



BSI Standards Publication

Hydraulic turbines, storage pumps and pump-turbines — Hydraulic transient analysis, design considerations and testing

National foreword

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TECHNICAL SPECIFICATION

Hydraulic turbines, storage pumps and pump-turbines – Hydraulic transient analysis, design considerations and testing



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**Hydraulic turbines, storage pumps and pump-turbines -
Hydraulic transient analysis, design considerations and testing**

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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 Technical Specification 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

Proper treatment of hydraulic transients is important for utilities, equipment owners, producers, as well as equipment suppliers. During transient operational modes, hydro power equipment can be exposed to large dynamic loads that, in extreme cases, can lead to catastrophic events. Insufficient understanding of basic transient phenomena and related modelling can lead to inadequate design safety factors that can put both equipment and related plant personnel at risk.

The occurrence of transient phenomena during hydro power production is unavoidable. Simulation and testing of hydraulic transients provide critical values throughout the project development that should be used to establish and validate guarantees for maximum and minimum allowable pressures, rotational speeds, torques, etc. The primary goal of a hydraulic transient analysis is to support the safety of the hydroelectric equipment and any personnel directly related to the environment produced by a hydraulic transient event.

1 Scope

This Technical Specification (1) describes hydraulic transient phenomena of hydro turbines, storage pumps and pump-turbines and the factors that affect them, (2) provides modelling and measurement best practice guidelines and resulting limitations (3) defines relevant equipment design criteria, and (4) identifies potential mitigation solutions. Definitions of the relevant terms and quantities are provided along with descriptions of the system components that are considered.

Various types of hydraulic transient phenomena are covered in the current specification, including waterhammer, surge tank mass oscillation, and speed variation of the hydraulic machine for operational modes such as start-up, normal shutdown, emergency shutdown, and load rejection and acceptance. Combinations of the individual transient modes are also considered. The links between these transient characteristics and operation across the relevant quadrants for each machine type are illustrated. Methods for mitigating extreme transient values of pressure, surge tank water levels, and unit rotational speed are suggested.

In this Technical Specification, abnormal transient phenomena are also defined and described, including component malfunction and catastrophic events like component rupture. The probability of the occurrence of these extreme events and how this can influence the relevant safety margin is described.

Different stages of hydroelectric development correspond to unique sets of available data with differing levels of detail and uncertainty. The various stages of project development are described in relation to transient analysis. The corresponding uncertainty margins associated with each stage and analysis technique are discussed.

This Technical Specification provides guidelines and commonly accepted practices to model and compute transient conditions. It provides a summary of the basic hypotheses and equations, together with relevant characteristic quantities and system time constants. Accepted methods of modelling hydraulic components, and related numerical simulation methods are identified. This specification details the input data, including best practices for model testing of hydraulic machines, valves, gates, etc. to acquire reliable transient modelling.

Different calculation methods with different levels of uncertainty are described so that the most suitable approach for the available data or project stage can be selected. Limitations of one-dimensional modelling methods are described. Additional means of investigation, such as physical model tests or CFD computation, can be used to improve the simulation results.

For various configurations of equipment and operating regimes, typical scenarios for consideration are defined.

Procedures to determine uncertainty margin, with respect to the modelling and computation methods and available input data are described.

Finally, this Technical Specification describes methodologies for on-site measurements with respect to transient such as load rejection tests, runaway tests, etc. Recommendations are provided for quantities to be monitored during these tests, with related instrumentation, calibration and data acquisition systems. Procedures for comparing on-site measurements with numerical simulation results are proposed.

Note that the following aspects influence hydraulic transients and are therefore introduced; however, the associated modelling and related design guidelines will not be addressed in the Technical Specification:

- waterway design and optimization;
- upstream and downstream free surface flows;
- fluid structure interactions;
- cavitation and water column separation;
- unsteady friction and viscoelasticity;
- two phase flows;
- turbine governor design;
- flow induced pressure fluctuations.

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 60041:1991, *Field acceptance test to determine the hydraulic performance of hydraulic turbines, storage pumps and pump-turbines*

IEC 60193:2019, *Hydraulic turbines, storage pumps and pump-turbines - Model acceptance tests*

IEC 60545:2021, *Guidelines for commissioning and operation of hydraulic turbines, pump-turbines and storage pumps*

IEC 60994:1991, *Guide for field measurement of vibrations and pulsations in hydraulic machines (turbines, storage pumps and pump-turbines)*

IEC 62884 (all parts), *Measurement techniques of piezoelectric, dielectric and electrostatic oscillators*

3 Terms, definitions, symbols and units

3.1 General

For the purposes of this document, the following common terms, definitions, symbols and units apply. Specialized terms and definitions are explained where they appear.

Clarification of any term, definition or unit of measure in question shall be agreed to in writing by the contracting parties in advance of the test.

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

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

3.2 General terminology

3.2.1 point

value established by one or more consecutive sets of readings or recordings at unchanged operating condition and settings, sufficient to calculate the performance of the machine at this operating condition and these settings

3.2.2 test

collection of points and results adequate to establish the performance of the machine over a specified range of operating conditions

3.2.3 hydraulic performance

all performance parameters attributable to the machine due to hydrodynamic effects

3.2.4 main hydraulic performance data

subset of the hydraulic performance parameters, i.e. power, discharge, specific hydraulic energy, efficiency, steady-state runaway speed and discharge.

Note 1 to entry: It is important to consider the influence of cavitation.

3.2.5 guarantee

specified performance data contractually agreed to

3.3 Units

The international System of Units (SI, see ISO 80000-4) has been used throughout this document.

All terms are given in SI base units or derived coherent units¹. The basic equations are valid using these units. This shall be taken into account if other than coherent SI units are used for certain data (e.g., kilowatt instead of watt for power, kilopascal or bar instead of pascal for pressure, min^{-1} instead of s^{-1} for rotational speed, etc.). Temperatures can be given in degrees Celsius since absolute temperatures (in kelvins) are rarely required.

Any other system of units can be used, but only if agreed in writing by the contracting parties.

¹ $\text{N} = \text{kg} \cdot \text{m} \cdot \text{s}^{-2}$ $\text{Pa} = \text{kg} \cdot \text{m}^{-1} \cdot \text{s}^{-2}$ $\text{J} = \text{kg} \cdot \text{m}^2 \cdot \text{s}^{-2}$ $\text{W} = \text{kg} \cdot \text{m}^2 \cdot \text{s}^{-3}$

3.4 Terms, definitions, symbols and units

3.4.1 List of terms, definitions, symbols and units

Subclause	Title
3.4.2	Subscripts and symbols
3.4.3	Geometric terms
3.4.4	Physical quantities and properties
3.4.5	Discharge, velocity, speed terms and time symbols
3.4.6	Pressure terms
3.4.7	Specific energy terms
3.4.8	Height and head terms
3.4.9	Power and torque terms
3.4.10	General terms relating to fluctuating quantities
3.4.11	Fluid dynamic and scaling terms
3.4.12	Dimensionless terms

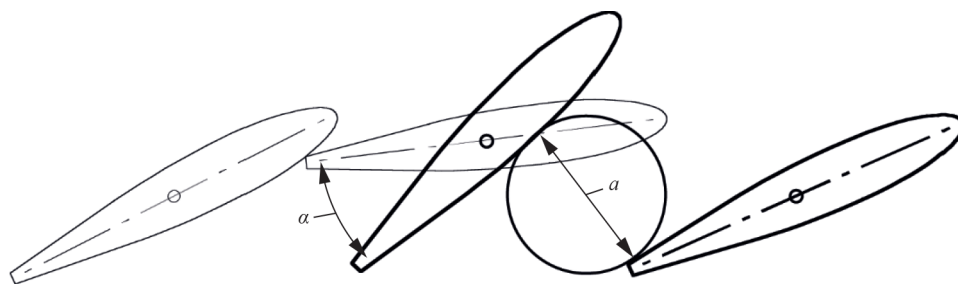
3.4.2 Subscripts and symbols

Subclause	Term	Definition	Subscript or symbol	Unit
	Free surface	location of water free surface	0	
	Pressure measurement	location of pressure measurement	1	
3.4.2.1	Absolute		abs	-
3.4.2.2	Atmospheric		atm	-
3.4.2.3	Number of blades		b	-
3.4.2.4	Best Efficiency Point		opt	-
3.4.2.5	Runner blade passing		bp	-
3.4.2.6	refer to water level		c	-
3.4.2.7	Downstream		d	-
3.4.2.8	Electrical		elect	-
3.4.2.9	Equivalent		equ	-
3.4.2.10	Number of fluxes of a hydraulic machine		Flux	-
3.4.2.11	refer to gas		g	-
3.4.2.12	Mechanical		m	-
3.4.2.13	Model		M	-
	Measurement instrument	location of pressure measurement instrument	M	
3.4.2.14	Prototype		P	-
3.4.2.15	Reference		ref	-
3.4.2.16	Number of stages of a hydraulic machine		Stage	-
3.4.2.17	Upstream		u	-
3.4.2.18	Total		tot	-
3.4.2.19	Top of the surge tank		weir	-

3.4.3 Geometric terms

Subclause	Term	Definition	Symbol	Unit
3.4.3.1	Area	net cross-sectional area normal to general flow direction	A	m ²
3.4.3.2	Blade angle	average runner blade setting measured from reference position	β	°
3.4.3.3	Cross sectional area of tunnel	net cross-sectional area of tunnel normal to general flow direction	A_t	m ²
3.4.3.4	Curvilinear abscisse		s	m
3.4.3.5	Elevation	level of a point in the system above the specified reference datum (usually mean sea level)	z	m
3.4.3.6	Elevation difference	difference between two elevations	Z	m
3.4.3.7	Elevation difference between the free surface water and the pressure measurement.		Z_0	m
3.4.3.8	Elevation difference between the pressure measurement and the pressure measurement instrument.		Z_1	m
3.4.3.9	Elevation difference between the free surface and the reference datum.		z_0	m
3.4.3.10	Elevation difference between the pressure measurement and the reference datum.		z_1	m
3.4.3.11	Elevation difference between the pressure measurement instrument and the reference datum.		Z_M	m
3.4.3.12	Guide vane angle	average vane angle measured from closed position (Figure 3-1)	α	
3.4.3.13	Guide vane pitch diameter	diameter of guide vane circle, measured at the stem axis	D_z	m
3.4.3.14	Horizontal cross-sectional area of surge tank	net cross-sectional area of surge tank parallel to the free surface Note 1 to entry: $1,5 \times A_{th}$ (a safety factor of 1,5 is taken on A_{th} in order to ensure the system stability).	A_{st}	m ²
3.4.3.15	Hydraulic diameter	ratio of cross-sectional area of a non-circular waterway and its wetted perimeter P : $Dh = 4 A / P$	D_{hyd}	m
3.4.3.16	Inner runner blade diameter	diameter at intersection of runner blade edge with runner crown at the high-pressure side	D_1	m
3.4.3.17	Length	<modelling> discrete pipe section with constant diameter	L_i	m
3.4.3.18	Length	<prototype> total length of a pipe	L	m
3.4.3.19	Maximal water level	maximal water level of a reservoir	Z_{max}	m
3.4.3.20	Minimal water level	minimal water level of a reservoir	Z_{min}	m
3.4.3.21	Outer runner blade diameter	diameter at intersection of runner blade edge with runner band at the high-pressure side	$D_{1\alpha}$	m
3.4.3.22	Pipe wall thickness	measured thickness of a pipe wall	e	m

