New JSME standard S008 “Performance Conversion Method for Hydraulic Turbines and Pump-Turbines”

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Abstract. JSME Standard S008 “Performance Conversion Method for Hydraulic Turbines and Pump-Turbines” is now being revised and will be published in 2016. This new revision follows the main theory of previous version S008-1999. It enables us to convert the performance of each flow passage component of spiral case, stay vane, guide vane, runner and draft tube of model turbines and pump-turbines to that of prototypes with one-step calculation. The relevant values needed for the performance conversion, e.g. dimension factor, flow velocity factor, relative scalable loss of components δECO, etc. are newly organized as functions of specific speeds of turbines and pump-turbines using polynomial expressions. Additional data for high specific speed turbines are included. The resultant factors for conversion of the specific energy efficiency scale factor FE, the discharge efficiency scale factor FQ and the power efficiency scale factor FT are determined by considering friction coefficient ratio for prototype to the model.

1. The object of revision

The previous edition of JSME S008 “Performance Conversion Method for Hydraulic Turbines and Pump-Turbines” [1] was published in 1999. This edition prescribes the performance conversion method to convert the model performance tested accurately under non-cavitating condition to prototype performance considering the scale effect caused by the differences of Reynolds number and surface roughness between the model and the prototype. The conversion formulae use the relative scalable loss of the model instead of the loss distribution coefficient and the friction coefficient ratio between the model and the prototype. This method takes into account of differences in machine type, specific speed...
and operating condition. The specific energy, discharge and power characteristics can be converted as well as efficiency characteristics. In 2009, IEC62097 [2] was published. It enables us to calculate component-wise step-up from the model performance to prototype performance. It provides basic formulae for the step-up of hydraulic efficiency with three efficiency scale factors and for shifting of performance following essentially the basis of JSME S008-1999, and refers to JSME S008-1999 for the values of relative scalable loss of machines.

In general, surface roughness in a spiral case, a distributor and a draft tube can be considered as the almost same value between the model and the prototype due to painting. In JSME S008-1999, the effect of the difference of surface roughness between the model and the prototype is calculated using the roughness ratio dependence coefficient, the ratio of having different surface roughness between the model and the prototype, for simplifying the conversion formulae. For the physical appropriateness for flow phenomena in machines, the difference of surface roughness should be taken into account for each component as well as its Reynolds number. In addition, refurbishment and replacement of an aged component are expected. Therefore, it is hoped that JSME S008 comes to be applicable to each component while enhancing its physical appropriateness.

2. Main points of the new edition of JSME S008
The main points of the new edition of JSME S008 are as follows:

(1) Component-wise conversion: In S008-1999, the scale effect was calculated for a whole machine. In this edition, the conversion formulae are improved so that the performance conversion can be done for each component as IEC62097 can do. For radial flow machines, the calculation of scale effect is implemented for a spiral case, stay vanes, guide vanes, a runner and a draft tube. For diagonal and axial flow machines, the passage from guide vane outlet to runner inlet is added. This improvement enables us to calculate the specific energy efficiency scale factor \( F_E \), the ratio of prototype specific energy efficiency to model specific energy efficiency, considering the difference in surface roughness of each component passage between the prototype and the model without using the roughness ratio dependence coefficient adopted in S008-1999.

(2) Relative scalable loss newly organized as functions of specific speeds: In S008-1999, relative scalable loss was approximated by a linear constant expression as a function of specific speed for each machine type. In this edition, imaginary standard models of turbines and pump-turbines are designed for a practical range of specific speeds on the basis of experiential data of the existing models. The ratio of the hydraulic diameter of flow passage to the reference diameter \( d_{CGO}/D \), the ratio of the length of flow passage to the reference diameter \( l_{CGO}/D \) and the ratio of the flow velocity to the peripheral speed \( v_{CGO}/u \) are given for each component passage as functions of specific speed. Those are corresponding to dimension factor \( \kappa_{CGO} \) and flow velocity factor \( \kappa_{vCGO} \) of IEC 62097-2009. Using these ratios, the relative scalable loss \( \delta_{CGO} \) of each component passage is calculated by one dimensional theory of loss evaluation, and standardized using a polynomial expression as a function of specific speed for each machine type. This standardization plays an important role in enhancing the physical appropriateness for flow phenomena in machines in this conversion method.

(3) Re-evaluation of relative scalable loss in axial flow turbines: There were only few cases evaluated for relative scalable loss in axial flow turbines (Kaplan and Bulb turbines) in S008-1999. In addition, as for Kaplan turbines, the scalable loss from the guide vane outlet to the runner inlet was not included. Therefore, numerical analysis [3] is carried out to calculate the relative scalable loss in each component passage of axial flow turbines in a practical range of specific speed.

