BS EN 62097:2009

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

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National foreword

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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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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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Up-to-date lists and bibliographical references concerning such national standards may be obtained on application to the Central Secretariat or to any CENELEC member.

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CENELEC

European Committee for Electrotechnical Standardization
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Europäisches Komitee für Elektrotechnische Normung

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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.

The following dates were fixed:

– latest date by which the EN has to be implemented at national level by publication of an identical national standard or by endorsement	(dop)	2009-12-01
– latest date by which the national standards conflicting with the EN have to be withdrawn	(dow)	2012-03-01

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.

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

CONTENTS

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	
model	M	component	CO	
prototype	P			
specific energy	E	spiral case	SP	in general term represented by CO (SP, SV, GV, RU, DT, ST)
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	$m^2\ s^{-1}$
Discharge	Volume of water per unit time flowing through any section in the system	Q	$m^3\ s^{-1}$
Mass flow rate	Mass of water flowing through any section of the system per unit time	(ρ Q)	$kg\ s^{-1}$
Discharge of machine	Discharge flowing through the high pressure reference section	Q_1	$m^3\ s^{-1}$
Leakage flow rate	Volume of water per unit time flowing through the runner seal clearances	q	$m^3\ s^{-1}$
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	$m^3\ s^{-1}$
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 Pump: Specific hydraulic energy produced by the impeller	E_m E_m	$J\ kg^{-1}$ $J\ kg^{-1}$
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	$Re_d = d_h\ v\ /\ \nu$	Re_d	-
Friction loss coefficient for pipe flow	Friction loss coefficient for a pipe. $\lambda = \frac{E_{Lf}}{\frac{L}{d}\frac{v^2}{2}}$ 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. $$C_f = \frac{E_{Lf}}{\frac{BL}{Q}\frac{w^3}{2}}$$ where B width of a flat plate (m) L length of a flat plate (m) Q discharge passing by the plate (m^3/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 %) $$\delta_E = E_{lf}/E$$	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 = 7×10^6	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 = 7×10^6	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	$d_{ECOref} = \frac{\delta_{ECOref}}{1 + 0{,}351(\kappa_{uCO} \times \kappa_{dCO})^{0{,}2}}$	d_{ECOref}	-
Scalable disc friction loss index	$d_{Tref} = \frac{\delta_{Tref}}{1 + 0{,}154\, \kappa_T^{0{,}4}}$	d_{Tref}	-
Loss distribution factor	Ratio of scalable loss to total loss $V = \frac{\delta}{1 - \eta_h}$	V	-
Specific speed	$N_{QE} = \frac{n Q_1^{0{,}5}}{E^{0{,}75}}$	N_{QE}	-
Speed factor	$n_{ED} = \frac{nD}{E^{0{,}5}}$	n_{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	$Q_{nD} = \frac{Q_1}{nD^3}$	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 as expression as shown below. [4, 6]

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

where

$Re_0 = 7 \times 10^6$ $\lambda_0 = 0{,}008\,5$

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 Ra as stated in 4.2.2. Regarding the relationship between the sand roughness k_S and Ra, 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:

$$\frac{k_S}{d_h} \cong 5 \frac{Ra}{d_h} \tag{2}$$

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;

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

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.

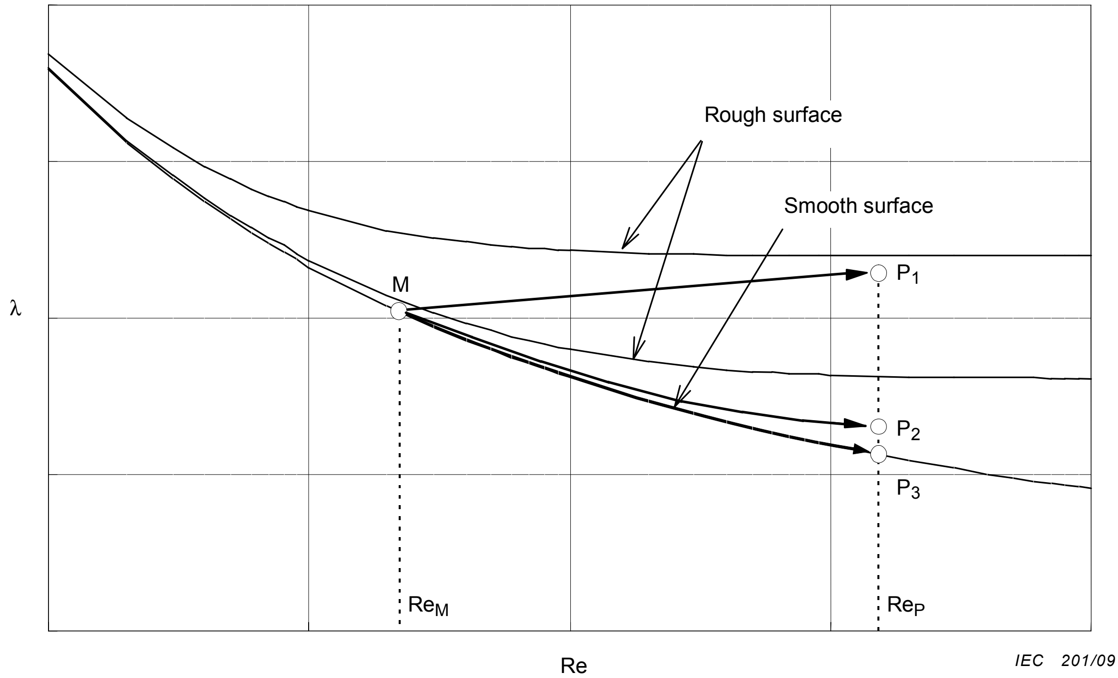


Figure 1 – Basic concept for step-up considering surface roughness

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)).

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

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.

$$\begin{aligned} C_f &= C_{f0}\left[0{,}80\left(10^5\frac{k_S}{L} + \frac{Re_0}{Re_f}\right)^{0{,}2} + 0{,}20\right] \\ &= C_{f0}\left[0{,}80\left(5\times10^5\frac{Ra}{L} + \frac{D\times u}{L\times w}\times\frac{Re_0}{Re}\right)^{0{,}2} + 0{,}20\right] \end{aligned} \tag{5}$$

where

$Re_0 = 7\times10^6$ $\quad$ $C_{f0} = 0{,}003\,2$

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

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).

$$C_m = C_{m0}\left[0{,}85\left(1{,}5\times10^4\times\frac{k_{ST}}{a}+\frac{Re_0}{Re_T}\right)^{0{,}2}+0{,}15\right]$$
$$= C_{m0}\left[0{,}85\left(7{,}5\times10^4\,\frac{Ra_T}{a}+\frac{D^2}{2a^2}\times\frac{Re_0}{Re}\right)^{0{,}2}+0{,}15\right] \qquad (6)$$

where

$Re_0 = 7\times10^6$ $\qquad C_{m0} = 0{,}0019$

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

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)):

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

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 $Re_{ref} = 7\times10^6$ 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).

$$\Delta_{ECO} = \frac{\Delta\eta_{ECO}}{\eta_{EM}} = \delta_{ECOref}\left(\frac{\lambda_{COM} - \lambda_{COP}}{\lambda_{COref}}\right)$$

$$= \delta_{ECOref}\left[\frac{\left(4\times10^5\,\kappa_{uCO}\,\frac{Ra_{COM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0,2} - \left(4\times10^5\,\kappa_{uCO}\,\frac{Ra_{COP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0,2}}{1+0,35(\kappa_{uCO}\times\kappa_{dCO})^{0,2}}\right] \quad (8)$$

$$= d_{ECOref}\left[\left(4\times10^5\,\kappa_{uCO}\,\frac{Ra_{COM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0,2} - \left(4\times10^5\,\kappa_{uCO}\,\frac{Ra_{COP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0,2}\right]$$

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]).

$$\Delta_{ERU} = d_{ERUref}\left[\left(4\times10^5\kappa^*_{uRU}\frac{Ra_{RUM}}{D_M}+\frac{7\times10^6}{Re_M}\right)^{0,2}-\left(4\times10^5\kappa^*_{uRU}\frac{Ra_{RUP}}{D_P}+\frac{7\times10^6}{Re_P}\right)^{0,2}\right] \quad (9)$$

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.

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

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.

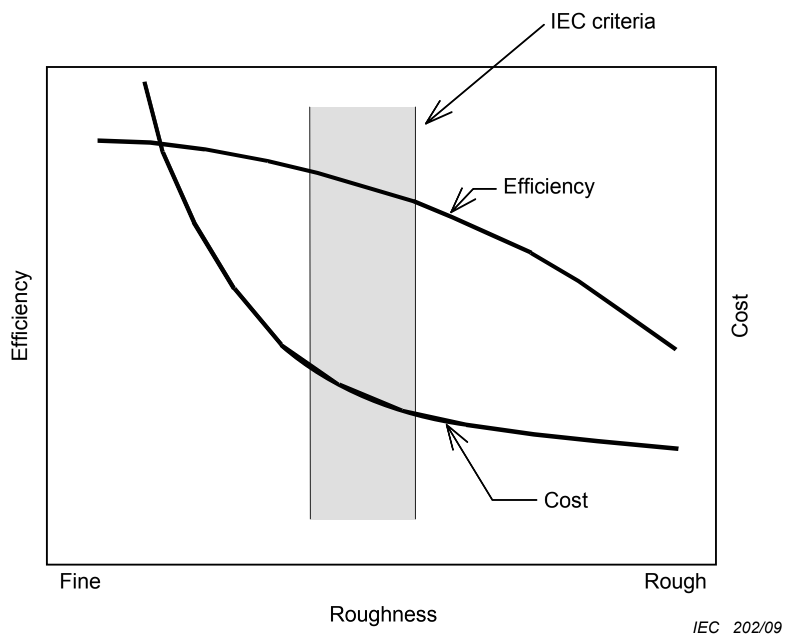


Figure 2 – IEC criteria of surface roughness given in Tables 1 and 2

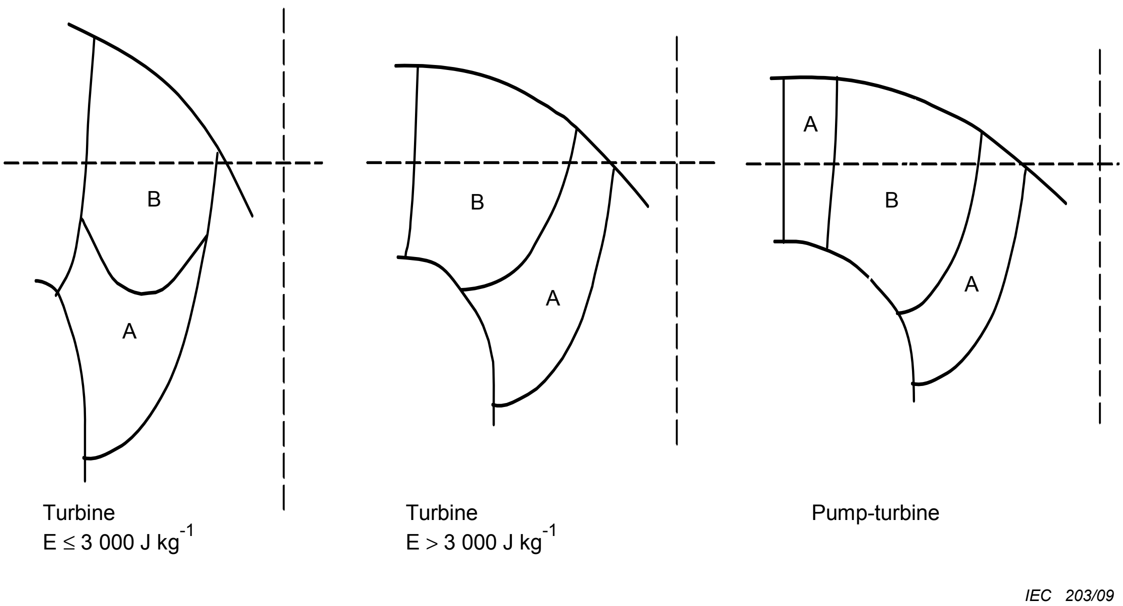
Tables 1 and 2 show the maximum recommended roughness for prototype runner and guide vanes of new turbines. These recommended roughness values supersede those given in IEC 60193.

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

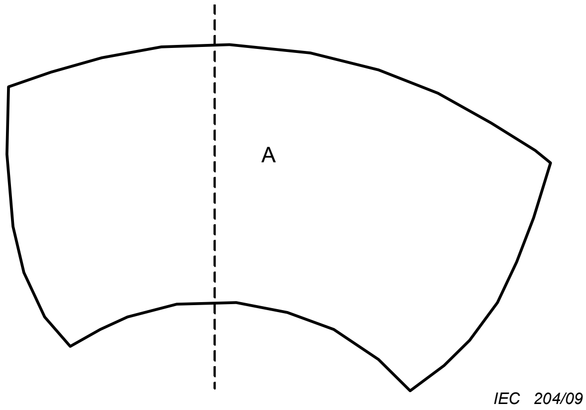
[a] Even though there are only 2 regions A and B in this table, it is well understood that an additional region along the blade inflow edge is often polished to a very low roughness, in order to avoid initiation of cavitation.

[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.



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

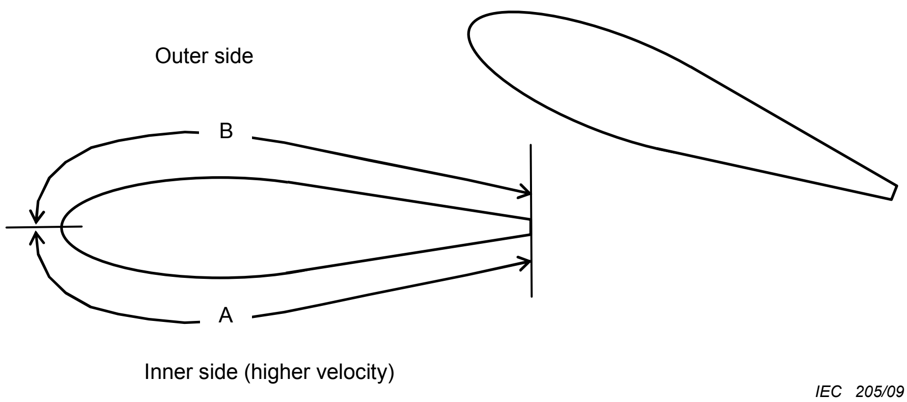


NOTE It is recommended to apply the roughness values specified for "Blade Suction" in Table 1 to both pressure and suction sides of the runner blades for axial flow machines.

Figure 4 – Runner blade axial flow

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

E ≤ 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)	2,3	2,3	2,3	6,3	3,2	12,5	6,3	12,5
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)	1,6	2,3	2,3	2,3	2,3	3,2	3,2	6,3



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).

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

$$= \delta_{Tref}\frac{\left(7{,}5\times10^4\,\kappa_T\,\frac{Ra_{TM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0{,}2} - \left(7{,}5\times10^4\,\kappa_T\,\frac{Ra_{TP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0{,}2}}{1 + 0{,}154\kappa_T{}^{0{,}4}}$$

$$\therefore\quad \Delta_T = d_{Tref}\left[\left(7{,}5\times10^4\kappa_T\frac{Ra_{TM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0{,}2} - \left(7{,}5\times10^4\kappa_T\frac{Ra_{TP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0{,}2}\right] \qquad (12)$$

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} = 7×10^6 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.

$$Ra_T = \frac{2 \times Ra_{TR} + Ra_{TS}}{3} \tag{13}$$

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

$$\Delta_Q = \frac{\Delta\eta_Q}{\eta_{QM}} = \frac{\eta_{QP}}{\eta_{QM}} - 1 \tag{14}$$

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} = 7×10^6 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.

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

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 $0{,}06 \le N_{QE} \le 0{,}30$ (see B.4, B.5 and B.6).		

2) Pump-turbines

a) Turbine operation

Table 5 – Scalable loss index d_{ECOref} and velocity index κ_{uCO} for pump-turbines in turbine operation

Component passage	d_{ECOref}	κ_{uCO}
Spiral case	$d_{ESPref} = 0{,}45/100$	$\kappa_{uSP} = -0{,}5\,N_{QE} + 0{,}34$
Stay vanes	$d_{ESVref} = (-N_{QE} + 0{,}45)/100$	$\kappa_{uSV} = -1{,}4\,N_{QE} + 0{,}57$
Guide vanes	$d_{EGVref} = (-2{,}9\,N_{QE} + 1{,}65)/100$	$\kappa_{uGV} = -3{,}3\,N_{QE} + 1{,}23$
Runner	$d_{ERUref} = (3{,}4\,N_{QE} + 1{,}35)/100$	$\kappa_{uRU} = -1{,}3\,N_{QE} + 0{,}87$
Draft tube	$d_{EDTref} = (0{,}5\,N_{QE} + 0{,}05)/100$	$\kappa_{uDT} = 0{,}31$
NOTE The above equations are valid for $0{,}06 \le N_{QE} \le 0{,}20$ (see B.4, B.5 and B.6).		

b) Pump operation

Table 6 – Scalable loss index d_{ECOref} and velocity index κ_{uCO} for pump-turbines in pump operation

Component passage	d_{ECOref}	κ_{uCO}
Spiral case	$d_{ESPref} = 0{,}45/100$	$\kappa_{uSP} = -0{,}5\,N_{QE} + 0{,}31$
Stay vanes	$d_{ESVref} = (-N_{QE} + 0{,}50)/100$	$\kappa_{uSV} = -1{,}4\,N_{QE} + 0{,}53$
Guide vanes	$d_{EGVref} = (-2{,}9\,N_{QE} + 1{,}65)/100$	$\kappa_{uGV} = -3{,}3\,N_{QE} + 0{,}96$
Runner	$d_{ERUref} = (3{,}4\,N_{QE} + 1{,}55)/100$	$\kappa_{uRU} = -1{,}3\,N_{QE} + 0{,}79$
Draft tube	$d_{EDTref} = (0{,}5\,N_{QE} + 0{,}05)/100$	$\kappa_{uDT} = 0{,}27$
NOTE The above equations are valid for $0{,}06 \le N_{QE} \le 0{,}20$ (see B.4, B.5 and B.6).		

3) Axial flow machines

Table 7 – Scalable loss index d_{ECOref} and velocity factor κ_{uCO} for axial flow machines

Component passage	d_{ECOref}	κ_{uCO}
Runner	$d_{ERUref} = 2{,}45/100$	$\tilde{\kappa}_{uRU} = 1{,}29$
All stationary parts	$d_{ESTref} = 1{,}23/100$	$\kappa_{uST} = 0{,}19$
NOTE The above equations are valid for $0{,}25 \le N_{QE} \le 0{,}70$ (see C.9).		

5.4 Parameters for power efficiency (disc friction) step-up

The following equations shall be used to obtain d_{Tref} and κ_T (see D.3). These equations are valid in the specific speed range shown for each equation.

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, they may be used for the evaluation of contractual model test results by mutual agreement among the concerned parties.

1) Francis turbines

$$d_{Tref} = \left(0{,}44 + \frac{0{,}004}{N_{QE}^{\ 2}}\right) \times \frac{1}{100} \text{ for } 0{,}06 \leq N_{QE} \leq 0{,}30 \quad (16)$$

$$\kappa_T = -5{,}7\,N_{QE} + 2{,}0 \quad \text{or} \quad 1{,}0\,, \quad \text{whichever larger} \quad (17)$$

2) Pump-turbines

a) Turbine operation

$$d_{Tref} = \left(0{,}97 + \frac{0{,}012}{N_{QE}^{\ 2}}\right) \times \frac{1}{100} \text{ for } 0{,}06 \leq N_{QE} \leq 0{,}20 \quad (18)$$

$$\kappa_T = -8{,}3\,N_{QE} + 2{,}7 \quad \text{or} \quad 1{,}0\,, \quad \text{whichever larger} \quad (19)$$

b) Pump operation

$$d_{Tref} = \left(1{,}23 + \frac{0{,}015}{N_{QE}^{\ 2}}\right) \times \frac{1}{100} \text{ for } 0{,}06 \leq N_{QE} \leq 0{,}20 \quad (20)$$

$$\kappa_T = -7{,}5\,N_{QE} + 2{,}7 \quad \text{or} \quad 1{,}0\,, \quad \text{whichever larger} \quad (21)$$

6 Calculation of prototype performance

6.1 General

The formulae set out from 6.2 to 6.6 concern the conversion of the hydraulic performance data from a homologous model to a prototype for hydraulically similar operating conditions.

Using the methods of measurement described in IEC 60193, absolute model test data such as η_M, E_M, Q_M, T_M, P_M, Re_M, etc. are obtained for each test point.

With additional absolute data of model and prototype such as n, D, g and ρ, the corresponding prototype performance data can be calculated.

For off-design points, Δ_E, Δ_T and Δ_Q calculated for the maximum efficiency point shall be used from Equation 22 to Equation 33. It should be noted that this procedure gives slightly less step-up of efficiency for off-design points comparative to the maximum efficiency point.

6.2 Hydraulic efficiency

The hydraulic prototype efficiency of a hydraulic machine can be calculated by the following formula:

$$\frac{\eta_{hP}}{\eta_{hM}} = \frac{\eta_{EP} \times \eta_{TP} \times \eta_{QP}}{\eta_{EM} \times \eta_{TM} \times \eta_{QM}} = (1 + \Delta_E)(1 + \Delta_T)(1 + \Delta_Q) \quad (22)$$

The mathematically strict derivation leads to a multiplier η_{hP}/η_{hM}. By omitting terms of second and higher order, the following equation can be applied. It shows negligible deviation from the strict formula but leads to the customary constant adder.

$$\Delta\eta_h = \eta_{hM}\left(\frac{\eta_{hP}}{\eta_{hM}} - 1\right) \cong \eta_{hM}\left(\Delta_E + \Delta_T + \Delta_Q\right) \tag{23}$$

In case of axial flow machines with homologous gaps, $\Delta_T = \Delta_Q = 0$. Then, the above formula is simplified to:

$$\frac{\eta_{hP}}{\eta_{hM}} = \frac{\eta_{EP}}{\eta_{EM}} = \left(1 + \Delta_E\right) \tag{24}$$

or

$$\Delta\eta_h = \eta_{hM} \times \Delta_E \tag{25}$$

In case the model hydraulic efficiency η_M is higher than "assumed maximum hydraulic efficiency: η_{hAmax}'", it is assumed that the standardized loss terms provided in this standard (d_{ECOref}, d_{Tref}, $1-\eta_{QM}$) are uniformly decreased by multiplying them by $(1-\eta_M)/(1-\eta_{hAmax})$. The attached Excel sheets give step-up values using thus modified loss terms. If these step-up values are used for contractual model tests, it shall be agreed on in advance among the concerned parties.

NOTE "Assumed maximum hydraulic efficiency: η_{hAmax}" is defined as the efficiency which is given by the values of δ_{Eref}, δ_{Tref} and volumetric efficiency η_Q given in this standard, assuming no kinetic loss is present.

$$\eta_{hAmax} = (1 - \delta_{Eref}) \times (1 - \delta_{Tref}) \times \eta_Q$$

6.3 Specific hydraulic energy

Under hydraulically homologous conditions, the specific hydraulic energy is converted by the following equations.

Turbine operation: (see Note in 6.6)

$$\frac{E_P}{E_M} = \left(\frac{n_P}{n_M}\right)^2 \times \left(\frac{D_P}{D_M}\right)^2 \times \left(\frac{\eta_{EM}}{\eta_{EP}}\right) = \left(\frac{n_P}{n_M}\right)^2 \times \left(\frac{D_P}{D_M}\right)^2 \times \left(\frac{1}{1+\Delta_E}\right) \tag{26}$$

Pump operation:

$$\frac{E_P}{E_M} = \left(\frac{n_P}{n_M}\right)^2 \times \left(\frac{D_P}{D_M}\right)^2 \times \left(\frac{\eta_{EP}}{\eta_{EM}}\right) = \left(\frac{n_P}{n_M}\right)^2 \times \left(\frac{D_P}{D_M}\right)^2 \times \left(1+\Delta_E\right) \tag{27}$$

6.4 Discharge

Under hydraulically homologous conditions, the discharge is converted by the following equations.

