

INTERNATIONAL STANDARD

NORME INTERNATIONALE



Guidelines to specification of hydraulic turbine governing systems

Lignes directrices pour la spécification des systèmes de régulation des turbines hydrauliques



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INTERNATIONAL
ELECTROTECHNICAL
COMMISSION

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IEC 61362 has been prepared by IEC technical committee 4: Hydraulic turbines. It is an International Standard.

This third edition cancels and replaces the second edition published in 2012. This edition constitutes a technical revision.

This edition includes the following significant technical changes with respect to the previous edition:

- a) adoption of parts of IEC 60308:2005 which deal with specification matters;
- b) introduction of several new technical topics;
- c) overall editorial revision.

The text of this document is based on the following documents:

Draft	Report on voting
4/500/FDIS	4/509/RVD

Full information on the voting for its approval can be found in the report on voting indicated in the above table.

The language used for the development of this International Standard is English.

This document was drafted in accordance with ISO/IEC Directives, Part 2, and developed in accordance with ISO/IEC Directives, Part 1 and ISO/IEC Directives, IEC Supplement, available at www.iec.ch/members_experts/refdocs. The main document types developed by IEC are described in greater detail at www.iec.ch/publications.

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INTRODUCTION

While a standard for the testing of hydraulic turbine governing systems had been existing for a very long time (IEC 60308 published in 1970)¹, guidelines for the specification of hydraulic turbine governing systems were missing until 1998. The need for such guidelines became more and more urgent with the fast development and the new possibilities especially of the digital components of the governor.

While the first edition was written more or less as a supplement to the already existing guide for testing, the objective of the second edition was to be the leading guide with respect to turbine governing systems.

The second edition of this document took into account the experience with the guide until 2012 as well as the progress in the state of the art of the underlying technologies.

This third edition was developed together with the third edition of the standard for the testing of hydraulic turbine governing systems (IEC 60308) in order to harmonize their contents and their publishing dates.

Furthermore, the standards are kept open for state of the art by introducing new topics and harmonizing the structure as well as the terms and definitions for both standards.

¹ IEC 60308:1970, *International code for testing of speed governing systems for hydraulic turbines*. This publication was withdrawn and replaced with IEC 60308:2005.

GUIDELINES TO SPECIFICATION OF HYDRAULIC TURBINE GOVERNING SYSTEMS

1 Scope

This document includes relevant technical data used to describe hydraulic turbine governing systems and to define their performance. It is aimed at unifying and thus facilitating the selection of relevant parameters in bidding specifications and technical bids. It serves also as a basis for setting up technical guarantees.

The scope of this document is restricted to the turbine governing level. Additionally some remarks about the control loops of the plant level and about interactions with the electrical grid in case of primary and secondary frequency control (see also Annex B and Annex C) are made for better understanding without making a claim to be complete.

Important topics covered by the guidelines are:

- speed, power, water level, opening and flow (discharge) control for reaction and impulse-type turbines including double regulated machines;
- means of providing actuating energy;
- safety devices for emergency shutdown, etc.

To facilitate the setting up of specifications, these guidelines also include data sheets, which are filled out by the customer and the supplier in the various stages of the project and the contract.

Acceptance tests and specific test procedures are outside the scope of this document; those topics are covered by IEC 60308.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

IEC 60308, *Hydraulic turbines – Testing of control systems*

IEC 61131-2, *Industrial-process measurement and control – Programmable controllers – Part 2: Equipment requirements and tests*

3 Terms and definitions

For the purposes of this document, the following terms and definitions apply.

ISO and IEC maintain terminology databases for use in standardization at the following addresses:

- IEC Electropedia: available at <https://www.electropedia.org/>
- ISO Online browsing platform: available at <https://www.iso.org/obp>

3.1 General terms and definitions

3.1.1

turbine governing system

technical equipment governing the opening (guide vane, runner blade, needle, deflector position) of hydraulic turbines

Note 1 to entry: At the present state of the art, the turbine governing system consists of an oil hydraulic and an electronic part, the "oil hydraulic governor" and the "electronic governor" and an interface between both, the electro/hydraulic converter.

3.1.2

controlled system

system controlled by the actuators of the governing system consisting of the hydraulic turbine, its water supply and discharge passages, the generator with voltage regulator and the electric power grid to which it is connected

3.2 Terms and definitions related to control levels, control modes and operational modes

3.2.1

control levels

3.2.1.1

turbine governing level

control functions directly related to the governing system of a single turbine

Note 1 to entry: The following control modes are related to the turbine governing level:

- speed control;
- power output control;
- water level control;
- opening control;
- flow control (the term flow used in this document has the same meaning as the term discharge).

3.2.1.2

unit control level

control functions directly related to the overall control of a single unit (turbine, generator, unit auxiliaries) including turbine governing, voltage regulation, start-stop-sequencing etc.

3.2.1.3

plant control level

control functions related to the overall control of a whole plant including the control of several units

Note 1 to entry: In automatic unit and plant control operation, the turbine governing system gets its modes and set-points from the unit and plant control level.

3.2.1.4

grid control level

control functions related to the overall control of the grid as a whole

Note 1 to entry: If applicable the turbine governing system participates either by primary or by secondary frequency control, or both (see Annex B).

3.2.2

control modes at the turbine governing level

3.2.2.1

speed control

mode of the governing system dealing with the control of the speed of the turbine

3.2.2.2

power output control

mode of the governing system dealing with the control of the power output of the generator

3.2.2.3

water level control

mode of the governing system dealing with the control of the water level of the headwater reservoir

3.2.2.4

opening control

mode of the governing system dealing with the control of the position of the main actuator(s) of the turbine

3.2.2.5

flow control

mode of the governing system dealing with the control of the flow through the turbine

3.2.3

main operation modes

3.2.3.1

no-load operation

mode of the governing system when the unit is not connected to a grid

3.2.3.2

island operation

operation of a generating unit that is interconnected with a relatively small number of other generating units

Note 1 to entry: Such a small number can occur after inadvertent tripping of circuit breakers that interconnect the island with a large interconnected power system.

3.2.3.3

isolated operation

specific case of islanded operation consisting of a single generating unit

3.2.3.4

grid operation

mode of the governing system when the unit is connected to a stable grid

3.3 Terms and definitions from control theory

3.3.1

differential equation

equation describing the dynamic system behaviour in the time-domain, as shown in Annex A

3.3.2

transient response

system response (output) to a step change of the input

3.3.3

frequency response

for a linear time-variant system with a sinusoidal input variable in steady-state the ratio of the phasor of the output variable to the phasor of the corresponding input variable, represented as a function of the angular frequency ω

Note 1 to entry: The frequency response coincides with the transfer function taken on the imaginary axis of the complex plane.

3.3.4**transfer function**

for a linear time-invariant system the ratio of the Laplace transformation of an output variable to the Laplace transform of the corresponding input variable, with all initial values equal to zero

Note 1 to entry: The Laplace operator s is the complex variable of Laplace transform, only used for transfer functions in the frequency domain.

3.4 Subscripts and prefixes

Sub-clause	Subscript/prefix related to	Definition	Symbol	Unit
3.4.1	rated	subscript indicating the rated operation point of the system	r	–
3.4.2	maximum minimum	subscript indicating maximum or minimum values of any term	max. min.	–
3.4.3	deviation	Prefix indicating the deviation of any value from a defined value (in case of linearization the defined value is the steady-state value)	Δ	–
3.4.4	wicket gate	subscript associating a quantity to wicket gate (guide vane) position	ga	–
3.4.5	runner	subscript associating a quantity to runner blade position	ru	–
3.4.6	needle	subscript associating a quantity to needle position	ne	–
3.4.7	deflector	subscript associating a quantity to deflector position	de	–
3.4.8	main inlet valve	subscript associating a quantity to main inlet valve	miv	-

3.5 Terms and definitions related to the plant and the units

Sub-clause	Term	Definition	Symbol	Unit
3.5.1	specific hydraulic energy of unit	specific energy of hydraulic water available between the high- and low-pressure side sections of the unit	E	$\text{J} \cdot \text{kg}^{-1}$
3.5.2	turbine head	$H = E/g$ definition of E , see 3.5.1 g = acceleration due to gravity = $9,81 \text{ m} \cdot \text{s}^{-2}$ (at sea level)	H	m
3.5.3	water level	elevation above the reference datum (usually mean sea level)	z	m
3.5.4	flow	volume of water per unit time flowing through any section in the system	Q	$\text{m}^3 \cdot \text{s}^{-1}$
3.5.5	rotational speed	number of revolutions per unit time Instead of s^{-1} the unit rpm is frequently used.	n	s^{-1}
3.5.6	frequency	number of cycles per second	f	Hz
3.5.7	generator power output	generator power measured at generator terminals	P_G	W
3.5.8	moment of inertia of mass	moment of inertia for calculation of fly-wheel effect. $I = M \times D^2/4 = M \times R^2$ (M = mass, D = diameter of gyration, R = radius of gyration)	I	$\text{kg} \cdot \text{m}^2$
3.5.9	torque deviation	power output deviation divided by instantaneous angular speed	ΔM	N·m
3.5.10	unit acceleration constant	ratio of the angular momentum of the unit to the rated torque	T_a	s
3.5.11	load acceleration constant	ratio of the angular momentum, caused by the power grid referred to the rated torque of the unit	T_b	s

Sub-clause	Term	Definition	Symbol	Unit
3.5.12	turbine control transmission ratio alternate term: turbine gain	At a considered servomotor position, the slope of the graph relating to the turbine torque m_t at constant speed and head to servomotor movement y (see Figure 1) $e_y = \frac{d(M_t/M_r)}{dy} = \frac{dm_t}{dy}$	e_y	
3.5.13	controlled system self-regulation factor	at the speed considered, the slope of the graph relating to the torque deviation to the speed at a specified servomotor position and a specified load condition of the grid. $e_n = e_g - e_t$ The torque m should be referred to the rated torque $M_r = P_r/\omega_r$ and the speed x_n referred to the rated speed ω_r (see Figure 2)	e_n	
3.5.14	turbine self-regulation factor	component of e_n due to the turbine (index t) $e_t = \frac{dm_t}{dx_n} = \frac{\omega_r}{P_r} \times \frac{d(P_t/\omega)}{dx_n}$ P_r and ω_r are the same reference values as used to determine e_n (see Figure 2)		
3.5.15	load self-regulation factor	component of e_n due to the load (index g) $e_g = \frac{dm_g}{dx_n} = \frac{\omega_r}{P_r} \times \frac{d(P_g/\omega)}{dx_n}$ Error! Bookmark not defined. P_r and ω_r are the same reference values as used to determine e_n (see Figure 2)		
3.5.16	penstock reflection time	time required for the pressure waves to travel 2 lengths of the penstock: $T_r = 2 \sum_{i=1}^n \frac{L_i}{a_i}$ where a_i is the velocity of wave propagation in each section of the penstock; L_i is the length of each penstock section.	T_r	s
3.5.17	water inertia time alternative term: water starting time	characteristic time at rated condition due to inertia of the water in the water passages defined as: $T_W = \frac{Q_r}{gH_r} \sum_{i=1}^n \frac{L_i}{A_i}$ where A_i is the area of each section; L_i is the corresponding length; Q_r is the rated discharge; H_r is the rated net head; g is the gravity constant; n is number of penstock sections including spiral case and draft tube. If T_W is calculated for other than rated conditions, this shall be mentioned clearly.	T_W	s
3.5.18	water hammer number alternative term: Allievi constant	ratio of water inertia time, T_W , to penstock reflection time, T_r at rated conditions. $h_W = \frac{T_W}{T_r}$		

