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Hydraulic machines — Guide for dealing with hydro-abrasive erosion in Kaplan, Francis, and Pelton turbines

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The UK participation in its preparation was entrusted to Technical Committee MCE/15, Hydraulic turbines.

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Guide relatif au traitement de l'érosion
hydro-abrasive des turbines Kaplan,
Francis et Pelton
(CEI 62364:2013)

Wasserturbinen -
Leitfaden für den Umgang mit
hydroabrasiver Erosion in Kaplan-,
Francis- und Pelton-Turbinen
(IEC 62364:2013)

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Foreword

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CONTENTS

INTRODUCTION.....	7
1 Scope.....	8
2 Terms, definitions and symbols	8
2.1 Units	8
2.2 Terms, definitions and symbols	9
3 Abrasion rate.....	11
3.1 Theoretical model.....	11
3.2 Introduction to the <i>PL</i> variable	13
3.3 Survey results	15
3.4 Reference model	16
4 Design.....	17
4.1 General.....	17
4.2 Water conveyance system.....	17
4.3 Valve.....	18
4.3.1 General	18
4.3.2 Selection of abrasion resistant materials and coating	18
4.3.3 Stainless steel overlays	19
4.3.4 Protection (closing) of the gap between housing and trunnion	19
4.3.5 Stops located outside the valve	19
4.3.6 Proper capacity of inlet valve operator.....	19
4.3.7 Increase bypass size to allow higher guide vane leakage	19
4.3.8 Bypass system design	20
4.4 Turbine.....	20
4.4.1 General	20
4.4.2 Hydraulic design.....	20
4.4.3 Mechanical design.....	22
4.4.4 Operation	28
4.4.5 Spares and regular inspections	29
4.4.6 Particle sampling and monitoring	29
5 Abrasion resistant materials	30
5.1 Guidelines concerning relative abrasion resistance of materials including abrasion resistant coatings.....	30
5.1.1 General	30
5.1.2 Discussion and conclusions.....	31
5.2 Guidelines concerning maintainability of abrasion resistant coating materials.....	32
5.2.1 Definition of terms used in this subclause	32
5.2.2 Time between overhaul for protective coatings	32
5.2.3 Maintenance of protective coatings.....	33
6 Guidelines on insertions into specifications	34
6.1 General.....	34
6.2 Properties of particles going through the turbine.....	35
6.3 Size distribution of particles	35
6.4 Mineral composition of particles for each of the above mentioned periods	36
Annex A (informative) <i>PL</i> calculation example.....	37
Annex B (informative) Measuring and recording abrasion damages	39

Annex C (informative) Water sampling procedure	52
Annex D (informative) Procedures for analysis of particle concentration, size, hardness and shape	53
Annex E (informative) Tests of abrasion resistant materials	56
Annex F (informative) Typical criteria to determine overhaul time due to abrasion erosion	67
Annex G (informative) Example to calculate the amount of erosion in the full model	68
Annex H (informative) Examples to calculate the TBO in the reference model.....	70
Bibliography.....	73
 Figure 1 – Estimation of the characteristic velocities in guide vanes, W_{gv} , and runner, W_{run} , as a function of turbine specific speed	13
Figure 2 – Example of flow pattern in a Pelton injector at different load	14
Figure 3 – Example of protection of transition area	19
Figure 4 – Runner blade overhang in refurbishment project	21
Figure 5 – Example of “mouse-ear” cavitation on runner band	22
Figure 6 – Detailed design of guide vane trunnion seals	23
Figure 7 – Example of fixing of facing plates from the dry side	25
Figure 8 – Head cover balancing pipes with bends.....	26
Figure 9 – Step labyrinth with optimized shape for hard coating	28
Figure 10 – Development of spiral pressure over time.....	33
Figure D.1 – Typical examples of particle geometry	55
Figure E.1 – Schematic of test rig used for test 1	56
Figure E.2 – ASTM test apparatus	58
Figure E.3 – Test coupon.....	59
Figure E.4 – Slurry pot test facility	60
Figure E.5 – High velocity test rig	61
Figure E.6 – Samples are located on the rotating disk	62
Figure E.7 – Comparison of two samples after testing.....	62
Figure E.8 – Whole test system of rotating disk	62
Figure E.9 – Schematic of test rig used for test 8.....	64
Figure E.10 – Testing of samples on hydro abrasive stand	65
Figure E.11 – Cover of disc	65
Figure E.12 – Curve of unit abrasion rate with circumference velocity for 3 kinds of materials.....	66
 Table 1 – Data analysis of the supplied questionnaire.....	16
Table 2 – Overview over the feasibility for repair C	33
Table 3 – Form for properties of particles going through the turbine.....	35
Table 4 – Form for size distribution of particles.....	36
Table 5 – Form for mineral composition of particles for each of the above mentioned periods	36
Table A.1 – Example of documenting sample tests	37
Table A.2 – Example of documenting sample results	38
Table B.1 – Inspection record, runner blade inlet form	44

Table B.2 – Inspection record, runner blade outlet form	45
Table B.3 – Inspection record, runner band form.....	46
Table B.4 – Inspection record, guide vanes form.....	47
Table B.5 – Inspection record, facing plates and covers form.....	48
Table B.6 – Inspection record, upper stationary seal form.....	49
Table B.7 – Inspection record, upper rotating seal form	49
Table B.8 – Inspection record, lower stationary seal form	50
Table B.9 – Inspection record, lower rotating seal form.....	51
Table E.1 – Relative wear resistance in laboratory test 1	57
Table E.2 – Relative wear resistance in laboratory test 2	57
Table E.3 – Relative wear resistance in laboratory test 3	58
Table E.4 – Relative wear resistance in test 4.....	59
Table E.5 – Results of test.....	60
Table E.6 – Results of test.....	61
Table E.7 – Results from test.....	63
Table E.8 – Relative wear resistance in laboratory test 8	64
Table E.9 – Results of relative wear resistance for some materials ($U = 40\text{m/s}$).....	66
Table G.1 – Calculations.....	69
Table H.1 – Pelton turbine calculation example.....	70
Table H.2 – Francis turbine calculation example	71

INTRODUCTION

Many owners of hydroelectric plants contend with the sometimes very aggressive deterioration of their machines due to particle abrasion. Such owners must find the means to communicate to potential suppliers of machines for their sites, their desire to have the particular attention of the designers at the turbine design phase, directed to the minimization of the severity and effects of particle abrasion.

