PERFORMANCE CONVERSION OF WATER TURBINES AND PUMP-TURBINES
------ RECENT DEVELOPMENT IN SCALE EFFECT------

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1. Introduction

Water turbine is one of the simplest single-stage turbomachine among various electric power generating machines, and nevertheless the efficiency amounts to 95% in a large-sized water turbine of middle specific speed. Moreover, since it is not very difficult to increase the capacity of a machine, a huge-sized machine of 780MW has been manufactured. A large economical effect is then expected only by a small increase in total efficiency, and thus the accuracy of 0.1% is required in efficiency measurements.

In manufacturing such water turbines, a shop test by a fully homologous model is made in the first step, the model performance characteristics are then converted to prototype performances, and the results are finally evaluated and judged. Another reason for adopting model test is that a prototype test is difficult to conduct.

As the operating Reynolds number of a prototype is very large, the friction loss of a prototype becomes in general much smaller than that of a model, which results in that the prototype efficiency becomes larger than the model efficiency. Such a deviation from a simple similarity law is called " **scale effect** ". For example, the prototype Reynolds number amounts at most to about 50 times the model Reynolds number, which results that the efficiency step-up is around 2%[1]. The precise estimation of scale effect is thus very important.

From a historical point of view, Camerer[2] is the first person to propose an efficiency scale-up formula. Since then many proposals have followed[3]~[6], however most of them are not fully supported by theoretical and/or experimental verification. The progress was thus much slower in establishing performance conversion method compared with a large progress attained in manufacturing and testing technique.

In 1978 Research Working Group (Prof. Osterwalder) in IAHR reported on the efficiency scale-up based on theoretical consideration[7]. WG-18 is organized under IEC/TC4 for the standardization of efficiency scale-up into IEC 193: "Model Acceptance Test". In 1991 after 15 years discussion, WG-18 finally came to publish IEC-995[8]. It is however until Tokyo Congress in 1994 that the Publication 995 was adopted as a part of IEC-193[9], and the efficiency scale-up formulae took a great progress.

On the other hand in Japan, the working group was organized under the Japan Society for Mechanical Engineers parallel with the discussion in IEC, and the scale-effect was actively discussed based on the Ida-theory[10]~[12]. In 1989 the results were published as the JSME Standard S-008[13]. This Japan Standard gave a strong impact upon the activity of IEC/TC4, and WG-3 was organized and started to work in order to establish more accurate performance conversion method. In Japan, the Ida-theory has thereafter largely progressed[14]~[16] taking further progressed results of loss analysis[17],[18] and a new conversion method is proposed in 1996. The new method is much more simplified with keeping the same theoretical strictness and preciseness as those in S-008, and the new JSME S-008 is going to be published in the next year.

Here, the conventional performance conversions are surveyed and the recent development is interpreted in the following chapter.

2. Some Problems in Conventional Scale-up Formulae

IEC Publication 995 specifies that total efficiency of a hydraulic machine be expressed as

$$\eta = \eta_h \cdot \eta_m \tag{1}$$

where, η_m is mechanical efficiency due to external mechanical loss such as bearing loss. Since η_m has no contribution to the scale effect, it is excluded from the topics of this paper. η_h is hydraulic efficiency and is given as the product of the following three internal efficiencies;

$$\eta_h = \eta_E \cdot \eta_Q \cdot \eta_R \tag{2}$$

where, η_E is specific energy efficiency concerning all specific hydraulic energy loss in the main water passage, η_Q is discharge efficiency concerning leakage loss, and η_R is power efficiency of runner/impeller concerning disk friction loss.

Between two hydraulic machines of geometrically homologous shape, the flows around both runners/impellers become dynamically homologous, only when the velocity triangles are homologous at the inlet of runner/impeller and the operating Reynolds numbers are equal. In this case the performance characteristics can be converted between both runners/impellers by a simple similarity law. But if the operating Reynolds numbers are not equal, the difference in friction loss (scalable loss) causes significant difference in performance characteristics.

In hydraulic turbines and pump-turbines, the difference in operating Reynolds numbers is considerable between prototype and model, as described above, and moreover the surface roughness in both bodies are largely different, which causes that the ratio of scalable loss to total hydraulic loss is different between the prototype and the model.

Since the friction mostly causes dissipation of specific hydraulic energy E of the machine, scale effect appears primarily in the specific energy efficiency η_E, and also appears in both the leakage loss and the disk friction loss of runner/impeller due to the difference of Reynolds number. The differences in η_E, η_Q and η_R should thus be considered in scale effect between the prototype and the model.