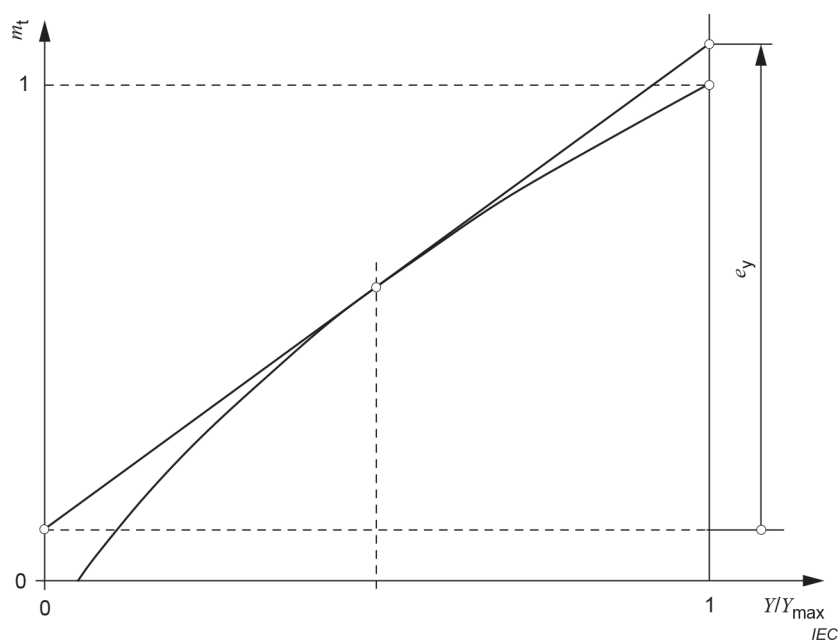


Figure 1 – Turbine control transmission ratio

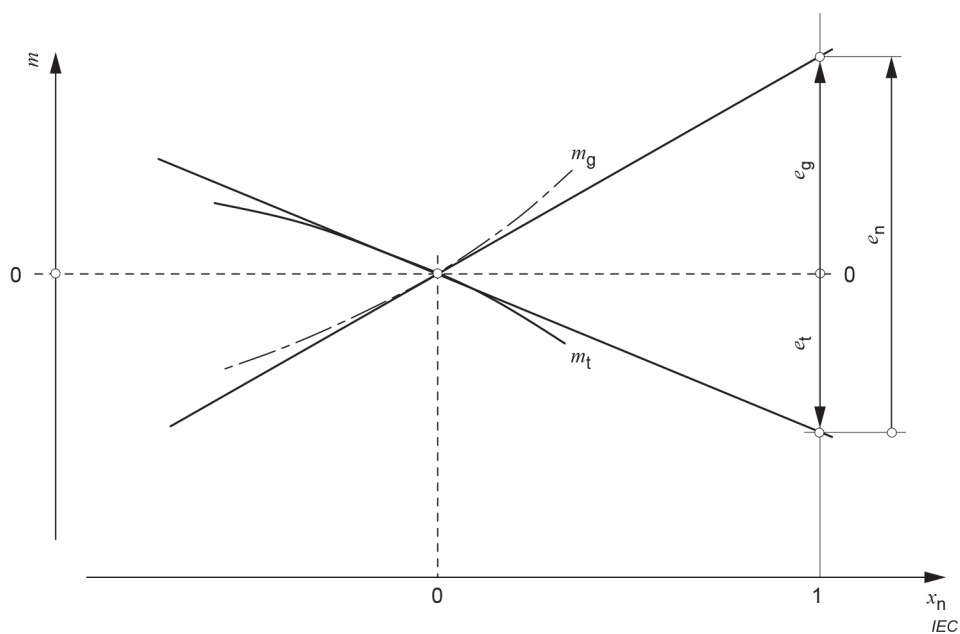


Figure 2 – Controlled system self-regulation factor

3.6 Terms and definitions related to the governing system

Sub-clause	Term	Definition	Symbol	Unit
3.6.1	controlled variable	variable which shall be controlled such as – absolute, dimensional value – relative value the relative value is calculated as absolute value related to the rated value, $x = X/X_r$ for – rotational speed – power output – opening – flow deviating from the definition above the relative value for water level z is calculated as: $x_z = (z - z_{\min})/(z_{\max} - z_{\min})$	X x x_n x_p x_y x_q x_z	var. – – – – – –
3.6.2	command signal	signal which can be set by an external adjustment: – absolute, dimensional value – relative value the relative value is calculated as absolute value related to the rated value, $c = C/C_r$ for – rotational speed – power output – opening – flow deviating from the definition above the relative value for water level z is calculated as: $c_z = (z - z_{\min})/(z_{\max} - z_{\min})$	C c c_n c_p c_y c_q c_z	var. – – – – – –
3.6.3	servomotor stroke	stroke of the main servomotor, either as – absolute value – relative value The servomotor moves either the – guide vane – runner blades – needles – deflectors the relative value is calculated as absolute value related to the maximum value, $y = Y/Y_{\max}$ Note 1 to entry The effective maximum servomotor stroke $Y_{\max}$ is the maximum mechanical stroke after commissioning. Taking the kinematics into account, it can also be related to the opening, see 5.4.5	Y y	m –
3.6.4	servomotor stroke set-point	set-point for the servomotor stroke (relative value)	y_o	–
3.6.5	main control valve stroke	stroke of the main control valve, either as – absolute value – relative value Note 1 to entry The effective maximum main control valve stroke is the maximum mechanical stroke after commissioning.	Y_v y_v	
3.6.6	controlled variable range	adjusting range for the setting of a controlled variable, consisting of – maximum value of the controlled variable – minimum value of the controlled variable Note 1 to entry In speed control or water level control, the controlled variable range depends on the average setting of the permanent droop (Figure 3).	$X_{\max}$ $X_{\min}$	– –

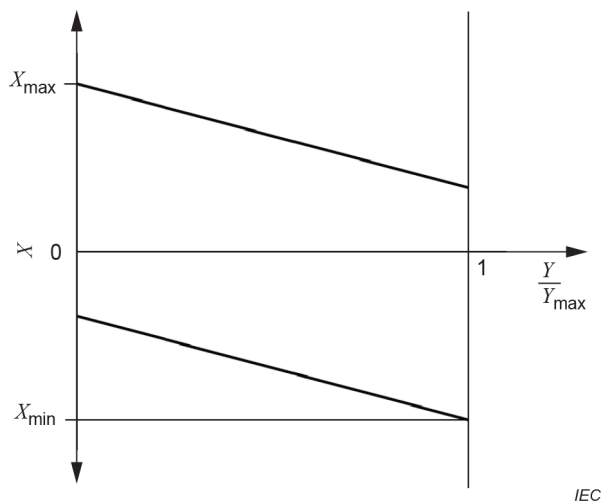


Figure 3 – Controlled variable range

Sub-clause	Term	Definition	Symbol	Unit
3.6.7	electronic governor output signal	output signal at the electronic governor corresponding to the input signal of the following servo-positioner as a relative value	<i>m</i>	–
3.6.8	valve position	position of the controlling valve or the pilot valve	<i>u</i>	–

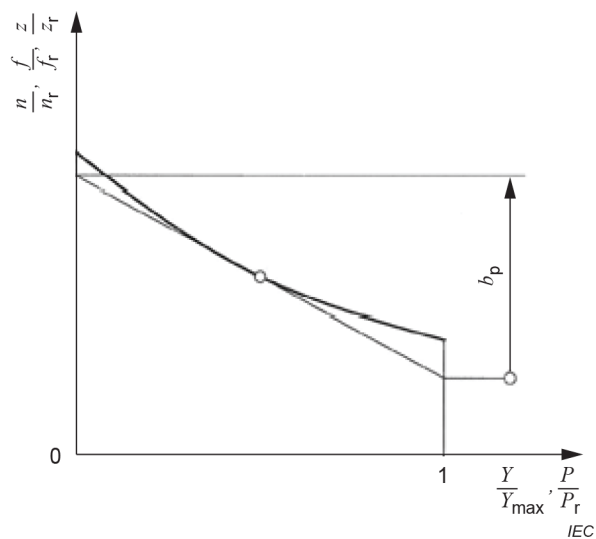


Figure 4 – Permanent droop

Sub-clause	Term	Definition	Symbol	Unit
3.6.9	permanent droop	function showing the relationship between – speed, frequency or water level versus – opening or power under steady state conditions, either – the slope of the function at a specific point of operation or – the slope of the straight line between the endpoints of the function SEE: Figure 4 Note 1 to entry Wherever the term "droop" is used throughout this document, the definition of "permanent droop" applies. The "temporary droop" of the past is not defined and not used anymore in the document. Note 2 to entry The following specific droops are practically used: – frequency-power droop or speed-power droop, – frequency-opening droop or speed-opening droop, – level-opening droop, – level-power droop.	b_p	%
3.6.10	proportional action coefficient alternative term: proportional gain	quotient of the variation of the output variable to the corresponding variation of the input variable for a proportional element Note 1 to entry As shown in Figure 5 the proportional action coefficient produces a controlling action that is proportional to the magnitude of the controller error in a PI controller.	K_p	–

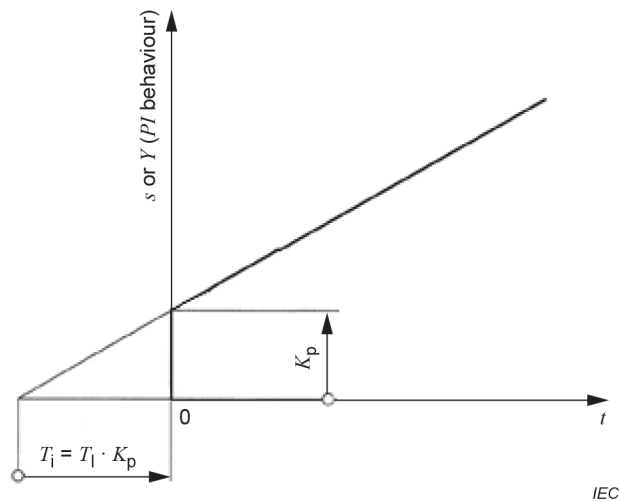


Figure 5 – Proportional action coefficient and integral action time

Sub-clause	Term	Definition	Symbol	Unit
3.6.11	integral action time	constant of the integral action of an idealized PID controller Note 1 to entry The integral action acts to trim the controller error to a value of zero. The rate of trim is proportional to the magnitude of the error and indirect proportional to the integral action time, as shown in Figure 5. Note 2 to entry The integral action coefficient K_I (also known as "integral gain") is the reciprocal value of the integral action time ($K_I = 1/T_I$). Reset time is defined by $T_i = K_p/K_I$.	T_I	s
3.6.12	derivative action time	constant of the derivative action of an idealized PID controller Note 1 to entry The transfer function ($T_D \times s$) can practically be realized only approximately by a DT_1 transfer function, i.e. a derivative term multiplied by a first-order lag element: $\frac{K_{1D} \times T_{1D} \times s}{1 + T_{1D} \times s}$ The step response of such a transfer function is shown in Figure 6. For small values of T_{1D} the following approximation applies, which represents the idealized derivative action: $T_D = K_{1D} \times T_{1D}$ Rate time is defined in parallel structured PID-controllers by $T_d = K_D/K_p$, with derivative action coefficient $K_D = T_D$.	T_D	s