Turbine operation: (see Note in 6.6)

$$\frac{Q_{1P}}{Q_{1M}} = \frac{n_P}{n_M} \times \left(\frac{D_P}{D_M}\right)^3 \times \frac{\eta_{QM}}{\eta_{QP}} = \frac{n_P}{n_M} \times \left(\frac{D_P}{D_M}\right)^3 \times \left(\frac{1}{1+\Delta_Q}\right) \tag{28}$$

Pump operation:

$$\frac{Q_{1P}}{Q_{1M}} = \frac{n_P}{n_M} \times \left(\frac{D_P}{D_M}\right)^3 \times \frac{\eta_{QP}}{\eta_{QM}} = \frac{n_P}{n_M} \times \left(\frac{D_P}{D_M}\right)^3 \times (1+\Delta_Q) \tag{29}$$

6.5 Torque

Under hydraulically homologous conditions, the torque is converted by the following equations.

Turbine operation:

$$\frac{T_{mP}}{T_{mM}} = \frac{\rho_{1P}}{\rho_{1M}} \times \left(\frac{n_P}{n_M}\right)^2 \times \left(\frac{D_P}{D_M}\right)^5 \times \left(\frac{\eta_{TP}}{\eta_{TM}}\right) = \frac{\rho_{1P}}{\rho_{1M}} \times \left(\frac{n_P}{n_M}\right)^2 \times \left(\frac{D_P}{D_M}\right)^5 \times (1+\Delta_T) \tag{30}$$

Pump operation:

$$\frac{T_{mP}}{T_{mM}} = \frac{\rho_{1P}}{\rho_{1M}} \times \left(\frac{n_P}{n_M}\right)^2 \times \left(\frac{D_P}{D_M}\right)^5 \times \left(\frac{\eta_{TM}}{\eta_{TP}}\right) = \frac{\rho_{1P}}{\rho_{1M}} \times \left(\frac{n_P}{n_M}\right)^2 \times \left(\frac{D_P}{D_M}\right)^5 \times \left(\frac{1}{1+\Delta_T}\right) \tag{31}$$

6.6 Power

Under hydraulically homologous conditions, the power is converted by the following equations.

Turbine operation: (see Note in 6.6)

$$\frac{P_{mP}}{P_{mM}} = \frac{\rho_{1P}}{\rho_{1M}} \times \left(\frac{n_P}{n_M}\right)^3 \times \left(\frac{D_P}{D_M}\right)^5 \times \left(\frac{\eta_{TP}}{\eta_{TM}}\right) = \frac{\rho_{1P}}{\rho_{1M}} \times \left(\frac{n_P}{n_M}\right)^3 \times \left(\frac{D_P}{D_M}\right)^5 \times (1+\Delta_T) \tag{32}$$

Pump operation:

$$\frac{P_{mP}}{P_{mM}} = \frac{\rho_{1P}}{\rho_{1M}} \times \left(\frac{n_P}{n_M}\right)^3 \times \left(\frac{D_P}{D_M}\right)^5 \times \left(\frac{\eta_{TM}}{\eta_{TP}}\right) = \frac{\rho_{1P}}{\rho_{1M}} \times \left(\frac{n_P}{n_M}\right)^3 \times \left(\frac{D_P}{D_M}\right)^5 \times \left(\frac{1}{1+\Delta_T}\right) \tag{33}$$

NOTE In usual practice of the step-up calculation of turbine performance, firstly the value of n_{EDM} corresponding to the specified head for the prototype turbine E_P is calculated. Then, on the model performance curves, the values of η_{hM} and Q_{EDM} (and/or P_{EDM}) corresponding to this n_{EDM} are read.

In this procedure, the value of n_{EDM} should be calculated by the following formula considering the scale effect on E_P.

$$n_{EDM} = \frac{n_P \times D_P}{\sqrt{E_P}} \times \frac{1}{\sqrt{1+\Delta_E}}$$

Then the model values of η_{hM} and Q_{EDM} (and/or P_{EDM}) are converted to the prototype values.

For the conversion of η_{hM}, Equation 22 should be applied.

For the conversion of other performance parameters, such as Q_{EDM} and P_{EDM}, the following formulae should be used considering the scale effect.

$$Q_{1P} = Q_{1M}\left(\frac{D_P}{D_M}\right)^2\left(\frac{n_P D_P}{n_M D_M}\right)\frac{1}{1+\Delta_Q} = Q_{1M}\left(\frac{D_P}{D_M}\right)^2\left(\frac{E_P}{E_M}\right)^{0,5}\frac{(1+\Delta_E)^{0,5}}{1+\Delta_Q}$$

$$\therefore\ Q_{1P} = Q_{EDM}\times D_P{}^2\times E_P{}^{0,5}\frac{(1+\Delta_E)^{0,5}}{1+\Delta_Q}$$

$$P_{1P} = P_{1M}\left(\frac{\rho_{1P}}{\rho_{1M}}\right)\left(\frac{D_P}{D_M}\right)^2\left(\frac{n_P D_P}{n_M D_M}\right)^3(1+\Delta_T) = P_{1M}\left(\frac{\rho_{1P}}{\rho_{1M}}\right)\left(\frac{D_P}{D_M}\right)^2\left(\frac{E_P}{E_M}\right)^{1,5}(1+\Delta_E)^{1,5}(1+\Delta_T)$$

$$\therefore\ P_{1P} = P_{EDM}\times\rho_{1P}\times D_P{}^2\times E_P{}^{1,5}(1+\Delta_E)^{1,5}(1+\Delta_T)$$

6.7 Required input data

Required input data for the calculation of the prototype performance are itemized in Table 8.

Table 8 – Required input data for the calculation of the prototype performance

			Model	Prototype	Note
a) For conversion of the maximum efficiency point					
Operating Data	Reference diameter		D_M	D_P	
	Speed		n_{Mopt}	n_P	n_P: rated speed
	Discharge		Q_{1Mopt}	–	
	Specific hydraulic energy		E_{Mopt}	–	or H_{Mopt}
	Hydraulic efficiency		η_{Mopt}	–	
	Water temperature		t_{WM}	t_{WP}	
Data for step-up of η_E	Roughness	Spiral case	Ra_{SPM}	Ra_{SPP}	
		Stay vanes	Ra_{SVM}	Ra_{SVP}	
		Guide vanes	Ra_{GVM}	Ra_{GVP}	
		Runner blades	Ra_{RUM}	Ra_{RUP}	
		Draft tube	Ra_{DTM}	Ra_{DTP}	
Data for step-up of η_T	Roughness	Outer surface of runner	Ra_{TRM}	Ra_{TRP}	
		Stationary part facing to runner	Ra_{TSM}	Ra_{TSP}	
The dimensions below are required only if the runner seal geometry of the model is not homologous to the prototype.					
Data for correction of η_Q (when geometry of runner seal is not homologous)	Seal clearances[a]		C_{c1M}	C_{c1P}	Outer seal, crown side
			C_{c2M}	C_{c2P}	Inner seal, crown side
			C_{b1M}	C_{b1P}	Outer seal, band side
			C_{b2M}	C_{b2P}	Inner seal, band side
	Radius of seals[a]		R_{c1iM}	R_{c1iP}	Outer seal, crown side
			R_{c2iM}	R_{c2iP}	Inner seal, crown side
			R_{b1iM}	R_{b1iP}	Outer seal, band side
			R_{b2iM}	R_{b2iP}	Inner seal, band side
	Seal length[a]		L_{c1iM}	L_{c1iP}	Outer seal, crown side
			L_{c2iM}	L_{c2iP}	Inner seal, crown side
			L_{b1iM}	L_{b1iP}	Outer seal, band side
			L_{b2iM}	L_{b2iP}	Inner seal, band side

		Model	Prototype	Note
b) For conversion of turbine/pump performance				
General	Density of water	ρ_M	ρ_P	
For conversion of Turbine performance	Speed	n_{EDM}	n_P	n_P: rated speed
	Discharge	Q_{EDM}		
	Specific hydraulic energy		E_P	or H_P
	Power [a]	(P_{EDM})		
	Hydraulic efficiency	η_{hM}		
For conversion of Pump performance	Speed		n_P	n_P: rated speed
	Discharge	Q_{nDM}		
	Specific hydraulic energy	E_{nDM}	E_P	or H_P
	Power [a]	(P_{nDM}) [a]		
	Hydraulic efficiency	η_{hM}		
[a] If the prototype power is calculated from ρ_P, η_{hP}, E_P and Q_P, these model values for power are not necessary.				

7 Calculation procedure

Summarizing, the procedure how to scale up hydraulic model performance data to prototype conditions consists of the following steps:

step 1: Determination of the specific speed N_{QE} at the maximum efficiency point.

step 2: Calculation of scalable loss index d_{ECOref} and velocity index κ_{uCO} of each component corresponding to the N_{QE} obtained above.

step 3: Calculation of loss index d_{Tref} and dimension index κ_T for disc friction loss step-up.

step 4: Determination of surface quality expressed by Ra.

step 5: Determine the geometrical data for runner seals, if they are not homologous.

step 6: Calculation of individual step-ups $\left(\Delta_E = \frac{\Delta\eta_E}{\eta_{EM}}, \Delta_Q = \frac{\Delta\eta_Q}{\eta_{QM}}, \Delta_T = \frac{\Delta\eta_T}{\eta_{TM}}\right)$.

step 7: Calculation of prototype performance.

The attached flow chart represents the whole procedure starting from the calculation of specific speed to the calculation of prototype performance data. As demonstrated in this flow chart, the application of the new method is, despite the new features, still easy to handle.

By utilizing the Excel sheets attached to this standard, the step-up calculation can be done simply by entering the required input data into the relevant cells of input form.

Step 1 Calculation of specific speed at maximum efficiency point: $N_{QE} = \frac{nQ_1^{0,5}}{E^{0,75}}$.

Step 2 Calculation of scalable loss factors δ_{ECOref} and velocity factor κ_{uCO} of each component.

Step 3 Calculation of scalable disc friction loss index δ_{Tref} and dimension factor κ_T.

Step 4 Determination of roughness quality expressed by Ra.

Step 5 Determination of dimensions of seal geometry, if they are not homologous.

Step 6 Calculation of individual step-up values with Equations. 8 to 14:

Δ_E: Equations 8 to 10

Δ_T: Equation 12

Δ_Q: Equation 14

Step 7 Calculation of prototype performance with Equations 22 to 33. The example shown below is for turbine operation:

$$\frac{\eta_{hP}}{\eta_{hM}} = (1+\Delta_E)(1+\Delta_Q)(1+\Delta_T)$$

$$\frac{E_P}{E_M} = \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \left(\frac{1}{1+\Delta_E}\right) \qquad \frac{T_{mP}}{T_{mM}} = \frac{\rho_{1P}}{\rho_{1M}} \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^5 (1+\Delta_T)$$

$$\frac{Q_{1P}}{Q_{1M}} = \frac{n_P}{n_M} \times \left(\frac{D_P}{D_M}\right)^3 \left(\frac{1}{1+\Delta_Q}\right) \qquad \frac{P_{mP}}{P_{mM}} = \frac{\rho_{1P}}{\rho_{1M}} \left(\frac{n_P}{n_M}\right)^3 \left(\frac{D_P}{D_M}\right)^5 (1+\Delta_T)$$

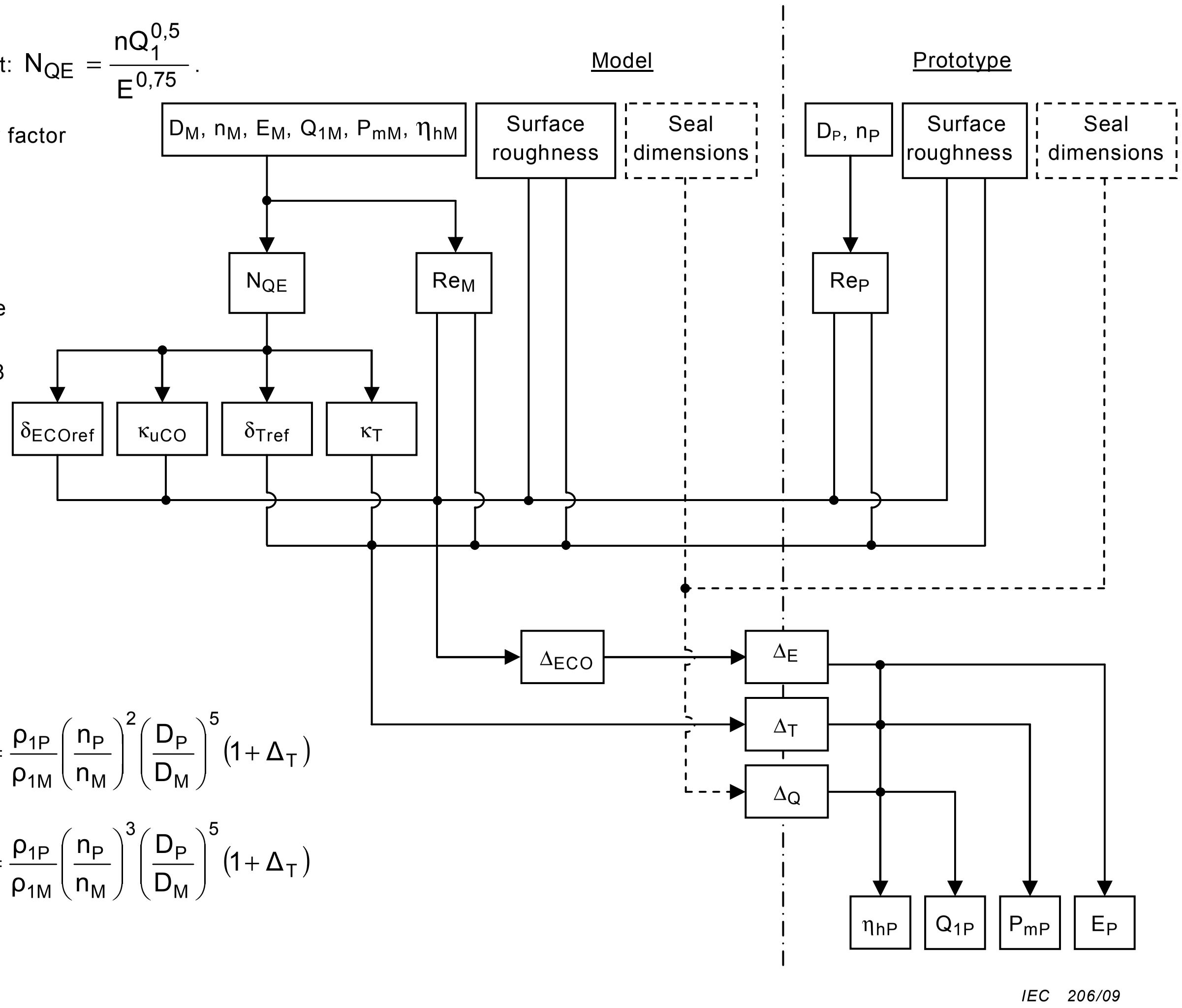


Figure 6 – Calculation steps of step-up values

Annex A
(informative)

Basic formulae and their approximation

A.1 Basic concept of loss structure and scale effect

The scale effect formulae set out in this standard are derived from the following basis.

1) Loss structure and efficiency components.

As illustrated in Figures A.1 and A.2, losses in hydraulic machines are classified into four component losses. (see Annex N of IEC 60193:1999 [1], [8], [10])

They are:

- specific hydraulic energy loss: E_L;
- leakage flow loss: q;
- disk friction loss: P_{Ld} ;
- bearing friction loss: P_{Lm}.

Corresponding to each loss, the following efficiency components are defined.

- specific hydraulic energy efficiency: η_E;
- volumetric efficiency: η_Q;
- power efficiency: η_T;
- mechanical efficiency: η_m.

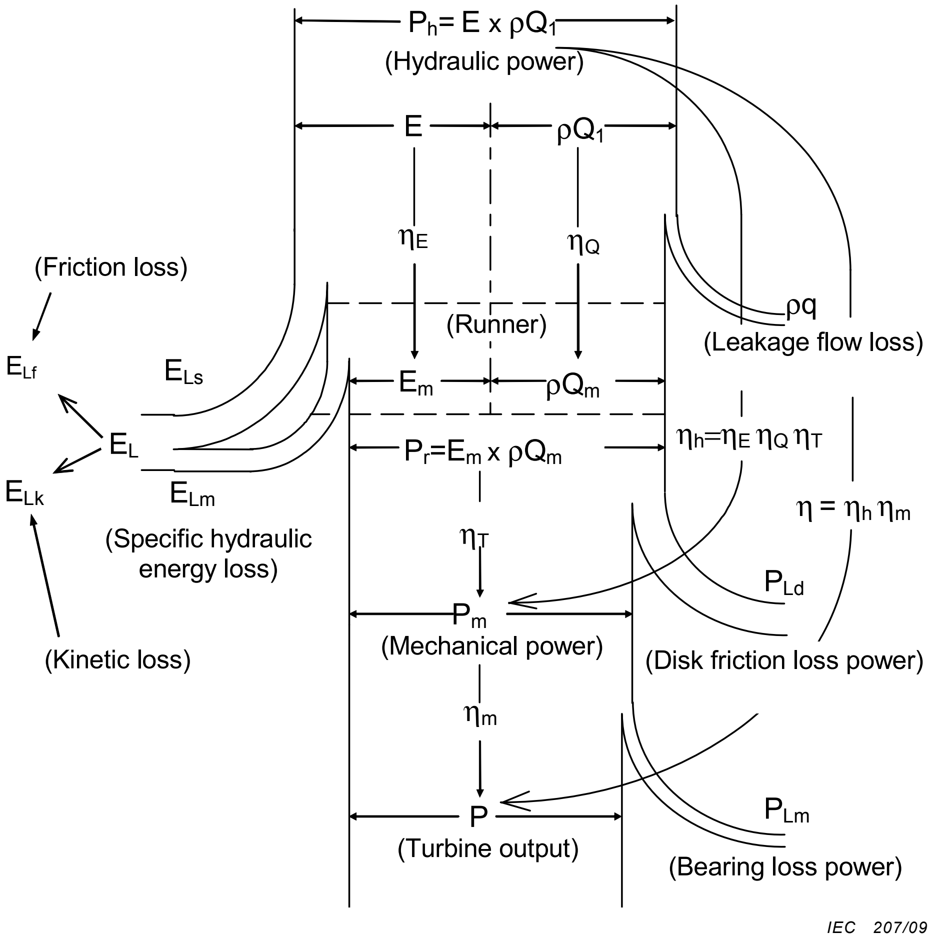


Figure A.1 – Flux diagram for a turbine

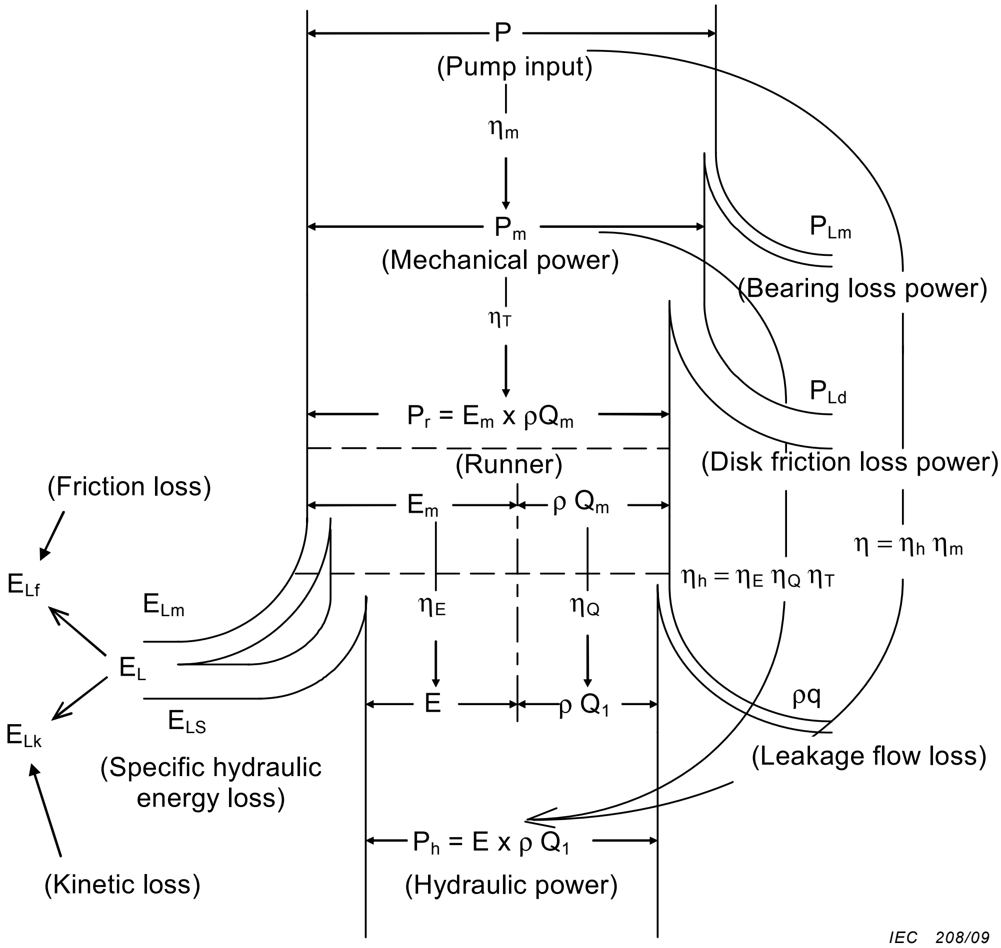


Figure A.2 – Flux diagram for a pump

The ratio of $\frac{P_m}{P_h}$ (for turbine) or $\frac{P_h}{P_m}$ (for pump) is defined as hydraulic efficiency η_h, which is expressed as the product of η_E, η_Q and η_T.

This standard deals with the scale effect on the hydraulic efficiency η_h and the mechanical efficiency η_m is excluded from the topic of this standard.

2) Homologous operating condition

Homologous operating condition of the runner/impeller between a model and a prototype can be achieved when the velocity triangles at both inlet and outlet of the runner/impeller are homologous. However, strictly speaking, homology of both the inlet and the outlet velocity triangles cannot be maintained simultaneously due to the scale effect on the internal flow in the runner/impeller. According to the theoretical assessment, it has been proved that, if the homology of the velocity triangle at the high pressure side of the runner/impeller is maintained, the deviation of the velocity triangle at its low pressure side is very minor and it does not affect its performance significantly. Therefore, it is considered in this standard that the homologous operating condition between model and prototype can be achieved when the homology of the velocity triangle at the high pressure side of the runner/impeller is maintained [2]. In case that such homologous operating condition is maintained between model and prototype, the performance parameters of the runner/impeller, E_m, Q_m and P_r can be converted by hydraulic similarity law as shown below without any shifting due to the scale effect.

$$E_{mP} = \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 E_{mM}\,,\; Q_{mP} = \left(\frac{n_P}{n_M}\right)\left(\frac{D_P}{D_M}\right)^3 Q_{mM} \text{ and } P_{rP} = \left(\frac{n_P}{n_M}\right)^3 \left(\frac{D_P}{D_M}\right)^5 P_{rM} \qquad \text{(A.1)}$$

3) Shifting of performance [7]

When η_E, η_Q and η_T of the prototype differ from those of the model due to the scale effect, the performance parameters of the prototype can be calculated by the following formulae considering that E_m, Q_m and P_r are homologous between model and prototype.

