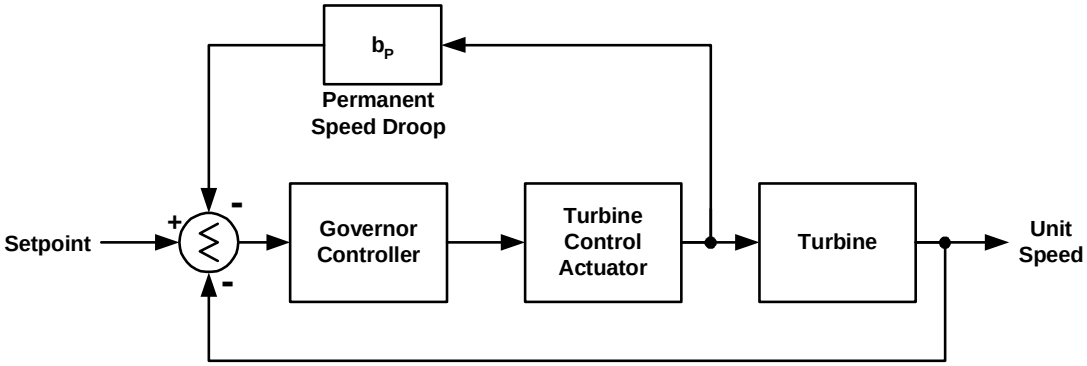


Figure 2—Typical governing system with permanent speed droop

The steady-state effect of adding the permanent speed droop feedback loop is a governor operating characteristic as shown in Figure 3. The value of permanent speed droop determines the slope of the characteristic curve. A typical permanent speed droop value is 5%, resulting in a unit that changes speed by 1% (0.6 Hz on a 60 Hz system) in response to approximately a 20% change in load (requiring a 20% change in wicket gate position) when operating connected to an isolated load. For a unit connected to a large interconnected power system, a unit with 5% permanent speed droop changes its gate position by 20% (approximately 20% change in output power) if the system frequency changes by 1%.

NOTE—Permanent speed droop relates changes in speed (or frequency) directly to changes in wicket gate position. Corresponding changes in turbine power output depend on the gate-to-power characteristics of the turbine.

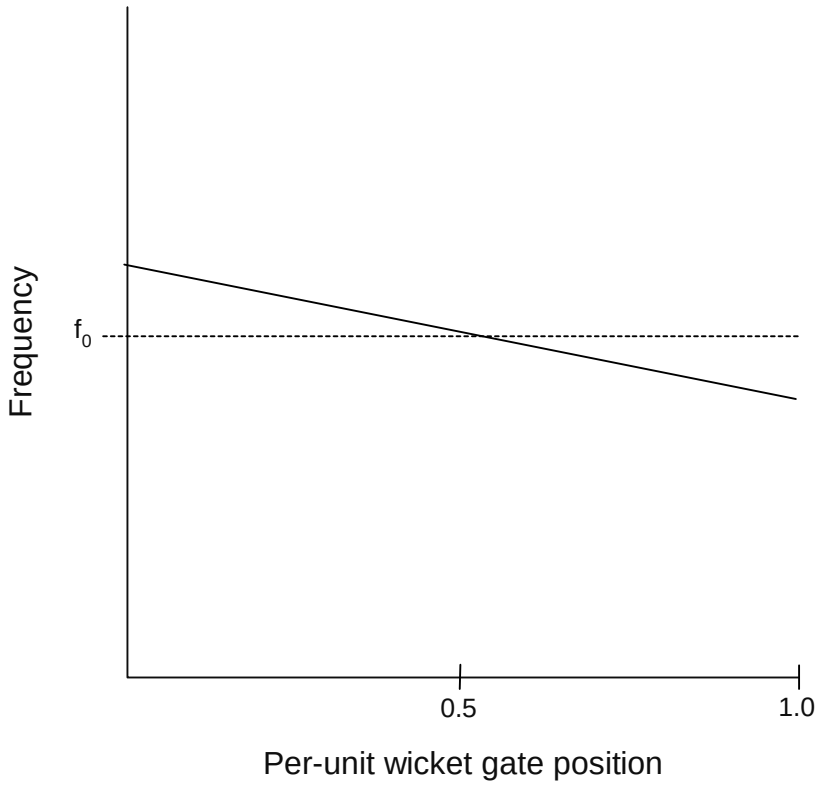


Figure 3—Permanent speed droop characteristics

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If two generators with different permanent speed droops are connected to a common load, they share changes in this load proportionally to their respective permanent speed droop characteristics, because their rotating speeds are dictated by the system frequency, which is common to the units. Figure 4 illustrates the steady-state load-sharing effect of permanent speed droop upon two units with different permanent speed droop settings.

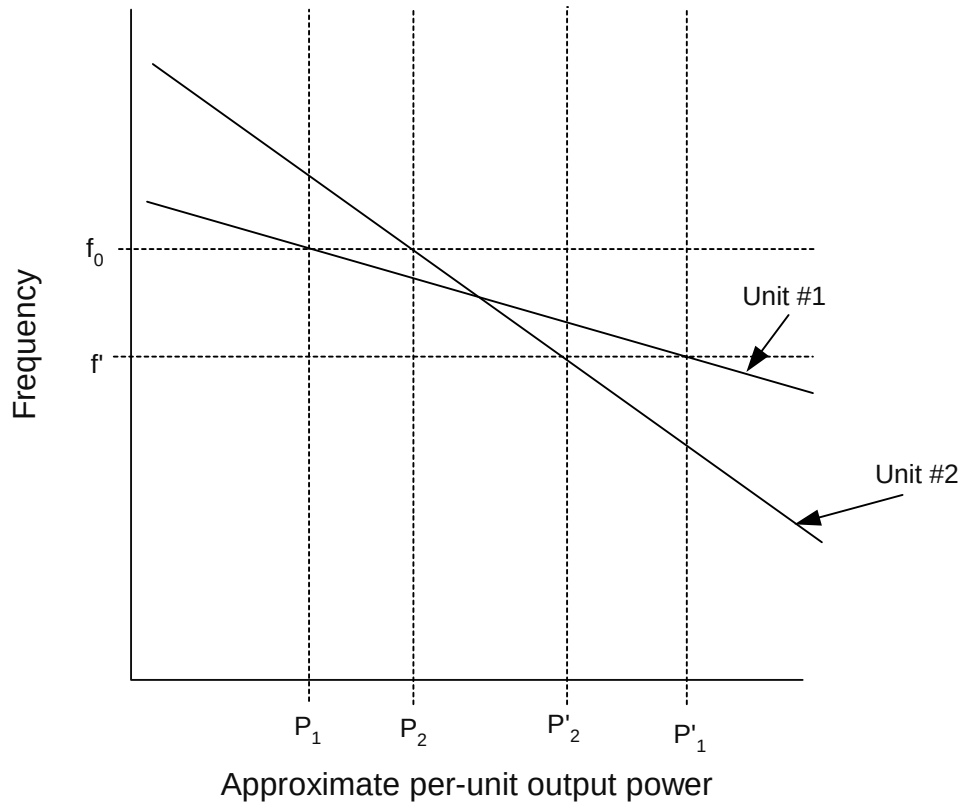


Figure 4—Allocation of unit loads with different governor permanent speed droops

It should be noted that if the governors on the interconnected units were adjusted for zero permanent speed droop, the units would not effectively share the system load. Differences in both the unit response times and in the governor calibrations would eventually result in one unit attempting to provide all of the load power, with the other unit being driven toward a motoring condition.

The governor parameter primarily used to control the operation of the unit is the “setpoint,” which is also known as the “speed reference,” the “speed adjustment,” the “speed-load adjustment,” or the “speed changer setting.” By changing the setpoint, the governor can be set to operate at the system frequency at any desired unit output. On a large interconnected system, the setpoint can be used to dispatch the desired generation into the grid when operating at rated system frequency. Figure 5 illustrates the steady-state effect of different governor setpoints on the output of the unit at rated frequency and 5% permanent speed droop.

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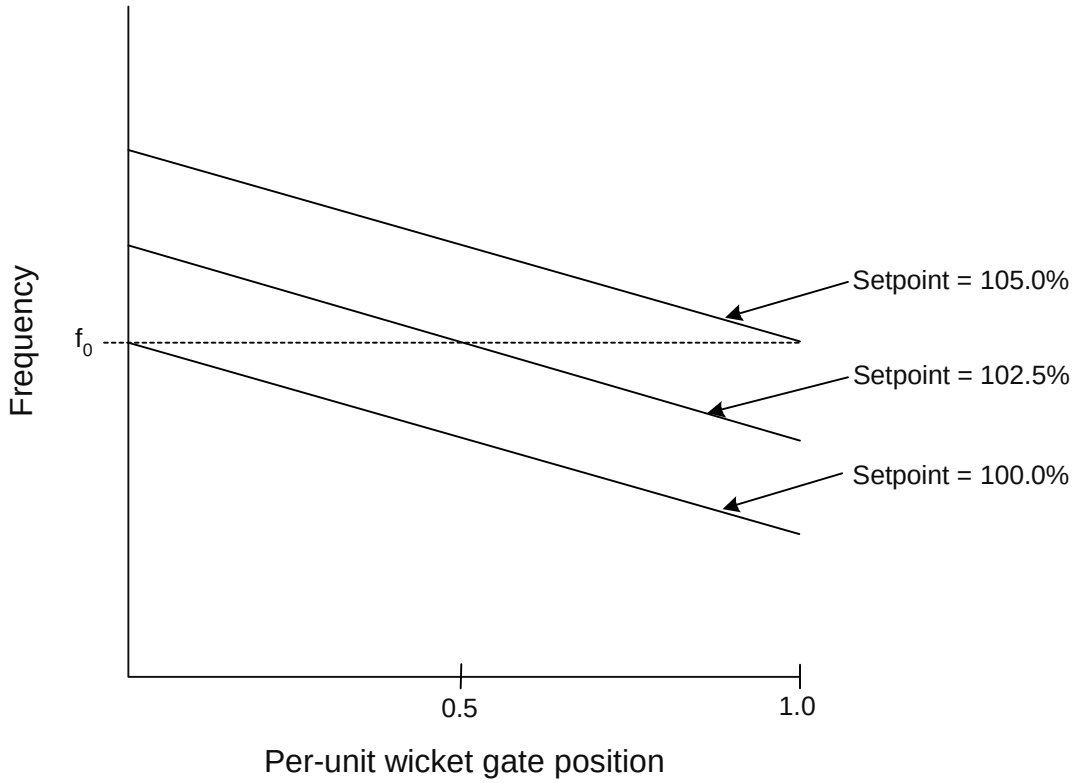


Figure 5—Unit loading with governor setpoint (at 5% permanent speed droop)

The principle of operation of a unit with permanent speed droop is essentially the same for both small, isolated systems and large, interconnected systems. The steady-state system frequency change after a significant disturbance, such as a large load trip or a large generation trip, is proportional to the size of the generation/load change. On an interconnected system, there is often a greater resulting frequency error than might be expected from the equivalent permanent speed droop of the units connected to the system (see Schultz [B41]). This additional frequency error typically results from nonlinear governor characteristics and units operating either at maximum generation or against their turbine control actuator position limits (limited or “nongoverning” operating mode).

It should be noted that, because of the permanent speed droop characteristics of the governors, after a system disturbance, the governors alone is not able to restore the system frequency to its predisturbance value (e.g., 60 Hz). The governors respond dynamically to the system disturbance, limiting the frequency disturbance to the permanent speed droop characteristic curve. Trimming out any residual frequency error as a result of the remaining load/frequency error may be done manually by operating personnel on small systems, or by an automatic generator control (AGC) on a large interconnected system.

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Figure 6 shows a frequency recording of a test performed in the Western Electricity Coordinating Council ("WECC," one of the North American reliability councils) where 1250 MW of generation was tripped. The total generation in the system was approximately 111 650 MW. The test was performed with all AGC systems in the WECC turned off. The settling frequency deviation 90 s after the trip was approximately –0.095 Hz, or –0.158%. As the 1250 MW generation trip was 1.12% of the total generation in the system, the "effective" permanent speed droop of the system is calculated to be approximately 14.1%. WECC's policy is to have all governors set to 5% permanent speed droop. The difference noted between the measured permanent speed droop and the theoretical permanent speed droop of the system is attributed to several phenomena occurring within the system, including nonlinearities in governors, which effectively change their permanent speed droop as a function of the unit loading levels and a number of machines operating in a limited (i.e., nongoverning) mode. The most likely largest contributing factor to the high value of measured system permanent speed droop is the effect of a significant number of units running in a limited mode of operation. If a unit is, for example, running against its gate limit, it cannot increase its generation in response to a drop in the system frequency. The equivalent permanent speed droop of such a limited unit is infinite, which raises the average equivalent permanent speed droop of the system in proportion to the rated generation of the limited unit.

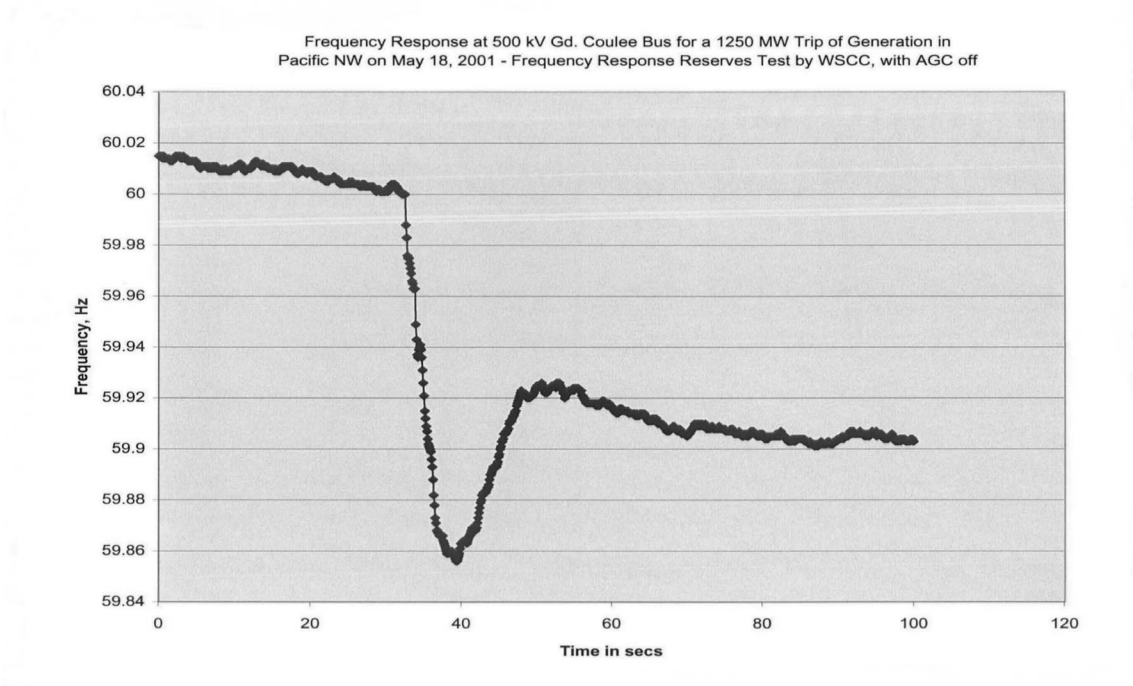


Figure 6—Example of testing the equivalent system permanent speed droop

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4.5.2 Speed regulation

A “speed regulation” (also known as “power droop”) governing system is similar to a permanent speed droop governing system, but with unit generation being used as the intermediate feedback from the controlled process instead of actuator position. Adding a portion of the unit power generation feedback to offset the detected unit speed error develops speed regulation. Figure 7 is a block diagram representation of a typical speed regulation governing system. The controller setpoint may be calibrated in terms of desired power output (e.g., megawatts) from the unit. The governor characteristic response for a speed regulation governor controller is essentially the same as the permanent speed droop characteristic described in Figure 3, except that the “per-unit output” axis is in terms of generated power rather than actuator position. Generally, a speed regulation characteristic can be beneficial when accurate dispatch of generation is required for system operation, or when a unit is required to operate at a constant base generation level despite any changes in operating head of the unit. However, using this type of feedback as the primary control feedback tends to de-stabilize governor operation if the unit ever experiences operation on a small, islanded, or isolated system.

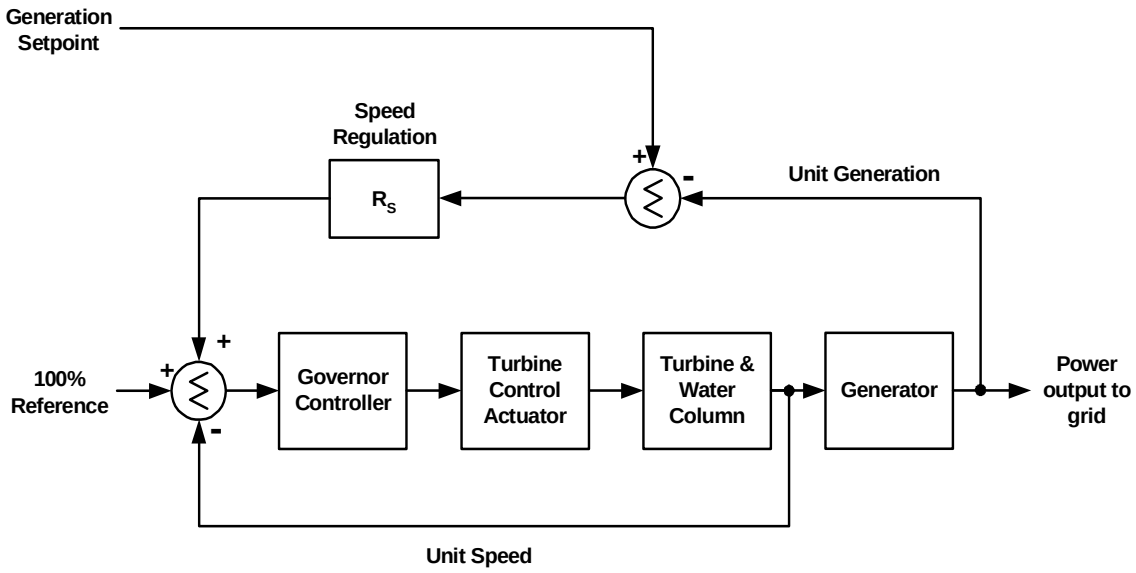


Figure 7—Governing system with speed regulation

Generally, governing systems using speed regulation are only appropriate for units that are generating into a large interconnected system, where the output of any single unit cannot have a significant impact on the system frequency. As with permanent speed droop governors, speed regulation governors respond to disturbances in the system frequency with changes in unit output that reduce the effects of a load/generation imbalance within the interconnected system. Using speed regulation governors on relatively small or islanded systems can result in the units responding in a manner detrimental to the system stability as a result of changes in system loads or other electrical system disturbances.

Synchronizing a unit with a speed regulation governor can sometimes be more difficult than with a permanent speed droop governor. This is because, with the unit breaker open, there is no power feedback to produce the permanent speed droop characteristic and the unit is essentially operating at zero permanent speed droop when synchronizing. A turbine governing system is inherently less stable at zero permanent speed droop, because a permanent speed droop feedback adds some degree of stabilizing influence to the control loop. The operating characteristics of some turbines at speed-no-load accentuate this influence. Operating the unit governor with conventional permanent speed droop when synchronizing, and then switching to speed regulation operation after closing the unit breaker, can avoid this stability problem.

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A speed regulation unit is inherently less stable than a permanent speed droop governor because additional dynamic influences from the water column are included in the primary feedback path (generated power) of the turbine governing system. The differences in these control systems can be seen by comparing Figure 2 with Figure 7. This difference in control loops generally requires that the speed regulation governor's damping adjustments (e.g., proportional plus integral plus derivative (PID) gains) be tuned for slower governor compensating action to achieve stable control using speed regulation. Additionally, dynamic water conditions such as draft tube surging have an influence on the generated power of the unit. These influences can result in undesirable movement of the turbine control actuator unless appropriate compensating measures are taken within the governor controller.

It is important to note that if a speed regulation governor controller uses a generation setpoint calibrated in units of generation (e.g., megawatts), the unit controls at its setpoint generation level only when the unit speed is at the 100% reference level. The composite error input to the governor controller algorithm is the summation of the generation error (multiplied by the speed regulation constant) and the speed error. Thus, the steady-state unit generation is a linear function of the unit speed, similar in nature to the permanent speed droop curve shown in Figure 3. The slope of the power droop characteristic response is determined by the speed regulation constant R_S.

4.6 Governor speed deadband

The governor speed deadband is a measure of the smallest speed change that can be detected and responded to by the turbine governing system. For a hydroelectric generating unit operating into an isolated system, the governor speed deadband determines the smallest band within which the unit can maintain the system frequency under steady-state loading conditions. Typically, a governor speed deadband of 0.02% is achievable and is commonly specified for hydroelectric turbine governing systems. Increasing the amount of speed deadband in a turbine governing system decreases the accuracy of frequency control that the governing system can achieve. Increased deadband also results in an increase in governor deadtime.

4.7 Blade control deadband

The blade control deadband is a measure of the smallest change in blade position setpoint that can be detected and responded to by the blade servomotor positioning system. The blade control deadband determines the accuracy of the gate/blade relationship for an adjustable-blade turbine. The accuracy of the gate/blade relationship determines the efficiency of the turbine as well as the amount of vibration and cavitation produced by off-peak operation. Typically, a blade control deadband of 1.0% is achievable and is commonly specified for hydroelectric turbine governing systems. Increasing the amount of blade control deadband decreases the accuracy of positioning the blades as a function of gate position. The resulting deviation from the ideal blade position reduces the efficiency of the turbine. This reduction in turbine efficiency may result in loss of revenue due to the reduced efficiency, increased cavitation damage to the turbine runner, and increased vibration damage to the turbine, generator, and bearings.

4.8 Governor deadtime

The governor deadtime is a measure of the amount of time elapsed between a change in speed to the first corrective action by the hydroelectric turbine governing system. The governor deadtime affects the peak overspeed after a load rejection as a result of the delay in governor response to the rising speed of the unit. The governor deadtime also affects the stability limit as achieved via the governor gain (or compensation) settings. Deadtime adds phase lag to the governing control system without a corresponding decrease in gain. This limits the amount of compensating gain that can be used in the governor controller. This limitation in governor stability requires the governor gains to be reduced to maintain control system stability. Reduction of governor gains makes the governor system slower to respond to system disturbances. Typically, a governor deadtime of 0.2 s is achievable and is commonly specified for hydroelectric turbine governing

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systems. However, modern control systems can often achieve shorter deadtimes due to improved control valve and control algorithm design.

4.9 Stability

The stability of a turbine governing system can be expressed as a damping ratio or as a settling time. These quantities cannot be measured directly, but they can be deduced from the measured response of the unit in response to a specified disturbance. Another method of specifying the stability of a governing system is to specify the relative size of successive peak deviations of the controlled speed after a disturbance. Typically, specifying the desired damping ratio and the settling time sufficiently defines the desired stability and responsiveness of the unit. As with many control systems for nonlinear processes, turbine speed governing systems may exhibit small oscillations around a steady-state operating point that are more related to the deadbands and nonlinearities of the system rather than to the stability of the control system.

4.10 Rated speed

The rated speed is the speed at which the generator frequency is at its rated value. If the generator is directly coupled to the turbine, the rated speed of the turbine is the same as the rated speed of the generator. If a speed increaser is used, the rated speed of the generator is greater than the rated speed of the turbine.

4.11 Overspeed

Any speed greater than the rated speed is referred to as overspeed and is typically expressed as a percent of the rated speed (e.g., 125% of rated speed is an overspeed condition). Sometimes, overspeed is expressed as a percent of unit speed in excess of 100% rated speed (e.g., 10% overspeed = 110% of rated speed).

4.12 Underspeed

Any speed less than the rated speed is referred to as underspeed and is expressed as a percent of the rated speed (e.g., 25% of rated speed is an underspeed condition).

4.13 Maximum momentary speed variation

The maximum momentary speed variation is the maximum change in unit speed when the unit load is changed by a specified amount. This variation is expressed as a percent of rated speed. The size of the load change, the characteristics of the turbine, the rotating inertia of the unit, the water column inertia, and the responsiveness of the unit governor determine this maximum momentary speed variation. The maximum momentary speed variation generally occurs when the unit governor responds to the change in unit speed and begins to return the unit speed toward its rated value.

4.14 Runaway speed

The runaway speed of a unit is influenced primarily by the turbine characteristics. Typical runaway speeds for reaction turbines range from 140% to 190% of rated speed for fixed-geometry turbines, and from 200% to 350% of rated speed for adjustable-blade turbines. Typical runaway speeds for impulse turbines range from 180% to 200% of rated speed. To avoid false operation of speed-related functions, the turbine governing system should be able to accurately measure and display unit speeds up to the runaway speed of the unit.

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4.15 Rated head

The rated head, as stated on the turbine nameplate, may be based on several different conventions. Often, the rated head is stated in terms of net head, which is the net pressure drop across the turbine runner, adjusted for the change in kinetic energy of the water as it passes through the turbine. The rated head may also be stated in terms of gross head, which is the net elevation difference between the headwater and the tailwater. The turbine is designed to operate over a range of head that reflects the normal head range that may be experienced during normal operation. The turbine nameplate rating may be stated as the maximum power output achievable at the minimum operating head, and it may also be stated as the maximum power output achievable at the maximum operating head. The nameplate convention used can vary, depending on the turbine manufacturer, the country of origin, and the requirements of the owner.

4.16 Steady-state governing speed band

The steady-state governing speed band is a measure of the peak-to-peak speed deviations that occur when the turbine governing system is controlling the speed of the turbine. This stability index is discussed in more detail in 7.1.1.1. This performance index is intended to represent the peak-to-peak speed deviations caused by the turbine governing system. In 9.2.7, a more detailed discussion of the interpretation of the steady-state governing speed band is presented.

4.17 Steady-state governing load band

The steady-state governing load band is a measure of the peak-to-peak deviations in generated power that occur when the turbine governing system is controlling the generation of the hydroelectric unit into a large interconnected power system. This stability index is discussed in more detail in 7.1.1.2. This measurement is done under constant-generation conditions, and it determines the peak-to-peak power deviations caused by the turbine governing system. Disturbances in unit-generated power induced by irregularities in water flow through the turbine should be excluded from the steady-state governing load band. Disturbances in unit generated power can also be induced by the actions of the excitation system for the generator. Normally, these disturbances are also excluded from the steady-state governing load band. However, if the turbine governing system is integrated into the same controller package as the excitation control system, the composite of disturbances caused by the turbine governing function and the excitation control system may be considered in computing the steady-state governing load band.

4.18 Speed

The instantaneous speed of the turbine is typically expressed either in percent of rated speed or in revolutions per minute. Normally, all speed-related indication, control, and protection functions use the same units of speed measurement.

4.19 Speed reference

A speed reference is either a fixed or an adjustable setting, usually expressed as a percentage of rated speed, which is compared with the actual speed of the turbine. An adjustable speed reference, sometimes called the “speed changer” or “speed adjustment,” is often the primary setpoint to the turbine governing system that is used to synchronize the unit to the interconnected power system. The speed reference is also often used to load the unit once it is paralleled to the interconnected power system. If a different governing system setpoint is used during online operation, such as a power setpoint or a flow setpoint, a fixed 100% speed reference is typically used for computing the unit speed error within the governing algorithm.

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4.20 Speed deviation

The speed deviation (also known as “speed error”) is the instantaneous difference between the speed reference and the actual turbine speed, usually expressed as a percentage of rated speed. The speed deviation may be used, via an adjustment of the speed reference, to control either the turbine speed or the generator power output.

4.21 Power output

The power output of a generator is generally measured at the generator terminals, in either kilowatts or megawatts. Typically, some electrical power is consumed during its operation for excitation, hydraulic pumps, and other equipment. The electrical power available for transmission to users outside the hydroelectric generating station is the difference between the power output of the generators and the power consumed within the station.

4.22 Rated power output

The rated power output of a generator is generally stated in terms of megawatts or kilowatts at a specified power factor and temperature rise. A generator may have more than one rated power output if different temperature rises or power factors are also specified.

The rated power output of a turbine is generally stated in terms of kilowatts or megawatts. The mechanical power output of the turbine is generally not measured directly because of the difficulties in measuring shaft torque.

4.23 Maximum power output

The maximum power output of the hydroelectric unit with maximum head and maximum gate may be greater than the generator can safely sustain for extended periods of time due to generator heating. If the maximum turbine output is significantly greater than the rated generator output, there may be a danger of slipping generator poles.

4.24 Governor controller

The governor controller accepts a setpoint command, compares this command against the feedback from the controlled process, and positions the turbine control actuator. The following subclauses discuss some of the types of governing controllers commonly used in the hydroelectric generation industry at the time of publication of this guide.

It is important to note that one common characteristic of all governor controllers is a responsiveness to the speed of the unit. Sometimes, when a unit is synchronized to a large interconnected system, the sensitivity of the turbine governing system to the unit speed is disabled as a result of certain operating practices common within the hydroelectric generating industry. A turbine governing system may be operated against its gate limit, or an artificial speed deadband may be tuned into the speed-sensing portion of the governor controller. This may be done to reduce unwanted control action induced by system frequency fluctuations or other control system irregularities. In these instances, the turbine controllers are no longer functioning as governing systems, and these units are not able to participate in unison with the governing systems of the other units connected to the power system that are dedicated to maintaining the system frequency.

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4.24.1 Temporary droop controller

A temporary droop governor controller uses a feedback function from the governor actuator position to "temporarily" cancel part of the error between the governor setpoint and the unit speed feedback. This feedback function may be described as a lead-lag, a reset, a washout, or a filtered derivative function. The net error resulting from this summation is integrated by the governor controller to position the turbine control actuator. The temporary droop feedback within the governor controller helps to stabilize the control of unit speed by reducing overtravel of the turbine control actuator. Typical stabilizing adjustments for temporary droop governor controllers are error integration gain, temporary droop (in percent), and damping device time constant (in seconds). Originally, temporary droop governor controllers were implemented using mechanical devices such as floating levers (for summation and gain functions) and dashpots (for stabilization using temporary droop). This same controller strategy has also been implemented both in analog and digital electronic control systems. A temporary droop governor controller is approximately equivalent to a proportional plus integral controller. A functional block diagram for a temporary droop governor controller is shown in Figure 8.

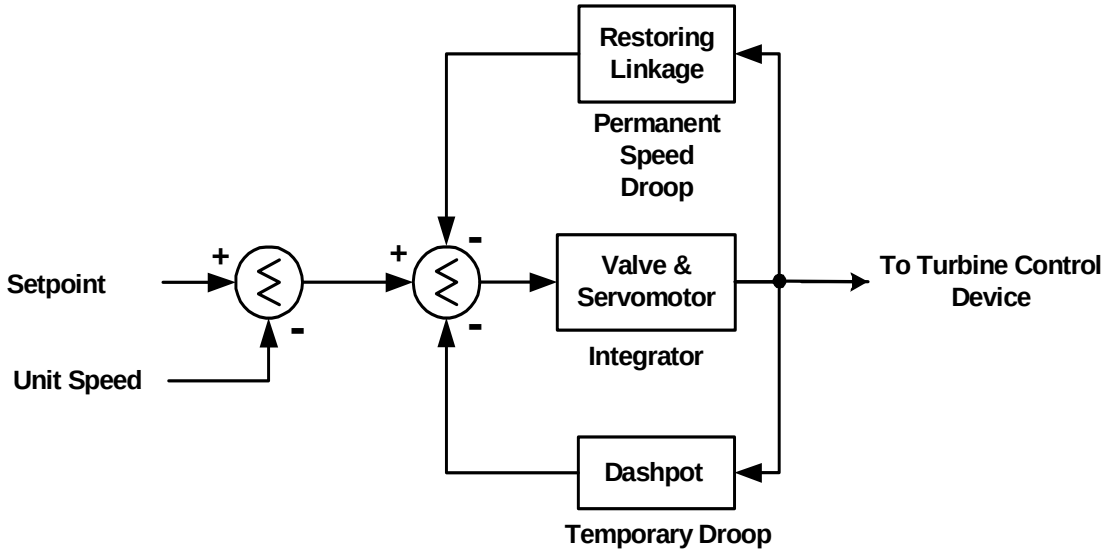


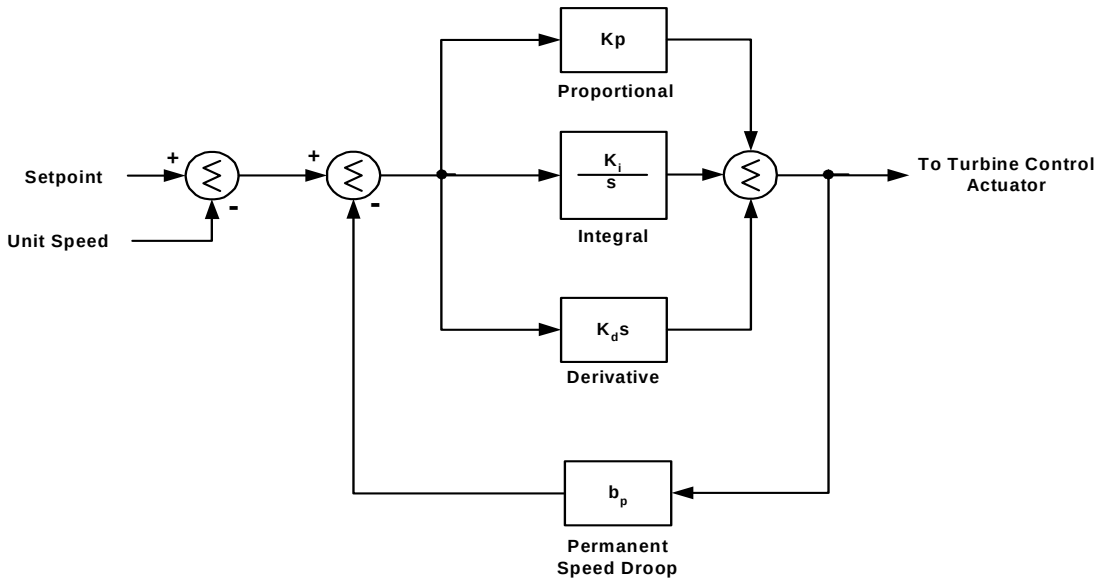
Figure 8—Typical temporary droop governor controller

4.24.2 PID controller

A PID governor controller uses proportional plus integral plus derivative terms to process its error input into a command signal to the turbine control actuator. The proportional term produces a control action proportional to the size of the error input. The proportional term produces an immediate response to an error level input, and typically, it has a significant influence on the stability of the governed system. The integral term produces a control action that accumulates at a rate proportional to the size of the error input. The integral term works in unison with the proportional term to determine the stability of the governed system. The integral term also trims out the error input to the governor controller to determine the steady-state accuracy of the governed system. The derivative term produces a control action that is proportional to the rate of change of the error input. The derivative term helps to extend the stability limits of the governed system by allowing higher proportional and integral gains while maintaining a stable control system. Typical stabilizing adjustments for a PID governor controller are proportional gain, integral gain (in inverse seconds), and derivative gain (in seconds). PID governor controllers have been implemented in both analog and digital electronic control systems. A functional block diagram for a typical PID governor controller is shown in Figure 9.

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where

K_P is the proportional gain, p.u.,

K_i is the integral gain, s^{-1},

K_d is the derivative gain, s,

b_P is the permanent speed droop, p.u.,

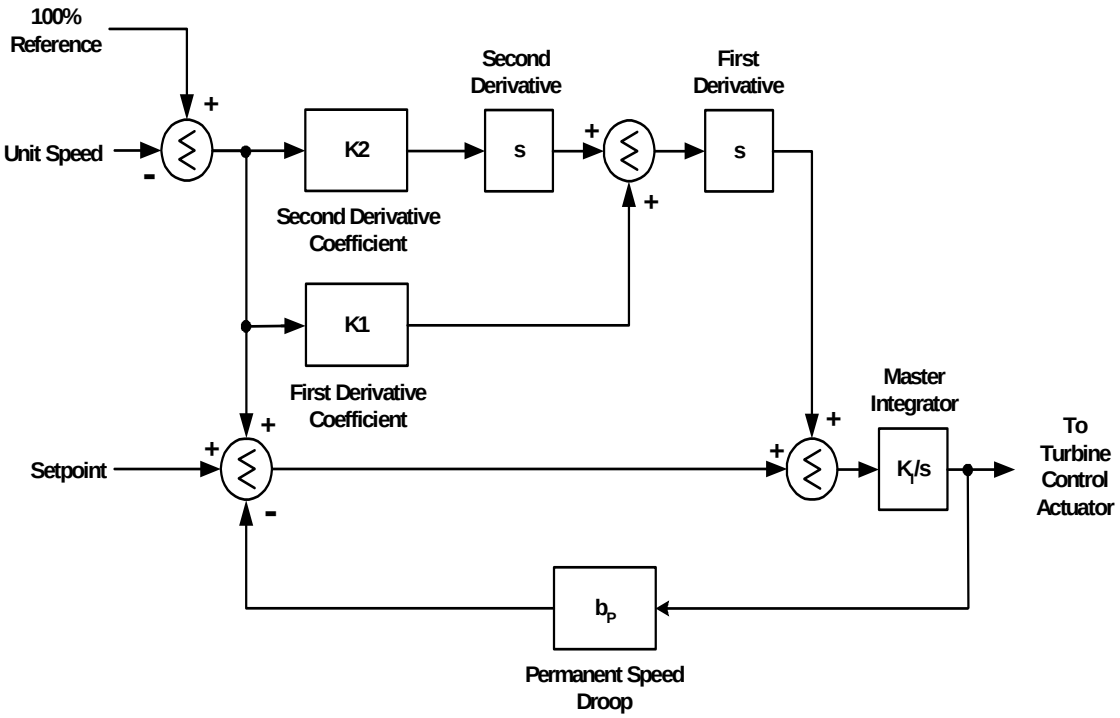
s is the Laplace operator.

Figure 9—Typical PID governor controller

4.24.3 Double-derivative controller

A “double derivative” governor controller is a variation of the PID governor controller. In this governor controller strategy, a proportional term, a first derivative term, and a second derivative term process the input error. The summation of these three terms is then integrated by the output stage of the governor controller. This final integrating stage may be either an electronic integrator or a hydraulic integrator. The double derivative governor controller strategy can result in a lower overspeed peak upon startup of the unit, and it can result in a smaller overtravel of the turbine control actuator when a position limitation is released, when compared with a corresponding PID controller. This controller strategy also eliminates the derivative and proportional influences from changes in the setpoint. Typical stabilizing adjustments for a double derivative governor controller are the first derivative gain (similar to the PID proportional term), second derivative gain (similar to the PID derivative term), and the overall integral gain. A functional block diagram for a typical double derivative governor controller is shown in Figure 10. Other structures may be implemented to achieve the same functionality as described in Figure 10.

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where

K_1 is the first derivative gain coefficient, s,

K_2 is the second derivative gain coefficient, s^2,

K_T is the overall integral gain, s^{-1},

b_P is the permanent speed droop, p.u.,

s is the Laplace operator.

Figure 10—Typical double-derivative governor controller

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4.24.4 Feedforward controller

Feedforward is a predictive control strategy that can be used in conjunction with any closed-loop controller to achieve a faster response to a change in setpoint without compromising the contribution of the unit governor controller to the overall system stability. Historically, many governing units were switched to a separate set of online damping parameters (also known as “dashpot bypass” or “online gains”) when synchronized to a large interconnected system. These online damping parameters allowed the unit to respond quickly to power dispatch commands. The unit derived its speed stability from its interconnection with the large interconnected power system. However, the high gains from speed to wicket gates could cause the system frequency to become unstable if the unit became islanded from the grid or if high online gains were used on a significant portion of units on the grid. Figure 11 shows how the setpoint feedforward strategy is used with a typical closed-loop governor controller.

