ABSTRACT

An efficiency of a low specific speed pump is in general relatively low and its prediction over the whole flow-rate range is still difficult. Multistage pumps are usually equipped with inducers and complicated thrust-balancing devices, which make the performance prediction more difficult. In order to improve pump efficiency, to establish performance prediction method and to elucidate the flow characteristics of a thrust balancing device, the method of predicting overall pump performance was presented together with the method of analysing internal flow in a thrust balancing device. The analysis developed here was confirmed to give satisfactory results by comparison with actual measurements. Using this method, the performance of a multistage pump of low specific speed and the viscous effect upon the performance were revealed, and the behavior of axial thrust and leakage in thrust-balancing device were determined over a wide flow-rate range. The difference in fluid properties were also revealed to influence little on the pump performance and the Reynolds number dependence was made clear.

INTRODUCTION

In pumping hazardous liquid, such as LPG, LNG, LH₂ or LOX, multistage diffuser pumps of low specific speed are widely used with inducers. The reliability and safety of these pumps depend largely upon axial thrust balancing and leakage sealing, and that the adoption of a thrust balancing device together with a submerged or a canned motor or a complicated sealing system is unavoidable.

In a performance test of these pumps a substitute liquid is usually used because of difficulty in testing by hazardous liquid, and the performance conversion method is applied to determine the actual pump performance. Pump performances expressed in the non-dimensional form are, in general, same for all kinds of fluid except for viscous effect. As for the viscous effect of a hydroturbine, the formulae proposed by Moody, Hutton and Ackeret have been widely used as a scale up formulae. The JSME Standard S-008 published in 1989 gives more accurate performance conversion based on theoretical treatment. However, as for a pump there are few data available for comparing a shop test (or model test) data with a field ones, which makes it difficult to evaluate the accuracy of performance conversion.

The present study is aimed to establish an accurate method of predicting the overall performances of multistage pumps of low specific speed over the whole flow-rate range by comparison with the actual measurements, and to reveal the viscous effect upon pump performances theoretically. The pump performance in the low flow-rate range is remarkably influenced by a leakage in a thrust-balancing device which is not yet well known. It is then also necessary to determine the method of analysing the flow characteristics in the thrust-balancing device.

OUTLINE OF THE PRESENT THEORY

Prediction of Pump Performance over the Whole Flow-rate Range

One of the present authors developed an analytical method of predicting a performance of centrifugal impellers taking the reverse flow at the impeller inlet and the outlet into account. This method is composed of a one-dimensional loss analysis of an impeller flow using the results of a quasi-three-dimensional potential flow analysis and a viscous flow analysis of an impeller outlet flow in a vaneless diffuser channel. The loss formulae proposed are all expressed in simple forms but the results have sufficient accuracy in the wide flow-rate range.

This method of performance prediction treats the impeller losses as consisting of shock loss and friction loss determined using the results of a quasi-three-dimensional analysis. The loss caused by flow separation and secondary flow in an impeller
The head coefficient $\Psi$ and the outlet pressure $P$ of the main impeller in the non-dimensional form are expressed as

$$\Psi = 2(1 - k - \phi \cos \beta / \eta_s) \cdot \zeta_s - \zeta_f$$

(1)

$$P = \Psi \cdot (v_2^2 - v_1^2) / u_2^2$$

(2)

where $k$, $\epsilon_2$, and $\eta_s$ are slip factor, contraction rate of impeller channel due to blade thickness and volumetric efficiency, respectively. $\zeta_s$ and $\zeta_f$ are the coefficients of shock loss and friction loss, and are given in a simple form (6).

The flow characteristics in a vaneless diffuser channel from the impeller outlet to the diffuser vane inlet is determined from a viscous flow analysis based on the boundary layer theory. The velocity variation along a streamline in this region is thus given by an analytical formula (7) and the pressure is determined by the balance between the radial pressure gradient and the centrifugal force: $dp/dr = \rho v_0^2 / r$. It is to be noted that the friction loss from an impeller outlet to a diffuser vane inlet is very large because of large peripheral velocity in a low specific speed impeller.

The hydraulic losses in a diffuser vane channel is composed of shock loss, friction loss and deceleration loss, which can be predicted by a conventional method (8).