For turbines:

$$E_{mP} = \eta_{EP} E_P \quad \text{and} \quad E_{mM} = \eta_{EM} E_M$$

$$\therefore \quad E_P = \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \left(\frac{\eta_{EM}}{\eta_{EP}}\right) E_M = \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \left(\frac{\eta_{EM}}{\eta_{EM} + \Delta\eta_E}\right) E_M$$

$$= \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \left(\frac{1}{1 + \dfrac{\Delta\eta_E}{\eta_{EM}}}\right) E_M = \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \left(\frac{1}{1 + \Delta_E}\right) E_M \qquad \text{(A.2)}$$

$$Q_{mP} = \eta_{QP} Q_{1P} \quad \text{and} \quad Q_{mM} = \eta_{QM} Q_{1M}$$

$$\therefore \quad Q_{1P} = \left(\frac{n_P}{n_M}\right)\left(\frac{D_P}{D_M}\right)^3 \left(\frac{\eta_{QM}}{\eta_{QP}}\right) Q_{1M} = \left(\frac{n_P}{n_M}\right)\left(\frac{D_P}{D_M}\right)^3 \left(\frac{\eta_{QM}}{\eta_{QM} + \Delta\eta_Q}\right) Q_{1M}$$

$$= \left(\frac{n_P}{n_M}\right)\left(\frac{D_P}{D_M}\right)^3 \left(\frac{1}{1 + \dfrac{\Delta\eta_Q}{\eta_{QM}}}\right) Q_{1M} = \left(\frac{n_P}{n_M}\right)\left(\frac{D_P}{D_M}\right)^3 \left(\frac{1}{1 + \Delta_Q}\right) Q_{1M} \qquad \text{(A.3)}$$

$$P_{rP} = \frac{P_{mP}}{\eta_{TP}} \quad \text{and} \quad P_{rM} = \frac{P_{mM}}{\eta_{TM}}$$

$$\therefore \quad P_{mP} = \left(\frac{n_P}{n_M}\right)^3 \left(\frac{D_P}{D_M}\right)^5 \left(\frac{\eta_{TP}}{\eta_{TM}}\right) P_{mM} = \left(\frac{n_P}{n_M}\right)^3 \left(\frac{D_P}{D_M}\right)^5 \left(\frac{\eta_{TM} + \Delta\eta_T}{\eta_{TM}}\right) P_{mM}$$

$$= \left(\frac{n_P}{n_M}\right)^3 \left(\frac{D_P}{D_M}\right)^5 \left(1 + \frac{\Delta\eta_T}{\eta_{TM}}\right) P_{mM} = \left(\frac{n_P}{n_M}\right)^3 \left(\frac{D_P}{D_M}\right)^5 (1 + \Delta_T) P_{mM} \qquad \text{(A.4)}$$

For pumps:

$$E_{mP} = \frac{E_P}{\eta_{EP}} \quad \text{and} \quad E_{mM} = \frac{E_M}{\eta_{EM}}$$

$$\therefore \quad E_P = \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \left(\frac{\eta_{EP}}{\eta_{EM}}\right) E_M = \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \left(\frac{\eta_{EM} + \Delta\eta_E}{\eta_{EM}}\right) E_M$$

$$= \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \left(1 + \frac{\Delta\eta_E}{\eta_{EM}}\right) E_M = \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 (1 + \Delta_E) E_M \qquad \text{(A.5)}$$

$$Q_{mP} = \frac{Q_{1P}}{\eta_{QP}} \quad \text{and} \quad Q_{mM} = \frac{Q_{1M}}{\eta_{QM}}$$

$$\therefore \; Q_{1P} = \left(\frac{n_P}{n_M}\right)\left(\frac{D_P}{D_M}\right)^3\left(\frac{\eta_{QP}}{\eta_{QM}}\right)Q_{1M} = \left(\frac{n_P}{n_M}\right)\left(\frac{D_P}{D_M}\right)^3\left(\frac{\eta_{QM}+\Delta\eta_Q}{\eta_{QM}}\right)Q_{1M}$$

$$= \left(\frac{n_P}{n_M}\right)\left(\frac{D_P}{D_M}\right)^3\left(1+\frac{\Delta\eta_Q}{\eta_{QM}}\right)Q_{1M} = \left(\frac{n_P}{n_M}\right)\left(\frac{D_P}{D_M}\right)^3(1+\Delta_Q)Q_{1M} \qquad \text{(A.6)}$$

$$P_{rP} = \eta_{TP}P_{mP} \quad \text{and} \quad P_{rM} = \eta_{TM}P_{mM}$$

$$\therefore \; P_{mP} = \left(\frac{n_P}{n_M}\right)^3\left(\frac{D_P}{D_M}\right)^5\left(\frac{\eta_{TM}}{\eta_{TP}}\right)P_{mM} = \left(\frac{n_P}{n_M}\right)^3\left(\frac{D_P}{D_M}\right)^5\left(\frac{\eta_{TM}}{\eta_{TM}+\Delta\eta_T}\right)P_{mM}$$

$$= \left(\frac{n_P}{n_M}\right)^3\left(\frac{D_P}{D_M}\right)^5\left(\frac{1}{1+\frac{\Delta\eta_T}{\eta_{TM}}}\right)P_{mM} = \left(\frac{n_P}{n_M}\right)^3\left(\frac{D_P}{D_M}\right)^5\left(\frac{1}{1+\Delta_T}\right)P_{mM} \qquad \text{(A.7)}$$

Scale effect on the performance at off-design points is complicated. In this standard, however, it is prescribed that the performance at off-design points shall be calculated in the same manner by using Equations A.2 to A.7 with Δ_E, Δ_Q and Δ_T obtained for the maximum efficiency point.

4) Scalable losses

As stated in 4.1.1, the following losses are subject to scale effect by the difference of Reynolds number and the relative roughness.

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

In the past international standards, all scalable losses were dealt with collectively. The magnitude of the scalable loss was estimated by the assumption that its ratio over the total loss, which had been denoted as V, could be given as a certain constant value for each type of hydraulic machines. This assumption gave larger value of the scalable loss to low efficiency turbines and, as a result, it gave unreasonably high efficiency step-up for them.

In this standard, however, it is considered 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 the machine. This enables to calculate the scale effect on each efficiency component individually and to calculate the shifting of each performance parameter as stated in 3) above.

A.2 Derivation of the scale effect formulae and the approximation introduced for simplifications

1) Scalable loss ratio in specific hydraulic energy δ_E and specific hydraulic energy efficiency η_E

Similarly to the conventional IEC standard, the relative scalable loss δ_E and the relative non-scalable loss δ_{ns} are defined. The relations among these values and the specific hydraulic energy efficiency η_E are shown below. It should be noted that the homologous quantity which is directly transposable from the model to the prototype by the hydraulic similarity law is E_m but not E. To explain simply the derivation of the formulae, new parameters, δ_E^* and δ_{ns}^*, defined by using E_m are introduced in the table shown below.

	Turbine	**Pump**
Definition of η_E	$\eta_E = \frac{E_m}{E} = \frac{E-\left(\sum E_{Lf} + \sum E_{Lk}\right)}{E}$	$\eta_E = \frac{E}{E_m} = \frac{E}{E+\left(\sum E_{Lf} + \sum E_{Lk}\right)}$
Definition of δ_E and δ_{ns}	$\delta_E = \frac{\sum E_{Lf}}{E} = \eta_E \frac{\sum E_{Lf}}{E_m}$ $\delta_{ns} = \frac{\sum E_{Lk}}{E} = \eta_E \frac{\sum E_{Lk}}{E_m}$	$\delta_E = \frac{\sum E_{Lf}}{E} = \frac{1}{\eta_E}\frac{\sum E_{Lf}}{E_m}$ $\delta_{ns} = \frac{\sum E_{Lf}}{E} = \frac{1}{\eta_E}\frac{\sum E_{Lf}}{E_m}$
New definition of δ_E^* and δ_{ns}^*	$\delta_E^* = \frac{\sum E_{Lf}}{E_m}$ Since E_m is homologous, δ_E^* can be scaled up by the ratio of the loss coefficient. $\left(\delta_{EP}^*/\delta_{EM}^*\right) = \left(\lambda_P/\lambda_M\right)$ $\delta_{ns}^* = \frac{\sum E_{Lk}}{E_m}$ Since E_m is homologous, δ_{ns}^* remains constant for both model and prototype: $\delta_{nsP}^* = \delta_{nsM}^*$	
Relationship between δ^* and conventional δ	$\delta_E = \eta_E \delta_E^*$ and $\delta_{ns} = \eta_E \delta_{ns}^*$	$\delta_E = \frac{\delta_E^*}{\eta_E}$ and $\delta_{ns} = \frac{\delta_{ns}^*}{\eta_E}$
Shifting of δ_{ns}	$\frac{\delta_{nsP}}{\eta_{EP}} = \frac{\delta_{nsM}}{\eta_{EM}}$	$\eta_{EP}\delta_{nsP} = \eta_{EM}\delta_{nsM}$
New expression of η_E using δ_E^* and δ_{ns}^*	$\eta_E = \frac{E_m}{E} = \frac{E-\left(\sum E_{Lf} + \sum E_{Lk}\right)}{E}$ $= 1 - \delta_E - \delta_{ns}$ $= 1 - \eta_E\left(\frac{\sum E_{Lf} + \sum E_{Lk}}{E_m}\right)$ $= 1 - \eta_E\left(\delta_E^* + \delta_{ns}^*\right)$ $\left(* \quad \therefore \delta_E = 1 - \eta_E - \delta_{ns}\right)$ □ $\eta_E = \frac{1}{1+\delta_E^* + \delta_{ns}^*}$	$\eta_E = \frac{E}{E_m} = \frac{E}{E+\left(\sum E_{Lf} + \sum E_{Lk}\right)}$ $= \frac{1}{1+\delta_E + \delta_{ns}}$ $= \frac{\eta_E}{\eta_E + \left(\frac{\sum E_{Lf} + \sum E_{Lk}}{E_m}\right)}$ $= \frac{1}{1+\frac{1}{\eta_E}\left(\delta_E^* + \delta_{ns}^*\right)}$ $\left(* \quad \therefore \delta_E = \frac{1}{\eta_E} - 1 - \delta_{ns}\right)$ □ $\eta_E = 1 - \delta_E^* - \delta_{ns}^*$
δ_E^* is stepped up by the ratio of friction coefficient. δ_{ns}^* remains constant for both model and prototype.		

2) Step-up of specific hydraulic energy efficiency η_E

As shown in A.2 1), η_E is expressed by different equations for turbines and pumps. This is caused by the difference in the term η_E; for turbines, E_m appears at the numerator and for pumps, E_m appears at denominator. Also, it should be noted that the non- scalable loss δ^*_{ns} is a common value for both model and prototype but δ_{ns} is not, and that the scalable loss δ^*_E can be scaled up by the ratio of the loss coefficient from model to prototype but δ_E can not.

Hence the following scale effect formulae for η_E can be derived:

	Turbine	Pump
$\Delta\eta_E$ calculated by using δ^*_E and δ^*_{ns}	$\Delta\eta_E = \eta_{EP} - \eta_{EM}$ $= \frac{1}{1+\delta^*_{EP}+\delta^*_{nsP}} - \frac{1}{1+\delta^*_{EM}+\delta^*_{nsM}}$ $= \frac{\left(\delta^*_{EM}-\delta^*_{EP}\right)+\left(\delta^*_{nsM}-\delta^*_{nsP}\right)}{\left(1/\eta_{EP}\right)\left(1/\eta_{EM}\right)}$ since $\left(\delta^*_{nsM}-\delta^*_{nsP}\right)=0$ $\Delta\eta_E = \eta_{EP}\eta_{EM}\left(\delta^*_{EM}-\delta^*_{EP}\right)$	$\Delta\eta_E = \eta_{EP} - \eta_{EM}$ $= \left(\delta^*_{EM}+\delta^*_{nsM}\right)-\left(\delta^*_{EP}+\delta^*_{nsP}\right)$ $= \left(\delta^*_{EM}-\delta^*_{EP}\right)+\left(\delta^*_{nsM}-\delta^*_{nsP}\right)$ since $\left(\delta^*_{nsM}-\delta^*_{nsP}\right)=0$ $\Delta\eta_E = \delta^*_{EM}-\delta^*_{EP}$
Conversion of friction loss	$\delta^*_{EP} = \delta^*_{Eref}\frac{\lambda_P}{\lambda_{ref}}$ $\delta^*_{EM} = \delta^*_{Eref}\frac{\lambda_M}{\lambda_{ref}}$	where: λ is friction loss coefficient
$\Delta\eta_E$ referring to δ^*_{Eref}	$\Delta\eta_E = \eta_{EP}\eta_{EM}\delta^*_{Eref}\left(\frac{\lambda_M}{\lambda_{ref}}-\frac{\lambda_P}{\lambda_{ref}}\right)$	$\Delta\eta_E = \delta^*_{Eref}\left(\frac{\lambda_M}{\lambda_{ref}}-\frac{\lambda_P}{\lambda_{ref}}\right)$
$\Delta\eta_E$ referring to δ_{Eref}	since $\delta^*_{Eref} = \frac{\delta_{Eref}}{\eta_{Eref}}$ $\Delta\eta_E = \frac{\eta_{EP}\eta_{EM}}{\eta_{Eref}}\delta_{Eref}\left(\frac{\lambda_M}{\lambda_{ref}}-\frac{\lambda_P}{\lambda_{ref}}\right)$ $\therefore \frac{\Delta\eta_E}{\eta_{EM}} = \frac{\eta_{EP}}{\eta_{Eref}}\delta_{Eref}\left(\frac{\lambda_M-\lambda_P}{\lambda_{ref}}\right)$	since $\delta^*_{Eref} = \eta_{Eref}\delta_{Eref}$ $\Delta\eta_E = \eta_{Eref}\delta_{Eref}\left(\frac{\lambda_M}{\lambda_{ref}}-\frac{\lambda_P}{\lambda_{ref}}\right)$ $\therefore \frac{\Delta\eta_E}{\eta_{EM}} = \frac{\eta_{Eref}}{\eta_{EM}}\delta_{Eref}\left(\frac{\lambda_M-\lambda_P}{\lambda_{ref}}\right)$
Approximation formula given in this standard	since $\frac{\eta_{EP}}{\eta_{Eref}} \approx 1$ $\Delta_E = \frac{\Delta\eta_E}{\eta_{EM}} \approx \delta_{Eref}\left(\frac{\lambda_M-\lambda_P}{\lambda_{ref}}\right)$	since $\frac{\eta_{Eref}}{\eta_{EM}} \approx 1$ $\Delta_E = \frac{\Delta\eta_E}{\eta_{EM}} \approx \delta_{Eref}\left(\frac{\lambda_M-\lambda_P}{\lambda_{ref}}\right)$

It should be noted that the equation to obtain $\Delta\eta_E$ is different for turbines and pumps. However, by introducing the approximation given in the lowest frames of the above table, the same formula is used for both turbines and pumps in this standard.

3) Step-up of volumetric efficiency η_Q

Similar to η_E, the equation of η_Q is expressed differently for turbines and pumps. Since the quantity that is directly transposable to the prototype is Q_m (not Q_1), the ratio of the leakage loss q over Q_m is expressed as shown below and the step-up amount of volumetric efficiency $\Delta\eta_Q$ is obtained.

	Turbine	Pump
Definition of η_Q	$\eta_Q = \frac{Q_m}{Q_1} = \frac{Q_m}{Q_m + q} = \frac{1}{1+\frac{q}{Q_m}}$	$\eta_Q = \frac{Q_1}{Q_m} = \frac{Q_m - q}{Q_m} = 1 - \frac{q}{Q_m}$
Conversion of leakage loss q	$\frac{q_P}{q_M} = \left(\frac{\zeta_{kM}+\zeta_{fM}}{\zeta_{kP}+\zeta_{fP}}\right)^{0,5}\left(\frac{A_P/D_P^2}{A_M/D_M^2}\right)\left(\frac{D_P}{D_M}\right)^2\left(\frac{E_{mP}}{E_{mM}}\right)^{0,5}$ $= \left(\frac{\zeta_k+\zeta_{fM}}{\zeta_k+\zeta_{fP}}\right)^{0,5}\left(\frac{A_P/D_P^2}{A_M/D_M^2}\right)\frac{Q_{mP}}{Q_{mM}}$ $\therefore \frac{q_P}{Q_{mP}} = \left(\frac{\zeta_k+\zeta_{fM}}{\zeta_k+\zeta_{fP}}\right)^{0,5}\left(\frac{A_P/D_P^2}{A_M/D_M^2}\right)\frac{q_M}{Q_{mM}}$ where ζ_k loss coefficient due to the kinetic loss for the leakage flow through the seal clearance (non-scalable) ζ_f loss coefficient due to the friction loss for the leakage flow through the seal clearance (scalable) A cross sectional area of seal clearance when the dimensions of the seal and the concerned parts are geometrically homologous, $\left(\frac{A_P/D_P^2}{A_M/D_M^2}\right) = 1$ $\therefore \frac{q_P}{Q_{mP}} = \left(\frac{\zeta_k+\zeta_{fM}}{\zeta_k+\zeta_{fP}}\right)^{0,5}\frac{q_M}{Q_{mM}}$	
$\Delta\eta_Q$ referring to q and Q_m (when the geometry of the seal is homologous)	$\Delta\eta_Q = \eta_{QP} - \eta_{QM}$ $= \frac{1}{1+(q_P/Q_{mP})} - \frac{1}{1+(q_M/Q_{mM})}$ $= \eta_{QP}\eta_{QM}\{(q_M/Q_{mM}) - (q_P/Q_{mP})\}$ $= \eta_{QP}(1-\eta_{QM})\left\{1-\left(\frac{\zeta_k+\zeta_{fM}}{\zeta_k+\zeta_{fP}}\right)^{0,5}\right\}$	$\Delta\eta_Q = \eta_{QP} - \eta_{QM}$ $= \frac{q_M}{Q_{mM}} - \frac{q_P}{Q_{mP}}$ $= (1-\eta_{QM})\left\{1-\left(\frac{\zeta_k+\zeta_{fM}}{\zeta_k+\zeta_{fP}}\right)^{0,5}\right\}$
Approximation formula given in this standard	Since both $(1-\eta_{QM})$ and $\left\{1-\left(\frac{\zeta_k+\zeta_{fM}}{\zeta_k+\zeta_{fP}}\right)^{0,5}\right\}$ are very small values, the step-up amount of the volumetric efficiency (negative) can be usually disregarded. $\therefore \Delta\eta_Q = 0$ (see Annex E)	

4) Step-up of power efficiency (disc friction) η_T

In this case, the power of the runner/impeller P_r is transposable directly from model to prototype by the hydraulic similarity law (not P_m). Then, the scalable disc friction loss δ_T, which is defined as $\frac{P_{Ld}}{P_m}$, is stepped up by the following formulae.

	Turbine	Pump
Definition of η_T	$\eta_T = \frac{P_m}{P_r} = \frac{P_r - P_{Ld}}{P_r} = 1 - \frac{P_{Ld}}{P_r}$	$\eta_T = \frac{P_r}{P_m} = \frac{P_r}{P_r + P_{Ld}} = \frac{1}{1 + \frac{P_{Ld}}{P_r}}$
Homologous condition	Since P_r is homologous between model and prototype, $P_{rM} = P_{rref}\left(\frac{n_M}{n_{ref}}\right)^3\left(\frac{D_M}{D_{ref}}\right)^5$, $P_{rP} = P_{rref}\left(\frac{n_P}{n_{ref}}\right)^3\left(\frac{D_P}{D_{ref}}\right)^5$	
Definition of δ_T	$\delta_T = \frac{P_{Ld}}{P_m} = \frac{1}{\eta_T}\frac{P_{Ld}}{P_r}$	$\delta_T = \frac{P_{Ld}}{P_m} = \eta_T \frac{P_{Ld}}{P_r}$
Expression of $\Delta\eta_T$	$\Delta\eta_T = \eta_{TP} - \eta_{TM}$ $= \eta_{TM}\delta_{TM} - \eta_{TP}\delta_{TP}$ $= \frac{P_{LdM}}{P_{rM}} - \frac{P_{LdP}}{P_{rP}}$	$\Delta\eta_T = \eta_{TP} - \eta_{TM}$ $= \eta_{TM}\eta_{TP}\left(\frac{\delta_{TM}}{\eta_{TM}} - \frac{\delta_{TP}}{\eta_{TP}}\right)$ $= \eta_{TM}\eta_{TP}\left(\frac{P_{LdM}}{P_{rM}} - \frac{P_{LdP}}{P_{rP}}\right)$
Step-up of η_T	Since 1) both P_{LdM} and P_{LdP} are proportional to the loss coefficient and 2) P_r is homologous between model and prototype, the following equations are derived. $P_{LdM} = P_{Ldref}\frac{C_{mM}}{C_{mref}}\left(\frac{n_M}{n_{ref}}\right)^3\left(\frac{D_M}{D_{ref}}\right)^5$ $P_{r\,M} = P_{r\,ref}\left(\frac{n_M}{n_{ref}}\right)^3\left(\frac{D_M}{D_{ref}}\right)^5$ $P_{LdP} = P_{Ldref}\frac{C_{mP}}{C_{mref}}\left(\frac{n_P}{n_{ref}}\right)^3\left(\frac{D_P}{D_{ref}}\right)^5$ $P_{r\,P} = P_{r\,ref}\left(\frac{n_P}{n_{ref}}\right)^3\left(\frac{D_P}{D_{ref}}\right)^5$	
	$\Delta\eta_T = \left(\frac{P_{Ld}}{P_r}\right)_{ref}\left(\frac{C_{mM}}{C_{mref}} - \frac{C_{mP}}{C_{mref}}\right)$ $= \eta_{Tref}\delta_{Tref}\left(\frac{C_{mM} - C_{mP}}{C_{mref}}\right)$ $\therefore \frac{\Delta\eta_T}{\eta_{TM}} = \frac{\eta_{Tref}}{\eta_{TM}}\delta_{Tref}\left(\frac{C_{mM} - C_{mP}}{C_{mref}}\right)$	$\Delta\eta_T = \eta_{TM}\eta_{TP}\left(\frac{P_{Ld}}{P_r}\right)_{ref}\left(\frac{C_{mM}}{C_{mref}} - \frac{C_{mP}}{C_{mref}}\right)$ $= \eta_{TM}\eta_{TP}\frac{\delta_{Tref}}{\eta_{Tref}}\left(\frac{C_{mM} - C_{mP}}{C_{mref}}\right)$ $\therefore \frac{\Delta\eta_T}{\eta_{TM}} = \frac{\eta_{TP}}{\eta_{Tref}}\delta_{Tref}\left(\frac{C_{mM} - C_{mP}}{C_{mref}}\right)$
Approximation formula given in this standard	since $\frac{\eta_{Tref}}{\eta_{TM}} \approx 1$ $\Delta_T = \frac{\Delta\eta_T}{\eta_{TM}} \approx \delta_{Tref}\left(\frac{C_{mM} - C_{mP}}{C_{mref}}\right)$	since $\frac{\eta_{TP}}{\eta_{Tref}} \approx 1$ $\Delta_T = \frac{\Delta\eta_T}{\eta_{TM}} \approx \delta_{Tref}\left(\frac{C_{mM} - C_{mP}}{C_{mref}}\right)$

As shown in the above table, the formula to obtain $\Delta\eta_T$ should be different for turbines and pumps. However, by introducing the approximation given in the lowest frames of the above table, a common formula is used in this standard for both turbines and pumps.