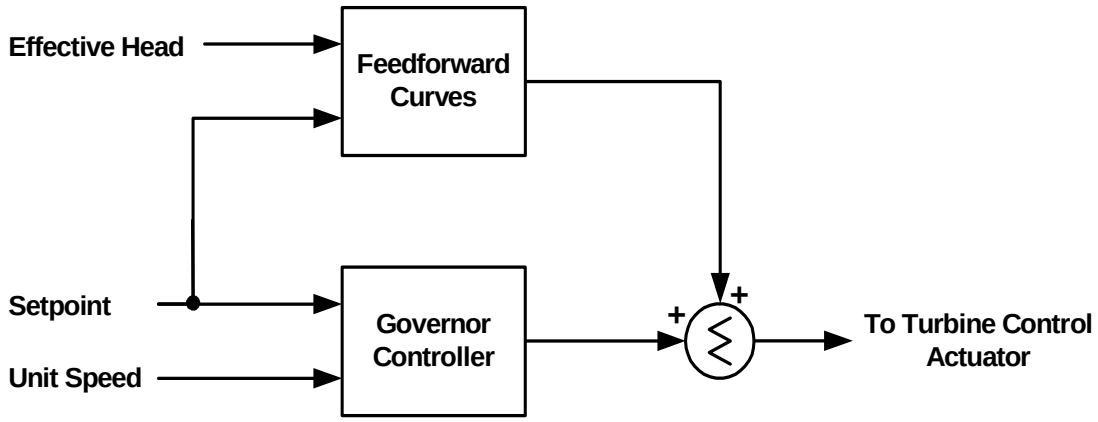


Figure 11—Typical governor controller with speed feedback and setpoint feedforward

It is important to note that a feedforward strategy be designed for operation only while the unit is paralleled with a large interconnected power system. The feedforward function is typically disabled when operating off-line, or when the unit becomes islanded or isolated, as detected by a significant disturbance in the unit speed. If a unit’s generation versus gate position characteristics are not significantly influenced by the unit’s operating head at a particular installation, the feedforward function may be simplified by eliminating the effective head input. If, however, the unit’s generation versus gate position characteristics are significantly influenced by other operating parameters, such as the flow through other units sharing the same water passage, these conditions should be accommodated by the feedforward curves to achieve acceptable performance by the governor controller. If the factors influencing the unit’s generation versus gate position characteristics become too complex, it may be more expedient to eliminate the feedforward function from the governor controller strategy and depend on the governor controller gains to provide the desired online response to the governor setpoint.

4.24.5 State space controller

A state space controller is a predictive control strategy, and it may be used to optimize the response of a well-defined system by modeling the system characteristics and taking the control action necessary to achieve the desired control response. An accurate model of the controlled process is required to implement a state space control strategy. A state space control system may be used in conjunction with a feedback control system. The purpose of the feedback control system is to trim out any errors that may occur as a result of inaccuracies in the state space controller model. A functional block diagram of a typical state space controller with a feedback control trim function is shown in Figure 12.

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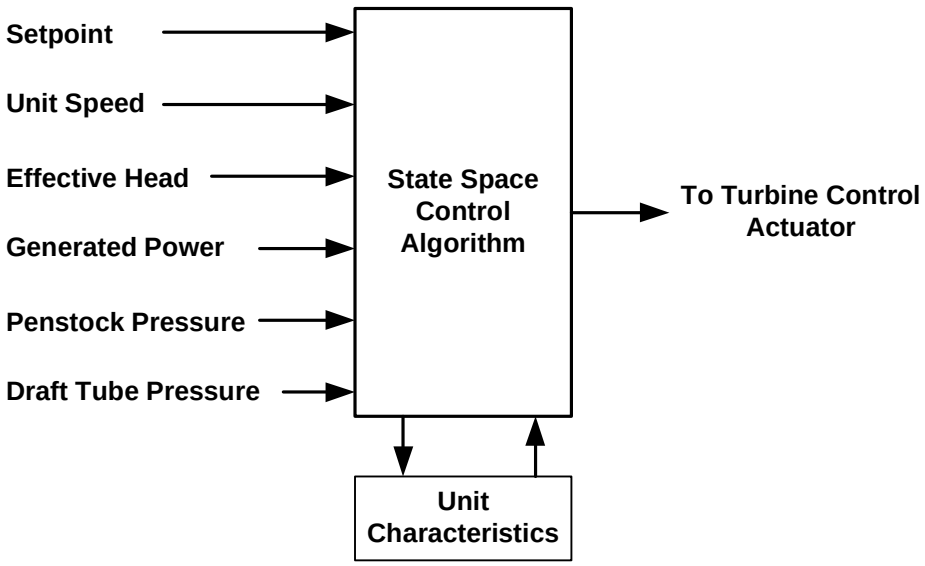


Figure 12—Typical state-space controller

4.24.6 Impulse turbine controller

An impulse turbine presents a unique challenge for a turbine governing system. The impulse turbine needles control the flow of water into the turbine. As many impulse turbines use relatively long water conduits, the water hammer effect can be significant, requiring a relatively long needle servomotor operating time. To achieve the desired dynamic operation, many impulse turbines include deflectors that divert the water stream away from the turbine buckets, thus removing some or all of the water energy from the turbine runner. The deflectors do not affect the flow rate of the water in the conduit, so the deflector servomotors may operate at relatively fast servomotor timings.

4.24.6.1 Water-saving control mode

A water-saving control mode for an impulse turbine achieves maximum turbine efficiency by positioning the deflectors outside the water stream, thus allowing all water discharged from the turbine needles to impinge on the turbine runner. To reduce the deadtime of bringing the deflectors into the water stream when needed, the deflectors are often placed as closely as possible to the edge of the water stream using either a two-dimensional function based on needle servomotor position or a three-dimensional function based on needle servomotor position and turbine operating head. Figure 13 is a block diagram representation of a classical water-saving governor controller strategy for impulse turbines.

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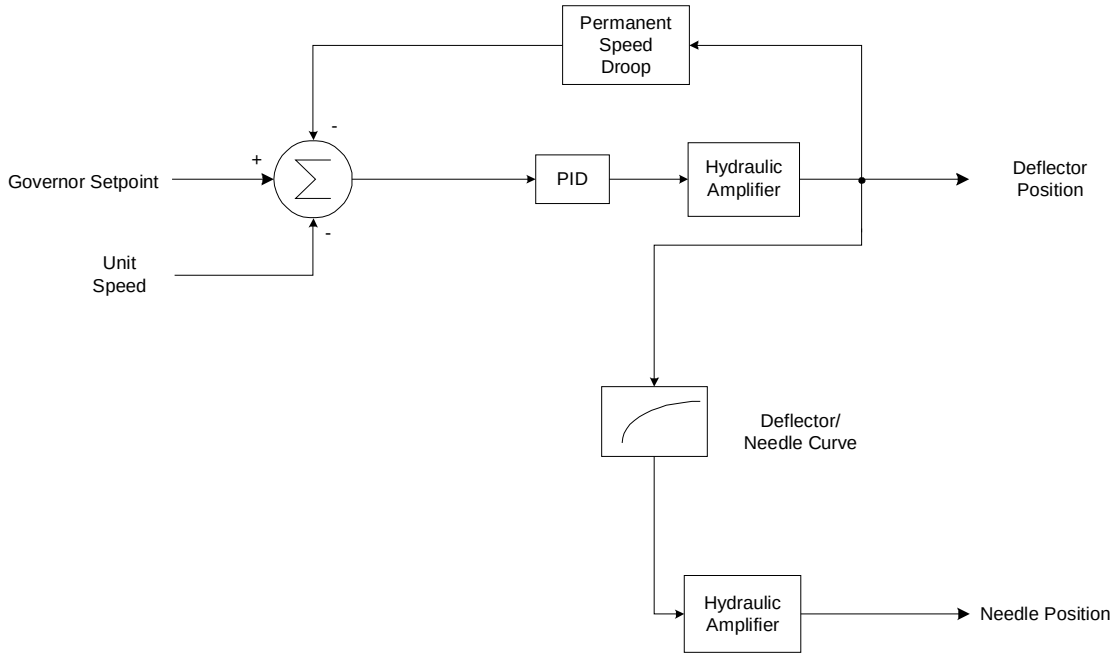


Figure 13—Classic water-saving governor controller for impulse turbines

4.24.6.2 Water-wasting control mode

Deflectors can be used to rapidly decrease the amount of water that impinges on an impulse turbine runner. However, if the deflector is outside the water stream, it cannot rapidly increase the amount of water impinging on the turbine runner. If an impulse turbine is used to control the frequency of an isolated electrical system, the deflectors can be positioned slightly within the water stream to allow for the rapid increase of water flow to the turbine runner in response to an increase in electrical load connected to the system. It is important to note that when the deflectors are in the water stream, their action dominates the control of the turbine speed. Because deflector motion does not result in a water hammer effect, the governor can be tuned to be very responsive in this mode. Once the deflectors are out of the water stream, they no longer have any effect on the turbine operation. At that point, only the needles have an effect on the turbine operation. If a water-wasting mode of operation is specified, it should specify a method of determining the amount of hydraulic power to be deflected away from the turbine runner. This determines the amount of load acceptance the turbine can handle while keeping the deflectors within the water stream. Any water-wasting mode of operation should have a method of coordinating the dynamic response of the turbine governing system when the deflectors are within the water stream and when the deflectors are outside the water stream. When operating in a water-wasting mode, the deflectors continuously deflect water away from the turbine runner. The deflectors should be designed to withstand the erosion effects of operating in this mode for extended periods of time.

4.24.6.3 Needle sequencing

To achieve higher operating efficiencies at different power levels, an impulse turbine may sequence its needles to use different numbers of needles to operate at different power levels. Normally, sets of needles are used that are symmetrically spaced to minimize the radial loading of the turbine guide bearings. Switch points for operating with 2, 3, 4, or 6 needles (for example) should be determined from the crossover points of the turbine efficiency curves.

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4.24.6.4 Needle/deflector dynamic coordination

To optimize the dynamic operating characteristics of the impulse turbine governing system, the governor controller may have separate compensating gains (e.g., PID gains) for operation with the deflectors within the water stream and with the deflectors outside the water stream. In this way, both the deflector servomotor and the needle servomotor can respond simultaneously to a frequency disturbance. The effects of these respective servomotor responses are determined by the mode of operation (e.g., water-saving or water-wasting). This approach should have some method of returning the deflectors and needles to the desired relationship during steady-state operating conditions. Figure D.4 is a block diagram representation of a dual-loop governor controller strategy for impulse turbines.

4.25 Stabilizing adjustments

For any turbine governing control system to achieve the desired performance, it should be tuned to accommodate the characteristics of the turbine, the water inertia, the rotating inertia, and the load characteristics. Annex F summarizes some methods used to tune governing systems. Brief descriptions of some typical stabilizing adjustments, and their impact on performance, are provided in 4.25.1 – 4.25.6.

4.25.1 Temporary droop

Temporary droop is used to limit overshoot of the turbine control servomotor during a transient condition. Temporary droop may be developed either mechanically or electronically in the governing controller. Mechanically, connecting a dashpot (hydromechanical lead-lag, or reset, device) develops temporary droop from the wicket gate position to the governor error summing point. Electronically, temporary droop may be developed by adding a filtered derivative of wicket gate position to the governor error summing point, or it may be approximated as a PI controller.

4.25.2 Time constant of the damping device

The time constant of the “damping device” determines the rate at which the effect of the temporary droop decays to zero. If temporary droop is developed mechanically with a dashpot, this time constant is adjusted with a needle valve to determine the equalization time of the dashpot pistons. If temporary droop is developed electronically with a filtered derivative term, the time constant of the damping device is the filter time constant of this derivative term. If temporary droop is approximated by using a PI controller, then the time constant of the damping device may be approximated as the proportional gain divided by the integral gain.

4.25.3 Integral gain

Integral gain produces a controlling action that is the time integral of the governor controller error in a PI or a PID governor controller. Integral gain is also known as “reset,” and it acts to trim the controller error to a value of zero. The rate of trim is proportional to both the magnitude of the error and to the integral gain setting. Integral gain is defined as the rate of change (in percent/second) of the integrator output divided by the governor controller error (in percent).

4.25.4 Proportional gain

Proportional gain produces a controlling action that is proportional to the magnitude of the controller error in a PI or a PID governor controller. The proportional gain contributes an immediate response to a governor controller error. Proportional gain is defined as the magnitude of the proportional element output (in percent) divided by the magnitude of the governor controller error (in percent).

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4.25.5 Derivative gain

Derivative gain produces a controlling action that is the time derivative of the governor controller error. The derivative gain term acts to limit controller overshoot by reducing controller action proportional to the rate of change of error input to the governor controller. Derivative gain is defined as the magnitude of the derivative element output (in percent) divided by the rate of change of the governor controller error input (in percent/second).

4.25.6 Permanent speed droop

Permanent speed droop determines the amount of change in output a unit produces in response to a change in unit speed. Permanent speed droop can be developed either by using the wicket gate position or by using the unit power generation. If unit power generation is used to develop the permanent speed droop characteristic, the permanent speed droop term is sometimes called “speed regulation” or “power droop.” Permanent speed droop (in per-unit terms) is defined as the change in unit speed (in percent rated speed) divided by the change in governor output (percent gate position or percent rated power generation). Permanent speed droop is often expressed in terms of a percentage, which is 100 times the per-unit value.

For isolated or islanded operation, the permanent speed droop, if derived from the turbine control servomotor position, can help to stabilize the unit speed control by providing an intermediate feedback that limits the overtravel of the turbine control servomotors while controlling the unit speed. For operation while synchronized to an interconnected power system, the permanent speed droop determines the amount of participation the unit produces when responding to disturbances in system frequency. This participation should be coordinated with the other interconnected units to assure stable operation of the power system.

4.26 Water inertia time

The water inertia time (often called the “water starting time”) affects the water hammer effect of changing the water flow through the turbine using the primary turbine control device, such as the wicket gates. This parameter also affects the small signal stability of the frequency control system. The dynamic performance of the control system is very much dependent on the penstock water time T_W. T_W is a simplified approximation of the relative inertia of the water column, and it does not include the effects of wave time on the dynamic influence of the water column. Water starting time is normally calculated for the unit at rated generation and at rated head. Other turbine operating conditions have different effective water starting times. For a simple penstock system, the classical treatment of water start time computes its value as in Equation (2):

$$T_W = Q/g \times H\sum L/A \tag{2}$$

where

T_W is the water start time, s,

Q is the rated water flow, m^3/s,

g is the gravitational acceleration, m/s^2,

H is the rated operating head, m,

L is the length of a given portion of water column with constant area, m,

A is the cross-sectional area of the corresponding portion of water column, m^2.

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Normally, dynamic performance degrades with increasing T_W. As T_W is directly proportional to the length of the penstock, control performance degrades with increasing penstock length. The computation of the water start time is meaningful only if the water start time is relatively short. For very long penstocks, the wave travel time of the water column becomes significant, and the reflected pressure waves in the water column cause the preceding treatment of water start time to no longer be valid. When the wave travel time approaches 25% of the T_W, the classic value of T_W should not be relied on, and the performance of the turbine governing system should be evaluated by considering the effects of both the water starting time and the wave travel time.

On long penstocks, a surge tank is often added. A surge tank consists of a branch in the water passageway that is typically vented to atmosphere. The elevation of the surge tank's vented end should be sufficient to prevent overflowing of water under normal operation of the unit. The surge tank reduces the effective water starting time by absorbing some of the inertial effects of the water flow, temporarily providing or storing water from the penstock in response to pressure changes within the penstock (see Chaudhry and Hanif [B6]). The surge tank also reduces the effect of the traveling pressure wave in the water column upon the unit operation by reflecting the pressure wave back toward the unit. This reduces the effective wave travel time, making it less likely to impact the performance of the turbine governing system (see Sanathanan [B37] and Trudnowski and Agee [B42].

4.27 Mechanical inertia time

The mechanical inertia time (often called the "mechanical starting time," [see Equation (3)] and the inertia constant [see Equation (4)] are normally used to characterize the mechanical inertia of the rotating parts:

$$T_M = I \times \omega^2 / P \tag{3}$$

where

T_M is the mechanical inertia time, s,
I is the combined rotating inertia of generator, turbine, shaft, flywheel, and generator speed increaser, kg-m^2,
P is the rated power output of the unit, W,
ω is the rated rotational speed of the turbine, rad/s.

$$H = T_M / 2 \tag{4}$$

where

H is the inertia constant, s.

Experience has shown that the turbine rotating inertia is normally approximately 5% of the generator rotating inertia. The choice of generator inertia is an important consideration in the design of a hydroelectric plant, as outlined in the following:

The speed rise of the turbine-generator unit under load rejection conditions is inversely proportional to T_M (and thus inertia). Choice of plant design parameters involves optimization of unit inertia, effective gate closing time, and penstock design, to control pressure rise, speed rise, and costs. Increasing T_M (rotating inertia) reduces the maximum transient speed rise, but at additional cost. Additional inertia may be provided either by increasing the mass of the generator rotor or by adding a flywheel.

The inertia constant, H, also has a significant effect on electrical system transient stability, as this factor influences the speed with which energy can be moved in or out of the generator to control the rotor angle acceleration during system fault conditions.

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4.28 Impact upon stability

Both the water inertia time (T_W), also known as the "water starting time," and the mechanical inertia time (T_M), also known as the "mechanical starting time," affect the achievable controllability of the turbine governing system. One benchmark that is often used to estimate the controllability of a hydroelectric generating unit's governing system is the ratio of the T_M to T_W. Although steady-state stability can generally be achieved for essentially any T_M/T_W ratio (see Wozniak [B44]), a ratio of approximately 1:1 is often considered to be a practical lower limit for achieving acceptable controllability for an isolated unit that experiences normal load transients. Classically, many older hydroelectric generating units had T_M/T_W ratios in the range of 3:1 to 4:1. Older mechanical-hydraulic governors using a temporary droop type of governor controller strategy could generally achieve acceptable controllability for T_M/T_W ratios as low as 2:1. Units with lower T_M/T_W ratios, down to approximately 1:1, generally require electrohydraulic governors using a PID (or similar) type of governor controller strategy to achieve acceptable controllability.

4.29 Automatic generation control

An interconnected power system consists of a number of control areas (each consisting of one or more power companies) interconnected with each other through tie lines. In each control area, a central computer-controlled AGC system controls to a target steady-state frequency and the net tie-line power exchanges between the control areas in the interconnected system. Generation and frequency control is thus performed on a decentralized basis. Each control area tries to maintain the amount of generation required to meet internal load, adjusted by the interchange demand. The frequency of the interconnected system is coordinated and maintained by the cooperative operation of all interconnected control areas.

Before the development of AGC, speed governors provided frequency control. This often placed an excessive burden on the governing systems of a few units. Furthermore, after a disturbance, governors with permanent speed droop could not restore frequency to the predisturbance frequency level. With AGC, the governing burden is distributed among many units at various plants so that each unit assumes its regulating share depending on unit capability, unit response, transmission availability, economic loading, and so on, as determined by the control area operator. The net interchange of power with neighboring control areas is maintained using real-time power measurements of all tie lines. After a system disturbance, such as a large generator outage when the system settles to a lower frequency (determined by the permanent speed droop settings), the AGC-controlled units increase their generation level to restore the system frequency to its predisturbance level. In small systems without AGC, operators perform this restoration of frequency manually on the governor frequency or power reference settings.

The AGC interfaces with the unit turbine governing systems either directly or via a plant-level control system. The AGC system issues raise/lower commands to adjust the setpoint for the unit or station control system, or the AGC system may issue a level setpoint (generally in terms of power generation) to determine the desired output of the unit or of the plant.

Hydro plants that are capable of rapid response are good candidates for AGC. One approach to achieve rapid stable response is to use setpoint feedforward within the turbine governing system. This approach allows the governor stabilizing parameters (i.e., gains and damping adjustments) to be adjusted so the hydroelectric generating unit contributes to the frequency stability of the interconnected system, and the feedforward function achieves the desired unit responsiveness.

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4.30 Efficiency optimization

On a unit basis, the efficiency of an adjustable-blade turbine may be optimized by use of a blade/gate relationship that is also dependent on the operating head of the unit. By positioning the turbine blades as a three-dimensional function of both gate position and operating head, the operating efficiency of the turbine is optimized, based on the most recent turbine efficiency testing of the turbine that produced the gate/blade relationship used in the blade controller. This optimization strategy is often called “3D blade control.”

With the proliferation of modern computer-based control systems, software is sometimes applied to optimize the operational efficiency of hydroelectric generating stations. These optimizing software programs normally reside in the plant control system with control interfaces to the unit governor control systems. These programs contain representations of the unit efficiency curves for various operating heads as determined from test measurements. Optimization of the station’s output is based on preestablished algorithms, which determine the unit generation levels to maximize the efficiency of the station under all conditions. Although these optimization programs usually reside outside the unit governing system, they typically interface with it through a station-level communication system.

In addition to the plant optimization control link, these optimization programs may also interface with the unit governors to optimize the unit efficiency of adjustable-blade turbines. This optimization is often done by computing a “cam offset” value and by transmitting this offset value to the unit governor’s blade control algorithm. The blade control optimization program may compute this cam offset as a function of the unit output, the unit flow, and the unit operating head. The blade control cam offset can be optimized using either the unit generation or the calculated unit efficiency as the parameter to maximize. This optimization strategy is sometimes called “automatic index testing.”

When implementing an optimization system, the amount of additional control activity that results from the optimization process should be considered. If the optimization system continuously searches for an optimum operating point, this may result in excessive movement of the unit governors. The result of this excessive movement can be both increased wear of the turbine control actuators and the turbine control devices and a reduced overall operating efficiency because of the increased percentage of time the unit or station is operated off its peak efficiency. Typically, this situation is avoided by “locking in” the computed optimization parameters for an operating point so that operating point is not reoptimized again until the optimization system is commanded to do so. An optimization system that uses stored efficiency curves to schedule the operating levels of the units in a station does not typically exhibit the symptoms of excessive searching activity.

5. Elements of the turbine governing system

As with any conventional control system, a turbine governing system comprises three basic components: the setpoint controller, the actuator or actuators, and the controlled process.

5.1 Setpoint controller

The setpoint controller portion of the turbine governing system determines the basic operating function of the hydroelectric generating unit. The operating parameters and controller strategy used in the setpoint controller determine what conditions the turbine governing system responds to, and what action is taken by the turbine governing system.

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Hydroelectric generating stations are sited for diverse reasons. Energy generation could be either the primary justification for the project or it could be a mere byproduct of the project that was justified for other reasons. For example, hydrogeneration may be an ancillary component in an irrigation, flood, or reservoir-level control project. In these instances, water flow or reservoir levels are the primary controlled variables. Whenever hydrogeneration capability is added to a project, control takes on new dimensions.

To be useful, the output of a hydroelectric generating station should have defined performance requirements for power, voltage, and frequency. The hydrogenerator, with its own specific unit control system, should be in harmony with the load or the grid to be supplied, and the total system should be stable to assure a reliable supply of electrical energy.

5.1.1 Error computation and processing

A typical governor's setpoint controller subtracts the process output (speed) from the setpoint input (speed reference). The error processing elements of the governor perform operations on the input–output difference (the error) to arrive at a correcting command for the actuators. The most common controlled process outputs for a hydroelectric turbine governing system are unit speed and generated power.

5.1.1.1 PID control

A common computing algorithm is the PID. Speed or power errors, or both, are inputs to a typical PID governor on a hydroelectric unit. The PID gains should be independently adjustable to achieve the desired dynamic performance. There are several possible structural implementations of the basic PID control algorithm that have been used in hydroelectric turbine governing systems.

5.1.1.2 Gain scheduling

The characteristics of a hydro turbine can vary significantly over its operating range. This means that if the turbine governing system is tuned for optimum performance at one operating point, its performance may be different at a different operating point with those same gain settings. With a single set of gains for the turbine governing system, the system is typically tuned at its "worst case" operating point, and the resulting performance at the other operating points prevails. One method to approach optimum performance at all operating points is to use a schedule of gains based on some operating parameter such as generation level. With this method, the unit is tuned for the best achievable performance at several different operating levels and the gains are stored in a data map that is used to set the controller gains as a function of the operating point. Gain settings between the stored data points can be interpolated to yield smooth transitions between operating levels.

5.1.1.3 Adaptive control

Adaptive control is a technique used to automatically adjust the dynamic characteristics of the control system to achieve optimum performance at any operating point. This technique also compensates for changes in unit characteristics and other system parameters. Although adaptive control can theoretically provide optimum performance under all conditions, the additional actuator movement inherent to many adaptive control approaches can have a negative impact on the longevity of many unit components. The term "adaptive control" is broad in scope. Many control techniques used in conjunction with feedback control systems, such as feedforward, gain scheduling, and 3D blade control, also fit the general definition of adaptive control.

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5.1.1.4 State space control

State space control is a control technique that uses more information than a single computed error for making corrective action to the control system actuator. This may include the processing of information from several independent parameters that affect the operation of the unit. Often, this information is input to a mathematical model of the controlled unit and the best control action is predicted from this model. The benefits of state space control include improved responsiveness and stability, along with improved robustness over a wide range of changing system parameters. Some elements of state space control techniques are in common usage in the hydroelectric power industry. However, turbine governing systems based primarily on state space control techniques are not common, as practical forms of this control strategy are still in their early stages of development.

5.1.1.5 Other control approaches

Several other control approaches to error processing have been used during the evolution of hydroelectric unit control. These include temporary droop control (approximately equivalent to a PI controller), gain-plus-reset (approximately equivalent to a PI controller), "double-derivative" implementation of the PID controller, and deadband control. The specific characteristics of these variations of control approaches are not addressed this guide.

5.1.2 Feedback

The process of continuously monitoring the output of a process to maintain it at a desired level is known as feedback control. The desired operating level, whether power, gate position, pond level, or frequency, is called the setpoint. An elementary feedback control system is shown in Figure 14.

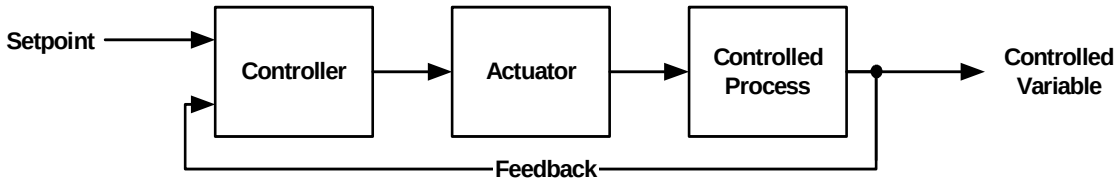


Figure 14—Elementary feedback control system

5.1.2.1 Multiple feedbacks

More than one feedback from a controlled process can be used in a feedback control system. Generally, one feedback comes from the process output and one feedback comes from an intermediate point in the controlled process. Using this control approach establishes a relationship (generally linear) between the two feedbacks and improves the stability of the control loop by reducing the phase lag in the control loop. This approach, however, also reduces the accuracy of the control system because the intermediate feedback allows the setpoint controller to be satisfied when a difference exists between the setpoint and the process output.

5.1.2.1.1 Permanent speed droop governing system

A "permanent speed droop" governing system is a common type of control that uses multiple feedbacks. Permanent speed droop determines the amount of change in the position of the turbine control servomotor a unit provides in response to a change in unit speed. Permanent speed droop is discussed in 4.5. Figure 2 is a block diagram representation of a typical governing system using permanent speed droop.

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5.1.2.1.2 Speed regulation governing system

A “speed regulation” (also known as “power droop”) governing system is similar to the permanent speed droop type of control system, but with generated power being used as the intermediate feedback from the controlled process instead of actuator position. Speed regulation determines the amount of power generation a unit provides in response to a change in unit speed. Speed regulation is discussed in 4.5.2. Figure 7 is a block diagram representation of a typical governing system using speed regulation.

5.1.3 Feedforward

The process of setting a controlled variable to a target level computed to achieve the desired level is known as feedforward control. Figure 15 is a simplified block diagram of a feedforward control system. Feedforward control differs from feedback control in that it does not actually measure the output of the controlled process. Instead, it computes the necessary level of the controlled variable or variables based on the setpoint and the known characteristics of the controlled process. Noncontrolled inputs, which affect the controlled process, other than the setpoint may also be used in this computation. The advantage of feedforward control is that the actuator and process dynamics are not included in the control action, allowing the control to be very responsive without risking instability of the control loop. The disadvantage of feedforward is that it cannot compensate for the inevitable deviations of an actual process from its model. Most real-world feedforward control systems include some supplemental feedback control to enhance the steady-state accuracy of the control. Refer to Figure 11 for a simplified block diagram of a governor controller using both feedback from the unit speed and feedforward from the setpoint to the controller output. This system allows the gains in the feedback controller to be set for stable operation while providing a very fast response to setpoint changes.

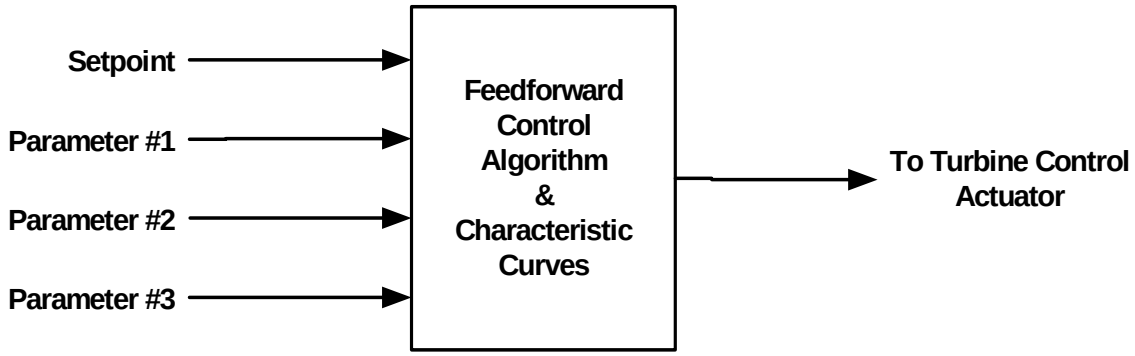


Figure 15—Feedforward control system

5.2 Actuator

An actuator is the connection between the setpoint controller and the controlled process. The output or the actuator should be able to effect a change in the controlled variable of the process. Most controlled variables related to the control of a hydroelectric generating unit require the mechanical positioning of the controlled variable, such as the wicket gates or turbine blades. Other processes may require an electrical current as the actuator output, such as the generator field or the demand of a shunt load bank.

The actuator system compares the desired turbine actuator position command or setpoint with the actual actuator position, and it provides the necessary work to hold the actuator output at the desired value. Several types of governor actuators are commonly used, but they all perform the same basic function.

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For dual-regulated turbines with adjustable blades, such as Kaplan or bulb turbines, separate actuator servomotors are generally used to independently control the wicket gates and the runner blades. On a double-regulated turbine such as an impulse turbine, the number of turbine control actuators used varies with the number of needles controlling the water flow to the wheel. It is common for each needle to be driven by a dedicated servomotor. It is also common for the needle valves of an impulse turbine to be designed with a loading spring built into the control servomotor to provide fail-safe operation to close the needles upon control system failure or loss of hydraulic control pressure. A separate actuator servomotor generally controls the deflector mechanism, which acts to deflect the water stream away from the turbine runner to reduce the developed torque of the turbine more rapidly than can be done with the slower moving needle servomotors.

The work capacity of the actuator used to control a turbine's actuating device should be greater than the maximum work necessary to move the device throughout its operating range. The maximum work capacity required to move the turbine actuating device is affected by the following items:

a) Type of turbine
b) Head across the turbine
c) Mechanical design of turbine actuator linkages
d) Maintenance history (e.g., lubrication) of turbine actuator linkages
e) Hydraulic design of the turbine actuator
f) Hydraulic design of the turbine distributor passageway
g) Rate of travel of the actuator

For new installations, the turbine manufacturer generally determines the maximum work requirement to move the turbine actuating device, and it often supplies the turbine control servomotor.

The required power to provide fast movement of turbine actuating devices usually requires that the actuator have access to an adequate supply of stored energy to perform the necessary work. All hydraulic turbine actuators should be of a reliable, fail-safe design. This means that a reserve supply of energy should be continuously available to be able to move the actuator to its fail-safe position, even when the station electrical service is lost. A hydraulic pressure supply accumulator tank generally provides this stored energy supply. A detailed discussion of the design considerations for the governor hydraulic pressure supply system is addressed in 6.3.17.

5.2.1 Mechanical actuators

A mechanical actuator is a device that produces a mechanical position output using mechanical power either from the turbine shaft or from a human operator as its source of energy. This type of actuator was common in the early days of turbine control, but the advent of hydraulic control in the early part of the twentieth century has relegated mechanical actuators, for the most part, to the role of a manual backup system for small units.

5.2.2 Mechanical hydraulic actuators

The mechanical hydraulic actuator is characterized by a mechanical setpoint position input that is amplified by a hydraulic amplifier driving the output servomotor. Figure 16 is a block diagram of a typical mechanical hydraulic actuator. The gate servomotor sets the position of the turbine gate linkage. The flow of hydraulic oil to the gate servomotor is controlled by the distributing valve and its servomotor rate limiters. The pilot valve and distributing valve provide the necessary amplification of the position error (difference) between the setpoint input and the servomotor position output.

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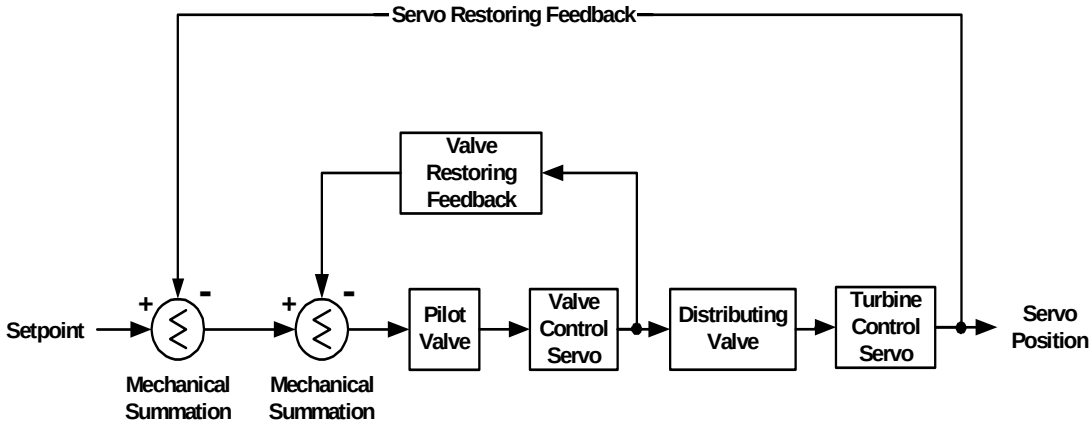


Figure 16—Typical mechanical hydraulic actuator

Figure 16 does not show the hydraulic pumping unit and pressure tank needed to supply pressurized oil to the pilot and distributing valves. The hydraulic oil pressure supply systems used in modern turbine governing systems often have a nominal pressure of 7 MPa or greater. Earlier mechanical hydraulic governor actuators used pressures as low as 1 MPa.

5.2.3 Electromechanical actuators

An electromechanical actuator uses mechanical power developed by an electric motor as its source of energy. This type of actuator typically uses a ballscrew or other gear reduction to transfer the mechanical power of the motor to actuate the controlled variable. Figure 17 is a block diagram of an electromechanical actuator. The fail-safe requirements of the control application when using an electromechanical actuator are important because this type of actuator generally requires electrical power to move in either direction. Therefore, the actuator may not be able to move to a fail-safe position after loss of station electrical service. In cases where an electromechanical actuator is used to control the wicket gates (or other flow-controlling devices) of a hydroelectric turbine, a backup means of shutting down the unit may be necessary to protect it from a sustained runaway condition resulting from loss of the station electrical service, or the failure of a key component of the actuator system.

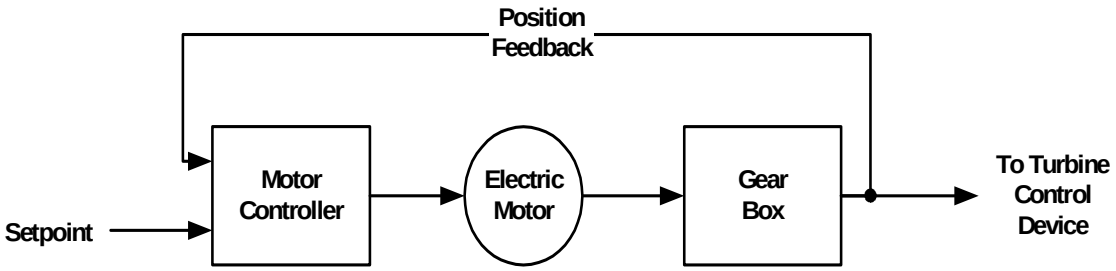


Figure 17—Electromechanical actuator

5.2.4 Electrohydraulic actuators

An electrohydraulic actuator uses an electronic setpoint signal as its input, and it amplifies this setpoint with an electrohydraulic amplifier to determine the position of the output hydraulic servomotor. Figure 18 is a simplified block diagram of a typical electrohydraulic actuator. An electrohydraulic actuator system is typically employed on large turbines where high oil flow rates are required for the gate servomotor. On smaller systems, a two-stage electrohydraulic valve can often provide acceptable control of the oil flow rate,

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thus eliminating the need for a separate pilot stage to operate the main valve. Although Figure 18 does not show any hydraulic shutdown valves, such valves are often used to override the main valve output for the purpose of protective shutdown of the unit.

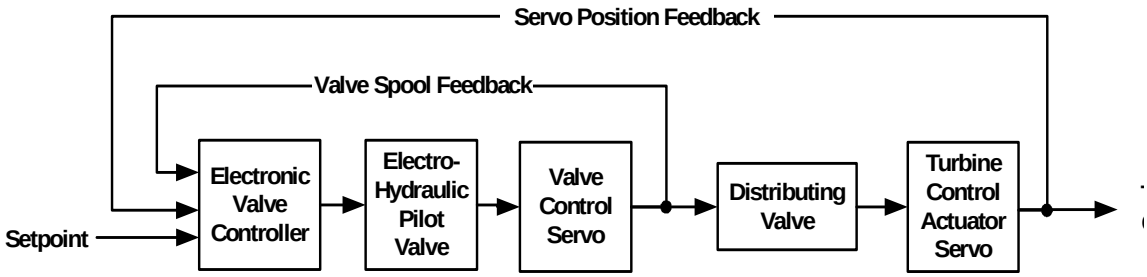


Figure 18—Typical electrohydraulic actuator

5.2.5 Hydraulic control valve design

For both mechanical-hydraulic and electrohydraulic actuator systems, two design details of the hydraulic valving have a significant impact on the performance of the actuator. These design details are valve lap and valve dither.

5.2.5.1 Valve lap

Valve lap, also called “valve overlap” or “positive lap,” is the amount that the valve control plunger overlaps the valve control ports. Valve lap contributes to both the deadband and the deadtime of the actuator system, because the valve control plunger needs to travel across the valve overlap area to begin passing oil to the actuator servomotor. A zero-lap control valve provides the best actuator performance in terms of both deadband and deadtime. However, manufacturing a zero-lap valve increases its cost, because the valve plunger generally needs to be custom-ground to match the spacing of the valve bushing ports. Another disadvantage of a zero-lap control valve is that the oil consumption under steady-state conditions is greater than with a positive-lap valve.

5.2.5.2 Valve dither

One technique that has historically been used to offset the negative effects of valve lap is valve dither. Valve dither is implemented by injecting a small oscillatory action into the pilot stage of a servomotor control valve. The amplitude of this dither is generally adjusted so the main stage of the servomotor control valve continuously travels across the limits of the valve lap. The frequency of the valve dither is generally adjusted to a value high enough so no perceptible dithering action is transmitted to the actuator output servomotor. Valve dither, when properly adjusted, effectively eliminates the contribution of the valve lap to the deadband of the actuator system. The dithering action also reduces the contribution of the valve lap to the deadtime of the actuator system. An additional benefit of valve dither is the reduction of friction-induced inaccuracy in the actuator system by keeping the valve components moving with respect to each other. This dithering action keeps an oil film between the moving parts of the control valve system, thus eliminating the effects of static friction on the performance of the actuator control valve system.

5.2.6 Electronic load actuators

A load “actuator” is designed to control the electrical load on the generator so the generator rotor is maintained at the desired speed. Figure 19 is a block diagram of an electronic load controller. Typically this type of actuator is only used on small hydroelectric units. The load bank is used to impose an electrical load on the generator, which is reflected as a mechanical load on the turbine. This mechanical loading is used to control the unit’s speed. The load bank may be either air-cooled or water-cooled.