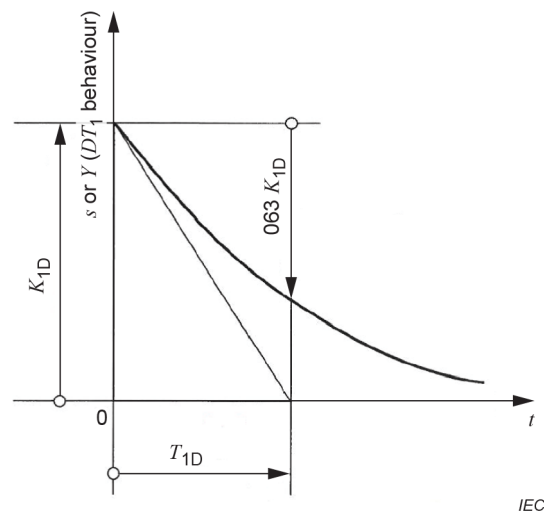


Figure 6 – Derivative time constant

Sub-clause	Term	Definition	Symbol	Unit
3.6.13	dead band	maximum band between two values inside of which the variation of the controlled variable does not cause any governing action SEE: Figure 7.	i_x	–
3.6.14	insensitivity	one-half of the dead band	$i_x/2$	–

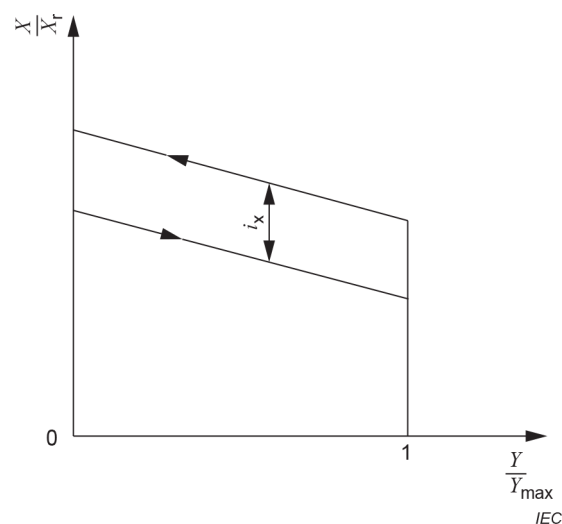
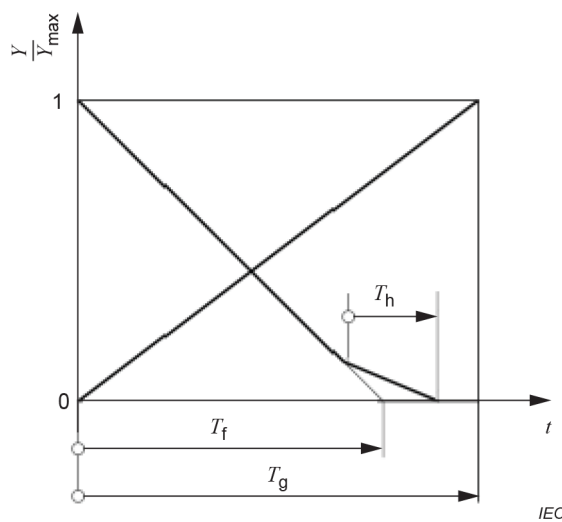


Figure 7 – Dead band

Sub-clause	Term	Definition	Symbol	Unit
3.6.15	minimum servo-motor opening/closing time	opening/closing time for one full servomotor stroke at maximum velocity, cushioning times disregarded SEE: Figure 8. Note 1 to entry: Minimum servomotor opening and closing times are the result of hydraulic transient calculations.	T_g, T_f	s
3.6.16	servomotor cushioning time	elapsed time during which the rate of servomotor travel is retarded, beginning at specified servomotor position Y_h/Y_{max} to full closed position SEE: Figure 8.	T_h	s



NOTE In case of stepped opening/closing velocities a diagram should be provided.

Figure 8 – Minimum servomotor opening/closing time

Sub-clause	Term	Definition	Symbol	Unit
3.6.17	time constant of the servo-positioner	time constant of an approximated behaviour of the servo positioner using a first-order lag element PT1: $\frac{Y}{Y_o} = \frac{1}{1 + sT_y}$ Note 1 to entry T_y can be calculated approximately from T_m : $T_y = T_m/K_{pm}$, see 5.4.3.	T_y	s
3.6.18	time constant of the servo-motor	reciprocal value of the slope of the curve which represents the servomotor velocity dy/dt as a function of the relative position u of the last controlling valve $T_m = \frac{1}{\frac{d(dy/dt)}{du}}$ Note 1 to entry For practical purposes T_m can be calculated by a linear approximation, see Figure 9. This approximation is only valid within an interval $-\frac{1}{T_f} < \frac{dy}{dt} < \frac{1}{T_g}$ because the velocity is limited outside of this interval	T_m	s

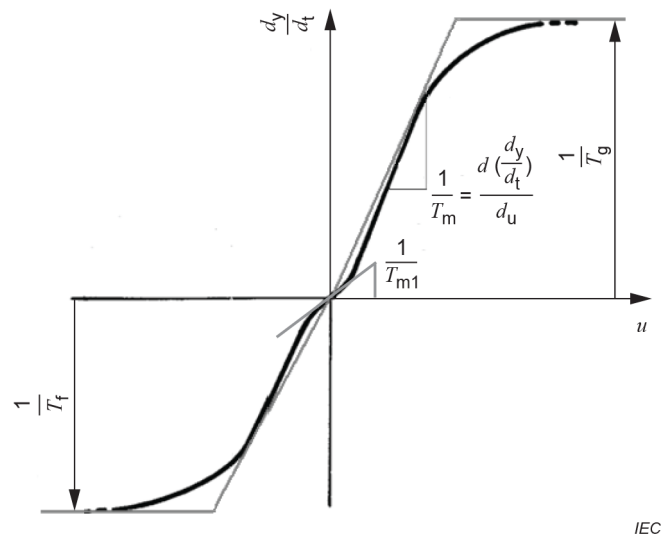


Figure 9 – Time constants of the servomotor/servo-valve combination

Sub-clause	Term	Definition	Symbol	Unit
3.6.19	time constant of the main control valve	reciprocal value of the slope of the curve which represents the main control valve velocity dy_v/dt as a function of the relative position u_v of the pilot valve $T_v = \frac{1}{\frac{d(dy_v/dt)}{du_v}}$ Note 1 to entry For practical purposes T_v can be calculated by a linear approximation (similar as in Figure 9 for T_m).	T_v	s
3.6.20	servo-positioner inaccuracy	maximum possible change in the servomotor position which can occur for a given constant value of the input signal of the servo-positioner SEE: Figure 10.	i_a	–

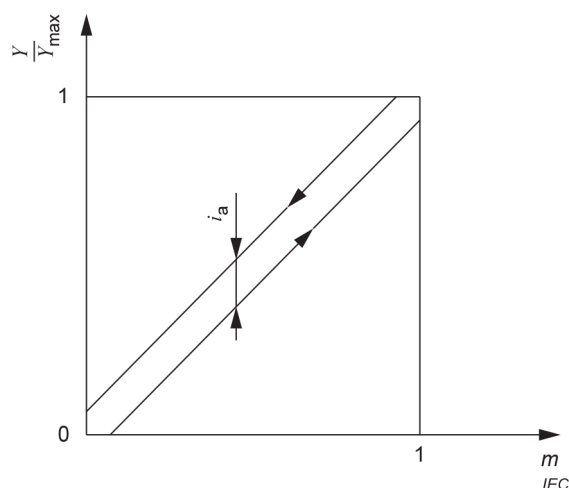


Figure 10 – Servo-positioner inaccuracy

Sub-clause	Term	Definition	Symbol	Unit
3.6.21	proportional gain main control valve controller	proportional gain (see 3.6.10) of the main control valve controller	K_{pv}	-
3.6.22	proportional gain servomotor controller	proportional gain (see 3.6.10) of the servomotor controller	K_{pm}	-
3.6.23	governing system dead time	time interval between a specified change in speed or command signal and the first detectable movement of the servomotor SEE: Figure 11.	T_q	s

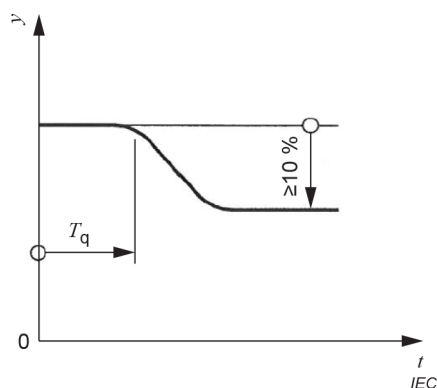


Figure 11 – Governing system dead time

Sub-clause	Term	Definition	Symbol	Unit
3.6.24	actuating energy	required energy for one servomotor stroke under the minimum required pressure $p_R = E_R/V_S$ considering the most critical operation conditions	E_R	N · m
3.6.25	servomotor volume	oil volume of the servomotors	V_S	m ³
3.6.26	tripping oil volume	oil volume of the pressure tank at the tripping point (between p_T and p_R , see Figure 28)	V_T	m ³
3.6.27	usable oil volume	usable oil volume between $p_{o \min}$ and p_R SEE: Figure 28.	V_u	m ³
3.6.28	residual oil volume	oil volume of the pressure tank after a full-load shut-down from the tripping point SEE: Figure 28. Note 1 to entry The residual oil is not usable.	V_{res}	m ³
3.6.29	design oil pressure	design pressure of the oil pressure tank	p_D	Pa ^{a)}
3.6.30	operating oil pressure	operating oil pressure under normal operating condition	p_o	Pa ^{a)}
3.6.31	tripping oil pressure	when the tripping pressure p_T is reached a shutdown is released, this implies $p_R < p_T < p_o < p_D$	p_T	Pa ^{a)}
3.6.32	minimum required pressure	minimum required pressure in the oil servo system	p_R	Pa ^{a)}

a) The unit bar is also used.

4 Governing system structure

4.1 General

The governing system of hydraulic turbines consists of the following main parts:

- electronic governor;
- oil hydraulic governor;
- electro hydraulic converter (as interface between both other parts).

The control functions are implemented in the electronic governor.

The oil hydraulic governor supplies the pressurized oil required for governing.

The servo-positioner performs the positioning of the servomotor and consists of the positioning control function in the electronic governor, the electrohydraulic converter and the required position measurement devices.

The combination of various control functions with the servo-positioners leads to certain typical governing system structures.

4.2 Main control functions

4.2.1 General

According to the control modes at the turbine governing level, the following major control functions can be distinguished:

- speed control;
- power output control;
- water level control;
- opening control;
- flow control.

In some systems, combinations of these control functions also occur.