Between the prototype P and the geometrically homologous model M, the specific hydraulic energy E, the discharge Q and the mechanical power P are converted by the following relations;

Turbine	Pump	
$\frac{E_P}{E_M} = \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \left(\frac{\eta_{EP}}{\eta_{EM}}\right)^{-1}$	$\frac{E_P}{E_M} = \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \left(\frac{\eta_{EP}}{\eta_{EM}}\right)$	(3)
$\frac{Q_P}{Q_M} = \left(\frac{n_P}{n_M}\right) \left(\frac{D_P}{D_M}\right)^3 \left(\frac{\eta_{QP}}{\eta_{QM}}\right)^{-1}$	$\frac{Q_P}{Q_M} = \left(\frac{n_P}{n_M}\right) \left(\frac{D_P}{D_M}\right)^3 \left(\frac{\eta_{QP}}{\eta_{QM}}\right)$	(4)
$\frac{P_P}{P_M} = \left(\frac{n_P}{n_M}\right)^3 \left(\frac{D_P}{D_M}\right)^5 \left(\frac{\eta_{RP}}{\eta_{RM}}\right)$	$\frac{P_P}{P_M} = \left(\frac{n_P}{n_M}\right)^3 \left(\frac{D_P}{D_M}\right)^5 \left(\frac{\eta_{RP}}{\eta_{RM}}\right)^{-1}$	(5)

where, n is rotational speed and D is reference diameter of runner/impeller. The last term in the above equations represent the scale effect and are to be put 1.0 in a simple similitude.

In the conventionally proposed scale-up formulae, the scale effect is only considered in the mechanical power by replacing η_R by η_h and by neglecting the specific energy efficiency η_E and the discharge efficiency η_Q. That is to say, the step-up of hydraulic efficiency is all converted to the mechanical power increase of runner/impeller. The conventional scale-up formulae can be classified to the following two categories.

① The hydraulic loss consists only of scalable loss(friction loss).

② The hydraulic loss consists of scalable loss and non-scalable loss.

The former is represented by the famous Moody 1/5-power formula, shown as;

$$1-\eta_P = (1-\eta_M)\left(\frac{D_P}{D_M}\right)^{-\frac{1}{5}} \tag{6}$$

Here, the relative loss $1-\eta$ is supposed to consist only of friction loss, and is converted under the assumption of hydraulically smooth surface (friction coefficient $C_f \propto \mathrm{Re}^{-1/5}$).

In the latter case, the efficiency scale-up is expressed as follows by putting the ratio of scalable loss to total loss as V, called as "loss distribution coefficient".

$$1-\eta_P=(1-\eta_M)(1-V+V\Lambda) \tag{7}$$

This expression is represented by Ackeret formula (V=0.5) for propeller turbine, and by Hutton formula (V=0.7)[4] for Kaplan turbine. Here, Λ is the ratio of friction coefficient of prototype to that of model, and is given under the assumption of hydraulically smooth surface in both formulae, that is

$$\Lambda \equiv C_{fP}/C_{fM}=\left(\frac{Re_P}{Re_M}\right)^{-\frac{1}{5}} \tag{8}$$

On the other hand in IEC Pub. 995, the relative loss $1-\eta$ is divided into two, the relative scalable loss δ and the relative non-scalable loss δ_{ns}, and only the relative scalable loss is converted from the model δ_M to the prototype δ_P assuming a smooth surface, $C_f \propto \mathrm{Re}^{-0.16}$. In this case the non-scalable loss δ_{ns} is constant over the whole Reynolds number range and δ is determined at the reference Reynolds number of $\mathrm{Re}_{ref}=7\times10^6$ by use of loss distribution coefficient V specified for each type of turbine and pump-turbine.

In the above-described conventional formulae several problems have been pointed out as follows;

(1) The efficiency step-up $\Delta\eta \equiv \eta_P-\eta_M$ at the optimum point is treated to be only mechanical power increase, however both the mechanical power P and the discharge Q actually increase and the corresponding operation point shifts in the prototype.

(2) The V-value must change depending on individual design and specific speed of a runner/impeller. V-value also changes at off-design operation, although a constant value is specified.

(3) The hydraulically smooth surface is assumed, but the surface roughness of prototype is usually in completely rough region..