Limited consensus and very little quantitative data exists on the steps which the designer could and should take to extend the useful life before major overhaul of the turbine components when they are operated under severe particle abrasion service. This has led some owners to write into their specifications, conditions which cannot be met with known methods and materials.

HYDRAULIC MACHINES – GUIDE FOR DEALING WITH HYDRO-ABRASIVE EROSION IN KAPLAN, FRANCIS, AND PELTON TURBINES

1 Scope

This Guide serves to:

- a) present data on particle abrasion rates on several combinations of water quality, operating conditions, component materials, and component properties collected from a variety of hydro sites;
- b) develop guidelines for the methods of minimizing particle abrasion by modifications to hydraulic design for clean water. These guidelines do not include details such as hydraulic profile shapes which should be determined by the hydraulic design experts for a given site;
- c) develop guidelines based on “experience data” concerning the relative resistance of materials faced with particle abrasion problems;
- d) develop guidelines concerning the maintainability of abrasion resistant materials and hard facing coatings;
- e) develop guidelines on a recommended approach, which owners could and should take to ensure that specifications communicate the need for particular attention to this aspect of hydraulic design at their sites without establishing criteria which cannot be satisfied because the means are beyond the control of the manufacturers;
- f) develop guidelines concerning operation mode of the hydro turbines in water with particle materials to increase the operation life;

It is assumed in this Guide that the water is not chemically aggressive. Since chemical aggressiveness is dependent upon so many possible chemical compositions, and the materials of the machine, it is beyond the scope of this Guide to address these issues.

It is assumed in this Guide that cavitation is not present in the turbine. Cavitation and abrasion may reinforce each other so that the resulting erosion is larger than the sum of cavitation erosion plus abrasion erosion. The quantitative relationship of the resulting abrasion is not known and it is beyond the scope of this guide to assess it, except to recommend that special efforts be made in the turbine design phase to minimize cavitation.

Large solids (e.g. stones, wood, ice, metal objects, etc.) traveling with the water may impact turbine components and produce damage. This damage may in turn increase the flow turbulence thereby accelerating wear by both cavitation and abrasion. Abrasion resistant coatings can also be damaged locally by impact of large solids. It is beyond the scope of this Guide to address these issues.

This guide focuses mainly on hydroelectric powerplant equipment. Certain portions may also be applicable to other hydraulic machines.

2 Terms, definitions and symbols

2.1 Units

The International System of Units (S.I.) is adopted throughout this guide but other systems are allowed.

2.2 Terms, definitions and symbols

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

NOTE They are also based, where relevant, on IEC/TR 61364.

Sub-clause	Term	Definition	Symbol	Unit
2.2.1	specific hydraulic energy of a machine	specific energy of water available between the high and low pressure reference sections 1 and 2 of the machine Note 1 to entry: For full information, see IEC 60193.	E	J/kg
2.2.2	acceleration due to gravity	local value of gravitational acceleration at the place of testing Note 1 to entry: For full information, see IEC 60193.	g	m/s ²
2.2.3	turbine head pump head	available head at hydraulic machine terminal $H = E/g$	H	m
2.2.4	reference diameter	reference diameter of the hydraulic machine Note 1 to entry: For Pelton turbines this is the pitch diameter, for Kaplan turbines this is the runner chamber diameter and for Francis and Francis type pump turbines this is the blade low pressure section diameter at the band Note 2 to entry: See IEC 60193 for further information.	D	m
2.2.5	abrasion depth	depth of metal layer that has been removed from a component due to particle abrasion	S	mm
2.2.6	characteristic velocity	characteristic velocity defined for each machine component and used to quantify particle abrasion damage Note 1 to entry: See also 2.2.20 to 2.2.24.	W	m/s
2.2.7	particle concentration	the mass of all solid particles per m ³ of water solution Note 1 to entry: In case the particle concentration is expressed in ppm it is recommended to use the mass of particles per mass of water, so that 1 000 ppm approximately corresponds to 1 kg/m ³ .	C	kg/m ³
2.2.8	particle load	the particle concentration integrated over the time, T , that is under consideration $PL = \int_0^T C(t) \times K_{\text{size}}(t) \times K_{\text{shape}}(t) \times K_{\text{hardness}}(t) dt$ $\left(\approx \sum_{n=1}^N C_n \times K_{\text{size},n} \times K_{\text{shape},n} \times K_{\text{hardness},n} \times T_{s,n} \right)$ $C(t) = 0$ if no water is flowing through the turbine. If the unit is at standstill with pressurized spiral case then $C(t)=0$ when calculating PL for runner and labyrinth seals, but $C(t) \neq 0$ when calculating PL for guide vanes and facing plates.	PL	kg × h/m ³
2.2.9	size factor	factor that characterizes how the abrasion relates to the size of the abrasive particles	K_{size}	
2.2.10	shape factor	factor that characterizes how the abrasion relates to the shape of the abrasive particles	K_{shape}	
2.2.11	hardness factor	factor that characterizes how the abrasion relates to the hardness of the abrasive particles	K_{hardness}	