4.2.2 Speed control

The purpose of the speed control basically is to maintain constant frequency. In the various modes of operation this means that:

- in the isolated operation with only one unit (small grid), the actual speed and therefore the frequency correspond to the command signal setting; in the island operation with more than one unit (medium grid), the speed control contributes to the frequency control through the permanent droop avoiding oscillation between the units;
- in the operation on a big grid, where the speed is determined by the grid frequency, the speed control contributes to the grid frequency control through the permanent droop and the dynamic characteristics of the controlled system;
- in the no-load mode (before synchronization and after separation from the grid), the actual speed corresponds to the command signal or the actual grid frequency with some small deviation.

4.2.3 Power output control

The power output control with a separate power control function is applied with the unit connected to the grid. Its purpose is to control the power output of the unit according to a power command signal irrespective of head variations. Frequency variations can influence the power output additionally via the permanent droop.

4.2.4 Opening control

The opening control serves to position the opening of the servomotor according to an opening command signal, either as a follow-up control in a superimposed control mode (for example speed control) or as a specific operating mode in grid control. In that last case, the usual configuration includes the permanent speed droop, which creates the relationship between the frequency of the grid and the opening of the servomotor, around the opening set-point from the unit control system.

4.2.5 Water level control

For run-of-the-river hydropower plants, it is often required to control the water level of the upper part of the river, in order to keep it relatively constant or inside a specified range around a fixed value. The corresponding water level control is usually operated by an external control function in the power plant control system. But in some cases, it can also be managed by the turbine governing system itself.

In the first case, by using an external water level control, the water level control operates as a secondary control function. For that, it compares a water level measurement with a level set-point or level limit values, and by a specific algorithm, modifies the command signal of the main controller of the turbine governing system (flow or opening or power controller), in order to control the water level. Attention should be paid, in case of participation of the unit to the frequency control of the grid, that time constant of the water level control shall be large enough, in order to allow the action of the primary frequency control, with a duration as required by the TSO (transmission system operator) according to the corresponding grid code.

In the second case, the internal water level control function also compares the level measurement with the level set-point or level limit values and modifies the command signal to the servo-positioners, or to the opening control including the permanent speed droop. If there are several units in the power plant, a level-opening droop (see 5.3.2) shall be implemented in each governor, in order to fix the operating point of each unit.

4.2.6 Flow control

For run-of-the-river hydropower plants, especially in case of several cascaded power plants along a river, it can be required to control the flow across the different turbines in operation in the concerned power plant (see 11.4). With that objective, a flow controller can be implemented as part of each turbine governing system or at a higher control level.

The actual flow is generally an indirect measurement, i.e. calculated using characteristic curves as a function of guide vane opening, runner blade opening or needle position, if necessary, also dependent on the head.

In case of participation of the unit to the frequency control of the grid, attention should be paid that time constant of the flow control has to be large enough, in order to allow the action of the primary frequency control, with a duration as required by the transmission system operator (TSO) according to the corresponding grid code.

4.3 Configurations of combined controllers

4.3.1 General

In combined systems, control modes are assigned to different control functions. However, the output of the combined system acts the same main servo-positioner through the opening set-point.

Thereby, a bump-free switch-over between modes requires attention. In case of separate control functions, parameters shall be set according to the respective control loop. The speed control function always remains functional for safety reasons, e.g., to take over in the case of a load rejection or transition to island grid.

Other configurations than described in the following are possible.

4.3.2 Parallel structure

Two control functions are arranged in parallel and actuate one or several servo-positioners via a selector or a summing point, see in Figure 12 an example with parallel speed and power output control functions.

If a selector is applied, it often includes a maximum/minimum function for the speed control loop to prevail in the case of a load rejection.

If a summing point is applied, the switching of signals is avoided, but the power output control function (or other control function) influences speed control additionally and shall be set to ensure stability.

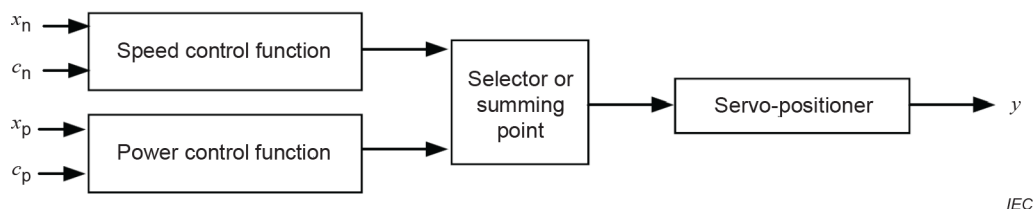


Figure 12 – Governing system with speed and power output control functions in parallel

Figure 13 shows a similar arrangement with water level control function.

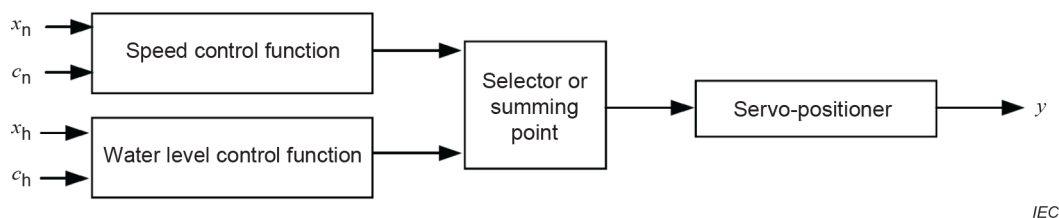


Figure 13 – Governing system with speed control function and water level control function in parallel

4.3.3 Series structures

The power output control function or the water level control function precedes the speed control function. They modify the set-point of the speed control function, see Figure 14.

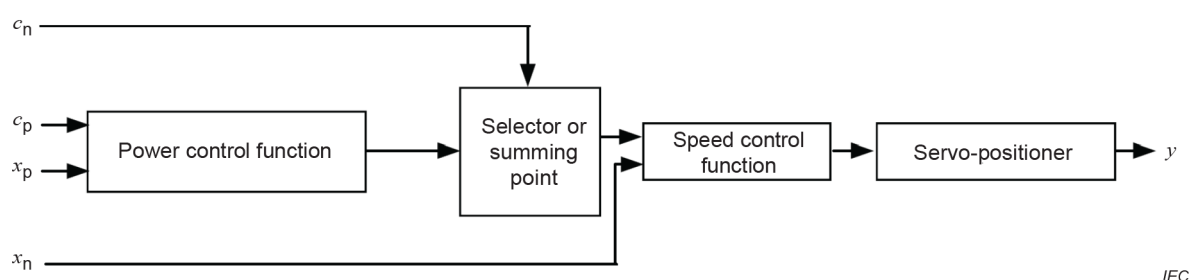


Figure 14 – Governing system with power output and speed control functions in series

4.4 Special control functions

4.4.1 Feed forward control

Feed forward is a control strategy that can be used with any closed loop controller to decouple the set-point response from the controller gain settings. Feed forward can be used to optimize the set-point response without having to increase the controller gains to values that would result in an unstable unit.

Feed forward can be implemented in many different ways depending upon the structure of the closed loop controller.

Figure 15 shows a general block diagram for implementing feed forward using the control function set-point also as set-point input for the feedforward function. The set-point input for the feedforward function can however also be derived in a more complex way.

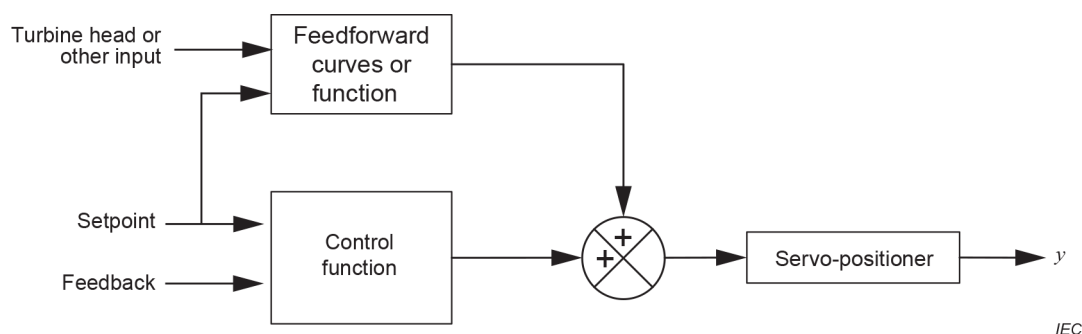


Figure 15 – Schematic diagram of a turbine governing system with feed forward

This method of feed forward allows a hydro unit to meet the response requirements set by the transmission system operator (TSO) and can be used to enable a hydro unit to provide additional ancillary services related to load following by scheduling or dispatch.

4.4.2 Surge tank level or pressure feedback

In certain configurations of hydropower plants, the PID traditional speed-load controllers of the turbine governing system are not sufficient to achieve a stable behaviour of the plant in grid connection mode. This could be the case of hydraulic turbines supplied by long penstocks, or by penstocks connected to a surge tank with a horizontal cross-section smaller than the Thoma area, due to economic considerations: in that case, the waterway can have an oscillatory response with an "anti-regulation" power effect and a low damping.

In such a situation, an additional control function including a waterway dynamics compensation system could be added to the digital governor, in order to improve the damping of the whole system, in a completely stable manner.

A typical design of such a compensation system could be as follows: the feedback compensation control function receives a measurement signal of the penstock water pressure (or of the surge tank water level), with the following objective. After a load reduction, when the water pressure or surge tank water level is increasing, the controller's action counteracts this increasing by opening the guide vane of the turbine. Concerning the design of the corresponding control function, it could be mainly based on a derivative action on the water pressure or level measurement.

This compensation system could then provide an improvement in the stabilization of the oscillations of water inside the penstock or surge tank after disturbances such as loss of neighbouring units, power set-point changes, frequency disturbances.

It should however always be kept in mind, that the positive effect of this method on the overall pressure fluctuation will have undesired effects in the initial response of the controlled variable (e.g. speed or power).

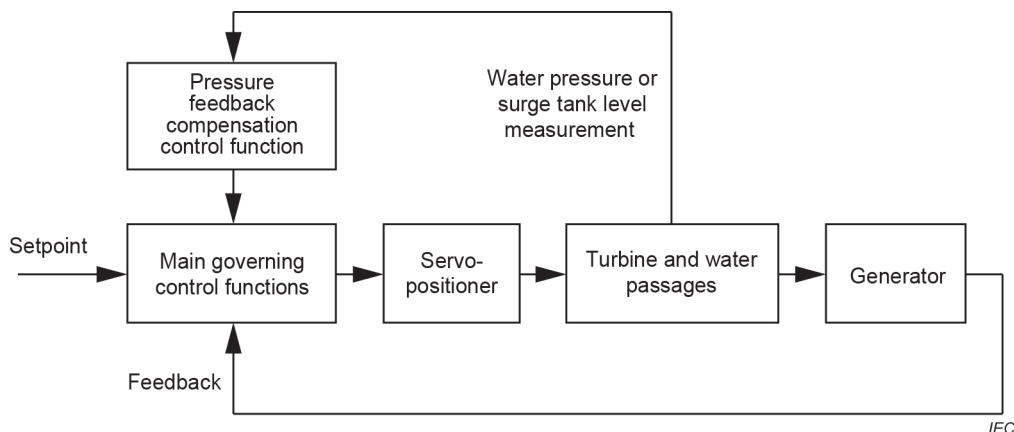


Figure 16 – Schematic diagram of a turbine governing system with an additional pressure feedback compensation control function

This method of pressure feedback compensation can also be used for some pumped storage power plants, where pump-turbines are difficult or even impossible to be synchronized when they operate too close to the hydraulic stability limit. The additional pressure feedback control function allows to extend the stability domain of the speed control loop in no-load operating mode, in order to cover the specified range of grid frequency.