To show some examples of efficiency step-up $\Delta\eta \equiv \eta_P-\eta_M$, Fig. 1 illustrates the calculated results[18] from Eq. (7) versus the model efficiency η_M for the case of Λ=0.6. It is revealed from Fig. 1 that a poor turbine with low efficiency has very high efficiency step-up based on V-value. The low efficiency of a poor turbine is mainly caused by a large non-scalable loss, and thus the efficiency step-up should be low. This result is because the denominator of the loss distribution coefficient V includes non-scalable loss, which changes largely by individual design and specific speed. Constant V-value thus gives unreasonably high efficiency step-up for poor turbines.

On the other hand, the JSME Standard S-008 has avoided all of the above problems, as the scale-up formulae are all based on the theory and are deduced from loss analysis which was verified by the model test data. The surface roughness effects are fully considered and the V-value is given as a function of the specific speed n_{SQ} as shown in Fig. 2,and moreover, the variation of V-value is also specified at the off-design condition. The complexity of the performance conversion method is necessarily accompanied in the JSME Standard.

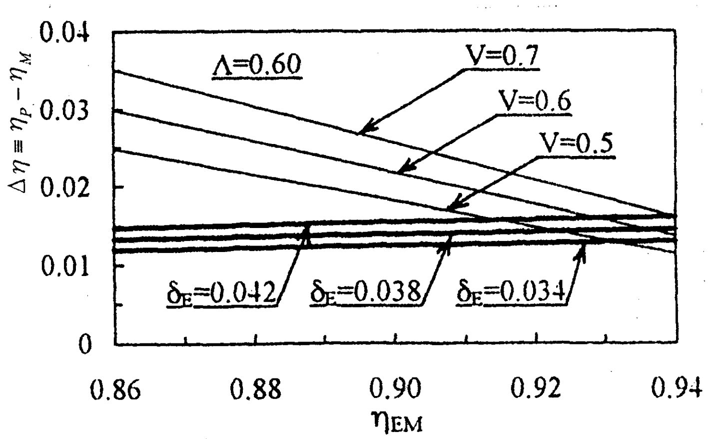


Fig. 1 Efficiency step-up $\Delta\eta$ versus model efficiency η_M and comparison of scale-up based on V-value with that based on δ_E-value

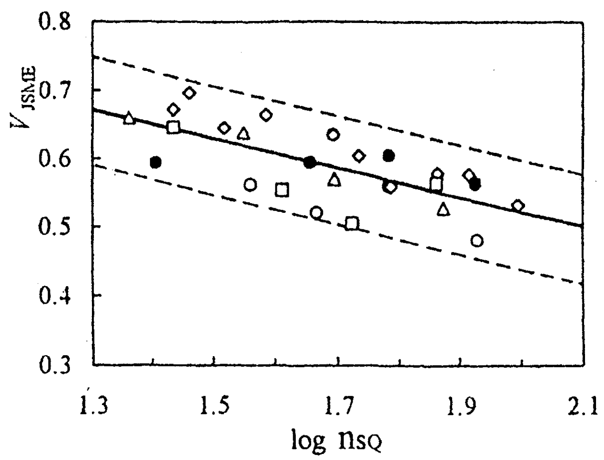


Fig. 2 V-value versus specific speed

3. Newly Proposed Performance Conversion Method

The Ida-theory has thereafter largely progressed under re-examination and newly proposed idea, and the performance conversion method is considerably simplified without losing the theoretical preciseness. One of the major changes from the existing method to new method is the use of " **relative scalable loss** δ_E ", defined as the ratio of scalable loss to total hydraulic specific energy, instead of loss distribution coefficient V. The δ_E -value changes little for the variation of n_{sQ} in Francis turbine, as shown in Fig. 3, and the efficiency step-up is nearly constant for the variation of model efficiency η_M and becomes rather small for a poor turbine of low efficiency. Thus the method based on δ_E gives much more reasonable results than that base on V (18). In Fig. 1 is also shown the results for the upper and the lower range of the scattering of δ_E. Apparently this scattering is much less than that of V, and the use of δ_E instead of V improves considerably the accuracy of conversion. Hence, δ_E is selected instead of V in the new Ida-theory, and the following conversion factors are also introduced for simplification of discharge efficiency η_Q and mechanical power efficiency η_R.

$$F_E = \frac{\eta_{EP}}{\eta_{EM}},\ F_Q = \frac{\eta_{QP}}{\eta_{QM}},\ F_R = \frac{\eta_{RP}}{\eta_{RM}} \tag{9}$$

