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Hydraulic machines, radial and axial — Performance conversion method from model to prototype

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

A list of organizations represented on this committee can be obtained on request to its secretary.

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English version

Hydraulic machines, radial and axial - Performance conversion method from model to prototype (IEC 62097:2009)

Machines hydrauliques,
radiales et axiales -
Méthode de conversion des performances
du modèle au prototype
(CEI 62097:2009)

Hydraulische Maschinen,
radial und axial -
Leistungsumrechnung
vom Modell zum Prototyp
(IEC 62097:2009)

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Foreword

The text of document 4/242A/FDIS, future edition 1 of IEC 62097, prepared by IEC TC 4, Hydraulic turbines, was submitted to the IEC-CENELEC parallel vote and was approved by CENELEC as EN 62097 on 2009-03-01.

The International Standard contains attached files in the form of Excel file. These files are intended to be used as complement and do not form an integral part of this publication.

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Annex ZA has been added by CENELEC.

Endorsement notice

The text of the International Standard IEC 62097:2009 was approved by CENELEC as a European Standard without any modification.

Annex ZA (normative)

Normative references to international publications with their corresponding European publications

The following referenced documents are indispensable for the application 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.

NOTE When an international publication has been modified by common modifications, indicated by (mod), the relevant EN/HD applies.

<u>Publication</u>	<u>Year</u>	<u>Title</u>	<u>EN/HD</u>	<u>Year</u>
IEC 60193	1999	Hydraulic turbines, storage pumps and pump-turbines - Model acceptance tests	EN 60193	1999

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INTRODUCTION

0.1 General remarks

This International Standard establishes the prototype hydraulic machine efficiency from model test results, with consideration of scale effect including the effect of surface roughness.

Advances in the technology of hydraulic turbo-machines used for hydroelectric power plants indicate the necessity of revising the scale effect formula given in 3.8 of IEC 60193. [1]¹ The advance in knowledge of scale effects originates from work done by research institutes, manufacturers and relevant working groups within the organizations of IEC and IAHR. [1 - 7]

The method of calculating prototype efficiencies, as given in this standard, is supported by experimental work and theoretical research on flow analysis and has been simplified for practical reasons and agreed as a convention. [8 – 10] The method is representing the present state of knowledge of the scale-up of performance from model to a homologous prototype.

Homology is not limited to the geometric similarity of the machine components, it also calls for homologous velocity triangles at the inlet and outlet of the runner/impeller. [2] Therefore, compared to IEC 60193, a higher attention has to be paid to the geometry of guide vanes.

According to the present state of knowledge, it is certain that, in most cases, the formula for the efficiency step-up calculation given in the IEC 60193 and earlier standards, overstated the step-up increment of the efficiency for the prototype. Therefore, in the case where a user wants to restudy a project for which a calculation of efficiency step-up was done based on any previous method, the user shall re-calculate the efficiency step-up with the new method given in this standard, before restudying the project of concern.

This standard is intended to be used mainly for the assessment of the results of contractual model tests of hydraulic machines. If it is used for other purposes such as evaluation of refurbishment of machines having very rough surfaces, special care should be taken as described in Annex B.

Due to the lack of sufficient knowledge about the loss distribution in Deriaz turbines and storage pumps, this standard does not provide the scale effect formula for them.

An excel work sheet concerning the step-up procedures of hydraulic machine performance from model to prototype is indicated at the end of this Standard to facilitate the calculation of the step-up value.

0.2 Basic features

A fundamental difference compared to the IEC 60193 formula is the standardization of scalable losses. In a previous standard (see 3.8 of IEC 60193:1999 [1]), a loss distribution factor V has been defined and standardized, with the disadvantage that turbine designs which are not optimized benefit from their lower technological level.

This is certainly not correct, since a low efficiency design has high non-scalable losses, like incidence losses, whereby the amount of scalable losses is about constant for all manufacturers, for a given type and a given specific speed of a hydraulic machine.

This standard avoids all the inconsistencies connected with IEC 60193:1999. (see 3.8 of [1]) A new basic feature of this standard is the separate consideration of losses in specific hydraulic energy, disc friction losses and leakage losses. [5], [8 – 10]

¹ Numbers in square brackets refer to the bibliography.

Above all, in this standard, the scale-up of the hydraulic performance is not only driven by the dependence of friction losses on Reynolds number Re , but also the effect of surface roughness Ra has been implemented.

Since the roughness of the actual machine component differs from part to part, scale effect is evaluated for each individual part separately and then is finally summed up to obtain the overall step-up for a complete turbine. [10] For radial flow machines, the evaluation of scale effect is conducted on five separate parts; spiral case, stay vanes, guide vanes, runner and draft tube. For axial flow machines, the scalable losses in individual parts are not fully clarified yet and are dealt with in two parts; runner blades and all the other stationary parts inclusive.

The calculation procedures according to this standard are summarized in Clause 7 and Excel sheets are provided as an Attachment to this standard to facilitate the step-up calculation.

In case that the Excel sheets are used for evaluation of the results of a contractual model test, each concerned party shall execute the calculation individually for cross-check using common input data agreed on in advance.

HYDRAULIC MACHINES, RADIAL AND AXIAL ± PERFORMANCE CONVERSION METHOD FROM MODEL TO PROTOTYPE

1 Scope

This International Standard is applicable to the assessment of the efficiency and performance of prototype hydraulic machine from model test results, with consideration of scale effect including the effect of surface roughness.

This standard is intended to be used for the assessment of the results of contractual model tests of hydraulic machines.

2 Normative references

The following referenced documents are indispensable for the application 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 60193:1999, *Hydraulic turbines, storage pumps and pump-turbines ± Model acceptance tests*

3 Terms, definitions, symbols and units

3.1 System of units

The International System of Units (SI) is used throughout this standard. All terms are given in SI Base Units or derived coherent units. Any other system of units may be used after written agreement of the contracting parties.

3.2 List of terms

For the purposes of this document, the terms and definitions of IEC 60193 apply, as well as the following terms, definitions, symbols and units.