The prediction of an inducer performance is very difficult, as there are few data so far reported, in which a flow separation and a large secondary flow is observed at off-design conditions. As an inducer head is not very large compared with that of a main impeller in a multistage pump, and hence a cascade theory is applied simply assuming that a stream surface is two-dimensional. The drag coefficient $C_d$ of a thin and lowly cambered aerofoil of an inducer blade and the overturning angle at the outlet are estimated referring to the NACA cascade data (9).

**Flow Analysis in Thrust-Balancing Device**

One of the present authors has revealed the behavior of axial thrust and leakage of radial flow turbomachinery, and has found that the accuracy of the thrust calculation is mainly dependent on leakage flow-rate (10). Axial thrust is caused by pressure acting on the rotating parts of a machine, thus axial thrust analysis consists of an analysis of the gap flow between a rotating and a stationary wall and of the determination of boundary values.

Generally, gap flow is of two types. One is the axial gap flow between a rotating disk and a stator such as that at the back of an impeller shroud, and the other is the annular gap flow such as that in an annular seal. The analyses of gap flows and the determination of boundary values are given in the literatures (1) (11).

**Calculation Procedure**

In predicting an impeller performance, the leakage flow-rate in the front and back spaces of an impeller shrouds amounts to a large part of the impeller flow at very low flow-rates. An accurate determination of a pump performance then requires an accurate determination of all the leakages.

However, the leakage in a balancing device varies largely depending on the balancing disk gap $s_d$ (see Fig. 4) determined from the balance of forces acting on all the rotating parts. Accordingly, the calculation is initiated by assuming zero leakage and the pump performance is calculated. Then the pressure distribution in the balancing device and in the front and the rear gaps of an impeller are determined, which gives the first approximation of each leakage. The pump performance and the leakage with higher accuracy are asymptotically determined.

**DESCRIPTION OF PUMPS TESTED**

Three kinds of multistage diffuser pumps used for LPG and LNG transportation were tested. These pumps are usually set submerged in a liquid tank vertically. A schema of each pump...
is shown in Figs. 1, 2 and 3. In Fig. 1 (Type A pump) the liquid entering through the 4 bladed inducer shown at the left end of the figure flows into the main impeller of two-stages. The fluid coming out of the 2nd stage impeller is discharged through six pipes arranged around the motor to the discharge pipe shown at the right end in Fig. 1. The pump of Type B (Fig. 2) has six or two stages and Type C (Fig. 3) two stages of different impellers with different blade angle as shown in the figures.

The thrust balancing device consists of the balancing disk and the balancing piston as shown in Fig. 4. The rotating parts including the balancing piston and the disk are movable to the axial direction, and form a self-balancing mechanism, in which the balancing disk plays an important role of balancing the axial force by adjusting the disk gap $s_d$.

A portion of the liquid at the last stage of impellers is lead into the balancing dram through the balancing piston and disk, and forms a leakage through the motor rotor housing and the casing orifice into the liquid tank (outside of pump). In order to lubricate the rear bearing, there is one more leakage pass from the discharge pipe at the right end in Fig. 1 to the casing orifice through the shaft end orifice and the motor housing. In Type B pump the leakage in the motor housing returns to the exit of the first stage impeller through the return tubes as shown in Fig. 2, and in Type C pump to the suction side as shown in Fig. 3.

The liquid used is LPG or LNG, but in a shop test a substitute liquid such as water or LN$_2$ is used, as mentioned above, and the properties of these liquids are listed in Table 1. The test Reynolds number based on the impeller tip speed and the radius was from $7 \times 10^6$ to $4 \times 10^7$.

**COMPARISON OF THEORY WITH MEASUREMENT**

**Pump Performances**

The predicted curves of pump head, shaft power and efficiency are compared with the measured results in Figs. 5, 6 and 7, which include the performances of inducer, main impeller, vaned diffuser, return channel and discharge pipe. They agree
well not only around the design point but also over a wide flow-rate range.

As for the shut-off power, it is still difficult to predict theoretically, because most of the shut-off power is due to a reverse flow at an impeller inlet, thus the following formula was adopted here for the shut-off power coefficient $\tau_s$ referring to Stepanoff

$$\tau_s = 0.137\left(\frac{b_2}{r_1}\right) + 0.0188 + 0.00079\left(\frac{b_2}{r_1}\right)^2$$

and the power due to the inlet reverse flow was expressed by a quadratic which decreases to zero at the designed flow-rate.