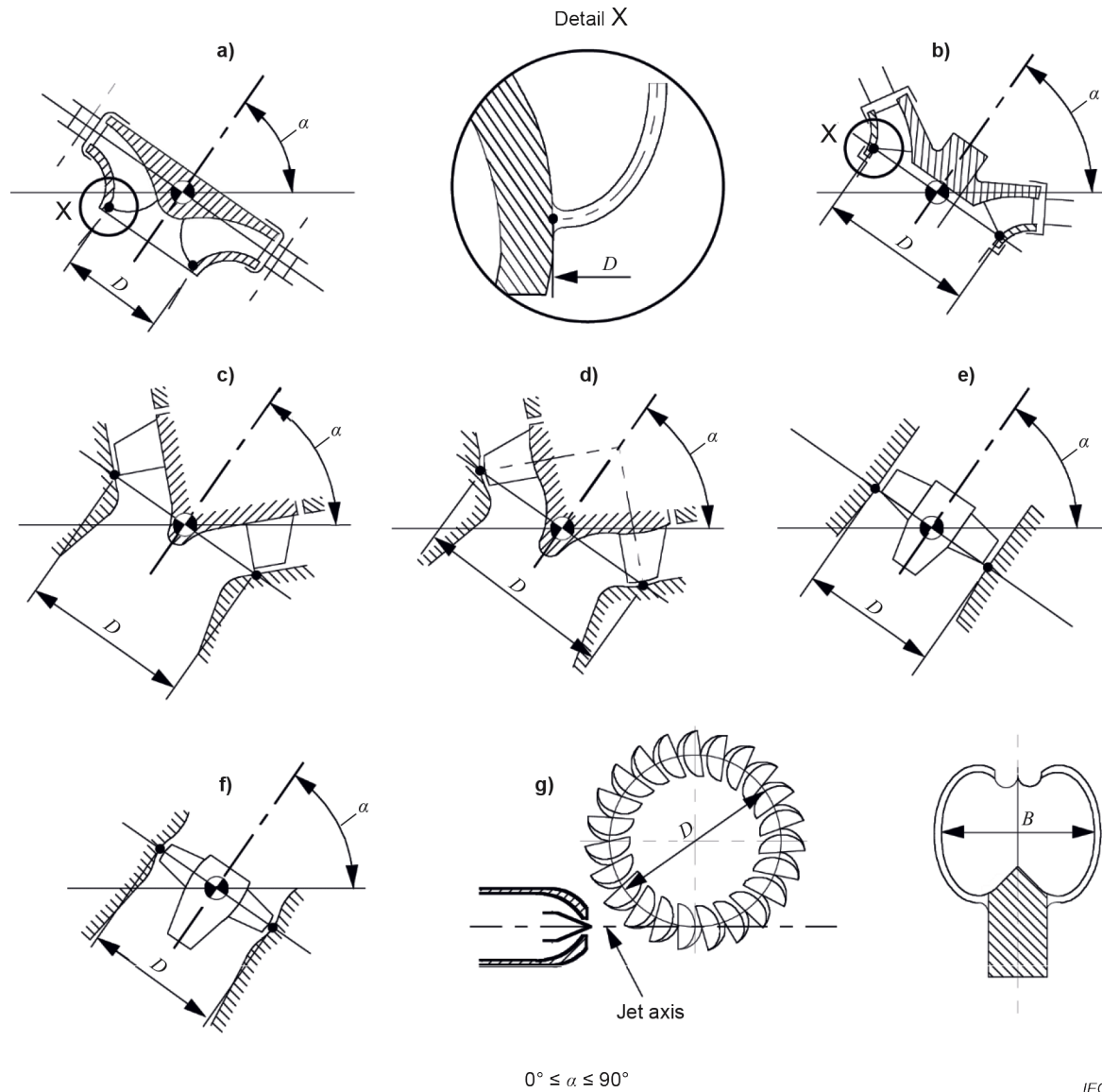
Subclause	Term	Definition	Symbol	Unit
3.4.3.23	Radius		R	m
3.4.3.24	Reference diameter	reference diameter of the hydraulic machine, see Figure 3-2	D	m
3.4.3.25	Scale of discharge		λ_{sq}	-
3.4.3.26	Scale of flow velocity		λ_{sv}	-
3.4.3.27	Scale of geometry		λ_{sl}	-
3.4.3.28	Scale of intensity of pressure		λ_{sp}	-
3.4.3.29	Scale of inertia of rotating parts		λ_{sJ}	-
3.4.3.30	Scale of power		λ_{sp}	-
3.4.3.31	Scale of rotational speed		λ_{sn}	-
3.4.3.32	Scale of roughness coefficient		λ_{sr}	-
3.4.3.33	Scale of time		λ_{st}	-
3.4.3.34	Scale of time constant		λ_{sTm}	-
3.4.3.35	Scale of water head		λ_{sh}	-
3.4.3.36	Spillway length	spillway length on the top of a surge tank	weir	m
3.4.3.37	Thoma cross-section area	smallest cross-sectional area of a surge tank that allows sufficient dampen oscillations	A_{Th}	m ²
3.4.3.38	Throat diameter	minimum band diameter of a runner or impeller	D_{th}	m
3.4.3.39	Integer number		Z	-



Closed position: $\alpha = 0^\circ$ or $a = 0$ mm

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Figure 3-1 – Guide vane opening and angle



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- a) Radial machines, such as Francis turbines, radial (centrifugal) pumps and pump-turbines; for multistage machines: low-pressure stage.
- b) Diagonal (mixed-flow, semi-axial, Deriaz) machines with fixed runner or impeller blades and with runner or impeller band.
- c) Diagonal (mixed-flow, semi-axial, Deriaz) machines with fixed runner or impeller blades, without runner or impeller band.
- d) Diagonal (mixed-flow, semi-axial, Deriaz) machines with adjustable runner or impeller blades.
- e) Axial machines, such as propeller turbines, tubular turbines, axial pumps and pump-turbines with fixed runner or impeller blades.
- f) Axial machines, such as Kaplan turbines, tubular turbines, axial pumps and pump-turbines with adjustable runner or impeller blades.
- g) Pelton turbines.

NOTE The term "tubular turbines" (see e) and f)) includes bulb, pit, rim generator and S-type units.

Figure 3-2 – Reference diameter and bucket width

3.4.4 Physical quantities and properties

Subclause	Term	Definition	Symbol	Unit
3.4.4.1	Acceleration due to gravity	local value of gravitational acceleration at the place of testing	g	$\text{m}\cdot\text{s}^{-2}$
3.4.4.2	Average gravity	gravity calculated between the free surface and the pressure measurement location	$\bar{g}$	$\text{m}\cdot\text{s}^{-2}$
3.4.4.3	Average density	density calculated between the free surface and the pressure measurement location	$\bar{\rho}$	$\text{kg}\cdot\text{m}^{-3}$
3.4.4.4	Axial strain	conduit clamping condition referred to axial strain	ε_{ax}	-
3.4.4.5	Axial stress	conduit clamping condition referred to axial stress	σ_{ax}	Pa
3.4.4.6	Bulk modulus of rock		E_r	Pa
3.4.4.7	Bulk modulus of water		E_w	Pa
3.4.4.8	Compressed air capacitance		C_g	m^2
3.4.4.9	Density	mass per unit volume	ρ	$\text{kg}\cdot\text{m}^{-3}$
3.4.4.10	Kinematic viscosity	ratio of the dynamic viscosity to the density of a fluid.	ν	$\text{m}^2\cdot\text{s}^{-1}$
3.4.4.11	Mass		m	kg
3.4.4.12	Masses of the rotating parts		G	kg
3.4.4.13	Moment of inertia	inertia of the rotation mass used for transient calculation defined as follows: $mR^2 = GD^2/4$	J	$\text{kg}\cdot\text{m}^2$
3.4.4.14	Pipe Young's modulus	modulus of elasticity of pipe wall material	E_p	Pa
3.4.4.15	Poisson's ratio	ratio of a material defining the ration of transverse strain to the axial strain	μ	-
3.4.4.16	Polytropic index		n_g	-
3.4.4.17	Sand grain equivalent	roughness height which gives the same actual resistance coefficient as the one caused by the real non-uniform wall roughness	k_s	-
3.4.4.18	Volume of the compressed air		V_g	m^3

3.4.5 Discharge, velocity, speed terms and time symbols

Subclause	Term	Definition	Symbol	Unit
3.4.5.1	First Eigenfrequency of the penstock	half of the inverse of the reflection time	f_{refl}	s^{-1}
3.4.5.2	Angular velocity	rate of change of angular position of a rotating body	ω	s^{-1}
3.4.5.3	Closure time	time required to close a flow control device	T_{closure}	s
3.4.5.4	Discharge	<volume flow rate> volume of water per unit time passing through any section in the system	Q	$\text{m}^3\cdot\text{s}^{-1}$
3.4.5.5	Discharge coefficient		K_v	-
3.4.5.6	Initial discharge		Q_0	$\text{m}^3\cdot\text{s}^{-1}$
3.4.5.7	Initial flow velocity		C_0	$\text{m}\cdot\text{s}^{-1}$

Subclause	Term	Definition	Symbol	Unit
3.4.5.8	Mass oscillation time	oscillation time between free water surfaces, usually between an upstream or downstream reservoir and a surge tank	T_{mo}	s
3.4.5.9	Maximal frequency		f_{max}	s ⁻¹
3.4.5.10	Mean flow velocity		C	m·s ⁻¹
3.4.5.11	Negative characteristic line	characteristic line in negative direction	C^-	m·s ⁻¹
3.4.5.12	Period	time interval	t	s
3.4.5.13	Positive characteristic line	characteristic line in positive direction	C^+	m·s ⁻¹
3.4.5.14	Real speed of sound in a duct		C_R	m·s ⁻¹
3.4.5.15	Reflection time	time required for the pressure wave to travel back and forth the length of the waterways	T_{refl}	s
3.4.5.16	Rotational speed	number of revolutions per unit time	n	s ⁻¹
3.4.5.17	Surge tank inlet discharge		Q_c	m ³ ·s ⁻¹
3.4.5.18	Time		t	s
3.4.5.19	Unit acceleration time	time to accelerate the rotating masses from zero to rated speed at rated torque	T_a	s
3.4.5.20	Water starting time	time required to accelerate the discharge under the effect of gravity from zero to rated discharge under rated head conditions	T_W	s
3.4.5.21	Waterways reflection time	time required for pressure waves to travel two lengths of the penstock.	T_{refl}	s
3.4.5.22	Wave speed	velocity of pressure wave propagating in pressurized elastic pipe	a	m·s ⁻¹
3.4.5.23	Wave speed in an infinite fluid	velocity of pressure wave propagating in an infinite fluid	a_F	m·s ⁻¹
3.4.5.24	Weir discharge		Q_{weir}	m ³ ·s ⁻¹

3.4.6 Pressure terms

Subclause	Term	Definition	Symbol	Unit
3.4.6.1	Ambient pressure	absolute pressure of the ambient air	p_{amb0}	Pa
3.4.6.2	Gauge pressure	difference between the absolute static pressure of a fluid at the reference level of the pressure measuring instrument and the ambient pressure at the place and time of measurement	p	Pa
3.4.6.3	Pressure		p	Pa

3.4.7 Specific energy terms

In the International System of Units, mass (kg) is one of the base quantities. The energy per unit mass, known as specific energy, is used in this document as a primary term instead of the energy per local unit weight, which is called head, and was exclusively used in previous publications.

The latter term (head) has the disadvantage that weight is a force which depends on the local value of acceleration due to gravity g , which changes mainly with latitude but also with altitude. Nevertheless, the term "head" will remain in use because it is very common. Therefore, both related energy terms are listed, the specific energy terms in this subclause and head terms in 3.4.8. They differ only by the factor g , which is the local value of acceleration due to gravity.