3.2.1 Subscripts' list

Term	Symbol	Term	Symbol	in general term represented by CO
model	M	component	CO	
prototype	P			
specific energy	E	spiral case	SP	
volumetric	Q	stay vane	SV	
torque or disc friction	T	guide vane	GV	
reference	ref	runner	RU	
hydraulic diameter	d	draft tube	DT	
velocity	u	stationary part	ST	
hydraulic	h			
optimum point	opt			
off design point	off			

3.2.2 Terms, definitions, symbols and units

Term	Definition	Symbol	Unit
Radial flow machines	Francis turbines and Francis type reversible pump-turbines	-	-
Axial flow machines	Kaplan turbines, bulb turbines and fixed blade propeller turbines	-	-
Reference diameter	Reference diameter of the hydraulic machine (see Figure 3 of IEC 60193)	D	m
Hydraulic diameter	4 times sectional area divided by the circumference of the section	d_h	m
Sand roughness	Equivalent sand roughness [11]	k_s	m
Arithmetical mean roughness	Deviation of the surface profile represented by the arithmetical mean value	Ra	m
Acceleration due to gravity	Local value of gravitational acceleration at the place of testing as a function of altitude and latitude (see IEC 60193)	g	m s^{-2}
Density of water	Mass per unit volume of water (see IEC 60193)	ρ	kg m^{-3}
Dynamic viscosity	A quantity characterizing the mechanical behaviour of a fluid	μ	Pa s
Kinematic viscosity	Ratio of the dynamic viscosity to the density of the fluid. Values are given as a function of temperature. (see IEC 60193)	ν	$\text{m}^2 \text{s}^{-1}$
Discharge	Volume of water per unit time flowing through any section in the system	Q	$\text{m}^3 \text{s}^{-1}$
Mass flow rate	Mass of water flowing through any section of the system per unit time	(ρQ)	kg s^{-1}
Discharge of machine	Discharge flowing through the high pressure reference section	Q_1	$\text{m}^3 \text{s}^{-1}$
Leakage flow rate	Volume of water per unit time flowing through the runner seal clearances	q	$\text{m}^3 \text{s}^{-1}$
Net discharge	Volume of water per unit time flowing through runner/impeller. It corresponds to $Q_1 - q$ in case of turbine and $Q_1 + q$ in case of pump.	Q_m	$\text{m}^3 \text{s}^{-1}$
Mean velocity	Discharge divided by the sectional area of water passage	v	m s^{-1}
Peripheral velocity	Peripheral velocity at the reference diameter	u	m s^{-1}
Rotational speed	Number of revolutions per unit time	n	s^{-1}
Specific hydraulic energy of machine	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 (see IEC 60193)	E	J kg^{-1}
Specific hydraulic energy of runner/impeller	Turbine: Net specific hydraulic energy working on the runner	E_m	J kg^{-1}
	Pump: Specific hydraulic energy produced by the impeller	E_m	J kg^{-1}
Specific hydraulic energy loss in stationary part	Specific hydraulic energy loss in stationary part which includes both friction loss and kinetic loss	E_{Ls}	J kg^{-1}
Specific hydraulic energy loss in runner/impeller	Specific hydraulic energy loss in runner/impeller which includes both friction loss and kinetic loss	E_{Lm}	J kg^{-1}
Friction loss of specific hydraulic energy	Specific hydraulic energy loss caused by the friction on the surface of water passages	E_{Lf}	J kg^{-1}

Term	Definition	Symbol	Unit
Kinetic loss of specific hydraulic energy	Specific hydraulic energy loss caused by the hydraulic phenomena other than surface friction, such as turbulence, separation of flow, abrupt change of water passage, etc.	E_{lk}	$J\ kg^{-1}$
Turbine net head or pump delivery head	$H = E / g$	H	m
Turbine output or pump input	The 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 Figures A.1 and A.2)	P	W
Hydraulic power	The power available for producing power (turbine) or imparted to the water (pump) $P_h = E (\rho Q_1)$	P_h	W
Mechanical power of runner/ impeller	The power transmitted through the coupling between shaft and runner (impeller).	P_m	W
Power of runner/impeller	Turbine: Power produced by the runner corresponding to $E_m (\rho Q_m)$ or $P_m + P_{Ld}$ Pump: Power produced by the impeller represented by $E_m (\rho Q_m)$ or $P_m - P_{Ld}$	P_r P_r	W W
Disc friction loss	Loss power caused by the friction on the outer surface of the runner/impeller	P_{Ld}	W
Bearing loss power	Loss power caused by the friction of the shaft bearing and shaft seal	P_{Lm}	W
Runner/impeller torque	Torque transmitted through the coupling of the runner/impeller and the shaft corresponding to the mechanical power of runner/impeller, P_m .	T_m	N m
Hydraulic efficiency	Turbine: $\eta_h = P_m / P_h$ Pump: $\eta_h = P_h / P_m$	η_h	-
Specific hydraulic energy efficiency	Turbine: $\eta_E = E_m / E_h$ Pump: $\eta_E = E_h / E_m$ (see Figures A.1 and A.2)	η_E	-
Volumetric efficiency	Turbine: $\eta_Q = Q_m / Q_1$ Pump: $\eta_Q = Q_1 / Q_m$ (see Figures A.1 and A.2)	η_Q	-
Power efficiency (disc friction efficiency)	Turbine : $\eta_T = P_m / P_r$ Pump : $\eta_T = P_r / P_m$ (see Figures A.1 and A.2)	η_T	-
Mechanical efficiency	Turbine: $\eta_m = P / P_m$ Pump: $\eta_m = P_m / P$ (see Figures A.1 and A.2)	η_m	-
Efficiency step-up	Difference between efficiencies at two hydraulically similar operating conditions	$\Delta\eta$	-
Efficiency step-up ratio	Ratio of efficiency step-up against model efficiency $\Delta = \frac{\Delta\eta}{\eta_M}$	Δ	
Reynolds number	Reynolds number of the machine $Re = D u / \nu$	Re	-
Reynolds number of component passage	$Re_d = d_h \nu / \nu$	Re_d	-
Friction loss coefficient for pipe flow	Friction loss coefficient for a pipe. $\lambda = \frac{E_{Lf}}{L \frac{\nu^2}{d^2}}$ where d pipe diameter (m) L pipe length (m)	λ	-