In Figs. 5, 6 and 7 the measurements were performed for LPG, LNG, LN$_2$, and water, of which operating Reynolds numbers were nearly equal as shown in Table 1, and then the measured performances should also be nearly equal theoretically except for the case of Type B pump (Fig.6). However, some differences are recognized in the measured data depending on the working fluid. As the shaft power was determined from the input current and voltage of a motor using the motor efficiency curve, there remains a possibility of an error due to difficulty in calibrating a submerged type motor.

The type B pump (in Fig.6) originally consists of six stages but the case of two stages were also measured using water, of which Reynolds number was about 1/5 of the six stage case (LN$_2$ and LPG tests). In the two-stages case the contribution of an inducer performance to a pump overall performance is relatively large compared with the six-stage case, and the negative gradient of the head curve becomes larger as shown by the prediction in Fig.6, because the inducer head curve has large gradient. In Fig. 6 some differences are recognized between the predicted and the measured head curves, which might be due to the difficulty in calibrating the inducer head in the low flow-rates, as this inducer has especially large angle near the boss side.

In order to reveal the reason why a low specific speed pump has low efficiency, the three efficiencies composing an overall efficiency $\eta$ are predicted theoretically, that is the hydraulic efficiency $\eta_h$, the volumetric efficiency $\eta_v$, and the mechanical efficiency $\eta_m$. However, the definition of these efficiencies is not clear for the case of a multistage pump with an inducer and a balancing device, and here the efficiencies are defined as

$$\eta = \eta_h \eta_v \eta_m$$
$$\eta_h = 1 / (H_{in} + H_{out})$$
$$\eta_v = (Q - \Delta Q_{out}) / (Q + \Delta Q_{out})$$
$$\eta_m = L_{inp} / (L_{inp} + L_{out} - \gamma \Delta Q_{out} H_{in})$$
$$L_{inp} = \gamma (Q + \Delta Q_{out}) (H_{in} + H_{out})$$
Fig. 8 Hydraulic, mechanical and volumetric efficiencies of Type A pump (\( \eta_{\text{imp}} \) is impeller volumetric efficiency)

where \( Q \) is the flowrate at pump suction, \( \Delta Q_{\text{imp}} \) is the leakage in the front shroud gap of main impeller, \( \Delta Q_{\text{out}} \) is the leakage through a balancing device and a motor, and the discharge is given as (Q-\( \Delta Q_{\text{out}} \)). The leakage in the rear shroud of main impeller is taken into account in predicting the diffuser vane performance. \( L_d \) is the disk friction power including the power due to impeller inlet reverse flow.

The predicted efficiencies are illustrated in Fig. 8 for the case of Type A pump. As there are several leakages passing through the gaps around an impeller and in a balancing device, the volumetric efficiency of a pump is different from that of an impeller, the latter of which is also shown in Fig. 8 by comparing with that of an ordinal pump of a single stage. Figure 8 reveals that the volumetric efficiency of this pump is relatively high compared with that of an ordinal pump, and that the mechanical efficiency is low because of large power consumed by the friction of rotating parts in the balancing device. The hydraulic efficiency is not low, as it includes that of an inducer.

**Viscous Effects on Pump Performance**

In order to clarify the viscous effect, the present prediction was further made for varying the Reynolds number from \( 5 \times 10^5 \) to \( 5 \times 10^6 \) for the case of Type A pump. The calculated results are shown in Figs. 9 and 10. It is clearly recognized from Fig. 9 that a pump overall efficiency increases remarkably with the Reynolds number, and that this increase is mainly due to the decrease in shaft power. If the Reynolds number is increased, it is, in general, expected that a head coefficient increases due to a decrease of wall friction. The friction loss coefficients in an impeller channel and a diffuser channel actually decreases as is expected, but the friction loss of the wall between the impeller outlet and the diffuser vane inlet increases against expectation. The absolute velocity here is especially large in a low specific speed pump, and hence the friction loss coefficient of this part occupies large part of the total hydraulic loss. According to the theory, it is revealed that the boundary layer thickness decreases with an increase in the Reynolds number and the roughness effects becomes remarkable to yield the increase in friction loss coefficient, which is larger than the decrease of friction loss coefficient in an impeller channel and diffuser vane channel. It is then suggested that a hydraulically smooth finishing of a casing wall at an impeller outlet contributes much to increase of a pump efficiency.