Sub-clause	Term	Definition	Symbol	Unit
2.2.12	material factor	factor that characterizes how the abrasion relates to the material properties of the base material	K_m	
2.2.13	flow coefficient	coefficient that characterizes how the abrasion relates to the water flow around each component	K_f	$\frac{\text{mm} \times \text{s}^{3,4}}{\text{kg} \times \text{h} \times \text{m}^a}$
2.2.14	sampling interval	the time interval between two water samples taken to determine the concentration of abrasive particles in the water	T_s	h
2.2.15	yearly particle load	the total PL for 1 year of operation, i.e. PL for $T = 8\,760$ h calculated in accordance with 2.2.8	PL_{year}	$\text{kg} \times \text{h}/\text{m}^3$
2.2.16	maximum concentration	the maximum concentration of abrasive particles over a specified time interval	C_{max}	kg/m^3
2.2.17	particle median diameter	the median diameter of abrasive particles in a sample, i.e. such diameter that the particles with size smaller than the value under consideration represent 50 % of the total mass of particles in the sample	dP_{50}	mm
2.2.18	wear resistance index	abrasion depth or volume of a reference material (generally some version stainless steel) divided by the abrasion depth or volume of the material in question, tested under the same conditions	WRI	-
2.2.19	impingement angle	the angle between the particle trajectory and the surface of the substrate		°
2.2.20	characteristic velocity in Francis guide vanes characteristic velocity in Kaplan guide vanes	flow through unit divided by the minimum flow area at the guide vane apparatus estimated at best efficiency point $W_{\text{gv}} = \frac{Q}{a \times Z_0 \times B_0}$	W_{gv}	m/s
2.2.21	characteristic velocity in guide vanes of Kaplan, Francis or tubular turbines	speed of the water flow at guide vane location $W_{\text{gv}} = 0,5 \times \sqrt{2 \times E}$	W_{gv}	m/s
2.2.22	characteristic velocity in Pelton injector	speed of the water flow at injector location $W_{\text{inj}} = \sqrt{2 \times E}$	W_{inj}	m/s
2.2.23	characteristic velocity in Kaplan or Francis tubular turbine runner	the relative velocity between the water and the runner blade estimated with below formulas at best efficiency point $W_{\text{run}} = \sqrt{u_2^2 + c_2^2}$ $u_2 = n \times \pi \times D$ $c_2 = \frac{Q \times 4}{\pi \times D^2}$ Note 1 to entry: In calculation of c_2 for Kaplan turbines, the hub diameter has been neglected in the interest of simplicity.	W_{run}	m/s
2.2.24	characteristic velocity in Pelton runner	speed of the water flow at a Pelton runner $W_{\text{run}} = 0,5 \times \sqrt{2 \times E}$	W_{run}	m/s
2.2.25	discharge (volume flow rate)	volume of water per unit time passing through any section in the system	Q	m^3/s
2.2.26	guide vane opening	average shortest distance between adjacent guide vanes (at a specified section if necessary)	A	m

Sub-clause	Term	Definition	Symbol	Unit
		Note 1 to entry: For further information, see IEC 60193.		
2.2.27	number of guide vanes	total number of guide vanes in a turbine	z_0	
2.2.28	distributor height	height of the distributor in a turbine	B_0	m
2.2.29	rotational speed	number of revolutions per unit time	n	1/s
2.2.30	specific speed	commonly used specific speed to of an hydraulic machine $n_s = \frac{60 \times n \times \sqrt{P}}{H^{5/4}}$ P and H are taken in the rated operating point and given in kW and m respectively	n_s	
2.2.31	output	output of the turbine in the rated operating point	P	kW
2.2.32	actual abrasion depth of target unit	the estimated depth of metal that will be removed from a component of the target turbine due to particle abrasion Note 1 to entry: For use with the Reference model.	$S_{\text{target, actual}}$	mm
2.2.33	actual abrasion depth of reference unit	the actual depth of metal that has been removed from a component of the reference turbine due to particle abrasion Note 1 to entry: For use with the Reference model.	$S_{\text{ref, actual}}$	mm
2.2.34	number of nozzles	number of nozzles in a Pelton turbine	z_0	
2.2.35	bucket width	bucket width in a Pelton runner	B_2	mm
2.2.36	number of buckets	number of buckets in a Pelton runner	z_2	
2.2.37	time between overhaul for target unit	time between overhaul for target unit Note 1 to entry: For use with the reference model.	TBO_{target}	h
2.2.38	time between overhaul for reference unit	time between overhaul for reference unit Note 1 to entry: For use with the reference model.	TBO_{ref}	h
2.2.39	turbine reference size	the reference size for calculation curvature dependent effects of erosion Note 1 to entry: For Francis turbines, it is the reference diameter, D (see 2.2.4). Note 2 to entry: For Pelton turbines it is the inner bucket width, B . Note 3 to entry: For further information in the inner bucket width, B , see IEC 60609-2.	RS	m
2.2.40	size exponent	exponent that describes the size dependant effects of erosion in evaluating RS	p	
2.2.41	exponent	numerical value of $0,4-p$ that balances units for K_f	α	

3 Abrasion rate

3.1 Theoretical model

In order to demonstrate how different critical aspects impact the particle abrasion rate in the turbine, the following formula is considered:

$dS/dt = f(\text{particle velocity, particle concentration, particle physical properties, flow pattern, turbine material properties, other factors})$

However, this formula being of little practical use, several simplifications are introduced. The first simplification is to consider the several variables as independent as follows:

$dS/dt = f(\text{particle velocity}) \times f(\text{particle concentration}) \times f(\text{particle physical properties, turbine material properties}) \times f(\text{flow pattern}) \times f(\text{turbine material properties}) \times f(\text{other factors})$

This simplification is not proven. In fact, many examples can be found where this simplification was not strictly valid. Nevertheless, based on literature studies and experience, this simplification is considered to be justified for hydraulic machines.

The next simplification consists in assigning values to the functions. In the following equations the numerical values for the parameters, without units, have to be used. The units in which the values should be based are given below:

- $f(\text{particle velocity}) = (\text{particle velocity})^n$. In the literature abrasion is often considered proportional to the velocity raised to an exponent, n . Most references give values of n between 2 and 4. In this guide we suggest to use $n = 3,4$. Particle velocity in m/s,
- $f(\text{particle concentration}) = \text{particle concentration in kg/m}^3$,
- $f(\text{particle physical properties, turbine material properties}) = K_{\text{hardness}}$ = function of how hard the particles are in relation to the material at the surface. At the present stage we suggest to use K_{hardness} = fraction of particles harder than the material at the surface,
- $f(\text{flow pattern}) = K_f/RS^p$ (K_f = constant for each turbine component, RS = turbine reference size in m, p = exponent for each turbine component). K_f considers impingement angle and flow turbulence. RS^p considers part curvature radius,
- $f(\text{particle physical properties}) = f(\text{particle size, particle shape, particle hardness}) = f(\text{particle size}) \times f(\text{particle shape}) = K_{\text{size}} \times K_{\text{shape}}$. Note that in this simplification it is assumed that there is no influence from the particle hardness for this function. The particle hardness is considered in the K_{hardness} factor,
- K_{size} = median diameter of particles in mm,
- $K_{\text{shape}} = f(\text{particle angularity})$. It is believed that K_{shape} will increase with the degree of irregularity of the particles. Specific data is not available at present but several literature references indicate that K_{shape} varies from 1 to 2 from round to sharp,
- $f(\text{turbine material properties}) = K_m$. In this guide we consider $K_m = 1$ for martensitic stainless steel with 13 % Cr and 4 % Ni and $K_m = 2$ for carbon steel. For coated components K_m should be smaller than 1,
- $f(\text{other factors}) = 1$.