Term	Definition	Symbol	Unit
Friction loss coefficient for a flat plate	Friction loss coefficient for a flat plate. $C_f = \frac{E_{Lf}}{BL \frac{w^3}{Q^2}}$ where B width of a flat plate (m) L length of a flat plate (m) Q discharge passing by the plate (m³/s) w relative flow velocity (m/s)	C_f	-
Disc friction loss coefficient	Friction loss coefficient for a rotating disc $C_m = \frac{P_{Ld}}{\frac{\pi^4}{8} \rho n^3 D_d^5}$ where D_d diameter of the rotating disc (m)	C_m	-
Relative scalable hydraulic energy loss	Scalable specific hydraulic energy loss divided by E, which is dependent on Reynolds number and roughness (in most cases, it is represented in %) $\delta_E = E_{if}/E$	δ_E	-
Relative non-scalable hydraulic energy loss	Non-scalable specific hydraulic energy loss divided by E, which remains constant regardless of Reynolds number and roughness $\delta_{ns} = E_{lk}/E$	δ_{ns}	-
Reference scalable hydraulic energy loss	δ_E value for a model with smooth surface operating at a reference Reynolds number $Re = 7 \times 10^6$	δ_{Eref}	-
Reference scalable hydraulic energy loss in component passage	δ_{Eref} for each component passage	δ_{ECOref}	-
Relative disc friction loss	Disc friction loss P_{Ld} divided by P_m $\delta_T = \frac{P_{Ld}}{P_m}$	δ_T	-
Reference disc friction loss	δ_T value for a model with fairly smooth surface operating at a reference Reynolds number $Re = 7 \times 10^6$	δ_{Tref}	-
Flow velocity factor for each component passage	Ratio of the maximum relative flow velocity in each component passage against the peripheral velocity u $\kappa_{uCO} = \frac{v_{CO}}{u}$	κ_{uCO}	-
Dimension factor for each component passage	Ratio of the hydraulic diameter of each component passage against the reference diameter $\kappa_{dCO} = \frac{d_{hCO}}{D}$	κ_{dCO}	-

Term	Definition	Symbol	Unit
Dimension factor for disc friction loss	Ratio of the diameter of the runner crown or runner band against the reference diameter $\kappa_T = \frac{D_d}{D}$ D_d : diameter of the runner crown or the runner band, whichever larger	κ_T	-
Scalable hydraulic energy loss index for each component passage	$d_{E\text{C}0\text{ref}} = \frac{\delta_{E\text{C}0\text{ref}}}{1 + 0,351(\kappa_{u\text{CO}} \times \kappa_{d\text{CO}})^{0,2}}$	$d_{E\text{C}0\text{ref}}$	-
Scalable disc friction loss index	$d_{T\text{ref}} = \frac{\delta_{T\text{ref}}}{1 + 0,154 \kappa_T^{0,4}}$	$d_{T\text{ref}}$	-
Loss distribution factor	Ratio of scalable loss to total loss $V = \frac{\delta}{1 - \eta_h}$	V	-
Specific speed	$N_{QE} = \frac{nQ_1^{0,5}}{E^{0,75}}$	N_{QE}	-
Speed factor	$\eta_{ED} = \frac{nD}{E^{0,5}}$	η_{ED}	-
Discharge factor	$Q_{ED} = \frac{Q_1}{D^2 E^{0,5}}$	Q_{ED}	-
Power factor	$P_{ED} = \frac{P_m}{\rho_1 D^2 E^{1,5}}$	P_{ED}	-
Energy coefficient	$E_{nD} = \frac{E}{n^2 D^2}$	E_{nD}	-
Discharge coefficient	$Q_{nD} = \frac{Q_1}{n D^3}$	Q_{nD}	-
Power coefficient	$P_{nD} = \frac{P_m}{\rho_1 n^3 D^5}$	P_{nD}	-

4 Scale-effect formula

4.1 General

4.1.1 Scalable losses

The energy flux through hydraulic machines and the various losses produced in the energy conversion process in a hydraulic machine can be typically illustrated by the flux diagrams shown in A.1. [4]

As a consequence, one of the main features of the new scale up formula as stated in this standard is the separate consideration on three efficiency components. They are specific

hydraulic energy efficiency η_E , volumetric efficiency η_Q and power efficiency η_T . In this standard, scale effect on each of these efficiency components is considered.

Among the losses corresponding to these efficiency components, the following losses are subject to scale effect by the difference of Reynolds number and the relative roughness. Then these losses are referred to as “scalable losses” in this standard.

- Specific hydraulic energy loss due to friction: E_{Lf}
- Leakage loss: q
- Disc friction loss: P_{Ld}

It is considered in this standard that the relative magnitude of each scalable loss to each corresponding performance parameter, except for discharge, ($\delta_E = E_{Lf}/E$ and $\delta_T = P_{Ld}/P_m$) is given as a function of the specific speed for each type of machine.

E_{Lf} is the sum of the friction loss in various parts of the machine and it is expressed as the sum of the friction loss in each component as $E_{Lf} = \sum E_{LfCO}$. The scale effect on this loss is caused by the difference of Reynolds number and relative roughness between model and prototype and assessed by the formula shown as Equation 1.

The rest of the specific hydraulic energy loss is called “kinetic loss” or “non-scalable loss” and expressed as $E_{Lk} = \sum E_{LkCO}$. It is considered that the ratio of E_{Lk} against E_m remains the same through the model and the prototype.

The scale effect on the leakage loss, q , is caused by the change of the friction loss coefficient of the seal clearance of the runner/impeller. In most cases, the leakage loss itself is minor and the scale effect on this loss is relatively very small.

Therefore, in case that the geometry of the seal is maintained homologous between the model and the prototype within the criteria given in Table 3, the scale effect on the leakage loss is disregarded and η_Q of the prototype is considered to be the same as that of the model. (See E.3)

In case that the geometry of the model is not homologous to the prototype, this standard recommends to use the correction formula for η_Q as set out in E.2.

Similarly to E_{Lf} , the scale effect on the disc friction P_{Ld} is caused by the difference in Reynolds number and the relative roughness of the outer surface of the runner/impeller between the model and the prototype. Due to the presence of the radial flow and the distortion of the boundary layer in the limited space between the runner/impeller and the stationary parts, the scale effect on P_{Ld} appears in a slightly different manner than on E_{Lf} . It is considered in this standard that the scale effect on the disc friction may be assessed by a scale effect formula shown as Equation 7. (See Annex D)

In case of axial flow machines, the friction loss of the surface of runner hub is negligibly small and its scale effect is disregarded.

Therefore, in this standard, only the scale effect on the losses corresponding to the efficiency components; η_E and η_T , are considered for radial flow machines and only η_E is considered for axial flow machines.