Annex B
(informative)

Scale effect on specific hydraulic energy losses of radial flow machines

B.1 Scale effect on friction loss

1) Scale effect on friction loss coefficient

The scale effect, that is the variation of the friction loss caused by the difference in Reynolds number and the relative roughness, is slightly different for a flat plate and for a pipe. However, it is prescribed in this standard that the friction loss coefficient in various passages of the machine, excluding runner blades of axial flow machines, varies according to Colebrook formula established for pipe flow.

Since the original Colebrook formula is given as an implicit function (see Figure B.1), it is not easy to obtain the value of the loss coefficient by a simple calculation. Therefore, in this standard, a new formula proposed by Nichtawitz, which is an explicit function to give almost the same values as of the Colebrook formula, is used. [4, 6]

The new formula is:

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

where

$Re_0 = 7 \times 10^6$;

$\lambda_0 = 0{,}008\,5$;

k_s sand roughness;

d_h hydraulic diameter of a pipe / conduit / water passage;

Re_d Reynolds number in a pipe, $Re_d = \frac{v d_h}{\nu}$.

The comparison between the original Colebrook formula and the new formula is shown in Figure B.1.

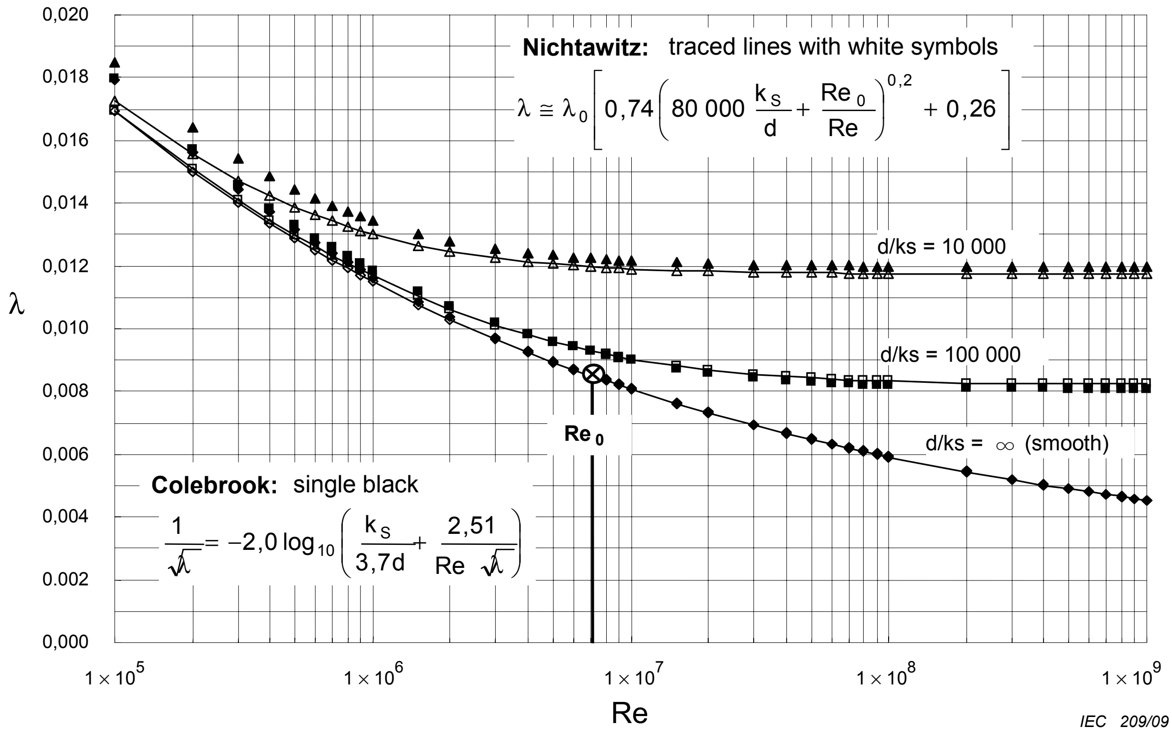


Figure B.1 – Loss coefficient versus Reynolds number and surface roughness

NOTE 1 In some experiments with sand roughness, it is observed that the friction loss of a rough surface having a roughness within a certain value is the same as a completely smooth surface. In such a case, the limit of the roughness is called "admissible roughness" and the surface with the roughness within this limit is regarded as "hydraulically smooth". (see curves B and C in Figure B.2).

Regarding the loss coefficient of rough surfaces, some different experimental results shown in Figure B.2 were reported in the past, in which the characteristics of the friction loss coefficient showed different trend in the transition zone between smooth and rough categories. [13 - 16]

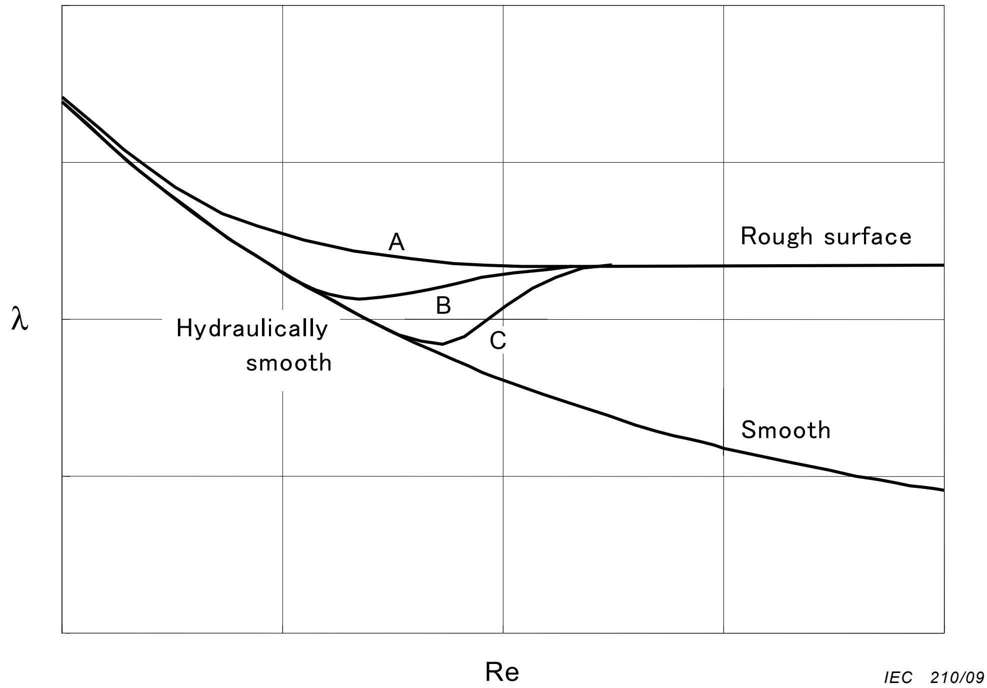


Figure B.2 – Different characteristics of λ in transition zone

The curve "A" is observed in the experiments with commercially rough pipe (Moody) or rough model turbine (Henry). [17] They show that the admissible roughness is very small and the friction loss characteristics show an asymptotic curve. Colebrook formula represents such characteristics. The admissible roughness in such case is nearly zero.

The curve "B" represents the experimental results with sand roughness (Nikuradse). In such case, the admissible roughness is given as approximately

$$\frac{k_{Sadm}}{d_h} \approx \frac{5}{Re_d}\sqrt{\frac{8}{\lambda}}$$

The characteristics like the curve "C" is observed in the experiments with a corrugated surface or a surface with isolated sharp sand grains. In such case, the admissible roughness becomes larger.

It is considered in this standard that the admissible roughness is very small and Colebrook formula may apply for the assessment of the scale effect on the friction loss.

2) Relationship between sand roughness k_S and arithmetical mean roughness Ra

The relationship between the sand roughness k_S and the arithmetical mean roughness Ra presently available in the literature is widely spread. [14] In this standard, however, it is considered that the arithmetical mean roughness can be converted to the sand roughness by the following equation (see Equation 2).

$$k_S = 5\,Ra \tag{B.2}$$

NOTE 2 In case of aged prototype machines with heavily rusted surfaces with Ra values larger than 50 µm, it is recommended to consider the followings in evaluating the surface roughness.

First is the difficulty in the measurement of roughness. On aged machines, the roughness values are often beyond any existing portable roughness tester range. In these situations, it is recommended to take molds at most representative locations using appropriate plastic material and measure the roughness of these molds by using a 'coordinate measuring machine' to find an equivalent Ra value. Other methods can also be used (like depth indicators, or roughness comparison coupons, etc) if a mutual agreement is reached among the concerned parties. In such a situation, however, the equivalent Ra roughness should be determined carefully, as it is affected by the roughness profile and the density of dispersed voids.

Secondly, a consideration should be taken in choosing the meaningful roughness values from the measurements. Based on the actual state of knowledge, it is believed that areas having scattered deep voids do not create as much losses as their measured value would indicate. Indeed, the stream lines over such areas pass over the voids without reaching the bottom and do not create significantly larger losses. Therefore, in such case, it is recommended to ignore areas with deep voids when measuring roughness (deep voids are considered as being depressions deeper than approximately 1,5 mm).

Once the above considerations have been taken into account, the relationship between k_S and Ra as expressed by Equation B.2 (or Equation 2) can be tentatively applied also to heavily rusted surfaces.

Then, Equation B.1 is expressed as follows:

$$\lambda = \lambda_0 \left[0{,}74 \left(4 \times 10^5 \frac{Ra}{d_h} + \frac{Re_0}{Re_d} \right)^{0{,}2} + 0{,}26 \right] \tag{B.3}$$

B.2 Componentwise step-up of specific hydraulic energy efficiency

1) Friction loss coefficient of each component [9]

When Equation B.3 is applied to each component passage, we obtain,

$$\lambda_{CO} = \lambda_0 \left[0{,}74 \left(4 \times 10^5 \frac{Ra_{CO}}{d_{hCO}} + \frac{Re_0}{Re_{dCO}} \right)^{0{,}2} + 0{,}26 \right] \tag{B.4}$$

where

subscript CO the values for each component passage;

Re_{dCO} Reynolds number for each component passage.

$$Re_{dCO} = \frac{v_{CO} d_{hCO}}{\nu}$$

Since the Reynolds number for the machine can be written as:

$$Re = \frac{uD}{\nu}$$

where

u peripheral velocity of the runner/impeller at the reference diameter;

D reference diameter of the machine.

The Reynolds number for the component passage can be expressed as follows:

$$Re_{dCO} = Re\frac{v_{CO}d_{hCO}}{uD}$$

By substituting Re_{dCO} in Equation B.4 by the above equation, we obtain:

$$\lambda_{CO} = \lambda_0\left[0{,}74\left(4\times10^5\frac{D}{d_{hCO}}\frac{Ra_{CO}}{D}+\frac{u\times D}{v_{CO}\times d_{hCO}}\frac{Re_0}{Re}\right)^{0,2}+0{,}26\right] \quad \text{(B.5)}$$

By introducing two new factors, κ_{dCO} and κ_{uCO}, Equation B.5 can be rewritten as follows:

$$\lambda_{CO} = \lambda_0\left[0{,}74\left(4\times10^5\frac{1}{\kappa_{dCO}}\frac{Ra_{CO}}{D}+\frac{1}{\kappa_{uCO}\times\kappa_{dCO}}\frac{Re_0}{Re}\right)^{0,2}+0{,}26\right] \quad \text{(B.6)}$$

where

κ_{dCO} dimension factor of component passage.

$$\kappa_{dCO} = \frac{d_{hCOM}}{D_M} = \frac{d_{hCOP}}{D_P} = \frac{d_{hCO}}{D} \quad \text{(B.7)}$$

where

κ_{uCO} flow velocity factor of component passage.

$$\kappa_{uCO} = \frac{v_{COM}}{u_M} = \frac{v_{COP}}{u_P} = \frac{v_{CO}}{u} \quad \text{(B.8)}$$

When the geometrical dimensions of the principal water passages as shown in Figure B.3 are given, the values of κ_{dCO} and κ_{uCO} can be calculated by Equations B.9 and B.10, respectively.

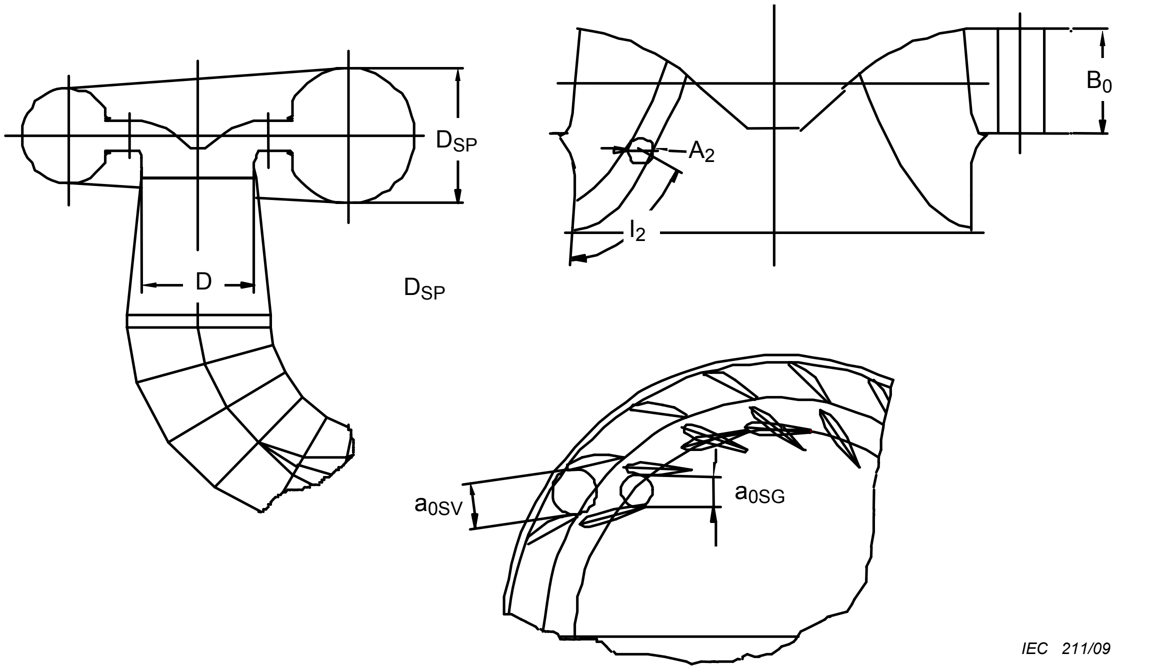


Figure B.3 – Representative dimensions of component passages

Flow velocity factor:

$$\kappa_{uSP} = \frac{v_{SP}}{u} = \frac{1}{u} \times \frac{4 \times Q_1}{\pi \times D_{SP}^2}, \ \kappa_{uSV} = \frac{v_{SV}}{u} = \frac{1}{u} \times \frac{Q_1}{Z_{SV} \times a_{0SV} \times B_0}, \ \kappa_{uGV} = \frac{v_{GV}}{u} = \frac{1}{u} \times \frac{Q_1}{Z_{GV} \times a_{0GV} \times B_0}$$

$$\kappa_{uRU} = \frac{v_{RU}}{u} = \frac{1}{u} \times \frac{Q_1}{Z_{RU} \times \int A_2 dl_2} = \frac{1}{u} \times \frac{Q_1}{Z_{RU} \times S_{0RU}}, \quad \kappa_{uDT} = \frac{v_{DT}}{u} = \frac{1}{u} \times \frac{4 \times Q_1}{\pi \times D^2} \qquad \text{(B.9)}$$

Dimension factor:

$$\kappa_{dSP} = \frac{D_{SP}}{D}, \ \kappa_{dSV} = \frac{2 \times a_{0SV} \times B_0}{D(a_{0SV} + B_0)}, \ \kappa_{dGV} = \frac{2 \times a_{0GV} \times B_0}{D(a_{0GV} + B_0)}$$

$$\kappa_{dRU} = \frac{4 \times \int_0^{l_2} A_2 dl_2}{D(2 \times l_2 + A_{2crown} + A_{2band})} = \frac{4 \times S_{0RU}}{D(2 \times l_2 + A_{2crown} + A_{2band})}, \ \kappa_{dDT} = 1 \qquad \text{(B.10)}$$

where

S_{0RU} sectional area of the flow passage between runner blades at the outlet section;

Z number of vanes or blades.

The values of κ_{uCO} and κ_{dCO} are calculated for the machines of average design currently used in the industry. Their standardized values are shown in B.5.

2) Derivation of the scale effect formula for component wise step-up

The standardized scalable loss δ_{ECO} is defined for each component passage as the scalable loss of a smooth model operating at Re_M=Re_{ref}. It means that the values of δ_{ECOref} correspond to λ_{COref}. Therefore, the equation shown at the end of the table in A.2, 2) can be rewritten for each component passage as follows:

$$\Delta_{ECO} = \frac{\Delta\eta_{ECO}}{\eta_{EM}} = \delta_{ECOref}\left(\frac{\lambda_{COM}-\lambda_{COP}}{\lambda_{COref}}\right) = \delta_{ECOref}\frac{\Delta\lambda_{CO}}{\lambda_{COref}} \quad \text{(B.11)}$$

The term of $\Delta\lambda_{CO}$ on the right side is expressed as follows by using Equation B.6.

$$\Delta\lambda_{CO} = 0{,}74\times\lambda_0\left[\left(4\times10^5\frac{Ra_{COM}}{D_M}\frac{1}{\kappa_{dCO}}+\frac{1}{\kappa_{uCO}\times\kappa_{dCO}}\frac{Re_0}{Re_M}\right)^{0{,}2}-\left(4\times10^5\frac{Ra_{COP}}{D_P}\frac{1}{\kappa_{dCO}}+\frac{1}{\kappa_{uCO}\times\kappa_{dCO}}\frac{Re_0}{Re_P}\right)^{0{,}2}\right] \quad \text{(B.12)}$$

The term of λ_{COref} is the loss coefficient when the Reynolds number of the machine is Re_{ref} or the Reynolds number of the component passage is $Re_{dCOref} = \kappa_{uCO}\times\kappa_{dCO}\times Re_{ref}$.

As Re_{ref} = Re_0 = 7×10^6 and the surface roughness of the reference model is smooth (namely, $\frac{Ra}{D}$ ≈ 0), λ_{COref} can be written as follows:

$$\lambda_{COref} = \lambda_0\left[0{,}74\left(\frac{Re_0}{\kappa_{uCO}\times\kappa_{dCO}\times Re_{ref}}\right)^{0{,}2}+0{,}26\right] = \lambda_0\left[0{,}74\left(\frac{1}{\kappa_{uCO}\times\kappa_{dCO}}\right)^{0{,}2}+0{,}26\right] \quad \text{(B.13)}$$

Then Δ_{ECO} is obtained by replacing $\Delta\lambda_{CO}$ and λ_{COref} in Equation B.11 by Equation B.12 and Equation B.13.

$$\Delta_{ECO} = \frac{\Delta\eta_{ECO}}{\eta_{EM}} = \delta_{ECOref}\left(\frac{\lambda_{COM}-\lambda_{COP}}{\lambda_{COref}}\right) = \delta_{ECOref}\frac{\Delta\lambda_{CO}}{\lambda_{COref}}$$

$$= \delta_{ECOref}\frac{\left(\frac{4\times10^5}{\kappa_{dCO}}\frac{Ra_{COM}}{D_M}+\frac{7\times10^6}{\kappa_{uCO}\kappa_{dCO}}\frac{1}{Re_M}\right)^{0{,}2}-\left(\frac{4\times10^5}{\kappa_{dCO}}\frac{Ra_{COP}}{D_P}+\frac{7\times10^6}{\kappa_{uCO}\kappa_{dCO}}\frac{1}{Re_P}\right)^{0{,}2}}{\left(\frac{1}{\kappa_{uCO}\times\kappa_{dCO}}\right)^{0{,}2}+\frac{0{,}26}{0{,}74}}$$

$$\therefore \Delta_{ECO} = \delta_{ECOref}\frac{\left(4\times10^5\kappa_{uCO}\frac{Ra_{COM}}{D_M}+\frac{7\times10^6}{Re_M}\right)^{0{,}2}-\left(4\times10^5\kappa_{uCO}\frac{Ra_{COP}}{D_P}+\frac{7\times10^6}{Re_P}\right)^{0{,}2}}{1+0{,}35(\kappa_{uCO}\times\kappa_{dCO})^{0{,}2}}$$

(B.14)

For simplification, the above formula is rewritten as follows:

$$\therefore\ \Delta_{ECO} = d_{ECOref}\left(\left(4\times10^5\,\kappa_{uCO}\,\frac{Ra_{COM}}{D_M}+\frac{7\times10^6}{Re_M}\right)^{0,2}-\left(4\times10^5\,\kappa_{uCO}\,\frac{Ra_{COP}}{D_P}+\frac{7\times10^6}{Re_P}\right)^{0,2}\right) \quad \text{(B.15)}$$

where

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

The standardized values of δ_{ECOref} are shown in B.4 and those of κ_{uCO} and κ_{dCO} are shown in B.5. The values of d_{ECOref} calculated from δ_{ECOref}, κ_{uCO} and κ_{dCO}, are shown in B.6.

Then, the step-up amount of the specific energy efficiency for the whole turbine $\Delta\eta_E$ can be calculated by the following formula:

$$\frac{\Delta\eta_E}{\eta_{EM}} = \Delta_E = \sum \Delta_{ECO} \quad \text{(B.16)}$$

B.3 Direct step-up for a whole turbine

By putting Equation B.15 into Equation B.16 and introducing the reference velocity index C_{u0}, we obtain:

$$\begin{aligned}\Delta_E &= \sum d_{ECOref}\left[\left(4\times10^5\,\kappa_{uCO}\,\frac{Ra_{COM}}{D_M}+\frac{7\times10^6}{Re_M}\right)^{0,2}-\left(4\times10^5\,\kappa_{uCO}\,\frac{Ra_{COP}}{D_P}+\frac{7\times10^6}{Re_P}\right)^{0,2}\right]\\ &= \sum d_{ECOref}\left[\left(4\times10^5\,\kappa_{u0}\,\frac{\kappa_{uCO}}{\kappa_{u0}}\frac{Ra_{COM}}{D_M}+\frac{7\times10^6}{Re_M}\right)^{0,2}-\left(4\times10^5\,\kappa_{u0}\,\frac{\kappa_{uCO}}{\kappa_{u0}}\frac{Ra_{COP}}{D_P}+\frac{7\times10^6}{Re_P}\right)^{0,2}\right]\end{aligned}$$

If the values of the terms $\left(\frac{\kappa_{uCO}}{\kappa_{u0}}\frac{Ra_{COM}}{D_M}\right)$ for all the model components can be regarded as the same and replaced by $\left(\frac{Ra_{0M}}{D_M}\right)$ and, similarly, those for the prototype component passages by $\left(\frac{Ra_{0P}}{D_P}\right)$, the above formula can be rewritten as follows:

$$\begin{aligned}\Delta_E &= \sum d_{ECOref}\left[\left(4\times10^5\,\kappa_{u0}\,\frac{\kappa_{uCO}}{\kappa_{u0}}\frac{Ra_{COM}}{D_M}+\frac{7\times10^6}{Re_M}\right)^{0,2}-\left(4\times10^5\,\kappa_{u0}\,\frac{\kappa_{uCO}}{\kappa_{u0}}\frac{Ra_{COP}}{D_P}+\frac{7\times10^6}{Re_P}\right)^{0,2}\right]\\ &= \sum d_{ECOref}\left[\left(4\times10^5\,\kappa_{u0}\,\frac{Ra_{0M}}{D_M}+\frac{7\times10^6}{Re_M}\right)^{0,2}-\left(4\times10^5\,\kappa_{u0}\,\frac{Ra_{0P}}{D_P}+\frac{7\times10^6}{Re_P}\right)^{0,2}\right] \quad \text{(B.17)}\\ &= d_{Eref}\left[\left(4\times10^5\,\kappa_{u0}\,\frac{Ra_{0M}}{D_M}+\frac{7\times10^6}{Re_M}\right)^{0,2}-\left(4\times10^5\,\kappa_{u0}\,\frac{Ra_{0P}}{D_P}+\frac{7\times10^6}{Re_P}\right)^{0,2}\right]\end{aligned}$$

The formula of Equation B.17 can be used for the direct step-up of the specific energy efficiency of the whole turbine.