Subclause	Term	Definition	Symbol	Unit
3.4.7.1	Specific hydraulic energy	specific energy of water available between the high and low pressure reference sections 1 and 2 of the machine, taking into account the influence of compressibility	E	$\text{N}\cdot\text{m}\cdot\text{kg}^{-1}$

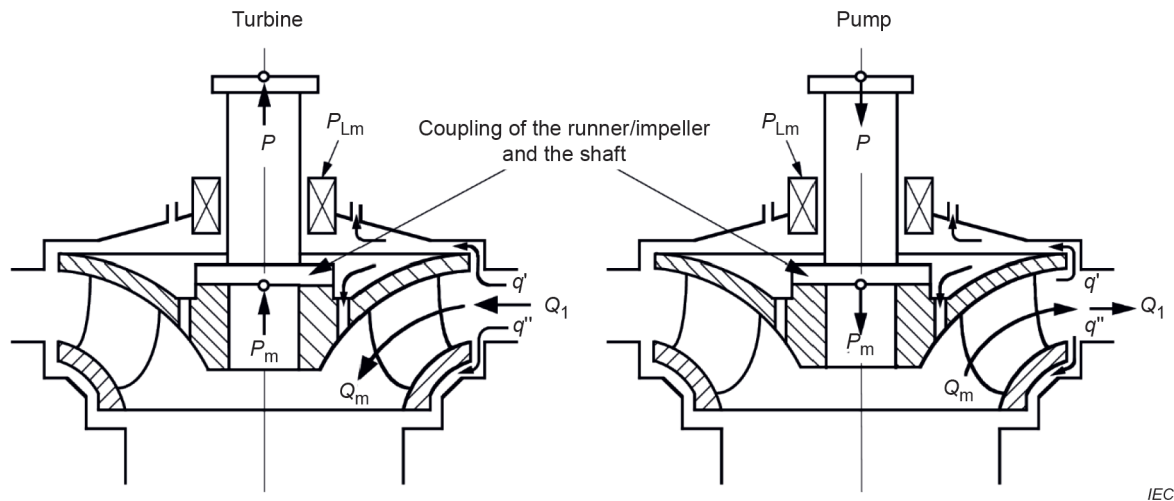
3.4.8 Height and head terms

Subclause	Term	Definition	Symbol	Unit
3.4.8.1	Churchill equation coefficient A	first coefficient of Churchill's equation to assess the Darcy-Weisbach friction factor	$A_{\text{churchill}}$	-
3.4.8.2	Churchill equation coefficient B	second coefficient of Churchill's equation to assess the Darcy-Weisbach roughness coefficient	$B_{\text{churchill}}$	-
3.4.8.3	Datum		Datum	m
3.4.8.4	Diaphragme singular head loss		ΔH_d	m
3.4.8.5	Darcy-Weisbach friction factor	friction factor utilized to determine friction losses in a pipe as a function of the pipewall roughness, pipe hydraulic diameter and the Reynolds number	λ	-
3.4.8.6	Gas absolute pressure head		H_g	m
3.4.8.7	Gross head	piezometric rated head for steady conditions at the turbine or pump level	H_0	m
3.4.8.8	Head	energy per local unit weight of water at any section $H = E / g$	H	m
3.4.8.9	Head loss coefficient		K_d	-
3.4.8.10	Hydraulic capacitance	<electrical analogy> discharge storage by the wall deformation and the fluid compressibility $dx \cdot g \cdot A / a^2$	C_h	m^2
3.4.8.11	Hydraulic inductance	<electrical analogy> water inertia in the pipe $dx / (g \cdot A)$	L_h	$\text{s}^2 \cdot \text{m}^{-2}$
3.4.8.12	Hydraulic resistance	<electrical analogy> headlosses in the pipe $(\lambda \cdot dx / (2g \cdot D \cdot A^2)) \cdot Q $	R_h	$\text{s} \cdot \text{m}^{-2}$
3.4.8.13	Inertia head		ΔH_I	m
3.4.8.14	Maximal gross head		H_{max}	m
3.4.8.15	Minimal gross head		H_{min}	m

Subclause	Term	Definition	Symbol	Unit
3.4.8.16	Negative waterhammer pressure wave	waterhammer amplitude of a negative waterhammer wave	ΔH^-	m
3.4.8.17	Net head		H_n	m
3.4.8.18	Positive waterhammer pressure wave	waterhammer amplitude of a positive waterhammer wave	ΔH^+	m
3.4.8.19	Reduced waterhammer	waterhammer amplitude when downstream pipe flow control device closes in a longer time than the pipe reflection time	$\Delta H_{\text{reduced}}$	m
3.4.8.20	Singular head losses		ΔH_r	m
3.4.8.21	Surge tank head		H_{st}	m
3.4.8.22	Surge tank hydraulic inductance		L_{st}	$\text{s}^2 \cdot \text{m}^{-2}$
3.4.8.23	Surge tank water level		H_c	m
3.4.8.24	Total hydraulic head loss	total hydraulic head loss of the waterway	H_r	m
3.4.8.25	Weir discharge coefficient		K_{vweir}	-
3.4.8.26	Weir Elevation		Z_{weir}	m

3.4.9 Power and torque terms

Subclause	Term	Definition	Symbol	Unit
3.4.9.1	Mechanical torque		T_m	N·m
3.4.9.2	Rated power		P_{rat}	W
3.4.9.3	Runner or impeller torque	torque transmitted through the coupling of the runner or impeller and the shaft and corresponding to the mechanical power of runner or impeller	T_m	N·m
3.4.9.4	Torque		T	N·m
3.4.9.5	Hydraulic power	hydraulic power available for producing power (turbine) or imparted to the water (pump) $P_h = E(\rho \cdot Q)_1$	P_h	W
3.4.9.6	Mechanical power of the machine (power)	mechanical power delivered by the turbine shaft or to the pump shaft, assigning to the hydraulic machine the mechanical losses of the relevant bearings and shaft seals (see Figure 3-3)	P	W
3.4.9.7	Mechanical power of runner(s)/impeller(s)	mechanical power transmitted through the coupling of the runner(s)/impeller(s) and the shaft (see Figure 3-3)	P_m	W



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$$q = q' + q''$$

Leakage discharge

$$Q_1 = Q_m + q$$

Discharge at section 1

$$P_h = E(\rho Q)_1$$

Hydraulic power

$$P = P_m - P_{Lm}$$

Mechanical power of the machine

$$\eta_m = \frac{P}{P_m}$$

Mechanical efficiency

$$\eta_h = \frac{P_m}{P_h}$$

Hydraulic efficiency

$$\eta = \frac{P}{P_h}$$

Efficiency

$$q = q' + q''$$

$$Q_1 = Q_m - q$$

$$P_h = E(\rho Q)_1$$

$$P = P_m + P_{Lm}$$

$$\eta_m = \frac{P_m}{P}$$

$$\eta_h = \frac{P_h}{P_m}$$

$$\eta = \frac{P_h}{P}$$

NOTE 1 The formulae ignore the compressibility of the water.

NOTE 2 For detailed analysis of internal losses, refer to IEC 60193:2019, Annex L.

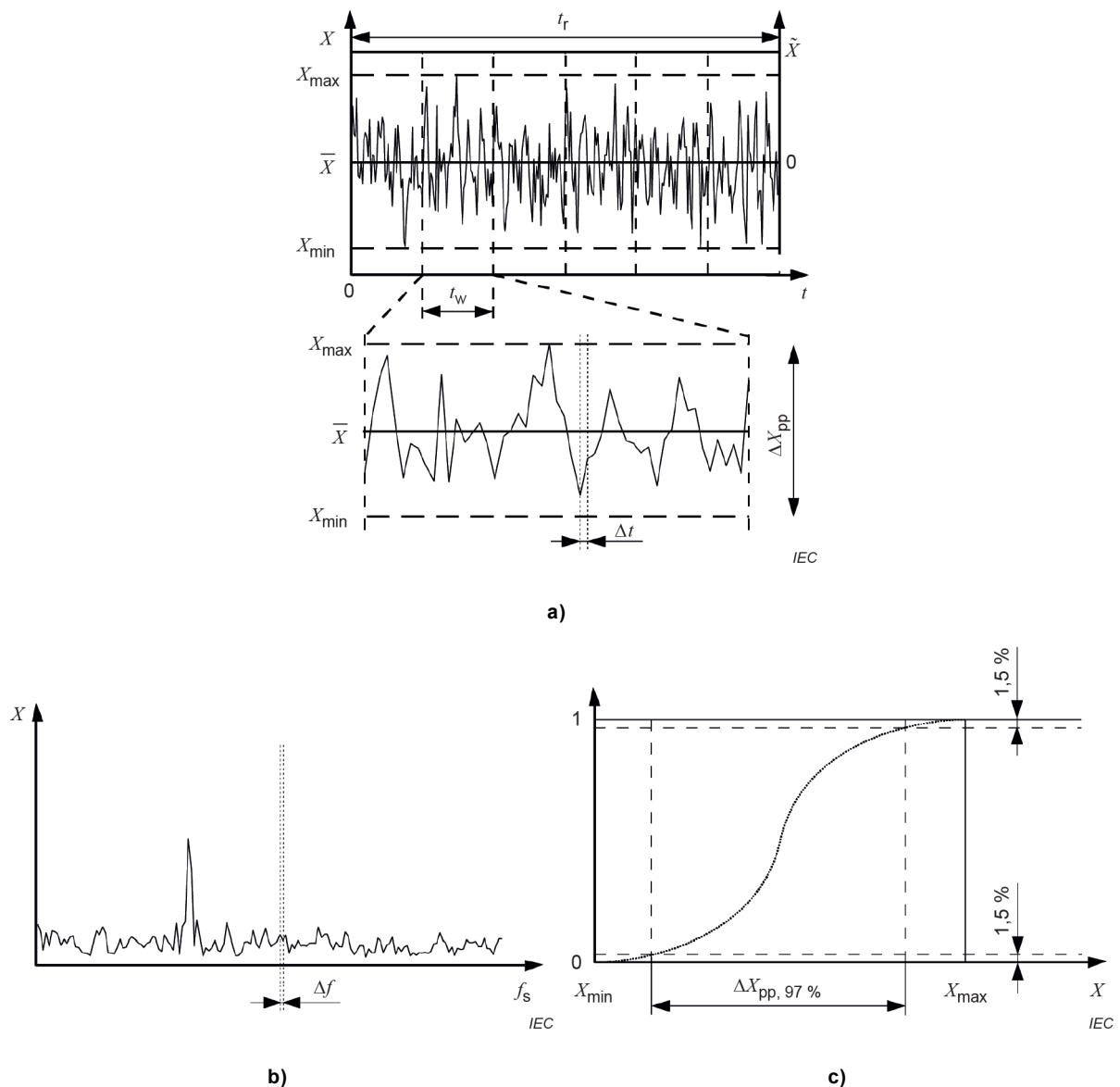
NOTE 3 The disk friction losses and leakage losses (volumetric losses) are considered as hydraulic losses in this formula. These "disk friction losses" are the friction losses of the outer surfaces of the runner/impeller not in contact with the flow Q_m passing the runner/impeller blades.

Figure 3-3 – Flux diagram for power and discharge

3.4.10 General terms relating to fluctuating quantities

IEC 60994:1991 provides a reference for terms relating to these quantities. The following table lists the terms relevant to this document, some of which are illustrated in Figure 3-4.

Subclause	Term	Definition	Symbol	Unit
3.4.10.1	Averaged peak value	maximum or minimum averaged value during the transient	X_{averaged}	-
3.4.10.2	Corrected Calculation		CC	-
3.4.10.3	Difference between measured and simulated transient quantities		Δ_{EV}	-
3.4.10.4	Fluctuating quantity	see Figure 8-3	Y	-
3.4.10.5	Fluctuating value	maximum or minimum fluctuating value during the transient	X_{filtered}	-
3.4.10.6	Frequency		f	s^{-1}
3.4.10.7	Iterations		i	-
3.4.10.8	Longitudinal position	position on the pipe centreline	x	m
3.4.10.9	Model uncertainty		MU	-
3.4.10.10	Multiplier (integer)		k	-
3.4.10.11	Number of elements of each pipe		Nb	-
3.4.10.12	Number of units		n	-
3.4.10.13	Random uncertainty		f_r	-
3.4.10.14	Reaches of length	number of equidistant reaches of length along a conduit	Δx	-
3.4.10.15	Relative opening of flow control device		y	-
3.4.10.16	Simulated transient change from steady state condition		Δ_{ss}	-
3.4.10.17	Systematic uncertainty		f_s	-
3.4.10.18	Total uncertainty		f_t	-



- a) A pressure fluctuation signal over a defined signal record time (t_r). A minimum of 5 window time lengths (t_w) shall be included in each signal record time duration.
- b) The signal transformed into the frequency domain using Fourier analysis. The frequency resolution (Δf) required for each analysis is determined by the speed of rotation of the machine.
- c) Using probability distribution, each measure point from Figure 3-4 a) is cumulated and extreme deviations from the mean fluctuation are removed. A 97 % interval of confidence is illustrated in this figure.

Figure 3-4 – Illustration of some definitions related to oscillating quantities

3.4.11 Fluid dynamic and scaling terms

Subclause	Term	Definition	Symbol	Unit
3.4.11.1	Reynolds number	ratio of inertia forces to viscous forces $Re = D \cdot C / \nu$	Re	

3.4.12 Dimensionless terms

Machine performance may be characterized by dimensionless terms based on $E = 1$, $D = 1$ and $\rho = 1$ or on $n = 1$, $D = 1$ and $\rho = 1$.

The units are: H (m); D (m); E ($\text{J} \cdot \text{kg}^{-1}$); n (s^{-1}); ρ ($\text{kg} \cdot \text{m}^{-3}$); T ($\text{N} \cdot \text{m}$); P (W); Q ($\text{m}^3 \cdot \text{s}^{-1}$).