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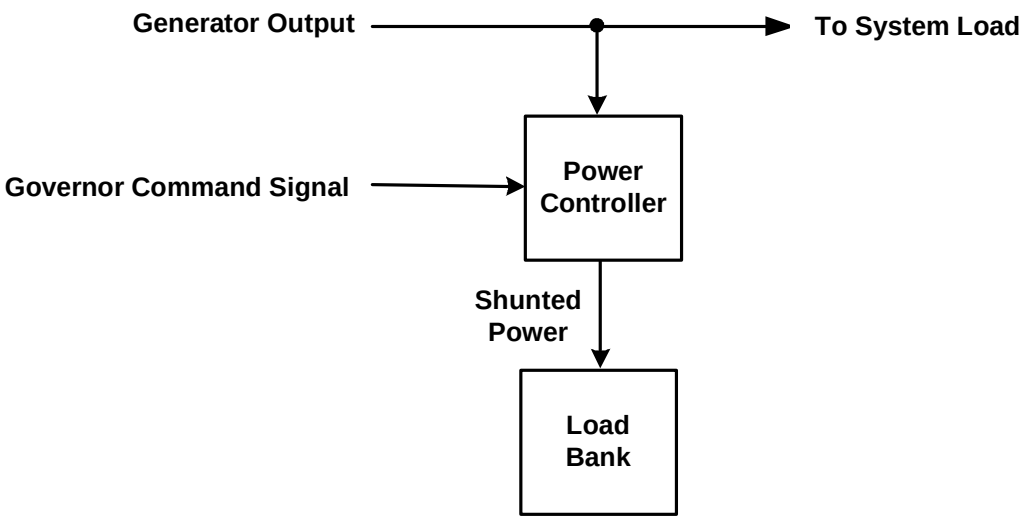


Figure 19—Electronic load actuator

5.3 Controlled process

The hydroelectric generating plant consists of four basic elements that are necessary to generate power from water: a means of creating head, a conduit to convey water, a hydraulic turbine, and an electric generator. The dam creates the operating head necessary to move the turbines, establishes the amount of water storage available for power production, and impounds the water supply necessary for the daily or seasonal stream flow release pattern. The turbine governing system controls the operation of the hydraulic turbine to achieve the desired operating results. The water levels, water flow rate, turbine speed, generator output frequency, and generator output power are parameters that are affected by the operation of the turbine governing system. The design of the turbine governing system determines which of these parameters has the highest priority in the operation of the hydroelectric turbine.

5.3.1 Turbine

The turbine converts the potential energy of water into mechanical energy, which in turn drives the generator. Water under pressure enters the turbine through the wicket gates and is discharged through the draft tube after its energy is extracted. The amount of power the turbine is able to produce depends on the head on the turbine, the rate of flow of water passing through the unit, and the efficiency of the turbine. Modern turbines can develop power from almost any combination of head and flow. The specific type of turbine selected for an application depends on many factors, but primarily on the operating head and available flow. Although there are many types of turbines, they fit into two basic categories: impulse turbines and reaction turbines. Annex B describes the impact of several turbine designs on the turbine governing system.

5.3.2 Generator

The generator converts the mechanical power produced by the turbine into electrical power. The two major components of the generator are the rotor and stator. The rotor is the rotating assembly, which is attached by a connecting shaft to the turbine. The stator is the stationary portion of the generator, which transmits the generated power to its destination. Generator interaction with the power system can be complex, and it can result in lightly damped response poles that can be stimulated by fast governor action, producing undesirable power oscillations.

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5.3.2.1 Generator inertia

The rotating inertia of the generator limits the rate of change of unit speed in response to an imbalance between the developed power of the turbine and the load power imposed on the generator. In general, increasing the generator inertia improves the capability of the turbine governing system to regulate the unit speed within acceptable limits.

5.3.2.2 Generator speed increaser

A generator speed increaser is sometimes used to increase the rotating speed of the generator above the rotating speed of the turbine. This allows a smaller, and less expensive, generator to be used. A speed increaser is typically used only on small turbine-generator units. The effective rotating inertia of this smaller generator, from the perspective of the turbine, is increased by the ratio of the speed increaser. This apparent multiplication of the generator's rotating inertia by the speed increaser partially compensates for the smaller rotating inertia of the higher speed generator.

5.3.3 Water passage

Water is the medium through which energy is delivered to a hydroelectric turbine for the purpose of generating electric power. Often, there are constraints placed on water levels, flows, or pressures that require certain elements of the water-handling system to be included within the boundaries of the "controlled process" for the turbine governing system. Brief descriptions of the main elements of the water system for a hydroelectric generating unit follow:

5.3.3.1 Head pond

The head pond is the water impoundment used as an energy source for the hydroelectric unit. Some stations may have a very large storage capacity, and others may have essentially no storage capacity. The size of the head pond storage affects the rate at which the hydroelectric generating unit can affect the head pond water level. The head pond may also be called the reservoir, forebay, headrace, headwater, or the pool elevation.

5.3.3.2 Water column

The water column comprises all of the structures used to convey water from the head pond to the turbine. The water column may include an intake structure, a penstock, one or more surge tanks, and a spiral case. The composite water column inertias and elasticity of these structures contribute to the water hammer effect that impacts the performance of the turbine governing system.

5.3.3.3 Draft tube

The draft tube conveys the water from the discharge side of the turbine to the tailrace. It is normally a part of the powerhouse structure, and it is designed to minimize exit losses. The inertia of the water in the draft tube also contributes to the total water inertia that impacts the performance of the turbine governing system. In some applications, this effect is significant.

5.3.3.4 Tail pond

The tail pond can be an open stream, the reservoir of a downstream project, a canal, or a tunnel exiting from an underground powerhouse. The level of the tail pond exerts back-pressure on the turbine. This affects the power output of the turbine. The size of the tail pond affects the rate at which the hydroelectric generating unit can affect the water level. Impulse turbines usually rotate in open air, so tail pond level does not affect the output of the unit generator. The tail pond may also be called the tailrace or lower reservoir.

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5.4 Shutdown control

Historically, protective shutdown of the unit has commonly been accomplished by using the primary turbine governing control actuator. Typically, a shutdown valve, either solenoid operated or mechanically operated, is used to override the turbine control actuator's control valve and force the primary turbine control servomotor to close at its maximum rate. Shutdown valving such as this is typically designed to be fail-safe, shutting down the unit if either the control power or any critical component of the shutdown system fails. In some cases, protective shutdown of the unit may be accomplished by a separate device, such as a fast-acting turbine shutoff valve in the water passage to the turbine. Sometimes, the primary turbine control actuator is biased toward the fail-safe direction with a counterweight, thus using gravity as the fail-safe actuating force.

5.5 System examples

Annex C contains some simplified block diagrams describing the scope of some common types of turbine governing systems related to the control of hydroelectric generating units.

5.6 System modifications

In some cases, a system may not perform as desired once it is installed. Some common reasons for the unsatisfactory performance include control specifications that are not consistent with the expected performance, unforeseen constraints on the system performance, or changes in system requirements that occurred after the specification of the system. Annex D contains some examples of systems that needed to be modified to achieve the desired governing system performance when the original design did not meet the application requirements.

6. Equipment specifications

The specifications for the turbine governing system should address the performance needs of the installation. Information from several sources is required for producing the equipment specification. Care should be taken that the turbine governing system provides all of the necessary functionality without unnecessarily duplicating functions provided by other systems at the hydroelectric turbine installation (see IEC 1116 [B22], IEEE Std 1010™-1997 [B24], IEEE Std 1020™-1993 [B25], IEEE Std 1147™-1991 [B26], and IEEE Std 1249™-1996 [B27].

6.1 Cooperation of manufacturers

For the governing system to provide the desired performance, the designs of the plant's civil structures, the turbine, the generator, and the governing system should be coordinated. Designs of one or more of these elements of a hydroelectric generating unit may place constraints on the governing system, preventing it from achieving the desired level of performance. The earlier the impact of these elements on governor system performance is addressed, the better the chance for eliminating possible constraints on governor system performance. Particular attention should be given to issues such as servomotor loading, servomotor timing, servomotor stroke, mechanical inertia time, water inertia time, and expected modes of operation.

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6.2 Governor equipment

Some general considerations for specification of the turbine governing system equipment are discussed in 6.2.1 – 6.2.3.

6.2.1 Governor construction

There are many possible basic arrangements for the construction of the governor actuator. The governor controller equipment may either be assembled as part of the governor actuator system, or it may be located separately. Three basic types of construction are commonly used: the integrated actuator, the remote servomotor actuator, and the cabinet actuator.

6.2.1.1 Integrated actuator

The integrated actuator includes the turbine control servomotor, the distributing valve, and the associated feedback and control components of the governor actuating system in a single assembly. A mechanical hydraulic governor using this type of actuator to operate the wicket gates via a rotating gate shaft is usually called a “gateshaft governor.”

6.2.1.2 Remote servomotor actuator

The remote servomotor actuator includes the distributing valve and associated control components of the governor actuator system in an assembly located remotely from the turbine control servomotors. The remote servomotor actuator may be mounted on a hydraulic pumping unit, or it may be mounted on a separate base.

6.2.1.3 Cabinet actuator

A cabinet actuator refers to a remote servomotor actuator type of governor control system that is enclosed by a cabinet. The cabinet panels are often used for mounting indication and control devices related to the operation of the governing system. A cabinet actuator typically includes the governor controller equipment along with the hydraulic control valve components of the turbine control actuator equipment.

6.2.2 Special considerations affecting electrohydraulic governors

The controller portion of an electrohydraulic governor requires electrical power to function. Consequently, the overall reliability of the turbine governing system is affected by the reliability of the electrical power source. Usually, the station battery system is considered the most reliable power source in the powerhouse. Station ac power may also be used to power the electrohydraulic governor, but this power source is often backed up by the station battery system, thus providing an uninterrupted power source for the turbine governing system.

6.2.2.1 Power supplies

The power supplies used to provide regulated dc power for the electronic circuitry in the electrohydraulic governing system should be very reliable to avoid compromising the reliability of operation of the governing system. The size and importance of the hydroelectric generating unit generally determine the degree of reliability required.

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6.2.2.2 Transient immunity

Transient immunity describes the effects of electromagnetic interference on the governing equipment. The unit should operate reliably under the conditions to which it is exposed during normal operation in the powerhouse. The degree of electromagnetic interference present is determined by the nature of the other equipment located near the governing equipment. Devices such as motor starters, handheld radios, switchgear, and other electrical equipment all contribute to the level of electromagnetic interference that the governing equipment sees. Depending on the expected level of electromagnetic interference, the specifications for the turbine governing system equipment should require appropriate testing to ensure reliable operation in this environment. Some typical standards for testing transient immunity are included in the following:

a) IEEE Std C37.90.1™-2002 [B28] “SWC test”
b) IEC 60255-22-1 [B16] class III and IEC 60255-22-4 [B17] class IV “Electrical disturbance test”
c) IEC 60255-5 [B15] “Insulation coordination test”
d) IEEE Std C37.90.2™-1995 [B29] “Radiated EMI test”
e) IEC 61000-4-2 [B18] “Electrostatic discharge test”
f) IEC 61000-4-3 [B19] “Electromagnetic field test”
g) IEC 61000-4-4 [B20] “Burst immunity test”
h) IEC 61000-4-5 [B21] “Surge immunity test”

6.2.2.3 Electronic equipment

Electronic equipment should be designed to operate reliably during the extreme ambient conditions expected in the powerhouse. Typical operating ranges are 0 °C to 70 °C for components and 0 °C to 50 °C for system assemblies. Specified humidity ranges are dependent on the climate at the site. Heaters within the electronic cabinets are often used to reduce the relative humidity within the cabinets when they are installed in very humid climates.

6.2.2.4 Test facilities

The electronic controls should be designed to allow easy troubleshooting of the system. Appropriate test points should be provided to allow easy measurement of control signals throughout the system. For digital control systems, provisions should be made to allow access of the output of each functional block within the control system.

6.2.2.5 Governor balance meter

A governor balance meter may be provided to indicate the amount of error between the governor command signal to the turbine control servomotor and the actual servomotor position. A separate indicating meter may provide this function, or it may be provided as a computed value displayed on the governor operating display panel.

6.2.2.6 Actuator lock

In some instances, it may be desirable to keep the turbine control servomotors in their last operating position after certain types of failures. This allows the unit to continue generating power after the occurrence of the specified failure conditions. Some typical conditions that may activate an actuator lock condition are failure of a speed signal or failure of the power supply to the governor controller. It is important to evaluate the implications of operating under an actuator lock condition, considering unit safety and the impact of other failures occurring when in this operating mode. It is important to assure that the unit can safely shut down if a critical failure occurs while operating in actuator lock.

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The actuator lock may be implemented mechanically, in a separately powered electronic controller, or within the main governor controller. The method of implementing the actuator lock mode should be determined by the design criteria for this mode of operation.

6.2.2.7 Digital governors

When a digital controller is used in a turbine governing system, the system is usually referred to as a "digital governor." Digital control hardware running an application program accomplishes the required control functions with this system. The application program should be stored in a secure, nonvolatile memory. Any configuration or tuning parameters should also be stored in a nonvolatile memory so reliable and repeatable operation of the system is not compromised. The controller hardware should routinely check the integrity of the application program, and it should either issue an alarm or it should revert to using a secure backup copy of the program if a program error is detected.

6.2.2.7.1 System reliability considerations

Digital controllers used for turbine governing systems are very flexible and can be used for functions not directly related to the turbine governing control function. Consideration should be given to the impact on system reliability of all functions accomplished by the turbine governing system controller. The level of reliability required for any particular turbine governing system is influenced by the following factors, which should be considered when designing the turbine governing system:

a) Importance of the unit to the security of the interconnected power system

b) Importance of the unit to the owner's revenue source

c) Impact of unscheduled downtime of the unit

The impact on system reliability of all functions performed by the digital controller should be considered in the design process. The worst-case processor burden of all functions, both governing and nongoverning, should be evaluated for their impact on the control response and reliability.

The reliability of a digital control system may be improved by designing redundancy into the system. Redundant processors or redundant critical I/O devices may be used to improve the overall system reliability. Downtime may also be reduced by designing self-diagnostic and testing features into the digital controller used in the turbine governing system.

The actuator portion of the turbine governing system is usually also used by the protective shutdown process for the turbine. Therefore, the reliability of the turbine control actuator is of prime importance to the security and safety of the unit.

6.2.2.8 Servomotor restoring feedback

Electrohydraulic governing systems may use either mechanical or electronic restoring feedback from the turbine control servomotors. The restoring feedback is used in conjunction with the servomotor control valves to determine the position of the servomotors. The servomotor restoring feedback system should be of a highly reliable design, and it should be fail-safe in the event of either electrical or mechanical failure of the restoring system. The fail-safe condition desired should be defined in the specifications for the turbine governing system. Typically, the fail-safe condition for wicket gates or needles is the closed position. The fail-safe condition for runner blades is usually in the steep position (i.e., parallel to the water flow), but sometimes the fail-safe condition for runner blades is specified to be in the flat position (i.e., perpendicular to the water flow).

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6.2.3 Governor actuator rating

Functionally, the turbine governing system's control actuator includes the turbine control servomotor, distributing valve, and other related components. However, the turbine control servomotor and the hydraulic control components are often supplied by different manufacturers. The designs of these components should be coordinated so the turbine governing system can reliably meet the desired performance specifications.

6.2.3.1 Servomotor size

The servomotor dimensions should be such that the servomotor can provide adequate force to operate its associated turbine control device. The operating hydraulic supply pressure times the servomotor control area times the number of turbine control servomotors determines the net available force to the turbine control device. Experience has shown that the turbine control servomotor should be designed so the maximum loading force from the turbine control device can be provided with an operating hydraulic pressure of 60% to 70% of the nominal system pressure. This design criterion typically provides an adequate margin of safety, and it provides sufficient pressure under all conditions to safely shut down the unit.

6.2.3.2 Distributing valve flow

The distributing valve should be capable of providing sufficient flow to the turbine control servomotors so they can meet the servomotor timing specifications. For the distributing valve to provide the desired flow, it should have sufficient valve porting area and pressure drop across the valve. Experience has shown that a distributing valve should have a net pressure drop of at least 25% to 35% (depending on the type of valve) of the operating hydraulic supply pressure across the valve at maximum flow. If the pressure drop across the valve at maximum flow is less than this criterion, the timing of the turbine control servomotor is more dependent on its mechanical loading and the viscosity of the oil. This pressure drop criterion is applicable with the servomotor traveling at its maximum rate with maximum servomotor loading, at the minimum normal hydraulic operating pressure.

6.2.3.3 Pressure drop in piping

The pressure drops in the piping or tubing from the hydraulic pressure supply accumulator tank to the distributing valve, from the distributing valve to the turbine control servomotor, from the turbine control servomotor back to the distributing valve, and from the distributing valve back to the oil sump tank should be considered in the design of both the distributing valve and the turbine control servomotor piping. To avoid excessive pressure drops in the piping, hydraulic control systems have historically been specified to have a maximum oil velocity of 5 m/s within the piping. This guideline produces acceptable designs in most cases. However, in certain instances of long piping runs or a relatively low nominal hydraulic supply pressure, the pressure drop in the oil piping can become a very significant consideration in the design of the turbine control actuator system. An increase in the servomotor piping size, an increase in the distributing valve porting area, or both may be necessary in some instances to meet the servomotor timing specification. Experience has shown that the pressure drop in the piping or tubing to the turbine control servomotor should not exceed approximately 5% to 8% of the available hydraulic system pressure. If the pressure drop in the hydraulic piping becomes too high, the servomotor timing is influenced to a greater degree both by the servomotor loading and by the oil viscosity.

6.3 Components or auxiliary devices

There are many auxiliary devices and functions that are often present as part of a turbine governing system. Some functions may be accomplished either by a separate device or by a function programmed into a digital controller.

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6.3.1 Speed changer

The speed changer is also known as the “speed adjustment” or the “speed reference”. The speed changer function determines the reference speed for the turbine governing system. For mechanical-hydraulic governors, the speed changer is typically either a motor-driven or a manually set pivot point in the speed sensing portion of the governor. For electrohydraulic governors, the speed changer may be either a motor-driven or a manually set potentiometer assembly. For digital governors, the speed changer is often an adjustable reference value that is embedded in the governor controller’s application program, or it may be established from an external signal input to the governor controller.

6.3.2 Servomotor limit

The servomotor limit determines the maximum (or minimum) position that the turbine control servomotor may achieve during normal operation of the turbine governing system. When applied to wicket gates, this servomotor limit is often called the “gate limit.” The servomotor limit may be accomplished either mechanically or electronically. Mechanical servomotor limits are typically used only when the servomotor restoring feedback is accomplished mechanically. When electronic servomotor restoring feedback is used, the servomotor limit function is typically done electronically within the governor controller. The servomotor limit position may be established as an adjustable reference value that is embedded in the governor controller’s application program, or it may be established from an external signal input to the governor controller.

6.3.3 Permanent speed droop changer

The permanent speed droop changer function determines the permanent speed droop characteristic for the turbine governing system. For mechanical-hydraulic governors, the permanent speed droop changer is typically either a motor-driven or a manually set pivot point in the speed sensing portion of the governor. For electrohydraulic governors, the permanent speed droop changer may be either a motor-driven or a manually set potentiometer assembly. For digital governors, the permanent speed droop changer is often an adjustable reference value that is embedded in the governor controller’s application program, or it may be established from an external signal input to the governor controller.

6.3.4 Speed regulation changer

The speed regulation changer function determines the permanent speed regulation characteristic for the turbine governing system. Mechanical-hydraulic governors do not typically offer speed regulation control characteristics. For electrohydraulic governors, the speed regulation changer may be a motor-driven or a manually set potentiometer assembly. For digital governors, the speed regulation changer is often an adjustable reference value that is embedded in the power controller’s application program, or it may be established from an external signal input to the power controller.

6.3.5 Servomotor velocity adjustment

Generally, some method of adjusting the velocity of turbine control servomotors is provided as part of the actuator portion of the turbine governing system.

6.3.5.1 Primary turbine control servomotors

Limiting the velocity of the turbine control servomotor is important to the safe operation of the hydroelectric unit. The wicket gates or needles control the flow of water through the turbine. Controlling the rate of travel of these servomotors constrains the water hammer effect to safe levels of pressure transients in the water passage, thus protecting these water passages from damage.

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6.3.5.2 Secondary turbine control servomotors

Secondary turbine control servomotors such as blades (on adjustable-blade turbines) and deflectors (on impulse turbines) may act more quickly to reduce the overspeed of the turbine as a result of a load rejection. Impulse turbine deflectors have negligible influence on the water hammer effect, and adjustable turbine blades generally have much less influence on the water hammer effect than do the wicket gates.

6.3.5.3 Velocity adjustment methods

The servomotor velocity limiter function can be accomplished by limiting the travel of the distributing valve spool (using "stop nuts" or the equivalent), by machining flow-limiting orifices within the distributing valve, by placing orifice plates in the servomotor control lines, or by installing flow control valves in the servomotor control lines. Devices placed in the servomotor control lines generally have some restrictive effect on servomotor response at servomotor velocities slower than the maximum velocity setting. Devices that limit the travel of the distributing valve spool do not affect performance until the spool hits its limit of travel. Some flow control valves have the ability to compensate for changes in oil velocity, resulting in a more consistent maximum velocity setting as oil temperature changes. Thin, sharp-edged orifices are fairly independent of oil viscosity and have proven to perform satisfactorily in many applications. Limiting the travel of the distributing valve spool is, essentially, limiting the maximum control orifice size within the distributing valve. Regardless of the method used for limiting servomotor velocity, the method should be reliable and reasonably independent of viscosity changes due to normal oil temperature changes.

In some applications, a distributing valve lacking sufficient valve porting area may be used. To achieve the desired maximum servomotor rate in these cases, an additional valve may be installed in parallel with the distributing valve to achieve the desired flow rate. This additional valve may be actuated only on shutdown of the unit, or it may be actuated during governing control of the unit in response to large transients to achieve the desired maximum servomotor rate. If the additional valve is used only for shutdown purposes, the impact of the slower governing servomotor rate should be considered for its impact on the unit's ability to contribute to the stability of the interconnected system.

6.3.6 Restoring connection

The method of connecting the servomotor feedback restoring system to the turbine control servomotor affects both the reliability of the system and the linearity of the positioning system. If the connection of the restoring system fails, the system should be designed for fail-safe operation of the servomotor positioning system. If electronic transducers are used for the servomotor feedback restoring system, the type of transducer and the method of connection determines the accuracy and linearity of the restoring signal. For example, connecting a linear servomotor to a rotary transducer generally imparts some degree of nonlinearity into the feedback signal. The impact of this nonlinearity on the system operation should be considered during the initial design phase, so satisfactory operation of the turbine governing system is achieved.

6.3.7 Speed sensor source

The rotating speed of the turbine is detected by the turbine governing system in order for the system to respond to changes in speed. Several approaches to detecting turbine speed are discussed in 6.3.7.1 – 6.3.7.2.

6.3.7.1 Mechanical speed sensing

A mechanical-hydraulic governor typically uses rotating weights that translate the rotating speed of these weights into a mechanical displacement that is connected to a hydraulic amplifier in the governor. This rotating weight assembly is called a "ballhead" assembly. The governor ballhead assembly is rotated at a speed directly proportional to the rotating speed of the turbine by one of several methods.

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6.3.7.1.1 Direct connection

A ballhead assembly may be directly connected to the turbine or generator shaft. This method requires that either the ballhead displacement or the pilot hydraulic output be connected to the main governor assembly. The distance involved in making this connection degrades both the accuracy and the responsiveness of this type of arrangement.

6.3.7.1.2 Mechanically coupled connection

A ballhead may be mechanically coupled to the turbine or generator shaft. This is typically done with gear-driven shafts or with belt-driven pulleys. This approach allows the governor ballhead assembly to be located within the main governor assembly. However, the inherent roughness of gear-driven shafts or pulleys driven by a long belt imparts some undesirable modulation of the speed signal. The measures necessary to minimize unwanted motion of the turbine control servomotors due to this speed signal modulation limit the overall responsiveness of the governing system.

6.3.7.1.3 Motor-driven ballhead

The ballhead assembly may be driven by an electric motor if the speed of the motor is directly proportional to the unit speed. In this way, the connection between the turbine and the governor is electrical and, therefore, independent of the distance between the turbine and the governor. This may be accomplished by driving the ballhead motor directly from the generator voltage transformers, which ensures that the motor speed is directly proportional to the unit speed. However, this approach requires additional capacity from the voltage transformers, and the ballhead assembly generally cannot rotate until the generator excitation has been applied (typically around 90% of rated speed). A permanent magnet generator (PMG) that is mechanically coupled to the turbine or the generator shaft may also be used to drive the ballhead motor. Using this approach causes the ballhead assembly to begin rotating at a very low speed. However, the mechanical connection between the PMG and the turbine or generator shaft should be designed to minimize unwanted modulation of the sensed speed.

6.3.7.2 Electronic speed sensing

Turbine speed can be measured electronically by converting a frequency input into an equivalent unit speed. The input frequency should be from a source producing a frequency that is directly proportional to the unit speed. This input frequency may be measured using analog technology, digital technology, or a combination of both technologies. The measurement method should have sufficient resolution to meet the required deadband requirement, and the update time and time constant of the measurement should allow the unit to meet its deadtime requirement. The input frequency source may be developed by several alternative methods, each having inherent benefits and disadvantages. The most common problem with electronic speed signals is an unwanted frequency modulation of the signal due to mechanical irregularities in the speed signal source. This unwanted modulation causes movement of the distributing valve and, in severe cases, movement of the turbine control servomotor. This movement increases both the consumption and the temperature of the fluid in the hydraulic pressure system of the turbine governing system.

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6.3.7.2.1 Speed signal generator (SSG)

An SSG consists of a gear or toothed wheel that is mechanically coupled to the turbine or generator shaft. Passive magnetic pickups or active proximity pickups sense the passage of the SSG gear teeth to produce a frequency that is proportional to the speed of the turbine. Active proximity pickups are able to sense the rotational speed down to zero speed, making this approach usable for a full range of speed switch functions including creep detection. Passive magnetic pickups require a minimum speed of rotation before the pickups produce sufficient output so the unit speed can be detected by the speed sensing circuitry. The mechanical coupling from the unit shaft to the gear or toothed wheel should be designed so the coupling does not impart unwanted modulation of the sensed speed signal. Lost motion in the coupling and asymmetry in the coupling are two major sources of speed signal modulation. The precision of manufacture and the concentricity of rotation of the gear or toothed wheel also affect the quality of the speed signal produced by the SSG. If an SSG gear is used to drive mechanical speed switches, the mechanical loading of these switches on the gear can impart a modulation of the speed signal due to windup of the gear coupling.

6.3.7.2.2 Shaft-mounted speed source

A wraparound gear or reflective tape may be used to sense the speed of rotation of the turbine shaft. Ideally, this wraparound device should be mounted as near as possible to a turbine shaft guide bearing to minimize the effects of shaft vibration and runout on the speed signal. Multiple speed pickups may also be distributed around the circumference of the wraparound device to average out these effects. Using this approach generally provides a very clean and stable speed signal. However, the dimensions of the wraparound device need to be precisely matched to the dimensions of the turbine shaft, and the mounting and adjustment of the multiple speed pickups require considerable design and installation time.

6.3.7.2.3 Voltage transformer frequency

The generator voltage transformer output can be used as a speed input to the governor speed sensor. Using this approach eliminates the possibility of mechanical irregularities causing unwanted frequency modulation of the speed signal. However, if this method of measuring unit speed is intended to be used prior to applying the generator field current, the speed sensing circuitry needs to be sensitive enough to sense the unit speed from the residual output of the generator before excitation is applied. There is a minimum unit speed at which the speed can be reliably measured, depending on the sensitivity of the speed sensing circuitry and the strength of the residual magnetism in the generator rotor. The speed sensing circuitry should be capable of rejecting switching transients that may appear on the output of the generator voltage transformer. Also, as there is a possibility that a generator or line fault may demagnetize the generator rotor, or a fuse in the voltage transformer circuit may fail, it is advisable to use at least one additional speed signal source as a backup source of speed measurement.

6.3.8 Manual control

Manual control generally describes a mode of operation where the turbine control servomotors are set by a manually adjusted reference. This mode of operation may be accomplished using a mechanical gate limit, if one is available. Manual control may also be accomplished either through a mode of operation using the main governor controller or by using a separately powered controller that controls only the position of the turbine control servomotors. In any case, normal protective interlocks are generally not functional when a system is placed in manual control, and the risks of operating in manual control should be carefully evaluated.

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6.3.9 Servomotor position indicator

The indication of servomotor position may be accomplished by a pointer that is mechanically driven from the servomotor, from an indicating meter that is driven from a servomotor position transducer, or by a numeric display on a graphic user interface. These methods are functionally equivalent, and if one method is preferred, it should be specifically stated in the specifications for the turbine governing system.

6.3.10 Automatic shutdown

Automatic shutdown is the process of stopping the unit under normal conditions. A typical automatic shutdown closes the turbine control servomotor at a controlled rate until the unit generation is approximately zero. At that time, the unit breaker is tripped and the turbine control servomotors are fully closed. The operation of field breakers, excitation systems, and other auxiliary systems are also included in the automatic shutdown process. There are many variations of the automatic shutdown process, and the desired process should be completely described in the specifications for the turbine governing system. The automatic shutdown process may be impacted by the functionality of specific devices used in the shutdown process such as limit switches, control switches, and transducers.

6.3.11 Emergency stop pushbutton

Generally, emergency stop pushbuttons are located strategically at major system components such as at the unit breaker, the governor control panel, and the unit control board. When an emergency stop is initiated, the unit is generally shut down by opening the unit breaker, closing the turbine control servomotors at their maximum rate, removing excitation, and shutting down all auxiliary systems. The unit overspeed experienced as a result of an emergency stop makes this method of stopping a unit undesirable except in cases of severe equipment problems. Emergency stop pushbuttons are generally large red mushroom pushbuttons, or pushbuttons protected by covers that are required to be lifted before the buttons can be pressed.

6.3.12 Generator air braking system

The generator air braking system may consist of a dedicated air compressor, pressure vessel, piping, brake/jacking cylinders, and pressure monitoring devices. Devices located with the turbine governing system may operate the air braking system, or it may be operated by other parts of the unit control system.

6.3.13 Air brake pressure gauge

An air brake pressure gauge provides an indication of the pressure available for operating the air brakes. This indication may be provided either by a pressure gauge connected to the air supply piping, or it may be displayed on a graphic user interface that is part of the turbine governing system or the unit control system.

6.3.14 Speed indicators

Speed indicators display the unit speed. Speed indicators may be mechanically driven from the turbine shaft, or they may be electronically driven from speed transducers. Electronically driven speed indicators may be either separate indicating meters, or they may be numeric displays on a graphic user interface.

6.3.15 Adjustable-blade turbine control

For adjustable-blade type turbines, the turbine runner blades are controlled in accordance with the turbine manufacturer’s requirements for an optimal relationship of the runner blade angle with the wicket gate position and the operating conditions. Operating conditions that influence the operation of the turbine runner blades are unit startup, unit synchronizing, online operation, and unit shutdown. The effective operating head of the unit also impacts the optimum operation of the unit.

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6.3.15.1 Blade controller

The blade controller should be capable of controlling the blade servomotor within 1% of the ideal blade position as determined by the turbine manufacturer for the turbine operating conditions. This tolerance should apply over the full range of operating head and wicket gate position. Upon loss of governor oil pressure, the blades should be allowed to move to the position of steepest pitch to limit the runaway speed of the turbine. The ideal blade position for any operating condition may be determined either as a two-dimensional (2D) function of gate position or as a three-dimensional (3D) function of gate position and effective head. The blade positioning function may be accomplished by either mechanical cams or by curves stored in data maps within a digital controller.

6.3.15.2 Blade lock

A blade lock is used to prevent further movement of the blade servomotor under specified conditions. The blade lock function may be accomplished mechanically by closing isolating valves in the blade servomotor control lines, or it can be implemented electronically by preventing the blade servomotor position setpoint from changing. Typically, the blade lock function is activated when the governor oil pressure drops below a preset level. By preventing further movement of the blade servomotor, the governor oil pumps are able to build the system oil pressure to a level with sufficient reserve to allow the blades to continue movement to their optimal position.

6.3.15.3 Manual blade control

Manual blade control generally describes a mode of operation where the blade servomotor is set by a manually adjusted reference. This mode of operation may be accomplished using a mechanical blade position limit, if one is available. Manual control may also be accomplished through a mode of operation using the main governor controller. Furthermore, manual control may be accomplished by using a separately powered controller that only controls the position of the blade servomotor. In any case, normal protective interlocks are generally not functional when a system is placed in manual control, and the risks of operating in manual control should be carefully evaluated before operating in this mode.

6.3.15.4 Blade angle indicator

The indication of blade servomotor position may be accomplished by a pointer that is mechanically driven from the servomotor, from an indicating meter that is driven from a servomotor position transducer, or by a numeric display on a graphic user interface. These methods are functionally equivalent, and if one method is preferred, it should be specifically stated in the specifications for the turbine governing system.

6.3.15.5 Blade tilt

A blade tilt feature may be incorporated to limit the maximum unit overspeed in the event of a load rejection. Usually, the blade tilt is initiated around 110% of normal speed, and it moves the turbine blades to their steepest position. Separate control devices such as a blade tilt solenoid can implement this function, or it can be integrated into the controller functions of the turbine governing system.

6.3.15.6 Startup blade position

The blades may be prepositioned to a steeper off-cam position for startup of the turbine. This provides greater developed torque for a faster breakaway. This steeper angle also results in less loading of the unit's thrust bearing on startup. Starting the unit with the blades positioned to a steeper off-cam position often results in more stable controllability of the speed of the turbine, resulting in easier synchronization of the unit to the power system. After synchronizing to the power system, the blades return to their on-cam position, which results in some increase of power output of the generator. This automatic pickup of generation after synchronization helps to avoid problems with reverse-power protective relaying on the unit.

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6.3.15.7 Blade oil shutoff

Automatically controlled blade servomotor oil shutoff valves can be provided in the blade servomotor control lines to implement the blade lock feature described in 6.3.15.2. Manually controlled blade servomotor oil shutoff valves can be provided for maintenance purposes to isolate the blade servomotor from the hydraulic pressure supply.

6.3.16 Pump turbine control

In the generate mode, the operation of a pump turbine is essentially the same as the operation of a conventional turbine. However, in the pump mode, there are some additional considerations in the operation of the unit.

6.3.16.1 Multiple gate rates

For pumping operation, a much slower opening rate is generally required to achieve a smooth pump loading. This slower gate rate may be achieved by auxiliary timing devices on the gate distributing valve, or the slower opening rate may be achieved by limiting the rate of change of the setpoint from the governor controller. The gate closing rate for a pump turbine is generally much faster than the opening rate to minimize or eliminate flow reversal in the penstock after a pump input rejection.

6.3.16.2 Pumping gate control

An automatic gate positioning control system can be designed to maintain optimum pumping efficiency throughout the entire pumping cycle. This control positions the wicket gates as a function of the effective head across the turbine. The pumping gate control may be accomplished either with mechanical cams of the desired profile (typically with mechanical-hydraulic actuators) or with pump gate curves stored in data maps within the digital turbine governor controller.

6.3.17 Hydraulic pressure supply system

The hydraulic pressure supply system uses various oil pressures and levels to control the pressure pumps, issue alarms, and to shut down the unit under abnormal conditions. Some design and operating parameters for hydraulic pressure supply systems are mandated by applicable safety standards for the jurisdiction of use. Specifications, design, and construction of the hydraulic pressure supply system should comply with these applicable safety standards.

6.3.17.1 Operating parameters

Typical values used in controlling a hydraulic pressure supply system are described in 6.3.17.1.1–6.3.17.1.14.

6.3.17.1.1 Maximum system pressure

The maximum system pressure is the pressure where the gas pressure relief valve on the pressure accumulator tank operates to bleed off excessive pressure to protect the hydraulic system from catastrophic component failure. This pressure is typically from 10% to 15% above the nominal system pressure.

6.3.17.1.2 Maximum pumping pressure

The maximum pumping pressure is the pressure where the oil pressure relief valve on the discharge of the oil pressure pump operates to prevent the pump from imposing excessive pressure on the system. This pressure is typically 5% to 10% above the nominal system pressure.

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6.3.17.1.3 High accumulator oil level alarm

An alarm may be issued to indicate a condition where the oil level in the hydraulic pressure accumulator tank is above its normal level. A second level, below the alarm level, may be detected and used in conjunction with the hydraulic supply pressure to initiate the operation of an automatic gas admission system used to readjust the operating oil level in the pressure accumulator tank.

6.3.17.1.4 Nominal system pressure (lead pump stop pressure)

The nominal system pressure is the design oil pressure for the hydraulic pressure supply system. This is the pressure at which the lead pump stops.

6.3.17.1.5 Lag pump stop pressure

The lag pump stop pressure is the pressure at which the lag pump stops. This pressure may be at or below the nominal system pressure.

6.3.17.1.6 Lead pump start pressure

The lead pump start pressure is the pressure at which the lead pump starts to replenish the oil in the pressure accumulator tank consumed by system operation. This pressure is typically around 90% of the nominal system pressure.

6.3.17.1.7 Lag pump start pressure

The lag pump start pressure is the pressure at which the lag pump starts to assist the lead pump in replenishing the oil in the pressure accumulator tank consumed by system operation. The lag pump start pressure is typically around 85% of the nominal system pressure.

6.3.17.1.8 Low pressure alarm

A “low pressure” alarm may be issued to indicate an abnormal, but not critical, operating condition. This alarm point is generally set below the lead pump start pressure and is sometimes set below the lag pump start pressure.

6.3.17.1.9 Low accumulator oil level alarm

A “low-accumulator oil level” alarm may be issued to indicate that the oil level in the hydraulic pressure supply accumulator tank is below its normal level. The oil level measurement may be coordinated with the hydraulic supply pressure measurement to determine when the abnormally low level condition occurs.

6.3.17.1.10 Low pressure shutdown alarm

A “low pressure shutdown” alarm may be issued when the system oil pressure is critically low, requiring the unit to be immediately shut down to protect the integrity of the unit.

6.3.17.1.11 Minimum accumulator oil level

The minimum accumulator oil level is the level at which the pressure accumulator tank cannot provide any additional oil volume to the hydraulic servomotor actuating system. For a gas-over-oil type of pressure accumulator tank, this is the level at which the tank’s float valve closes. For a transfer barrier type of pressure accumulator tank, this is the level at which the bladder or piston bottoms out within the accumulator tank.

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6.3.17.1.12 Minimum system pressure

The minimum system pressure is the system pressure that corresponds to the minimum accumulator oil level. Historically, this is approximately 75% of the nominal system pressure. In order to provide sufficient pressure so the turbine control servomotors can safely shut down the unit under all conditions, the minimum system pressure should be at least 5% greater than the maximum servomotor loading.

6.3.17.1.13 Sump tank high level alarm

A sump tank high level alarm may be issued to indicate that the oil level in the sump tank is above its normal maximum level. This alarm may occur when there is excessive oil in the system or when excessive gas admission to the pressure accumulator tank has occurred.

6.3.17.1.14 Sump tank low level alarm

A sump tank low level alarm may be issued to indicate that the oil level in the sump tank is below its normal minimum level. This alarm may indicate loss of oil from the system or loss of gas cushion from the pressure accumulator tank.

6.3.17.2 Oil pressure pumps

It is common practice to use two pressure pumps to maintain system hydraulic oil pressure for operating the turbine control servomotors. Using multiple pumps offers a degree of redundancy, making the available oil pressure supply more reliable. Some larger systems may employ more than two pumps to achieve the required pumping capacity using more readily available pump and motor sizes. The flow capacity of the oil pressure pumps should allow the system to reestablish the hydraulic system pressure in a reasonable period of time after a controlling movement of the turbine control servomotors. Industry practice is to specify the combined flow capacity of the hydraulic pressure pumps to be at least 25% of the flow required to move the turbine control servomotors at their maximum rates.