Since the friction loss in runner and guide vanes shares about two thirds of total friction loss, the average value of κ_{uRU} and κ_{uGV} is used as the reference velocity index κ_{u0}. Also, the average value of Ra_{GV} and Ra_{RU} is used as the representative roughness of the machine Ra_0.

$$\kappa_{u0} = \frac{\kappa_{uGV} + \kappa_{uRU}}{2} \tag{B.18}$$

$$Ra_0 = \frac{Ra_{GV} + Ra_{RU}}{2} \tag{B.19}$$

The values of $d_{Eref} = \sum d_{ECOref}$ and κ_{u0} are calculated from the standardized values of d_{ECOref}, κ_{uRU} and κ_{uGV} shown in B.5 and B.6 and shown in Table B.1.

Table B.1 – d_{Eref} and κ_{u0} for step-up calculation of whole turbine

Francis turbine		d_{Eref} = 3,05/100	$\kappa_{u0} = -2{,}3N_{QE} + 1{,}10$
Pump-turbine	(turbine operation)	d_{Eref} = 3,95/100	$\kappa_{u0} = -2{,}3N_{QE} + 1{,}05$
	(pump operation)	d_{Eref} = 4,20/100	$\kappa_{u0} = -2{,}3N_{QE} + 0{,}88$

For the application of Equation B.17, it is required to keep $\frac{\kappa_{uCO}}{\kappa_{u0}}\frac{Ra_{COM}}{D_M} \approx \frac{Ra_{0M}}{D_M}$ and $\frac{\kappa_{uCO}}{\kappa_{u0}}\frac{Ra_{COP}}{D_P} \approx \frac{Ra_{0P}}{D_P}$. In other words, the surface roughness of each component passage is required to be within the range $\frac{Ra_{COM}}{D_M} \approx \frac{\kappa_{u0}}{\kappa_{uCO}}\frac{Ra_{0M}}{D_M}$ and $\frac{Ra_{COP}}{D_P} \approx \frac{\kappa_{u0}}{\kappa_{uCO}}\frac{Ra_{0P}}{D_P}$. The values of $\frac{\kappa_{u0}}{\kappa_{uCO}}$ obtained from the values of κ_{uCO} given in B.5 and the required range of roughness for the application of Equation B.17 are shown in Table B.2.

Table B.2 – Criteria for the surface roughness for the application of the direct step-up formula

Component passage	$\frac{\kappa_{u0}}{\kappa_{uCO}}$						Required roughness range	
	Francis turbines		Pump-turbine (T)		Pump-turbine (P)		Model	Prototype
	Range[a]	Ave.	Range[a]	Ave.	Range[a]	Ave.		
SP	3,00 ~ 2,31	2,69	2,92 ~ 2,58	2,75	2,56 ~ 2,06	2,31	(2,0~4,0) Ra_{0M}	< 3,0 Ra_{0P}
SV	1,40 ~ 2,03	1,60	1,78 ~ 2,56	2,03	1,53 ~ 2,04	1,72	(1,5~3,0) Ra_{0M}	< 2,5 Ra_{0P}
GV	0,87 ~ 1,25	0,99	0,88 ~ 1,01	0,93	0,95 ~ 1,26	1,07	(0,7~1,3) Ra_{0M}[b]	< 1,3 Ra_{0P}[b]
RU	1,17 ~ 0,83	1,01	1,16 ~ 1,00	1,08	1,06 ~ 0,83	0,94	(0,7~1,3) Ra_{0M}[b]	< 1,3 Ra_{0P}[b]
DT[c]	4,86 ~ 2,33	3,40	4,26 ~ 2,94	3,54	3,89 ~ 2,37	3,03	(2,5~4,5) Ra_{0M}	< 4,0 Ra_{0P}

[a] The values on the left indicate those for the lowest specific speed (N_{QE} = 0,06) and those on the right indicate the values for the highest specific speed (N_{QE} = 0,30 for Francis turbine, N_{QE} = 0,20 for pump-turbine).

[b] Since the average value of Ra_{GV} and Ra_{RU} is defined as Ra_0, when Ra_{GV} is selected as 1,3 Ra_0, Ra_{RU} should be 0,7 Ra_0.

[c] In case of draft tube, κ_{uDT} is defined at the upstream end of the draft tube, where the diameter is the same as the reference diameter and the velocity to calculate κ_{uDT} is the highest in the draft tube section. To evaluate the roughness effect in the draft tube, it seems reasonable to use the average flow velocity, which is approximately estimated to be 0,7 times the velocity at the upstream section. From this viewpoint, $\frac{\kappa_{u0}}{0{,}7 \times \kappa_{uDT}}$ is indicated in the row of $\frac{\kappa_{u0}}{\kappa_{uCO}}$ for the draft tube.

B.4 Relative scalable hydraulic energy loss of radial flow machines

1) Definition

On the basis stated in B.2, the scalable loss dealt with in this standard is defined for each component passage (spiral case, stay vanes, guide vanes, runner, draft tube) as follows:

$$\delta_{ECOref} = \frac{E_{LfCO}}{E}$$

where

δ_{ECOref} scalable specific hydraulic energy loss ratio of each component;

E_{LfCO} specific hydraulic energy loss due to surface friction of each component at the maximum efficiency point when the machine is operated at the reference Reynolds number;

E specific hydraulic energy of the machine.

The following values were derived from numerical analysis conducted on the industrial models designed by different manufacturers. [7] In order to quantify the friction loss in the water passages, various methods which reflected the present state of the art are used.

For spiral case and draft tube:

- Friction loss as an equivalent pipe according to Colebrook formula, Moody diagram, Blasius formula or Nikuradse formula.

For stay vanes and guide vanes:

- Friction loss as a flat plate applied to surrounding walls of a rectangular passage.
- Boundary layer calculation based on the velocity distribution of the main flow obtained by inviscid CFD analysis.

For runner:

- Boundary layer calculation based on the velocity distribution of the main flow obtained by inviscid CFD analysis.

The evaluation of the friction loss by boundary layer calculation was conducted by one of the following methods:

- Integration of the loss energy due to the shear stress in boundary layer over whole surface area.
- Dissipation of velocity energy obtained from the lack of fluid velocity energy downstream the trailing edge of the blade/vane which can be calculated by the energy thickness of the boundary layer.

The values of δ_{ECOref}, κ_{uCO}, κ_{dCO} and d_{ECOref} set out in Annex B are substantiated by analytical or experimental data for the following specific speed ranges:

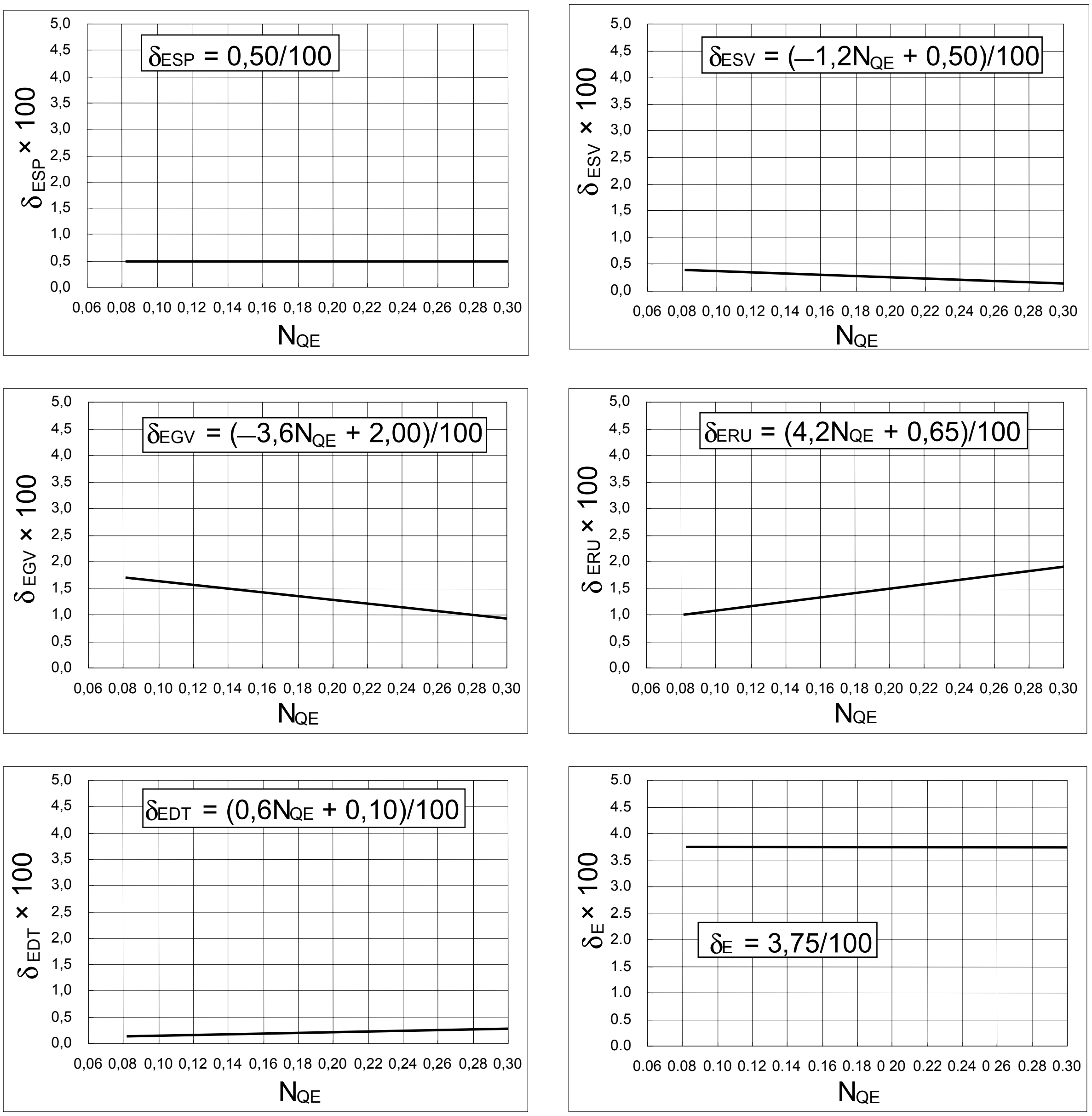
- For Francis turbines $0{,}06 \le N_{QE} \le 0{,}30$;
- For pump-turbines $0{,}06 \le N_{QE} \le 0{,}20$.

Outside of these ranges, their values may not be correct. Therefore, if the step-up equations in this standard are applied to the evaluation of the contractual model test results beyond the above specific speed ranges, prior agreement shall be made among the concerned parties.

Total friction loss of a whole turbine $\delta_{Eref} = \sum \delta_{ECOref}$, which is used for direct step-up of the hydraulic efficiency of a whole turbine, is also shown at the end of the figures.

2) Relative scalable hydraulic energy loss δ_E of Francis turbine

The values of δ_{ECOref} calculated for some typical models are plotted against specific speed and shown below. For the convenience, the plots are approximated by linear functions.



IEC 212/09

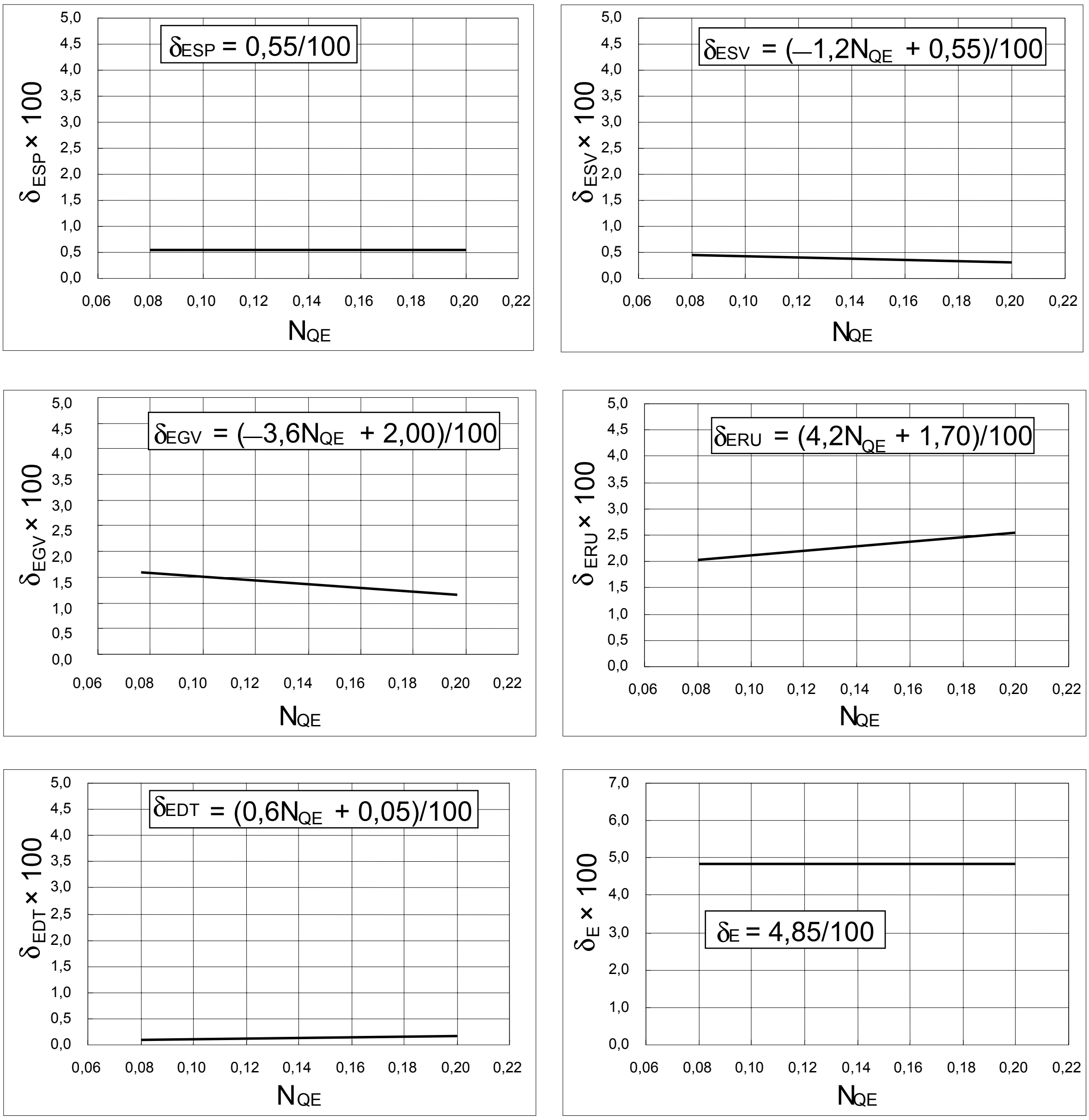
Figure B.4 – Relative scalable hydraulic energy loss in each component of Francis turbine

It should be noted that the abscissa N_{QE} is the dimensionless specific speed defined in IEC 60193, which is defined as $N_{QE} = nQ_1^{0,5} / E^{0,75}$, where n is rotating speed in terms of sec^{-1} and E is specific hydraulic energy of the machine in terms of J kg^{-1}.

3) Relative scalable hydraulic energy loss δ_E of reversible pump-turbine

The values of scalable loss ratio of pump-turbines are separately calculated for each turbine or pump operation. They are plotted against the specific speed calculated for the maximum efficiency point in turbine or pump operation, respectively.

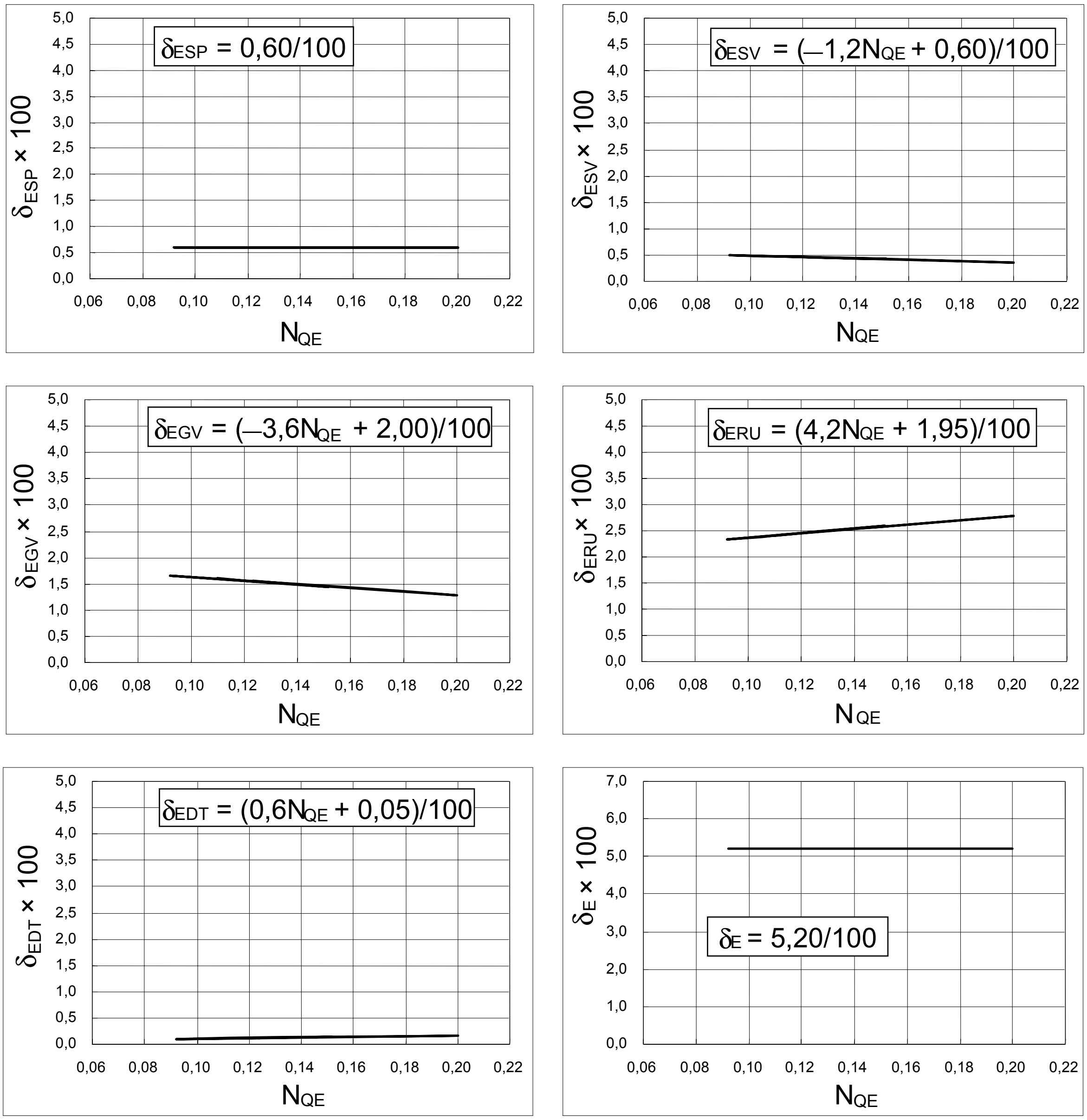
a) Turbine operation



IEC 213/09

Figure B.5 – Relative scalable hydraulic energy loss in each component of pump-turbine in turbine operation

b) Pump operation



IEC 214/09

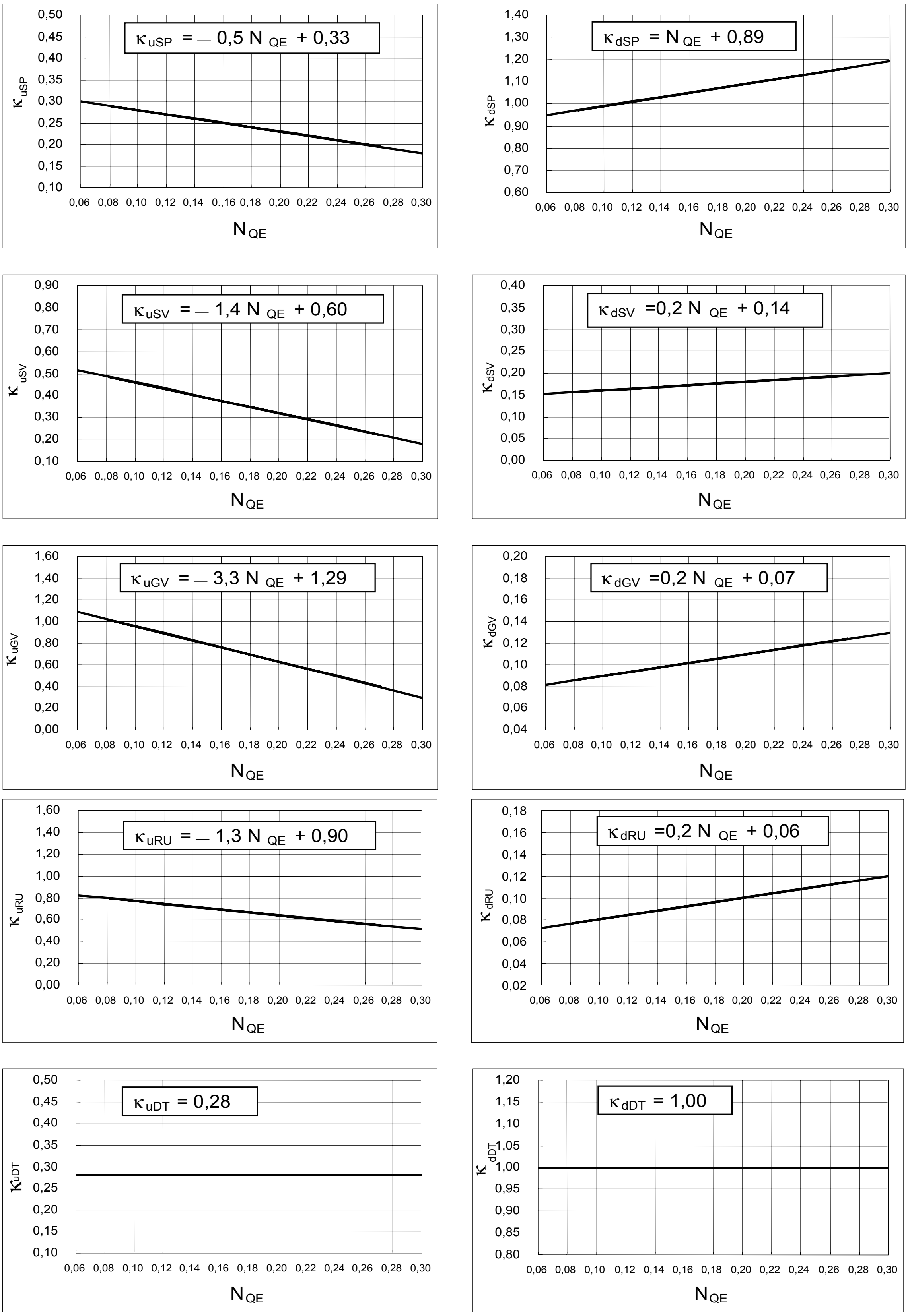
Figure B.6 – Relative scalable hydraulic energy loss in each component of pump-turbine in pump operation

B.5 Flow velocity factor κ_{uCO} and dimension factor κ_{dCO} of radial flow machines [9]

1) Definition

Based on standardized geometry data of hydraulic machines, flow velocity factor κ_{uCO} as defined by Equation B.9 and dimension factors κ_{dCO} as defined by Equation B.10 set out in B.2 are calculated. Since these parameters are used for calculating d_{ECOref} and for the final scale effect formula (Equation 8 or Equation B.17) in the term with exponent of 0,2, some deviation can be tolerated. Therefore, the calculated results are approximated by linear lines for simplification.