Subclause	Term	Definition	Symbol	Unit
3.4.12.1	Discharge factor	factor defined as : $Q_{ED} = Q / (D^2 \cdot E^{0,5})$	Q_{ED}	-
3.4.12.2	Specific speed	speed defined as: $N_{QE} = n Q^{0,5} / E^{0,75}$	N_{QE}	-
3.4.12.3	Speed factor	factor defined as: $n_{ED} = n D / E^{0,5}$	n_{ED}	-
3.4.12.4	Torque factor	$T_{ED} = T_m / (\rho \cdot D^3 \cdot E)$	T_{ED}	-

3.5 Abbreviated terms

Abbreviated terms	Meaning
AV	Air inlet valve
C	Catastrophic load cases
CFD	Computational fluid dynamics
CFL	Courant-Friedrichs-Lewy condition
CO	Any combination of load cases
E	Exceptional load cases
ESD-A	Automatic emergency shutdown
ESD-AF	Automatic emergency shutdown – Serious faults in power station
ESD-AO	Automatic emergency shutdown – Overspeed detection
ESD-PB	Push-button emergency shutdown
EXC	Exceptional load cases
FDM	Finite difference method
FFT	Fast Fourier transform
GCOL	Governor-controlled operating laws
LR	Load rejection
MIV	Main inlet valves
MOC	Method of characteristics
NOR	Normal load cases
ODE	Ordinary differential equations
P	Pump load case
PDE	Partial differential equations
PDV	Pump discharge valve
PRV	Pressure relief valve
PSV	Penstock shut-off valve
PU	Pump mode
QSD-E	Quick shutdown, electrical faults
QSD-M	Quick shutdown, mechanical faults
RA	Unit runaway
ROL	Rapid operating laws
SC	Synchronous condenser mode
SNL	Speed-no-load
SO	Shut-off

Abbreviated terms	Meaning
SS	Standstill
SYN	Unit synchronized
T	Turbine load case
TU	Turbine mode

4 Hydraulic transient basics

4.1 Elements related to hydraulic transient

4.1.1 General

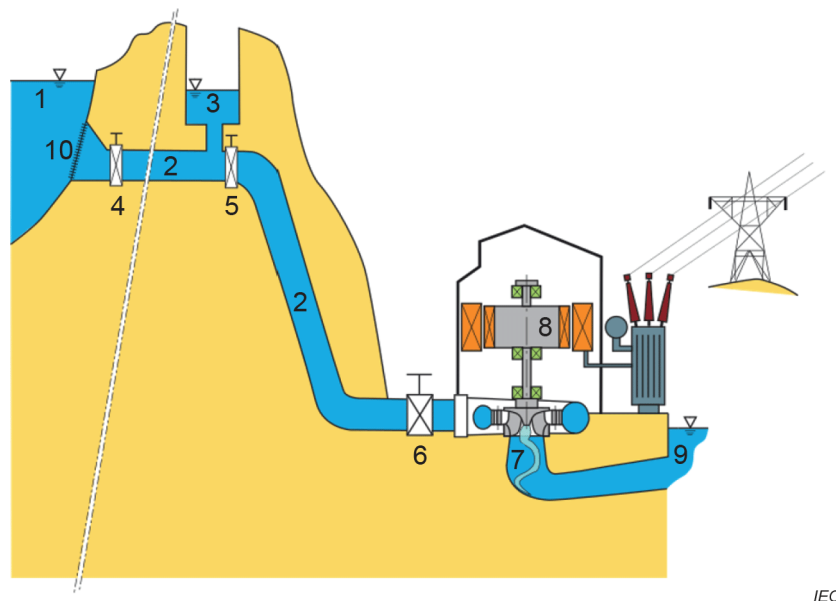
Current best practice, when setting up a hydraulic transient calculation, is based on using constant values for the following parameters for all elements modelled:

- density of water;
- gravitational acceleration constant.

4.1.2 Waterways system components

4.1.2.1 General

Figure 4-1 schematically shows waterway system components that are modelled for a hydraulic transient calculation.



Key

- | | |
|---------------------------------|----------------------------|
| 1 Reservoir / variable inlet | 2 Tunnel / pipe / penstock |
| 3 Surge shaft / surge tank | 4 Intake gate |
| 5 Penstock shut-off valve (PSV) | 6 Main inlet valve (MIV) |
| 7 Hydraulic machine | 8 Motor-generator |
| 9 Reservoir / variable outlet | 10 Trashrack |

Figure 4-1 – General schematic of a typical hydropower plant arrangement

4.1.2.2 Reservoir

The reservoir defines the upper or lower boundary conditions for the hydraulic system. It is characterised by a free surface water level and its level is considered constant for the time of the transient event.

Any hydraulics phenomenon that can be observed nearby the reservoir such as potential vorticity, flow separation and velocity approach will not be considered but can be added as an additional loss to the nearest element.

Typical input parameter for reservoirs is:

- constant water level.

4.1.2.3 Variable free surface head

If a reservoir is of considerably small volume, the upstream or downstream reservoir can be modelled using a variable head.

A surge tank, as described in 4.1.2.5, can also be used to model any of scheme's main inlet or outlet reservoir, where a significant variation in head can be expected during a transient event or where spilling might be of interest (e.g., small weirs).

The typical input parameter for variable head inlet or outlet reservoir is:

- water level as function of plant discharge or time.

4.1.2.4 Conduit or tunnel or pipe

Sections located between two free surfaces (e.g., reservoir or surge tank) can be either tunnels, lined tunnels or pipe sections. Note that the type of conduit or tunnel or pipe considered in this document are pressurized.

Typical input parameters for conduit are:

- Equivalent hydraulic diameter, depending on conduit shape (circular, horseshoe, D-shaped, etc.);
- Conduit cross-section area, depending on conduit shape (circular, horseshoe, D-Shaped, etc);
- Length of conduit;
- Elevation along the conduit;
- Pressure wave celerity in conduit;
- Friction factor.

A tunnel is normally excavated from the rock, either by drill and blast or by using a tunnel boring machine. The construction type depends strongly on the required finished (internal) diameter of the tunnel as well as the quality of the rock. The construction method will also have an impact on the expected headlosses along the tunnel. Based on the most economic design a tunnel internal surface is normally either unlined or concrete lined. In some cases, a tunnel can be partly or completely steel lined when rock quality is insufficient, external pressure is high or severe internal pressure changes variations are expected.

Tunnel friction factors depend on their construction method and lining.

Pipes features circular cross section and can either be above ground or underground. For the same discharge, it has a smaller diameter than a tunnel. As this leads to higher velocities, a smooth finishing is selected in order to reduce headlosses.

Common pipe materials are:

- coated steel;
- concrete;
- high Density Polyethylene (HDPE);
- glass/Fibre-reinforced plastics (GRP);
- wood-stave.

4.1.2.5 Surge tank

A surge tank can also be called a surge shaft. It is common practice to use the term surge shaft instead of surge tank if the inertia of the water in the surge tank has a significant influence on dynamic behaviour. The closer the surge tank is to the hydraulic machine, the more effective it is.

A surge tank in this context is a variable head storage capacity with volume and cross-section small enough for the water level to vary over the course of a transient event and hence have a significant impact on the transient behaviour of the system.

A surge tank is open to atmospheric pressure and hence can spill into the surrounding environment. It shall be designed so that during operation the tank will not drain and admit air into the turbine penstock or pump discharge line.

Surge tanks are installed in waterways:

- to reflect the pressure waves in the waterways due to discharge variations, thus reducing the magnitude of waterhammer at the hydraulic machine;
- to temporarily store and release water in the waterways;
- to protect tunnel/pipe from pressure variations during transient phenomena;
- to enable faster reaction time of the hydraulic machine, thus improving its regulating performance.

Common types of surge tanks are shown in Figure 4-2:

- surge shaft – (Sketch 1)
- inclined surge shaft with upper and lower chamber – (Sketch 2)
- throttled surge tank – (Sketch 3)
- differential surge tank – (Sketch 4)
- as surge tanks are integrated in the topology, many other surge tank designs exist.

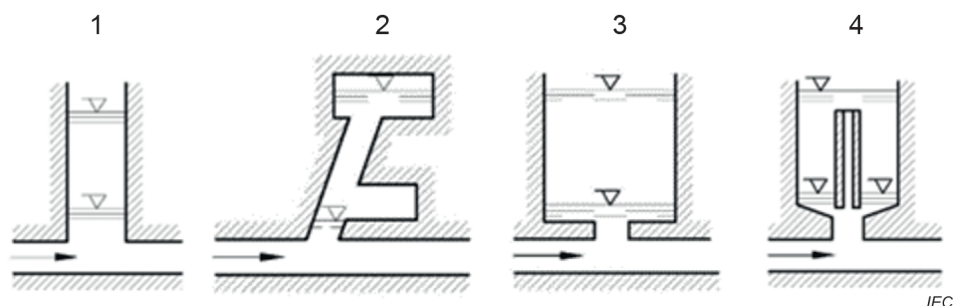


Figure 4-2 – Possible configurations of surge tanks – Sketches 1 to 4

Typical input parameters for surge tank are:

- area as function of the elevation;
- reference area of orifice;
- loss coefficient of inflowing or outflowing discharge;
- inertia of the water in the surge tank.

Further information on types of surge tanks and their design parameters are covered in several standards or guides in the industry, for instance IS 7396 series.

4.1.2.6 Pressurized air chambers

Pressurized air chambers, shown schematically in Figure 4-3, fulfils the same requirements as a surge tank regarding the operation and safety of hydro-electric power plants. However, unlike a surge tank it is not open to atmosphere, but it is fully enclosed.

Upon watering-up of the waterways, air gets naturally entrapped in the air chamber. Usually, compressor devices are used to compensate the air losses which can result from water diffusion or rock leakage to achieve design water level in the pressurized chamber.

During operation and in case of a transient event the air in the chamber gets compressed during rising water levels and decompressed during dropping levels.

There is no risk of overflowing. Compressibility of the trapped air shall be considered. Minimum and maximum pressure shall be respected and air shall not enter the penstock as well as ensuring that the chamber cannot be completely drained.

It is also possible to inject other gases into the chamber in case of specific application.

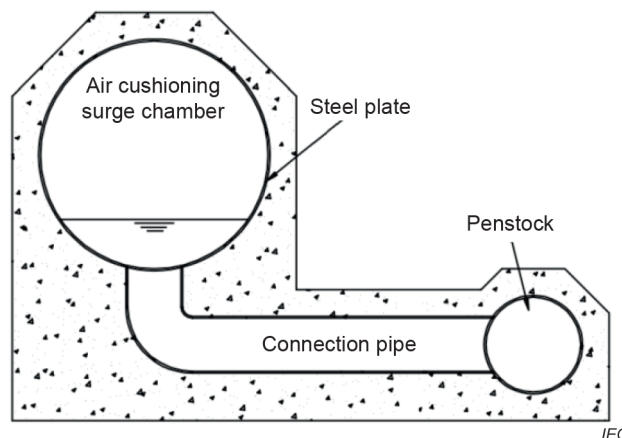


Figure 4-3 – Schematic of a simple pressurised air chamber

Typical input parameters for a pressurized air chamber:

- Area of the pressurized air chamber as function of the elevation;
- elevation of the main pressurized air chamber ;
- polytropic exponent of gas in air chamber;
- reference area of orifice ;
- loss coefficient of inflowing and outflowing discharge;
- initial volume and pressure of the air.

4.1.2.7 Penstock and pressure shaft

Penstocks are pipe sections in the waterways located between a free surface (upstream reservoir or surge tank) and the hydraulic machine. Penstocks can be above ground, buried or laid in rock depending on the environmental, legal, geological and geographical requirements as well as the choice of material. Its routing, material, diameter, thickness and degree of anchoring will be determined at the earliest phase of a project through an optimization process that considers the headloss, the hydraulic transient, the manufacturing and installation cost. Losses in penstock are computed in the same manner as for the tunnel or pipe sections.

Common penstock materials are:

- steel;
- concrete;
- cast iron;
- High Density Polyethylene (HDPE);
- Glass fibre-reinforced plastics (GRP);
- wood-stave.

Typical input parameters for penstocks are:

- diameter;
- length of penstock;
- elevation along the penstock;
- pressure wave celerity in conduit;
- friction factor.