The change in pump performance at the best efficiency point (BEP) is illustrated in Fig. 10. The flow-rate coefficient at the BEP is seen to change little for large change in the Reynolds number while the maximum efficiency and the shaft power coefficient at the BEP changes largely. It is also revealed that the performance obtained by use of substitute liquid such as water or LN\(_2\) differs little from that by hazardous liquid such as LPG or LNG, as the...
test Reynolds number differs little as shown in Table 1.

Axial Thrust Performance

As described above, the axial thrust varies rapidly with gap $s_d$ at the variable balancing disk. Accordingly, calculations were performed to demonstrate the effect of variations in this gap. The results are shown in Fig. 11, in which thrust toward the discharge side (upward direction in actual setup) is assumed to be positive. It is seen that the axial thrust and the leakage decrease considerably with a decrease in the range of the gap $s_d < 0.1$ mm, while for the wider range of $s_d$ the change becomes little.

The operating gap is obtained by the intersection point between the thrust curve and the weight of the rotating parts, as shown in Fig. 11, which results that the present balancing device is safely operating by the gap $s_d$ of about 0.07 mm for a wide flow-rate range and 0.1 mm in high flow-rates. It is also revealed that the movement of the shaft is little against a wide variation of axial thrust. The stability of the thrust balancing device can be estimated by the gradient $\frac{dC_t}{ds_d}$ of the thrust curve, and about 40 tons are needed to move the axis by 0.1 mm near the point of $C_t=0$. This indicates that this device has strong stiffness and is stable against thrust variation. However, in high flow-rates the axial thrust curve becomes gentle and the stiffness decreases.

CONCLUSION

A theoretical method of predicting overall performance, axial thrust and leakage of multistage diffuser pumps of low specific speed has been established over the whole flow-rate range. Using this method, the viscous effect on the pump overall performance are elucidated together with the thrust and leakage performances. The results are summarized as follows:

(1) Comparison with the actual measurements revealed that the performance prediction presented here yields satisfactory results including the prediction of axial thrust and leakage behavior.

(2) Pump efficiency rises remarkably with an increase in the Reynolds number because of a decrease in disk friction coefficient. Pumping head coefficient changes little with the Reynolds number because of the behavior of boundary layer on a casing wall at an impeller outlet.

(3) The pump performances obtained by use of substitute liquid such as water or LN$_2$ differ little from those by use of hazardous liquid such as LNG or LPG.

(4) The axial thrust of a multistage pump of low specific speed greatly varies with the gap of a balancing disk, and the present balancing device has significant stiffness against thrust variation, although the operating gap is as narrow as 0.07 mm at the BEP. The operating gap decreases little in the smaller pump discharge range and increases in the larger discharge range.

Fig. 11 Axial thrust variation against balancing disk gap $s_d$

NOMENCLATURE

- $b_2$ = impeller outlet width
- $C_t = \frac{T}{\rho \pi r_2^2 u_2^2}$; axial thrust coefficient ($T$=axial thrust)
- $N_S =$ specific speed of pump [$\text{m}, \text{m/\text{min}}, \text{rpm}$] eaker
- $P = \frac{2p}{\rho u_2^2}$: non-dimensional pressure
- $Q, \dot{Q} =$ pump flowrate and leakage flowrate
- $r_1, r_2 =$ radius at main impeller inlet and outlet, respectively
- $Re = \frac{r_2 u_2}{v}$ : Reynolds number
- $s_d =$ axial gap width at balancing disk
- $u_2 =$ main impeller tip speed
- $\beta_2 =$ outlet angle of main impeller
- $\eta =$ efficiency
- $\psi = \frac{H}{(u_2^2/2g)}$ head coefficient of pump ($H$=pump head)
- $\nu, \rho =$ kinetic viscosity and density ($\nu=\rho g$) of fluid, respectively
- $\tau = \frac{L \rho A_3 u_2^3}{A_4}$; power coefficient ($L$= power, $A_3 = 2\pi r_2 b_2$)
- $\phi = \frac{Q_{\text{discharge}}}{A_2 u_2} =$ discharge coefficient of pump

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