6.3.17.2.1 Echelon control

An echelon control system can be provided that allows any of the pressure pumps to be selected as the lead pump. This selection can either be done manually or automatically. Periodic changes in the lead pump designation tends to equalize the amount of running time for each of the oil pressure pumps.

6.3.17.2.2 Makeup pump

In some cases, a smaller makeup pump, also known as a “jockey pump” or a “pony pump,” may be provided to maintain system pressure during steady-state operating conditions. The makeup pump helps to reduce the number of start/stop cycles of the lead pump when the turbine control servomotors are at a relatively constant operating position. Generally, the makeup pump is designed to be able to provide two times the expected oil consumption during steady-state operating conditions. The flow capacity of the makeup pump can be included in the total oil pumping capacity when determining the required size of the pumps for the oil pressure supply system. Any makeup pump flow capacity in excess of the average required steady-state oil consumption can be diverted through a kidney loop filtering system to automatically maintain the oil system cleanliness at an acceptable level, and it can be passed through an oil warming system to maintain a desirable oil viscosity. Oil warming at a low intensity can be provided by passing the oil through a back-pressure valve before returning the oil to the oil sump tank. Ideally, an oil temperature approximately 5°C above the ambient air temperature should be maintained to help drive out condensation as well as to provide a desirable oil viscosity level.

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6.3.17.2.3 Oil pressure safety relief valve

Generally, an oil pressure safety relief valve is installed at the discharge of each oil pump to prevent excessive pressure buildup. Typically, the oil pressure safety relief valve operates at 105% to 110% of the nominal system oil pressure. The actual setting of the pressure safety relief devices should comply with the applicable safety standards for the jurisdiction of use.

6.3.17.2.4 Black start capability

In certain applications, a unit will be required to perform a "black start," which requires the ability to start the unit without any AC power available to the unit. Often, where black start capability is required, a DC-powered oil pressure pump is provided to produce sufficient hydraulic oil pressure to start the unit. This DC-powered oil pressure pump uses the station battery system for operating power. Generally, the flow output from this DC-powered oil pressure pump is not required to produce the oil flow specified for the main AC-powered oil pressure pumps. For black start capability, it is only necessary to be able to build sufficient oil pressure to start the unit within a specified time frame. Therefore, a unit with a separate oil pressure pump used to build oil pressure for a black start may have an output flow of one-fourth or less of the flow produced by the main AC-powered pumps. After a black start, the main AC-powered oil pressure pumps are usually powered by the output of the generator just started, thus allowing continued normal operation of the unit.

6.3.17.3 Pressure accumulator tanks

The pressure accumulator tanks store the hydraulic energy needed to move the turbine control servomotors. The pressure accumulator tank should have sufficient stored energy to safely shut down the unit when the pumping system has failed. Industry practice is to provide a pressure accumulator tank that holds an active oil volume (the volume of oil above the minimum accumulator oil level) of at least 20 times the combined oil volume of the turbine control servomotors. The normal operating oil level in the pressure accumulator tank is then chosen to achieve the desired energy reserve in the tank. Experience has shown that operating with 15 servomotor volumes of gas in the pressure accumulator tank while operating at nominal system pressure generally results in an acceptable relationship between the system pressure and the reserve volume of oil within the pressure accumulator tank. To achieve sufficient accumulator tank volume in a limited space, a separate tank or system of tanks is often used to store the gas cushion for the pressure accumulator system. These gas cushion tanks are generally connected to a point at the top of the pressure accumulator tank containing the system oil.

At higher operating pressures, the design of air-over-oil pressure accumulator tank systems contends with the effects of entrainment of air with oil at high pressure. This can lead to localized combustion of the oil with entrained air occurring within the pump, resulting in blackening of the oil due to the carbon in the byproducts of the combustion. Consideration of air entrainment control should be exercised in the design of a hydraulic pressure system with a nominal operating pressure exceeding 7 MPa. Air entrainment can be reduced by proper sump tank design, as discussed in 6.3.17.4. Air-over-oil accumulator systems have been successfully operated at nominal system pressures as high as 10 MPa. Some alternatives to air-over-oil accumulators include nitrogen over oil accumulators (because nitrogen does not support combustion of the oil), bladder accumulators, and piston accumulators.

To safely shut down the unit, the pressure accumulator tank should to provide at least enough oil to close the turbine control servomotors at a pressure sufficient to overcome the mechanical loading on the servomotors. Typical specifications require from 1.5 to 3.0 servomotor volumes of oil to be available from the low pressure shutdown alarm oil level to the minimum accumulator oil level of the pressure accumulator tank. If a float valve is not provided within a gas-over-oil pressure accumulator tank, then the design should ensure that the oil level in the pressure accumulator tank is always above the point where gas may begin to enter the discharge of the pressure accumulator tank. Allowing gas to enter the hydraulic pressure piping can result in violent instability of the hydraulic control valves and significant damage to the control valves, the turbine control devices, and (in severe cases) the turbine water passages. To allow that the unit to always be able to safely shut down the unit, the system oil pressure should remain above the maximum servomotor loading pressure in the shutdown direction during a low pressure shutdown alarm condition.

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To prevent the lag pump from starting excessively, it is industry practice to specify that the pressure accumulator tank be sized large enough to provide 1.5 turbine control servomotor volumes of oil before the system pressure drops to the "lag pump start pressure." The classic pressure accumulator tank design with 20 active servomotor volumes of oil meets this requirement.

Regardless of how the pressure accumulator tank is specified, an overriding concern is that the system is able to provide sufficient oil flow and pressure to safely shut down the unit in the event of failure of all pressure pumps.

6.3.17.3.1 High pressure accumulator tanks

Hydraulic pressure supply systems operating at nominal system pressures above 10 Mpa generally use a transfer barrier type of pressure accumulator tank. A transfer barrier type of pressure accumulator tank uses a bladder or piston to separate the oil from the pressurized gas cushion. This transfer barrier prevents the absorption of gas into the oil. Typically, dry nitrogen is used to charge the gas cushion in a transfer barrier type of pressure accumulator tank. A transfer barrier type of accumulator system is often constructed from two or more accumulator tanks connected together. This allows the system to achieve the desired accumulator volume by using commercially available accumulator tanks (often called "bottles"). The usable capacity of a transfer barrier type of pressure accumulator tank can be increased by connecting a gas bottle (often called a "capacitor") to the gas reservoir connection of the transfer barrier accumulator tank.

6.3.17.3.2 Float valve

A gas-over-oil pressure accumulator tank may be equipped with a float valve device that closes at low oil levels, preventing the entry of gas into the hydraulic pressure supply lines. Due to the low viscosity and compressible nature of gas, the introduction of gas into the hydraulic pressure supply lines of a servomotor control system can have catastrophic results. Servomotor control valves can become unstable, causing violent oscillations of the turbine control devices. The rate of flow of gas through oil metering devices can greatly exceed the safe limits for the actuating devices. The presence of gas in a hydraulic servomotor positioning system can, therefore, result in the catastrophic failure of turbine control devices, turbine/generator devices, and water passageways. If a float valve is provided as part of a gas-over-oil pressure accumulator tank, the float valve should be of a proven, highly reliable design and construction. If a gas over oil pressure accumulator tank is provided without a float valve device, the resulting compromise of the safety and integrity of the hydraulic servomotor control system with either a loss of power to the hydraulic pressure pumps or the failure of the oil pressure pumps should be evaluated.

6.3.17.3.3 Gas pressure safety relief valve

For a gas-over-oil pressure accumulator tank, a gas pressure safety relief valve is often supplied to limit the maximum system pressure to a level within the safe design limits of the pressure accumulator tank and the other hydraulic components in the hydraulic pressure supply system. Generally, the discharge from the gas pressure relief valve should not be connected to any piping or sound muffling devices, because these connections can affect the maximum flow capacity of the relief valve. Also, the connection of the gas pressure relief valve should not have any isolating valving between the pressure accumulator tank and the relief valve to prevent the inadvertent disabling of this safety device.

6.3.17.3.4 Manual gas bleed valve

A manually operated valve may be provided to allow the bleeding off of the gas pressure cushion from the oil pressure accumulator tank. This valve is generally used only for maintenance and testing purposes. The discharge from the manual gas bleed valve is often connected through a sound muffling device and suitable piping to divert the gas discharge to a location that does not present a safety hazard to nearby personnel.

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6.3.17.3.5 Manual oil bleed valve

A manually operated valve may be provided to allow the bleeding off of oil from the pressure accumulator tank. This valve is generally used only for maintenance and testing purposes. The discharge from the manual oil bleed valve should be submerged below the oil level in the sump tank to prevent excessive aeration of the oil.

6.3.17.3.6 Pump cycle time

For oil pressure systems without a makeup pump, the lead pump starts at regular intervals to replace the oil consumed by the servomotors and their control valves. During steady-state operation of the unit, it is generally considered acceptable if the time between pump starts is 10 minutes or greater. For small systems, or for systems with a high steady-state oil usage, a larger pressure accumulator tank may be required to achieve an acceptable lead pump cycle time. A properly sized makeup pump effectively eliminates the lead pump cycling under steady-state conditions.

6.3.17.4 Sump tank

The sump tank is the reservoir of hydraulic oil stored at atmospheric pressure. The size of the sump tank should be sufficient to store the system oil that can be drained into it without overflowing. Typically, the sump size is specified to be 110% of the system oil that can be drained by pressure or by gravity into the sump tank. The sump tank should have some method of displaying its oil level, such as a sight glass or level transducer and indicator. The sump tank should also have access to its interior for inspection and maintenance. Designing the floor of the sump tank with a slight incline toward a drain connection facilitates the removal of oil during maintenance activities. The design of the sump tank should provide an oil return path that is long enough to allow the adequate release of entrained gas before the returning oil reaches the pressure pump inlets. A method of maintaining a minimum oil temperature, such as sump oil heaters or unloader back-pressure control, facilitates the release of entrained gas from the oil within the sump tank. The sump tank should also be designed to adequately control the ingress of airborne contaminants into the oil system. This is typically accomplished by sealing all openings into the sump tank and using a filtered breather vent for the passage of air into the sump tank. The breather filter should be fine enough to prevent the majority of airborne particles from entering the system. Experience has shown that a 10 micron or finer filter provides acceptable sump air filtration. The breather filter should also be sized so an excessive pressure change does not occur in the sump tank during maximum-rate movement of the turbine control servomotors. The sump tank is often used as a base for mounting the pressure pumps and the governing system distributing valves.

6.3.17.5 Automatic gas admission

The proper fluid-to-gas ratio should be maintained so the pressure accumulator tank has sufficient stored energy for the safe operation and shutdown of the hydroelectric generating unit. Over time, a gas-over-oil pressure accumulator tank loses some of its gas cushion charge through absorption of the gas into the oil. If an automatic gas admission system is specified, it should be capable of maintaining this gas to fluid ratio. A compressed gas supply (typically either air or dry nitrogen) of sufficient pressure should be part of the automatic gas admission system, along with suitable pressure and level sensing devices. A procedure for admitting gas should be defined and implemented as part of the gas admission system. The actual control of this automatic gas admission system can be implemented within a separate controller, as a function within the unit controller, or as a function within the turbine governing system controller.

6.3.17.6 Electric motors and starters

Electric motors and their associated motor starters are used to drive pumps to provide hydraulic pressure supply for the turbine governing system. The motor starters may be located on the governor cabinet, on the pump base (sump tank), in a separate motor control center (MCC).

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6.3.17.7 Piping and fittings

The hydraulic piping and fittings should be of a leak-resistant design and constructed so the system can be disassembled for cleaning and maintenance purposes. Flanged, bolted joints are generally used on piping greater than 25 mm in diameter. Control piping less than 12 mm diameter is often seamless hydraulic tubing using suitable fittings. The piping should be sized so the maximum fluid velocity within the pipes does not exceed 5 m/s. This criterion generally keeps the pressure drops within the oil piping to negligible levels. However, the impact of piping and fitting losses should be considered as a percentage of operating pressure. Therefore, for longer piping runs or lower operating pressures, the oil velocity may need to be reduced so there is sufficient operating pressure at the turbine control servomotor. The pressure measured across the turbine control servomotor is an indication of the hydraulic force available to move the turbine control device.

Oil piping to the runner blade servomotor of an adjustable-blade turbine should have provisions to prevent the flow of electrical currents. This prevents the erosive effects of circulating currents upon turbine and generator bearings. This electrical isolation is typically accomplished by using insulating gaskets in the servomotor pipe flanges at the turbine oil head.

6.3.17.8 Pressure gauges

Pressure gauges are used to indicate the hydraulic oil pressure, the brake system supply pressure, the brake cylinder pressure, or the turbine control servomotor cylinder pressures. Pressure gauges may be separate gauges connected to the points of interest, indicating meters connected to transducers measuring the pressures of interest, or numeric displays on a graphic user interface displaying the output of transducers connected to the pressures of interest.

6.3.17.9 Oil cleanliness control

The cleanliness of the hydraulic oil used in the turbine governing system has a significant impact on both the performance of the system and the longevity of the system components. It is estimated that at least 75% of all failures of hydraulic control components are related to oil contamination. The major areas of concern in maintaining an acceptable degree of hydraulic oil system cleanliness are discussed in 6.3.17.9.1 – 6.3.17.9.5.

6.3.17.9.1 Initial component cleanliness

The initial cleanliness of the components used in the hydraulic turbine control servomotor system is one of the most important influences in establishing the desired degree of oil system cleanliness. During the installation of the hydraulic system components, it is important to mechanically clean all system components prior to final installation. This often means that the components need to be disassembled and mechanically cleaned after the initial fabrication and assembly is completed so all rust, scale, dirt, preservatives, and byproducts of welding and grinding have been removed. This procedure should be followed for all system components that contain hydraulic oil, including the piping, the pressure accumulator tank, the sump tank, the turbine control servomotors, and the servomotor control valves. After cleaning of these components is completed, they should be coated with oil and sealed against the ingress of contaminants until the final assembly of the components is done. This procedure is especially important if any on-site welding or grinding is performed as part of the installation of the system.

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6.3.17.9.2 Initial oil cleanliness

The cleanliness of the oil introduced into the turbine servomotor control system should be verified before this oil is introduced into the system. Often, the new oil is passed through an on-site filtration system with a nominal rating of 10 micron or finer prior to being introduced into the turbine servomotor control system. Note that, to be effective, the filter's beta ratio should be at least 75 at the specified micron rating of the filter. This approach eliminates the possibility of introducing contamination from the oil-transporting vessel into the hydraulic control system.

6.3.17.9.3 Initial in-system filtration

Once the hydraulic oil pressure system has been commissioned, the cleanliness of the oil should be carefully checked at regular intervals to verify that an acceptable degree of oil cleanliness has been achieved. Several movements of the turbine control servomotors should be accomplished at regular intervals while continuously filtering the oil in the system through a kidney-loop filtration system. This procedure should be continued until the desired degree of oil cleanliness has been achieved. Experience has shown that 12 full-rate opening and closing cycles per day of the turbine servomotors for one week effectively dislodges most trapped contamination within a hydraulic system, allowing the filtration system to remove the suspended contamination from the oil. Generally, an ISO 4406:1999 cleanliness level of 17/15/12 provides an acceptable level of performance and longevity. Experience has shown that an ISO 4406:1999 cleanliness level of 18/16/13 generally provides acceptable operation for systems operating above 3 MPa, and an ISO 4406:1999 cleanliness level of 20/18/15 provides acceptable operation for systems operating below 3 MPa.

6.3.17.9.4 Contamination control

One of the most common sources of oil system contamination is from airborne particles. Each time oil is pumped from the sump into the hydraulic pressure accumulator, the oil sump draws in atmospheric air to replenish the volume of oil used from the sump. Particles suspended in the air can then settle into the sump tank and contaminate the oil. A discussion of pertinent sump tank design considerations is provided in 6.3.17.4.

Another source of oil system contamination is from the wearing of moving parts in the system. These moving parts include the oil pressure pumps, the servomotor control valves, and the turbine control servomotors. Some of these metallic particles can be removed from the oil system by using a magnetic trap.

6.3.17.9.5 Operational filtration

During normal operation of a hydraulic oil pressure supply system, some degree of oil contamination occurs. Generally, the pilot stages of the servomotor control valves have filtration systems designed to remove contaminant particles that could damage or downgrade the performance of these pilot stages. Additionally, kidney loop filtration of the entire oil system can be accomplished during the normal operation of the system. If this kidney loop filtration system is run continuously, the oil cleanliness is maintained throughout the system. Alternatively, this kidney loop filtration system can be connected to the hydraulic pressure system at specified intervals and run until the measured oil cleanliness meets the desired specification.

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6.3.18 Nameplate

The turbine governing system should have a permanent metal nameplate clearly marked or stamped to show the following information:

a) Manufacturer's name and address

b) Model or type number of the turbine governing system

c) Serial number of the turbine governing system

d) Maximum safe working governor hydraulic pressure

e) Minimum normal working governor hydraulic pressure

f) Rated capacity in volume per unit time at the minimum normal working governor hydraulic pressure

6.3.19 Switches

Position switches, level switches, pressure switches, and similar devices may be used in circuits to control and protect certain functions of the turbine governing system. These switch functions may also be accomplished by using position, level, pressure, and similar transducers as inputs to the turbine governing system. These transducer signals may then be compared against the desired switch operation points within the application software of the turbine governing system to accomplish the desired control functions. Using transducers can provide controls that are easier to set up for accomplishing the desired control performance, because all switching functions are developed from the one transducer signal. However, the impact of a single-point failure of the transducer should be considered in the reliability of the system design. To enhance the reliability of the system, multiple transducers can be used, or independent sensing devices can be used as backup for critical functions.

6.3.19.1 Overspeed switch

Historically, the preferred method of detecting unit overspeed for protective shutdown of the unit has been the use of a mechanically driven speed switch. Mechanically driven overspeed switches have been specified for the purpose of achieving highly reliable overspeed detection for this protective function. Any intermediate mechanical coupling devices between this mechanically driven overspeed switch and the turbine shaft reduces the reliability of the overspeed detection system. The reliability of current electronic technology is sufficient, in most cases, to allow the use of electronic speed switches for the protective tripping of the unit due to overspeed. In considering the overall reliability of any overspeed detection system, the maintenance and calibration needs of the system should be considered. If a system requires too much time to be properly maintained and calibrated, it is unlikely that the required calibration and maintenance will be performed very often. This may result in a system that is unreliable or inoperable, which is not desirable for a protective function such as the main overspeed trip system.

Some approaches that have been used for the protective tripping of the unit due to overspeed are discussed in 6.3.19.1.1 – 6.3.19.1.4.

6.3.19.1.1 Mechanically driven centrifugal device operating an electrical switch

This type of overspeed switch uses rotating weights mechanically coupled to the turbine or generator shaft. Centrifugal force at the trip speed of the overspeed switch overcomes the spring preloading of these weights, allowing these rotating weights to swing outward to operate an electrical switch. To check the calibration of this device, the centrifugal weight device may be uncoupled from the turbine shaft and driven by a test motor system. The couplings that connect the centrifugal weight device to the turbine shaft are a potential failure point of the system, and their impact on the overall reliability of the overspeed detection system should be considered when evaluating the overall reliability of the overspeed detection system, All moving parts should be properly lubricated and maintained for reliable operation of this system.

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6.3.19.1.2 Electronic switch using magnetic or optical pickups

Magnetic or optical pickups normally sense the speed of the turbine shaft from the frequency produced by the passage of the teeth, slots, or optical stripes on a device that is mechanically coupled to the turbine or generator shaft. To check the calibration of this system, a test frequency can be injected into the electronic speed switch without actually rotating the unit. Any mechanical couplings between the turbine shaft and the toothed wheel used for speed detection are potential failure points of the system. The impact of the reliability of these couplings on the overall reliability of the overspeed detection system should be considered. To improve the reliability of the overspeed detection system, double or triple redundant speed pickups and overspeed switches may be used in a fault-tolerant arrangement.

6.3.19.1.3 Electronic overspeed switch driven by voltage transformer frequency

To check the calibration of this system, a test frequency may be injected into the electronic speed switch without actually rotating the unit. The electronic switch should be sensitive enough to detect the voltage transformer frequency at low residual voltages as produced without generator excitation. One condition that should be considered is the possible demagnetization of the generator rotor due to an electrical fault, resulting in a residual voltage that is too low to detect. Another condition to consider is the loss of one or more voltage transformer fuses. To improve the reliability of the overspeed detection system, double or triple redundant speed switches may be used (each sensing a different phase of the voltage transformer) in a fault-tolerant arrangement.

6.3.19.1.4 Shaft-mounted centrifugal device operating a hydraulic shutdown valve

To check the calibration of this device, it is necessary to run the unit up to the trip speed. Because of the high peripheral velocity of the trip arm, there is the likelihood of damage to the shutdown valve upon tripping of the overspeed device. For reliable operation of this system, all moving parts should be properly lubricated and maintained.

6.3.19.2 Speed switches

Speed switches are devices used to operate an electrical contact as a function of the unit speed. Speed switches may be driven mechanically from the turbine shaft, or they may be driven electrically from a speed probe sensing the passage of the teeth of a gear or toothed wheel. Speed switch functions may also be derived from sensing the frequency of the generator voltage transformer. Speed switch functions may consist of separate devices for each switch contact, consist of a single device with multiple contacts for different speed settings, or be derived within the turbine governing system by comparing the measured turbine speed against the desired operating speeds of the speed switch functions. Integrating certain speed switch functions into the turbine governing system can improve the reliability of the system and reduce wiring and maintenance costs. It is important to consider the reliability of the method used to develop the speed switch functions when designing a turbine governing system.

6.3.19.3 Servomotor limit position switch

A servomotor limit position switch operates when the turbine control servomotor reaches the end of its travel. This function can be achieved by installing a switch that is mechanically or magnetically actuated by the servomotor or gate ring at the end of the servomotor travel. A servomotor limit position switch function can also be implemented by comparing the signal produced by a servomotor position transducer with the desired limit position setting.

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6.3.19.4 Servomotor position switch

Servomotor position switch functions operate switch contacts at specified intermediate positions of the turbine control servomotor. This function can be achieved by installing a switch that is mechanically or magnetically actuated by the servomotor or gate ring at the specified position. A servomotor position switch function can also be implemented by comparing the signal produced by a servomotor position transducer with the desired position switch setting.

6.3.19.5 Hydraulic system pressure, level, and temperature switches

Hydraulic system pressure, level, and temperature switch functions provide control, alarm, and shutdown alarm level sensing for these system parameters. Individual switches that are adjusted to operate at the desired pressure, level, or temperature may implement these switch functions. These switch functions may also be implemented by comparing the signal produced by a transducer measuring these quantities with the desired switch operating level. Only critical shutdown alarm levels may require a dedicated switch device, with other operating and alarm levels being detected using the appropriate transducer signals. Depending on the size and importance of the hydroelectric unit, it may be acceptable to detect all operating, alarm, and shutdown alarm conditions from the appropriate transducer signals.

6.3.19.6 Coincidence switch

A coincidence switch detects when the servomotor is operating against its position limit. This indicates that the turbine governing system is no longer able to increase the output of the unit. This limited condition is often referred to as a “blocked-load” condition. Under this limited condition, the turbine governing system no longer contributes to the frequency stability of the interconnected system.

A coincidence switch may be implemented mechanically on a turbine governing system that uses mechanical position restoring feedback. For electrohydraulic turbine governing systems, the coincidence switch function is generally implemented within the governor controller. The coincidence function may be used externally as an indication of a blocked-load condition, or it may be used internally to the governor controller to prevent windup of the applicable control algorithm setpoint.

6.3.19.7 Speed changer auxiliary switches

A speed changer auxiliary switch function is used to indicate to the unit control system that the speed reference for the turbine governing system is at a specific level. A speed changer auxiliary switch may be a mechanical switch actuated from a mechanical speed reference or from a motorized potentiometer assembly that provides a speed reference to the turbine governing system. A speed changer auxiliary switch may also be implemented in a digital controller by comparing the speed reference against the desired switching levels and then appropriately activating contact outputs from the system. When using a digital controller, many of the functions accomplished by speed changer auxiliary switches can be more effectively implemented within the application program for the digital controller.

6.3.19.8 Air brake pressure switch

The air brake pressure switch indicates that the air pressure supply for the braking system is below a specified level. This switch is generally used to initiate an alarm. Typically, the air brake pressure switch is a separate pressure switch, but this function can also be implemented by comparing the signal from an air brake pressure transducer (if available) with the desired switching level.

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6.3.20 Automatic servomotor lock control

If specified, a servomotor locking system may be provided on the primary turbine control servomotors. The purpose of this locking system is to provide a degree of safety against inadvertent startup of the unit by assuring that the primary turbine control device remains closed, even upon loss of hydraulic control pressure. A hydraulic cylinder typically operates the servomotor locking system, and the automatic operation of the servomotor locking system can be implemented either within the turbine governing system or within the unit control system. Typically, the automatic servomotor locking system is engaged as part of the shutdown sequence after the turbine control servomotor reaches its fully closed position. The desired method of operation of the automatic servomotor locking system should be clearly specified by either the turbine manufacturer or the owner.

6.3.21 Remote control

The term “remote control” can have several meanings in the hydroelectric generating industry. One common usage of the term “remote control” refers to control of the unit from a point within the powerhouse that is physically removed from the turbine governing system. Another usage of the term remote control refers to control of the unit from an off-site location. For the purpose of this guide, remote control refers to control of the unit from a point within the powerhouse that is physically removed from the turbine governing system. Typically, off-site control of a turbine governing system is implemented through a plant-level controller, and the actual control commands to the turbine governing system are issued from the plant controller to the unit control system.

Some typical functions within the turbine governing system that are controlled remotely are as follows:

a) Speed reference (speed changer)
b) Generation setpoint (load changer)
c) Servomotor limit
d) Permanent speed droop
e) Speed regulation
f) Automatic generator braking system
g) Governor gain settings

6.3.22 Remote indication

The turbine governing system may provide electrical outputs signals to drive remote-indicating instruments to indicate operating conditions of the unit. Some typical remote indication outputs that may be provided by a turbine governing system are as follows:

a) Servomotor position
b) Servomotor adjustable limit setting
c) Speed reference (speed changer) setting
d) Generation reference (load changer) setting
e) Permanent speed droop setting
f) Speed regulation setting
g) Unit speed
h) Turbine start/stop status
i) Unit creep
j) Governor hydraulic pressure
k) Pressure accumulator tank fluid level
l) Governor controller error
m) Coincidence of servomotor position with adjustable limit setting

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6.3.23 Automatic control

Automatic control of a unit refers to the automatic starting and stopping of the turbine and generator, speed governing of the unit, and automatic control of the governor pressure system. Automatic control may be accomplished by a separate control system for each subsystem, by the unit control system, or it may be integrated into the turbine governing system. A turbine governing system or a unit control system that uses a digital controller can integrate these automatic control features into the application program for the controller (see IEEE Std 1249-1996 [B27]).

6.3.24 Electrical control power

Electrical control power for a turbine governing system should be from a reliable source. Typically, dc power from the station battery is used to provide reliable control power. Control power may also be provided from an uninterruptible power supply (UPS) with sufficient operating time capacity to meet the needs of the station (see IEC 1116 [B22] and IEEE Std 1020-1993 [B25]).

6.3.25 Auxiliary components for pump turbines

Pump turbines require some features that are not generally present on conventional turbines. Some pump turbine control functions that may be specified as part of the turbine governing system include

a) Turbine runner drain valve control
b) Adjustable or multiple gate servomotor rate controls
c) Gate servomotor versus operating head control function
d) Tailwater depression control

6.3.26 Rotor creep detector

A creep detector may be used to determine if the unit starts to rotate while the unit is shut down. The creep detection often initiates some actions to protect the unit's thrust bearings, such as starting the high-pressure lubrication system when the unit is operating below rated speed. A typical creep detection threshold specified is 3° of arc, and the creep detection system is usually enabled during the shutdown process after a complete stop ("deadstop") has been detected. It is important that the creep detector be very reliable and insensitive to normal vibrations present in the powerhouse. A direction-sensitive sensing method (such as multiple proximity pickups) may be used so the specified angular displacement is detected before rotor creep is detected and alarmed. Mechanical detectors that contact the shaft after deadstop may also be used to detect creep.

6.3.27 Fire protection system

The design of a fire protection system is outside the scope of this guide. However, it should be noted that hydraulic fluid is combustible, and therefore, the hydraulic pressure system, the hydraulic system sump tank, and the hydraulic control valve system may be subject to inclusion in the fire protection system. This may include the mounting of sensors and suppression equipment on or near these system components.

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6.3.28 Alarms

The turbine governing may be used to issue alarms for significant failures or abnormal conditions. The following alarm points are commonly provided:

a) Unit overspeed
b) Speed signal failure
c) Unit creep detection
d) Gate limit coincidence switch for blocked gate
e) Governor hydraulic system high and low pressures
f) Governor sump tank high and low oil levels
g) Governor sump tank high oil temperature
h) Governor hydraulic system lag system pump operation
i) Governor power supply failure
j) Generator air brake supply pressure low
k) Wicket gate automatic lock failure
l) Pilot valve strainer obstruction

6.3.29 Spare parts and accessories

Spare parts required for reliable long-term operation of the unit should be given careful thought. Parts that are critical to the operation of the unit should be given the highest priority. The normal lifetime and the lead-time of obtaining parts should be considered when deciding which parts should be stocked. Some parts have a predictable failure or wear-out pattern, and some part failures may be difficult to predict. Also, some parts have a limited shelf life, or they may require special storage procedures.

The availability of parts from suppliers should also be considered. If parts are available from only one supplier, the risk to long-term support of the control system is greater. Long-term support agreements may be made with the supplier, or manufacturing details may be held in escrow at an agreed-on location to assure the long-term availability of spare parts.

Software support for digital control systems is an issue similar to the spare parts issue. If the control system programming language or hardware platform is of a proprietary nature, then the same issues as spare parts availability apply when considering long-term support. Long-term support agreements may be made with the system supplier, and proprietary programming software, source code, and other documentation may be held in escrow at an agreed-on location to assure long-term maintainability of the system’s application software.

6.4 Types of turbine governing system installations

Turbine governing system designs may be applied to varying situations. Each application situation requires consideration of different system constraints in the design of the turbine governing system. Some common application situations are discussed in 6.4.1 – 6.4.3.

6.4.1 New installations

New installations, where there is no existing turbine or turbine governing system, require coordination of the turbine design, the system performance requirements, and the design of the turbine governing system.

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6.4.2 Replacements

Replacement of the turbine governing system requires the coordination of the turbine governing system with the system performance requirements and the design of the existing turbine. The replacement turbine governing system may either interface to the existing turbine control servomotor or it may include a replacement of the turbine control servomotors. The replacement turbine governing system may also include a replacement of the hydraulic pumping unit, if the replacement turbine governing system uses a hydraulic turbine control actuator. The designs of the replacement turbine control actuator and the replacement hydraulic pumping unit must be coordinated with the requirements of the turbine to assure acceptable system performance.

6.4.3 Conversions

Conversion of a turbine governing system generally includes replacement of the governor controller (also called the "governor head"), along with an interface to the existing governor control valve. Generally, the hydraulic pumping unit is either reused (if it is in acceptable operating condition) or it may be refurbished to its original operating condition. It is important to note that if the conversion of the turbine governing system coincides with the replacement of the turbine runner, the requirements of the new turbine should be carefully evaluated to assure that the existing turbine control actuator and hydraulic pressure unit will be able to perform acceptably with the new turbine.

7. Performance specifications

There are many ways to specify the performance of a turbine governing system. The intended method of operation of the unit determines which performance specifications are applicable in any particular instance. Some of the more common performance specifications for hydroelectric turbine governing systems are discussed in 7.1 – 7.3.

7.1 Stability

The stability of a power system refers to its ability to remain in a state of operating equilibrium under normal operating conditions and to regain an acceptable state of equilibrium after being subjected a disturbance. During the disturbance, and its recovery from it, the system is subjected to dynamic oscillations of frequency and voltages (see Kundur [B32] and Ramey and Skooglund [B36]). It is often stated that when a generator is connected to a large power system, its governor no longer controls speed, but it controls power. Although this is true from the perspective of the individual unit operation during normal conditions, the composite speed governing of all of the units connected to the system is the primary control that limits the magnitude of a system frequency deviation after a disturbance.

As discussed in 4.29, in large interconnected systems, the AGC system maintains the overall steady-state frequency and the net tie-line power exchange between control areas in the interconnected system. Generation and frequency control is then performed on a decentralized basis by AGC, and each control area tries to maintain frequency and its scheduled interchange of power. However, after a disturbance, such as a large unit or major load tripping out, the AGC is designed to allow governors to operate first. Thus, the frequency deviation causes the governors to respond and restore the frequency in accordance with the permanent speed droop settings of the units. The AGC action subsequently trims out any remaining error within the interconnected power system. When the power system undergoes a major cascading disturbance that may split up the system into electrical "islands," speed governors are the only control devices that can control and maintain the frequency within acceptable deviations. Therefore, governors of key facilities in potential system islands should be adjusted to enable generator operation under islanded conditions, assisted by load or generation special tripping schemes to maintain generation/load balance, allowing the system governors to control the frequency (see Hovey [B13], Murty and Hariharan [B34], Schleif and Martin [B39], and Schleif and Wilbor [B40]). During these disturbances, generator excitation systems maintain the system voltages within their specified limits.

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Electrical loads connected to the power system generally consist of dynamic (motor) loads and static (lighting and heating) loads. These differing load characteristics have an impact on the damping of dynamic frequency oscillations. The effective composite load of the system often varies significantly with the time of day, and seasonally. One approach to achieving system frequency stability is to model the governing response of a unit assuming it to be a single unit connected to a single load in an isolated system (see 7.1.3.2 for a discussion on load impact). In such a model, the effects of the generator, excitation system, power system stabilizer (PSS), and transmission system are neglected. Effectively, this is similar to a generator with a high gain voltage regulator maintaining constant terminal voltage regardless of speed, and the resistive load is in essence a constant power load when connected to an isolated generator. An inertialess load gives the smallest system inertia constant (only the generator provides system inertia) and, therefore, the fastest rate of change of speed for a generation/load imbalance (see the IEEE Working Group Report [B31]).

The isolated load usually has a component that is sensitive to frequency, but a conservative modeling approach generally uses the inertialess, resistive constant-power load to achieve a robust solution.

As power is torque multiplied by speed, a resistive load contributes negative damping torque to oscillations in speed. That is, when speed increases, the electrical (braking) torque of the load on the generator decreases. This tends to result in a further increase in speed. Governor modeling using the isolated system, constant power load model thus presents a conservative modeling approach that is widely used. Governor responses using such an isolated model is presented in Annex E, and the examples in this annex show the effects of varying the various parameters in the model.

Annex E also provides comparisons of governor and system frequency responses to a small, interconnected system model that includes generators and excitation systems. All such models are at best an approximation, and in reality, frequency oscillations in the system are invariably accompanied by voltage oscillations that result in fluctuating loads and complex interactions with other control systems such as excitation systems and PSS of units in the system. Nonlinearites in the system also play a large part in the differences between model simulations and real-time recordings of system events. The overall objective in modeling is to make sure that sufficient damping is available for oscillations, and if oscillations do persist, an overall coordinated review of all control systems including governors should be performed.

7.1.1 Sustained condition

Two measures of the stability of a turbine governing system are the steady-state governing speed band and the steady-state governing load band. These are also called the “speed stability index” and the “power stability index.” When synchronized to a large interconnected power system, a unit’s contribution to the grid stability is often measured as its “speed regulation” parameter.

7.1.1.1 Speed stability index

The speed stability index (also called the “steady-state governing speed band”) is the peak-to-peak variation in unit speed, expressed as a percentage of rated speed, that is caused by the turbine governing system when the unit is operating disconnected from an interconnected power system and connected to a constant local electrical load. Ideally, the unit under test should be operated both under governor control and with a blocked gate position to determine how much of the speed variation is due to water turbulence and similar non-governor causes, and how much is caused by the turbine governing system. The difference in the peak-to-peak speed deviations under free governor operation and under blocked gate operation is the traditional governor speed stability index. However, the overall speed stability index under governor control may be used as a measure of the governor’s ability to maintain a steady speed. The effectiveness of different governor tuning parameters can be determined by comparing the overall speed stability index with different governor parameter adjustments. A speed stability index of 0.3% with the unit operating at 5% permanent speed droop has historically been specified for hydroelectric turbine governing systems. A smaller speed stability index can generally be achieved by using more advanced control systems and higher control gains.

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However, achieving a smaller speed stability index may also result in increased control action of the turbine control devices. Weighing the benefits of a smaller speed stability index against the additional wear caused by increased control action depends on the needs of each individual site.

Measuring the speed stability index requires operating the unit or units into a load that is isolated from an interconnected power system. In many cases, this arrangement may be difficult or impractical to implement. Because of inherent hydraulic transients in the turbine, the speed stability index may not always be a meaningful index of governing performance. These limitations are discussed in 9.2.7. Therefore, most speed stability index measurements are performed with no load connected to the generator as a measure of the governor's ability to maintain a steady speed for synchronizing the unit to the interconnected power system.

7.1.1.2 Power stability index

The power stability index (also called the "steady-state governing power band") is the peak-to-peak variation in unit power generation, expressed as a percentage of rated power generation, that is caused by the turbine governing system when operating connected to a large interconnected power system with a constant turbine governor setpoint. To determine how much of the measured peak-to-peak power deviation is caused by the turbine governing system, and how much of this deviation is due to water turbulence and similar non-governor causes, the unit should be operated in both the blocked gate position mode and in the free governor mode. The difference in the peak-to-peak power deviations under free governor operation and under blocked-gate operation is the power stability index. A power stability index of 0.4% of rated power with the unit operating at 5% permanent speed droop or speed regulation is commonly specified for hydroelectric turbine governing systems.

7.1.1.3 Contribution to interconnected grid stability

The amount of contribution that a given generating unit provides to the overall frequency regulation of an interconnected power system is determined both by the speed regulation ("power droop") characteristic and by the transient response of the turbine governing control system. The governor setpoint, the speed regulation characteristic, and the system frequency determine the operating point of the unit (see Dandeno et al. [B8] and Undrill and Woodward [B43]). The contribution of the speed regulation characteristic to the unit operation is expressed by Equation (5):

$$P = \frac{100 \times F}{R_s} \tag{5}$$

where

P is the change in unit power generation, % rated generation,

F is the change in system frequency, % rated frequency,

R_s is the unit speed regulation characteristic, %.

On an interconnected power system, the speed regulation characteristic determines how much each unit's power changes in response to a system frequency change.

The transient response of the integrated power system to both load and generation changes is determined by the aggregate sum of each speed governing system's individual transient response. A common approach to tuning individual speed governors is to ensure that the individual unit is stable and well damped for isolated load changes. In this way, the individual unit contributes positive damping to the integrated power system. Some common approaches to tuning turbine governing systems are described in more detail in Annex F (see Schleif and Angell [B38]).

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7.1.2 Load rejection

The maximum unit speed after a load rejection is a function of gate closing time, unit rotating inertia, and water column inertia. The maximum permissible speed rise is determined by the mechanical characteristics of the turbine-generator design (typically 140–190% of rated speed for Francis units, and up to 270% for many adjustable-blade type units). The maximum permissible unit speed after a load rejection is generally determined by the construction of the generator rotor.

7.1.3 Speed control, fluctuating isolated load basis

Speed control for an isolated load is generally defined as the maximum deviation from nominal frequency, caused by changes in the power demanded by the isolated load. Normally, the level of regularly occurring load changes are identified during the design process, and simulation studies conducted to determine the resulting frequency changes with various amounts of generator inertia, water column inertia, and gate operating times, to verify that the frequency deviations can be maintained within the desired limits.

7.1.3.1 Islanded performance

Islanded operation is similar to isolated operation, with the exception that the frequency of the islanded system is impacted by the performance of all of the interconnected generating units and loads within the "island" of operation. Generally, if all interconnected units within the island of operation are tuned for good governing stability under isolated conditions, then they will also perform well as an islanded group.