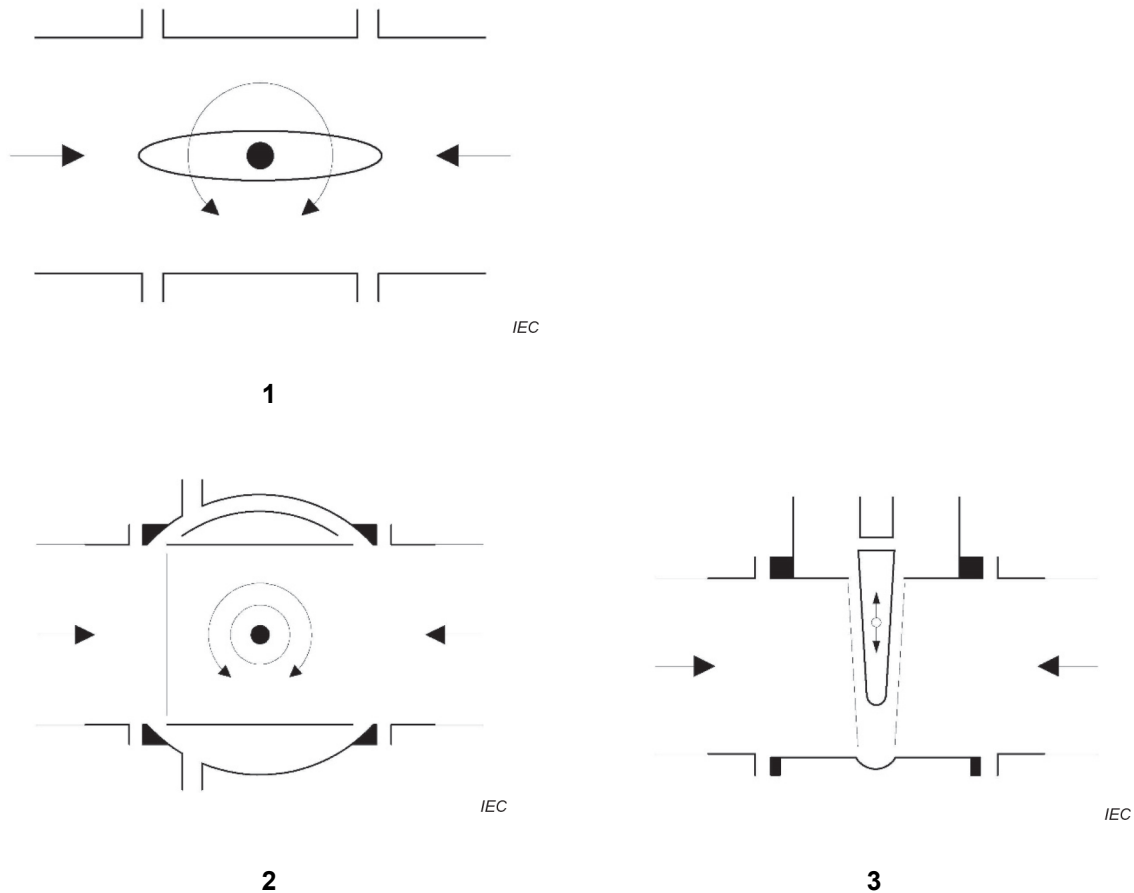
4.1.3 Equipment

4.1.3.1 Valve

4.1.3.1.1 General

A valve is an isolating device and can be designed as an emergency discharge shut-off device. Common designs are shown in Figure 4-4:

- butterfly valve (for pressure heads typically <250 m) – 1
- spherical valve (for pressure heads typically >200 m) – 2
- gate valve, for low head application (for pressure heads typically <100 m) – 3



SOURCE: IEC TR 61364:2019, Figure 10.

Figure 4-4 – Common valve designs

Application range of all type of valve can vary depending on economic consideration of the power plant. Spherical valves have normally lower headlosses than butterfly and gate valves.

Depending on their location in the hydraulic system, the following types of valves are commonly differentiated:

- Main Inlet Valve also called Turbine Inlet Valve or Turbine Shutoff Valve (MIV/TIV/TSV, located immediately upstream of the turbine);
- Pump Discharge Valve (PDV, located at the high-pressure side of the pump);
- Penstock Shut-off Valve (PSV, located at the top of a penstock);
- Turbine Outlet Valve (TOV, located downstream of low head turbines).

Typical input parameters for valves in hydropower schemes are:

- reference diameter or area;
- valve loss characteristics (discharge coefficient vs. opening);
- closing behaviour (linear, nonlinear).

Optional input parameters for valves for a more detailed analysis would include:

- valve hydraulic torque characteristics vs. opening;
- torque from closing weight vs. opening;
- torque from (braking) servomotor vs. opening.

More information on types of valves can be found in IEC 60534-1.

Optional tests to improve input data reliability are provided in Annex B.

4.1.3.1.2 Valve seal

Spherical valves can have moveable sealings, see Figure 4-5. Spherical valves without movable seals are also known as ball valves.

Especially at high head designs there will be a significant discharge when the valve rotor is in closed position, but the moveable sealing is still open. In execution phase the behaviour of the moveable sealing shall be considered to define the safe closing time for the moveable sealing. The seals shall also be checked against potential risks of resonance in the penstock and self-excitation. Such phenomena are described in literature, e.g. [12]², [13], [14].

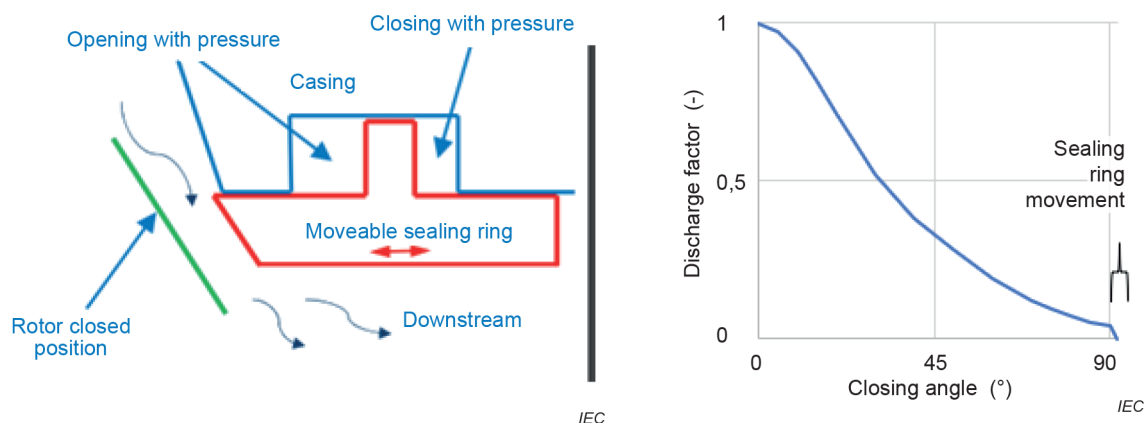


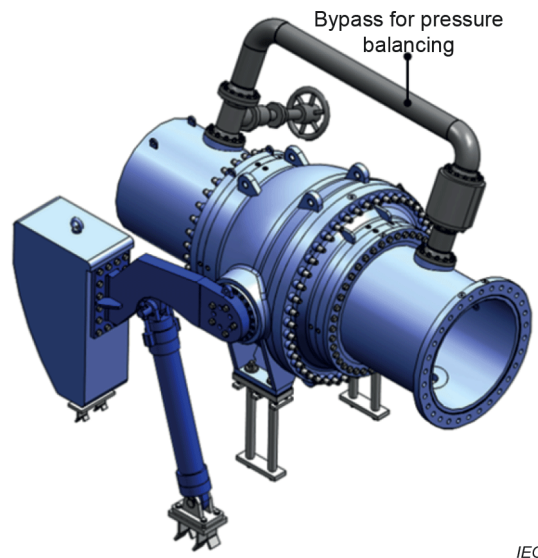
Figure 4-5 – Schematic of sealing ring arrangement (left) and corresponding loss characteristic of valve with sealing ring between 89,9 ° and 90 °

4.1.3.1.3 Pressure balancing bypass

Valves can be equipped with small (relative to the main valve size) connections from high to low pressure side, see Figure 4-6. This connection is opened to balance pressure upstream and downstream of the valve, prior to opening the main valve.

The opening of the pressure balancing bypass leads to flow for a short time until pressure is balanced. As for any regulating device, if this flow change happens within the reflection time, it can cause high pressure peaks. Either a controlled slow opening shall be foreseen, or the resulting flow shall be limited by an orifice.

² Numbers in square brackets refer to the Bibliography.



SOURCE: This figure is courtesy of Voith Hydro Holding GmbH & Co. KG. It belongs to the company and can only be reproduced with the permission of Voith.

Figure 4-6 – Schematic of spherical valve with pressure balancing bypass

4.1.3.2 Gate

The headloss caused by a gate in the fully open position are part of the total losses in the system. A gate shaft with a significant free surface can be modelled as a surge shaft.

If a gate can be used as a shut-off device, i.e. it can close under flow, the gate closing behaviour shall be included in the simulation. Vents located downstream of the gate can be used to limit negative pressure during its closure.

Gates are characterized by their locations in the hydraulic system:

- intake gate;
- surge tank gate;
- draft tube gate;
- outlet gate.

For Kaplan and Tubular turbines, the draft tube gate can be used for energy dissipation during sluicing mode operation. Such operation is intended to allow significant discharge flowing through the turbine while not connected to the electrical grid and keeping the unit in runaway at reduced head.

If gates are to be included in a transient calculation, typical input parameters are the same as for a valve, see 4.1.3.1.

4.1.3.3 Bypass valve

A bypass valve is used to maintain a specified minimum discharge in a waterway system, even if the turbine is shut down, e.g. for ecological reasons.

The closing slope of the bypass valve is slow compared to the turbine(s). This kind of bypass valve is not included in the transient analysis of the hydropower station.

In some cases, a special type of bypass valve is used to limit pressure rise in the system. Such a bypass valve is commonly located close to the turbine. As its operation is linked to the turbine transients, this kind of valve is described as part of the hydraulic machine in 4.1.4.

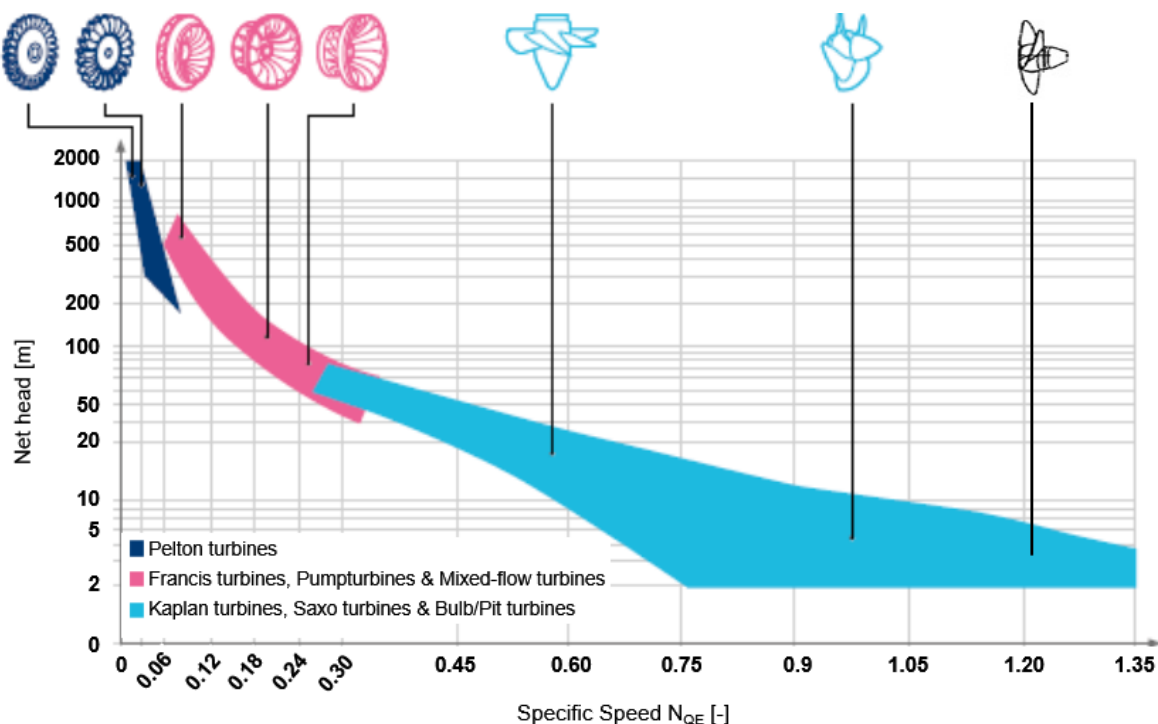
If bypass valves are to be included in a transient calculation, typical input parameters are the same as for a valve, see 4.1.3.1.

4.1.4 Hydraulic machine

4.1.4.1 General

Hydraulic machines are typically classified based on their specific speed value N_{QE} as shown in Figure 4-7 below. The specific speed value is an indicator of the type of hydraulic machine suitable for expected prototype net head conditions as well as the runner profile.