(4) Improvement of discharge efficiency scale factor \( F_Q \): The discharge efficiency scale factor \( F_Q \), the ratio of prototype discharge efficiency to model discharge efficiency, was given as a constant value in the previous edition. To improve this, the effect of Reynolds number is taken into account in friction coefficient ratio.
Improvement of power efficiency scale factor $F_T$: The power efficiency scale factor $F_T$, the ratio of prototype power efficiency to model power efficiency, was given as a function of specific speed in the previous edition. The effect of Reynolds number and surface roughness are considered in this edition.

3. Basic formulae and calculation procedure (turbine operation)

This conversion method should be applied to the model test conducted in accordance with JIS/IEC/JEC code or equivalent. A model turbine has a stipulated tolerance to maintain geometrical similarity to the prototype turbine. The angle of guide vanes and runner vanes of the model and the prototype shall be identical. As for surface roughness, the relevant code shall be applied. The test Reynolds number should be close to the reference Reynolds number ($7 \times 10^6$).

In this paper, the basic formulae and calculation procedure are explained only for turbine operation. The prototype hydraulic efficiency $\eta_{hp}$ is converted from the model hydraulic efficiency $\eta_{hm}$ using three efficiency scale factors as follows:

$$\eta_{hp} = F_E F_Q F_T \eta_{hm}$$

(1)

where,

$$F_E = \frac{\eta_{EP}}{\eta_{EM}}, \quad F_Q = \frac{\eta_{EQ}}{\eta_{QM}}, \quad F_T = \frac{\eta_{TP}}{\eta_{TM}}$$

(2)

The prototype performance is converted from the model performance by the following equations.

$$E_p = n_{EDM}^2 D_p^2 F_E^{-1}$$

(3)

$$Q_{ap} = Q_{EDM} D_p^{3/2} E_p^{1/2} F_Q^{-1}$$

(4)

$$P_{mp} = P_{EDM} D_p^{3/2} E_p^{3/2} \rho_p F_E^{3/2} F_T$$

(5)

3.1. Specific energy efficiency scale factor $F_E$

The specific energy efficiency scale factor $F_E$ is given by Eq. (6).

$$F_E = \frac{\eta_{EP}}{\eta_{EM}} = \frac{1}{1 - \sum_{CO} \delta_{ECO} (1 - \lambda_{CO})}$$

(6)

where, $CO$ denotes flow passage components of $SP$, $SV$, $GV$, $RU$ and $DT$ for Francis turbines and Francis pump-turbines, and of $SP$, $SV$, $GV$, $GR$, $RU$, $DT$ for diagonal flow turbines and axial flow turbines. These subscripts are $SP$: spiral case, $SV$: stay vane, $GV$: guide vane, $GR$: flow passage from guide vane outlet to runner inlet, $RU$: runner and $DT$: draft tube. $\lambda_{CO}$ denotes the ratio of prototype friction coefficient to the model friction coefficient of each passage component.

In order to determine the relative scalable loss $\delta_{ECO}$ and friction coefficient ratio $\lambda_{CO}$ for the model test, the design of the imaginary model is employed. First of all, non-dimensional specific speed $N_{QE}$ is calculated by Eq. (7) using the speed factor $n_{EDopt}$ and discharge factor $Q_{EDopt}$ at the optimum operating point of the model test result.
The speed factor \( n_{E_D\text{std}} \) and discharge factor \( Q_{E_D\text{std}} \) of the imaginary model of the standard design are determined as functions of the non-dimensional specific speed \( N_{QE} \) as shown in Figs. 1 and 2.

\[
N_{QE} = n_{E_D\text{opt}} \sqrt{Q_{E_D\text{opt}}}
\]  

(7)

The ratio of the hydraulic diameter of flow passage to the reference diameter \( d_{CO}/D \), the ratio of the length of flow passage to the reference diameter \( l_{CO}/D \) and the ratio of the flow velocity to the peripheral speed \( v_{CO\text{std}}/u \) are also given as functions of \( N_{QE} \) as shown in Figs 3 to 14.
Fig. 7 Reference dimension factor $l_{GV}/D$