7.1.3.2 Impact of connected load

The nature of the load connected to an isolated generating unit impacts the ability of the turbine governing system to control the output frequency of its generator. A motor-driven load with high inertia could add stability to a hydro governed system by increasing the ratio of the total rotating inertia to the water column inertia. However, it has also been noted in system-wide stability studies of small oscillations that, depending on the interconnected system configuration, system stresses, and spread of generation and load in various parts of the system, induction motor loads can negatively influence the system stability. An inertialess load, such as resistive heaters, adds no equivalent rotating inertia to the system. The load torque reflected from an electrical load to the turbine shaft varies with frequency for most types of loads. For a resistive heater load, the load torque imparted on the turbine decreases as frequency increases. This type of load has a destabilizing influence, and it places a greater demand on the turbine governing system in maintaining the system frequency. For motor-driven fans and compressors, the load torque imparted on the turbine generally increases as frequency increases. This type of load thus has a stabilizing influence. Certain types of motor-driven positive-displacement pumps tend to impart no change in load torque on the turbine shaft with frequency changes. Electrical loads are generally a combination of each of these types of individual loads, so the composite sensitivity of the load torque to frequency changes determines the net stabilizing or destabilizing effect of the connected load on the governed frequency stability. The composite characteristics of a connected load can change seasonally in any given area, so the ideal tuning and performance of the turbine governing system changes accordingly. For example, the ideal tuning of a turbine governor for an electrical system that has air conditioners as its dominant load (such as may occur during the summer months) can be significantly different from the ideal tuning of a turbine governor for an electrical system that has resistive heaters as its dominant load (such as may occur during the winter months).

7.1.3.3 Interaction with excitation system

A poorly adjusted governor can cause or worsen a local prime-mover oscillation involving its water column, in turn influencing interarea system oscillations. Equally, a poorly adjusted automatic voltage regulator (AVR) or PSS can lead to a similar overall stimulation of oscillations in machine response to the interconnected power system (see Bollinger and Gu [B2], Bollinger and Nettleton [B3], and DeMello and Concordia [B10]).

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For operation with the generator connected to an isolated electrical load, a modern static exciter can be tuned to respond quickly to maintain a fairly constant terminal voltage at the generator. Comparatively, older rotating exciters are often much slower to respond, resulting in changes in terminal voltage at the generator as a result of either changes in system load or changes in generator speed. With a slow-acting exciter, the dynamic changes in generator terminal voltage as a result of changes in generator speed can actually have a stabilizing influence on the governed speed of the unit. This stabilizing influence is the result of the electrical braking effect of the load as a function of the transient speed-dependent voltage output from the generator. For example, an increase in generator speed results in a transient increase in generator voltage. This increase in voltage generally results in an increase in the reflected torque demand from the load, which tends to limit the rise in speed. There have been instances where an excitation system has been upgraded to incorporate a new, more responsive AVR, and the loss of the speed-stabilizing effect of the old slow-responding AVR has reduced the achievable speed stability of the turbine governing system when operating into an isolated electrical load.

In the tuning process of governors and power system stabilizers for hydroelectric generating units, there is usually no significant interaction between the two control systems. The governors are adjusted to perform proper stabilization of the frequency of an isolated system, along with adequate step-load response. Power system stabilizers are adjusted to compensate for the phase lag in the excitation system and generator characteristics to cause an increase in the electrical damping torque component of the generator.

There has not been much discussion in the technical arena on the interdependence or interaction between the respective adjustments and control functions of hydro governors and power system stabilizers. However, the large differences in the time responses between the two control systems are generally recognized (see the IEEE Working Group Report [B30]).

Studies to control power swings arising from hydro turbine draft tube surging, utilizing pressure and electric power signals as inputs to the PSS, indicate that only a limited damping effect can be achieved. No change in the original surging was noted. Only a reduction in the power system swings in reaction to the surging was observed (see Frizell and Agee [B12]).

The desired action of the PSS is to cause a component of electrical damping torque in the generator to damp the power swings of the machine connected to the power system. In certain cases, it also counteracts the negative damping that may arise in high-gain AVRs (see DeMello and Concordia [B10]). The hydroelectric unit's governor remains in its historical role of controlling the steady-state frequency and low-frequency (<0.1 Hz) disturbances by damping water column energy oscillations and turbine torque oscillations associated with low-frequency power swings. Because the electrical time constants associated with the generator excitation system are generally much shorter than the mechanical time constants associated with the water column and rotating elements of a hydroelectric generating unit, the PSS can quickly damp out the local-mode and interarea power swings in the power system without interaction with the governor.

7.1.4 Stability studies

Computerized simulations of turbine governing systems may be used to predict the performance of the system under the expected operating conditions. By properly simulating the turbine, the water passages, the generator, the turbine governing system, and the connected electrical load, the performance of the actual system can be predicted within a reasonable tolerance. If simulated stability studies are done early enough in the design process of the hydroelectric generating unit, there may be time to revise the design of one or more constraints to achieve the desired performance level. Simulated stability studies can also be used to predict the ideal tuning of the turbine governing system. This greatly reduces the amount of time required to tune the unit in the field.

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7.2 Permanent speed droop and speed regulation

The desired settings of the permanent speed droop or speed regulation are generally dictated by the needs of the systems operation. Generally, a value of 5% permanent speed droop or speed regulation assures adequate participation of all units on a large interconnected system. Lower values of permanent speed droop may be necessary to stabilize the frequency of a smaller interconnected system.

7.3 Deadband

The deadband inherent in a turbine governing system determines the minimum speed deviation that causes a response from the governing system. The primary contributors to the deadband of a turbine governing system include the resolution of the speed measurement system and the overlap of the governor's servomotor control valves. Typically, a maximum deadband value of from 0.01% to 0.02% has provided acceptable performance for hydroelectric generating units. Larger values of deadband may result in acceptable performance for some units, but they are not able participate in supporting the system frequency for the smaller frequency deviations.

For adjustable-blade turbines, the blade positioning deadband determines the accuracy at which the blades follow the ideal blade position setpoint for achieving maximum turbine efficiency. Typically, a blade control deadband is specified to be no greater than 1% of full blade servomotor stroke. Increasing the blade control deadband reduces the average efficiency of operation of the turbine.

7.4 Deadtime

The deadtime of the turbine governing system determines the amount of time that is required after a disturbance in unit speed before the turbine governing system makes a corrective action. Increasing the deadtime of a turbine governing system reduces its contribution to the stability of the interconnected system. Increasing the deadtime of the turbine governing system can also introduce a resonance in the system that falls within the normal controlling spectrum of the turbine governing system. This can result in violent oscillations of the turbine control servomotor under certain conditions. The primary contributors to the turbine governing system deadtime include the update time of the governor controller and the amount of overlap in the servomotor control valves. Typically, a maximum governing system deadtime value of 0.2 s has provided acceptable performance for hydroelectric generating units.

7.5 Range of governor speed changer adjustment

The range of the governor speed changer adjustment (also known as the governor speed reference, or speed setpoint) should be sufficient to synchronize the unit to the interconnected system at the lowest expected system frequency, and to fully load the unit when it is connected to the system at normal frequency. Typically, the range of governor speed changer adjustment is specified to be from a minimum setting of 85% to 90% of rated speed to a maximum of 110% of rated speed.

7.6 Manual control

Manual control is used to position the turbine control servomotors to the position setpoint without any influence from the normal governing parameters of speed, head, or generation. The manual control mode may be implemented in several ways, each with an impact on the security and flexibility of operation of the system. Manual control is often used to operate the turbine control servomotors during commissioning and testing of units. The manual control mode may also be used as a default mode of operation to keep a unit generating online when certain failures occur. Adequate protective systems should be functional when a unit is operating online in the manual control mode.

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7.6.1 Fully independent manual control

A fully independent manual control mode of operation implies that the manual control can operate the turbine control servomotors without the governing controller in operation. This mode can be accomplished with either mechanical-hydraulic or electrohydraulic actuators. If an electrohydraulic actuator uses electronic servomotor position feedback when in the manual control mode, the feedback device and the control amplifier should be separately powered so this mode of operation can be fully independent from the governor controller operation.

7.6.2 Selectable manual control mode

The manual control mode of unit operation can be a selectable option that is integrated into the governor controller. To operate using this implementation of manual control, the governor controller needs to be operational. All of the normal inputs required for governing operation do not need to be functional.

7.7 Turbine control servomotor time adjustment

For the safe operation of the hydroelectric generating unit, the speed of the turbine control servomotors should stay within the limits established either by the turbine manufacturer or by the field testing of the unit. For the primary turbine control servomotors that control the water flow through the turbine, pressure transients due to the water inertia effects need to be balanced against the limitations on the maximum-allowable overspeed imposed on the unit.

7.7.1 Controlling maximum turbine control servomotor rate

Turbine control servomotor time can be expressed in several ways. One common convention is in seconds required for full servomotor travel from 0% stroke to 100% stroke (or vice versa). Another way to express the servomotor timing is as a rate, commonly given in percent per second. Discussions of various ways to control the maximum rate of turbine control servomotor travel follow in 7.7.1.1 – 7.7.1.3.

7.7.1.1 Distributing valve travel limit

One method of setting hydraulic servomotor timing is by using a mechanical limitation of the travel of the main distributing valve plunger. This limitation is often done via adjustable stops, also called “stop nuts” that limit both the opening and the closing flow rate to the turbine control servomotor. This method of timing is relatively easy to adjust, and the opening rate can be set independently from the closing rate. With this approach to servomotor timing, the total available distributing valve control porting should be large enough to allow the servomotor to travel slightly faster than the maximum specified servomotor rate. This allows adjustment of the servomotor timing, using the stop nuts, to achieve the specified rate. One benefit of using distributing valve stop nuts for setting servomotor timing is that the stop nuts have no effect on the dynamic control of the servomotor until the distributing valve plunger contacts the stop nuts.

7.7.1.2 Rate-limiting orifice

Another method of controlling servomotor travel rate is by using a rate-limiting orifice. This method is implemented by placing an orifice in series with the oil flow to the servomotor. This rate-limiting orifice can be on a separate orifice plate inserted between two mating pipe flanges in the servomotor control piping. The rate-limiting orifice can also be incorporated into the structure of the distributing valve. Depending on which approach is selected, separate timing rates for the opening and closing directions may or may not be achievable. Using orifices to determine the maximum servomotor rate results in a simple and very reliable timing limitation. If the timing orifice is located at the turbine control servomotor, it can also provide some assurance that the specified maximum servomotor rate cannot be exceeded in the unlikely event of a rupture of a servomotor control line. However, making changes to the servomotor timing orifice during the

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commissioning of the unit is much more time-consuming than when using adjustable stop nuts. Another consideration when using a timing orifice is that the timing orifice has some dynamic effect on the servomotor positioning system at all flows, which may impact the responsiveness and settling time of the turbine control servomotor positioning system. For the timing orifice to be effective, the available distributing valve porting should be significantly larger than when using stop nuts. This results in less achievable resolution for moving the turbine control servomotor at an intermediate rate of travel.

7.7.1.3 Flow control valves

The use of adjustable flow control valves is yet another way to control the maximum servomotor rate. These flow control valves are adjustable and can be arranged to allow separate opening and closing maximum rate adjustments. Temperature-compensated flow control valves are available, which keep the maximum servomotor rate fairly constant over a wide range of oil temperature. Due to cost issues, flow control valves are generally only used on relatively small servomotor systems. Flow control valves have some impact on the dynamic performance of the servomotor positioning system at all flow rates, similar to the rate-limiting orifice approach to servomotor timing.

7.7.2 Servomotor loading

The mechanical loading imposed on the turbine control servomotor can affect the maximum achievable servomotor rate. The mechanical loading experienced by the turbine control servomotor is the summation of the frictional loading of the devices connected to the servomotor and the loading created by the impingement of the water passing through the turbine. The loading due to water forces is greater when the flow is changing than when maintaining a constant flow through the turbine. The mechanical force available from the turbine control servomotor should be sufficient to overcome these forces and move at the prescribed maximum rate. Experience has shown that when the maximum servomotor rate is set by orifices or by limiting the travel of the distributing valve, the maximum loading of the servomotor should not result in a differential pressure across the servomotor of greater than 70% of the minimum hydraulic supply pressure. When using flow control valves to set the maximum servomotor rate, a slightly higher maximum differential pressure can be experienced before the servomotor loading affects the servomotor timing. If the servomotor loading is increased beyond the 70% criterion, the travel rate of the servomotor is decreased. If the loading is increased to near 100%, the servomotor stalls and the unit cannot be controlled beyond this point. Depending on the nature of this stalling of the servomotor, the results can include a catastrophic runaway of the unit.

The mechanical loading of a turbine control servomotor is reflected as a differential pressure across the servomotor piston. The difference between the operating hydraulic pressure and the servomotor loading pressure includes the pressure drops both in the servomotor piping and in the servomotor timing device. Figure 20 illustrates the relationship between the servomotor loading and the servomotor timing. Note that for servomotor loadings above 70% of nominal system pressure, the servomotor timing begins to rapidly increase. Also note that a typical pressure accumulator tank using a float valve has approximately 75% of nominal system pressure at the point where the float valve closes. Therefore, if the maximum servomotor loading is always less than 70% of the nominal system pressure, the turbine governing system is able to close the turbine control servomotor under all operating conditions without stalling. At operating pressures below the nominal system pressure, the servomotor travel rate is slower than the specified rate, but the servomotor is still able to move, without stalling, to safely shut down the unit.

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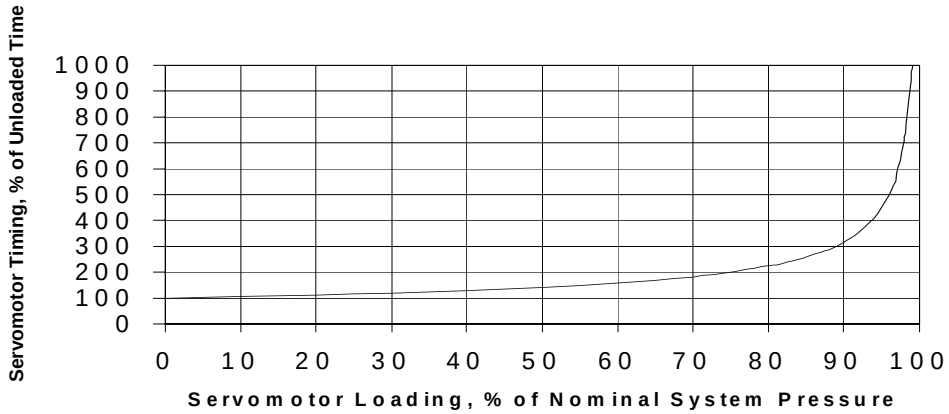


Figure 20—Effect of servomotor loading on servomotor timing at nominal system pressure

For existing units, an increase in servomotor loading is often the result of friction in the connecting linkages or by wear in the associated bushings or bearings. Proper lubrication or replacement of worn parts is likely to solve this problem. For new or replacement units, the problem of excessive servomotor loading may also be due to excessive water loading imposed on the control device of the turbine. This water loading is a function of both the operating head and the flow through the unit. Overcoming this type of servomotor loading problem may require significant changes in the servomotor and turbine system design. Solutions to this problem may include increases in the hydraulic system pressure, changing the servomotor sizing, or changing the pump sizing.

7.8 Governor damping adjustments

The response of the turbine governing system should be tuned to match the rotating inertia, the water column inertia, the turbine control servomotor timing, and the characteristics of the connected electrical load. To achieve the desired performance from the turbine governing system, the damping adjustments should have sufficient adjustment range to provide the proper response and control system damping. Annex F discusses in more detail several approaches that have been used to successfully tune turbine governing systems.

7.8.1 Temporary droop governor adjustments

The primary damping adjustments for a temporary droop type of governing controller are the temporary droop (b_t) and the time constant of the damping device (T_d). A typical range of adjustment for the temporary droop is from 0% to 150%. A typical range of adjustment for the time constant of the damping device is from 0 s to 30 s. The actual range of adjustment needed for any particular installation may vary from these guidelines.

7.8.2 PID governor adjustment

The primary damping adjustments for a PID type of governing controller are the proportional gain (KP), the integral gain (KI), and the derivative gain (KD). A typical range of adjustment for the proportional gain is from 0 to 20. A typical range of adjustment for the integral gain is from 0 per second to 10 per second. A typical range of adjustment for the derivative gain is from 0 s to 5 s. The actual range of adjustment needed for any particular installation may vary from these guidelines.

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8. Information to be provided by the manufacturer

For a turbine governing system to be maintainable over its useful life, the manufacturer of the system should provide adequate information about the hardware and, if applicable, the application software.

8.1 Information to be provided at the time of submission of proposals

At the time of submission of a proposal for a turbine governing system, the manufacturer should provide functional diagrams of the proposed system, a complete description of operation of the system, outline drawings of the equipment including major dimensions, a clear and complete definition of the scope of supply of the proposed system, and any appropriate control system mathematical models that could be used in dynamic computer simulations of the system performance.

8.2 Drawings

An appropriate submittal schedule of drawings for the turbine governing system may be established as part of the contractual agreement for supplying the turbine governing system. Generally, arrangement drawings showing confirmed overall dimensions and weights of the components of the system are submitted first. After approval of these submitted drawings, detailed drawings showing foundation requirements, required erection procedures, piping schematics, wiring diagrams, functional descriptions, and schematic diagrams of all components and auxiliary devices in the turbine governing system should be submitted. As part of the contractual agreement between the manufacturer and the purchaser, the drawing submittal schedule, if required, should be completely defined.

8.3 Operation and maintenance manuals

Operation and maintenance manuals should provide adequate information for proper operation of the turbine governing system. Complete information on all components of the system should be provided to fully define the functionality, assembly, dismantling, diagnosis of trouble, reassembly, and operation of each component or assembly that is part of the turbine governing system. Typically, six bound copies of the operation and maintenance manuals are required to be submitted in draft form three months prior to delivery of the turbine governing system. The number of actual copies and their submittal date should be clearly stated as part of the contractual agreement between the purchaser and the supplier of the turbine governing system. Submittal of the operation and maintenance manuals in an electronic format should be considered as an option to a bound paper version of these manuals.

8.3.1 Long-term maintenance

In addition to the operation and maintenance manuals, the purchaser of the turbine governing system should be given documented assurance that the equipment can be maintained adequately over its useful lifetime. There should be some assurances that replacement parts and system support can be obtained over this period of time. The stated normal lifetime of a turbine governing system may vary, but an expected lifetime of 20 years to 50 years is not uncommon. The purchaser may require that the manufacturer of the turbine governing system guarantee that replacement parts for components of proprietary designs be available for this period of time. Proprietary components may include both hardware and software elements of the system. An alternative to this guarantee of availability may be to agree to hold in escrow the necessary machine drawings and software source code to allow maintenance of the system in the event that the manufacturer of the turbine governing system becomes unable or unwilling to provide the necessary parts or support for maintaining the system.

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9. Acceptance tests

Generally, certain tests are specified as a basis for acceptance of the turbine governing system. Some tests are more appropriately conducted under factory or laboratory conditions, and some tests are better done in the field as part of the commissioning and acceptance process.

9.1 Factory acceptance tests

The following acceptance tests are commonly specified to be performed at the turbine governor manufacturer's testing facility. The purchaser may require that a representative of the purchaser be present to witness these tests prior to agreeing to shipment of the turbine governing system equipment to site.

9.1.1 Deadband test

A factory deadband test is often specified to verify that the turbine governing system meets the speed deadband specified for the system. Generally, this testing should be done in accordance with the procedures outlined in ASME PTC29-1980 or in IEC 60308.

9.1.2 Deadtime test

A factory deadtime test is often specified to verify that the turbine governing system is capable of responding to a frequency change within the specified deadtime. Generally, this testing should be done in accordance with the procedures outlined in ASME PTC29-1980 or in IEC 60308. Shop testing is not generally done with full-size turbine control servomotors, so the results obtained from factory testing of deadtime may vary from the results obtained in the field. The factory testing of deadtime, however, demonstrates the turbine governing system's capability of responding within the specified deadtime.

9.1.3 Gain verification test

A gain verification test may be specified to establish that the settings of the compensating gains within the turbine governing system represent the actual response of the governing equipment. The actual gains tested, and the method used to test them, depends on the functional structure of the governing controller.

9.1.4 Transient immunity test

To assure that the turbine governing system continues to work reliably in the presence of electromagnetic interference normally experienced in the field, a transient immunity test may be specified. These tests may require the demonstration of undisturbed operation in the presence of operating handheld radios such as may be used in the field. The turbine governing system may also be required to be tested according to IEEE Std C37.90.1™-2002 [B28] to demonstrate its surge withstand capability. Other testing standards consistent with the expected field-operating environment may also be specified.

9.2 Field acceptance tests

The following tests are often specified by the purchaser to be performed at site prior to acceptance of the turbine governing system equipment for commercial operation. The results of these tests are generally documented and included in the commissioning report for the unit.

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9.2.1 Servomotor timing test

The turbine control servomotors are generally tested to verify their rate of travel, usually expressed as the equivalent time for 100% servomotor travel. Generally, this testing should be done in accordance with the procedures outlined in ASME PTC29-1980 or in IEC 60308. The servomotor timing is usually tested and adjusted before the unit is watered up, and tested and readjusted again (if necessary) after the unit is watered up.

9.2.2 Upset stability test

An upset stability test generally consists of running the unit off-line at rated speed, and then disturbing the speed by a specified amount using the gate limit or some equivalent means. The stability of the unit is judged by observing the pattern by which the unit returns the unit speed to its rated value. Governor damping adjustments are used to achieve the desired governing response. Figure 21 shows a speed response from an upset stability test with a damping that is generally considered desirable.

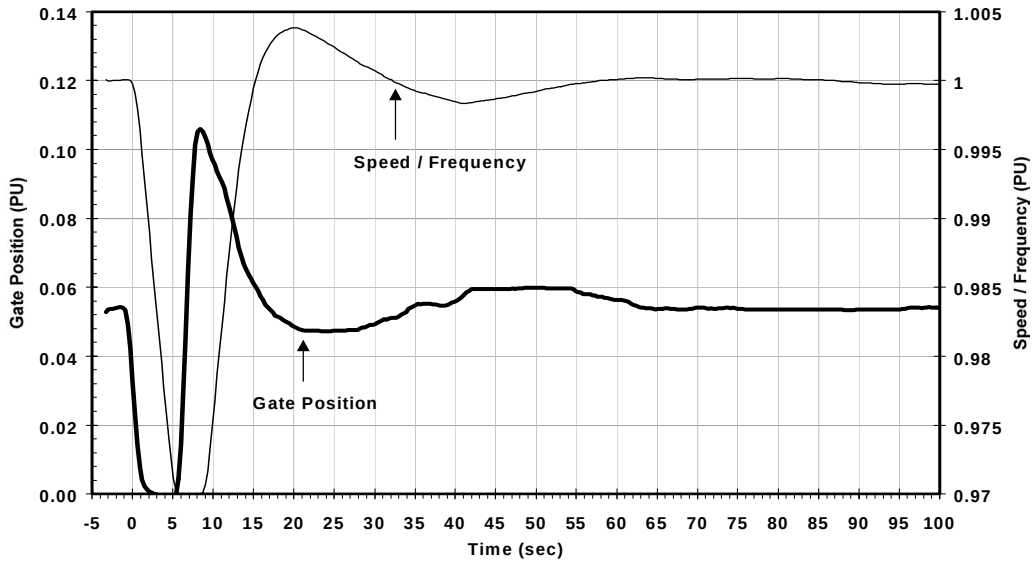


Figure 21—Upset stability test, desirable response

Figure 22 shows a speed response from an upset stability test that is generally considered underdamped.

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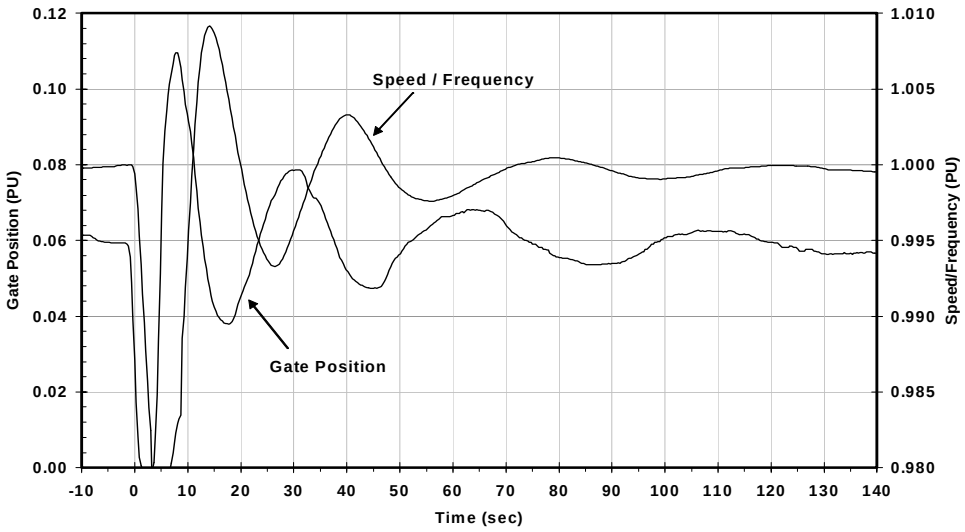


Figure 22—Upset stability test, underdamped response

9.2.3 Load rejection response test

A load rejection response test is performed to judge the turbine governing system's ability to return the unit to stable speed control and to evaluate the pressure rise in the penstock or intake water passage after a load rejection. This test is generally done by adjusting the unit generation into the interconnected system to a specified level and then by tripping the unit breaker. Several levels of load rejection may be specified for commissioning a new hydroelectric generating unit, whereas only one level of load rejection may be considered sufficient to determine acceptability of a new or modified turbine governing system. Generally, the unit speed, intake water passage pressure, and the turbine control servomotor position are recorded during this test to record the unit's response to the load rejection. The turbine control servomotor timing is used to achieve an acceptable peak transient overspeed and pressure rise, and the governor damping adjustments are used to achieve the desired settling characteristics of unit speed. Figure 23 shows a typical load rejection response test.

It is important to note that the control strategy of a turbine governing system often activates a "partial shutdown" function upon the tripping of the unit breaker. This partial shutdown function automatically positions the gate limit to a position slightly above the no-load gate position. By activating the partial shutdown function, the turbine governing system assures that, in response to a load rejection, the gates are driven downward at the maximum achievable gate servomotor rate to limit the maximum transient overspeed to the lowest possible value. To evaluate the response of the turbine governing system to a load rejection, the load rejection response test may require that the partial shutdown function be disabled. This approach identifies if the damping adjustments of the turbine governing system may cause the wicket gates to reopen prior to bringing the unit speed down to rated speed after a load rejection. This reopening of the wicket gates may cause an unacceptably long settling time for the turbine governing system to return the unit to its rated speed after a load rejection, in the event that the partial shutdown function fails to operate for some reason.

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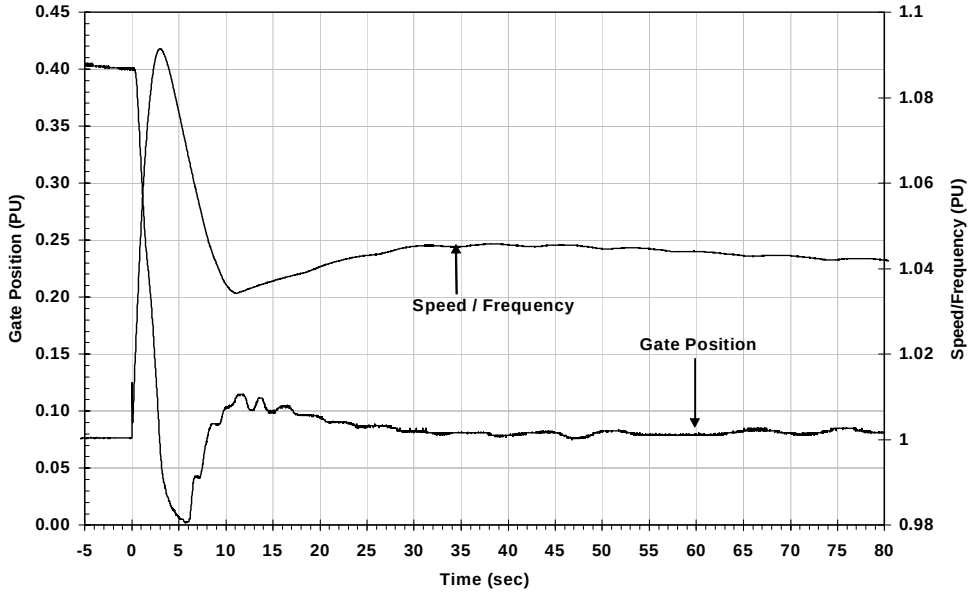


Figure 23—Load rejection response

9.2.4 Online generation response test

The online generation response test is used to judge the responsiveness of the turbine governing system to a setpoint change when synchronized to the interconnected power system. The online generation response of a unit is important for coordinating the action of the AGC system among units connected to the interconnected power system. The online response is generally specified as the time required, from the initiation of the setpoint change, for the unit generation to settle within a specified band of the setpoint.

If the turbine governing system uses setpoint feedforward for online loading response, this feature should be enabled and tuned to achieve the desired online response. If the turbine governing system uses a separate set of governor damping adjustments (e.g., online gains or dashpot bypass) for online loading response, these online parameters should be enabled and tuned to achieve the desired loading response. Generally, the turbine governing system setpoint and the unit generation are recorded for the online response test. Figure 24 shows a typical on-line response test.

9.2.5 Online servomotor response test

The online servomotor response test is used to judge the responsiveness of the turbine control servomotor to a setpoint change while the unit is synchronized to the interconnected power system. The online servomotor response is generally specified as the time required, from the initiation of the setpoint change, for turbine control servomotor to settle within a specified acceptance band of the setpoint.

Generally, the turbine control servomotor setpoint and the turbine control servomotor position are recorded for the online servomotor response test. Figure 25 shows a typical on-line servomotor response test.

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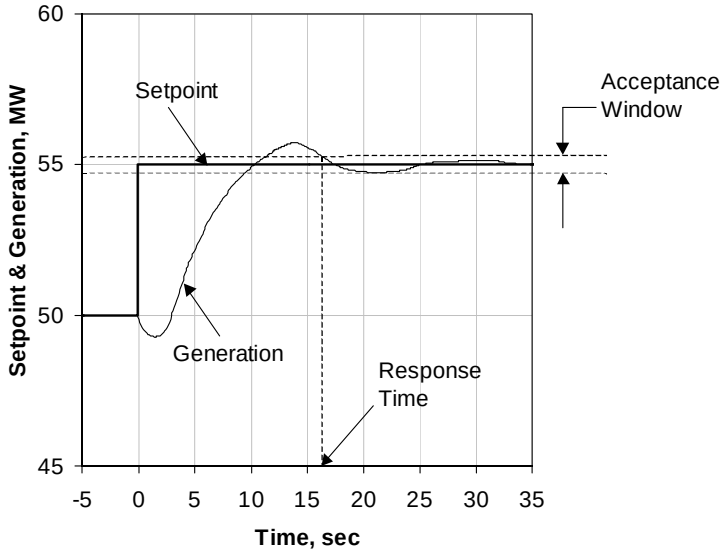


Figure 24—Online generation response test

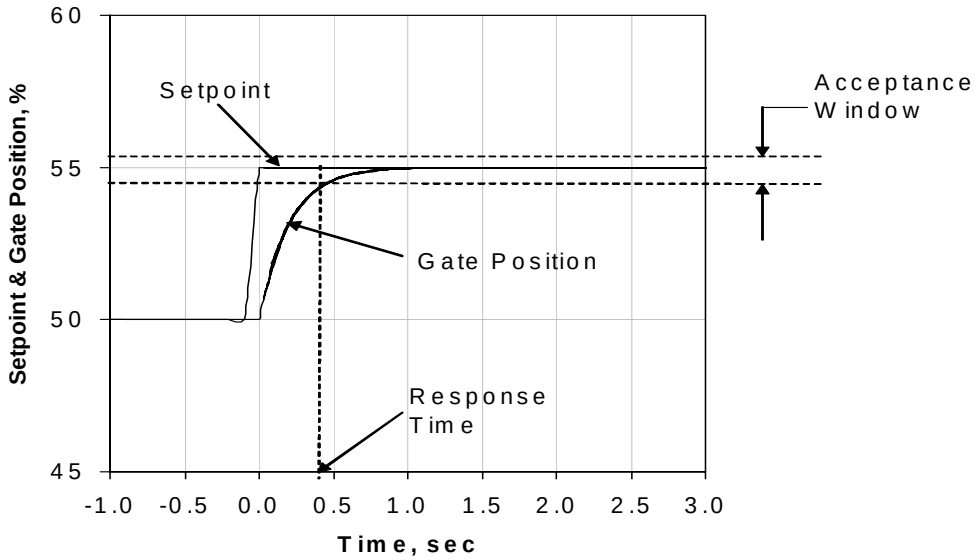


Figure 25—Online servomotor response test

9.2.6 Deadtime test

A field deadtime test may be specified to verify the deadtime of the turbine governing system using the actual turbine control servomotors. This test consists of performing a load rejection of at least 10% of the unit's rated generation and of recording the time difference between tripping the unit breaker and the first movement of the turbine control servomotor. Generally, this testing should be done in accordance with the procedures outlined in ASME PTC29-1980 or in IEC 60308.

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9.2.7 Speed stability index test

A speed stability index test may be specified to verify the speed stability index of the turbine governing system while controlling the speed of the turbine. According to IEEE Std 125-1988, this test consists of measuring the peak-to-peak speed deviations of the turbine that are caused by the turbine governing system. This test has historically been done by first recording the peak-to-peak speed deviations while running the unit disconnected from an interconnected power system and connected to a constant local electrical load with the gates limited (not moving). Next, the peak-to-peak speed deviations are recorded while running the unit under the same electrical load conditions while operating under governor control. The difference between these peak-to-peak speed deviation measurements is interpreted as the speed stability index. Figure 26 illustrates a typical speed stability index test, based on the historical interpretation of this index. Refer to 7.1.1.1 for further discussion of this test.

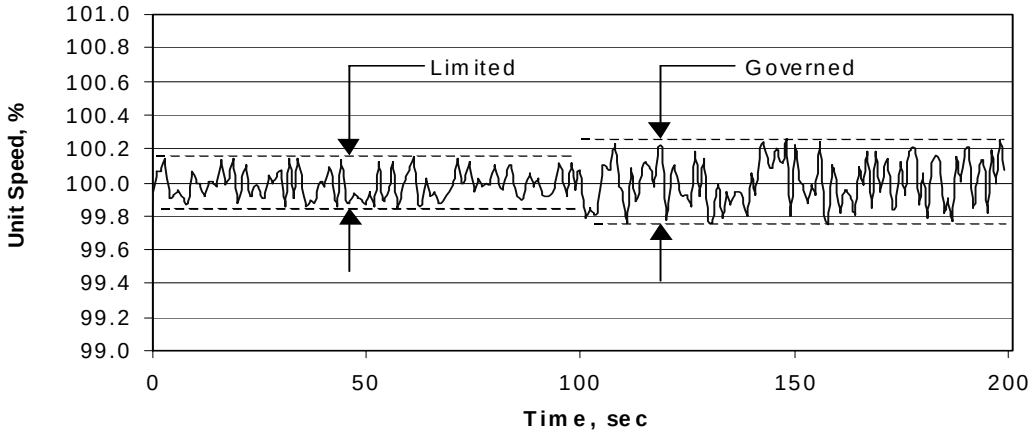


Figure 26—Speed stability index test (historical version)

Note that the speed stability index test measures the combined results of the turbine governing system stability and the inherent turbine stability. It is possible that a modern electronic governor may be tuned so the peak-to-peak speed deviations of a unit under governor control are smaller than the peak-to-peak speed deviations while operating against the gate limit. However, there may be some perturbations in turbine speed that are caused by hydraulic transients within the turbine that cannot be compensated for by the turbine governing system. Because of these limitations, the speed stability index test is not generally considered a useful index of the performance of the turbine governing system. The measurement of the speed stability index is more useful as an indicator that aids in evaluation of different tuning strategies for a particular unit under test.

9.2.8 Power stability index test

A power stability index test may be specified to verify the speed stability index of the turbine governing system while controlling the speed of the turbine. According to IEEE Std 125-1988, this test consists of measuring the peak-to-peak power deviations of the turbine that are caused by the turbine governing system. This test has historically been done by first recording the peak-to-peak power deviations while running the unit connected to an interconnected power system with the gates limited (not moving). Next, the peak-to-peak power deviations are recorded while running the unit under the same electrical load conditions while operating under governor control. The difference between these peak-to-peak power deviation measurements is interpreted as the power stability index. The chart recording documenting the power stability index test is similar to the speed stability index chart illustrated in Figure 26, with the exception that the vertical axis represents generated power rather than unit speed. Refer to 7.1.1.2 for further discussion of this test.

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9.2.9 Simulated speed step test

A simulated speed step test shows the response of gate position to a simulated step in speed (or frequency). This type of test can be used to verify the overall response of the governor system and can be compared with governor model responses to improve model parameters. A typical mechanical governor responds as shown in Figure 27, whereas a typical electrohydraulic governor responds as shown in Figure 28.

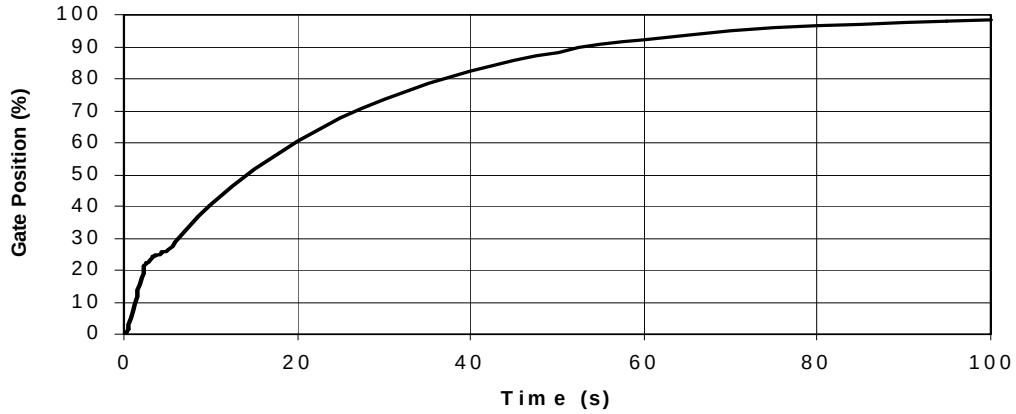


Figure 27—Simulated speed step test for a typical mechanical-hydraulic governor

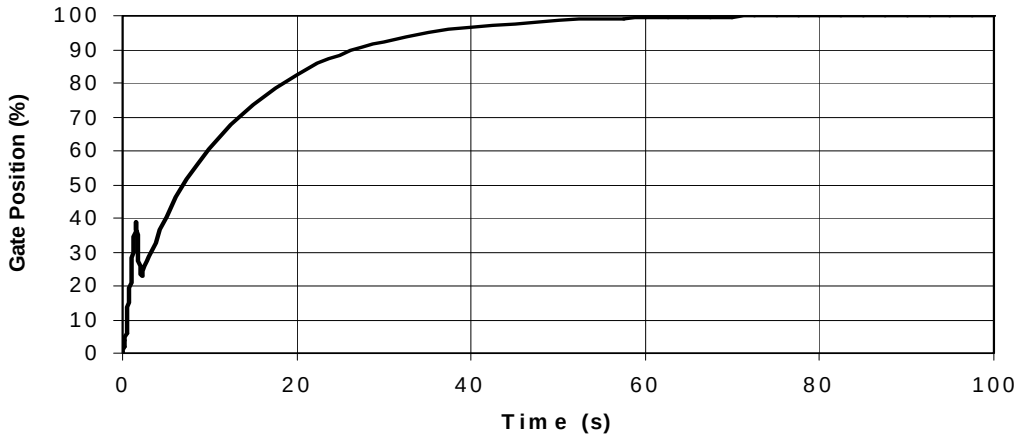


Figure 28—Simulated speed step test for a typical electrohydraulic governor

9.3 Performance auditing

To ensure stable and reliable operation of an interconnected system, power coordinating councils, or similar regulating agencies may require periodic verification of key elements of turbine governing systems. Permanent speed droop, control system damping, dispatch responsiveness, and response to system disturbances are some of the pertinent quantities that may need to be tested and documented to meet the interconnection requirements of the power pool. Annex G discusses some current and proposed testing requirements developed by certain power pool administrators.