4.5 Pump-turbine control

4.5.1 General

For pump-turbines two main control alternatives are available: the conventional pump-turbine control and the adjustable variable speed pump-turbine control.

4.5.2 Conventional pump-turbine control

Conventional pump-turbines operate principally in two different modes, the turbine mode and the pump mode. In turbine mode all other clauses of this document apply.

In pump mode the required governor function is restricted to the determination of the optimum guide vane opening as a function of head as shown in Figure 17.

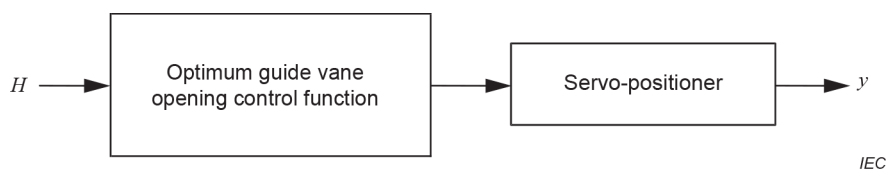


Figure 17 – Governor function in conventional pump mode

4.5.3 Variable speed pump-turbine control

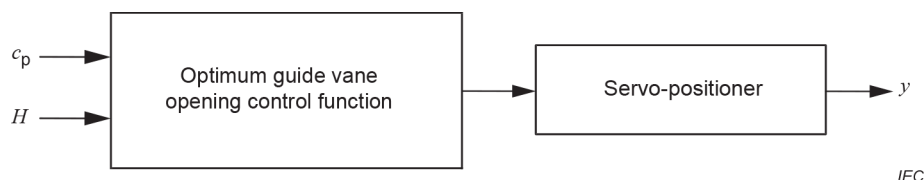
4.5.3.1 General

In variable speed pumped storage systems an additional degree of freedom exists: in case of turbine mode speed as well as power output can be controlled independently of each other in a combination of turbine control and generator control.

4.5.3.2 Pump mode

In pump mode, the task of the governor is to position the guide vane opening at the optimum as a function of the head and of the set-point of the power. The configuration is shown in Figure 18.

The power of the generator-motor is controlled by the generator converter; the shaft speed is a result of the equilibrium between this power and the rest of the system.



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Figure 18 – Governor function of variable speed pumped storage systems in pump mode

4.5.3.3 Turbine mode

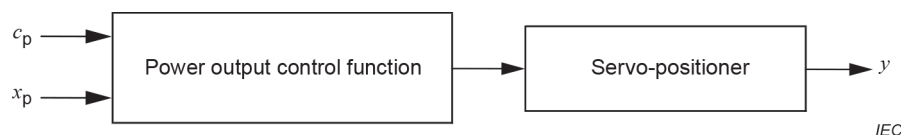
4.5.3.3.1 General

In turbine mode, two alternatives of control exist.

4.5.3.3.2 Power based control in turbine mode

On the power-based control in turbine mode, the task of the governor is to control the power according to the output power set-point. The configuration is shown in Figure 19.

In this case the rotating speed is controlled by the generator converter.



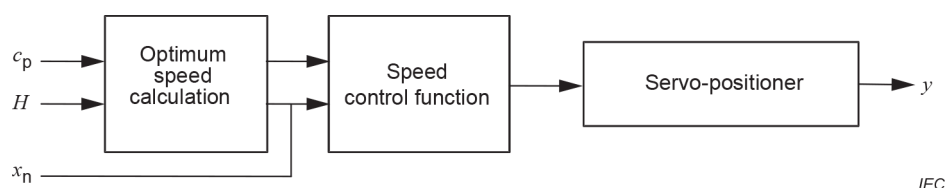
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Figure 19 – Governor function of variable speed pumped storage systems in turbine mode with power output based control

4.5.3.3.3 Rotating speed based control in turbine mode

On the rotating speed based control in turbine mode, first the optimum set-point for the rotating speed is calculated as a function of the head and the power. Using this optimum set-point the rotating speed of the unit is controlled. The configuration is shown in Figure 20.

In this case the power output is controlled by the generator converter.



IEC

Figure 20 – Governor function of variable speed pumped storage systems in turbine mode with rotating speed based control

4.6 Manual control

Manual control is a way to set the turbine opening(s) using the servo-positioner(s) or directly via the actuator(s). In any case, the unit's safety shall be kept by the overspeed protection.

Manual control can be desirable to allow a continuation of power generation in the case where the governor control loop or parts of it are out of order. Manual control can also be helpful in the commissioning phase of turbines and for maintenance activities.

The following general possibilities can be implemented as manual controls.

a) Manual control using the servo-positioner as part of the electronic governor

The positioning loop for manual control is an integral part of the electronic governor, that means manual control is only available when the electronic governor functionality is not impaired. All supervision functions are active during the operation of the turbine.

If higher availability including the manual control is required, a redundant configuration of the electronic governor can be used. After the failure of one of the systems, the redundant one will take the control keeping the complete functionality.

b) Manual control by independent servo-positioner

In this case the servo-positioner is fully independent of the electronic governor and the input to it is changed over from the output of the electronic governor to manual operation. An independent power source and other redundancies can be arranged to enhance the system availability.

c) Manual control by electronic actuation

The distribution valve assembly or other separate valves are actuated by electronical means over a path independent of output of the electronic governor.

d) Manual control by mechanical actuation

The distribution valve assembly or other separate valves are actuated by mechanical means independently of the output of the electronic governor

4.7 Linearization

For some control functions it is advisable that the relationship between the set-point value of the controlled variable and the servomotor position is linear. In order to achieve that, a linearization function can be introduced. This function can be implemented as a single curve or a set of curves including additional input signals, e.g. the value of the head.

For example, the non-linear relation between the servomotor position and the power output can be linearized introducing a function including additionally the head as input.

4.8 Limitation functions

In order to avoid unfavourable conditions of the unit, it's often required that either opening or power output limitation functions, or both are implemented in the electronic governor.

Examples for such limitation functions are:

- in case of a temporary problem of the penstock connected to a turbine which implies to reduce the discharge through the turbine, an opening limitation, with dependence of the head, can be introduced to limit the corresponding discharge;
- in case of turbine's, generator's or transformer's technical problems (for example overheating or insufficient quality of insulation), a simple solution is to reduce the output power of the unit at a certain value, until such problems have been solved; this can be made by the implementation of a power output limitation in the turbine governing system, and by tuning the corresponding value of limitation.

Before introducing such limitations, special attention shall be taken to the safety topics and specially to the question of the consequences of failures in the electronic governor. The opening and power output limitation should generally ensure quick action and always show a stable behaviour with small (or better without) overshoot and as small as possible residual power oscillations.

4.9 Bumpless control modes transition

When a transition from one control mode to another is performed switching a selector (Figure 12, Figure 13, Figure 14) the control functions which become active shall follow up in a bumpless way.

Examples

- The power output control function command signal follows the actual power output signal in the speed control mode.
- The speed command signal follows the actual frequency signal in the power output control mode.
- The manual control set-point follows the actual servomotor position in the various control modes.
- In the case of remote control, the respective local controls shall follow the respective command signals.

4.10 Optimization control

Special control configurations can be provided to optimize the system with respect to the overall plant efficiency, the running smoothness or other criteria.

Examples:

- optimal relation between guide vane and runner blade position (cam relation, see 6.4);
- control of multiple needles in Pelton-turbines (see 6.3.4);
- individual control of guide-vanes (see 6.3.4);
- variable speed control for pump-turbines (see 4.5.3).

5 Functional performance

5.1 General

This clause deals with the overall performance criteria for a governing system. As the performance of a turbine governing system will strongly depend on the characteristics of the individual controlled system, some guidance is offered first regarding its modelling and digital simulation.

After that, recommendations are given for the ranges of parameter settings for a PID configuration as the most common example for control algorithms in the governor. Other control strategies can be applied if suitable or desirable for superior performance in relation to PID-controllers.

Additionally, some considerations are listed for the tuning of governor systems.

5.2 Modelling and digital simulation

5.2.1 General

Generally speaking, in the case of new hydropower schemes, a mathematical model of the total system is valuable for an optimization of the transient behaviour, unless the system is either straightforward or similar, or both to existing plants. The same applies to the modernization of existing plants. The purpose of such computations can relate to three areas:

- physical dimensioning of components of the plants;
- demonstrating the dynamic behaviour of the system (resonance phenomena, etc.);
- governing system analysis and optimization.

These computations shall be based on a representative model of the system components, such as:

- the water passages;
- the turbine with its mechanism;
- the essential generator characteristics either in the island grid mode or the interconnected grid mode, or both;
- the grid characteristics;
- the turbine governing system.

All the mentioned areas of interest can in principle be served by the same models while the mathematical approach can vary. Whilst physical dimensioning of components of the plant shall be based on computations in the time domain, the dynamic behaviour of the total system can also be evaluated in the frequency domain. Control performance can be treated by both complementary studies, depending of the complexity of the installation or of the stability difficulties, or both:

- in the frequency domain with respect to small deviations from the steady state, in order to observe the poles of the system in the closed loop and to calculate the stability margins;
- in the time domain, either for small deviations in order to examine the overshoot and the damping of oscillations, but also for large deviations where non-linearities are significant.

If mathematical investigations of the dynamic behaviour in the frequency domain are applied, a suitable variable such as the guide vane opening shall be subjected to sinusoidal variations (frequency analysis). Thereby all relevant frequency ranges shall be considered at which excitations can occur, e.g. suction tube vortices in Francis turbines or resonances such as with natural frequencies of tunnel, penstock or the generator, or combinations of them. Thereby it should be noted that calculated natural frequencies of the hydro system can be inaccurate because the wave travel speed cannot be determined precisely.

For investigations with the aim of an optimization of the parameter settings of the governor, calculations in the time domain offer the advantage of considering non-linearities. Usually an integral criterion is applied, e.g.

$$\int |x - x_c| \times dt = \text{minimum}$$

or

$$\int t \times |x - x_c| \times dt = \text{minimum}$$

There are computer programs available which systematically vary the parameters and select a set of optimal values. By applying this method to the complete operating range, the setting of an adaptive governor can also be determined.

Optimization of the parameter setting of the governor in the frequency domain requires a linearized model, which can be derived from the complete non-linear model. It allows the analysis of stability margins. The set of optimal parameters can, for example, be determined by positioning the poles, i.e., the roots of the characteristic equation for optimal performance. This requires some experience.

The degree of detail in the modelling of a plant depends on the requirements with respect to controllability of the plant.

The effort even for smaller systems can be relatively costly. The following can help to make a judgment about how far the modelling should be carried in individual cases.

5.2.2 Water passages

- For the simulation of the water passages, the compressibility of the fluid and the elasticity of the penstock material shall be taken into account. For dimensioning and resonance studies, this should also be applied on tunnels and on galleries and shafts of surge tanks. If in the time domain the length and section of the waterway changes, then water and walls of this part can usually be regarded as incompressible and inelastic.
- A separate analysis of the tunnel/surge tank section and the penstock/turbine section is desirable to determine extreme values of the surge tank water level and maximum unit transient variables such as speed and pressure rises, respectively. System oscillations and governing system behaviour can only be reliably judged on the basis of the total system description.
- In surge tank calculations, energy dissipators such as throttles and the fluid inertia shall be taken into account.
- In low head plants, the inertia of water masses in the head and tailwater housings of the turbine shall be taken into account, while the elasticity can be neglected. Surge phenomena in either headwater or tailwater canals, or both can also be relevant.