Hydraulic machines with a low N_{QE} are more suited for high head low discharge applications, whilst higher N_{QE} machines normally operate under low head, high discharge conditions.



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Figure 4-7 – Hydraulic machine application diagram

4.1.4.2 Pelton type turbine

The Pelton turbine is an impulse type turbine. It differs significantly from other turbine types. The water coming from the distributing pipe flows through one or more nozzles and is released as a “free surface” jet into the atmosphere, entirely converting the head into velocity, see Figure 4-8. The jet then hits the turbine bucket and is deflected transferring its kinetic energy to the runner.

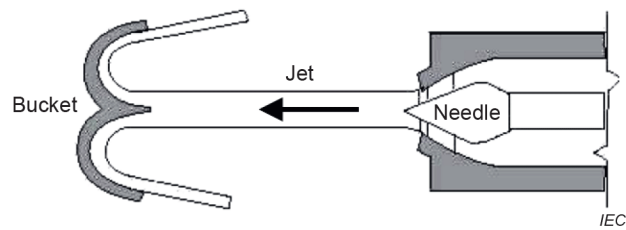
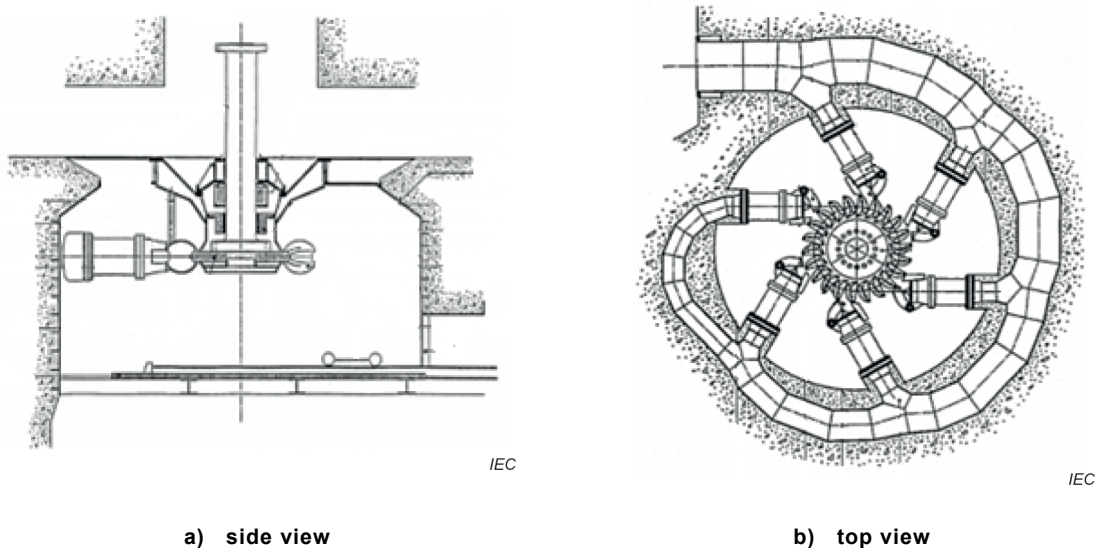


Figure 4-8 – Schematic view of water jet between nozzle and bucket

The runner rotates in the air, see Figure 4-9 and Figure 4-10, which has the following consequences:

- The runner setting level is above the downstream free-flow surface level, which means a few meters of head are not used.
- The hydraulic power and therefore nozzle opening to operate at speed-no-load is very small.
- The jet can be deflected off the runner in case of disconnection from the electric grid to limit speed rise. With deflectors activated, nozzles can be closed slowly to limit waterhammer.



SOURCE: IEC TR 61364:2019, Figure 29 and Figure 30.

Figure 4-9 – Schematic view of 6-jet vertical Pelton-type turbine

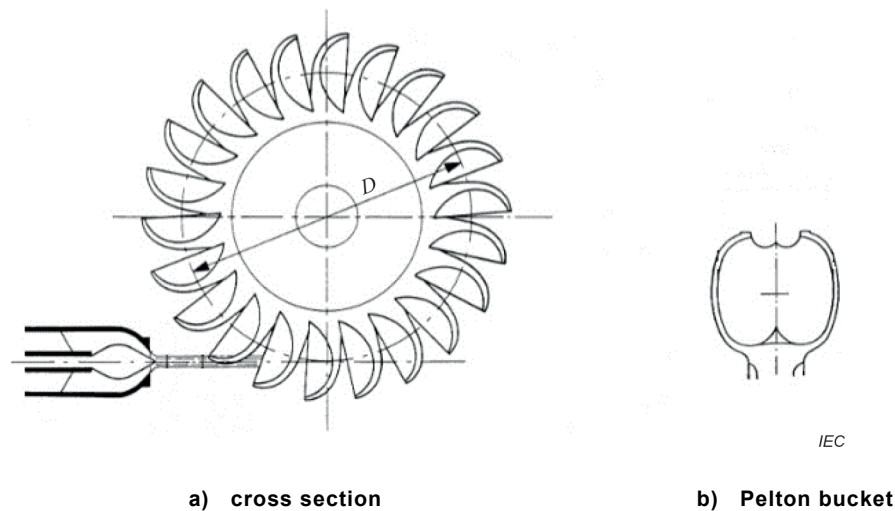


Figure 4-10 – Schematic view of Pelton-type turbine runner and bucket

The flow and load control devices of a Pelton turbine are the injectors and the deflectors.

– Injector:

A Pelton injector typically looks like a needle or spear valve and consisting in an external pipe housing with an internal concentric torpedo bearing the needle or spear shaft. The needle or spear moves forward and backward relative to the housing, to control the discharge. The servomotor can be either inside the torpedo or outside the housing.

– Deflector:

A Pelton jet deflector is a moving plate located just beside the nozzle outlet. It can move into the jet, deviating its trajectory away from the runner buckets. Since the flow is not stopped, the deflector creates no waterhammer and can move very fast into the jet (typically about 2 s), bringing the mechanical torque to zero, for example in case of a load rejection.

In Figure 4-11 the deflector is the push-out type. In case of special requirements, like primary control mode, a so-called cut-in deflector can be used.

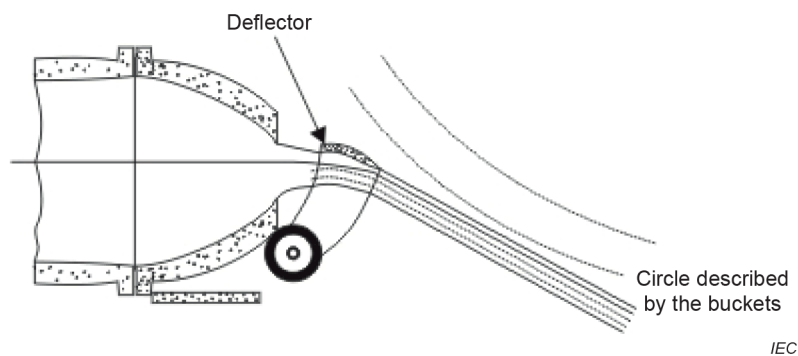


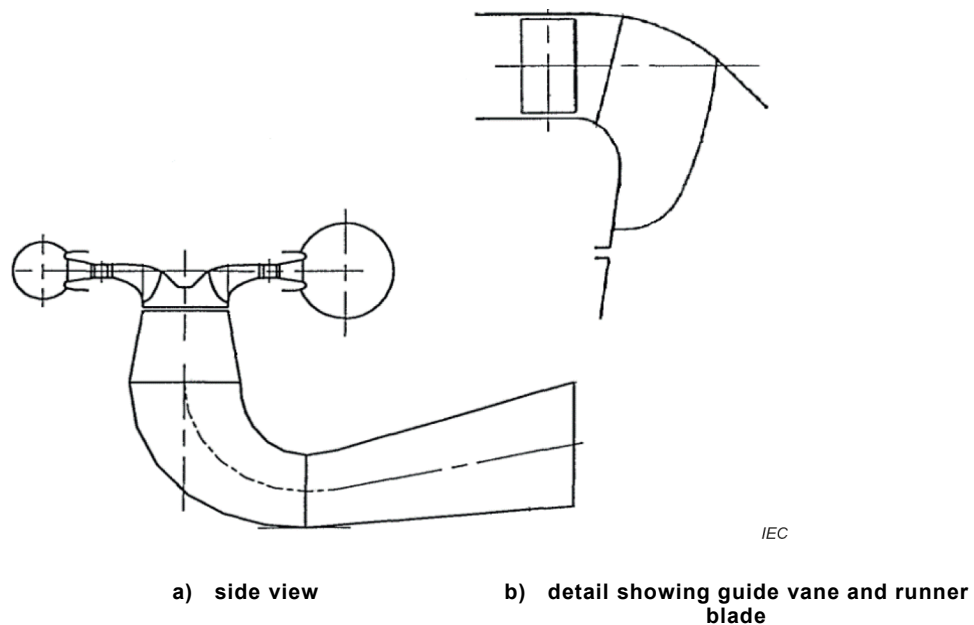
Figure 4-11 – Schematic view of Pelton-type turbine deflector

Typical input parameters for Pelton type turbines are as follows:

- turbine characteristics (nozzle openings, speed factors, discharge factors and torque factors);
- deflector characteristics;
- reference diameter;
- rotational speed;
- kinematics between servomotor stroke and nozzle opening.

4.1.4.3 Francis type turbine

The Francis type turbine is a radial turbine, see Figure 4-12. It has adjustable guide vanes in a cylindrical distributor arrangement.



SOURCE: IEC TR 61364:2019, Figure 38.

Figure 4-12 – Schematic view of Francis-type turbine for runner and spiral inlet

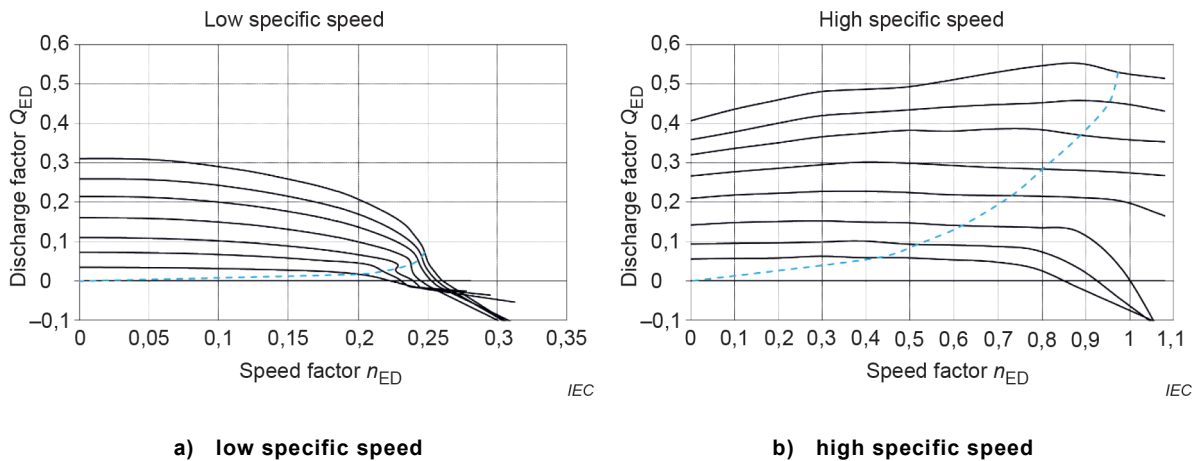


**Figure 4-13 – Francis typical discharge characteristics;
lines of constant guide vane opening (solid) and runaway curve (dashed)**

Examples of a typical Francis-type characteristics for two different specific speeds are shown in Figure 4-13 above.

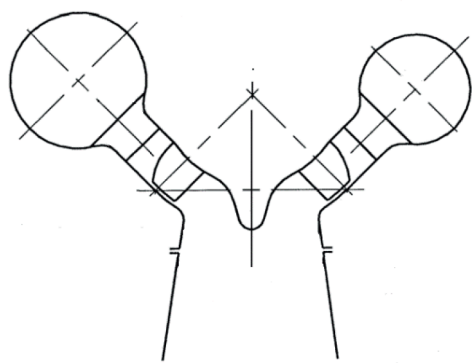
Special units referred as dual Francis are known to have unexpected behaviour, see [15]. Before any modifications on an arrangement with dual Francis, it is strongly recommended to perform on-field transient testing. Depending on the specific speed of the turbine design, the discharge decrease caused by the speed rise can be higher than the one caused by the closing of the distributor.

Typical input parameters for Francis type turbines are as follows:

- turbine characteristics (guide vane openings, speed factors, discharge factors and torque factors);
- reference diameter;
- rotational speed;
- kinematics between servomotor stroke and guide vane opening angle.