Fig. 8 Reference velocity factor $v_{GVstd}/u$

Fig. 9 Reference dimension factor $d_{hGR}/D$

Fig. 10 Reference velocity factor $v_{GRstd}/u$

Fig. 11 Reference dimension factor $l_{RU}/D$

Fig. 12 Reference velocity factor $v_{RUstd}/u$

Fig. 13 Reference dimension factor $d_{hDT}/D$

Fig. 14 Reference velocity factor $v_{DTstd}/u$
The ratio of the flow velocity to the peripheral speed \( \frac{v_{CO}}{u} \) is calculated by Eqs. (8) and (9).

\[
\frac{v_{CO}}{u} = \frac{v_{CO\text{opt}}}{u} \times \left[ \frac{1}{2} + \frac{16}{\pi^2} \left( \frac{Q_{ED\text{opt}}}{n_{ED\text{opt}}} \right)^2 \right]^{1/2} / \left[ \frac{1}{2} + \frac{16}{\pi^2} \left( \frac{Q_{ED\text{std}}}{n_{ED\text{std}}} \right)^2 \right]^{1/2} \tag{8}
\]

( the other COs )

\[
\frac{v_{CO}}{u} = \frac{v_{CO\text{std}}}{u} \times \left( \frac{Q_{ED\text{opt}}}{Q_{ED\text{std}}} \right) \left( \frac{n_{ED\text{opt}}}{n_{ED\text{std}}} \right) \tag{9}
\]

The standard relative scalable loss \( \delta_{ECO\text{std}} \) is given by the function of the non-dimensional specific speed \( N_{QE} \) as a value at \( n_{ED\text{std}}, Q_{ED\text{std}} \) and \( R_{eref} \) with hydraulically smooth passage surface. The relative scalable loss \( \delta_{ECO\text{opt}} \) are obtained from the results of one dimensional loss analysis of each passage component and shown in Figs.15 through 20. The standard relative scalable loss of a whole machine \( \delta_{Estd} \) is shown in Fig. 21.

The friction coefficients are calculated both at the reference Reynolds number \( R_{eref} \) and at the model test Reynolds numbers \( R_{EM} \). The friction coefficients \( \lambda_{COM} \) of spiral case, passage from guide vane outlet to runner inlet and draft tube are calculated using the equation of a pipe given by Eq. (10) for a hydraulically smooth surface and Eq. (11) for a rough surface, while the coefficients \( C_{fCOM} \) of stay vane, guide vane and runner using the equation of a flat plate by Eq. (12) for a hydraulically smooth surface and Eq. (13) for a rough surface. The component Reynolds numbers are calculated by \( R_{COM\text{pipe}}= (d_{CO}/D)(v_{CO}/u)R_{EM} \) for a pipe and \( R_{COM\text{flat}}= (l_{CO}/D)(v_{CO}/u)R_{EM} \) for a flat plate respectively. The equivalent sand roughness \( k_s \) are used in Eqs. (11) and (13). Those are calculated based on the relation between equivalent sand roughness \( k_s \) and arithmetic mean roughness \( R_a \). The larger of two friction coefficients of a smooth surface and of a rough surface is adopted. The reference friction coefficients \( \lambda_{COM\text{ref}} \) and \( C_{fCOM\text{ref}} \) are calculated under the condition of smooth surface and the reference Reynolds number \( R_{COM\text{ref}} \) by Eqs.(10) or (12) substituting \( R_{COM\text{ref}} \) instead of \( R_{COM} \).

\[
\frac{1}{\sqrt{\lambda_{COM}}} = 2.0\log_{10} \left( R_{COM} \sqrt{\lambda_{COM}} \right) - 0.8 \tag{10}
\]

\[
\lambda_{COM} = \left( 2 \log_{10} \frac{d_{COM}}{k_s_{COM}} / 2 + 1.74 \right)^{-2} \tag{11}
\]

\[
C_{fCOM} = 0.455 / \left( \log_{10} \left( R_{COM} \right) \right)^{2.58} \tag{12}
\]

\[
C_{fCOM} = 1 / \left( \left[ 1.89 - 1.62 \log_{10} \left( k_s_{COM} / l_{COM} \right) \right]^{2.5} \right) \tag{13}
\]

The relative scalable loss of each component \( \delta_{ECO} \) in Eq. (6) can be calculated by Eqs. (14) to (16) at each test operating condition by considering the test Reynolds number \( R_{EM} \) and ratio of discharge factor \( Q_{ED}/Q_{ED\text{opt}} \). In these equations, \( K_{FE} \) gives the ratio of the relative scalable loss at an operating condition to that at the optimum point, as the same way as the previous edition of JSME S008[1].
Fig. 15 Relative scalable loss $\delta_{\text{ESPstd}}$