The efficiency of the prototype is then given by

$$\eta_P = F_E F_Q F_R \eta_M \tag{10}$$

and the conversion factors F_E, F_Q and F_R are written as;

Turbine | Pump

$$F_E = \frac{1}{1-\delta_E(1-\Lambda)} \qquad F_E = 1+\delta_E(1-\Lambda) \tag{11}$$

$$F_Q = \frac{1}{\eta_{QM}+\Lambda_Q(1-\eta_{QM})} \qquad F_Q = \frac{(1-\Lambda_Q)}{\eta_{QM}}+\Lambda_Q \tag{12}$$

$$F_R = \frac{1-\Lambda_R}{\eta_{RM}}+\Lambda_R \qquad F_R = \frac{1}{(1-\Lambda_R)\eta_{RM}+\Lambda_R} \tag{13}$$

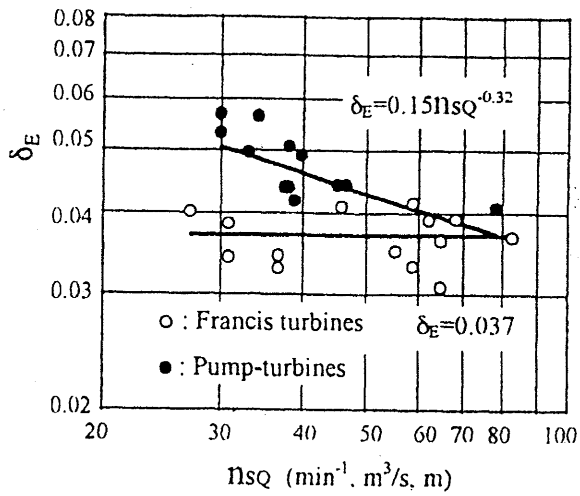


Fig. 3 Scalable loss δ_E versus specific speed

Here, Λ, Λ_Q and Λ_R are the ratios of scalable loss, leakage flowrate coefficient and disk friction coefficient of the prototype to that of the model, respectively. According to the new Ida-theory the formulae for Λ, Λ_Q and Λ_R are simplified to the following final form without losing accuracy in the range of actual operating conditions.

$$\Lambda = \left(\frac{D_P}{D_M}\right)^{-0.18}\left\{0.3+0.7\left(\frac{e_P}{e_M}\right)^{0.18}\right\} \tag{14}$$

$$\Lambda_R = \left(\frac{D_P}{D_M}\right)^{-0.27}\left(\frac{e_P}{e_M}\right)^{0.12} \tag{15}$$

$$\Lambda_Q = 1.1 \tag{16}$$

where, e is surface roughness. The admissible roughness is used for e if the roughness is less than the admissible. The formulae for δ_E and η_R in Eqs. (11)~(13) are found to have a simple form after many calculations at actual operating conditions. Taking for an example Francis turbine and pump-turbine, these values at the optimum point are expressed as the followings obtained from the studies using various numerical calculations(15).

Francis turbine | Francis pump-turbine

$$\delta_E = 0.037 \qquad \delta_E = 0.050\, n_{sQ}^{-0.32} \tag{17}$$

$$\eta_R = 0.995 - 5.00\, n_{sQ}^{-2} \qquad \eta_R = 0.986 - 21.0\, n_{sQ}^{-2} \tag{18}$$

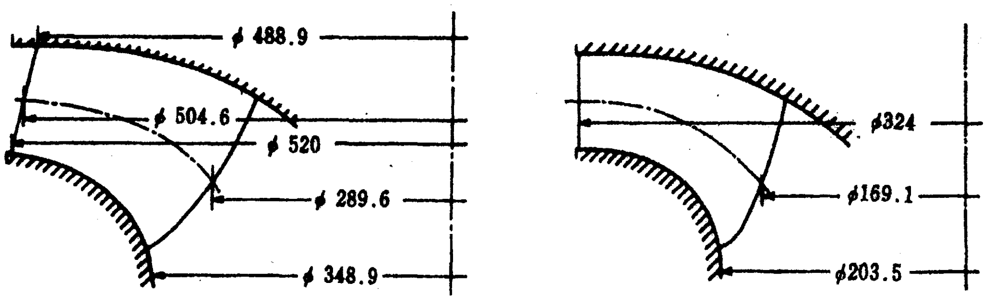
As the value of η_Q in both types are nearly equal to 1.0, the conversion factor of discharge efficiency F_Q can be put as $F_Q = 0.998$ in Eq. (12) within the error of 0.2% in $\Delta\eta_h$.