4.1.2 Basic formulae of the scale effect on hydrodynamic friction losses

Another new feature of the new scale effect formula is the consideration of surface roughness. The basic physical background for consideration of surface quality is the Colebrook diagram. By some manipulation and simplification, the implicit Colebrook formula can be converted into an expression as shown below. [4, 6]

$$\lambda = \lambda_0 \left[0,74 \left(8 \times 10^4 \times \frac{k_S}{d_h} + \frac{Re_0}{Re_d} \right)^{0,2} + 0,26 \right] \quad (1)$$

where

$$Re_0 = 7 \times 10^6$$

$$\lambda_0 = 0,0085$$

k_S sand roughness

d_h hydraulic diameter of the water passage

Re_d Reynolds number of the water passage $Re_d = \frac{d_h \times v}{\nu} = \frac{d_h \times v}{D \times u} Re$

Practically, the surface roughness of model and prototype are represented by the arithmetical mean roughness R_a as stated in 4.2.2. Regarding the relationship between the sand roughness k_S and R_a , wide spread results have been reported so far. In this standard, however, it is considered that the relationship can be expressed by the following formula:

$$\frac{k_S}{d_h} \cong 5 \frac{R_a}{d_h} \quad (2)$$

NOTE For very rough surfaces, considerations as described in (2) and in Note 2 of B.1 should be taken into account.

Then, Equation 1 is rewritten as follows;

$$\lambda = \lambda_0 \left[0,74 \left(4 \times 10^5 \times \frac{R_a}{d_h} + \frac{D \times u}{d_h \times v} \times \frac{Re_0}{Re} \right)^{0,2} + 0,26 \right] \quad (3)$$

Figure 1 sketches the basic concept for the step-up from model to prototype conditions including surface roughness. Example P_3 shows the case of a smooth prototype machine. P_2 shows the case of a prototype machine of reasonable roughness, whereby P_1 shows the example of a very rough surface where even a decrease of efficiency compared to the model will occur.

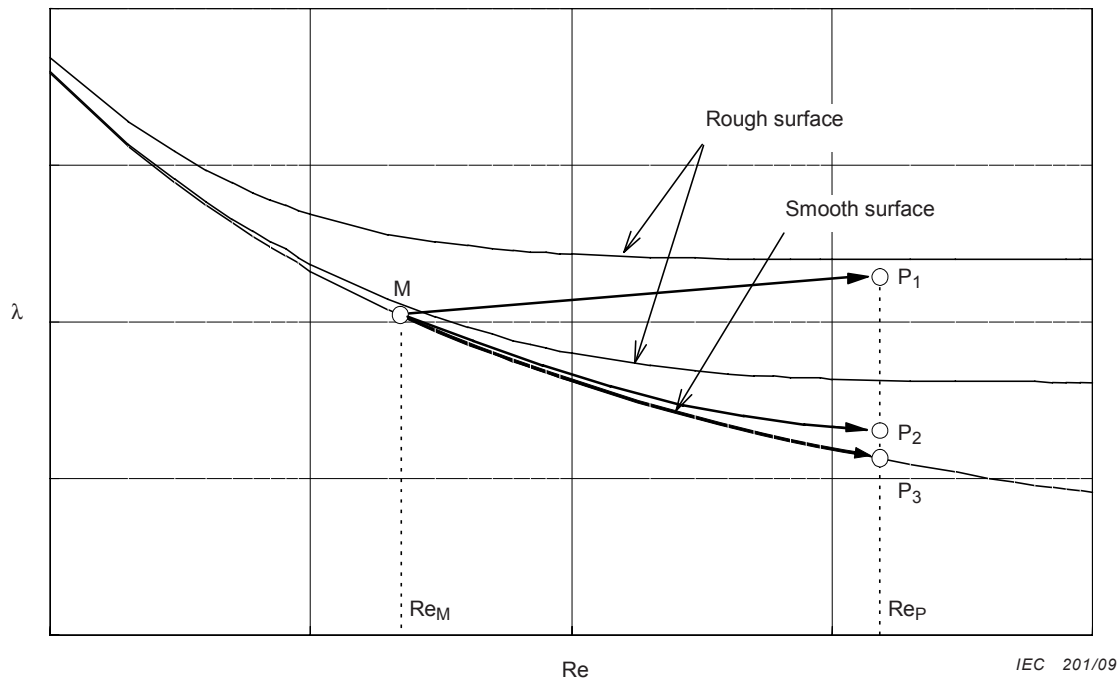


Figure 1 ± Basic concept for step-up considering surface roughness

In order to calculate the difference of hydraulic efficiency between two hydraulically similar operating points M and P at different Reynolds numbers and different surface roughness conditions, the following formulae can be derived by using Equation 3 (see A.2 (2)).

$$\Delta E = \frac{\Delta \eta_E}{\eta_{EM}} = \delta_{Eref} \left(\frac{\lambda_M - \lambda_P}{\lambda_{ref}} \right) \quad (4)$$

The Colebrook diagram is valid for pipe flow, but it can be demonstrated that also friction loss coefficients of flat plate flow can be approximated with sufficient accuracy by similar equations as shown below.

$$\begin{aligned} C_f &= C_{f0} \left[0,80 \left(10^5 \frac{k_S}{L} + \frac{Re_0}{Re_f} \right)^{0,2} + 0,20 \right] \\ &= C_{f0} \left[0,80 \left(5 \times 10^5 \frac{Ra}{L} + \frac{D \times u}{L \times w} \times \frac{Re_0}{Re} \right)^{0,2} + 0,20 \right] \end{aligned} \quad (5)$$

where

$$Re_0 = 7 \times 10^6 \quad C_{f0} = 0,003 \, 2$$

$$Re_f \quad \text{Reynolds number of the plate} \quad Re_f = \frac{L \times w}{\nu} = \frac{L \times w}{D \times u} Re$$

L length of the plate

w relative flow velocity on the plate

By replacing λ in Equation 4 by C_f given by Equation 5, Equation 4 is used to calculate the scale effect of the friction loss of runner blades of axial flow machines.