5.2.3 Turbine, generator, electrical grid

- The turbine characteristics should be defined as input data for the investigation. The speed control of Pelton turbines can pose difficulties due to the lack of a negative torque and of the non-linearities introduced by the deflector. For island operation, the quality of the control can be increased by controlling the deflectors.
- For investigations on the resonances of the unit in interconnected grid mode, the damping factor of the generator should be taken into account.
- The stability of frequency control in island grids depends on the type of load, such as resistor, motor or combined loads. The resistor type load is the most stringent requirement.

5.2.4 Control concept

It is to be expected that PID control functions will also in the future remain in use for many plants for speed, power and water level control. Higher order algorithms, e.g., state control schemes will be used for the more complex system requirements. These higher order control schemes, while necessitating more effort to be implemented, are justified where better behaviour with respect to the magnitude of deviations from steady state and its return to steady state has to be achieved.

It is to be noted that the behaviour of a PID control function can also be enhanced considerably by readily available special additional means, such as feedforward control (see 4.4.1 or the feedback of secondary variables (see 4.4.2).

This in turn justifies the intention of this document to use the PID control function as a basis and reference for recommendations relating to the governing system. The recommended ranges in parameter adjustment will suffice in all normal cases.

Special conditions – extremely low inertias, extremely long penstocks, undersized surge tanks – should be subjected to modelling and digital simulation and can require an extension of the recommended parameter adjustment range.

5.2.5 Hardware-in-the-loop simulation

The time domain based models described above implemented on real-time-simulators can be used for hardware-in-the-loop tests of the digital governing system, see also IEC 60308.

5.2.6 Personnel training

Personnel training in plant operation can be performed either in the workshop or on site. If training is to be carried out on the original governing system, a real-time simulator of the plant can be used connected to the existing SCADA system.

5.2.7 Inaccuracy of plant simulators

All simulations are prone to inaccuracies. This applies to plant simulators, calculations of pressure surges and control parameters, all of which consist of similar elements. There are different reasons for the inaccuracies. In the following, the unavoidable inaccuracies are listed according to their origin.

Regarding input data, distinction shall be made between

- unforeseeable inaccuracies of the input data, and
- specific simplifications made in input data processing.

Examples of unforeseeable inaccuracies of the input data are

- deviations of model characteristic diagrams and curves of the turbines and valves from those of the real plant components;
- deviations of the moment of inertia of the unit;
- deviations of the friction resistances and the resistance coefficients of individual components;
- deviations of the servomotor velocities of guide vane and valves due to hydraulic forces and moments.

These factors cannot be measured in advance and can result in significant differences between simulation and reality.

When preparing the input data, the following systematic simplifications are for instance made:

- simplification of distributing conduits, branches, cones in the water passages;
- adjustment of the partial sections or wave velocities of staggered conduit systems to the computed time intervals;
- simplified consideration of the mass inertia of the water masses in the turbines.

Generally, the effects of these systematic simplifications can be kept within acceptable limits when being dealt with carefully.

For relatively slowly evolving processes, which are subject of governor design and testing, inaccuracies due to the following assumptions can be accepted:

- using the streamline flow theory thus neglecting multi-dimensional influences;
- neglect of the speed energy of flow;
- using steady-state characteristic diagrams and curves;
- Treatment of continuous transient processes with a discrete-time simulation based on a chosen cycle time.

5.3 Characteristic parameters for PID-controllers

5.3.1 General

The following subclauses relate to the characteristic parameters of a PID control function with permanent droop. It does not cover relevant parameters for other higher algorithms or control strategies.

5.3.2 Permanent droop

For units participating to the grid frequency control, the permanent droop b_p establishes a defined relationship in the steady-state condition between the relative frequency deviation (i.e. rotational speed deviation), and the relative deviation of the position of the servomotor or of the power output, e.g.: $x_n + b_p \times y = 0$

- using the relative position of the servomotor, b_p is usually defined as "speed-opening droop" or "frequency-opening droop";
- using the relative power output, b_p is usually defined as "speed-power droop" or "frequency-power droop".

Recommended minimum setting range of the permanent droop for frequency control: 0 % to 10 %.

For example, with a value of 5 % = 0,05 of frequency-power droop, a unit connected to the grid with a steady-state frequency deviation of -1 % (i.e. -0,5 Hz on a 50 Hz power system) will increase its power output by $-(-1 \%)/(0,05) = 20$ % of the rated power output P_{Gr} .

For additional information regarding the permanent droop in grid frequency control, see Annex B. A principle functional scheme of permanent droop using the output power is given in Figure B.1.

For units participating to a water level control (with a water level control function implemented in the governor), the permanent droop – defined in this case as "level-opening droop" – establishes a defined relationship in the steady-state condition between the relative water level deviation and the relative deviation of the servomotor position, e.g.: $x_z + b_p \times y = 0$.

5.3.3 Proportional action coefficient K_P , integral action time T_I , and derivative action time T_D

The parameters K_P , T_I and T_D establish the transient response of the governor. The desired transient response can be achieved with different representations of the PID structure, see Annex A.

The suitable adjustment of the parameters depends on the controlled system and shall be selected to provide a satisfactory transient response. Depending on the mode of operation, different sets of adjustments can be necessary, e.g.

- with speed control:
 - in no-load mode;
 - in an island grid mode (in some cases required only for part load);
 - in operation on the grid (over the complete power range).
- with combined power output and speed control:
 - in parallel, see Figure 12
 - in series, see Figure 14
- with combined water level and speed control, see Figure 13.

For different control modes the following values are typically set:

a) Proportional action coefficient K_P

- speed controllers: between 0,6 and 10
- power output controllers: between 0,1 and 10.

b) Integral action time T_I

between 0,1 s and 20 s.

(For water level control, considerably higher values can be applicable)

c) Derivative action time T_D

between 0 and 2 s.

where the relation $T_{1D}/T_D = 1/K_{1D}$ (see 3.6.11) is generally between 0,1 and 0,2.

5.3.4 Tuning of turbine governing systems

5.3.4.1 General

One significant difference of the hydropower plants in comparison with the thermal plants is the influence of the waterway dynamics (inertia, action of water hammer) and its effect on the hydraulic power input at low frequencies: this leads to the non-minimum phase effect, which limits the attainable closed loop bandwidth for stability of the whole governing system including the turbine and the corresponding water passages.

New digital turbine governing systems can flexibly be tuned for the different operating modes of the unit, such as no-load, interconnected grid connection or island grid connection, with different corresponding sets of parameters.

5.3.4.2 Dynamic performance analysis

The determination of adapted parameters for no-load operation is generally done on-site during the commissioning period of the governing system with corresponding field tests, whereas the parameters for the grid connection modes can be determined in advance, generally by using computational simulations before the on-site implementation during field tests.

Subclause 5.2 gives some general ideas concerning the modelling and digital simulation that can be carried out in order to predetermine the tuning of a turbine governing system of the hydropower units.

With this objective, the general control theory and corresponding tuning methods can be applied, by using time and in special cases also frequency domain analysis, with the aim to find an adequate compromise between fast response time, stability and robustness.

The dynamic performance analysis can be made for the interconnected grid operating mode and in the island grid operating mode.

In the case of the interconnected grid operating mode, the analysis should also take into account the requirements of the corresponding grid code: in general, when the power plant is operating in primary and secondary grid frequency control (see Annex B), these requirements are mainly detailed in terms of maximum allowed value of response time of the power output for a given frequency disturbance, and sometimes also with a maximum allowed value of overshoot. The scope of the analysis shall be defined in each individual case and can include a number of different operating points (e.g. values of water head and power output set-points).

Attention should be paid that because of physical reasons in certain cases such requirements can be difficult or even impossible to be met, (see Annex C). In some of those cases the choice of a more complex structure of control system, with adequate corresponding parameters, can be an advantage.

For the island grid operation mode, the analysis should consider small and large load disturbances. For large disturbances, a special attention should be paid on the minimum servomotor opening and closing times, and the analysis can determine the maximum value of the load variation which leads to the required maximum value of frequency deviation. As for the interconnected grid operation mode, for the analysis different operating points can be defined.

5.4 Other parameters of the governing systems

5.4.1 Command signal adjustments for controlled variables (speed, power output, etc.) and load limiter

a) Command signal ranges

Typical adjusting range:

- for speed controls: -10% to $+10\%$.

b) Command signal rates

The rates shall be in accordance with the fastest servomotor stroke rates as defined by the limiting orifices (see also 5.4.4).

5.4.2 Governor insensitivity $i_x/2$

The governor insensitivity includes the whole chain from the measuring sensor to the position of the servomotor.

Recommended limits:

- speed control: $i_x/2 < 2 \times 10^{-4}$
- power output control: $i_x/2 < 1 \times 10^{-2}$
- water level control: $i_x/2 < 1 \times 10^{-2}$
- flow control: $i_x/2 < 1 \times 10^{-2}$

In case of less stringent requirements relative to grid frequency control, $i_x/2 < 2 \times 10^{-2}$ can be acceptable for the speed control function. This can apply for grids, in which larger frequency deviations occur frequently.

NOTE For water level control deviating from Figure 5 instead of the rated value X_r the range of variation ($X_{\max} - X_{\min}$) shall be used.

5.4.3 Dynamic characteristics of servo-positioner

Figure 21 shows a linear dynamic model which can be used to represent a servo-positioner, assuming that the characteristics of all parts are suitable to guarantee good conditions of linearity to the system in the range of operation. The linear model does not contain the limitation of the maximum velocity of the servomotor.