Again, these functions are engineering approximations in order to obtain useful results for hydraulic machines. We then have the following formula

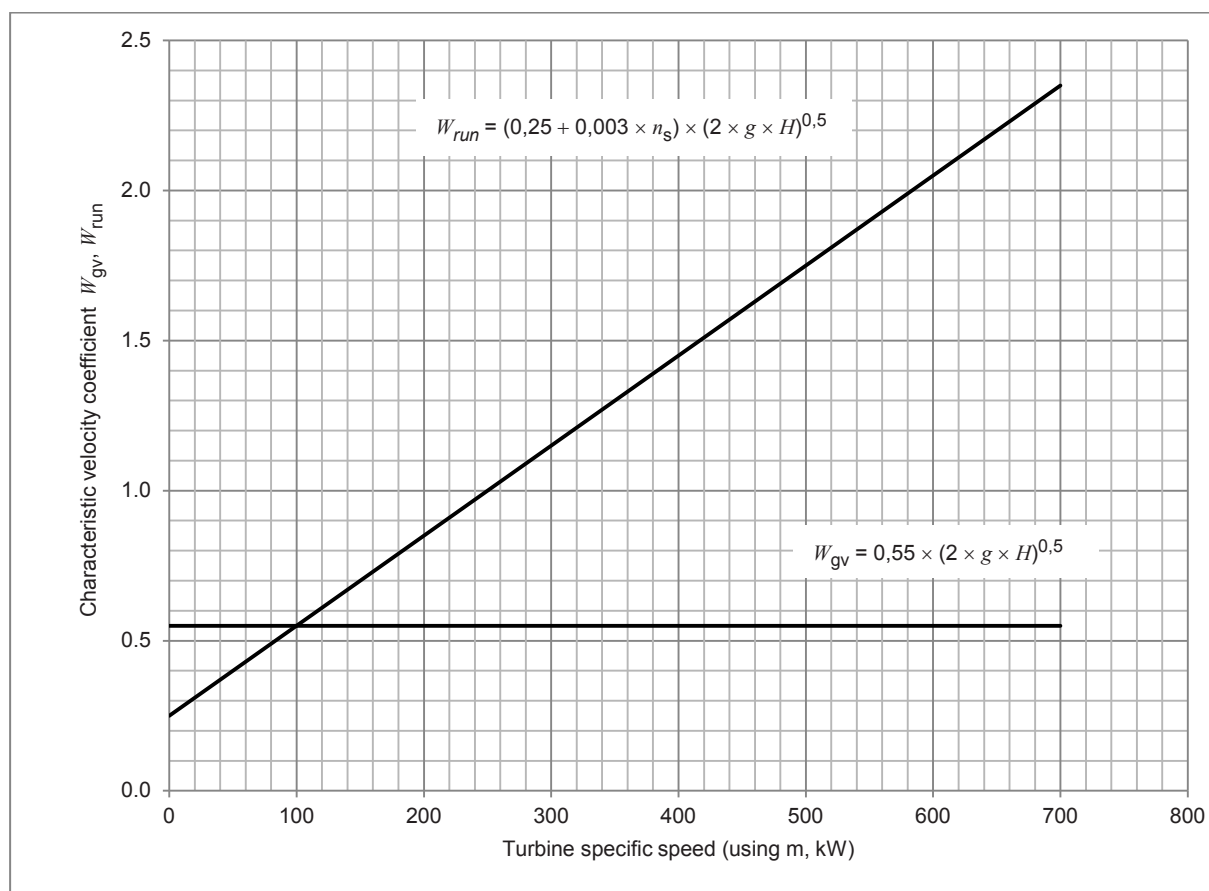
$$dS/dt = (\text{particle velocity})^{3,4} \times C \times K_{\text{hardness}} \times K_{\text{size}} \times K_{\text{shape}} \times K_f/RS^p \times K_m$$

The final step is to integrate this formula with respect to time. When we do this we find three distinct different types of variables with respect to their variations in time:

- 1) particle velocity and K_f : these variables vary with the water flow relative to the individual component, which in turn may vary with the head and flow;
- 2) C , K_{hardness} , K_{size} and K_{shape} : these variables vary with the particle properties. Integrated over time these variables become particle load, PL (see 2.2.8 for definition of PL and Annex A for a sample calculation);
- 3) RS , p and K_m : these variables are constant in time.

To find a simple and reasonably accurate estimate of the time integral, the PL variable (see 2.2.8) is introduced. PL integrates C , K_{hardness} , K_{size} and K_{shape} over time. When using PL , the particle velocity and K_f can be considered approximately constant over a limited variation of head and flow (see 3.2). Since these variables are considered constant, K_f and p were used as calibration factors to obtain good agreement between actual test data and the formula. The particle velocity can be replaced with the characteristic velocity, W , defined in 2.2.20 to 2.2.24.

W may be calculated for a specific turbine based on main data and dimensions. Since the effect of velocity on abrasion is proportional to the velocity raised to a power of 3,4 it is very important to estimate it accurately. For new turbines during design and bid stage, W for different components should be provided by the turbine manufacturer. When this is not possible, W can be estimated approximately from the diagram in Figure 1.



NOTE Values of n_s and H in this figure refer to the rated operating point while the characteristic velocities are given for the points noted in 2.2.

Figure 1 – Estimation of the characteristic velocities in guide vanes, W_{gv} , and runner, W_{run} , as a function of turbine specific speed

So the final, time integrated formula becomes:

$$S = W^{3,4} \times PL \times K_m \times K_f / RSP$$

S is the numerical value of the abrasion depth in mm.

3.2 Introduction to the PL variable

In this code the PL variable has been introduced, which has not been widely used before. One common way to integrate abrasion over time has been to consider the total weight of particles

that pass the turbine. However, this approach has usually not considered the effects from variation in flow or head in the turbine and could therefore lead to erroneous conclusions.

To illustrate this consider the following example. A Pelton injector (see Figure 2) operates for one day. Assume the head is 800 m and the abrasive particle concentration is $0,1 \text{ kg/m}^3$.

Case 1: At full opening (top half of Figure 2) the water with particles flows over the seat ring with a velocity of $(2 \times g \times H)^{0,5} = 125 \text{ m/s}$. In one day the amount of particles that pass the injector is $2 \text{ m}^3/\text{s} \times 3\,600 \text{ s/h} \times 24 \text{ h/day} \times 0,1 \text{ kg/m}^3 \times 1 \text{ day} = 17 \text{ tons}$.