2) κ_{uCO} and κ_{dCO} for Francis turbine

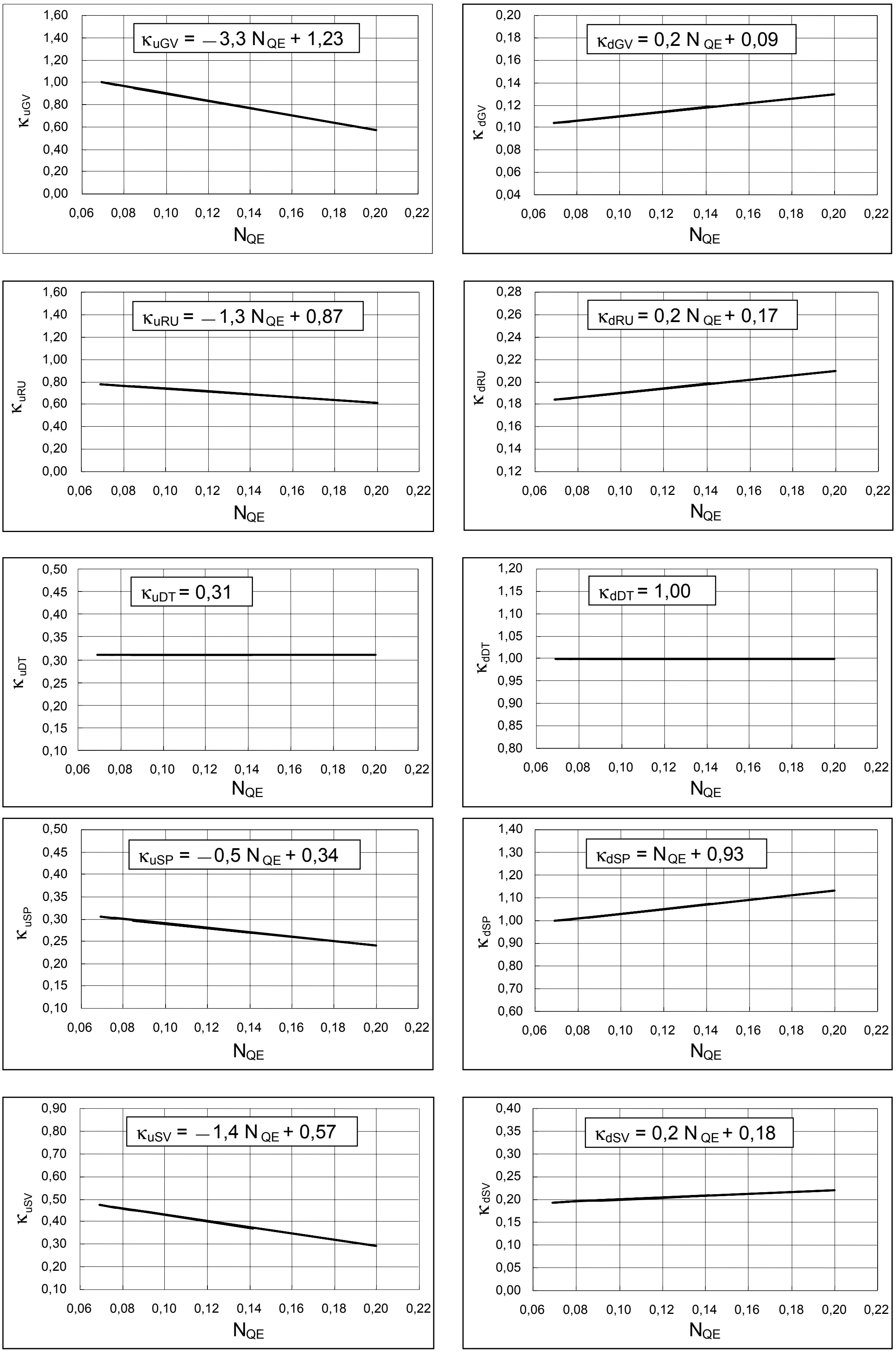


IEC 215/09

Figure B.7 – κ_{uCO} and κ_{dCO} in each component of Francis turbine

3) κ_{uCO} and κ_{dCO} for pump-turbine

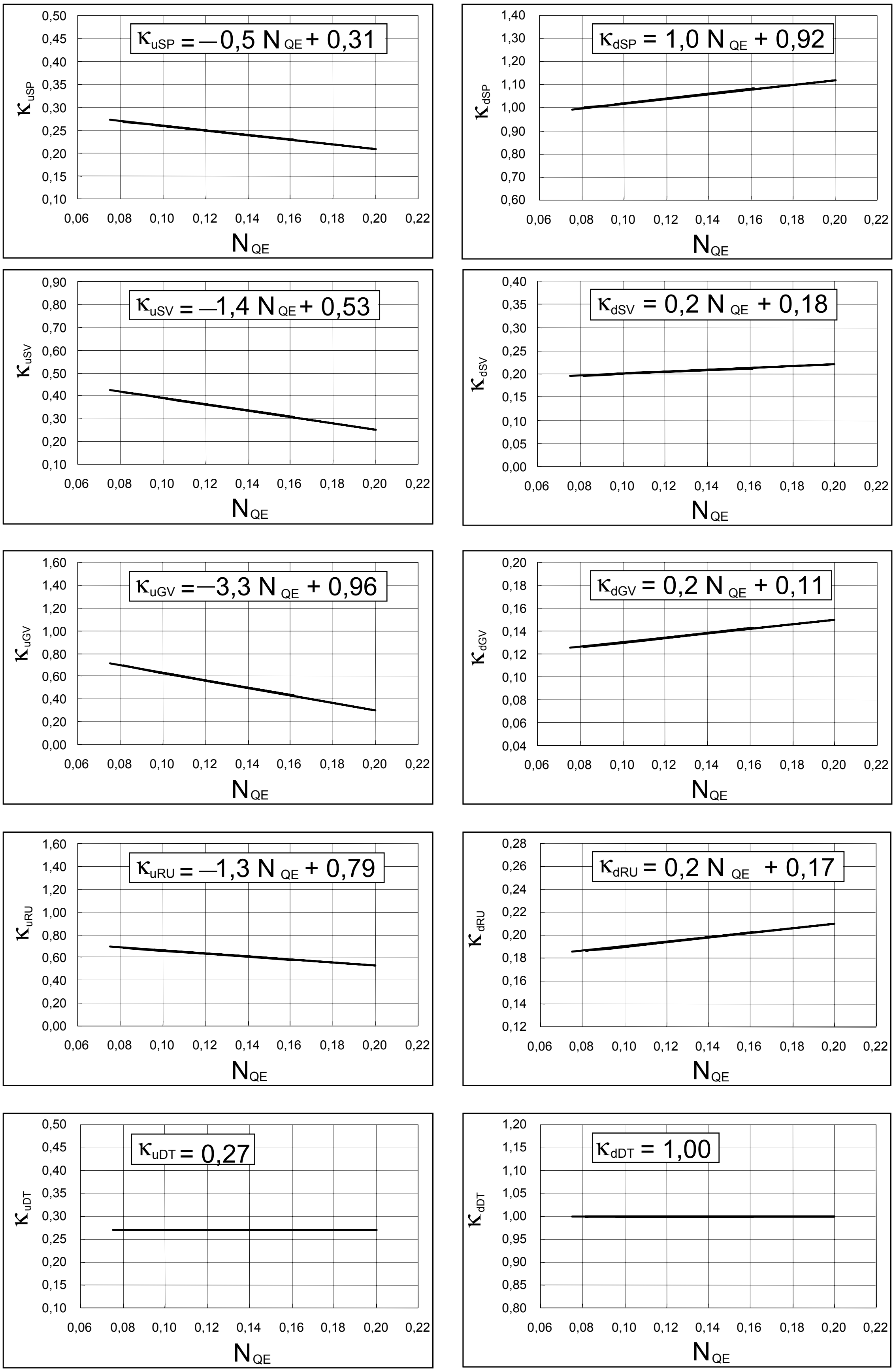
a) Turbine operation



IEC 216/09

Figure B.8 – κ_{uCO} and κ_{dCO} in each component of pump-turbine in turbine operation

b) Pump operation



IEC 217/09

Figure B.9 – κ_{uCO} and κ_{dCO} in each component of pump-turbine in pump operation

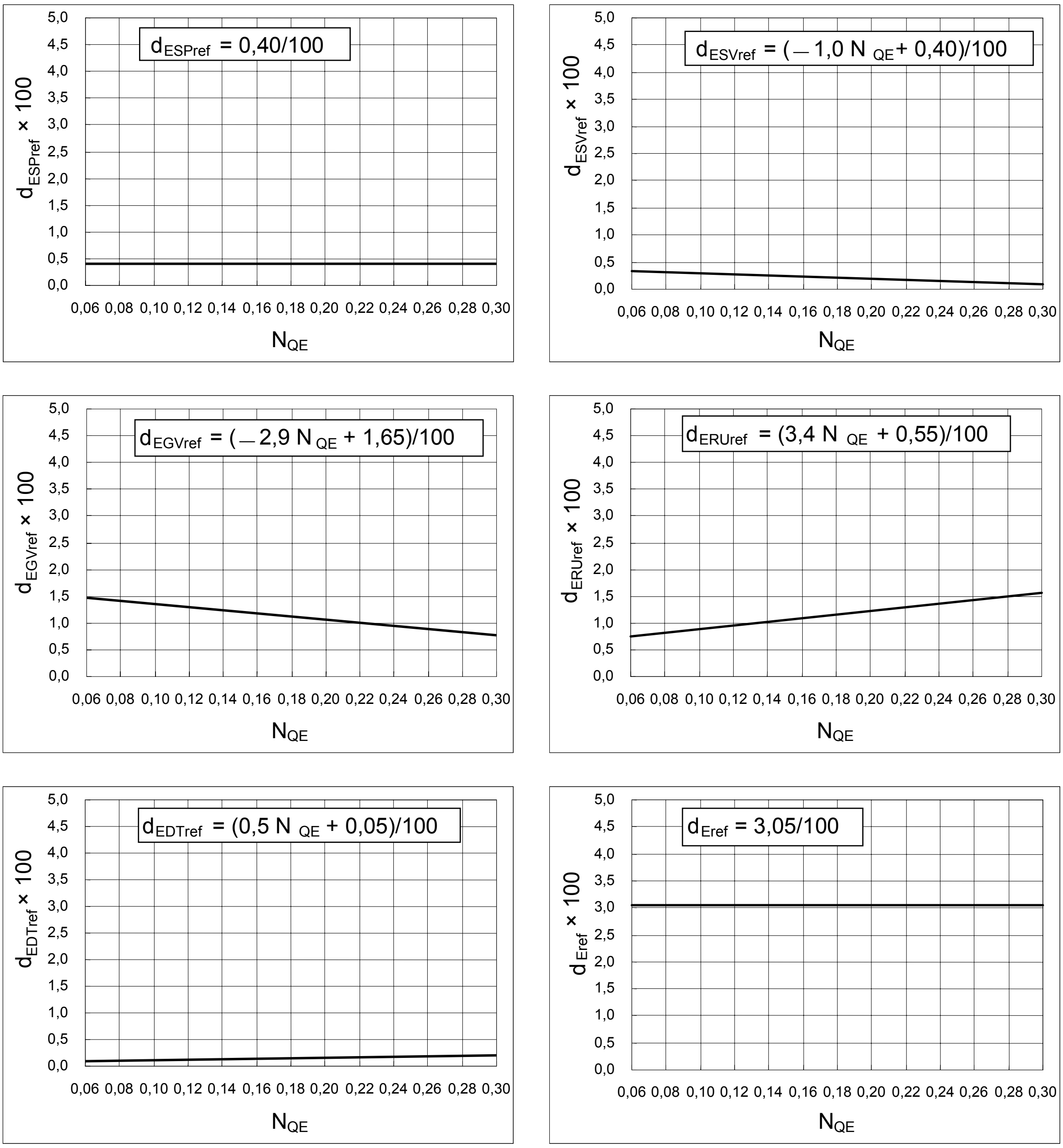
B.6 Scalable loss index d_{ECOref}

1) Definition

Based on δ_{ECOref}, flow velocity factor κ_{uCO} and dimension factor κ_{dCO}, scalable loss index d_{ECOref} is calculated as explained in 4.2.1. The calculated results of d_{ECOref} are approximated by linear function for simplification.

Total scalable loss index d_{Eref}, which is to be used for direct step-up of the specific hydraulic energy efficiency of a whole turbine (see B.3), is also shown at the end of the figures.

2) d_{ECOref} and d_{Eref} for Francis turbine

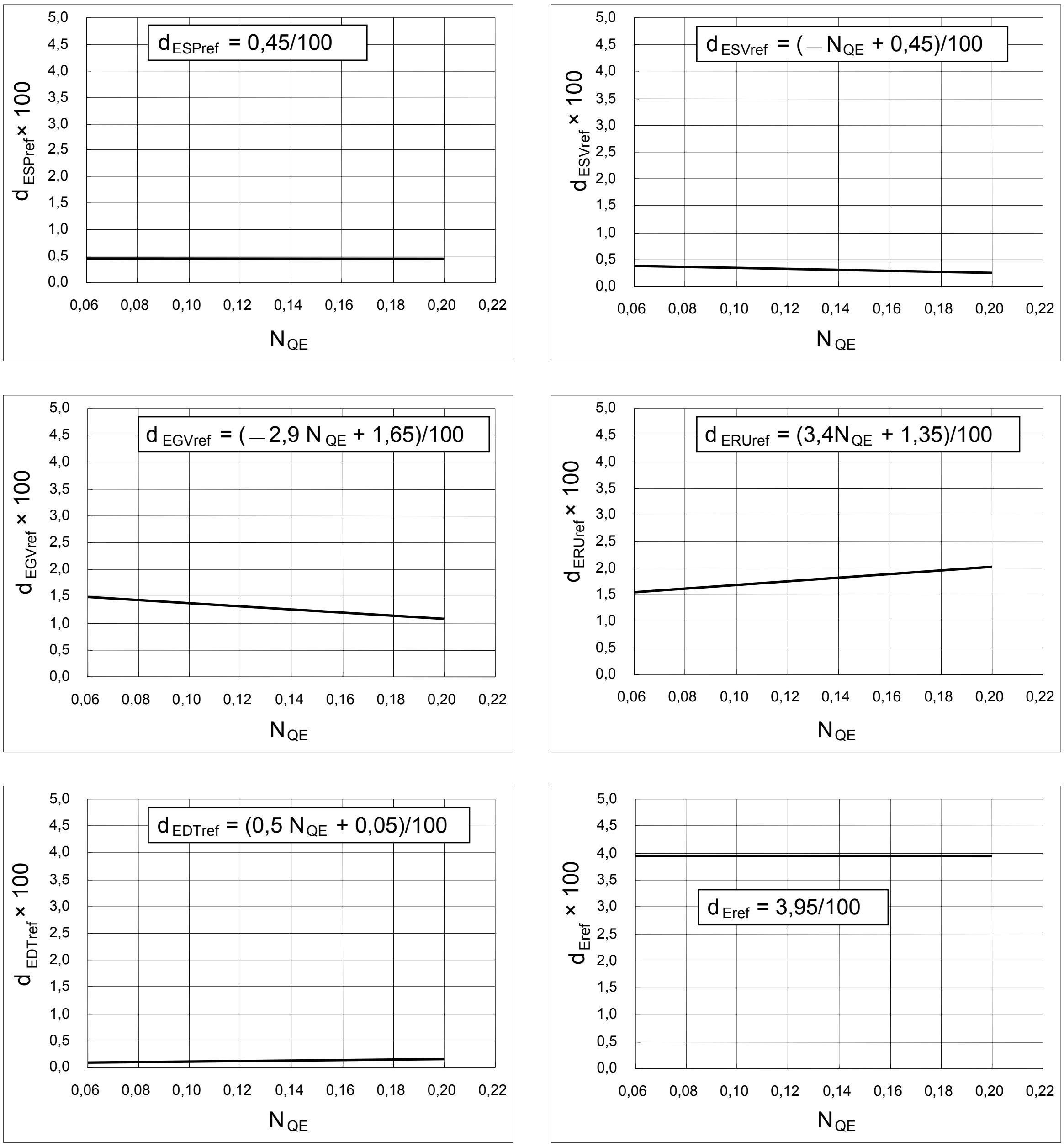


IEC 218/09

Figure B.10 – d_{ECOref} and d_{Eref} for Francis turbine

3) d_{ECOref} and d_{Eref} for pump-turbine

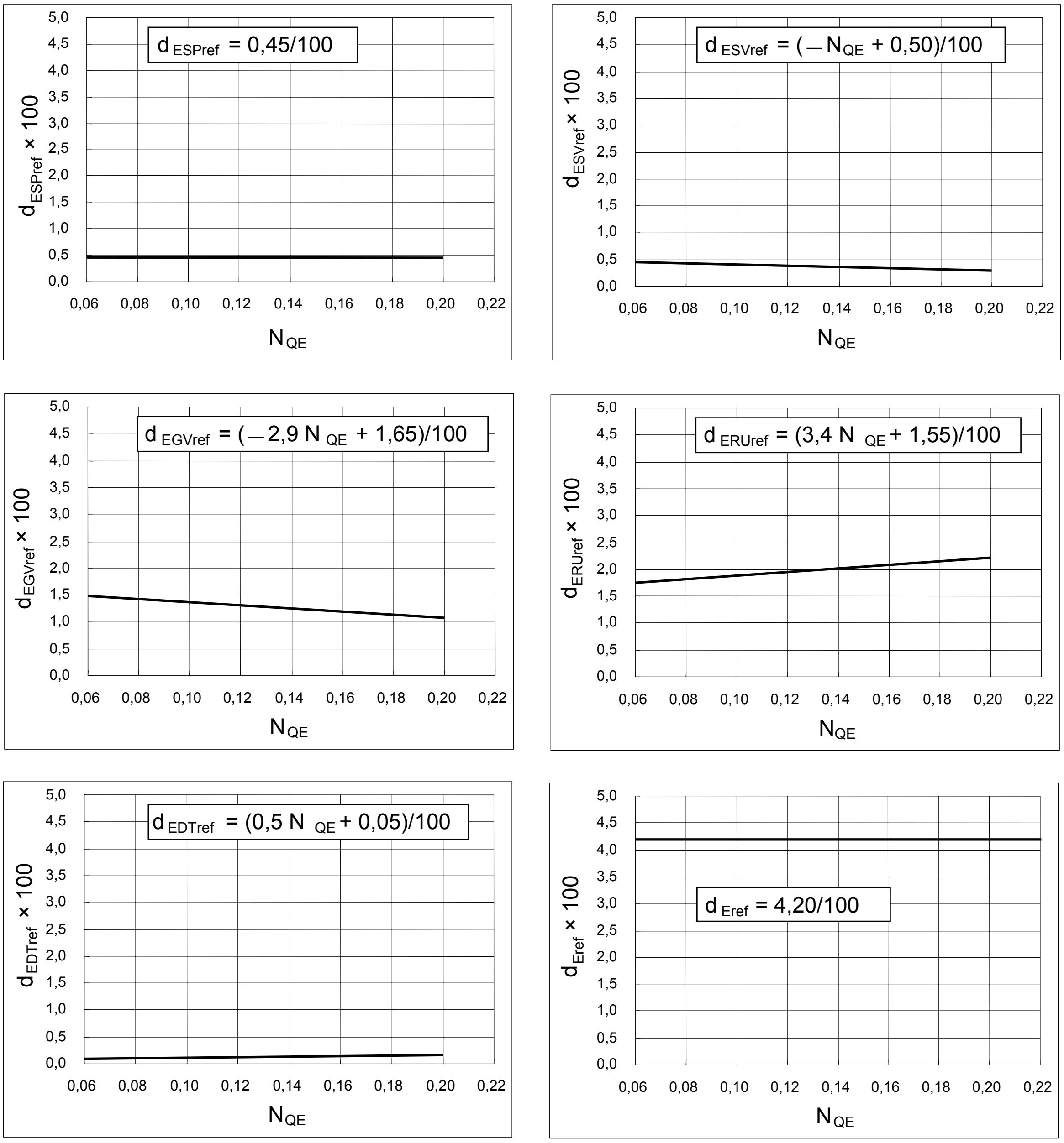
a) Turbine operation



IEC 219/09

Figure B.11 – d_{ECOref} and d_{Eref} for pump-turbine in turbine operation

b) Pump operation



IEC 220/09

Figure B.12 – d_{ECOref} and d_{Eref} for pump-turbine in pump-operation

Annex C
(informative)

Scale effect on specific hydraulic energy losses of axial flow machines [10]

C.1 Scalable losses of axial flow machines

Although detailed analysis on the scalable losses of axial flow machines is not available at present, it is prescribed in this standard that they can be dealt with in two parts, one part for runner blades and the other one for all other stationary components.

For the scalable loss of runner blades, the scale effect formula for flat plate (Equation 5) is applied. For the stationary parts, the formula for pipe flow (Equation 1) is applied in the same way as for radial flow turbines.

C.2 Scale effect formula for runner blades [9]

From the scale effect formula for flat plate (Equation 5), the following step-up formula for runner blades is derived:

$$\Delta_{ERU} = \frac{\Delta\eta_{ERU}}{\eta_{EM}} = \delta_{ERUref}\left(\frac{C_{fRUM} - C_{fRUP}}{C_{fRUref}}\right)$$

$$= \delta_{ERUref}\left[\frac{\left(5\times10^5\,\frac{Ra_{RUM}}{L_M} + \frac{D_M\times u_M}{L_M\times w_M}\times\frac{Re_0}{Re_M}\right)^{0,2} - \left(5\times10^5\,\frac{Ra_{RUP}}{L_P} + \frac{D_P\times u_P}{L_P\times w_P}\times\frac{Re_0}{Re_P}\right)^{0,2}}{\left(\frac{D_M\times u_M}{L_M\times w_M}\right)^{0,2} + 0,25}\right]$$

$$= \delta_{ERUref}\left[\frac{\left(5\times10^5\kappa_{uRU}\,\frac{Ra_{RUM}}{D_M} + \frac{Re_0}{Re_M}\right)^{0,2} - \left(5\times10^5\kappa_{uRU}\,\frac{Ra_{RUP}}{D_P} + \frac{Re_0}{Re_P}\right)^{0,2}}{1 + 0,25\left(\kappa_{dRU}\times\kappa_{uRU}\right)^{0,2}}\right]$$

$$= d_{ERUref}\left[\left(5\times10^5\kappa_{uRU}\,\frac{Ra_{RUM}}{D_M} + \frac{Re_0}{Re_M}\right)^{0,2} - \left(5\times10^5\kappa_{uRU}\,\frac{Ra_{RUP}}{D_P} + \frac{Re_0}{Re_P}\right)^{0,2}\right]$$

(C.1)

where

δ_{ERUref} standardized reference scalable loss for runner blades when the machine Reynolds number Re_M is equal to the reference Reynolds number (7×10^6);

L length of runner blade;

w relative flow velocity at the runner exit;

u peripheral velocity of runner blades;

κ_{uRU} standardized flow velocity factor for runner blade passage:

$$\kappa_{uRU} = \frac{w_M}{u_M} = \frac{w_P}{u_P}$$

κ_{dRU} standardized dimension factor for runner blade passage:

$$\kappa_{dRU} = \frac{L_M}{D_M} = \frac{L_P}{D_P}$$

d_{ERUref} scalable loss index for runner blades:

$$d_{ERUref} = \frac{\delta_{ERUref}}{1 + 0,25(\kappa_{dRU} \times \kappa_{uRU})^{0,2}} \quad \text{(C.2)}$$

The above Equation C.1 can be transformed to Equation C.3 as shown below by introducing modified flow velocity factor κ^*_{uRU}. This formula has the same form as Equation 8, which is applied to all the water passages of radial flow machines and the stationary parts of axial flow machines.

$$\Delta_{ERU} = d_{ERUref}\left[\left(5\times10^5\,\kappa_{uRU}\,\frac{Ra_{RUM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0,2} - \left(5\times10^5\,\kappa_{uRU}\,\frac{Ra_{RUP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0,2}\right]$$

$$= d_{ERUref}\left[\left(4\times10^5(1,25\times\kappa_{uRU})\frac{Ra_{RUM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0,2} - \left(4\times10^5(1,25\times\kappa_{uRU})\frac{Ra_{RUP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0,2}\right] \quad \text{(C.3)}$$

$$= d_{ERUref}\left[\left(4\times10^5\,\kappa^*_{uRU}\,\frac{Ra_{RUM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0,2} - \left(4\times10^5\,\kappa^*_{uRU}\,\frac{Ra_{RUP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0,2}\right]$$

where

κ^*_{uRU} modified flow velocity factor for runner blades:

$$\kappa^*_{uRU} = 1,25 \times \kappa_{uRU}$$

Since κ_{uRU} is approximately 1,03 for all axial machines, κ^*_{uRU} is finally given as follows:

$$\kappa^*_{uRU} = 1,25 \times \kappa_{uRU} = 1,29 \quad \text{(C.4)}$$

C.3 Scale effect formula for stationary parts

From the scale effect formula for pipe flow (Equation 1), the step-up formula to obtain Δ_E is derived. The formula is shown in the main text as Equation 8.