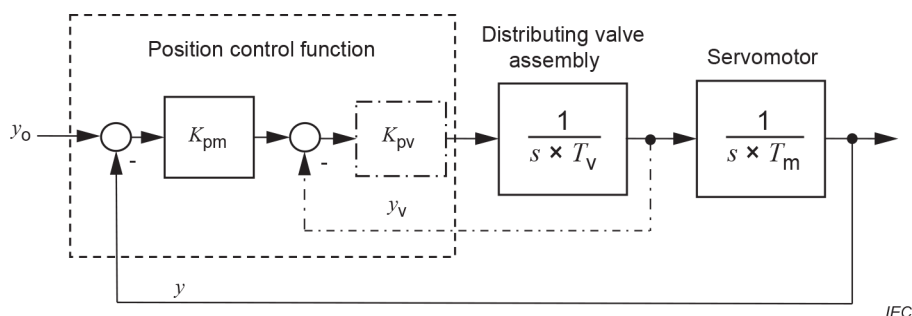


Figure 21 – Servo-positioner control loop – simplified dynamic model with P-controllers

For two stage configurations, the servo-positioner control loop can be regarded as two cascaded closed loops:

- a) Outer loop (servomotor position control loop, with y as opening signal feed-back)
- b) Inner loop (distributing valve assembly control loop (optional), with y_v as feed-back)

These loops have different time constants–(the inner loop is significantly faster than the outer loop). For simplified analysis, the time constant of the inner loop can be neglected and the cascaded closed loops of Figure 21 can be simply represented by the transfer function of Figure 22:

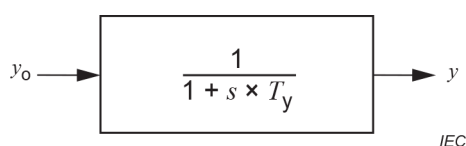


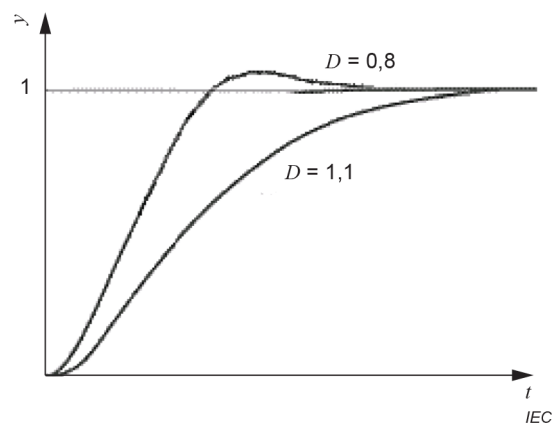
Figure 22 – Servo-positioner control loop – simplified dynamic model

Considering the above-mentioned simplification, the time constant of the whole servo-positioner T_y is approximately

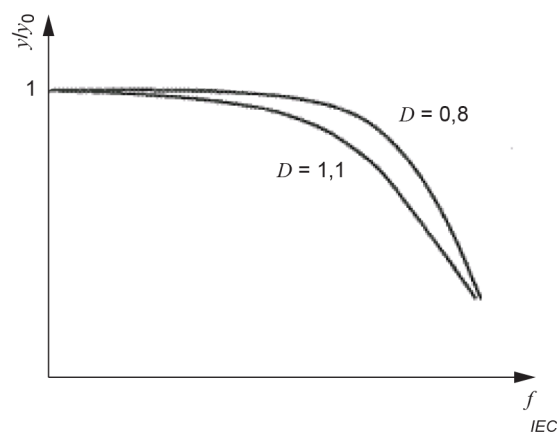
$$T_y = T_m/K_{pm}.$$

In order to provide a good global governing system performance, it is necessary to keep servo-positioner dynamic parameters in the correct range. With the objective of increasing the reaction velocity of the servo-positioner, sometimes a derivative action is introduced in the positioning circuit in parallel to the proportional action.

In practical cases the setting shall yield a response corresponding to a damping ratio D between 0,8 and 1,1 (see Figure 23).



a) Time step response



b) Frequency response

$D < 1$ periodically damped case

$D > 1$ aperiodic case

Figure 23 – Time step response and frequency response of the output of the servo-positioner

5.4.4 Parameters of servo-positioner

- a) Minimum servomotor opening time T_g and closing time T_f for the entire servomotor stroke, which are separately determined to satisfy water hammer and overspeed limitations. The limiting orifices or other suitable devices are dimensioned such that the actual stroke times in the presence of the highest oil supply pressure and the lowest required regulating capacity will not be lower than the allowable stroke time.
- b) Time constant of the servo-positioner T_y

This value is used for modelling and digital simulation of the system.

Recommended values for T_y :

- guide vane/needle: between 0,1 s and 0,25 s;
- runner blade: between 0,2 s and 0,8 s;
- deflector: between 0,1 s and 0,15 s.

Near the zero displacement, higher values of T_y prevail due to overlap and other special geometry features of the control edges of the distribution valves (see T_{y1} in Figure 9).

NOTE 1 If the graph is stepped or if, in a frequency response measurement, the limit velocities of a servomotor are reached, an effective averaged time constant (as a function of amplitude) can be used for comparison.

c) Servo-positioner inaccuracy in follow-up arrangement, i_a

It can have a major influence on the dead band i_x and therefore shall be kept small.

Recommended value:

$i_a < 0,4 \% ^2$ for the complete servo-positioning system.

d) Governing system dead time T_q

The governing system dead time T_q results from the combination of a dead time in the electronic controller (sampling time in the microprocessor if applicable) an overlap in the control valve or a series arrangement of several control valves, and clearances in mechanical connections.

Recommended value:

$T_q < 0,20 \text{ s}^2$.

5.4.5 Servo-positioner non-linearity by kinematics

The measured position of the sensor can vary from mechanical position of the actuating device, for example guide vanes or runner blades, due to the kinematic linkages.

In digital governing systems the nonlinearity of the measurement can be compensated by additional correction functions.

In order to improve the quality of the governing system it is advisable to keep servo-positioner non-linearity in a manageable range.

6 Servo positioner configurations

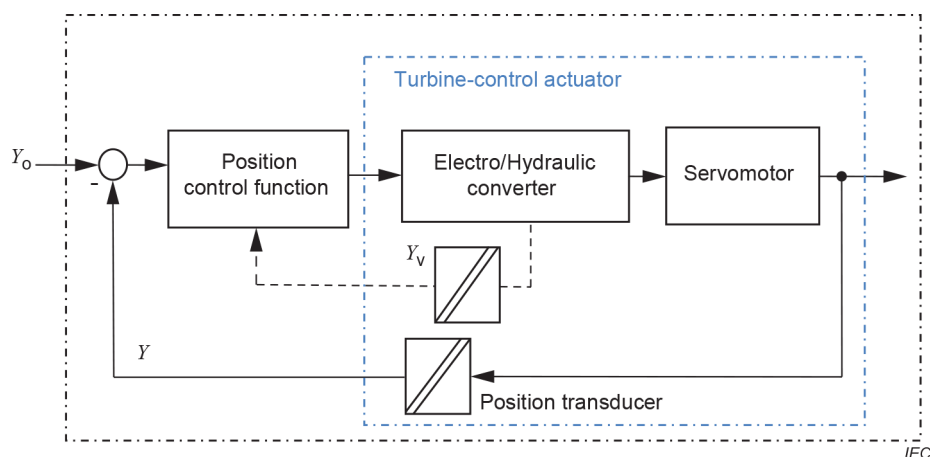
6.1 Servo-positioners

The servo-positioner receives an opening set-point from the main control loop of the turbine governor and positions the servomotor.

The servo-positioner typically includes the following parts (see Figure 24):

- servomotor;
- position transducer;
- electro/hydraulic converter;
- position control function.

² For small power stations, these values can be increased to: $i_a < 0,6 \%$, $T_q < 0,30 \text{ s}$.



Y_o opening set-point

Y servomotor position

Y_v main distributing valve position (in case of two-stage configuration)

Figure 24 – Servo-positioner block diagram

6.2 Configurations of servo-positioners

Depending on the oil volumes to control, the control loop can consist of:

- control valves can be either proportional valves or other types like on/off valves
- distributing valve assembly can be configured as follows:
 - one stage: distributing valve assembly consists only of control valve, which acts directly on the servomotor, electronical or mechanical one feedback to the digital governor of the servomotor position;
 - two stage integrated: the distributing valve assembly has an internal hydraulic amplifier with a second stage, the feedback signal is integrated in the valve, one feedback signal to the digital governor of the servomotor position;
 - two stages: pilot control valve acts on main distributing valve, which drives the servomotor, two feedback signals to the digital governor from main distributing valve and servomotor.

The type of configuration has influence on positioning accuracy and manual control options.

6.3 Multiple actuator control

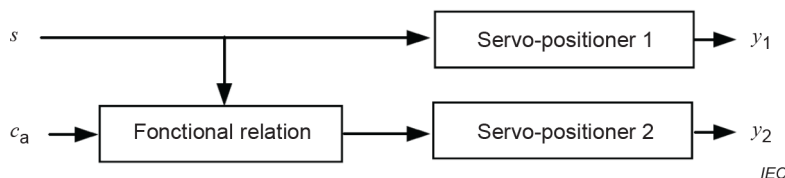
6.3.1 General

In case of multiple actuator control (e.g.: dual control of a turbine with controllable guide vanes and runner blades) the following arrangements are distinguished:

- parallel (see Figure 25);
- series (see Figure 26);
- individual (see Figure 27).

The functional relationship can be defined non-linearly through a single or a set of curves or tables (if applicable an additional signal can be used; e.g.: the head can be used to influence the guide vane – runner blade position relationship).

6.3.2 Parallel structure



s output signal of the electronic governor

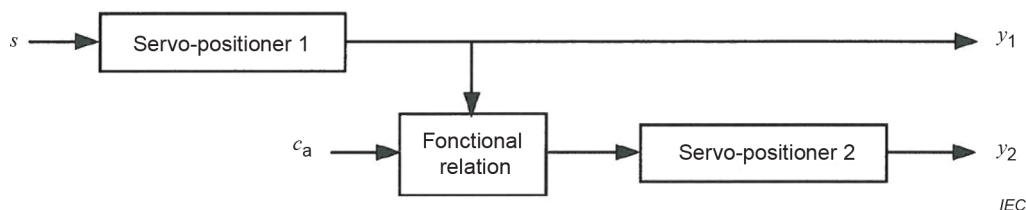
y_1 output signal of servo-positioner 1

y_2 output signal of servo-positioner 2

c_a input signal for the functional relation

Figure 25 – Parallel structure with defined functional relation and an additional signal superimposition

6.3.3 Series structure



s output signal of the electronic governor

y_1 output signal of servo-positioner 1

y_2 output signal of servo-positioner 2

c_a input signal for the functional relation

Figure 26 – Series structure with defined functional relation and additional signal superimposition

6.3.4 Individual control

In the case of individual servomotor control, individual set-points for each servo-positioner are calculated by the controller (e.g.: individual guide vane or individual needle control).

A special design of the turbine control is the provision of individual guide vane servomotors and electro-hydraulic controls, requiring synchronisation, to maintain all guide vanes at the same position. The synchronisation is achieved with the help of mechanical linkages or electronically, depending on the system adopted. In order to avoid instability, the set-points for each servo-positioner can be different.

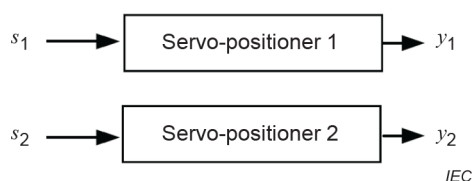


Figure 27 – Structure with different set-points for each servo-positioner

In the case of guide vanes or needles with individual servomotors, monitoring the synchronism of the positioning is recommended. For this purpose, the deviation of the position of each servomotor from the average of the total is monitored. When the deviation exceeds a given limit, a warning or a shut-down is initiated. Also the control deviation of the individual positioning loop can be monitored. If it does not return to approximately zero in a given time span a warning or a shutdown is initiated.

6.4 Dual regulation of turbines with controllable guide vane and runner blade angles

The functional relationship between guide vane and runner blade position is called cam relation.

The head H can be arranged to influence the functional relationship additionally. In the series structure, either the guide vane or the runner blade servo can be in the lead.

The defined functional relationship between guide vane and runner blade angle is generally based on specific or series model test results. The relationship can be verified or corrected on site by index measurements or measurements of the efficiency or other quantities.

6.5 Dual control of turbines with needles and deflectors

The deflector's purpose is to limit the speed increase during load rejections and to control speed under island operation when large perturbations occur. The control can be accomplished in the following ways:

- parallel control of needles and deflector;
- direct control of needles, control of deflector as series structure;
- direct control of the deflector, control of needles as series structure.