Similar formula of friction loss coefficient for disc friction is established as follows [9]; (See Annex D).

$$C_m = C_{m0} \left[0,85 \left(1,5 \times 10^4 \times \frac{k_{ST}}{a} + \frac{Re_0}{Re_T} \right)^{0,2} + 0,15 \right]$$

$$= C_{m0} \left[0,85 \left(7,5 \times 10^4 \frac{Ra_T}{a} + \frac{D^2}{2a^2} \times \frac{Re_0}{Re} \right)^{0,2} + 0,15 \right] \quad (6)$$

where

$$Re_0 = 7 \times 10^6 \quad C_{m0} = 0,0019$$

k_{ST} equivalent sand roughness corresponding to Ra_T $k_{ST} = 5Ra_T$

Ra_T weighted average of the arithmetical mean roughness of the outer surface of the runner and the surface of the stationary part facing to the runner as given by Equation 13

Re_T Reynolds number of the disc

$$Re_T = \frac{a^2 \omega}{\nu} = \frac{a^2 \omega}{Du} Re = \frac{2a^2}{D^2} Re = \frac{D_d^2}{2D^2} Re$$

a radius of runner crown or band, whichever larger (m)

$$a = \frac{D_d}{2}$$

ω angular velocity of the disc (rad/s)

By using Equation 6, step-up formula for power efficiency (disc friction) is obtained as follows (see A.2 (4)):

$$\Delta_T = \frac{\Delta \eta_T}{\eta_{TM}} = \delta_{Tref} \left(\frac{C_{mM} - C_{mP}}{C_{mref}} \right) \quad (7)$$

4.2 Specific hydraulic energy efficiency

4.2.1 Step-up formula

The scalable losses δ_{Eref} as appeared in Equation 4 are referred to those of a model with smooth surface operating at a reference Reynolds number $Re_{ref} = 7 \times 10^6$ and have been established as a function of type and specific speed of a hydraulic machine. They are standardized and set out in Annex B for radial flow machines and Annex C for axial flow machines.

By putting the new scale effect formula Equation 3 into Equation 4, the following formula for the individual step-up for a machine component is derived (see B.2).

$$\begin{aligned}
\Delta_{ECO} &= \frac{\Delta\eta_{ECO}}{\eta_{EM}} = \delta_{ECOref} \left(\frac{\lambda_{COM} - \lambda_{COP}}{\lambda_{COref}} \right) \\
&= \delta_{ECOref} \left[\frac{\left(4 \times 10^5 \kappa_{uCO} \frac{Ra_{COM}}{D_M} + \frac{7 \times 10^6}{Re_M} \right)^{0,2} - \left(4 \times 10^5 \kappa_{uCO} \frac{Ra_{COP}}{D_P} + \frac{7 \times 10^6}{Re_P} \right)^{0,2}}{1 + 0,35(\kappa_{uCO} \times \kappa_{dCO})^{0,2}} \right] \quad (8) \\
&= d_{ECOref} \left[\left(4 \times 10^5 \kappa_{uCO} \frac{Ra_{COM}}{D_M} + \frac{7 \times 10^6}{Re_M} \right)^{0,2} - \left(4 \times 10^5 \kappa_{uCO} \frac{Ra_{COP}}{D_P} + \frac{7 \times 10^6}{Re_P} \right)^{0,2} \right]
\end{aligned}$$

where

- δ_{ECOref} standardized reference scalable loss for each component passage when the machine Reynolds number Re_M is equal to the reference Reynolds number (7×10^6) (see A.2 (2) and B.2 (2))
- κ_{uCO} standardized flow velocity factor for each component passage (see B.2 (1))
- κ_{dCO} standardized dimension factor for each component passage (see B.2 (1))
- δ_{ECOref} scalable loss index for each component passage (see B.2 (2))

$$d_{ECOref} = \frac{\delta_{ECOref}}{1 + 0,35(\kappa_{uCO} \times \kappa_{dCO})^{0,2}}$$

For radial flow machines, Equation 8 allows to calculate the individual step-ups in the various components, using d_{ECOref} and κ_{uCO} which have been established for each individual component from spiral case to draft tube.

The values of d_{ECOref} and κ_{uCO} for each component passage of Francis turbine and pump-turbine are standardized and shown in 5.3 (1) and (2).

For axial flow machines, the scalable loss is divided into two parts, runner blades and stationary parts. The efficiency step-up ratio for the scalable loss of stationary parts, Δ_{EST} , can be obtained by Equation 8 in the same way as for radial flow machines. In this case, it is considered that the representative flow velocity factor κ_{uST} for all stationary parts can be represented by 0,8 times the flow velocity factor for guide vanes; namely, $\kappa_{uST} = 0,8 \times \kappa_{uGV}$. The value of κ_{uST} is shown in 5.3 (see Annex N of IEC 60193:1999 [1]).

As stated below Equation 5 in 4.1.2, scale effect formula for flat plate represented by Equation 5 is supposed to be applied to runner blades. However, as demonstrated in C.2, the scale effect formula based on Equation 5 can be transformed to the same formula as Equation 8 by introducing the modified flow velocity factor κ_{uRU}^* instead of κ_{uRU} . Therefore, the following formula similar to Equation 8 can be applied to runner blades by using κ_{uRU}^* given in 5.3 (see Annex N of IEC 60193:1999 [1]).

$$\Delta_{ERU} = d_{ERUref} \left[\left(4 \times 10^5 \kappa_{URU}^* \frac{Ra_{RUM}}{D_M} + \frac{7 \times 10^6}{Re_M} \right)^{0,2} - \left(4 \times 10^5 \kappa_{URU}^* \frac{Ra_{RUP}}{D_P} + \frac{7 \times 10^6}{Re_P} \right)^{0,2} \right] \quad (9)$$

Then the step-up of the specific hydraulic energy for the whole machine can be calculated by the equation below:

$$\Delta_E = \frac{\Delta \eta_E}{\eta_{EM}} = \sum \Delta_{ECO} \quad (10)$$

The structure of formula is valid for all types of hydraulic reaction machines. Also it can be applied for both turbines and pumps.

4.2.2 Roughness of model and prototype

When applying Equation 8 for the contractual model test to examine whether the model efficiency meets the guarantee or not, the values of surface roughness (Ra) as stipulated below shall be used in the formula.

– Roughness of the model

The values measured on the model shall be used. The model components are known to have a very good uniformity of roughness per component. When this is the case, 2 to 4 measuring points per component shall suffice. For repetitive components, like stay vanes, guide vanes and runner blades, measurement on at least 2 repetitive components is recommended.

– Roughness of the prototype

Design values for the prototype roughness, which are offered by the supplier, shall be used as the roughness of the prototype. When the turbine components are completed in the factory, the surface roughness shall be measured and it should be verified that the average value of the measured roughness of each component is equal or finer than the design roughness for the component.