Case 2: At 10 % opening (bottom half of Figure 2) the water with particles flows over the seat ring with the same velocity as in case 1 (125 m/s). In one day the amount of particles that pass the injector is $0,2 \times 3\,600 \times 24 \times 0,1 \times 1 = 1,7 \text{ tons}$.

In both cases the seat ring has been subject to abrasion with the same particle concentration, the same water velocity and the same amount of time. Therefore, the expected abrasion damage is the same. The PL variable also gives the same value in both cases. However, the total weight of particles that has passed the unit is 10 times higher in case 1 compared to case 2. So, PL is expected to correlate better with abrasion damage than the total weight of particles that has passed the seat ring.

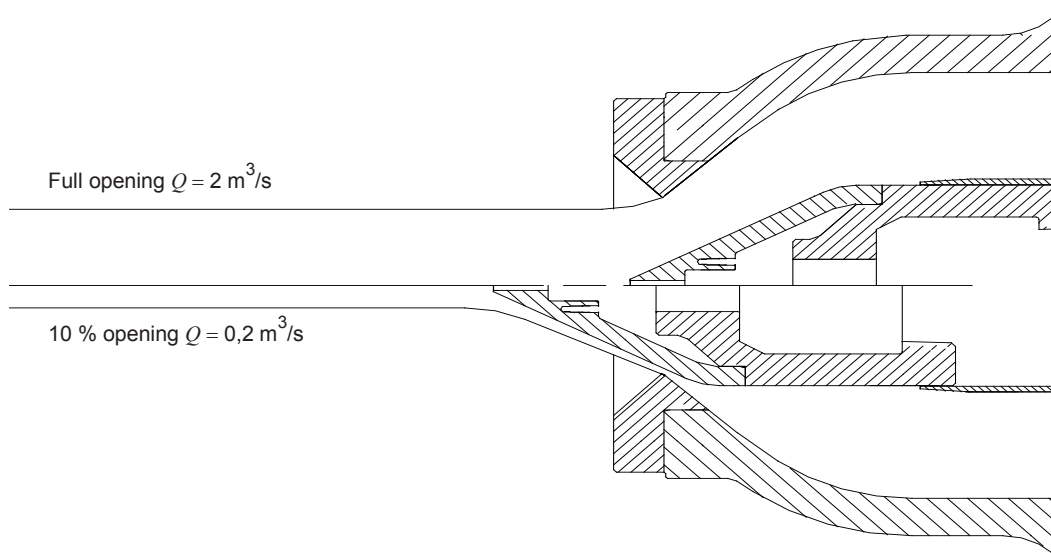


Figure 2 – Example of flow pattern in a Pelton injector at different load

The same type of reasoning can also be applied to other components subject to abrasion. In the following is a condensed summary of such analysis.

- Pelton needle tip

Very good correlation between PL and abrasion damage with minor influence of turbine discharge or head is expected. Some influence from the turbine flow since the water velocity is lower further inside the injector, where the needle is located at high flows. Some influence from turbine head since the water velocity is proportional to the square root of the head. With head and flow variations that are normal in Pelton projects this influence is disregarded in the interest of simplicity.

- Pelton runner

Good correlation between PL and abrasion damage with minor influence of turbine discharge or head is expected. Some influence from the turbine flow since the water film is thicker at higher flows and therefore more particles may be pressed towards the outside surface due to centrifugal forces. Some influence from turbine head since the relative water velocity in the

runner depends on the head. With head and flow variations that are normal in Pelton projects, this influence is disregarded in the interest of simplicity.

- Francis and Kaplan guide vanes and covers / facing plates

Good correlation between PL and abrasion damage with minor influence of turbine discharge or head is expected. Some influence from the turbine flow since the water velocity is higher at low discharge and the pressure difference between the two sides of the guide vane varies with flow. In particular, if the unit is at standstill with pressurized spiral case the leakage flow through the guide vanes has high velocity. Some influence from the turbine head since the relative water velocity in the guide vanes depends on the head. With head and flow variations that are normal in Francis and Kaplan projects, this influence is disregarded in the interest of simplicity.

- Francis runner seals / labyrinths

Very good correlation between PL and abrasion damage with minor influence of turbine discharge or head is expected. Some influence is expected from the turbine flow and head since they influence the pressure before and after the seal and thus the leakage flow through the seal. With head and flow variations that are normal in Francis projects, this influence is disregarded in the interest of simplicity.

- Francis runner blade inlet

Good correlation between PL and abrasion damage with minor influence of turbine discharge or head is expected. Some influence from the turbine discharge is expected since the water velocity is higher at low discharge. Moreover, the pressure difference between the two sides of the guide vanes varies with opening, resulting in more leakage between the guide vanes and the covers which in turn results in more unfavourable flow conditions at the runner inlet. Also discharge and head variations from the optimum operating point, will result in more unfavourable flow conditions at the runner inlet. With head and flow variations that are normal in Francis projects this influence is disregarded, as long as inlet cavitation is not present, in the interest of simplicity.

- Francis runner blade outlet

Reasonable correlation between PL and abrasion damage with minor influence of turbine discharge or head is expected. At part load there are two main phenomena that influence the wear. One is that the average velocity (defined as the total flow divided by the flow passage area) will decrease with decreasing discharge. The other is that the degree of turbulence will increase and the flow distribution will lose uniformity at low discharge (typically below 50 % to 80 % of maximum discharge). These two phenomena influence the wear in opposite ways, but it is expected that the turbulence effect will dominate and thus that the wear will increase at partial load. However, due to lack of supporting data this influence is disregarded in the interest of simplicity.

- Kaplan runner blade

Very good correlation between PL and abrasion damage with minor influence of turbine discharge or head is expected. With head and flow variations that are normal in Kaplan projects this influence is disregarded in the interest of simplicity.

- Kaplan runner chamber

Good correlation between PL and abrasion damage with minor influence of turbine discharge or head is expected. With head and flow variations that are normal in Kaplan projects this influence is disregarded in the interest of simplicity.

3.3 Survey results

A questionnaire was sent to plant operators at sites known for their exposure to particle abrasion problems. The purpose of this questionnaire was to collect and analyse data on

particle abrasion rates on as many combinations of water quality, operating conditions, component materials, and component properties as possible.