6.6 Other relationships

The following examples are related to equipment according to 11.1 to 11.3:

- runner blade and guide vane angle and possibly the position of the servomotor of a draft tube gate following a load rejection to minimize surge;
- guide vane angle and the position of a bypass valve following a load rejection to limit dynamic pressure variations and speed increases;
- guide vane angle and bypass position in normal operation and following a load rejection in the case of combined operation of a power station with an irrigation scheme.

7 Instrumentation

7.1 General

For the various control tasks the following variables are essentially measured:

- rotational speed;
- power;
- water level;
- pressure;
- stroke.

With electrical transducers these items are important:

- freedom from backlash in the mechanical components;
- environmental resistance;
- linearity;
- adjustability.

7.2 Rotational speed

The rotational speed is measured by different methods.

- toothed wheel or pole band, with proximity switch or impulse transducer (e.g. based on Hall-effect);
- speed measurement at the generator via transducers (utilizing the generator voltage or the residual remanence voltage);
- tacho-generators (especially if a free shaft end is available);
- others.

7.3 Power output

The power output is measured via a transducer. The input signal to the governor shall be sufficiently filtered (with care in order to avoid the introduction of an inadequate delay).

7.4 Water level

The water level is measured either directly or indirectly (e.g.: via a float or by using a pressure transducer or a pneumatic transmitter, or other sensors).

7.5 Actuator position (stroke)

Positions (e.g.: feedback signals) are measured by rotational or linear transducers.

In case of linkages, backlash and undue forces shall be avoided (e.g.: in the case of overstroke). In case of cables and driving band, sufficient pre-tension is important. The design of the system shall avoid resonance phenomena with suitable margins of safety.

7.6 Signal transmission from electronic transmitters

For the variables as per 7.2 to 7.5, usually 0 to 20 mA or 4 mA to 20 mA are used for signal transmission from the transmitter to the controller. Signals < 4 mA and > 20 mA are often used for signal monitoring. For signal transmission, shielded or twisted cables are required in order to suppress induction noise.

Alternatively, sensors can be connected remotely in digital form by network protocol. For long distances the use of optical transmission can be necessary.

8 Safety functions and devices

8.1 General

The different types of shutdown sequences of the unit are related to the unit control system and the protection system. The operation of the turbine governing system and associated safety devices or circuits shall be in accordance with the corresponding requirements.

8.2 Quick shutdown and emergency shutdown

8.2.1 General

For the definition of the tripping strategies (e.g.: quick shutdown and emergency shutdown), tripping actions, servomotor shutdown initiating devices, tripping criteria shall be distinguished and combined as follows.

8.2.2 Tripping actions

- moving of the servomotor to the closed position;
- opening of the circuit breaker.

8.2.3 Servomotor shutdown initiating devices

- turbine governor system;
- governor-independent shutdown valve.

8.2.4 Tripping criteria

- mechanical fault;
- electrical fault in the unit;
- serious fault in the governing system;
- emergency shutdown push-button pressed.

8.2.5 Tripping strategies

There are several different tripping strategies widely used as common practice today depending on a combination of different tripping criteria, different servomotor shutdown initiating devices and the corresponding sequence of tripping actions.

The terms "quick shutdown" and "emergency shutdown" cannot be standardized at the time being because the terms are used differently and contradictory today in the international community.

Annex D contains two different widely used strategies and emergency/quick shutdown definitions as examples.

8.3 Overspeed protection device

The following types of protection devices are used:

- electrical speed contacts of a measuring system integrated with and monitored by the governor;
- electrical speed contacts of a measuring system independent of the governor;
- electrical speed contacts switched by a mechanical safety pendulum;
- oil hydraulic tripping device directly activated by mechanical safety pendulum.

8.4 Interlocks

The following types of interlocks are used:

- either electrical or hydraulic interlocks, or both between the governing system and the main shut-off valve or gate to avoid erroneous control modes or dangerous conditions;
- either mechanical or hydraulic interlocks, or both of the guide vane to protect against possible re-opening after quick or emergency shut-downs;
- electrical interlock between needle control and braking (counter) nozzle.

9 Provision of actuating energy

9.1 General

The necessary actuating energy is provided predominantly by oil hydraulics.

The minimum required pressure p_R follows from the required regulating capacity E_R ³ and the volume of the servomotors:

$$p_R = E_R / V_S$$

There are systems with and without accumulators to be distinguished.

Systems with accumulators are preferred, where quick delivery of large amounts of actuating energy is called for (e.g., in the case of power output-frequency control or when stringent requirements in frequency control prevail).

9.2 System with an accumulator

9.2.1 Pressure tank (air-oil accumulator)

For maximum applicable pressure and calculation, the respective rules apply, see Bibliography.

To prevent compressed air from entering into the hydraulic control system, which could result in rapid servomotor travel rates and potentially dangerous hydraulic transients, an oil cut off valve, such as a float valve or external shut off valve shall be used.

The design of such device and its control system shall consider the following aspects:

- safe closing only if risk of air entering the system;
- safe not closing under normal operation.

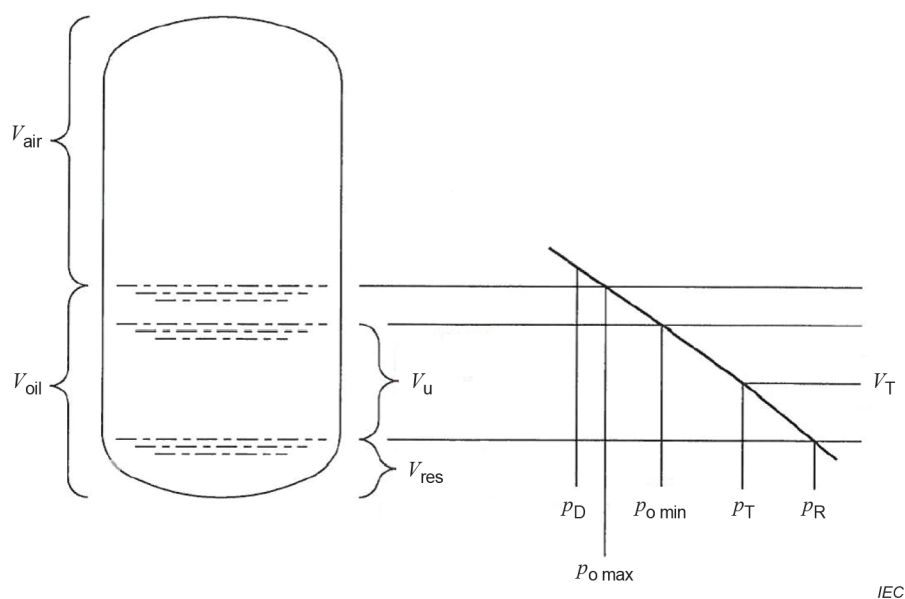


Figure 28 – Pressure tank content and pressure ranges

³ Friction included.

Design pressure, p_D

- Operating pressure range

$$p_{o \max} \quad \text{to} \quad p_{o \min}$$

$$p_{o \max} = (0,80 \text{ to } 1,0) p_D^4$$

$$p_{o \min} = (0,75 \text{ to } 0,9) p_D$$

- Tripping pressure (minimal pressure for emergency shutdown)

$$p_T (p_{o \min} > p_T > p_R)^5$$

- Minimum required pressure

$$p_R = (0,5 \text{ to } 0,75) p_D$$

Recommended usable oil volume

- Single loop control

$$V_u = 3 V_S$$

- Dual control

$$V_u = 3 V_{Sga} + (1,5 \text{ to } 2,0) V_{Sru}$$

$$V_u = 3 V_{Sde} + (1,5 \text{ to } 2,0) V_{Sne}$$

- Main inlet valve

$$V_u = 2 V_{Smiv}$$

where

V_S is the oil volume of all servomotors;

V_{Sga} is the total volume of all guide vane servomotors;

V_{Sru} is the volume of the runner blades servomotor;

V_{Sde} is the total volume of all deflector servomotors;

V_{Sne} is the total volume of all needle servomotors;

V_{Smiv} is the total volume of all main inlet valve servomotors.

In special island grid operating conditions, higher values can be required.

Minimum usable oil volume

If island operation is not required, the usable oil volume can be reduced to a minimum, which corresponds to the tripping oil volume V_T according to:

$$V_T = V_S + q_l \times t_l + V_{res}$$

where

q_l is the leakage of the whole oil supply system;

t_l is the time available up to ensuring a mechanical locking of the servomotors or closing time of inlet valve;

⁴ When determining the maximum operating pressure, the rules concerning the opening and final pressure of the safety valve apply.

⁵ p_T is chosen such that pressure after closure does not drop below p_R .

V_{res} is the volume reserve in the lower part of the pressure tank, including a safety margin and some volume to prevent air flowing into the system.

9.2.2 Piston accumulators

Commercial piston accumulators with a hermetic separation between oil and inert gas (mostly nitrogen) allow the application of higher pressures than those for air-oil accumulators. A residual oil volume and an automatic gas replenishment do not have to be provided.

For the design of the volumes of the piston accumulator system, the range of ambient temperature in the area of the accumulator shall be considered, in order to assure that extreme low or high ambient temperatures do not lead to situations related to the position of the piston, in which the closing safety of the turbine is not guaranteed.

The same values for pressure ranges and oil volumes apply as for air/oil accumulators, see 9.2.1. The value of the volume reserve V_{res} can be set to zero.

9.2.3 Bladder accumulators

As in this case the oil/gas volumes cannot directly be supervised, bladder accumulators shall not be used if the safe closing of the unit depends on the amount of storage energy in the accumulator. If closing of the turbine is guaranteed by other means (e.g. closing weight, closing spring) bladder accumulators can be allowed.

As a breakage of the bladder cannot totally be prevented, this case including the consequent transport of gas dissolved in the oil to the rest of the system shall be considered. In parts of the system with lower pressure levels the gas will be released in form of bubbles. The gas bubbles can collect in some points of the system or flow through valves and orifices. Both situations can lead to a malfunction of the positioning (instabilities, vibrations or irregular partially too fast movements of the servomotors). Therefore specially in case of critical configurations regarding water hammer (e.g. Pelton turbines with long penstocks) the use of bladder accumulators should be avoided or only be allowed after having carried out detailed investigations.

9.2.4 Other systems

For shutdown safety, weight or spring-loaded accumulators (low-head turbines) and water pressure taken directly from the penstock (high-head turbines) partly in combination with oil pressure systems (combined systems) are used.

These systems shall be dimensioned in such a way that the turbine can be shut down safely, i.e. also in the case of failure of the oil pressure supply in combined systems.

The opening is in most cases carried out with oil pressure.

9.2.5 Pumps for accumulator systems

Two induction motor driven pumps are normally foreseen, each with a capacity of one combined servomotor volume of turbine per minute or with a capacity which can refill the oil volume ranging from $p_{o \text{ min}}$ to $p_{o \text{ max}}$ within one minute. As a general rule the smaller pump capacity of both criteria will be valid.

For combined oil hydraulic pressure units for both turbine and inlet valve, the combined capacity of both pumps shall allow to open the inlet valve within 1 min. Such oil hydraulic pressure units are safe, if the closing of the inlet valve is performed by a closing weight or by water pressure from the penstock.

In case of higher requirements regarding either the starting time of the unit or the regulating activity, or both, bigger pumps should be used.