When applying Equation 8 to the scalable loss of all stationary parts, the following two simplifications are introduced.

1) Flow velocity factor to represent the flow velocity in all the stationary parts is considered to be 0,8 times the flow velocity factor of guide vane passage, κ_{uGV}. The value of κ_{uGV} is approximately 0,29 for low specific speed and 0,19 for high specific speed axial machines.

Then, it is simplified in this standard that κ_{uGV} is 0,24 for all axial machines taking the middle value.

2) Roughness representing all the stationary parts can be given by the arithmetical mean of the roughness of guide vanes and stay vanes.

Then the following step-up formula is applied to the scalable loss of stationary parts.

$$\Delta_{EST} = d_{ESTref}\left[\left(4\times10^5\kappa_{uST}\frac{Ra_{STM}}{D_M}+\frac{7\times10^6}{Re_M}\right)^{0,2}-\left(4\times10^5\kappa_{uST}\frac{Ra_{STP}}{D_P}+\frac{7\times10^6}{Re_P}\right)^{0,2}\right] \quad (C.5)$$

where

κ_{uST} flow velocity factor representing stationary parts:

$$\kappa_{uST} = 0,8\times\kappa_{uGV} \approx 0,19 \quad (C.6)$$

Ra_{ST} representative roughness of stationary parts:

$$Ra_{ST} = \frac{Ra_{SV} + Ra_{GV}}{2} \quad (C.7)$$

C.4 Scale effect for other efficiency components

C.4.1 Volumetric efficiency

If the runner tip clearance is homologous to the prototype, scale effect on volumetric efficiency can be neglected and $\Delta\eta_Q$ is regarded as 0.

Since the influence on η_Q caused by non-homologous tip clearance is not exactly known, no correction formula for non-homologous tip clearance can be provided. Therefore, it is a primary requirement to maintain the homology of the tip clearance between model and prototype turbines within the tolerances given in Table 3.

C.4.2 Power efficiency (disc friction)

Since the disc friction loss of runner hub is negligibly small, $\Delta\eta_T$ is regarded as 0.

C.5 Step-up of hydraulic efficiency

As stated above, in case of axial flow machines, only the scale effect on specific hydraulic energy efficiency is considered. Then the step-up amount of hydraulic efficiency is obtained by the following formula:

$$\frac{\eta_{hP}}{\eta_{hM}} = \frac{\eta_{EP}}{\eta_{EM}} = (1+\Delta_E) \qquad \therefore \Delta\eta_h = \Delta_E\times\eta_{hM} \text{ (see Equation 25)}$$

Therefore,

$$\Delta\eta_h = (\Delta_{ERU} + \Delta_{EST})\times\eta_{hM} \quad (C.8)$$

C.6 Determination of δ_{ECOref} of axial flow turbines

Although detailed analysis on the relative scalable hydraulic energy losses, δ_{ECOref}, in axial flow machines is not available at present, some reference materials give outlines of these values. One of these materials provides the scalable losses at the maximum efficiency point of Kaplan turbines as shown in Figure C.1 (see Note) [7].

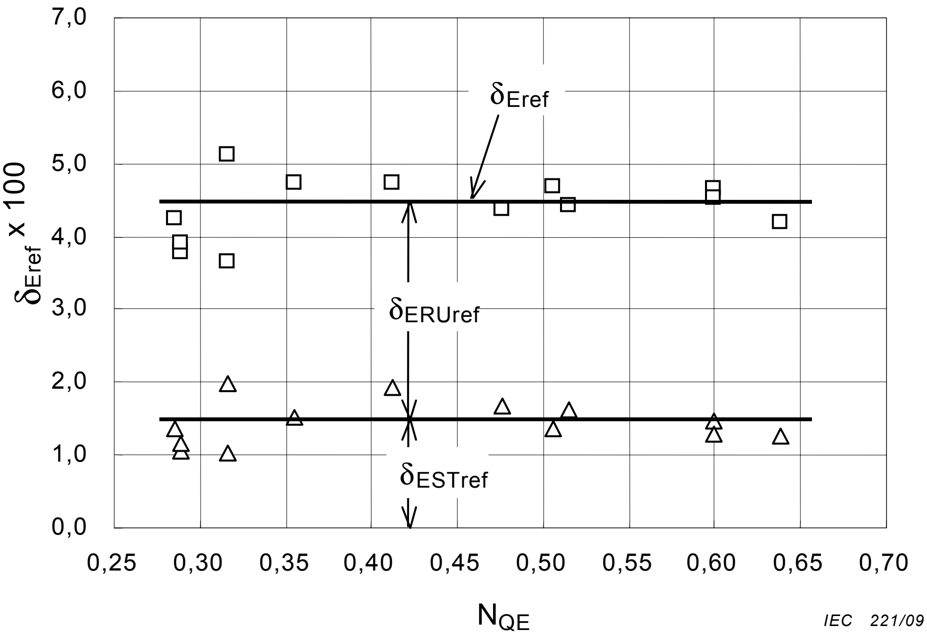


Figure C.1 – δ_{Eref} for Kaplan turbines

where

δ_{ESTref} scalable loss in stationary part;

δ_{ERUref} scalable loss in runner blades;

δ_{Eref} total scalable loss for whole turbine.

As shown in Figure C.1, dependence of δ_{ECOref} and δ_{Eref} on specific speed is very minor. Hence, for all Kaplan turbines, the following constant values are adopted in this standard.

$$\delta_{ESTref} = 0,015 \tag{C.9}$$

$$\delta_{ERUref} = 0,030 \tag{C.10}$$

$$\delta_{Eref} = 0,045 \tag{C.11}$$

These values are also applied for propeller (fixed blade) turbines.

NOTE JSME S008 – 1999 [7] provides three separate values of scalable losses for runner blades, draft tube and other stationary parts of Kaplan turbines. However, it is known that its value for stationary parts is slightly underestimated. Therefore, scalable losses modified from JSME by adequate correction are adopted in this standard. They are regrouped into two separate losses for runner blades and all other stationary parts including draft tube.

C.7 Determination of δ_{ECOref} of bulb turbines

The scalable loss of runner blades of bulb turbines is considered to be the same as of Kaplan turbines. Then, δ_{ERUref}= 0,030.

Regarding the scalable loss in stationary part, no data is available at present to determine the friction loss in the stationary part of bulb turbines. However, it is considered that the friction loss in the upstream part including the annular passage around the bulb is smaller than that of the spiral case of Kaplan turbines. On the other hand, the friction loss in the guide vane area is considered to be slightly larger than that of Kaplan turbines because of narrower passages. At present, the exact amount of these subtraction or addition of friction loss against that of Kaplan turbines is not known.

In any case, it is estimated that the friction loss in the stationary part of both Kaplan and bulb turbines is somewhere around 1,0 – 2,0 %. Therefore, if we adopt the assumption that the above subtraction and addition could cancel with each other, the error of δ_{ESTref} caused by this assumption would not exceed 0,5 %. Then the probable error of the step-up amount calculated from this δ_{ESTref} would be in the range of 0,05 – 0,1 %. Hence, it is thought that this assumption is acceptable.

Based on the above considerations, it is prescribed in this standard that δ_{ECOref} and δ_{Eref} for bulb turbines shall be the same as of Kaplan turbines.

C.8 Derivation of scalable hydraulic energy loss index, d_{Eref}

C.8.1 Scalable loss index for runner blades

Regardless of the specific speed of the machine or the number of runner blades, the values of κ_{dRU} and κ_{uRU} defined at the blade tip are approximately given as follows:

$$\kappa_{dRU} = \frac{L}{D} \approx 0{,}55 \qquad \text{(C.12)}$$

$$\kappa_{uRU} = \frac{w}{u} \approx 1{,}03 \qquad \text{(C.13)}$$

Then d_{ERU} is obtained by using Equation C.2.

$$d_{ERUref} = \frac{\delta_{ERUref}}{1 + 0{,}25(\kappa_{dRU} \times \kappa_{uRU})^{0{,}2}} = \frac{0{,}030}{1 + 0{,}25(0{,}55 \times 1{,}03)^{0{,}2}} \approx 0{,}024\ 5 \qquad \text{(C.14)}$$

C.8.2 Scalable loss index for stationary parts

It is difficult to define κ_{dST} and κ_{uST} representing all the stationary parts. Then, instead of calculating d_{ESTref} by using κ_{dST} and κ_{uST}, the value of d_{ESTref} is estimated by using the relationship between δ_{ECOref} and d_{ECOref} for the stationary parts of radial flow turbines.

Based on the values of δ_{ECOref} and d_{ECOref} for stationary parts of high specific speed Francis turbine (N_{QE}=0,30) and those for high specific speed reversible pump-turbine (N_{QE}=0,20), we can obtain the ratio of $\frac{d_{EST}}{\delta_{EST}} = \frac{\sum d_{ECO}}{\sum \delta_{ECO}}$ as shown in Table C.1 hereafter.

Table C.1 – Ratio of $\frac{d_{EST}}{\delta_{EST}}$ for Francis turbines and pump-turbines

	FT(N_{QE}=0,30)		PT(T) (N_{QE}=0,20)		PT(P) (N_{QE}=0,20)	
	δ_E	d_E	δ_E	d_E	δ_E	d_E
SP	0,50	0,40	0,55	0,45	0,60	0,45
SV	0,14	0,10	0,31	0,25	0,36	0,30
GV	0,92	0,78	1,28	1,07	1,28	1,07
DT	0,28	0,20	0,17	0,15	0,17	0,15
ST=Σ	1,84	1,48	2,31	1,92	2,41	1,97
$\frac{d_{EST}}{\delta_{EST}}$	0,804		0,831		0,817	

The average value of the above $\frac{d_{EST}}{\delta_{EST}}$ is approximately 0,82. Then, the value of d_{EST} for the stationary part of axial flow machines is determined as follows:

$$d_{EST} = \delta_{EST} \times 0{,}82 = 0{,}015 \times 0{,}82 = 0{,}012\ 3 \qquad \text{(C.15)}$$

C.9 Summary of the scale effect formula for axial flow machines

As explained in C.2, the step-up formula for runner blades (Equation C.3) can be expressed by an equation same as Equation C.5 for stationary part or Equation 8 for radial flow turbines. Then, Equation 8 can be applied commonly to runner blades and stationary parts of axial flow machines.

The parameters to calculate Δ_{ECO} for axial flow machines are given in the Table C.2 below:

Table C.2 – Parameters to obtain Δ_{ECO} for axial flow machines

CO	d_{ECOref}	κ_{uCO}
RU	0,024 5	1,29 *
ST	0,012 3	0,19
* The value marked by * is the one originally defined as κ^*_{uRU}.		
NOTE The modified flow velocity factor for runner blades, κ^*_{uRU} is hereafter expressed as κ_{uRU} to use the common symbol to those for radial flow machines or for stationary part of axial flow machines.		

The roughness value for stationary part, Ra_{ST}, shall be the arithmetical mean value of those measured at guide vanes and stay vanes (see Equation C.7).

After Δ_{ERU} and Δ_{EST} are obtained by the above formula, the step-up amount of hydraulic efficiency for a whole machine is obtained by Equation C.8.

The values of δ_{ECOref}, κ_{uCO}, κ_{dCO} and d_{ECOref} set out in Annex C are substantiated by analytical or experimental data for the specific speed range of $0{,}25 \leq N_{QE} \leq 0{,}70$.

Outside of these ranges, their values may not be correct. Therefore, if the step-up equations in this standard are applied to the evaluation of the contractual model test results beyond the above specific speed range, prior agreement shall be made among the concerned parties.

C.10 Direct step-up for a whole turbine

Similar to the direct step-up method for radial flow machines, the direct step-up method for axial flow machines is shown hereafter.

To represent the whole machine, the reference flow velocity index κ_{u0} and the representative roughness of the machine Ra_0 need to be defined.

As observed in Figure C.1, the scalable loss in runner is twice as large as of stationary part. By considering this, κ_{u0} and Ra_0 are defined as follows:

$$\kappa_{u0} = \frac{2\kappa_{uRU} + \kappa_{uST}}{3} = \frac{2 \times 1{,}29 + 0{,}19}{3} \approx 0{,}92 \qquad \text{(C.16)}$$

$$Ra_0 = \frac{2Ra_{RU} + Ra_{ST}}{3} \qquad \text{(C.17)}$$

As explained in B.3, if $\left(\frac{\kappa_{uCO}}{\kappa_{u0}} \frac{Ra_{COM}}{D_M}\right)$ of runner and stationary part of the model can be regarded as the same and represented commonly by $\left(\frac{Ra_{0M}}{D_M}\right)$ and, also, those for the prototype can be represented by $\left(\frac{Ra_{0P}}{D_P}\right)$, the following formula for direct step-up for a whole turbine can be derived:

$$\begin{aligned}
\Delta_E &= \sum d_{ECOref}\left[\left(4\times10^5\,\kappa_{uCO}\,\frac{Ra_{COM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0,2} - \left(4\times10^5\,\kappa_{uCO}\,\frac{Ra_{COP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0,2}\right] \\
&= \sum d_{ECOref}\left[\left(4\times10^5\,\kappa_{u0}\,\frac{\kappa_{uCO}}{\kappa_{u0}}\,\frac{Ra_{COM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0,2} - \left(4\times10^5\,\kappa_{u0}\,\frac{\kappa_{uCO}}{\kappa_{u0}}\,\frac{Ra_{COP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0,2}\right] \\
&= d_{Eref}\left[\left(4\times10^5\,\kappa_{u0}\,\frac{Ra_{0M}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0,2} - \left(4\times10^5\,\kappa_{u0}\,\frac{Ra_{0P}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0,2}\right]
\end{aligned}$$

(C.18)

where

$$d_{Eref} = d_{ERUref} + d_{ESTref} = 0{,}036\,8 \qquad \text{(C.19)}$$

Annex D
(informative)

Scale effect on disc friction loss

D.1 Loss coefficient formula for disc friction

As demonstrated in Annex B, a new explicit formula to give loss coefficient for pipe flow, which is proposed by Nichtawitz, gives almost the same value as the implicit Colebrook formula (see Figure B.1). It is reasonable now to assume that a similar formula is also able to describe the disc friction loss coefficient.

General loss coefficient formula as proposed by Nichtawitz [9] is:

$$C_m = C_{m0}\left[m\left(A_T\frac{k_{ST}}{a}+\frac{Re_0}{Re_T}\right)^n+(1-m)\right] \quad \text{(D.1)}$$

However, for the case of disc flow, an approximation formula similar to the Colebrook formula does not exist. Therefore, the above general formula was applied to physical model measurements done by Fukuda [18] and others [15,19]. It was found that a best fit to the test results could be reached by the following coefficients:

C_{m0}=0,001 9

Re_0=7 $\times 10^6$

A_T = 1,5 $\times 10^4$

m = 0,85

n = 0,2

where

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

D_d maximum diameter of the runner crown or band, whichever larger (m);

κ_T dimension factor of the disc $\kappa_T = \frac{2a}{D} = \frac{D_d}{D} \quad \therefore\ a = \frac{\kappa_T \times D}{2}$;

Re_T Reynolds number of the disc $Re_T = \frac{a^2 \times \omega}{\nu} = \frac{a^2 \times \omega}{D \times u} Re = \frac{2a^2}{D^2} Re = \frac{1}{2}\kappa_T{}^2 Re$;

ω angular velocity of the disc (rad/s).

NOTE 1 Since disc friction loss is proportional to 5th power of the disc diameter, the larger diameter of either runner crown or runner band has dominant influence on the disc friction loss. Therefore, the dimension factor for the disc, κ_T, is defined by the larger diameter of either the runner crown or the runner band.

Then, the basic equation for disc friction loss coefficient is given as follows:

$$C_m = C_{m0}\left[0{,}85\left(1{,}5\times10^4\,\frac{k_{ST}}{a}+\frac{Re_0}{Re_T}\right)^{0{,}2}+0{,}15\right]$$

$$= C_{m0}\left[0{,}85\left(7{,}5\times10^4\,\frac{2\times Ra_T}{\kappa_T\times D}+\frac{2}{{\kappa_T}^2}\,\frac{Re_0}{Re}\right)^{0{,}2}+0{,}15\right]$$

$$\therefore\quad C_m = C_{m0}\left[0{,}85\left(\frac{2}{{\kappa_T}^2}\right)^{0{,}2}\left(7{,}5\times10^4\,\kappa_T\,\frac{Ra_T}{D}+\frac{Re_0}{Re}\right)^{0{,}2}+0{,}15\right]$$

$$= C_{m0}\left[\left(\frac{0{,}976}{{\kappa_T}^{0{,}4}}\right)\left(7{,}5\times10^4\,\kappa_T\,\frac{Ra_T}{D}+\frac{Re_0}{Re}\right)^{0{,}2}+0{,}15\right] \tag{D.2}$$

$$= C_{m0}\left(\frac{0{,}976}{{\kappa_T}^{0{,}4}}\right)\left[\left(7{,}5\times10^4\,\kappa_T\,\frac{Ra_T}{D}+\frac{Re_0}{Re}\right)^{0{,}2}+0{,}154{\kappa_T}^{0{,}4}\right]$$

where

k_{ST} sand roughness of the disc averaged on both sides of runner and stationary part (m) $k_{ST} = 5\times Ra_T$;

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 (m).

$$Ra_T = \frac{2\times Ra_{TR}+Ra_{TS}}{3} \tag{D.3}$$

Ra_{TR} average arithmetical mean roughness measured near the outer periphery of the runner crown and band (m);

Ra_{TS} average arithmetical mean roughness measured on the stationary parts facing to the measuring points of the runner crown and band (m).

NOTE 2 The experiments carried out by Kurokawa [3, 20] indicate that the roughness of the rotating part has more dominant effect on the disc friction loss torque of the runner than the roughness of the stationary part. The roughness effect on disc friction loss can be represented by the weighted mean value of the roughness of both sides as shown by Equation D.3.

D.2 Step-up formula for power efficiency

As shown in A.2 4), the step-up formula for power efficiency is expressed as shown below:

$$\Delta_T = \frac{\Delta\eta_T}{\eta_{TM}} = \delta_{Tref}\left(\frac{C_{mM}-C_{mP}}{C_{mref}}\right) \tag{D.4}$$

The friction loss coefficient C_{mref} for the reference model with $Ra_T \approx 0$ at reference Reynolds number $Re_{ref} = 7\times10^6$ is obtained as follows:

$$C_{mref} = C_{m0}\left(\frac{0{,}976}{{\kappa_T}^{0{,}4}}\right)\left(1+0{,}154{\kappa_T}^{0{,}4}\right) \tag{D.5}$$

By replacing C_{mM} and C_{mP} in Equation D.4 by Equation D.2 and C_{mref} by Equation D.5, we obtain,

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

$$= \delta_{Tref}\left[\frac{\left(5A_T \times \kappa_T \frac{Ra_{TM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0,2} - \left(5A_T \times \kappa_T \frac{Ra_{TP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0,2}}{1+0,154\,\kappa_T{}^{0,4}}\right] \quad \text{(D.6)}$$

$$= d_{Tref}\left[\left(7,5\times10^4\,\kappa_T \frac{Ra_{TM}}{D_M} + \frac{7\times10^6}{Re_M}\right)^{0,2} - \left(7,5\times10^4\,\kappa_T \frac{Ra_{TP}}{D_P} + \frac{7\times10^6}{Re_P}\right)^{0,2}\right]$$

where

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

D.3 Standardized dimension factor κ_T and disc friction loss index d_{Tref}

1) Disc friction loss ratio δ_{Tref}

Based on the experimental studies conducted by Kurokawa [12], the disc friction losses for Francis turbines and pump-turbines of average design are estimated as follows:

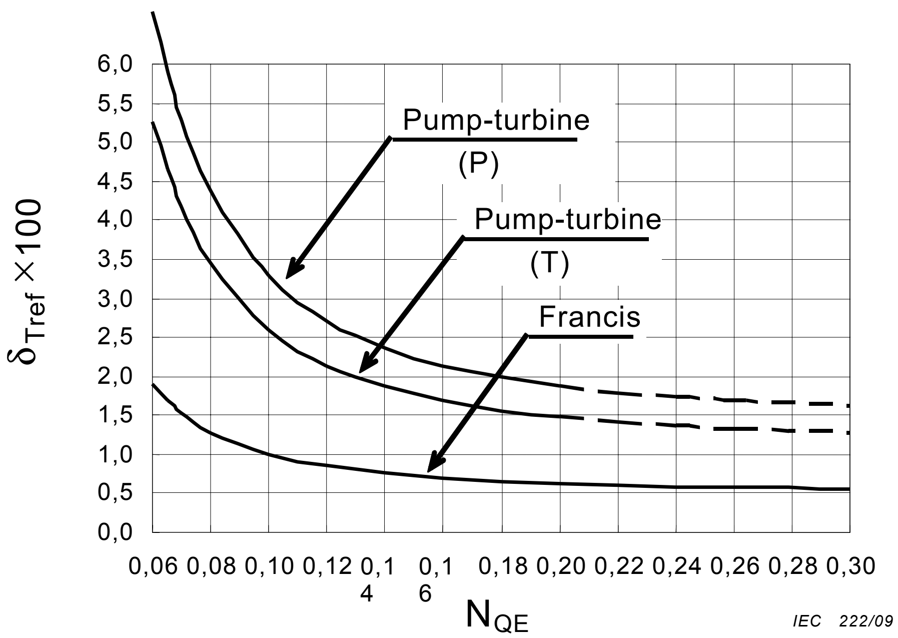


Figure D.1 – Disc friction loss ratio δ_{Tref}

These curves are approximated by the following formulae.