Fig. 16 Relative scalable loss $\delta_{\text{ESVstd}}$

Fig. 17 Relative scalable loss $\delta_{\text{EGVstd}}$

Fig. 18 Relative scalable loss $\delta_{\text{EGRstd}}$

Fig. 19 Relative scalable loss $\delta_{\text{ERUstd}}$

Fig. 20 Relative scalable loss $\delta_{\text{EDTstd}}$

Fig. 21 Relative scalable loss $\delta_{\text{Estd}}$
\( \delta_{ECO} = K_{FE} \left( \lambda_{COM} / \lambda_{COMref} \right) \delta_{ECOstd} \left( Q_{EDopt} / Q_{EDstd} \right)^2 \) \hspace{1cm} (14)

\( \delta_{ECO} = K_{FE} \left( C_{fCOM} / C_{fCOMref} \right) \delta_{ECOstd} \left( Q_{EDopt} / Q_{EDstd} \right)^2 \) \hspace{1cm} (15)

\( \delta_{ERU} = K_{FE} \left( C_{fCOM} / C_{fCOMref} \right) \delta_{ERUstd} \left( \frac{\pi^2}{2} n_{EDopt}^2 + \frac{8}{\pi^2} Q_{EDopt}^2 \right) / \left( \frac{\pi^2}{2} n_{EDstd}^2 + \frac{8}{\pi^2} Q_{EDstd}^2 \right) \) \hspace{1cm} (16)

where,

for Francis turbines and Francis pump-turbines[1]:

\[ K_{FE} = 1.0 \ \text{at} \ Q_{ED} / Q_{EDopt} \leq 1.0 \] \hspace{1cm} (17)

\[ K_{FE} = 0.91 + \left( Q_{ED} / Q_{EDopt} - 0.7 \right)^2 \ \text{at} \ Q_{ED} / Q_{EDopt} > 1.0 \] \hspace{1cm} (18)

for diagonal flow turbines and axial flow turbines[1]:

\[ K_{FE} = \left( n_{ED} / n_{EDopt} \right)^2 / \left( Q_{ED} / Q_{EDopt} \right)^{0.5} \] \hspace{1cm} (19)

The friction coefficient of each flow passage of the prototype is calculated in the same manner as the model. The friction coefficients of spiral case, passage from guide vane outlet to runner inlet and draft tube are calculated by the equation for a pipe (\( \lambda_{COM} \)), while stay vane, guide vane and runner by the equation for a flat plate (\( C_{fCOM} \)). The friction coefficient ratio \( A_{CO} \) in Eq. (6) can be calculated by Eq. (20) or (21).

\( \lambda_{CO} = \lambda_{COR} / \lambda_{COM} \) \hspace{1cm} (20)

\( A_{CO} = C_{fCOR} / C_{fCOM} \) \hspace{1cm} (21)

3.2. Discharge efficiency scale factor \( F_Q \)

The discharge efficiency scale factor \( F_Q \) for Francis turbines and Francis pump-turbines is given by Eq. (22) as equal to that of the optimum point.

\[ F_Q = F_{Qopt} = \frac{\eta_{Qopt}}{\eta_{QMopt} + A_Q \left( 1 - \eta_{QMopt} \right)} \] \hspace{1cm} (22)

where, \( \eta_{QMopt} \) denotes the discharge efficiency at the optimum point given as a function of the non-dimensional specific speed \( N_{QE} \), and \( A_Q \) friction coefficient ratio is given by a function of the ratio of Reynolds number \( Re_{Mopt} / Re_{P} \) and the non-dimensional specific speed \( N_{QE} \).
For example, calculation result of $F_Q$ for Francis turbine is shown in Fig. 22.

3.3. Power efficiency scale factor $F_T$

The power efficiency scale factor $F_T$ for Francis turbines and Francis pump-turbines is given by Eq. (23).

$$ F_T = \frac{\eta_{TP}}{\eta_{TM}} = \frac{1}{(1 - \Lambda_T)} \eta_{TM} + \Lambda_T $$

(23)

where, $\eta_{TM}$ denotes the power efficiency at each model test point.

$$ \eta_{TM} = \frac{1}{1 + \left( \frac{1}{\eta_{TMopt}} - 1 \right) \left( \frac{n_{EDopt}}{n_{EDopt}} \right)^{\frac{1}{3}} \left( \frac{P_{EDopt}}{P_{ED}} \right)} $$

(24)