When applying Equation 8 for the assessment of the efficiency improvement in a rehabilitation project, the roughness of the prototype components shall be measured on the existing unit. The improvement of the efficiency achievable by the replacement of some components can be assessed by comparing the efficiencies calculated with the roughness measured on the existing components and with the design values for the new components.

In case of rehabilitation projects, roughness data of those components not to be replaced shall be provided by the owner with the specification. For the measurement of rough surfaces of old turbines, the recommendations described in Annex B (at the end of B.1) for Ra values larger than 50 µm shall be taken into consideration.

When the roughness is measured on the model or the prototype, measurement shall be made carefully so that the measured values may represent the roughness of each component adequately.

For spiral case, stay vanes and draft tube, the sample points shall be selected so as to represent the average roughness of the component correctly. For guide vanes and runner, the sample points shall be selected so as to represent the average roughness of the high flow velocity area of their passages. It is recommended to measure the roughness at sample points as shown below and to use their arithmetic average for each component.

- Spiral case: 9 points or more; at 3 radial sections: entrance, middle, end of casing.
- 2 Stay vane channels: 6 points or more per stay vane channel; 2 points per side of the vane, 1 point on the top of the channel, 1 point on the bottom of the channel.

- 2 Guide vane channels: 10 points or more per guide vane channel between 2 guide vanes; 6 points on the inner side of the guide vane, 2 points on the outer side of the guide vane, 1 point on the top of the channel, 1 point on the bottom of the channel.
- Runner: 20 points or more; with 70 % of them on the high flow velocity area (region A, as defined in Table 1). The number of measuring points on pressure and suction sides of the blade shall be identical.
- Draft tube: 10 points or more; with 70 % of them upstream of the bend.

The surface roughness shall be measured as it appears in actual operation. Painted surface shall be measured over the paint coat.

For axial flow machines, the roughness value given by Equation 11 shall be used as a representative roughness for all stationary parts.

$$Ra_{ST} = \frac{Ra_{SV} + Ra_{GV}}{2} \quad (11)$$

As known by Equation 8, larger efficiency step-up can be achieved by polishing the prototype finer. Nevertheless, the roughness of the prototype should not be finer than the roughness expected after some period of operation (i.e. guaranteed period). Also, very fine polishing is not cost effective, as shown in Figure 2.

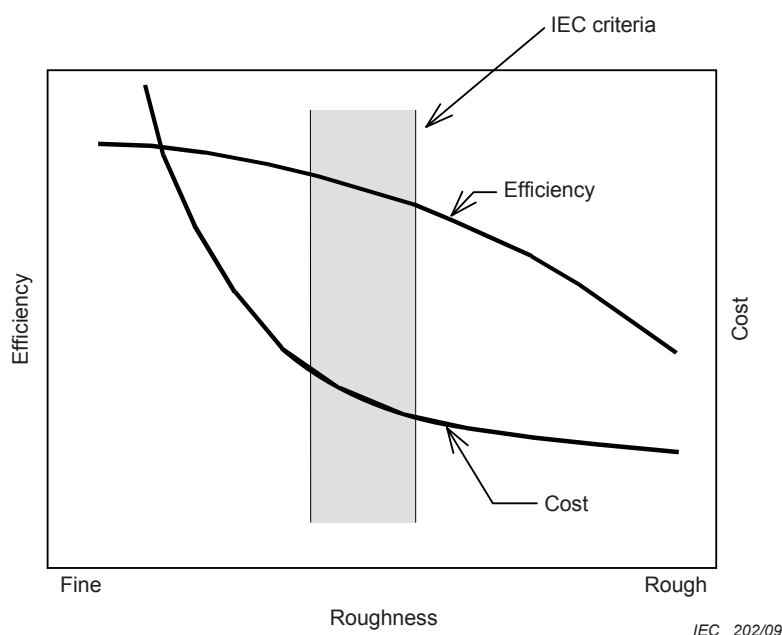


Figure 2 ± IEC criteria of surface roughness given in Tables 1 and 2

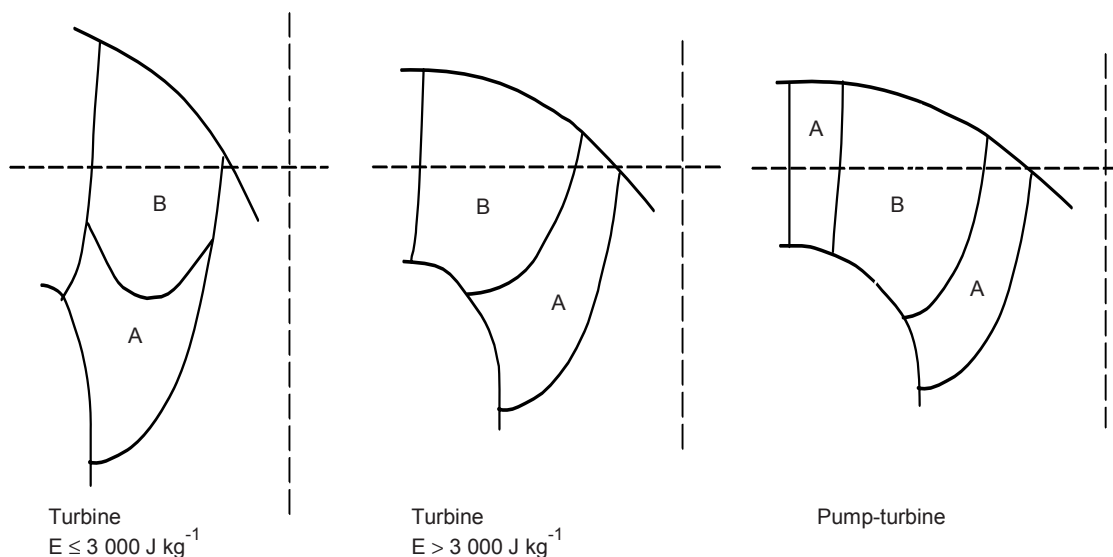
Tables 1 and 2 show the maximum recommended roughness for prototype runner and guide vanes of new turbines. These recommended roughness values supersede those given in IEC 60193.