4.1.4.4 Deriaz type turbine with adjustable runner blades

A Deriaz is one kind of diagonal type turbine, see Figure 4-14. The transient behaviour is between a Francis and a Tubular turbine. While it could have fixed or adjustable blades, this document focuses on adjustable runner blades and an adjustable conical guide vane distributor arrangement. It can also be used as a reversible pump-turbine.



IEC

a) cross section

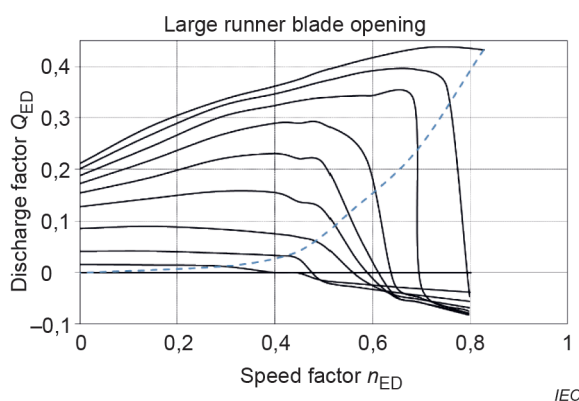


IEC

b) model runner

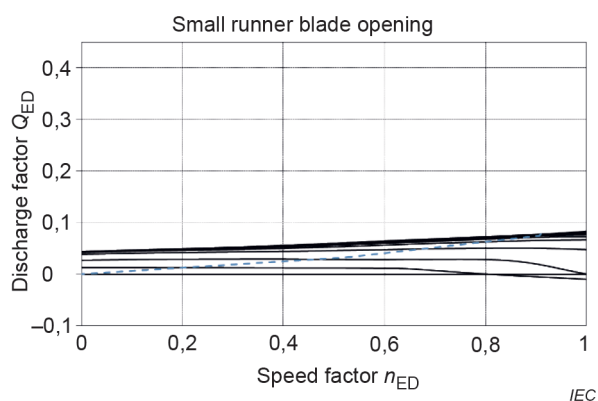
SOURCE: IEC TR 61364:2019, Figure 41.

**Figure 4-14 – Deriaz sketch and model runner
(3 of 6 runner blades removed)**



IEC

a) large runner blade opening



IEC

b) small runner blade opening

Figure 4-15 – Deriaz typical discharge characteristics – lines of constant guide vane opening (solid) and runaway curve (dashed)

Figure 4-15 shows a typical Deriaz discharge characteristic for fully open runner blade position (left) and closed runner blade position (right).

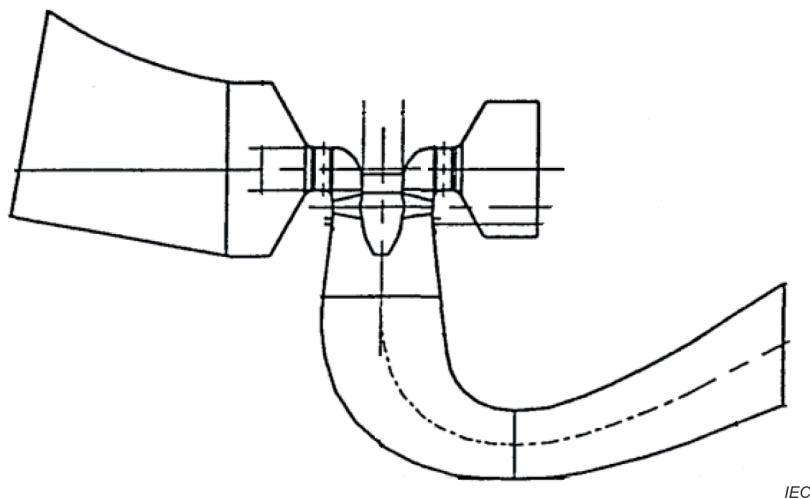
The behaviour of the Deriaz turbine during load rejection is a mixture of the initial behaviour of a Tubular type turbine (discharge increases with speed) followed by the behaviour of a Francis type turbine (discharge decreases with speed).

Typical input parameters for Deriaz type turbines are as follows:

- turbine characteristics (runner blade openings, guide vane openings, speed factors, discharge factors and torque factors);
- reference diameter;
- rotational speed;
- kinematics between servomotor stroke and guide vane opening angle;
- kinematics between servomotor stroke and runner blade opening angle.

4.1.4.5 Kaplan type turbine

A Kaplan type turbine is a vertical shaft turbine with adjustable runner blades and cylindrical distributor arrangement, schematically shown in Figure 4-16. Discharge characteristics for small and large blade angles is shown below in Figure 4-17.



SOURCE: IEC TR 61364:2019, Figure 42.

Figure 4-16 – Schematic view of Kaplan-type turbine

The spiral case inlet arrangement will have an impact on equivalent length and diameter, which shall be considered for the hydraulic transient analysis

Common types of spiral case depending on head are:

- full spiral case, or
- semi-spiral case.

Kaplan turbine discharges will increase with speed after a load rejection as shown below.

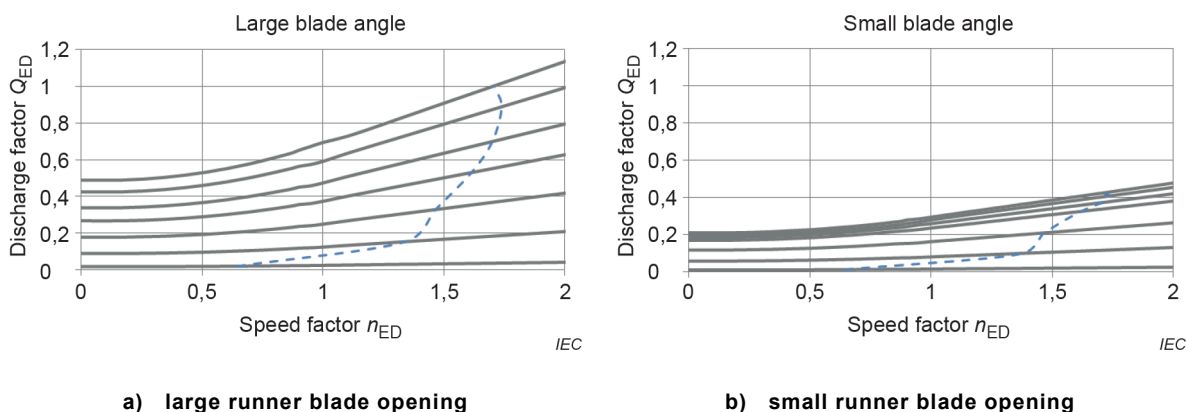


Figure 4-17 – Kaplan typical discharge characteristics – lines of constant guide vane opening (solid) and runaway curve (dashed)

In steady-state operation, Kaplan turbine normally operates based on an implemented optimised relationship between runner blade angle and guide vane angle.

The hydraulic transient analysis starts from on-cam relationship, but after disconnection from the grid, the runner blade angle versus guide vane angle relationship is lost, and guide vanes and blades move independently according to their respective operative laws.

Typical input parameters for Kaplan turbine are as follows:

- turbine characteristics (guide vane openings, blade openings, speed factors, discharge factors and torque factors);
- reference diameter;
- rotational speed;
- kinematics between servomotor stroke and guide vane opening angle;
- kinematics between servomotor stroke and runner blade opening angle;
- optimised relationship between guide vane angle, runner blade angle and head.

4.1.4.6 Saxo type turbine

Saxo type turbine is a vertical axis turbine with adjustable blades and adjustable conical guide vanes distributor arrangement, see Figure 4-18. The transient behaviour and load cases are equal to the Kaplan turbine.

The Saxo type unit is characterised by an elbow shape water passage. The turbine drives an external mounted generator on a vertical shaft arrangement.

Typical input parameters are similar to the input parameters of a S-type turbine, described in 4.1.4.7.

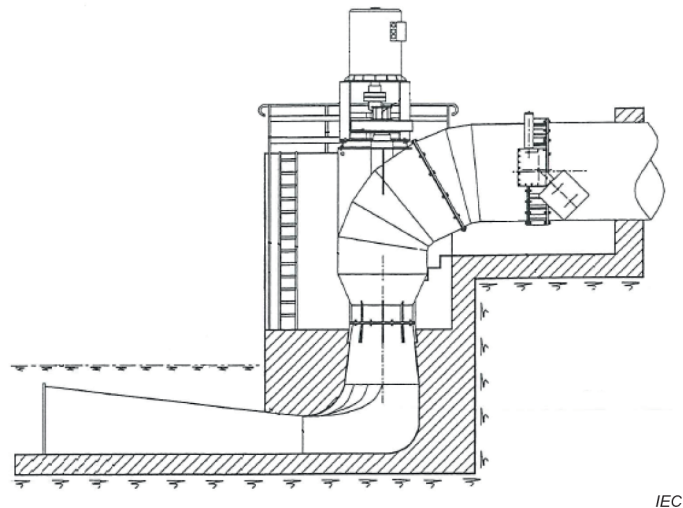


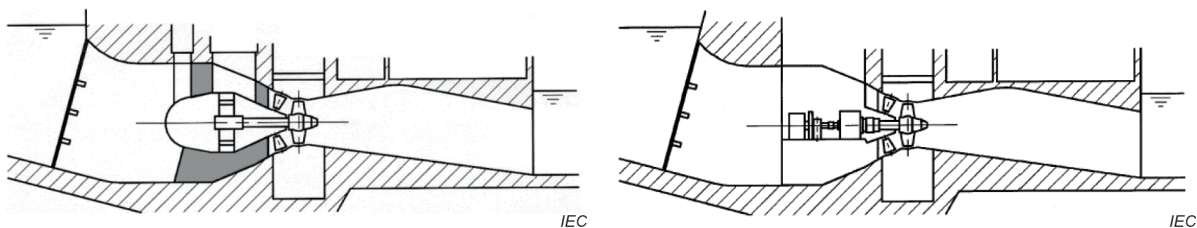
Figure 4-18 – Schematic view of a Saxo-type turbine

4.1.4.7 Tubular turbine

Tubular turbines can be pit, bulb or S-type turbines, see Figure 4-19 and Figure 4-20. They are horizontal axis turbines with adjustable blades and an adjustable guide vane distributor arrangement, similar to a Kaplan turbine, but conical guide. The transient behaviour and load cases are equal to the Kaplan turbine.

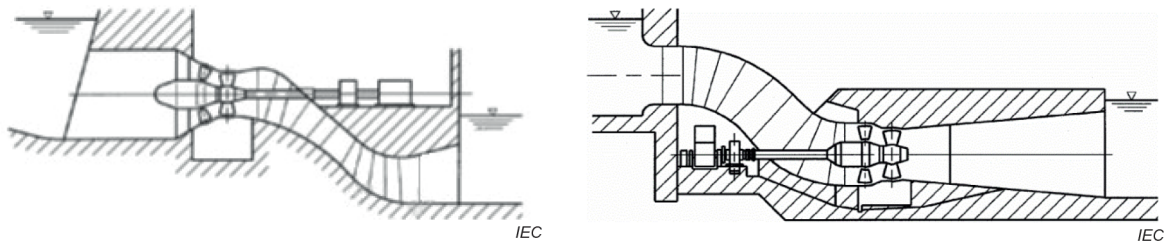
Based on the arrangement and location of the generator, the following types of turbines can be differentiated:

- The generator is housed in a bulb in the water passage. The unit can be directly driven, or it could be equipped with a gearbox.
- The generator is housed in a pit in the water passage. The generator is most frequently connected to the turbine shaft through a gearbox. The pit allows direct dismantling of the generator and the gearbox from above.
- The S-downstream type unit is characterised by an S Shape water passage. The turbine drives an external mounted generator downstream of the turbine which can be driven directly or by a gearbox.
- The S-upstream type unit is characterised by an S Shape water passage. The turbine drives an external mounted generator upstream of the turbine which can be driven directly or by a gearbox.



SOURCE: IEC TR 61364:2019, Figure 3 and Figure 4.

Figure 4-19 – Schematic layout of a bulb (left) and pit (right) turbine



SOURCE: IEC TR 61364:2019, Figure 6 and Figure 7.

Figure 4-20 – Schematic layout of a S-downstream (left) and S-upstream (right) turbine

The subtype of turbine or the arrangement has no influence on the set-up for the hydraulic transient analysis.