In Eq. (24), $P_{Edopt}$ is the power factor at the optimum point. The power efficiency of the model at the optimum point $\eta_{TMopt}$ is given by Eq. (25) from the standard power efficiency $\eta_{TMstd}$ at $Re_{ref}$. The standard power efficiency $\eta_{TMstd}$ is given by a function of the non-dimensional specific speed $NQE$.

$$ \eta_{TMopt} = 1 - \left( 1 - \eta_{TM} \right) \left( \frac{Re_{ref}}{Re_{Mopt}} \right)^{0.2} $$

(25)

The friction coefficient ratio is given by Eq. (26) as the ratio of friction torque coefficient.

$$ \Lambda_T = C_{LdP} / C_{LdM} = \left( \log_{10} K_{SM} \right)^2 / \left( \log_{10} K_{SP} \right)^2 $$

(26)

In Eq. (26), $K_{SM}$ and $K_{SP}$ are model relative roughness and prototype relative roughness respectively. To determine each of them, the relative roughness of smooth surface (subscript s) and that of rough surface (subscript r) are calculated by Eqs. (27) and (28), and the larger of two values is taken.

$$ K_{SMr} = \frac{200}{Re_{Mopt} \kappa_{DS}^2}, \quad K_{SPr} = \frac{200}{Re_{P} \kappa_{DS}^2} $$

(27)

$$ K_{SMr} = \frac{2ks_M}{D_{Mr} \kappa_{DS}^2}, \quad K_{SPr} = \frac{2ks_P}{D_{Pr} \kappa_{DS}^2} $$

(28)

where, $\kappa_{DS}$ is the ratio of maximum diameter of runner to reference diameter of runner given as a function of non-dimensional specific speed $NQE$. $k_{SM}$ and $k_{SP}$ are weighted averages of the two equivalent sand roughness mean values given as $ks_M, ks_P = (2k_{Sr} + ks_S)/3$. One of the values, $k_{Sr}$ is the average value of the outer surface of runner (crown and band of model or prototype) measured around the outer periphery. The other, $ks_S$ is the average value measured at the stationary wall facing to the runner where $k_{Sr}$ is measured. For example, calculation result of $F_T$ for Francis turbine is shown in Fig. 23.
4. Calculation example
Examples of the calculation for efficiency step-up $\Delta \eta_h$ for optimum operating points of a Francis turbine and a Kaplan turbine are shown in Fig.24 and Fig.25. Calculation condition is shown in Table 1. In Table 1, additional abbreviations are HS: hydraulically smooth, TR: back surface of runner crown and band and TS: back chamber surface facing to the runner crown and band.

The efficiency step-up $\Delta \eta_h$ by IEC 60193 are corresponding to the one given by the hydraulically smooth condition. Therefore, those results are almost the same as the results for the smooth surface given by the JSME S008-New and IEC 62097-2009. The differences between the results of the previous edition of JSME(JSME S008-1999) and the JSME S008-New are due to the difference of the assumed surface roughness condition. The assumed surface roughness of the JSME S008-1999 was between the hydraulically smooth condition and the practical rough condition. Difference between JSME-New and IEC 62097 for the Kaplan turbine is slightly larger than the one for the Francis turbine for both of the smooth and practical rough condition because of the following reason. Assumed relative scalable loss for a Kaplan turbine by IEC62097 is constant ($=0.045$) for whole $N_{Q_e}$ range, but the one by JSME-New varies due to the $N_{Q_e}$. The substantial differences of the efficiency step-up $\Delta \eta_h$ due to the surface roughness are given by the JSME S008-New and IEC 62097. So impact on hydraulic efficiency due to surface roughness of a turbine can be discussed.

| Table 1 Calculation condition |
|-----------------------------|
| Condition | SP | SV | GV | GR | RU | DT | TR | TS |
| Francis turbine | $N_{Q_e}=0.164$, $n_{EDopt}=0.345$, $Q_{EDopt}=0.225$ | Smooth surface | HS | HS | HS | HS | HS | HS |
| Kaplan turbine | $N_{Q_e}=0.330$, $n_{EDopt}=0.608$, $Q_{EDopt}=0.294$ | Smooth surface | HS | HS | HS | HS | HS | HS |

5. Concluding remarks
New JSME Standard S008 enhances the physical appropriateness for flow phenomena in machines by introducing performance conversion of each flow passage component from the model to the prototype. Main points of the revision, conversion formulae, calculation procedure and calculation examples were illustrated.

References
[1] JSME 1999 JSME Standard S008 Performance Conversion Method for Hydraulic Turbines and Pump-Turbines
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