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10. Data to be furnished by the purchaser

The following is a representative list of some information that is typically required by the turbine governing system provider to properly design the system. Some of this information may not be needed for a particular application, and some applications may require additional information. The impact on the design of the turbine governing system should be considered when determining what information the purchaser needs to provide.

10.1 Rated turbine output

The rated turbine output is generally specified at the rated head for the turbine. To properly design the instrumentation for the turbine governing system, the turbine output should also be specified for both the maximum and the minimum turbine operating heads.

10.2 Rated head

The rated head is usually the designed net operating head for the turbine. For installations where the operating head is expected to vary over a significant range, the maximum and minimum expected head should also be stated. It is not possible to directly measure the "net head" on a hydroelectric generating unit.

NOTE—The term "net head" refers to the total change in available energy of the water column (potential energy plus kinetic energy) across the turbine runner.

Normally, the "gross head" can be measured, which is the net elevation difference between the water level upstream of the plant and the water level downstream of the plant. Another parameter that is commonly measured is the "piezometric head," which is the operating head measured from pressure taps upstream and downstream of the turbine. The piezometric head measurement automatically compensates for the flow losses in the water passage, but it does not account for "velocity head," which is the change in kinetic energy due to the difference in cross-sectional area of the water passage at the points of the pressure taps. The method of measuring the operating head should be consistent with the data used in the design of the turbine governing system. For example, if a 3D blade controller data map is constructed using net head information from the model testing by the turbine manufacturer, and the actual governor controller receives gross head information from the installed water level transmitters, a significant reduction in the operating efficiency occurs as a result of this discrepancy.

10.3 Rated speed

Rated speed is the rotating speed, generally stated in revolutions per minute (rpm), of the turbine when the unit generator is operating at its rated frequency output.

10.4 Rated discharge

The rated discharge is the flow (in meters cubed/second) through the turbine when it is operating at rated head and rated power output.

10.5 Type of setpoint parameter

The type of setpoint parameter is the primary setpoint to be used by the turbine governing system for operating the hydroelectric generating unit. This setpoint parameter may be in terms of unit speed, gate position, flow, power generation output, or water level. The turbine governing system may have more than one operating mode, each of which may accept a different setpoint parameter.

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10.6 Ambient conditions

The ambient operating conditions should be thoroughly specified so the turbine governing system can operate reliably under these conditions. Generally, the ambient operating conditions should include maximum and minimum limits for the temperature and humidity.

10.7 Seismic requirements

If the geographical location of the hydroelectric generating station is subject to seismic activity, the equipment should be designed to withstand the worst of the expected seismic activity without sustaining significant damage affecting the safety of the equipment or nearby personnel. The degree of expected seismic activity should be specified, and the design calculations for the equipment that document its ability to withstand this activity should be submitted for approval during the design phase of the turbine governing system.

10.8 Surge tank dimensions and type

The surge tank, if part of the water passage design, has a significant impact on the operation of the turbine governing system. Full design details of the surge tank, along with the dimensions of the rest of the water passage, should be provided so the designer of the turbine governing system can evaluate the controllability of the unit.

10.9 Water inertia time

If the purchaser has computed the water inertia time for his system, the results of this calculation may be provided to the manufacturer of the turbine governing system instead of the full dimensional details of the water passages.

10.10 Pressure regulator valve capacity under full head

If a pressure regulator valve is provided in the turbine's water column, its flow capacity should be defined to coordinate its operation with the operation of the turbine control servomotors. It is preferable to provide the flow characteristics of the relief valve under at least the minimum and the maximum head conditions expected at site. Flow characteristics at intermediate head levels should be provided if the range of operating heads is great enough to significantly affect the performance of the pressure regulator valve.

10.11 Unit mechanical inertia

The rotating mechanical inertia of the hydroelectric generating unit has a significant effect on the performance of the turbine governing system. Both the inertia of the generator and the inertia of the turbine should be provided to the manufacturer of the turbine governing system. If the rotating inertia of the turbine is very small compared with the rotating inertia of the generator, which is often the case, providing only the rotating inertia of the generator may be sufficient to meet the needs of the turbine governing system manufacturer.

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10.12 Station ac and dc voltages

The available ac and dc supply voltages should be provided to the manufacturer so the equipment can be designed to operate from these voltages. If more than one power source is to be used to power the turbine governing system equipment, the power supply that is to be used as the primary and the power supply that is to be used as secondary should be specified.

10.13 Powerhouse drawings showing suggested location of equipment

Powerhouse drawings are generally required to determine the dimensional constraints that affect the allowable size for the components of the turbine governing system. Dimensional constraints may exist on either the final location of the equipment or on the path for moving the equipment into its final location.

10.14 Combined servomotor volume, stroke, and timing

The combined servomotor volume, stroke, and timing of the turbine control servomotors impact the design of the hydraulic pressure supply system, the servomotor piping, and the governor distributing valves. This information should be provided to the designer of the turbine governing system.

10.14.1 Wicket gate servomotor

For wicket gate servomotors, the bore diameter, the rod diameter, the maximum servomotor stroke, the maximum rate of servomotor travel, the number of servomotors, and the servomotor position at which the slow closure feature becomes effective should be provided to the manufacturer of the turbine governing system.

10.14.2 Runner blade servomotor

For adjustable-blade turbines, the blade servomotor bore diameter, the maximum servomotor stroke, the maximum rate of servomotor travel, and the diameters of both the servomotor pipe and the servomotor connecting rod should be provided to the manufacturer of the turbine governing system.

10.14.3 Deflector servomotor

For impulse turbine deflector servomotors, the bore diameter, the rod diameter, the maximum servomotor stroke, the number of servomotors, and the maximum rate of servomotor travel should be provided to the manufacturer of the turbine governing system.

10.14.4 Needle servomotor

For impulse turbine needle servomotors, the bore diameter, the rod diameter, the maximum servomotor stroke, the number of servomotors, and the maximum rate of servomotor travel should be provided to the manufacturer of the turbine governing system. If the needle servomotor is spring-loaded in the closing direction, the preload and scale of this spring should also be provided.

10.15 Servomotor design operating pressure

The servomotor design operating pressure is typically the maximum pressure at which the servomotor is expected to operate. It is at this pressure that the servomotor is capable of providing its maximum output force. The servomotor positioning system should have an adequate margin of safety in its design. A general practice is to design the turbine control servomotors such that the expected maximum mechanical loading of the servomotor does not exceed 60% to 70% of the available maximum force from the servomotor.

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10.16 Turbine control servomotor connection sizes

The type and size of the piping connection to the turbine control servomotors should be specified. The actual size of the connecting piping may be larger than the connection size at the servomotor if this is necessary to limit the velocity losses within the piping.

10.17 Servomotor travel direction to close

To properly design the feedback restoring system, it is necessary that the direction of travel for the turbine control servomotors be defined.

10.18 Minimum differential pressure required to close

Both the frictional loading and the hydraulic loading on the turbine control devices determine the servomotor differential pressure required to move the servomotor in the opening or closing direction. The frictional loading is only present when the turbine control device is being moved. The hydraulic force of the water on the turbine control devices has a steady-state value depending on flow and the position of the turbine control device. There may also be a dynamic force resulting from the effects of changing water flow with the turbine control device. This dynamic force is only present when the turbine control device is being moved to change the water flow through the turbine. Accordingly, any information available about the expected loading on the turbine control servomotors should be provided to the designer of the turbine control manufacturer.

10.19 Gate shaft or deflector shaft direction and angular travel to close

If the turbine control servomotor is used to rotate a gate shaft or deflector shaft, the direction of rotation and the total angular travel of this shaft should be provided to the manufacturer of the turbine governing system.

10.20 Required governor capacity

The required governor capacity is generally specified in terms of total energy, in Newton-meters, to move the turbine control servomotor through its full travel, and it should be provided to the designer of the turbine governing system. The sizing of the turbine governing system components is also affected by the required rate of change of energy to the turbine control servomotors.

10.21 Turbine control servomotor time: opening and closing

Generally, the rate of travel for turbine control servomotors should be provided, and it is normally specified in terms of total time for 100% servomotor travel. This rate may also be specified in terms of percent servomotor stroke per second. For servomotors with more than one rate specification over its travel, the rate may be specified in terms of total travel time over a specified portion of the servomotor stroke.

10.22 Results of turbine model tests or index tests

If model testing or index testing has been performed for the turbine for which the turbine governing system is being designed, this test information should be provided to the manufacturer of the turbine governing system. For double-regulated turbines, this information is necessary to establish the optimum control relationship between the primary turbine control device (e.g., gates or needles) and the secondary turbine control device (e.g., blades or deflectors).

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10.23 Switchboard instrument specifications

The type, size, and any special characteristics for required switchboard instruments should be provided to the manufacturer of the turbine governing system. Graphic user interface panels may provide many display functions historically provided by individual switchboard instruments.

10.24 Speed switch specifications

The operating and reset speeds for speed switch functions should be provided to the manufacturer of the turbine governing system. If separate devices performing speed-sensitive switch functions are required, the type and speed range should be specified. If electrical contacts are required for use by the purchaser from any speed switch function, the purchaser should specify the voltage and current-carrying and current-interrupting capacity of these contacts.

10.25 Brake actuating medium

The supply pressure, required flow rate, and type of medium used for operating the generator brakes should be specified to the manufacturer of the turbine governing system if control of the generator brakes is within the scope of supply of the turbine governing system.

10.26 Interface to purchaser equipment

Any interfacing requirements should be fully defined for the manufacturer of the turbine governing system. Mechanical interfacing requirements, electrical interfacing requirements, and performance requirements should be adequately described to allow the proper design of the turbine governing system equipment.

10.27 Special design considerations

If any special functions are required for the operation of the purchaser’s station, these functions should be described and specified in sufficient detail to allow the proper design of the turbine control system.

10.28 Required initial adjustments

The purchaser should specify both the desired initial adjustment and the range of adjustment for all components of the turbine governing system that have adjustable parameters.

10.29 Complete prototype turbine data

If available, complete four-quadrant prototype turbine data or turbine model data (together with model to prototype scaling information) should be provided to the manufacturer of the turbine governing system. These data can be used to mathematically model the turbine to evaluate the adequacy of the performance achievable by the turbine governing system.

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Annex B

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Impact of turbine design on governing performance

B.1 Impulse turbine

The impulse turbine (commonly called a Pelton turbine) has a runner with numerous spoon-shaped "buckets" attached to its periphery, which are driven by one or more jets of water issuing from fixed or adjustable nozzles. Impulse turbines extract power from the impact of the water on their runners. Many impulse turbines use deflectors to allow rapid reduction of the power delivered to the turbine runner in cases where the flow cannot be changed quickly. Large Pelton units are typically used at heads above 300 m. Smaller "standardized" units can operate at reasonable efficiencies at heads of 30 m and less. Impulse turbines operate best at nearly constant heads, and they have a relatively flat efficiency curve down to approximately 20% of rated output. This is a useful characteristic where flow range is wide. Unit sizes range up to 300 MW.

Figure B.1 shows typical efficiency curves for an impulse turbine. The governing system for an impulse turbine can reduce the generated power very rapidly by using the deflectors. The rate of increasing the generated power is determined by the much slower moving needles, placing some limitations on operating the unit in an isolated mode. Control strategies such as "water-wasting" can be implemented to improve isolated operation by running the unit with the deflectors partially into the water stream. Also, sequencing of needle operation on multiple-needle units can improve efficiencies at lower generation levels.

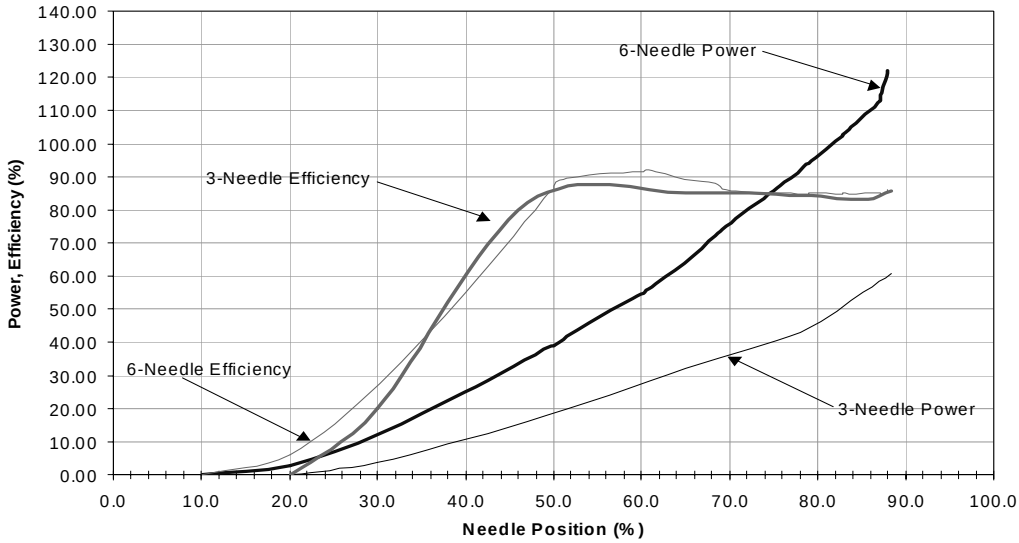


Figure B.1—Typical impulse turbine characteristics

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B.2 Reaction turbine

Reaction turbines extract power from the kinetic energy of water as a result of the difference in pressure between the front and the back of each runner blade as the water flows through the runner. There are three major types of reaction turbines: Francis turbines, fixed-blade propeller turbines, and adjustable-blade propeller turbines.

B.2.1 Francis turbine

The Francis turbine is constructed so that water enters the runner radially and then flows toward the center and along the turbine shaft axis. These units are most often applied for turbine operating heads ranging from 30 m to 450 m, and they are usually the economic choice in this range. However, small Francis units can operate satisfactorily under heads as low as 6 m. Unit sizes range from 1 kW to 1000 MW.

Figure B.2 shows typical power and efficiency curves for a Francis turbine. The governing system uses the wicket gate position to control the speed and generation of the unit.

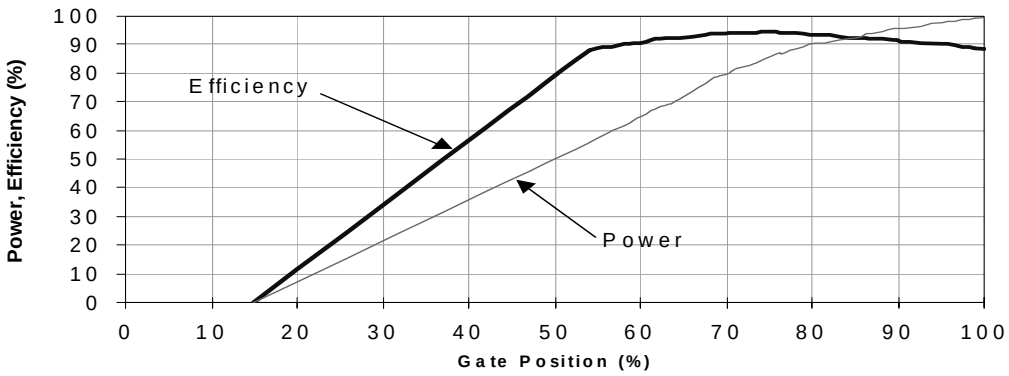


Figure B.2—Typical Francis turbine characteristics

B.2.2 Fixed-blade propeller turbine

The propeller turbine passes water through its propeller blades in an axial direction. Propeller turbines can be designed for heads ranging from 3 m to 60 m, but they are usually an economic choice in the 15 m to 45 m head range. Units as small as 0.5 MW can be obtained, but most are 10 MW or larger (up to 150 MW).

Figure B.3 shows typical power and efficiency curves for a fixed-blade propeller turbine. This type of turbine operates efficiently over a fairly limited range of output. Therefore, it is normally used where it can be operated close to its design discharge. The governing system uses the guide vane (wicket gate) position to control the speed and generation of the unit.

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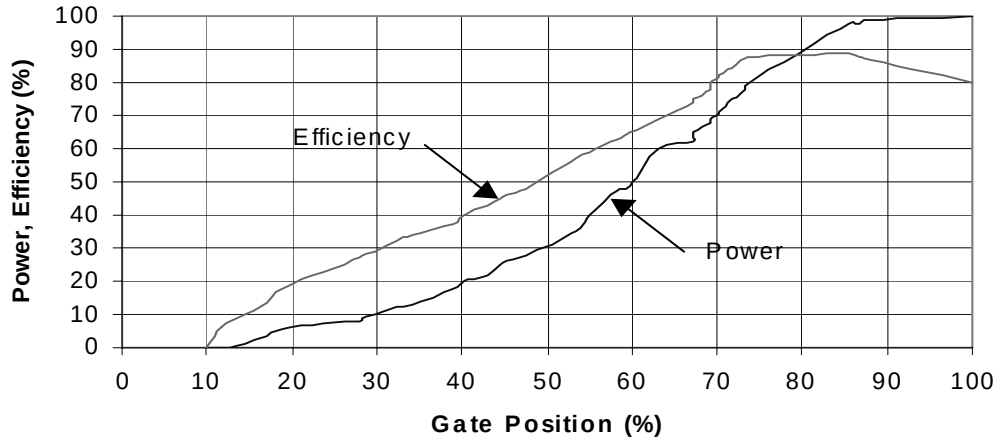


Figure B.3—Typical fixed-blade propeller turbine characteristics

B.2.3 Adjustable-blade propeller turbine

Propeller turbines with adjustable blades can operate in the same general head range as those with fixed blades. Using an adjustable-blade turbine generally results in a relatively flat efficiency curve over a wide range of head and flow. Some types of adjustable-blade propeller turbines include Kaplan turbines, bulb turbines, tubular turbines, pit turbines, and rim turbines. These turbines are available in unit sizes ranging from 1 kW to in excess of 150 MW. Some types have vertical shafts, and some have horizontal or inclined shafts. Depending on the application, the added expense of adjustable blades on the turbine can be offset by the increased efficiency at varying flow rates and power outputs.

Figure B.4 shows typical power and efficiency curves for an adjustable-blade propeller turbine. The governing system typically uses the guide vane (wicket gate) position to control the speed and generation of the unit. The blade angle is usually adjusted to obtain the best turbine efficiency for the operating head and flow, but some control strategies may also use blade angle control to improve the dynamic performance of the unit during transient conditions.

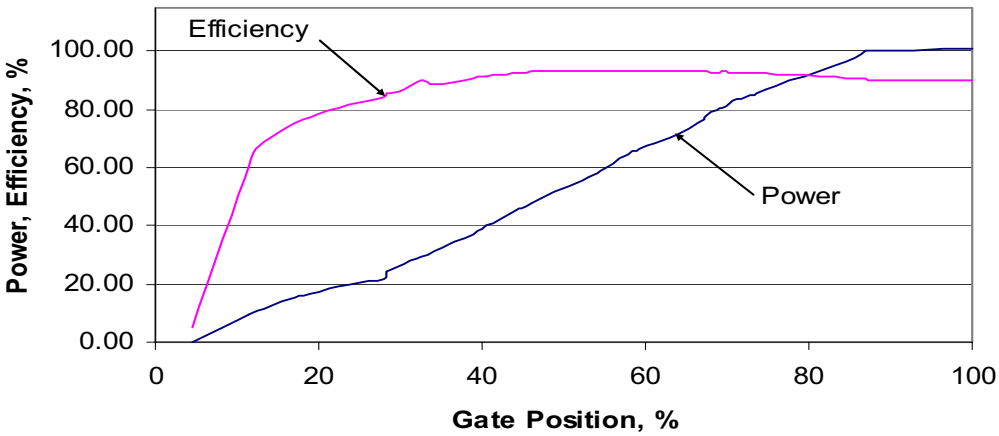


Figure B.4—Typical adjustable-blade turbine characteristics

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Annex C

(informative)

Examples of turbine governing systems

C.1 Speed governor

Figure C.1 shows a block diagram of a typical speed governor for a hydroelectric generating unit connected to an isolated electrical load. The “setpoint” input to this system is the speed reference (“speed adjustment”), and the feedbacks used are the unit’s actual speed and the gate position. The “process output” of this system is the speed of the turbine, which determines the frequency of the generator output. The “controlled process” here includes the turbine, the water passage, and the generator.

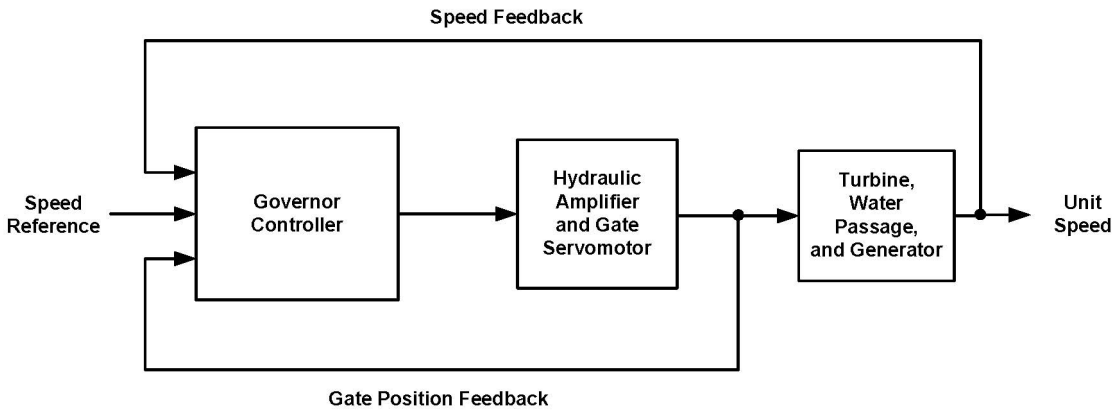


Figure C.1—Typical speed governing system

C.2 Pond level controller

Figure C.2 shows a block diagram of a head pond level controller. The “setpoint” input to this control system is the desired pond level, and the feedbacks used are the actual pond level and the unit’s wicket gate position. The “process output” of this system is the head pond level. The “controlled process” here includes the turbine, the water passages, and the water reservoir. This same basic control approach can be used to control tail pond. The only difference in approach for controlling tail pond level would be the reversal of polarity of the summation of the tail pond level signal within the controller function.

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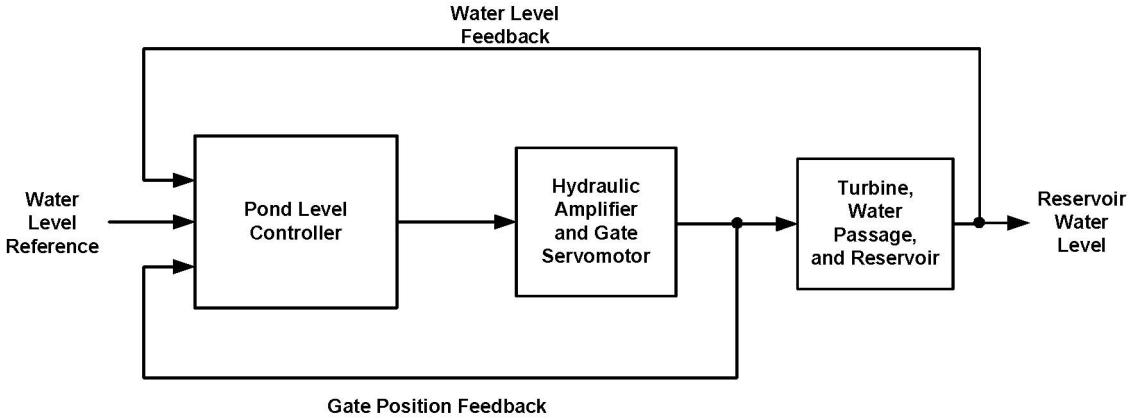


Figure C.2—Typical head pond level control system

C.3 Grid-connected generation controller

Figure C.3 shows a block diagram of a grid-connected generation controller. The "setpoint" input to this control system is the combination of the desired generation level (in megawatts) and the speed reference, and the feedback used is the unit generation. The "process output" of this system is the unit generation. The "controlled process" here includes the turbine, the water passages, the water reservoir, and the generator. The unit speed feedback is used to produce the speed regulation or "power droop" characteristic of the grid-connected unit.

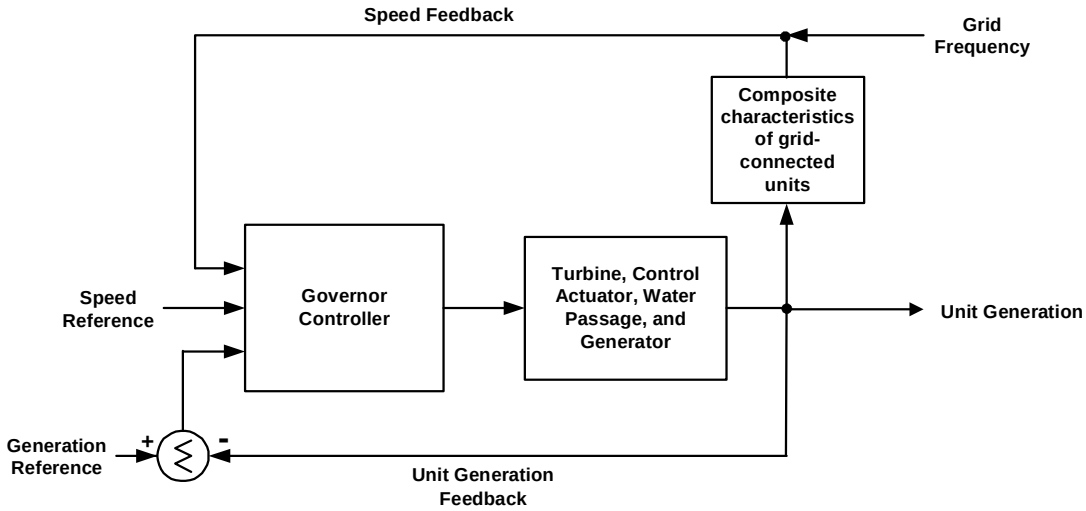


Figure C.3—Typical generation control system

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C.4 Load governor

Figure C.4 shows a block diagram of a typical load governing system. The “setpoint” input to this control system is the desired system frequency (“speed reference”), and the feedbacks used are the system frequency and the load power command signal. The “process output” of this system is the system frequency. The “controlled process” here includes the turbine, the generator, and the load bank.

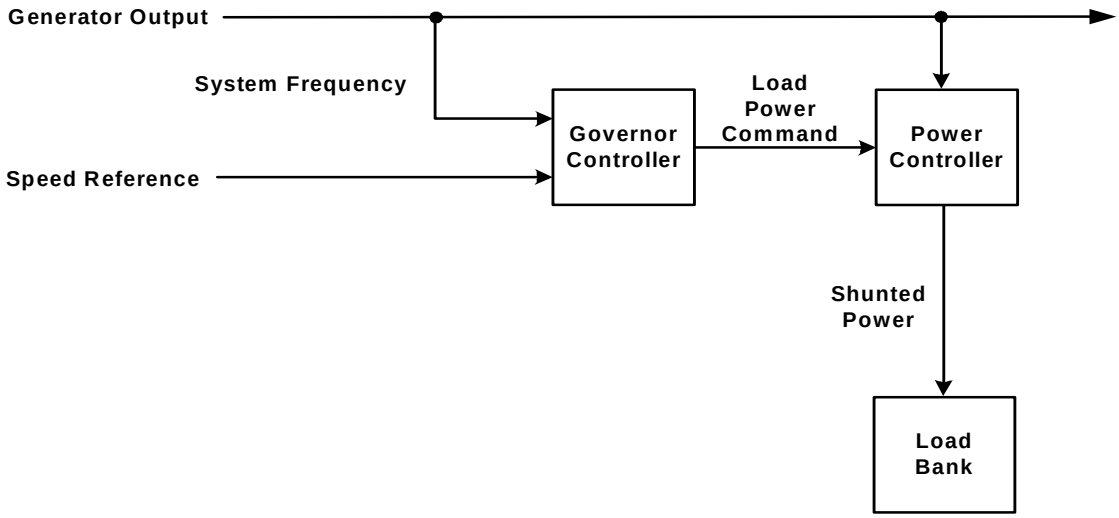


Figure C.4—Typical load governing system

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Annex D

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Experience gained from challenging applications

Over time, it is inevitable that certain situations occur that present unique challenges to the designer of a turbine governing system. In some cases, after a governing system is installed, it does not meet the expected performance criteria. This can occur for a variety of reasons. Some factors may not have been considered in the system design, some changes may have been made in other parts of the system during the design/build stage that impacted the governing system's ability to perform as desired, or experience gained while operating the station may have revealed some previously unknown constraints on the system. Brief descriptions discussing the challenges encountered and the solutions implemented in addressing these challenges are provided in D.1 – D.4.

D.1 Addition of a full-capacity load governor

An instance of adding a load governor to an existing installation occurred when the conventional mechanical-hydraulic governor did not provide acceptable frequency regulation when operating isolated from the grid. The installation was at a location where the hydroelectric generating unit was added to use some of the water that was being continuously spilled over the dam at a lock provided on the river for navigational purposes. Because of regulatory complications, the unit could not be connected to the utility power, so the unit was operated with the lock house as an isolated load. Because of the ratio of water column inertia to rotating inertia, the conventional governor was not able to keep the frequency within acceptable limits for normal load transients. This caused severe operating inconveniences for the operating personnel at the lock.

A load governor was then designed to be a full-capacity governing system using a resistive load bank rated for the maximum output of the generator of 60 kW. The original mechanical-hydraulic governor was set for full flow through the turbine during normal operation, and it was left in service to be used only as a turbine shutdown device for safety purposes. As the load governor regulates unit speed without requiring changes in the flow through the turbine, the large water column inertia was no longer a factor in the controllability of the system frequency. After installation of the load governor, the system was able to absorb much larger load transients than before. There was the added benefit of reduced maintenance requirements for the wicket gates, because they were no longer being moved continuously in trying to control the system frequency. An unexpected benefit of the load governor was that the portable resistive load bank could be moved into the powerhouse during cold weather to use the dissipated energy to heat the powerhouse, thus reducing the cost of fuel oil for keeping the powerhouse at a comfortable temperature for the personnel on duty. Figure D.1 is a block diagram representation of the load governing system implemented at this plant.

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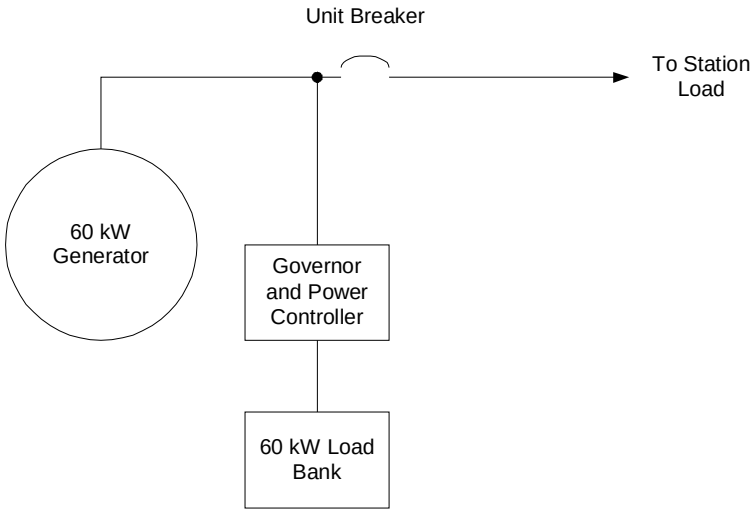


Figure D.1—60 kW load governing system

D.2 Addition of a load governor for synchronizing

During the design of a conventional governing system, a routine evaluation of the ratio of the water inertia to the rotating inertia revealed that a conventional turbine governing system would not be able to provide stable speed control for the 10 MW generating unit. The design of the water column and other significant portions of the station were such that it would not be economical to change either the water starting time or the mechanical starting time enough to allow stable control for synchronizing. In this application, the unit would always be connected to a large grid when in operation, so it would never need to operate into an isolated load. The decision was made that the turbine governing system would only need to control the speed of the turbine well enough to synchronize it to the grid. After that, it was acceptable for the unit to derive its speed stability from the grid.

A 500 kW shunt load bank was added to the system to be used in a load governing system for unit speed control during the synchronizing of the unit to the grid. The wicket gates would be positioned to a point where the no-load speed of the unit was slightly over synchronous speed. At that point, the synchronizer would give command signals to the load bank power controller to reduce the speed of the unit down to synchronous speed. In this way, during the synchronizing, the flow of water through the unit would not change significantly, and the water hammer effects would not interfere with the synchronizing action. It was determined that a load bank rated for approximately 5% of the maximum generator output would yield acceptable performance. As this generating unit was rated at 10 MW, the load bank needed to be 500 kW. The load bank chosen was an air-cooled unit, and provisions needed to be made in the powerhouse for exhausting the heat produced by the load bank while the unit was being synchronized. After synchronizing to the grid, the load bank is automatically disconnected from the unit generator, and the generation of the unit into the grid is controlled in a conventional manner using the wicket gates. Figure D.2 is a block diagram representation of the load governing synchronizing system implemented at this plant.

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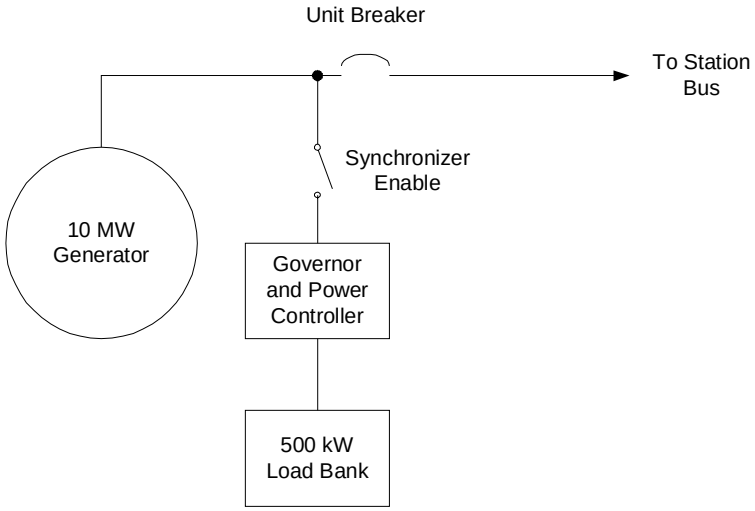


Figure D.2—Load governing system used for synchronizing

D.3 Mechanical implementation of a water-wasting mode of operation

An impulse unit at this station operates into a strong grid when online. The unit was very difficult to synchronize. The tunnel supplying water to the unit is approximately five miles long. Even after overhauling the governing unit, the synchroscope indicated some instability when attempting to synchronize to the grid. This unit's governor uses a cam on the mechanical restoring feedback from the deflector servomotor to produce the mechanical setpoint for the needle position hydraulic amplifier. To stabilize the unit at the low needle position, the deflector/needle cam was ground down around the speed-no-load position to allow the deflectors to be in the water stream while synchronizing. In this way, the control action of the deflectors would dominate the control of the unit. The much faster timing of the deflector servomotors, along with the elimination of the water hammer effect (since the deflectors do not control water flow) from the control loop, allowed the unit to control speed in a much more stable manner while synchronizing. This change eliminated all noticeable instability, as displayed by the synchroscope. Figure D.3 illustrates the modification made to the existing blade control cam on the mechanical-hydraulic turbine governing system.

D.4 Implementation of separately-tuned needle and deflector control loops

Two 63 MW impulse units are operated into a relatively weak grid when online, and occasionally into an isolated load. The tunnel supplying water to these turbines is approximately 5 km (3 miles) long. As this particular station is a substantial portion of the total system generating capacity, it is essential to system integrity that the unit is able to respond quickly to load changes. Traditional impulse wheel control strategies use a PID controller loop for the deflectors as the primary turbine control device, with needles following on a deflector/needle 2D "cam" relationship. Simulation studies indicated that this approach would not provide acceptable frequency control performance in this application.

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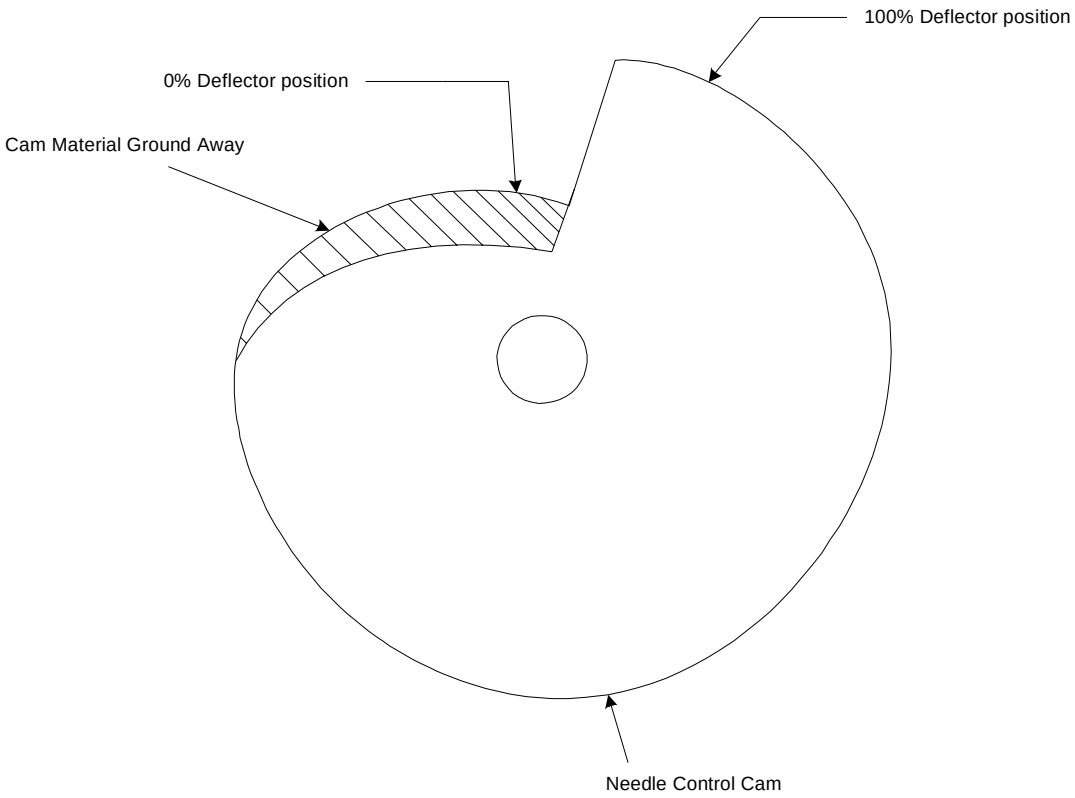


Figure D.3—Modification to needle control cam made to implement water-wasting mode

This application inspired a very robust solution involving the use of two PID control loops in the governing system. One PID loop controls the deflectors, whereas the other controls the needles. The two PID loops communicate with each other so that at steady state, the deflectors remain just outside the water stream from the needles (according to the 2D needle/deflector curve). This results in the most efficient unit operation with large load shedding capability. During one test, a partial system rejection was performed (50 MW load suddenly reduced to 30 MW) with two units operating. A maximum of 105% of normal speed was experienced, with speed dropping back to within 1% of nominal speed within a few seconds. Peak speed for full load rejections was approximately 110% of normal speed. Figure D.4 is a block diagram illustrating this control strategy.