4. Some Examples and Comparison with Conventional Method

In this chapter some examples of converted performance characteristics are illustrated and are evaluated. For the evaluation of scale-up formulae the best way is to use the measured performances of prototype, but the accuracy of field test necessarily contains at most 2% of error. Accordingly, the second best way might be that the theoretical method is verified by the model performance characteristics measured at shop test and thereafter the scale-up formulae, IEC Code and JSME Standard, are evaluated by use of the prototype performances predicted by the theoretical method by Ida-theory.

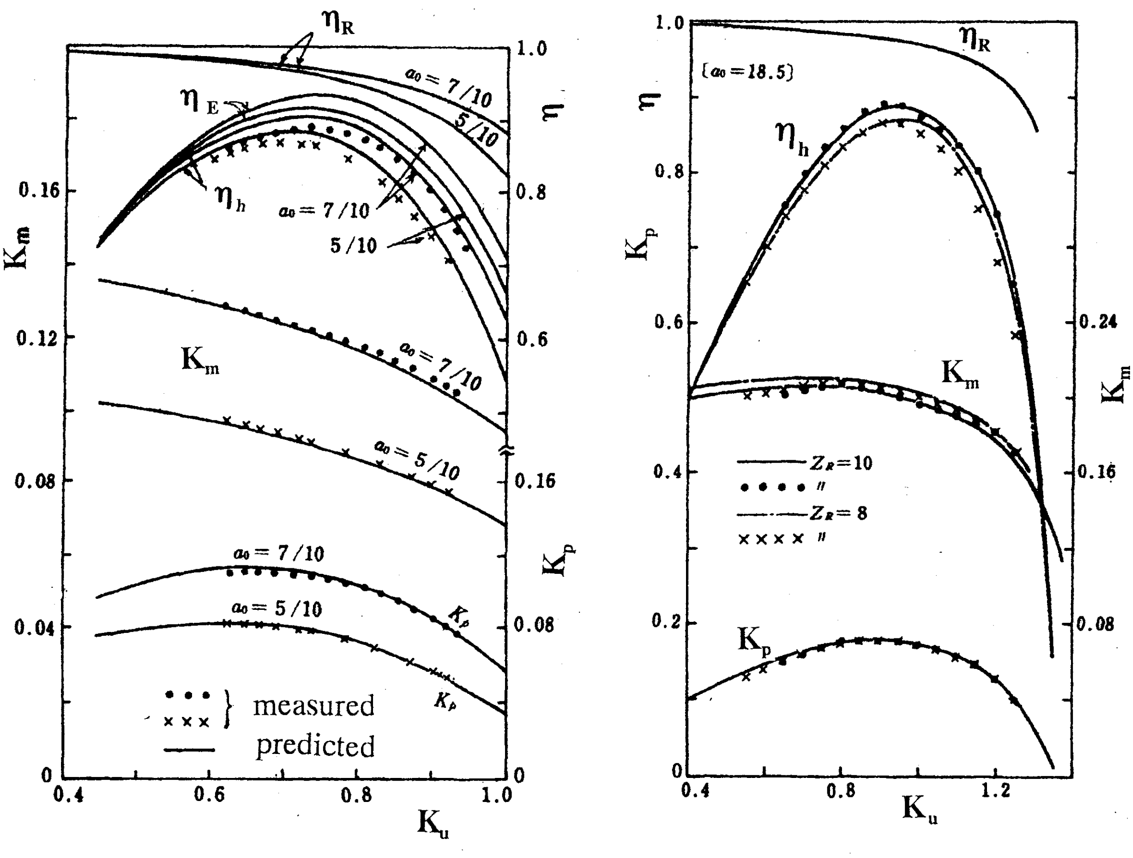
4. 1 Verification of Ida-theory in Model Francis Turbine and Pump-turbines

Taking as an example, the performances of model Francis turbine and pump-turbine shown in Figs. 4(a) and 4(b) are predicted by Ida-theory and the results are compared with the model test data in Figs. 5(a) and (b).



(a) Francis turbine (b) Francis pump-turbine

Fig. 4 Model turbines used for verification of performance prediction by Ida-theory



(a) Francis turbine shown in Fig. 4(a) (b) Pump-turbine shown in Fig. 4(b)

Fig. 5 Verification of Ida-theory by comparison with the model test data.