Table 1 ± Maximum recommended prototype runner roughness for new turbines (µm)

E ≤ 3 000 J.kg^{±1}								
Reference diameter	1 m – 2 m		2 m – 4 m		4 m – 7 m		7 m – 10 m	
Region	A^a	B^a	A	B	A	B	A	B
Roughness (Ra) Pressure side	2,3	3,2	6,3	12,5	12,5	25 ^b	12,5	25 ^b
Roughness (Ra) Suction side	2,3	2,3	2,3	3,2	3,2	6,3	6,3	6,3
E > 3 000 J.kg^{±1}								
Reference diameter	1 m – 2 m		2 m – 4 m		4 m – 7 m		7 m – 10 m	
Region	A	B	A	B	A	B	A	B
Roughness (Ra) Pressure side	2,3	2,3	2,3	3,2	3,2	6,3	6,3	6,3
Roughness (Ra) Suction side	1,6	1,6	2,3	2,3	2,3	3,2	3,2	4,5

^a Even though there are only 2 regions A and B in this table, it is well understood that an additional region along the blade inflow edge is often polished to a very low roughness, in order to avoid initiation of cavitation.

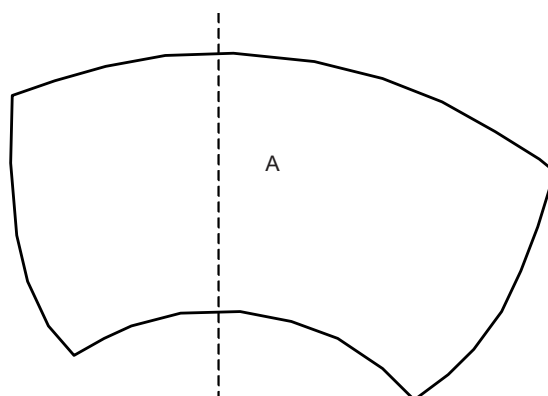
^b These roughness values may seem excessive for these regions. However, the above values were established based on comparable roughness losses between different machine sizes, having different Reynolds number. So, bigger machines, having bigger Reynolds number can afford more roughness. However, it is reasonable to use smaller roughness values than the ones recommended, if the parties involved feel that it is more practical or more economical for the project concerned.



IEC 203/09

NOTE Concerning the surface roughness along the runner band and the runner crown, a mid value between the "Pressure side" region and the "Suction side" region is recommended.

Figure 3 ± Francis Runner blade and fillets



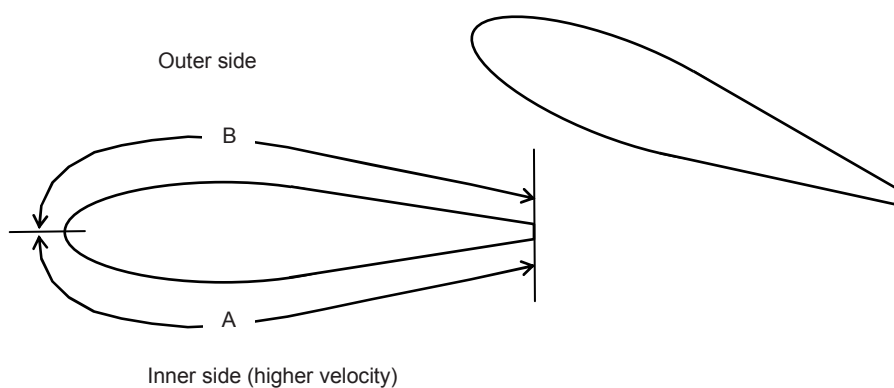
IEC 204/09

NOTE It is recommended to apply the roughness values specified for "Blade Suction" in Table 1 to both pressure and suction sides of the runner blades for axial flow machines.

Figure 4 ± Runner blade axial flow

Table 2 ± Maximum recommended prototype guide vane roughness for new turbines (μm)

$E \leq 3\,000 \text{ J.kg}^{\pm 1}$								
Reference diameter	1 m – 2 m		2 m – 4 m		4 m – 7 m		7 m – 10 m	
Region	A	B	A	B	A	B	A	B
Roughness (Ra)	2,3	2,3	2,3	6,3	3,2	12,5	6,3	12,5
$E > 3\,000 \text{ J.kg}^{\pm 1}$								
Reference diameter	1 m – 2 m		2 m – 4 m		4 m – 7 m		7 m – 10 m	
Region	A	B	A	B	A	B	A	B
Roughness (Ra)	1,6	2,3	2,3	2,3	2,3	3,2	3,2	6,3



IEC 205/09

NOTE Concerning the surface roughness along the guide vane passage top and bottom, a mean value of A and B is recommended.

Figure 5 ± Guide vanes

4.2.3 Direct step-up for a whole turbine

When the surface roughness of a component passage is finished adequately, corresponding to the flow velocity of each component passage, the step-up of the specific hydraulic energy efficiency for the whole turbine Δ_E can be calculated directly without calculating Δ_{ECO} for the components. Such simplified procedure is described in B.3 for radial flow machines and in C.10 for axial flow machines. Those simplified formulae may be used upon prior agreement among the concerned parties.

4.3 Power efficiency (disc friction)

4.3.1 Step-up formula

Disc friction has a significant impact on the efficiency of low specific speed radial machines. The following step-up formula, Equation 12, is obtained by putting Equation 6 into Equation 7. It describes the variation of power efficiency of radial flow machines due to the difference in Reynolds number and surface roughness (see Annex D).

$$\begin{aligned}\Delta_T &= \frac{\Delta\eta_T}{\eta_{TM}} = \delta_{Tref} \left(\frac{C_{mM} - C_{mP}}{C_{mref}} \right) \\ &= \delta_{Tref} \frac{\left(7,5 \times 10^4 \kappa_T \frac{Ra_{TM}}{D_M} + \frac{7 \times 10^6}{Re_M} \right)^{0,2} - \left(7,5 \times 10^4 \kappa_T \frac{Ra_{TP}}{D_P} + \frac{7 \times 10^6}{Re_P} \right)^{0,2}}{1 + 0,154 \kappa_T^{0,4}} \\ \therefore \Delta_T &= d_{Tref} \left[\left(7,5 \times 10^4 \kappa_T \frac{Ra_{TM}}{D_M} + \frac{7 \times 10^6}{Re_M} \right)^{0,2} - \left(7,5 \times 10^4 \kappa_T \frac{Ra_{TP}}{D_P} + \frac{7 \times 10^6}{Re_P} \right)^{0,2} \right] \quad (12)\end{aligned}$$

where

$$\delta_{Tref} = 1 - \eta_{Tref}$$

$$d_{Tref} = \frac{\delta_{Tref}}{1 + 0,154 \kappa_T^{0,4}}$$

κ_T : dimension factor for the disc relating to disc friction loss

$$\kappa_T = \frac{2a}{D} = \frac{D_d}{D}$$

Ra_T : representative roughness given by Equation 13.