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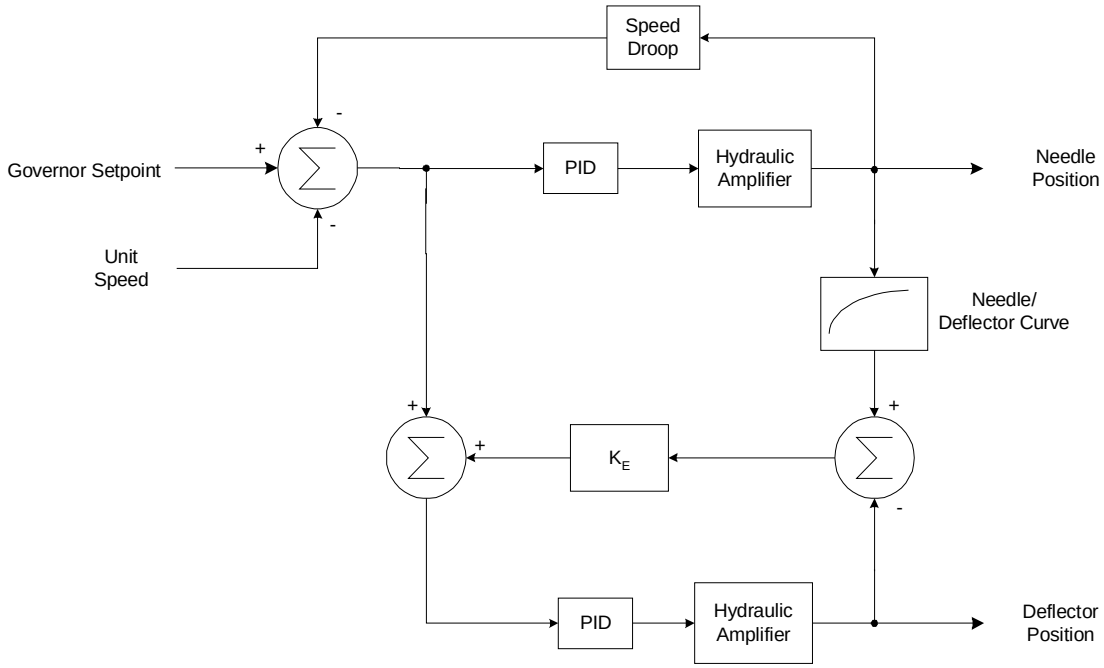


Figure D.4—Dual PID control loop strategy for impulse turbine

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Annex E

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Governor simulations to demonstrate sensitivity of governor parameters on performance

This annex illustrates the use of computerized modeling to predict the performance of a turbine governing system. Several cases are presented, some cases represent isolated operation and some cases represent interconnected operation. For each mode of operation, multiple transient responses are modeled to illustrate the effects of certain system parameters on the performance of the turbine governing system (see Codrington et al. [B7]).

E.1 Isolated system governor simulations

The following simulations were performed using a mechanical-hydraulic temporary droop governor model, for a reaction turbine driving a hydroelectric generating unit connected to an isolated system. The isolated system in this model consists of one generating unit supplying one resistive load. The model includes the temporary droop mechanical governor shown in Figure E.1, a turbine-penstock hydraulic model, and a model of the unit rotating inertia. It does not model generator electrical effects (see Bollinger et al. [B4], Carson [B5], Paynter [B35], Ramey and Skooglund [B36], and Schleif and Wilbor [B40]).

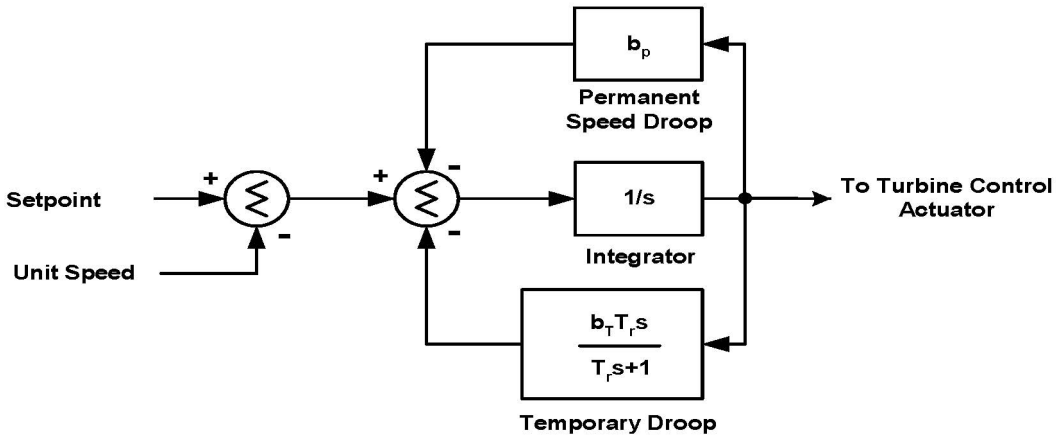


Figure E.1—Temporary droop governor model

The base case model parameters are as follows:

Permanent speed droop b_p = 0.05 (5%)

Inertia Constant H = 4.75

Mechanical starting time $T_M = 2 \times H$ = 9.5 s

Water inertia time constant T_W =1.24 s

Damping (Reset) time constant T_d = 5 s (using an empirical guideline: T_d = 4 to 5 times T_W)

Temporary droop b_t = 0.27 (27%) (using an empirical guideline: b_t = 2 to 2.5 times T_W/T_M)

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where

b_P is the permanent speed droop, per-unit,

b_t is the temporary droop, per-unit,

T_r is the damping reset time constant, s,

s is the Laplace operator.

Figure E.2 illustrates the modeled response of the isolated unit response to a 5% step decrease in load, using the system parameters listed in the base case of the simulation.

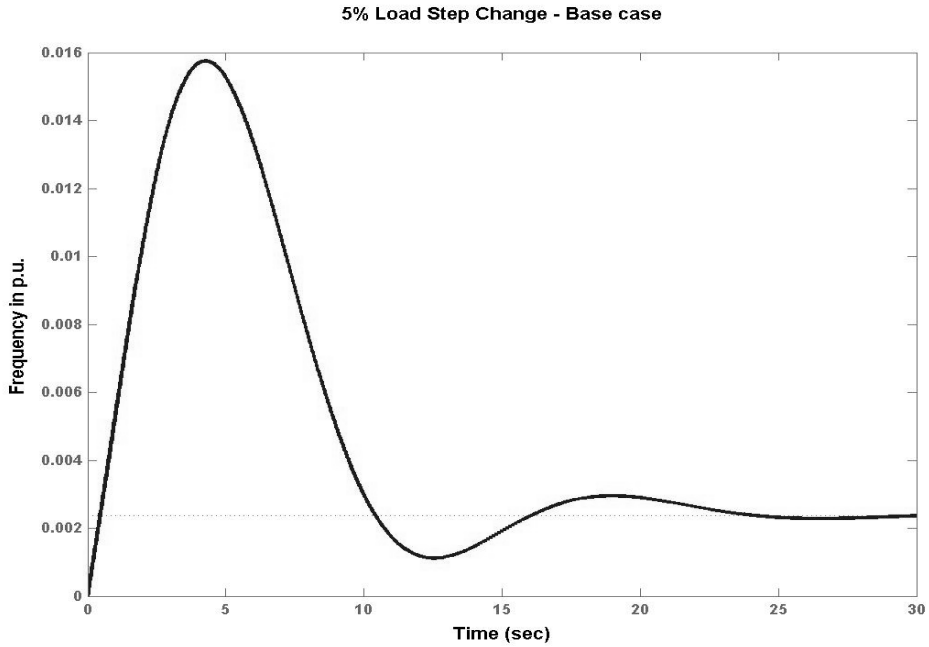


Figure E.2—Base case: Speed (frequency) vs. time response for a 5% load step change

Figure E.3 through Figure E.7 illustrate the effects of varying the values of the *H* constant, the water starting time (T_w), the temporary droop (b_t), the damping time constant (T_d), and the permanent droop (b_p) upon the response of the simulated system to the base 5% load step change.

Note that when the inertia constant *H* is decreased from 4.75 to 3.0, the slope of the step response is steeper and less damped for the larger unit inertia. When the inertia constant *H* is increased from 4.75 to 6.0, the slope of the step response is less steep and the damping is greater.

The transient response peak decreases with a smaller T_w by decreasing water inertia time constant T_w from 1.24 s to 0.8 s. The opposite effect is achieved when the water inertia time constant is increased from 1.24 s to 1.6 s.

Decreasing temporary droop from 27% 12% increases the speed of response and oscillations. Increasing temporary droop from 27% to 40% decreases the speed of response and of the transient oscillations.

Decreasing T_d from 5 s to 3 s decreases damping. Increasing T_d from 5 s to 10 s increases damping. Note that the peak of the speed transient remains the same in all three simulations and only damping is affected.

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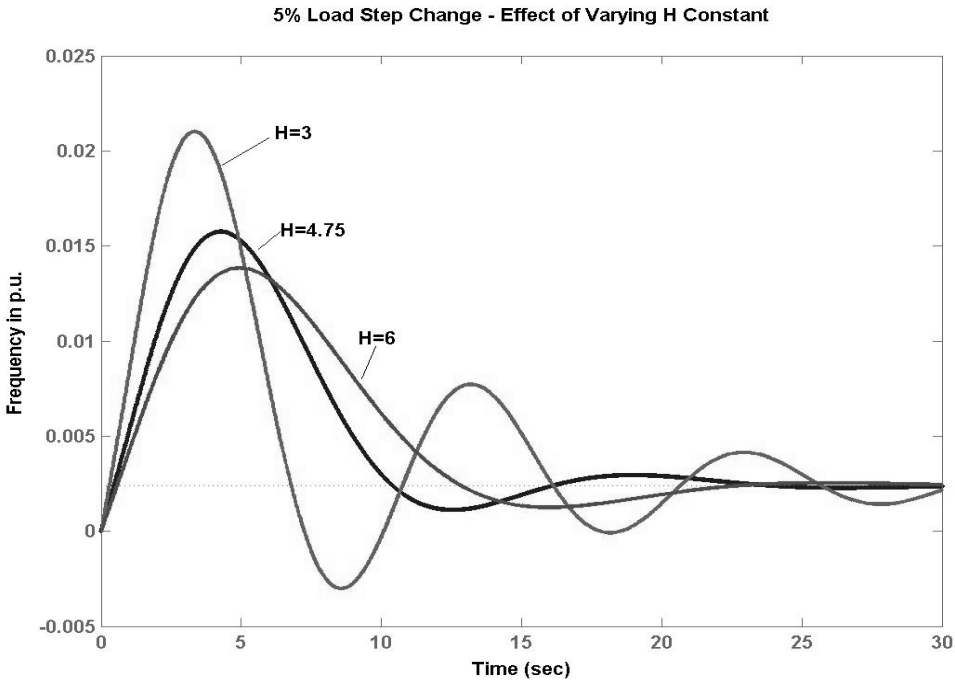


Figure E.3—Unit response with varying inertia constant *H*

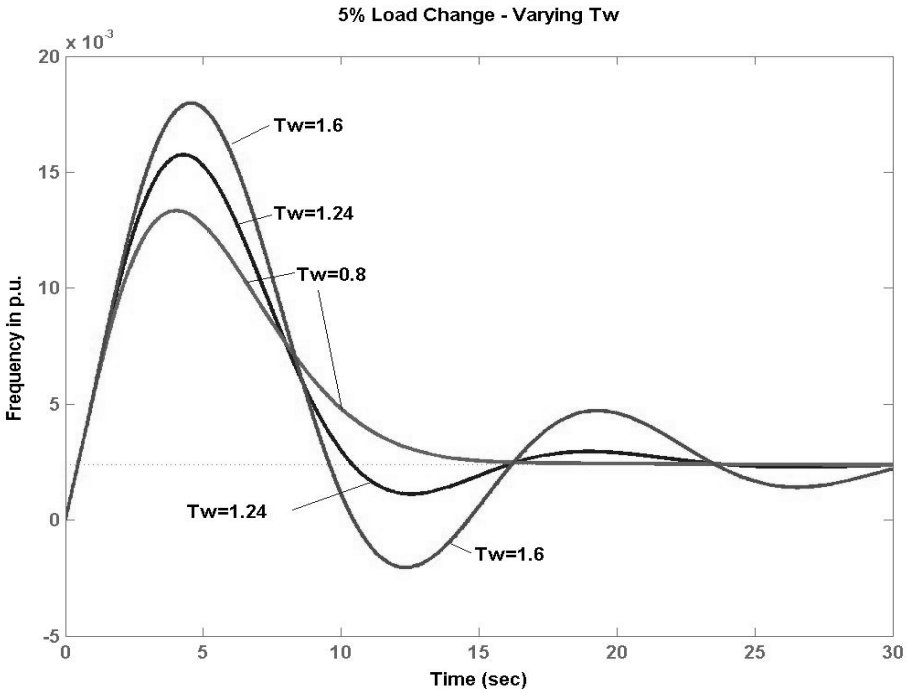


Figure E.4—Unit response with varying water inertia time constant T_W

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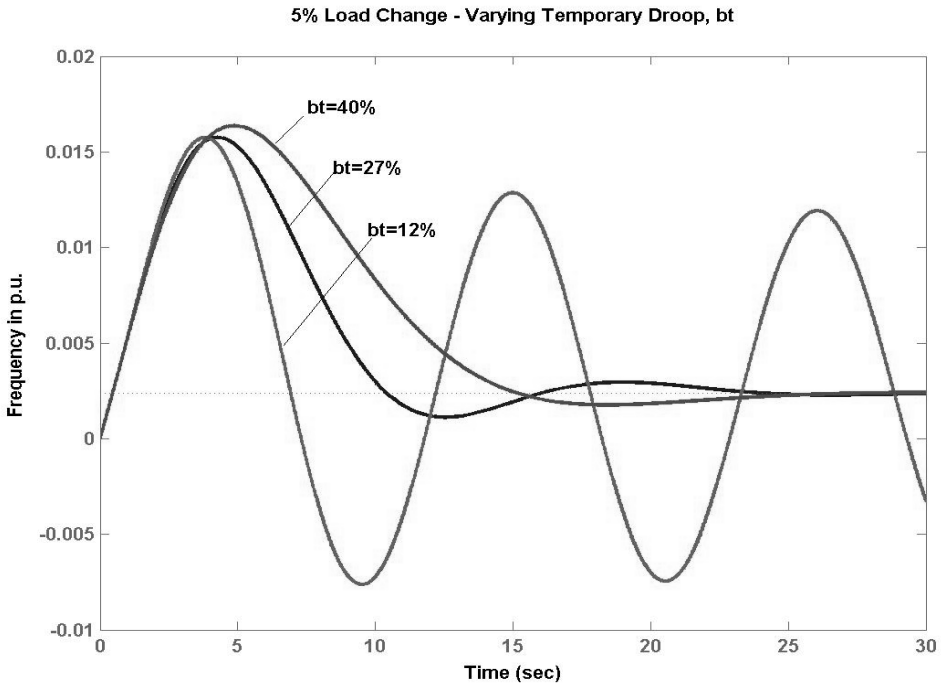


Figure E.5—Unit response with varying temporary droop (b_t)

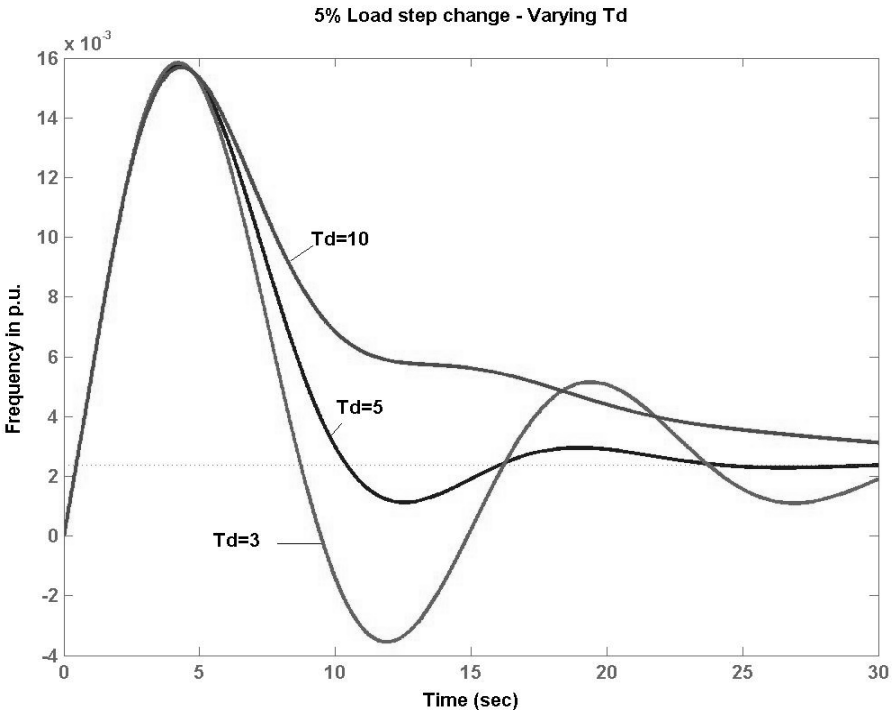


Figure E.6—Unit response with varying damping time constant (T_d)

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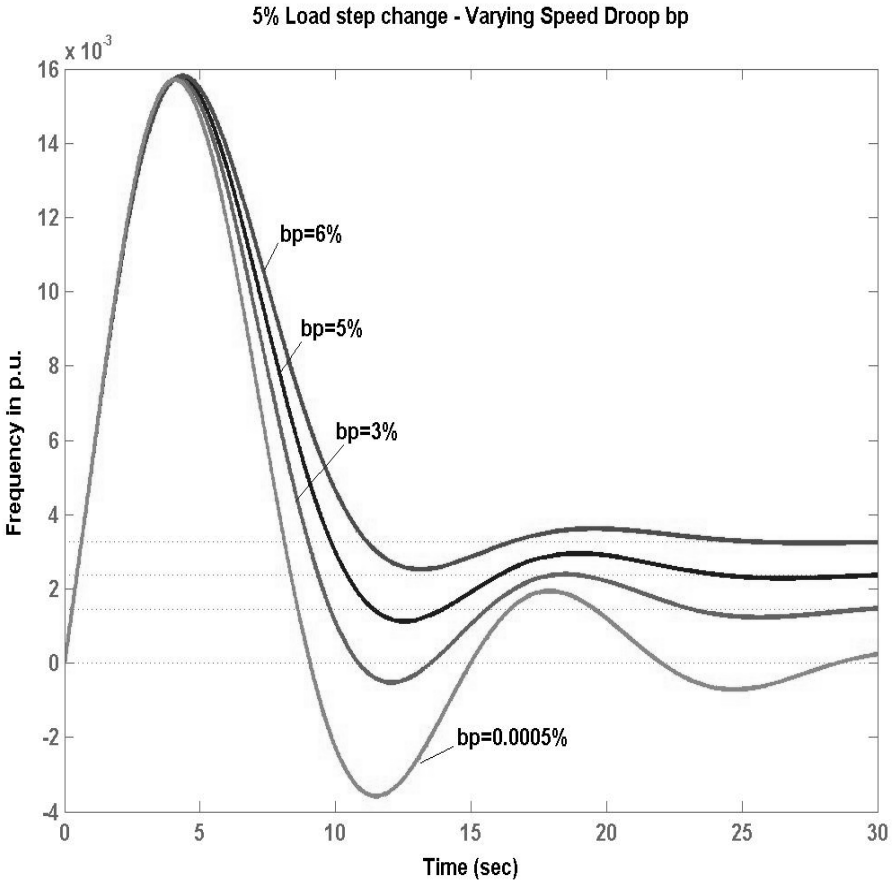


Figure E.7—Showing the effect of varying permanent speed droop (b_p) for a 5% step load change

The steady-state frequency deviation after the initial transient effects disappear is the product of the permanent speed droop and the size of the load change. Thus, for a 5% load change, the steady-state speed offset is 0.15% for 3% permanent speed droop, 0.25% for 5% permanent speed droop, and 0.3% for 6% permanent speed droop. Note that the bottom response curve is essentially isochronous operation with a permanent speed droop of 0.0005% giving a final frequency deviation of almost zero (0.0025%).

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E.2 Interconnected system governor simulations

The following simulations were performed with a simulation program to demonstrate the governor response of a 120 MW hydro unit connected to a small 2000 MW interconnected system. The interconnected system was modeled as two 120 MW hydro generators connected to a 2000 MW synchronous generator representing the system through a double circuit line comprising 25 km of 230 kV transmission line. The system model consisted of a single 2000 MW unit operating with permanent speed droop = 5%, T_w = 1.24 s, and T_M = 9.5 s. All machines are modeled with IEEE model representations for turbines, governors, generators, and excitation systems. The unit governors are modeled with a mechanical-hydraulic temporary droop hydro governor model similar to the isolated system example case. A 100 MW inertialess, constant-power load connected at the hydro plant is tripped off to represent a 5% load trip within the system. Figure E.8 illustrates the response of the units to this disturbance. The interconnected system case turbine/governor model parameters are as follows:

Permanent speed droop b_p =0.05 (5%)

Inertia Constant H = 4.75

Mechanical starting time $T_M = 2 \times H$ = 9.5 s

Water inertia time constant T_w =1.24 s

Damping (Reset) time constant T_d = 5 s

Temporary droop b_t = 0.27 (27%)

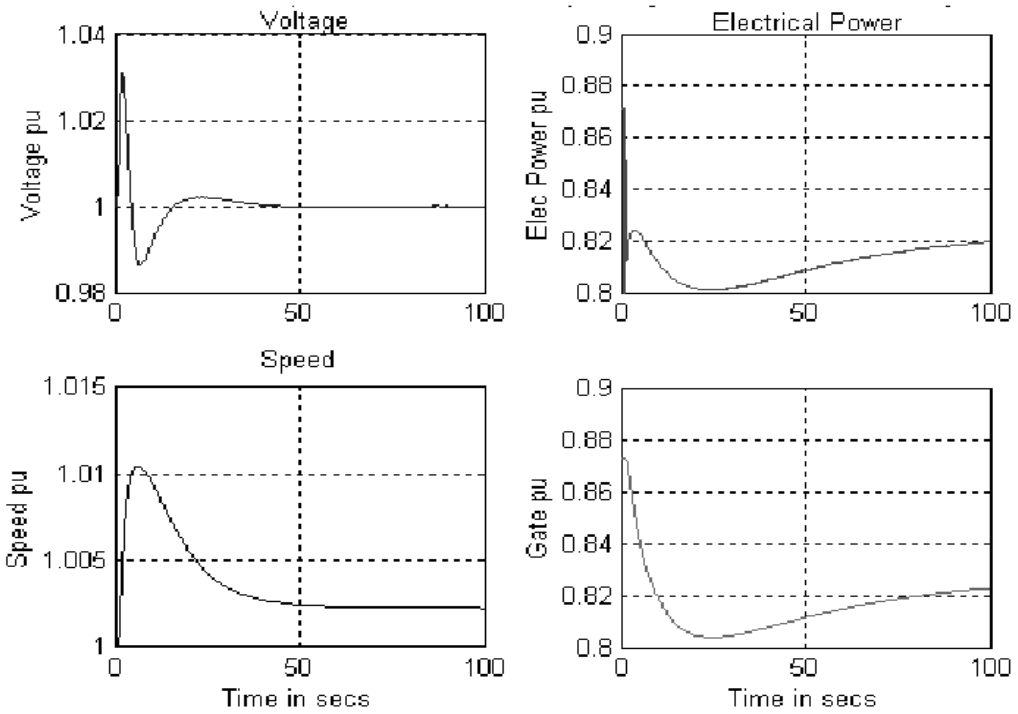


Figure E.8—Base case: Simulated unit response for a 5% interconnected system load step change

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Each generator drops 5% of its rating in accordance with the permanent speed droop characteristic, the 120 MW dropping 6 MW in the final steady state. The steady-state system speed deviation is 0.25% (0.05 x 0.05 x 100).

Note that in contrast with the isolated system case discussed in E.1, the first peak of the speed transient is less in magnitude and subsequent oscillatory behavior in the interconnected system is more damped. The steady-state speed deviation of 0.25 is achieved in a longer time period of 100 s. The greater damping of this response is primarily provided by the modeling of the electrical system characteristics within the system model, as contrasted with the isolated unit model illustrated in Figure E.2, which does not model the electrical system characteristics.

The responses for a 5% decrease in load to a 120 MW hydro unit in a 2000 MW system are shown in Figure E.9, where the only parameter changed was the temporary droop on the unit being modeled. Note that, for the sake of clarity, the curves of per-unit speed are not labeled with their corresponding value of temporary droop, but they appear on the graph in the same positional order as the curves of gate position. This graph shows the sensitivity of the temporary droop parameter on this hydro unit's interconnected system performance. The 12% temporary droop case exhibits the fastest response as seen in the gate position response, which gets progressively slower as the temporary droop setting is increased. Responses are not oscillatory in comparison with the single unit isolated system model responses. This is primarily due to the damping effect of the modeling of the electrical system characteristics in the interconnected response simulation, which was not modeled in the single unit isolated system model.

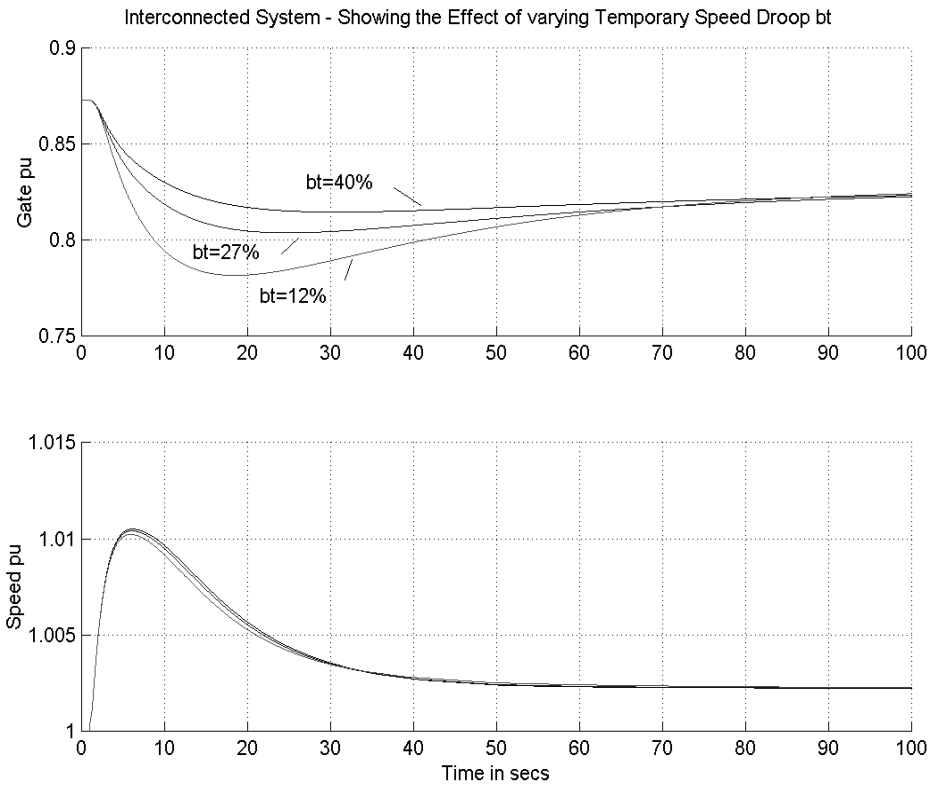


Figure E.9—Unit response with varying temporary droop (b_t) for a 5% interconnected system load step change

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E.3 Simulations of dual-regulated turbine performance

The preceding examples have used a model of a single-regulated turbine governing system, such as would be used on a Francis turbine. Figure E.10 shows a function block diagram of a model that could be used to simulate a typical mechanical governor used on a dual-regulated turbine, such as an adjustable-blade turbine or an impulse turbine. The process of modeling system response for a dual-regulated turbine is very similar to the approach used with the single-regulated turbine model. In the interest of brevity, sample simulation results for a dual-regulated turbine governing system are not presented in this guide.

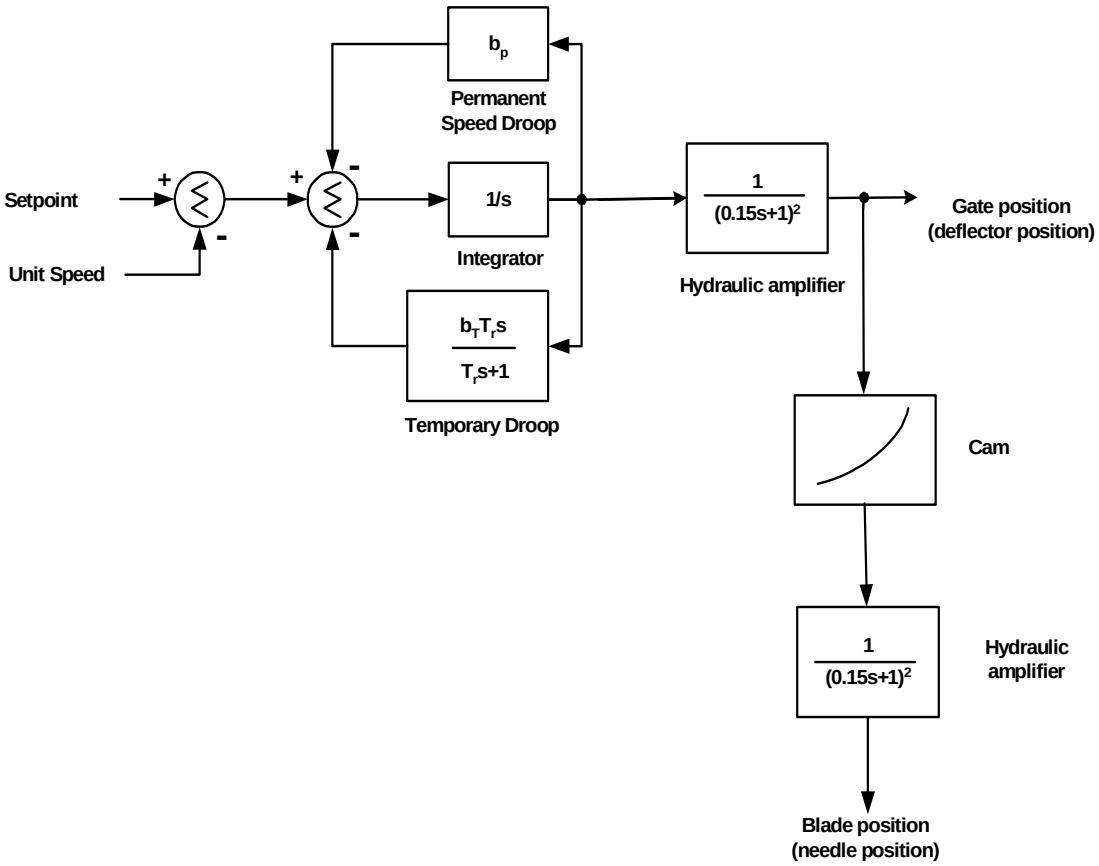


Figure E.10—Temporary droop governor model used with dual-regulated turbines

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Annex F

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Tuning of turbine governing systems

There are many approaches for tuning hydroelectric turbine governing systems for the desired performance, and there are many indices that can be used to measure the dynamic response of these systems. This annex describes some commonly used tuning methods and performance indices.

F.1 Tuning methods

Governor tuning methods are selected depending on performance objectives and system-data availability. Tuning for operation may differ from tuning for specification verification. Conventional tuning methods include several approaches, which generally apply to disturbances less than 10% of a unit's total rated generation level. Governors can usually compensate and stabilize for these load changes. Larger load changes may saturate servo actuators, which are rate-limited due to conduit pressure limits or other considerations (see Agee and Girgis [B1], Sananthanan [B37], Trudnowski and Agee [B42], and Wozniak [B44]).

Dynamic control should factor into the design of power generating units. When the unit is one of multiple generating units in a large interconnected network, the composite response of the units in the network, rather than the local response of each unit, controls the system frequency. However, by design or by accident, most units at times are required to supply isolated loads or to become a part of a small island of generation in which each unit should control power level and frequency, compensating for both unit and load dynamics. Therefore, most governors of hydroelectric generators are tuned to be able to govern isolated loads.

F.1.1 Stability tests

Certain tests are generally specified to verify that the turbine governing system is properly tuned and meets the specified stability criteria. Typically, these tests consist of either a time response or a frequency response plot of the turbine governing system reaction to a system disturbance. The data used to produce these plots may either be measured from the actual control system response to a disturbance or be computed from a mathematical model of the turbine governing system. Simulated isolation, a hybrid technique to determine online stability, is described in Annex H (see Sananthanan [B37]).

F.1.2 Small signal dynamic response

A typical dynamic response to a step input of a feedback control system is shown in Figure F.1. The principle characteristics of interest are the 10% to 90% rise time, overshoot, peak time, and settling time, as indicated.

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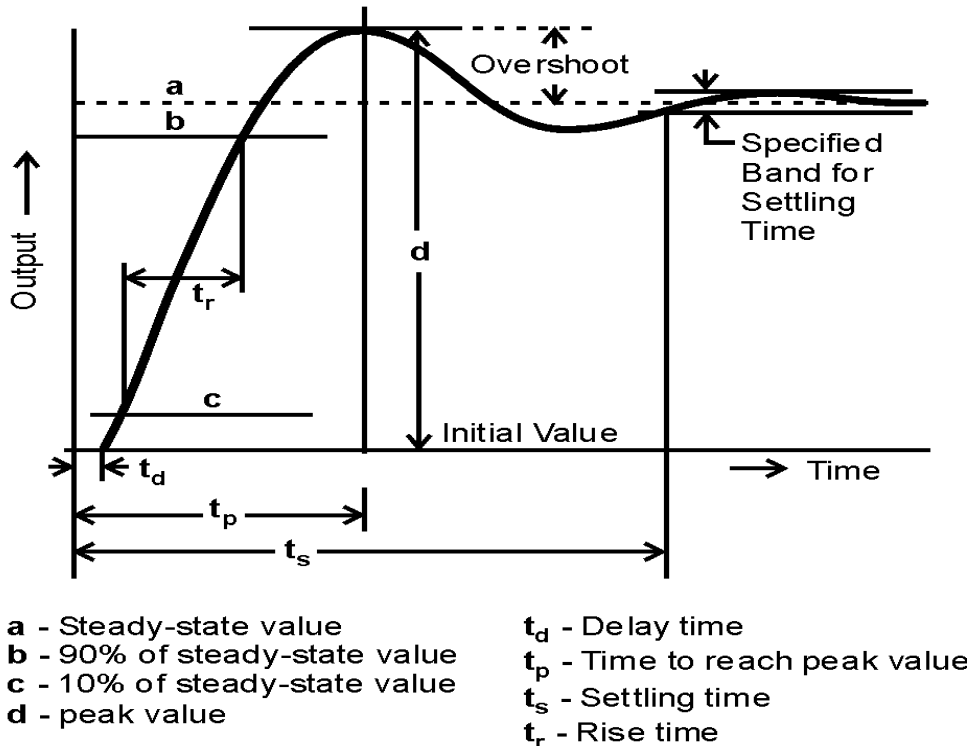


Figure F.1—Typical dynamic response of a turbine governing system to a step change

F.1.3 Governor system time response

A time response is generally obtained by making an oscillographic recording of the unit speed and possibly the turbine control servomotor (wicket gate, needle, or deflector) position in response to a disturbance induced into the operation of the unit. This disturbance may be induced either by injecting an offset into the setpoint of the turbine governing system (for a true step response) or by forcing the unit speed down by a specified amount using the gate limit and then rapidly releasing the limit. Figure 21 shows a typical time response of a unit running off-line using the gate limit method, demonstrating the stability of speed control for the turbine governing system.

F.1.4 Frequency response

In a linear system, the frequency-dependent relation, in both gain and phase difference, between steady-state sinusoidal inputs and the resultant steady-state sinusoidal outputs is called a frequency response. Both open-loop and closed-loop frequency responses are useful for assessing the performance of feedback control systems. In governor systems, open-loop frequency responses of the governor, water column, and inertia and closed-loop responses of the overall speed control system are of primary interest. These data can be collected and plotted in several ways. The most common ways of presenting frequency response data include Bode plots and Nyquist plots.

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F.1.4.1 Bode plot

A Bode plot consists of adjacent plots of gain and phase versus frequency. The gain plot is generally plotted using log-log axes, and the phase plot is generally plotted using semi-log axes. Both open-loop and closed-loop frequency responses can be shown on Bode plots. Figure F.2 shows an open-loop Bode plot. The principal characteristics of interest are the low-frequency gain G, gain-crossover frequency ω_c, and phase crossover Φ_c.

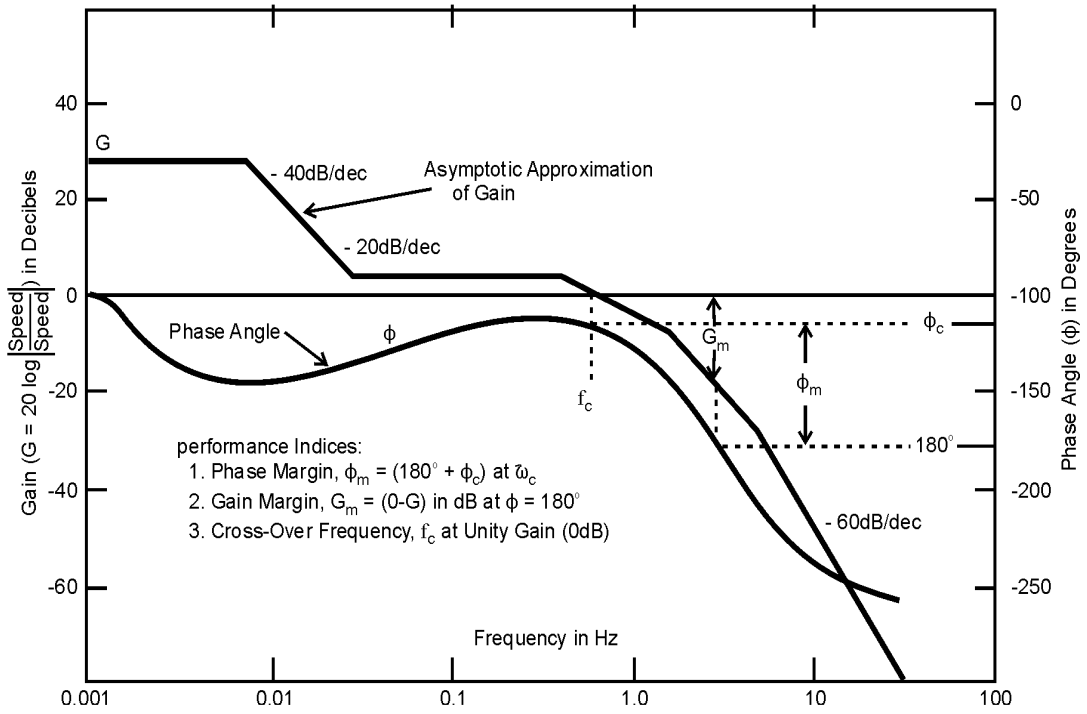


Figure F.2—Typical open-loop frequency response

Open-loop frequency response characteristics are useful in determining gain margin and phase margin, both of which are measures of relative stability of the closed-loop system. Relative stability of a closed-loop control system can be determined from the properties of the Bode plot of the open-loop transfer function. However, this method can be used only if the open-loop transfer function does not have any poles and/or zeros in the right half *s*-plane (minimum phase characteristics).

Relative stability of a feedback control system is measured in terms of the gain and phase margins. For a minimum phase system, which is stable with the feedback loop open, the gain and phase margins should be positive in order for the system to be stable with the feedback loop closed. Negative gain and phase margins means that the system will be unstable with the feedback loop closed. In general, a gain margin of 6 dB or more and a phase margin of 40º or more is recommended for most feedback control systems.

The process of determining stability of non-minimum phase (poles and/or zeros in the right half *s*-plane) and conditionally stable systems is relatively more involved and beyond the scope of this guide. A conditionally stable system is said to be stable for a given range of values of gain but becomes unstable if the gain is either reduced or increased sufficiently.

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The corresponding closed-loop frequency response is shown in Figure F.3. Here the parameters of interest are the bandwidth ω_B, the peak value M_p of the gain characteristic, and the frequency ω_m at which the peak value occurs.