This data was analyzed and the factor K_f and the exponent p determined for each component to get the best possible correspondence between the calculated and observed amount of erosion. The average K_f was then determined for all observations with components of the same type. Table 1 below shows the resulting K_f and p for various components as well as the number of observations. The ratio between the measured and calculated values of the abrasion depth was determined and the standard deviation calculated.

Table 1 – Data analysis of the supplied questionnaire

Component	K_f	Exponent p (for RS)	Number of observations	Standard deviation %
Francis guide vanes	$1,06 \times 10^{-6}$	0,25	7	42
Francis facing plates	$0,86 \times 10^{-6}$	0,25	7	38
Francis labyrinth seals	$0,38 \times 10^{-6}$	0,75	7	30
Francis runner inlet	$0,90 \times 10^{-6}$	0,25	6	26
Francis runner outlet	$0,54 \times 10^{-6}$	0,75	6	41

Although the values of standard deviation in the table above shows that the formula gives reasonable accuracy, it should be kept in mind that the amount of observations is limited and that further observations may improve the formula.

It is only for Francis turbines that enough data has been available for a meaningful analysis. Not enough data is available for Kaplan and Pelton turbines to give detailed guidelines.

In general, it is challenging to obtain complete and unambiguous observations from existing measurements. It is hoped that additional observations can be made in the future to further calibrate and revise the erosion model.

3.4 Reference model

In the Reference model presented in this guide the *TBO* of two turbines are compared to each other. To do this the *TBO* of one turbine (here called reference turbine) and the differences in the influencing parameters to another turbine (here called target turbine) have to be known to calculate the *TBO* of the target turbine. Note that the same overhaul criteria have to be applied for both the target and reference turbines.

The aim of the Reference model is not to calculate the erosion depth (S). Therefore a calibrated model for the depth is not necessary. The criteria for the *TBO* can be the relative amount of damage, the efficiency loss or some other criteria but has to be the same for both turbines.

There are a few differences in the way the formula is built up between the Reference model and the absolute model as follows:

- since the Reference model does not calculate the erosion depth of individual components, constants valid for the whole turbine are used instead of different constants for different components;
- a larger turbine can normally withstand more abrasion depth than a small turbine before it needs overhaul. For this reason the exponent for turbine reference size, p , is chosen as 1 in the Reference model;
- for Pelton turbines it is assumed that the critical component for overhaul is the runner. In addition to the factors described above, the K_f for Pelton runners is assumed to be

proportional to the number of nozzles and the speed and inversely proportional to the number of buckets;

- for Pelton turbines the reference size is taken as the bucket width, B_2 , instead of the runner diameter.

The *TBO* for the target turbine can be calculated as follows:

$$\frac{TBO_{\text{target}}}{TBO_{\text{ref}}} = \frac{W_{\text{ref}}^{3,4}}{W_{\text{target}}^{3,4}} \times \frac{PL_{\text{ref}}}{PL_{\text{target}}} \times \frac{K_{m,\text{ref}}}{K_{m,\text{target}}} \times \frac{K_{f,\text{ref}}}{K_{f,\text{target}}} \times \frac{RS_{\text{target}}^p}{RS_{\text{ref}}^p}$$

In this equation we use the following values for the relationships:

$$\text{Pelton turbines: } K_{f,\text{ref}} / K_{f,\text{target}} = z_{0,\text{ref}} \times n_{\text{ref}} \times z_{2,\text{target}} / (z_{0,\text{target}} \times n_{\text{target}} \times z_{2,\text{ref}})$$

$$\text{Francis and Kaplan turbines: } K_{f,\text{ref}} / K_{f,\text{target}} = 1$$

$$\text{Size exponent: } p = 1$$

The accuracy of the Reference model might decrease when the differences between the reference and target turbines become large.

The sensitivity of the calculated *TBO* value to variances in the input variables can also be studied with the same formula.

4 Design

4.1 General

The following guidelines explain some recommended methods to minimize particle abrasion and the effects thereof, by modifying the design for clean water.

It should be understood that every hydraulic powerplant is a compromise between several requirements. While it is possible to design a unit to be more resistant against particle abrasion this may adversely affect other aspects of the turbine. Some examples are:

- thicker runner blades may result in decreased efficiency and increased risk of vibrations from von Karman vortices,
- fewer runner blades (in order to improve the access to the blade surfaces for thermal spray surface treatment) may result in reduced cavitation performance,
- abrasion resistant coatings may initially result in increased surface roughness, which may reduce the efficiency,
- reduced runner blade overhang may result in reduced cavitation performance, which in turn may reduce the output that can be achieved for a turbine upgrade,
- many abrasion resistance design features will increase the total cost of the powerplant.

The optimum combination of abrasion resistant design features should be considered and selected for each site based on its specific conditions.

4.2 Water conveyance system

An important consideration for the water conveyance system is to remove as many particles as possible already before they enter the high velocity zones in the machinery. Large reservoirs may be very useful for this purpose. If a large reservoir is not available, so-called desilting chambers may be built. It appears that the minimum particle size that can be removed by desilting chambers is to the order of 0,1 mm to 0,3 mm unless the cost and size of the structures becomes prohibitive. The detailed design of desilting chambers is outside the scope of this Guide.

It is also important that any transient conditions that the powerplant may experience do not disturb the sand in sand traps, or other places where sand may accumulate, so that it is drawn into the turbine. Therefore, the design of sand traps should also consider possible transient conditions.

Even small amounts of large particles, such as stones, can cause severe damage since they may not be able to pass the turbine until they have been crushed into smaller pieces. This is due to the centrifugal force in the rotating water between runner and guide vanes. It is therefore important that tunnels and penstocks are clean and tidy at initial startup and after maintenance work.

Due to generally low velocity the water conveyance system itself seldom sees significant abrasion damage and normal coating paint is usually enough to protect it.

4.3 Valve

4.3.1 General

If solid particle abrasion is expected, as a general rule all mechanical disturbances in the flow are subject to high abrasive attack. Therefore a spherical valve should be preferred instead of a butterfly valve where the sealing disc is continuously exposed to the abrasive water flow.

As a general rule for the design, the area exposed to the abrasive water should be as small as possible. Discontinuities and sharp transitions or direction change of the flow should also be avoided.

The shape of the housing around the sealing of the rotor or disc shall be smooth without sharp edges and big changes in the flow direction.

In case of several units on a single penstock and with a spherical valve or a butterfly biplane valve as inlet valve, it is preferable to have a maintenance seal in addition to the service seal. This will make it possible to do maintenance of the downstream seal while the other units are in service.