Francis turbines: $$\delta_{Tref} = \frac{\left(0{,}5 + \frac{0{,}005}{N_{QE}^2}\right)}{100} \text{ for } 0{,}06 \le N_{QE} \le 0{,}30 \quad \text{(D.7)}$$

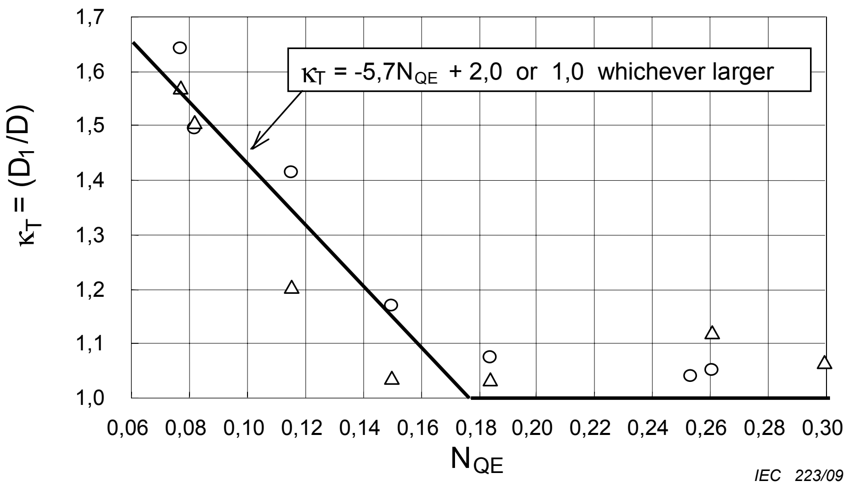
Pump-turbines (turbine mode) (T): $$\delta_{Tref} = \frac{\left(1{,}1 + \frac{0{,}015}{N_{QE}^2}\right)}{100} \text{ for } 0{,}06 \le N_{QE} \le 0{,}20 \quad \text{(D.8)}$$

Pump-turbines (pump mode) (P): $$\delta_{Tref} = \frac{\left(1{,}4 + \frac{0{,}019}{N_{QE}^2}\right)}{100} \text{ for } 0{,}06 \le N_{QE} \le 0{,}20 \quad \text{(D.9)}$$

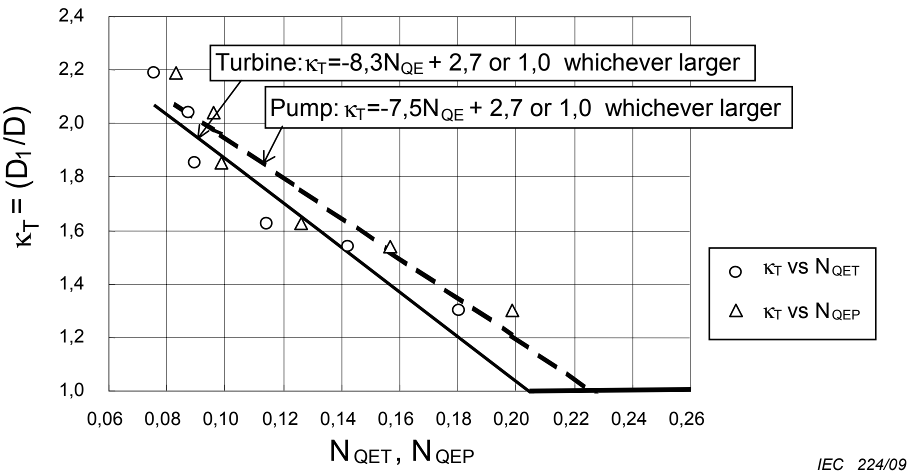
NOTE The above equations are not substantiated by analytical or experimental data beyond the specific speed range specified for each formula. However, these equations may be extrapolated beyond the specified range and used for the step-up calculation of contractual model test results by mutual agreement of the concerned parties.

2) Dimension factor of the disc κ_T

The values of κ_T calculated for some typical models are plotted against specific speed and shown below. For convenience, the plots are approximated by linear equations.



a) Dimension factor κ_T for Francis turbine



b) Dimension factor κ_T for pump-turbine (valid for $N_{QET} = N_{QEP} = 0,06 – 0,20$)

Figure D.2 – Dimension factor κ_T

3) Disc friction loss index d_{Tref}

By combining δ_{Tref} and κ_T, we can obtain the values of d_{Tref} as a function of specific speed. They are shown on Figure D.3. For simplification, they are approximated by hyperbolic equations.

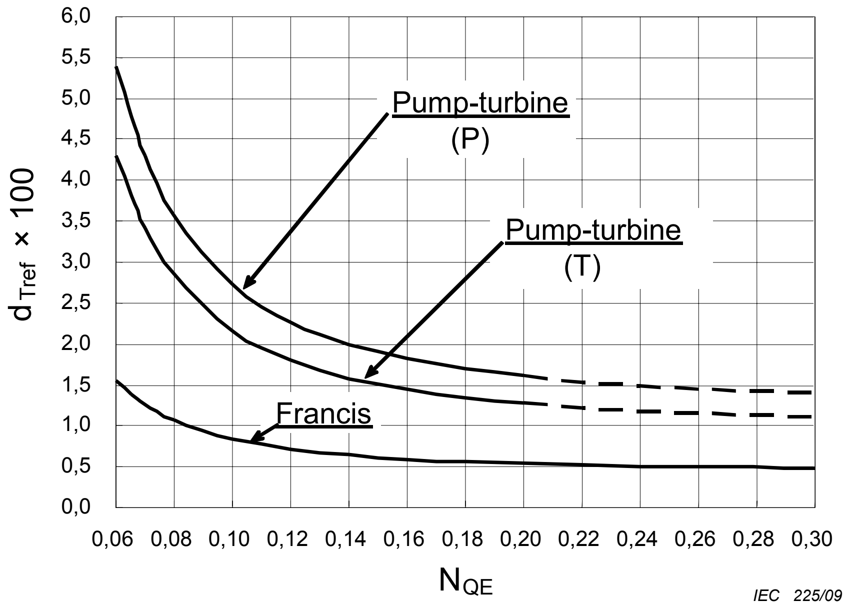


Figure D.3 – Disc friction loss index d_{Tref}

These curves are approximated by the following formulae.

Francis turbines: $$d_{Tref} = \frac{\left(0{,}44 +, \dfrac{0{,}004}{N_{QE}^{2}}\right)}{100} \quad \text{for } 0{,}06 \le N_{QE} \le 0{,}30 \qquad \text{(D.10)}$$

Pump-turbines (turbine mode): $$d_{Tref} = \frac{\left(0{,}97 + \dfrac{0{,}012}{N_{QE}^{2}}\right)}{100} \quad \text{for } 0{,}06 \le N_{QE} \le 0{,}20 \qquad \text{(D.11)}$$

Pump-turbines (pump mode): $$d_{Tref} = \frac{\left(1{,}23 + \dfrac{0{,}015}{N_{QE}^{2}}\right)}{100} \quad \text{for } 0{,}06 \le N_{QE} \le 0{,}20 \qquad \text{(D.12)}$$

NOTE The above equations are not substantiated by analytical or experimental data beyond the specific speed range specified for each formula. However, these equations may be extrapolated beyond the specified range and used for the step-up calculation of contractual model test results by mutual agreement of the concerned parties.

Annex E
(informative)

Leakage loss evaluation for non homologous seals

E.1 Loss coefficient of runner seal

In the main text of this standard, only the step-up for a homologous seal is given ($\Delta\eta_Q = 0$). However, due to the difficulty in manufacturing the model or to the structural restraint for installation of sensors, etc., the model seal design often cannot meet with the requirement given in Table 3. In that case, the procedure given in this annex may be used for the evaluation of the volumetric efficiency of the prototype upon the mutual agreement of the concerned parties.

An equivalent dimensionless loss coefficient of the seal, K, which is defined by the following formula is introduced:

$$K = \left[\sum_i \left(\frac{\zeta_{ki}}{A_i^2}\right) + \sum_j \left(\frac{\zeta_{fj}}{A_j^2}\right)\right] \times D^4$$
$$\propto \left[\zeta_{k1}\left(\frac{1}{R_1 \times c}\right)^2 + \zeta_{k2}\left(\frac{1}{R_2 \times c}\right)^2 + \sum_j \left[\zeta_{ksj}\left(\frac{1}{R_{sj} \times c}\right)^2\right] + \sum_j \left[\zeta_{fj}\left(\frac{1}{R_{fj} \times c}\right)^2\right]\right] \times D^4 \quad \text{(E.1)}$$

where

ζ loss coefficient $\zeta = \dfrac{E}{(q/A)^2/2}$

q leakage flow through the seal concerned

NOTE It is not the total leakage flow through both seals on crown and band.

A cross sectional area of the seal clearance

R radius of seal

c radial clearance of seal

i representing 1, 2 or s

j number of steps/grooves or seal clearances

subscripts:

k kinetic loss

f friction loss or values of each seal clearance

1 values at the inlet of the seal

2 values at the outlet of the seal

s values at the intermediate step or groove

When the loss coefficient K is calculated by the above formula, the loss coefficients ζ are prescribed in this standard as follows:

$$\left.\begin{aligned}
&\text{Inlet loss of the seal:} && \zeta_{k1} = 0{,}5\\
&\text{Outlet loss of the seal:} && \zeta_{k2} = 1{,}0\\
&\text{Intermediate step or groove:} && \zeta_{ks} = 1{,}0\\
&\text{Friction loss:} && \zeta_f = \lambda_C \frac{L}{2c}
\end{aligned}\right\}\qquad \text{(E.2)}$$

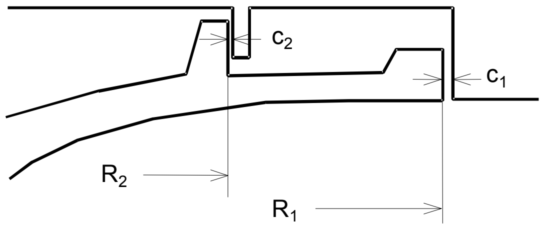
where

λ_C friction loss coefficient. λ_C = 0,04

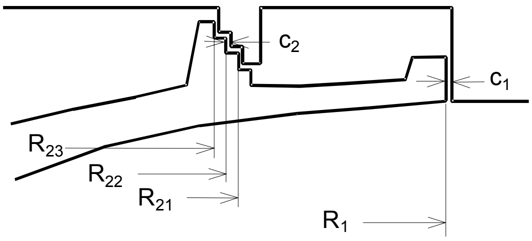
NOTE Scale effect on λ_C is neglected.

L length of each seal clearance

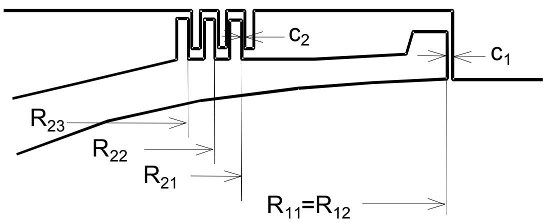
Some typical examples of the runner seal design on the crown side are illustrated on Figure E.1 and those on the band side on Figure E.2.



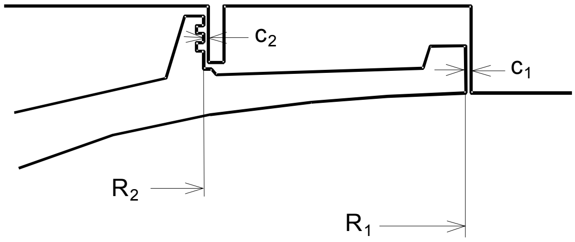
$$K_{crown} \approx (\zeta_{k1} + \zeta_f + \zeta_{k2}) \left[\left(\frac{1}{R_1 c_1} \right)^2 + \left(\frac{1}{R_2 c_2} \right)^2 \right] \times D^4$$



$$K_{crown} \approx \left[\begin{array}{l} (\zeta_{k1} + \zeta_f + \zeta_{k2}) \left(\frac{1}{R_1 c_1} \right)^2 + (\zeta_{k1} + \zeta_f + \zeta_{ks}) \left(\frac{1}{R_{21} c_2} \right)^2 \\ \quad + (\zeta_f + \zeta_{ks}) \left(\frac{1}{R_{22} c_2} \right)^2 + (\zeta_f + \zeta_{k2}) \left(\frac{1}{R_{23} c_2} \right)^2 \end{array} \right] \times D^4$$



$$K_{crown} \approx \left[\begin{array}{l} (\zeta_{k1} + \zeta_f + \zeta_{k2}) \left(\frac{1}{R_1 c_1} \right)^2 + (\zeta_{k1} + \zeta_f + \zeta_{ks}) \left(\frac{1}{R_{21} c_2} \right)^2 \\ \quad + (\zeta_f + \zeta_{ks}) \left(\frac{1}{R_{22} c_2} \right)^2 + (\zeta_f + \zeta_{k2}) \left(\frac{1}{R_{23} c_2} \right)^2 \end{array} \right] \times D^4$$



$$K_{crown} \approx \left[(\zeta_{k1} + \zeta_f + \zeta_{k2}) \left(\frac{1}{R_1 c_1} \right)^2 + \left(\zeta_{k1} + 2\zeta_{ks} + \sum_3 \zeta_f + \zeta_{k2} \right) \left(\frac{1}{R_2 c_2} \right)^2 \right] \times D^4$$

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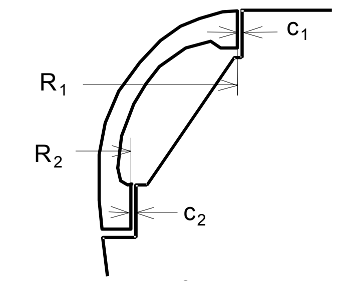
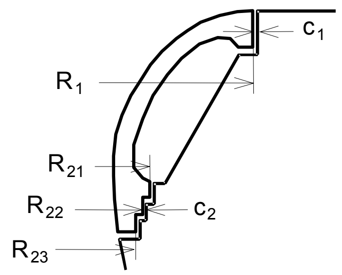
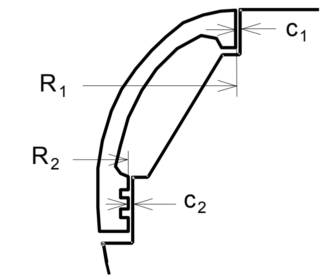
Figure E.1 – Examples of typical design of runner seals (crown side)



$$K_{band} \approx \left[\left(\zeta_{k1} + \zeta_f + \zeta_{k2}\right)\left(\frac{1}{R_1 c_1}\right)^2 + \left(\zeta_{k1} + \zeta_f + \zeta_{k2}\right)\left(\frac{1}{R_2 c_2}\right)^2 \right] \times D^4$$



$$K_{band} \approx \left[\begin{array}{l} \left(\zeta_{k1} + \zeta_f + \zeta_{k2}\right)\left(\frac{1}{R_1 c_1}\right)^2 + \left(\zeta_{k1} + \zeta_f + \zeta_{ks}\right)\left(\frac{1}{R_{21} c_2}\right)^2 \\ + \left(\zeta_f + \zeta_{ks}\right)\left(\frac{1}{R_{22} c_2}\right)^2 + \left(\zeta_f + \zeta_{k2}\right)\left(\frac{1}{R_{23} c_2}\right)^2 \end{array} \right] \times D^4$$



$$K_{band} \approx \left[\left(\zeta_{k1} + \zeta_f + \zeta_{k2}\right)\left(\frac{1}{R_1 c_1}\right)^2 + \left(\zeta_{k1} + 2\zeta_{ks} + \sum_3 \zeta_f + \zeta_{k2}\right)\left(\frac{1}{R_2 c_2}\right)^2 \right] \times D^4$$

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Figure E.2 – Examples of typical design of runner seals (band side)

The value of the loss coefficient K given by Equation E.1 is calculated individually for outer or inner seals on the runner crown and those on the runner band, respectively. Then total loss coefficient K for the whole machine is calculated by the following equation:

$$K = \frac{K_c \times K_b}{\left(\sqrt{K_c} + \sqrt{K_b}\right)^2} \tag{E.3}$$

where,

K_c sum of the dimensionless loss coefficient for the seals on the runner crown;

K_b sum of the dimensionless loss coefficient for the seals on the runner band;

K representative dimensionless loss coefficient for the whole machine.

NOTE Equation E.3 is derived by assuming that the differential pressure across the runner seals on both crown and band sides are identical. It disregards pressure gradient in the space between runner and stationary part and, also, the loss head in balance holes or equalizer pipes.

If the values of differential pressure across the runner seals on both sides are not identical, this equation is not applicable. In such case, detailed analysis is required.

E.2 General formula to obtain $\Delta\eta_Q$ for non-homologous seal

By using the representative loss coefficients for the model and the prototype, a general formula for $\Delta_Q = \dfrac{\Delta\eta_Q}{\eta_{QM}}$ can be written as follows (see A.2 3)):

$$\left.\begin{aligned}
&\text{For turbine: } \Delta_Q = \frac{\Delta\eta_Q}{\eta_{QM}} = \frac{\eta_{QP}}{\eta_{QM}}\left(1-\eta_{QM}\right)\left[1-\left(\frac{\zeta_{kM}+\zeta_{fM}}{\zeta_{kP}+\zeta_{fP}}\right)^{0,5}\right] \cong \left(1-\eta_{QM}\right)\left[1-\left(\frac{K_M}{K_P}\right)^{0,5}\right]\\
&\text{For pump: } \Delta_Q = \frac{\Delta\eta_Q}{\eta_{QM}} = \left(1-\eta_{QM}\right)\left[1-\left(\frac{\zeta_{kM}+\zeta_{fM}}{\zeta_{kP}+\zeta_{fP}}\right)^{0,5}\right] \cong \left(1-\eta_{QM}\right)\left[1-\left(\frac{K_M}{K_P}\right)^{0,5}\right]
\end{aligned}\right\} \qquad \text{(E.4)}$$

where

K_M representative loss coefficient for the model;

K_P representative loss coefficient for the prototype.

In the above formula, η_{QM} is considered to be 0,99 in this standard.

E.3 Evaluation of scale effect in case of a homologous straight seal

In case of a homologous seal with normal straight seal design,

$$\frac{D_M^2}{A_{iM}} \equiv \frac{D_M^2}{A_{aveM}} \equiv \frac{D_P^2}{A_{iP}} \equiv \frac{D_P^2}{A_{aveP}}$$

Therefore,

$$\left(\frac{K_M}{K_P}\right)^{0,5} = \left[\frac{\sum_i\left(\dfrac{\zeta_{kiM}}{A_{iM}^2}\right)+\dfrac{\zeta_{fM}}{A_{aveM}^2}}{\sum_i\left(\dfrac{\zeta_{kiP}}{A_{iP}^2}\right)+\dfrac{\zeta_{fP}}{A_{aveP}^2}}\right]^{0,5}\left(\frac{D_M}{D_P}\right)^2 = \left(\frac{\sum_i\zeta_{kiM}+\zeta_{fM}}{\sum_i\zeta_{kiP}+\zeta_{fP}}\right)^{0,5}$$

In normal straight seal design, $\left(\zeta_f/\sum\zeta_{ki}\right)\approx 0,5\cdots1,5$

If the scale effect on ζ_f is considered,

$\left(Re_P/Re_M\right)\approx 5\cdots40$ (in usual model test condition)

Then, this would give $\left(\zeta_{fP}/\zeta_{fM}\right)\approx\left(Re_P/Re_M\right)^{-0,2}\approx\left(5\cdots40\right)^{-0,2}\approx 0,5\cdots0,7$.

Since the kinetic loss is non-scalable, $\sum\zeta_{kiP} = \sum\zeta_{kiM}$.

Therefore, in case of homologous straight seal when scale effect on ζ_f is considered:

$$\left(\frac{K_M}{K_P}\right)^{0,5} = \left(\frac{\sum\zeta_{kiM} + \zeta_{fM}}{\sum\zeta_{kiP} + \zeta_{fP}}\right)^{0,5} = \left[\frac{\sum\zeta_{kiM} + \zeta_{fM}}{\sum\zeta_{kiM} + \zeta_{fM}\left(\zeta_{fP}/\zeta_{fM}\right)}\right]^{0,5}$$

$$\cong \left[\frac{1+(0,5\cdots1,5)}{1+(0,5\cdots1,5)(0,5\cdots0,7)}\right]^{0,5} \cong \left[\left(\frac{1,5}{1,25\cdots1,35}\right)\cdots\left(\frac{2,5}{1,75\cdots2,05}\right)\right]^{0,5}$$

$$\cong (1,11\cdots1,43)^{0,5} \cong 1,05\cdots1,20$$

Since $(1-\eta_Q) \cong 0,01$, $\Delta\eta$ for homologous straight seal may be estimated as follows:

$$\Delta_Q = \frac{\Delta\eta_Q}{\eta_{Qm}} = (1-\eta_{QM})\left[1-\left(\frac{K_M}{K_P}\right)^{0,5}\right]$$

$$\cong 0,01\times\left[1-(1,05\cdots1,20)\right] = -(0,000\,5\cdots0,002\,0)$$

or

$$\Delta\eta_Q = -(0,05\cdots0,20)\ \%$$

This is regarded as "0 %" in this standard for simplification.

E.4 Straight seal with non-homologous radial clearance

As an example, the case where the radii of the seal are homologous but the radial clearances are not homologous is examined. Then,

$$\frac{D_M}{R_{iM}} = \frac{D_M}{R_{aveM}} = \frac{D_P}{R_{iP}} = \frac{D_P}{R_{aveP}} \tag{E.5}$$

The term $\left(\frac{K_M}{K_P}\right)^{0,5}$ appeared in Equation E.4 can be written as follows:

$$\left(\frac{K_M}{K_P}\right)^{0,5} = \left[\frac{\dfrac{\zeta_{k1}}{(R_{1M}c_M)^2} + \dfrac{\zeta_{k2}}{(R_{2M}c_M)^2} + j\times\dfrac{\zeta_{ks}}{(R_{sM}c_M)^2} + \dfrac{\zeta_f}{(R_{aveM}c_M)^2}}{\dfrac{\zeta_{k1}}{(R_{1P}c_P)^2} + \dfrac{\zeta_{k2}}{(R_{2P}c_P)^2} + j\times\dfrac{\zeta_{ks}}{(R_{sP}c_P)^2} + \dfrac{\zeta_f}{(R_{aveP}c_P)^2}}\right]^{0,5} \left(\frac{D_M}{D_P}\right)^2$$

$$= \left[\frac{\dfrac{\zeta_{k1}D_M^2}{R_{1M}^2} + \dfrac{\zeta_{k2}D_M^2}{R_{2M}^2} + j\times\dfrac{\zeta_{ks}D_M^2}{R_{sM}^2} + \dfrac{\zeta_f D_M^2}{R_{aveM}^2}}{\dfrac{\zeta_{k1}D_P^2}{R_{1P}^2} + \dfrac{\zeta_{k2}D_P^2}{R_{2P}^2} + j\times\dfrac{\zeta_{ks}D_P^2}{R_{sP}^2} + \dfrac{\zeta_f D_P^2}{R_{aveP}^2}}\right]\left(\frac{D_M/c_M}{D_P/c_P}\right) \tag{E.6}$$

By Equation E.5, both the numerator for the model and the denominator for the prototype of the ratio in the square bracket of Equation E.6 becomes the same. Then, the above equation is simply expressed as follows;

$$\left(\frac{K_M}{K_P}\right)^{0,5} = \frac{c_P/D_P}{c_M/D_M} \tag{E.7}$$

Therefore,

$$\Delta_Q = \frac{\Delta\eta_Q}{\eta_{QM}} = (1-\eta_{QM})\left[1-\left(\frac{K_M}{K_P}\right)^{0,5}\right] \approx 0,01\times\left[1-\frac{(c_P/D_P)}{(c_M/D_M)}\right] \tag{E.8}$$

Hence, it is known that, if the radial seal clearance of the prototype is relatively smaller compared with the model, the volumetric efficiency of the prototype becomes higher than the model.

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