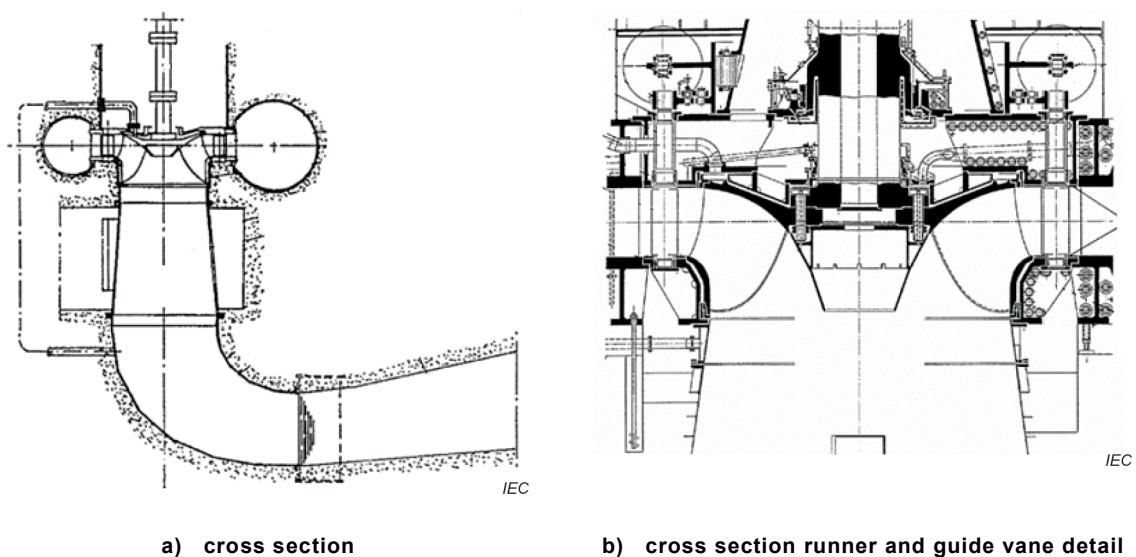
Typical input parameters for Bulb/Pit and S-type turbines are as follows:

- turbine characteristics (guide vane openings, blade openings, speed factors, discharge factors and torque factors);
- reference diameter;
- rotational speed;
- kinematics between servomotor stroke and guide vane opening angle;
- kinematics between servomotor stroke and runner blade opening angle;
- optimised relationship between guide vanes, blades and head.

4.1.4.8 Pump-turbine type turbine

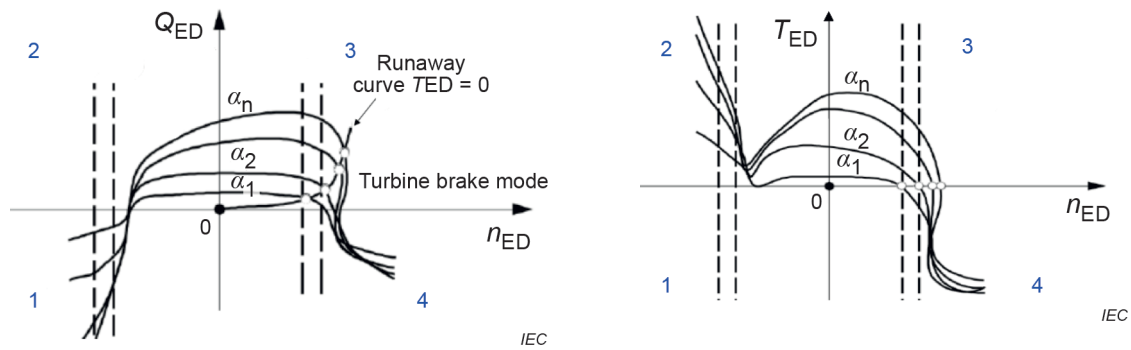
A reversible Francis pump-turbine is designed as a centrifugal pump with adaptation for turbine mode operation, see Figure 4-21.

The nomenclature follows the one for Francis-type turbines as shown in Figure 4-12 above.



SOURCE: IEC TR 61364:2019, Figure 15 and Figure 17.

Figure 4-21 – Schematic view of (radial) pump-turbine

**Key**

- 1 Pump
- 2 Pump break
- 3 Turbine
- 4 Reverse pump

SOURCE: IEC 60193:2019, Figure 115.

Figure 4-22 – Pump-turbine typical 4-quadrant characteristics for a single-regulated (radial) pump-turbine

Typical input parameters for Pump-turbines are as follows:

- pump-turbine characteristics see Figure 4-22 above (guide vane openings, speed factors, discharge factors and torque factors) measured at high sigma value, see influence of cavitation in Clause B.2;
- reference diameter;
- rotational speed;
- kinematics between servomotor stroke and guide vane opening.

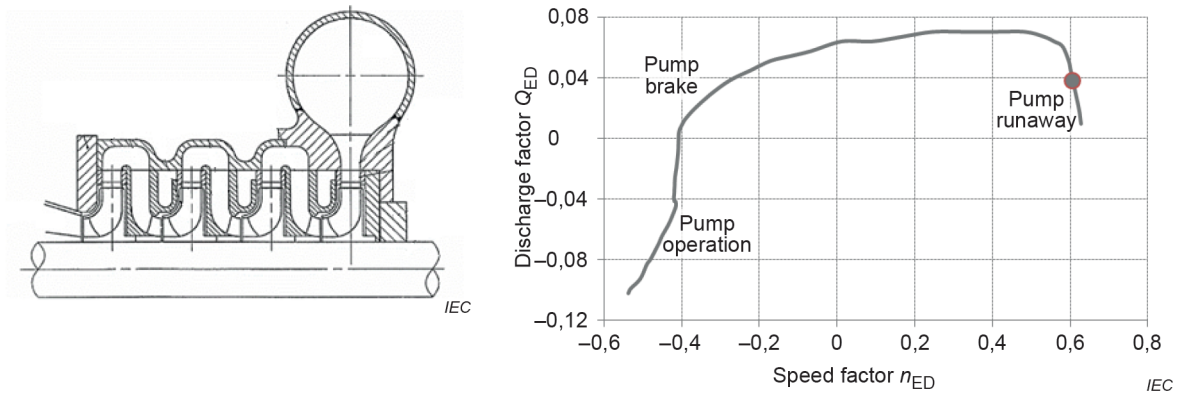
4.1.4.9 Storage pump

Typically, a storage pump has one or multiple radial runners, Figure 4-23 (left). Unlike a turbine, it draws power from the grid instead of providing. If it has no adjustable guide vane, its operating point with regard to discharge and input power can be determined solely due to the applicable pump head and rotational speed.

In order to increase the head range of a storage pump, it is possible to connect several runners in series (stages) with the total available pumping head then being the sum of all pump stages.

Runners can also be mounted in parallel, then the total amount of water being pumped is the equivalent to the sum of all parallel runners.

A typical pump 4 quadrant characteristics is shown in Figure 4-23 (right).



SOURCE: The left figure is taken from IEC TR 61364:2019, Figure 21.

Figure 4-23 – Multi-stage pump (left), and 4-quadrant pump characteristics (right)

Typical input parameters for storage pumps are:

- pump characteristics (guide vane opening (if applicable), speed factors, discharge factors and torque factors);
- number of stages/parallel runners;
- reference diameter;
- rotational speed;
- kinematics between servomotor stroke and guide vane opening (if applicable).

4.1.4.10 Ternary unit

A ternary unit is a complex arrangement of several machines on one shaft line, Figure 4-24. It consists of a motor-generator, a turbine runner, a clutch usually combined with a hydraulic torque converter and a storage pump.

Unlike reversible units a ternary unit has only one direction of rotation corresponding both to pumping for the pump runner and generating for the turbine runner sharing the same shaft.

Generating mode: the clutch is disengaged, and the pump runner is at standstill. The power can be regulated by acting on the turbine flow control device.

Pumping mode: the clutch is engaged, meaning the pump runner rotates. Normally, the turbine runner spins in air to reduce energy losses.

A part of the pumped discharge can be diverted into the turbine to regulate unit power by acting on the turbine flow control device. As a part of the flow runs in closed loop, this mode of operation is also called “hydraulic short circuit”.

The purpose of the torque converter is to start the pump without disconnecting the motor-generator from the grid. During the pump startup, the PDV is closed so that there is no interaction with the waterway system.

Typical parameters can be found in the appropriate machine type sections.

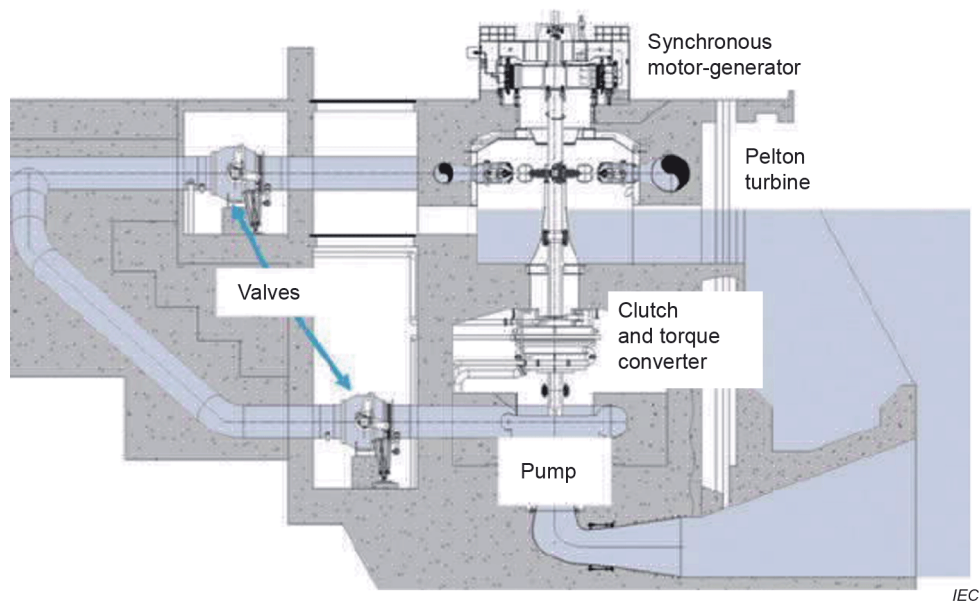


Figure 4-24 – Schematic view of ternary set with Pelton type turbine and storage pump

4.1.4.11 Servomotor

The servomotor acts on the distributor of the hydraulic machine and thereby controls the opening and closing of the unit.

The relationship between servomotor stroke and distributor opening angle can be linear or non-linear and is sometimes referred to as the 'distributor kinematics'. Examples of possible relationships are shown in Figure 4-25 below and this is an input parameter to the hydraulic transient calculation.

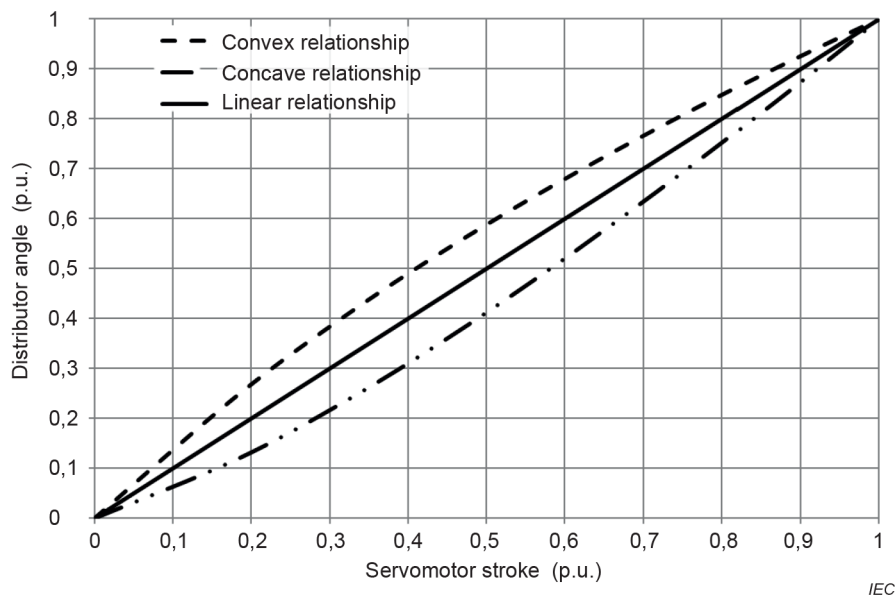


Figure 4-25 – Relationship between servomotor stroke and distributor opening angle

Advice on the treatment of the relationship between servomotor stroke and opening angle for the hydraulic transient analysis is provided as part of the modelling philosophy provided in Table 5-7.