A ring gate is a special type of valve and similar considerations apply to the ring gate and the main inlet valve.

4.3.2 Selection of abrasion resistant materials and coating

The selection of materials and possible abrasion resistant coatings for components which are subject to abrasive wear is important. See also Clause 5.

Because corrosion with simultaneously acting abrasion increases the wear rate, stainless steel is the preferred selection. Weldable stainless steel materials are preferred. On the basis of the larger hardness, martensitic steel is favored over austenitic steel. Weldability and erosion resistance are often contradictory and an optimum should be sought in each case.

The possibility to weld in situ is an advantage for future repair.

In case abrasion resistant coatings are not applied, it is recommended to make a mechanical component design where such coatings can easily be applied at a later stage.

The whole sealing area should be made of stainless steel and, if subject to abrasive wear, a coating should be applied.

4.3.3 Stainless steel overlays

Stainless steel welded overlays with sufficient thickness may be used instead of solid stainless components, if the abrasion area is not too large.

4.3.4 Protection (closing) of the gap between housing and trunnion

The area between trunnion and housing is especially susceptible for abrasive wear. Since the transition trunnion to the rotor is one of the highly stressed areas of the inlet valves this area has to be especially protected. Completely stainless or welded stainless overlay protected trunnions are recommended. Through a pre-labyrinth, the transitions can be protected against the direct attack of abrasive particles (see Figure 3).

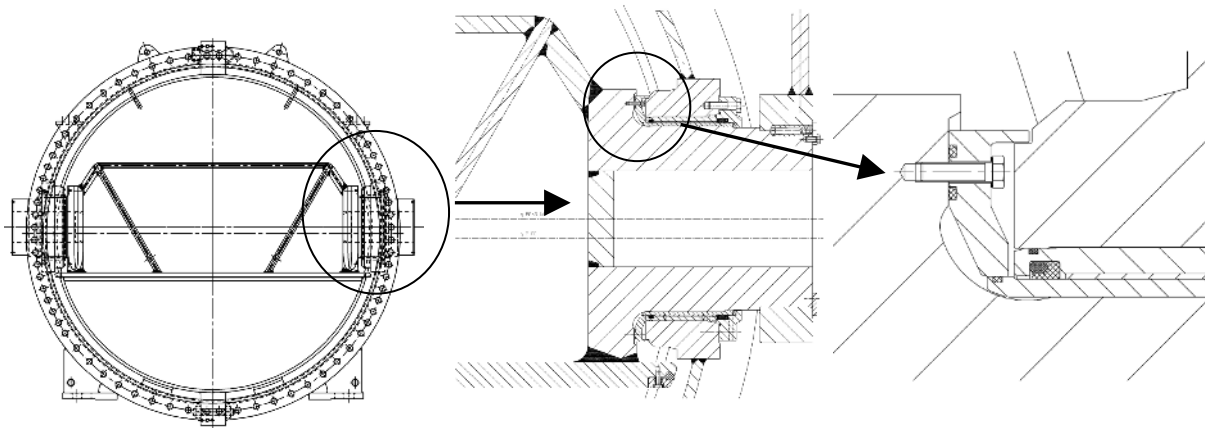


Figure 3 – Example of protection of transition area

4.3.5 Stops located outside the valve

In abrasive conditions, it is recommended that stops, which limit the angle of rotation of the rotor, are placed outside of the flow in the servomotor or adjacent to the lever.

4.3.6 Proper capacity of inlet valve operator

Normally inlet valves will be opened or closed with an approximately balanced pressure that is established by the bypass or the movable sealing rings.

If excessive abrasion occurs at the guide vanes, the differential pressure for the closing or opening of the inlet valve may be bigger than allowed or fixed in the layout. It is therefore recommended that the design takes into account a higher differential pressure for opening or closing. If the inlet valve is designed as an emergency shutoff valve it may already be able to open against a higher differential pressure.

Please see 5.2.2 for an example of how the pressurization of the spiral case can develop with abrasive wear in the guide vanes.

4.3.7 Increase bypass size to allow higher guide vane leakage

As already mentioned in 4.3.6 above, due to the excessive abrasion at the guide vanes the leakage water flow will increase to such an extent, that the balance water flow through the bypass of the inlet valve is not sufficient to achieve the pressure balance between the inlet pipe and spiral casing.

It is therefore recommended to increase the capacity of the bypass system.

4.3.8 Bypass system design

It is recommended to use an external bypass system instead of an internal one.

4.4 Turbine

4.4.1 General

These guidelines do not include details such as hydraulic profile shapes which should be determined by the hydraulic design experts for a given site.

4.4.2 Hydraulic design

4.4.2.1 Selection of type of machine

It is advantageous to select a type of machine that has low water velocity, that can easily be serviced and that can easily be coated with abrasion resistant coatings. Some general guidelines are as follows:

- in the choice between a vertical shaft Kaplan and a Bulb, the Kaplan will normally have lower velocity (see Figure 1). The serviceability and ease of coating is approximately equal between the two;
- in the choice between a Kaplan and a Francis, the Francis will normally have lower velocity. On the other hand, the Kaplan runner has better access for applying abrasion resistant coatings. The serviceability is approximately equal between the two;
- in the choice between a Francis and a Pelton the Francis will normally have lower maximum velocity. However, the parts in a Pelton turbine that are subject to the maximum velocity (i.e. the needle tips and seat rings) are small and have better access for applying abrasion resistant coatings. The Pelton turbine is also easier to service.

4.4.2.2 Specific speed

For the same plant lower specific speed machines are normally bigger and have lower water velocities in the runner outlet. However, the water velocities are not lower in the guide vanes and in the runner inlet. For Kaplan, Bulb and low head Francis turbines, most of the abrasion damage will be in the runner so the specific speed is important. For high head Francis turbines much of the abrasion damage will be in the guide vane apparatus, so the specific speed is not so important.

For Pelton turbines the water velocity does not depend on the specific speed. However, a lower number of jets is beneficial for a Pelton turbine since the buckets will be larger which in turn gives less water acceleration in the buckets and thus less abrasion damage. A lower number of jets will automatically result in a lower specific speed.

4.4.2.3 Variable speed

Even though variable speed machines are not frequent, they are less prone to cavitation, even under a wide head range operation. Due to this characteristic, the variable speed machine may better resist particle abrasion.

4.4.2.4 Turbine submergence

Cavitation and abrasion will mutually reinforce each other. For this reason it is recommended that the turbine submergence is higher for plants where abrasion is expected.