The scalable disc friction loss d_{Tref} as appeared in Equation 12 is referred to the model at the reference Reynolds number $Re_{ref} = 7 \times 10^6$ with smooth surface. The values of d_{Tref} and κ_T have been established as a function of type and specific speed of a radial flow machine. They are standardized and set out in 5.4.

For axial flow machines, the surface friction of runner hub is negligibly small. Therefore, it is considered in this standard that Δ_T is zero for axial flow machines.

4.3.2 Roughness of model and prototype

Generally the rules stated in 4.2.2 apply to the roughness concerning the disc friction except the requirement for the sample points as set out below.

Since the roughness near the outer periphery of runner crown and runner band has dominant influence on the disc friction loss, it is recommended to measure the roughness at the sample points as set out below.

- Runner crown: 2 points or more near outer periphery.
- Runner band: 2 points or more near outer periphery.
- Stationary part: 4 points or more at the areas facing to the sample points of runner.

Since the roughness of the rotating part has dominant influence of the disc friction torque, the weighted mean roughness as given by the following formula shall be used for Ra_T in Equation 12.

$$Ra_T = \frac{2 \times Ra_{TR} + Ra_{TS}}{3} \quad (13)$$

where

Ra_{TR} average roughness of those measured on the rotating part;

Ra_{TS} average roughness of those measured on the stationary part.

4.4 Volumetric efficiency

An estimation of the influence of Reynolds number to the volumetric efficiency demonstrates that the influence is almost negligibly small in case that the geometrical configuration of clearances, labyrinths, balancing holes/pipes is similar at both model and prototype. Therefore, in case that the geometry of the seal of the model is made homologous to the prototype within the deviation as set out in Table 3 below, the volumetric efficiency are considered to be the same at model and prototype (see E.3).

Table 3 ± Permissible deviation of the geometry of model seals from the prototype

Dimensions and design	Permissible deviation from the prototype
Radial clearance of runner seals *	0 ~ + 20 %
Diameter of seal	± 5 %
Axial length of seal clearances *	0 ~ - 20 %
Number of steps or grooves	should be the same
Shape of steps or grooves (see Annex D)	shall be homologous
NOTE In case of axial flow machines, the words marked by * should be read as "blade tip clearances" and "thickness of blade tip", respectively. Only these two criteria for radial clearance and thickness of blade tip should be applied.	

However, normally it is quite difficult, sometimes not practicable and sometimes impossible, to fabricate the runner seals of the model in complete homology with the corresponding prototype. In all these cases, the leakage flow has to be calculated separately for both model and prototype and the volumetric efficiency has to be adjusted accordingly. In this case one can write

$$\Delta_Q = \frac{\Delta\eta_Q}{\eta_{QM}} = \frac{\eta_{QP}}{\eta_{QM}} - 1 \quad (14)$$

If there is no agreement between the concerned parties about the calculation of Δ_Q , the formula given in E.2 may be applied.

5 Standardized values of scalable losses and pertinent parameters

5.1 General

The values of d_{ECOref} and κ_{uCO} to calculate the step-up of specific hydraulic energy efficiency and those of d_{Tref} and κ_T to calculate the step-up of power efficiency (disc friction) are shown in this clause. They are referred to a reference Reynolds number $Re_{ref} = 7 \times 10^6$ and correspond to the machines with smooth surface.

5.2 Specific speed

A hydraulic machine of any type can be described by its specific speed at the point of maximum efficiency. Therefore in the first step, the specific speed N_{QE} of the tested machine at its maximum efficiency has to be calculated.

$$N_{QE} = \frac{n \times Q_1^{0,5}}{E^{0,75}} \quad \text{or} \quad N_{QE} = n_{ED} Q_{ED}^{0,5} = \frac{Q_{nD}^{0,5}}{E_{nD}^{0,75}} \quad (15)$$

where

n rotational speed (s^{-1});

Q_1 discharge of machine (m^3/s);

E specific hydraulic energy of machine ($J \, kg^{-1}$).

For reversible pump-turbines, the specific speed at each maximum efficiency point when operating as a turbine or as a pump, should be calculated and taken as a reference to obtain the scalable losses in turbine or pump operation, respectively.

As the specific speeds of different machines from different manufacturers for the same specified prototype conditions are quite close, it is possible to fix d_{ECOREf} , κ_{uCO} , d_{Tref} and κ_T in advance in a specification. Also, for a comparative model test, common values of d_{ECOREf} , κ_{uCO} , d_{Tref} and κ_T should be defined.

5.3 Parameters for specific hydraulic energy efficiency step-up

Once an investigated hydraulic machine is described by its specific speed, the factors d_{ECOREf} and κ_{uCO} for a smooth model, which are required to apply the step-up formula, can be determined by the equations shown in Tables 4, 5, 6 and 7.

These equations are valid in the specific speed range shown below each table.

NOTE Beyond these specific speed ranges, the equations are not substantiated by analytical or experimental data and may not be correct. However, even beyond these specific speed ranges, the attached Excel sheets give step-up values which are calculated by extrapolating the equations. These step-up values are shown primarily for information. If they are used for the evaluation of contractual model test results, agreement shall be made in advance among the concerned parties.

1) Francis turbines

Table 4 ± Scalable loss index d_{ECOREf} and velocity factor κ_{uCO} for Francis turbines

Component passage	d_{ECOREf}	κ_{uCO}
Spiral case	$d_{ESPref} = 0,40 / 100$	$\kappa_{uSP} = -0,5 N_{QE} + 0,33$
Stay vanes	$d_{ESVref} = (-N_{QE} + 0,40) / 100$	$\kappa_{uSV} = -1,4 N_{QE} + 0,60$
Guide vanes	$d_{EGVref} = (-2,9 N_{QE} + 1,65) / 100$	$\kappa_{uGV} = -3,3 N_{QE} + 1,29$
Runner	$d_{ERUref} = (3,4 N_{QE} + 0,55) / 100$	$\kappa_{uRU} = -1,3 N_{QE} + 0,90$
Draft tube	$d_{EDTref} = (0,5 N_{QE} + 0,05) / 100$	$\kappa_{uDT} = 0,28$
NOTE The above equations are valid for $0,06 \leq N_{QE} \leq 0,30$ (see B.4, B.5 and B.6).		