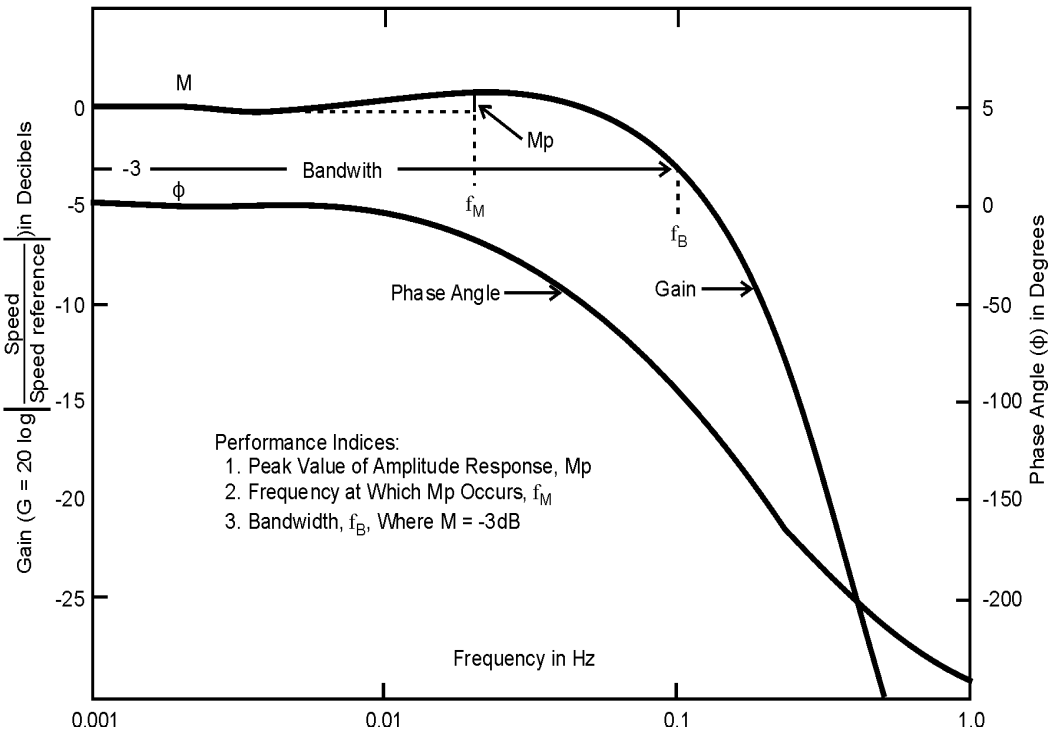


Figure F.3—Typical closed-loop turbine governing system frequency response

With respect to the closed-loop frequency response characteristics, the peak value M_p (in decibels) of the amplitude response is also a measure of relative stability. A high value of M_p (>1.6 dB) is indicative of an oscillatory system exhibiting large overshoot in its dynamic response. In general, 1.1 dB < M_p < 1.6 dB is considered good design practice for most feedback control systems.

Bandwidth ω_B is a significant closed-loop frequency response performance index because it is indicative of the rise time T_r or speed of the dynamic response of the system; it measures, in part, the ability of the system to reproduce input signals. In feedback control systems having a step response exhibiting less than 10% overshoot, rise time T_r in seconds is related to bandwidth ω_B in radians/second by the approximate relationship:

$$T_r\omega_B = 0.30 \text{ to } 0.45$$

In general, the product $T_r\omega_B$ increases as the overshoot in the system dynamic response increases; values in the range of 0.3 to 0.35 correspond to negligible overshoot; values in the region of 0.45 correspond to systems with about 10% overshoot.

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F.1.4.2 Nyquist plot

A Nyquist plot consists of a plot in rectangular coordinates of the ratio of the output of the control system to its input. The vertical axis of the plot represents the imaginary component of this plotted ratio, and the horizontal axis represents the real component of this plotted ratio. Points are generally plotted from zero frequency (or a practical minimum response frequency) to infinite frequency (or a practical maximum response frequency). Nyquist plots are normally used with open-loop frequency response data to determine the stability of a feedback control system. The criterion for stability of a feedback control system is that the Nyquist diagram shall not encircle the point X = –1, Y = 0. Figure F.4 shows a typical Nyquist plot for a turbine governing system, including the governor, water column, and unit inertia.

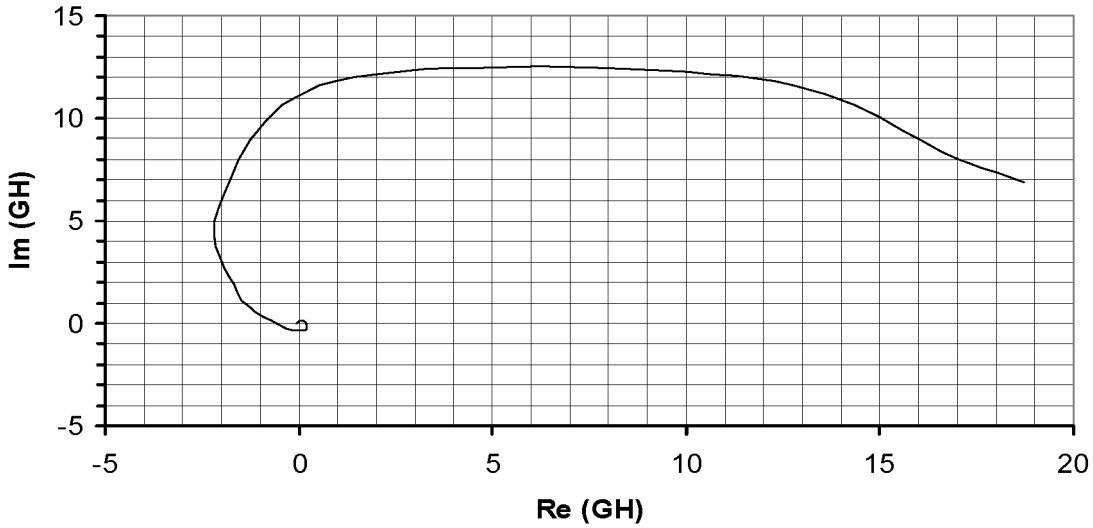


Figure F.4—Typical Nyquist plot of a turbine governing system frequency response

F.1.5 Complex frequency domain (*s*-plane)

The dynamic characteristics of a feedback control system can be represented by plotting the eigenvalues (or characteristic roots) of its transfer function in the complex frequency domain or *s*-plane. Typical root locations of a governor control system are shown in Figure F.5.

Real poles and zeros ($s = \sigma$) are plotted on the horizontal axis of the *s*-plane. Complex pole and zero pairs ($s = \sigma \pm j\omega$) are typically plotted showing only the positive-frequency root ($\sigma + j\omega$), with the negative-frequency root ($\sigma - j\omega$) implied. The poles are the roots of the denominator of the transfer function (each indicated by "X"), whereas the zeros are the roots of the numerator (each indicated by "0").

Poles that are farther to the left of the $j\omega$(vertical) axis represent modes that are more quickly damped than those nearer to the $j\omega$ axis. Poles that are on or to the right of the $j\omega$ axis represent unstable modes and, thus, indicate a control system that is unstable.

The locations of the open-loop poles and zeroes in Figure F.5 depend on the dynamic characteristics of transfer functions G1 and G2. Although the loop gain K has no effect on the open-loop poles and zeroes, it has a large effect on the closed-loop poles.

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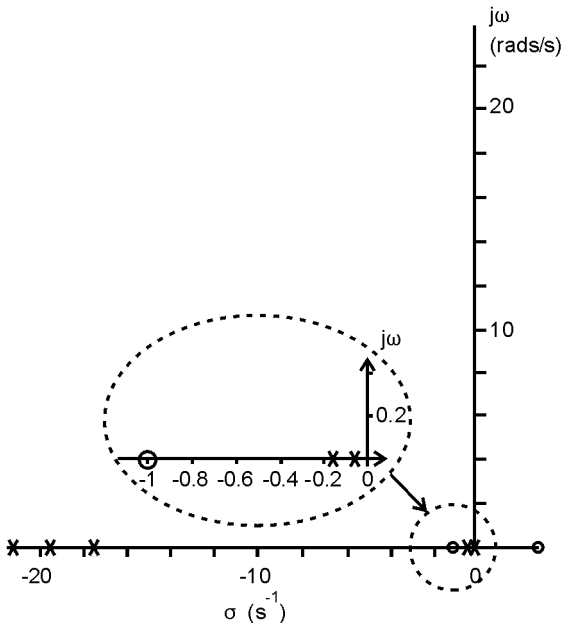


Figure F.5—Pole/zero plot of a typical open-loop turbine governing system

F.1.6 Root locus plot

A root locus plot typically maps the locations of the closed-loop *s*-plane poles as the loop gain is varied from zero to infinity. Figure F.6 shows a root locus plot of the governor control system with the open-loop characteristics of Figure F.5. The poles of the closed-loop system are plotted in the *s*-plane as the value of gain K is varied. At a value of gain $K = 0$, the closed-loop poles are the same as the open-loop system poles. At a value of gain $K = K_{co}$, complex poles cross over the $j\omega$ axis into the right half-plane, indicating instability. If the gain K is adjustable, the operating gain $K = K_{op}$ can be selected for acceptable gain margin (G_m) and damping ratio (ξ).

F.1.7 Computerized system simulation

The expected response of a turbine governing system can be predicted using a computerized system simulation. Mathematical models of the characteristics of the turbine, the water passages, the generator, the connected load, and the expected system disturbances are entered into the simulation software, and the predicted results can be displayed as a graphic. The correlation between the results of the simulation and the results achieved during actual field testing of the unit is dependent on the accuracy of the mathematical models used in the simulation.

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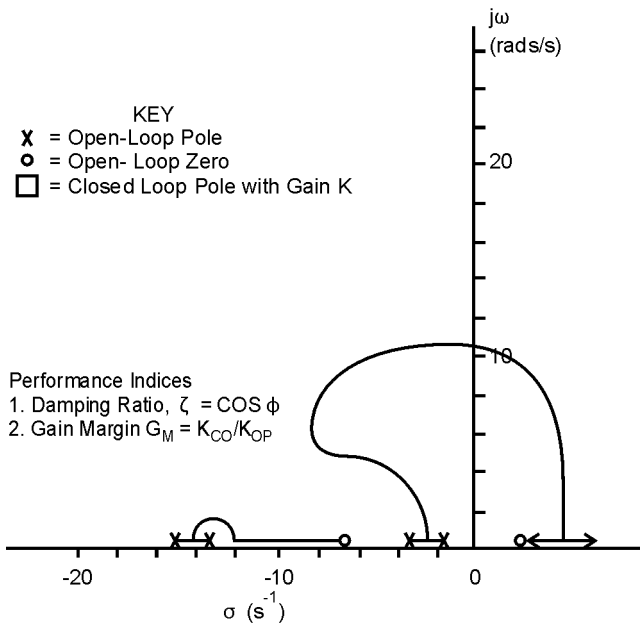


Figure F.6—Root locus plot of a typical closed-loop turbine governing system

F.2 Performance indices

Performance indices are representative parameters that can be either measured directly or computed indirectly from measured data on an operating unit. It is important to choose performance indices that can be obtained relatively easily. Generally accepted values of the performance indices described above characterizing good feedback control system performance are presented in Table F.1.

Table F.1—Generally accepted values of indices characterizing good feedback control system performance

Gain margin	6 dB
Phase margin	40°
Overshoot	0% to 15%
M_p	1.1 to 1.6 (0.8 dB to 4.0 dB)
Damping ratio	0.6 to 1.0

It is not possible to define such generally acceptable ranges of values for the other small signal performance indices such as rise time, settling time, and bandwidth. These indices are measures of relative speed and stability of control action. In most feedback control systems, they are determined primarily by the dynamic characteristics of the system element whose output is the ultimately controlled variable. In the case of a governor control system, the dynamic characteristics of the water column, synchronous machine (inertia constant, etc.), and the location in the power system where the machine is connected are the determining factors.

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It should be noted that simultaneous optimization of all small signal performance indices is not possible. For example, low M_p, high damping ratio, high gain margin, and high phase margin are not compatible with maximum bandwidth. The performance indices, which are of primary importance, depend on the individual application of each feedback control system, and no universally applicable "best criteria" can be recommended in standards.

Typical ranges of values of small signal performance indices for a governor control system are given in Table F.2. These data have been derived analytically using the anticipated extreme ranges (longest to shortest) of water starting time, T_W, synchronous machine inertia time constant, T_M (or $2H$), and all internal governor time constants likely to be encountered. Generally, values outside the ranges in Table F.2 should be avoided by adjusting control parameters; however, some governor systems may exhibit larger overshoot and smaller damping ratios, especially when the synchronous machine is disconnected from the power system.

Table F.2—Range of governor control system small signal dynamic performance

Performance index	Range of expected values
Gain margin	2 dB to 20 dB
Phase margin	20° to 80°
M_p	1 to 4 (0 dB to 12 dB)
Bandwidth	0.03 Hz to 1 Hz
Overshoot	0% to 40%
Rise time	1 s to 25 s
Settling time	2 s to 100 s
Damping ratio	0.25 to 1.0

The performance indices in the tables are applicable to any feedback control system having a single major feedback loop, that is, a single ultimately controlled variable. As such, they are applicable to a governor control system with the synchronous machine open-circuited or with the synchronous machine serving an isolated load. A loaded synchronous machine connected to a power system is a complex multiloop, multivariable feedback control system.

For a loaded synchronous machine connected to a multi-machine interconnected power system, performance indices such as those in the tables lose much of their significance. Analysis and synthesis techniques that are applicable to these types of systems should be used. One approach to assessing the performance of this complex system is to model the system using state-space techniques and calculate the eigenvalues (characteristic equation roots) for the range of system parameters of interest. The state-space model can be derived from known plant and system parameters, operating conditions, and experimental frequency response data, if the latter is available. The calculation of eigenvalues gives a direct indication of system stability for the linearized system and serves as an efficient initial step before testing the selected parameters in a more extensive study such as a transient stability study. The quadratic performance index and other indices that use the state-space models are measures of the ability of a multiloop control system to meet specified criteria.

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F.2.1 Practical time domain stability indices

Stability is defined as the capability of the governor and the system controlled by the governor to limit and damp speed oscillations following rejection of load and under isolated load conditions following sudden load changes. The magnitude of load changes and the type of isolated load (resistive, inductive, and capacitive) should be specified. Suitably damped response in the time domain is demonstrated by attenuation of successive speed deviations following the initial speed change. The relative magnitude of successive oscillations is determined by the damping ratio. Some commonly referenced damping ratios are as follows:

a) Damping ratio = 1.000; results in no overshoot (critically damped response)

b) Damping ratio = 0.707; results in each overshoot/undershoot equal to 25% of the previous undershoot/overshoot (quarter decay response)

c) Damping ratio = 0.000; results in each overshoot/undershoot equal to the previous undershoot/ overshoot (unstable operation)

It is possible under most circumstances to adjust a PID governor (or any governor with two tunable zeroes in its transfer function) to give "quarter decay" response for all values of T_M and T_W under worst-case resistive load conditions. Assuming optimal setting of the PID elements for quarter decay response, the initial overshoot and settling time are determined by the values of T_M and T_W. Normally, high values of T_W and low values of T_M create larger magnitudes of initial overshoot and settling time.

F.2.2 Practical frequency domain stability indices

Frequency domain stability indices are useful for determining the range and effectiveness of the governor damping capabilities for power system stability, and they are useful for setting governor gains. They are generally determined by a combination of modeling and simulation, although online measurements using swept-sine or white noise techniques can also be used. Online techniques are generally only practical when the unit is operating into a small or isolated load system, due to practical limitations of measuring frequency deviations on a large interconnected system. Frequency domain tools are generally limited to small signal analysis. Therefore, large signal time responses should be performed to validate governor parameters established using frequency domain methods. Also, due to nonlinearities in the turbine, the values can change substantially through the operating range of the turbine. These changes in unit characteristics should be considered when identifying the applicable models for a given unit.

The following characteristics are defined for the open-loop frequency response of the governor and turbine from the speed input of the governor to the speed of the turbine shaft at the least stable operating point and at signal levels within the linear response range:

a) Gain Margin at Phase Crossover: The gain margin is the amount that the gain can be raised before instability results. On a Bode plot, it is the amount that the gain is below 0 dB at the frequency where the output signal lags the input signal by 180 degrees.

b) Phase Margin at Gain Crossover: The phase margin is the amount that the phase is above –180 degrees when the system gain drops to 0 dB (unity gain).

Generally, the least stable operating condition for a turbine governing system is when a single unit is feeding an isolated electrical load at a high gate (or needle) opening (e.g., 80% or greater). Performing online tests under these conditions is typically difficult to arrange. This tends to limit the practicality of online frequency domain tests.

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F.3 Governor tuning for isolated operation

It is often stated that when a generator is connected to a large power system, its governor no longer controls speed, but it controls power. Although this is true from the perspective of the individual unit operation, the composite speed governing of all of the units connected to the system is one of the factors that creates the constant speed of the large system. The other factors are probability (load fluctuations tend to cancel each other) and system size (any one load is not large enough to cause a major impact upon the system). When the power system undergoes a major disturbance, governors may be the only one of these factors left. Therefore, governors of key facilities in all possible system islands should be adjusted to enable generator operation while carrying an isolated, inertialess, resistive load.

The isolated, inertialess, resistive load is one of the most difficult connections to govern. Isolation means that there is no other generator connected to the load to restore the system frequency. An inertialess load gives the smallest system inertia constant (only that of the generator) and, therefore, the fastest rate of change of speed. A resistive load has special properties due to the high gain voltage regulators used on most generators and the relationship between power and torque.

With a high gain voltage regulator, a generator maintains essentially constant terminal voltage regardless of speed. With constant applied voltage, a resistive load consumes constant electric power regardless of the frequency of the turbine-generator output. Therefore, a resistive load is in essence a constant power load when connected to an isolated generator. As power is equal to torque multiplied by speed, a resistive load contributes negative damping torque to oscillations in speed. That is, when speed increases, the electrical (braking) torque of the load on the generator decreases. This tends to cause a further increase in speed. Governors of key facilities should be tuned to control this otherwise unstable situation.

When governors for hydroelectric power plants are tuned to produce stable operation with an isolated, inertialess, resistive load, the overall open-loop transfer functions (governor, water column, turbine, and rotating inertia) have characteristics similar to the Bode plot shown in Figure F.7. The crossover of gain and phase occurs at approximately 0.07 Hz. Gain and phase margins are small but positive in this full load example. These gain and phase margins generally increase at lower loading levels.

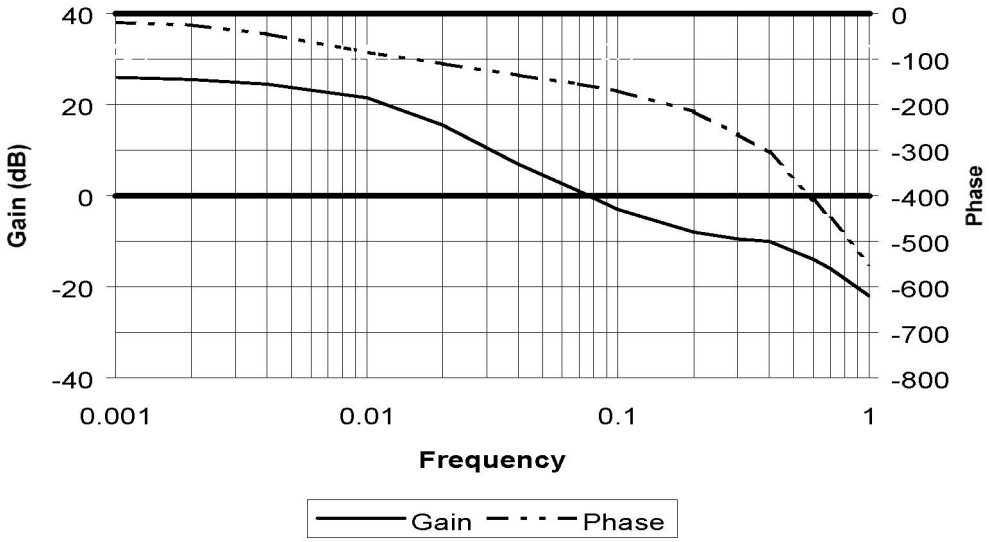


Figure F.7—Typical overall open-loop frequency response of an isolated unit

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The governor alone (including the actuator) generally has a frequency response characteristic similar to Figure F.8. The low-frequency pole (0.007 Hz) reduces the transient gain of the system. The rise in gain between 0.1 Hz and 1 Hz (from phase lead compensation) is necessary to keep the phase compensation correct in the previous decade (between 0.01 Hz and 0.1 Hz).

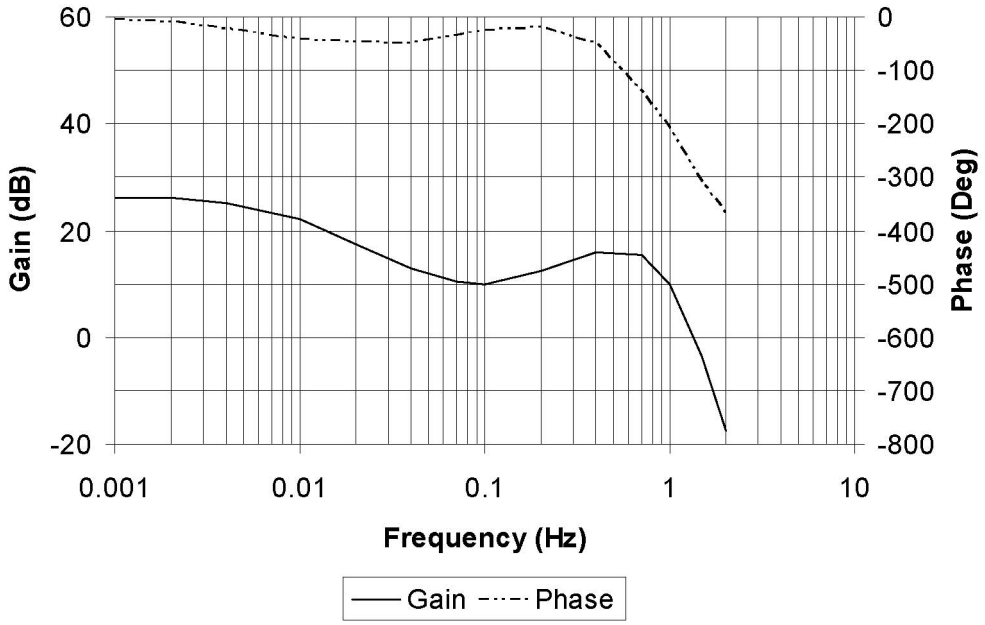


Figure F.8—Typical frequency response of a governor tuned for isolated operation

The transfer function of the water column is shown in Figure F.9. The gain is essentially constant, whereas the phase lags significantly at frequencies above 0.07 Hz. With larger values of T_W, the gain may even increase as the phase is lagging.

For ideal governing, mechanical (accelerating) torque changes should respond in phase with electrical (braking) torque changes. Speed would never change if this was possible. Practical governing provides mechanical torque responses to electrical torque changes that lag by less than 180 degrees for all frequencies where the control loop gain is greater than unity. For higher frequencies where it is not possible to provide a proper phase relationship, the control loop gain should be reduced below unity to maintain a stable control system.

F.4 Impact on power system performance

For optimum damping of power system oscillations, mechanical torque changes due to governing action should respond in exactly opposite phase from speed changes. This principle applies both to isolated operation and to operation into an interconnected system. This governing action is only physically possible for very low frequencies. However, some damping is provided as long as the phase lag from negative speed changes to mechanical torque is less than 90 degrees. Governing action contributes to damping at frequencies below 0.1 Hz for the governor response shown in Figure F.10, which includes the response of the controller, the actuator, and the water column. (Governing action is not the only source of damping for power system oscillations, because generator damping windings, field windings, and power system stabilizers all contribute to damping within their respectively achievable frequency ranges; see. Frizell and Agee [B12], Kundur [B32], and Kundur et al. [B33].)

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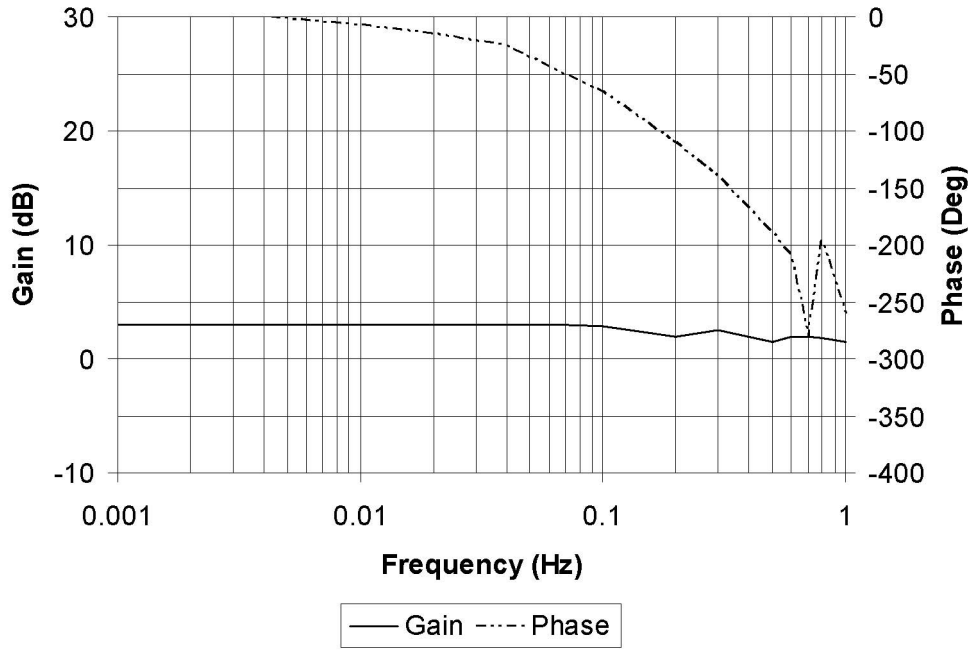


Figure F.9—Typical frequency response of a hydraulic turbine water column

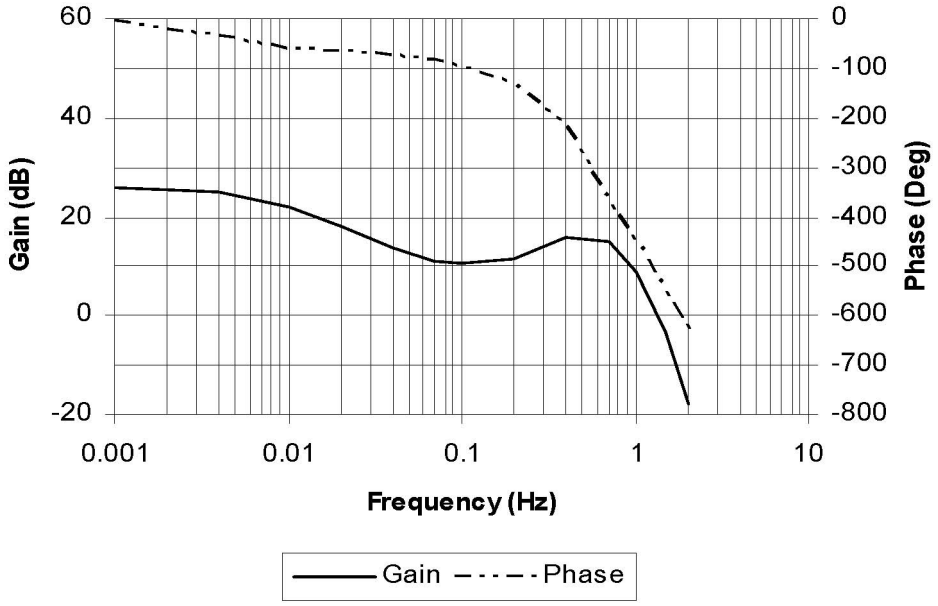


Figure F.10—Typical frequency response from speed to torque for a hydraulic turbine

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At frequencies above 0.1 Hz, the phase of mechanical torque drops very rapidly due to water column effects; however, the gain rises because the water column gain is essentially constant and the electronic governor gain is increasing. Considering a single governor embedded in a power system, it is possible for the governor to contribute to either damping or undamping the system oscillations at these higher frequencies, depending on the exact frequency of the oscillation. Fortunately, the other sources of power system damping can provide damping over a wider frequency range and can overwhelm any undamping caused by the governor system.

Although governors cannot provide damping at all frequencies, they can contribute to the stable control of steady-state system frequency. When the system is taken as a whole, governor mechanical torque changes should act through the inertia of the entire power system to produce changes in system frequency. Thus, the open-loop system returns to a form similar to that shown in Figure F.7. The system is generally stable if the system inertia constant is similar to the individual unit inertia that was used as the basis of the governor tuning.

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Annex G

(informative)

Verification of turbine governing system performance[11]

G.1 Background

With a global trend toward restructuring of the electric utility industry, increasing attention is being given to the performance of equipment controlling the dynamic performance of generating units. Reliable and repeatable performance of this equipment, which includes the turbine governor, is essential to the reliable performance of the utility grid to which the generating unit is connected (see EPRI TR-101080 [B11]). There are numerous documented instances where widespread system outages were either attributed to poor governor performance or poor governor performance was identified as a contributing factor. To aid in the process of identifying sources of poor or marginal governor performance and taking appropriate remedial measures, a number of countries have either instituted or are contemplating establishing programs for governor system performance testing. Several countries have programs in place. In North America, proposals are being considered to institute such programs. This annex is a status report on the nature of such programs principally based on proposals under consideration in North America.

G.2 Approach

In North America, proposals have been initiated to formalize system modeling data requirements and include recommendations for validating generator modeling data (including hydraulic turbine governors) through field verification and testing. Recent developments in the evolving restructuring and reorganization of the North American utility industry plus ongoing system analysis work of some North American regional reliability councils expand on the original proposals. Processes are in place that may lead in the future to mandated testing requirements, with the possibility of incentives for testing compliance.

System planners have developed an inventory of standardized block diagrams for hydraulic turbine governors that are used in a variety of system stability studies. These models have been refined and additional models introduced over the past three decades as needs for better representation for short- and long-term dynamic stability studies became evident. Most North American reliability council study groups use models summarized in work developed by the Power System Dynamic Performance Committee and published as a IEEE Working Group Report “Hydraulic Turbine and Turbine Control Models for Power System Studies” [B31]. The thrust of proposed testing and validation measures for hydraulic turbine governors is to have accurate representations of governors in a power plant conform to one of the models contained in the Working Group report cited (see Dandeno et al. [B8], Hannett and Fardnish [B14], and IEEE Working Group Report [B30]).

[11]Because of the evolving nature of a number of nontechnical factors affecting this topic, this annex reflects the current status of governor system performance testing in North America as of the publication of this document.

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Typically, a proposed program for validating hydraulic turbine governor performance might consist of the following:

a) Develop required supplementary new dynamic data regarding control area hydraulic turbine governors for incorporation in the power system stability model database. Work includes the following:
   1) Ensure proper governor models are included in the system stability study inventory.
   2) Verify model response versus actual unit tests.
   3) If possible, validate model response using actual disturbance data.

b) Identify governor parameters affecting governor response, which are adjustable, and establish a relationship between these settings and model parameters.

c) Perform necessary transient and mid-term stability studies to determine impact of governor settings on operational stability enhancement and control procedures to include
   1) Load shedding procedures
   2) Regional intertie transfer capabilities under import and export conditions
   3) AGC
   4) Unit load rejection performance

d) From the preceding steps, develop recommended governor settings and perform necessary transient and mid-term stability studies using recommended settings to optimize and verify the proposed settings under a variety of system conditions.

e) Implement recommend settings and establish a continuing testing and monitoring program to include
   1) Developing guidelines for governor tuning and testing applicable to the generation control area
   2) Conducting plant tests to verify actual governor performance agrees with study results
   3) Performing governor or model remedial work as necessary to obtain agreement in performance between the adopted model and the actual equipment
   4) Performing system tests to verify governor response to system AGC signals
   5) Recording and analyzing governor performance during actual system disturbances
   6) Establishing procedures for periodic testing of governor performance to ensure compliance with established control area criteria

Subclause G.3 contains an example of a typical procedure for testing hydraulic turbine governors developed by one North American utility.

It should be noted that no entity in North America has yet initiated a vigorous comprehensive testing program such as the one just outlined as of the publication of this document. Within North America, although some utilities and some regional reliability entities have initiated voluntary efforts in the governor testing and validation arena, most are awaiting further refinement and clarification of the regulatory processes as they evolve with utility industry restructuring before initiating efforts to validate performance.

G.3 Example of a performance testing procedure

G.3.1 Identifying servomotor time constants and permanent speed droop

With the generator operating at moderate load

a) Disable the dashpot by opening the dashpot needle valve to give the shortest possible dashpot time constant. In electronic dashpot-analogy governors, set the relaxation time to its minimum value.

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Carefully note the initial setting before making the adjustment so that the original setting can be restored after the test.

b) Increase the speed reference (speed adjustment) by a small amount to induce a step change into the system.
   1) Record the gate position signal versus time.
   2) Permanent speed droop is determined by the change in speed reference divided by the steady-state change in gate position. Both of these values are commonly expressed in per-unit terms.
   3) Servomotor time constants are determined by trial-and-error trying to match simulations to the actual recorded gate position response. This matching process can be done with a power system stability program or with a general-purpose software modeling tool.

G.3.2 Identifying temporary droop and dashpot relaxation time constant

With the generator operating at moderate load

a) Return the dashpot to the initial position.

b) Manually, quickly increase the speed reference by a small amount to induce a step change into the system.

c) Temporary droop and dashpot time constants are determined by trial-and-error trying to match simulations to the recorded gate position. This matching process can be done with a power system stability program or a general-purpose software modeling tool.

G.3.3 Identifying maximum gate opening and closing rates

With the generator operating at moderate load

a) Lower the gate limit (at the governor cabinet) quickly to about 10% to 20% lower than the initial gate position.
   1) Record the gate position signal versus time.
   2) Measure the maximum gate closing rate directly from the test record.

b) Raise the gate limit quickly above the initial gate position.
   1) Record the gate position signal versus time.
   2) Measure the maximum gate opening rate directly from the test record.

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Annex H

(informative)

Techniques for evaluating speed control performance of turbine governing systems

This annex summarizes some techniques that may be used to evaluate the speed control performance of a turbine governing system.

H.1 Tests with isolated resistive load

Where a controllable resistive load is available, such as an electrolytic plant, a water rheostat, or a DC transmission terminal, response of speed following either a small or a large step change of load is the simplest, most conclusive test available. The contribution of self-regulating influences from the turbine is automatically included. Test results are in the time domain.

H.2 Tests by simulated isolation

This test technique permits operating the turbine generating unit in parallel with an interconnected power system, and it allows wide flexibility in load magnitude and representation of load characteristics. The generator power output to the system is measured, and that signal is integrated to produce a simulated speed signal just as the actual turbine torque deviations (Δm) would be integrated by the Wk^2 of the turbine and generator in the speed response of the unit when isolated. The integration rate or gain is thus $1/T_M$ times the torque deviation. The computation of the per-unit speed deviation using this technique is illustrated as follows:

$$\Delta n = \frac{1}{T_M}\int_O^T \Delta m\, dt \tag{6}$$

where

Δn is the per-unit speed deviation,

T_M is the mechanical starting time of the turbine and generator, s,

T is the elapsed time, s,

Δm is the per-unit torque deviation,

t is the time, s.

The power deviation signal to be integrated is the difference between total power output and an adjustable reference. When the simulated speed signal so produced is substituted for the actual speed deviation signal, the turbine governor responds to the change of the simulator power reference just as an isolated unit responds to a change of load. The simulated speed deviation signal may also be delivered to the governor input in addition to the regular speed deviation signal because the latter remains essentially constant while the test unit remains synchronized to the power system.

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For simulating a purely resistive load, a positive (destabilizing) feedback from the simulated signal to the junction for summing the power signal and its reference is necessary to make the power deviation proportional to the product of speed and torque. The turbine self-regulating effect can also be represented by a negative (stabilizing) feedback from the simulated speed deviation signal to the junction for summing the power signal and its reference. Net feedback to be represented is the sum of these two terms. With Francis type turbines near full generation output, the self-regulation is near unity, and these two effects essentially cancel each other. Other load regulations and turbine self-regulations can be modeled by adjusting the net feedback, if desired. Inertia in the load, along with inertia of the unit, can be reflected in the integration rate as appropriate for representing other than resistive loads (for example, rotating loads).

Simulated isolation tests are performed by applying step inputs to the adjustable power reference. Transient response of the simulated speed deviation signal following the step changes is observed in the time domain. The functions are illustrated in Figure H.1.

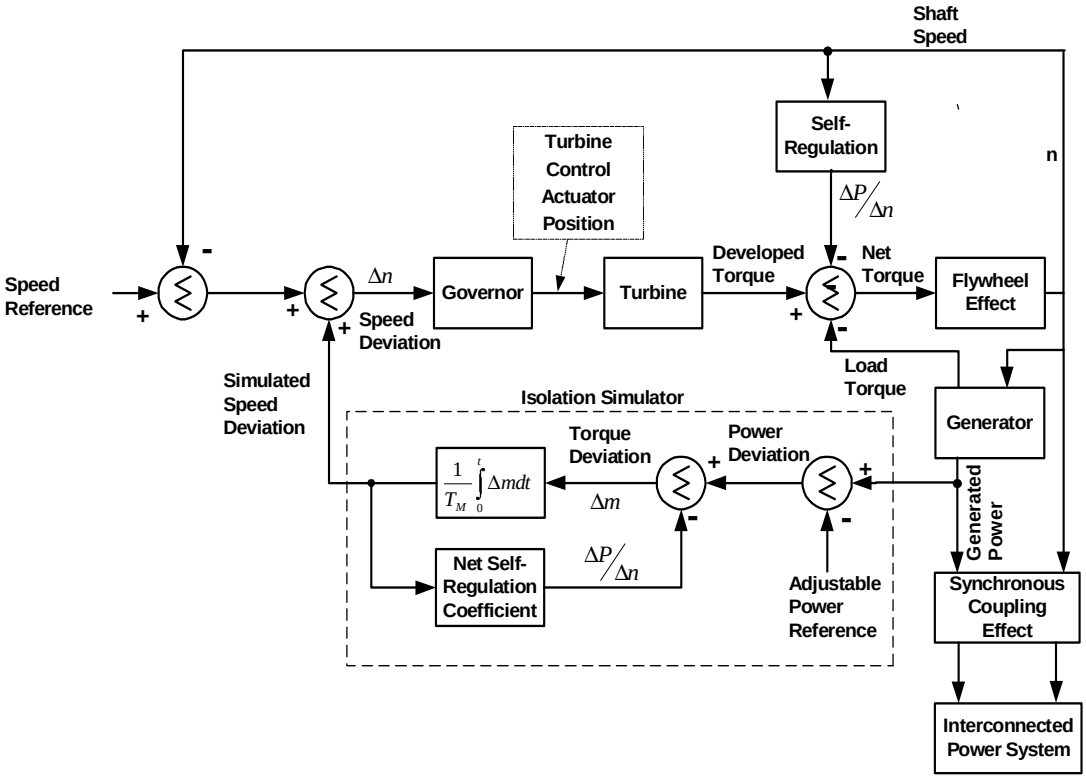


Figure H.1—Governor speed control loop closed through isolation simulator

H.3 Damping determinations in the frequency domain

Frequency response may be determined purely by computation, but accuracy of the method hinges on the accuracy with which characteristics of all components are known. Also, the influence of inadvertent nonlinearities in equipment such as valves and dashpots is not revealed.

Measurement of the frequency response of the governor and turbine from the governor speed input to generator power output, whereas the unit is operating in parallel with a power system is more direct and representative in that all actual characteristics are included in the measurement, except the influence of turbine self-regulation. Influence of the rotational inertia of the unit and turbine self-regulation should be computed, but the computation is less tedious, and error from inadvertent nonlinearities is less likely.

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Frequency domain-simulated isolation tests may also be performed. This is most easily accomplished in two steps. The first step involves inserting a sinusoidally varying input into the adjustable power reference of the isolation simulator. The frequency response from the simulated speed signal to gate position is taken during this step.

The second step involves inserting a sinusoidally varying input into the governor speed summing junction. The frequency response from gate position to the simulated sped signal is taken during this step. By cascading these two frequency responses, the open-loop frequency response from governor speed input to shaft speed can be obtained. Gain and phase margins can then be determined directly from this open-loop response. Various load regulation and load inertia properties can also be included in this test.

H.4 Computation of time response

This computation, which may be made by analog or digital computer, consists of computing performance of a mathematical model of the closed-loop control system. It requires accurate knowledge of the characteristics of the governor system and turbine self-regulating characteristics or the complete turbine performance data.

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