4.4.2.5 Runner blade overhang

During the refurbishment of a Francis turbine wheel it is sometimes necessary to significantly increase the turbine output. One way to do this is to extend the runner band inside the hub.

tube cone, in order to increase the blade area and therefore improve the cavitation performance, see Figure 4. However, this creates additional turbulences at the entrance of the draft tube cone that will increase metal removal if particles are present in the water. A secondary effect of the overhang blades is to create a lower pressure zone downstream of the runner band seal, thus creating higher seal leakage and more particle abrasion at the band seal.

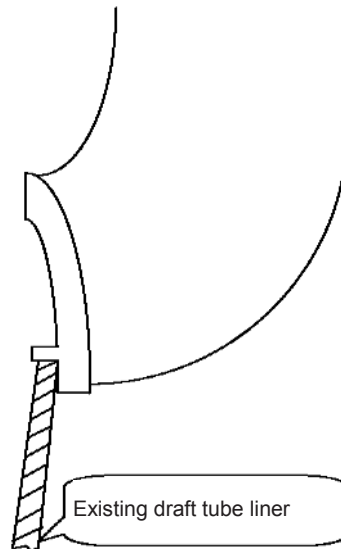


Figure 4 – Runner blade overhang in refurbishment project

4.4.2.6 Thicker runner blades and guide vanes

Increased runner blade thickness, particularly at the outflow edge, gives some extra margin before the removal of material on the runner blades becomes critical for the structural integrity of the runner. Designing a thicker blade should be done with care. A thicker blade may reduce the turbine efficiency and increase the risk for issues with von Karman vortices. Also, the risk of “mouse-ear” cavitation (cavitation damage on the runner band, downstream of the blade, see Figure 5) may increase. In this context it can be mentioned that abrasion resistant coatings may provide a means to design thin profiles with only a marginal increase in the thickness due to the coating.

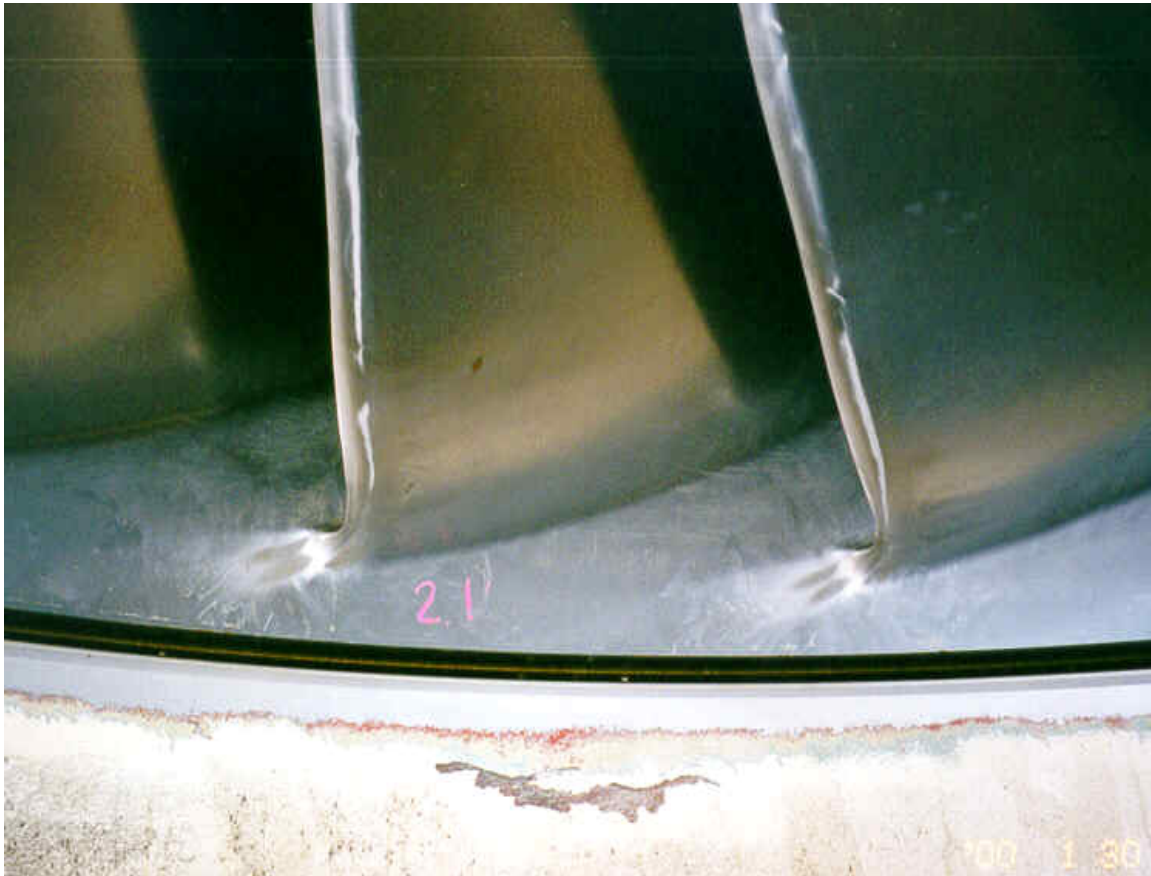


Figure 5 – Example of “mouse-ear” cavitation on runner band

Thicker guide vanes may also give an additional margin on abrasion, especially at the guide vane – trunnion area, although the critical areas in the guide vane apparatus are usually the guide vane end faces and the head covers and bottom rings, or facing plates.

4.4.2.7 Guide vane overhang

When guide vane overhang exists, the area underneath the guide vane will experience high turbulence and high recirculation and particle abrasion may be significant in that region. The high turbulence may also influence the runner inlet at the band. It is recommended to make the overhang as small as possible.

4.4.3 Mechanical design

4.4.3.1 General

If abrasion is expected and the turbine type is defined, not only the hydraulic design but also the mechanical design can take some precautions to reduce the abrasion rate and to allow easy maintenance or replacement of the abraded parts. In this subclause some features are mentioned.

If a special coating is foreseen, the design of the coated parts shall allow the type of application.

As a general rule for the design, the area exposed to the abrasive water should be as small as possible. As well, discontinuities and sharp transitions or direction change of the flow should be avoided.

The following subclauses are divided in direct measures to reduce abrasion and measures to allow an easy maintenance to dismantle the abraded components.