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Radial Thrust in Centrifugal Pumps with a Single-vane Impeller*

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Experiments on radial thrust in centrifugal pumps with a single-vane impeller are carried out. Three impellers with different subtended angles of vane and two delivery casings, i.e., volute casing and circular one, are designed. In the experiments the effect of the subtended angle of vane and that of the geometry of a delivery casing on radial thrust are examined. An analysis is also made to calculate the radial force acting on a single-vane impeller. From the present analysis and experiments, the following results are obtained.

- (1) It is confirmed that the radial thrust consists of the force caused by a delivery casing and that caused by an impeller itself.
- (2) The magnitude of the radial thrust in the case of the circular casing is less than that in the case of the volute casing.
- (3) The effect of the subtended angle of vane on radial thrust is small.

1. Introduction

A single-vane impeller is often applied for centrifugal pumps for liquid with sewage, because it has a good non-clogging performance given by a wide impeller passage. Since the unbalanced radial hydraulic force induced in a single-vane impeller causes vibrations of pump shaft and shortens the life of bearings and that of shaft seals, it is necessary to estimate the magnitude of radial thrust in the design of a single-vane impeller. But the radial thrust in pumps with a single-vane impeller has not been investigated enough compared with usual centrifugal pumps with a multi-vane impeller. It seems that the radial thrust consists of the force caused by a delivery casing and that caused by an impeller itself. Since the radial thrust caused by a multi-vane impeller itself is small because of its symmetric shape, the radial thrust caused by a delivery casing is dominant. On the other hand, in pumps with a single-vane impeller the radial thrust caused by the impeller itself may be considerably great. In this study, to confirm that the radial thrust consists of these two forces, the radial thrust in centrifugal pumps with a single-vane impeller was examined experimentally.

2. Nomenclature

b_2 : width of impeller exit, m
 d_2 : impeller diameter, m
 g : acceleration due to gravity, m/s²
 F_r : radial thrust, kgf
 H : total head, m
 Q : capacity, m³/min

r_2 : outer radius of impeller, m
 u_2 : peripheral velocity, m/s
 β : blade angle, rad
 η : pump efficiency
 θ : subtended angle of vane, rad
 ρ : density of water, kgf s²/m⁴
 ϕ : flow coefficient = $(Q/60)/(\pi d_2^2 u_2/4)$
 ψ : head coefficient = $H/(u_2^2/2g)$

3. Test pump and apparatus

3.1 Test pump

The test pump with a device for measuring the radial thrust is shown in Fig.1. Considering the subtended angle of vane and the geometry of a delivery casing as the main parameters which affect the radial thrust in pumps with a single-vane impeller, three open type impellers designated A, B and C, and two delivery casings designated C and V were designed. The twin-vane impeller shown in Fig.2(d) was also designed to investigate the effect of the number of vanes. Principal data on test pumps are shown in Table 1. The geometries of impellers and the photograph

Fig.1 Test pump

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Table 1 Principal data on test pumps

Impellers	Subtended angle of blade	Number of vanes	Blade exit angle
A	400°	1	9°30'
B	360°	1	8°36'
C	320°	1	7°39'
D	165°	2	3°12'

Revolution speed : 1410 rpm

Delivery casing : Volute type and circular one

Impeller : Open type (other than impeller D)

Fig.2 Impellers

Fig.3 Impeller A

of the impeller A are shown in Fig.2 and Fig.3 respectively. To reduce the axial thrust radial ribs were mounted on the back side of the impeller hub. The axial clearance between the impeller vane and the casing wall was 0.5 mm. Two kinds of delivery casings, volute type and circular type, were designed. The volute casing is adopted in usual centrifugal pumps. For pumps handling the sewage water, the circular casing is considered to be better than the volute one from the stand point of nonclogging performance, since the former has a larger radial space between impeller periphery and casing wall. It has been reported that the radial thrust of pumps with a multi-vane impeller and a circular casing changes greatly with a change in the ratio of inner-diameter of circular casing d_c to impeller diameter d_2 . According to the reference (1), when the ratio d_c/d_2 is smaller than 1.29, the casing diameter has some effects on the magnitude of radial thrust. As the ratio d_c/d_2 of present test pumps is equal to 1.53, the effect of the diameter of casing will be negligible.

3.2 Apparatus for measuring radial thrust

Fig.5 shows the experimental apparatus. The device for measuring radial thrust was installed between the test pump and the torque pick-up. A detail of the device is shown in Fig.6. The radial forces acting on the impeller are transferred to four thin ribs which support the sleeve for bearings. The strain of each rib is measured by strain gauges mounted on the corresponding rib. The four-gauge method using rosette strain gauges was employed to reduce the zero drift caused by a temperature rise of the bearings. Two Wheatstone bridges were composed so that the horizontal and vertical components of radial force can be measured separately. The device was calibrated statically using weights applied at the impeller center line.

Fig.4 Delivery casings

Fig.5 Experimental apparatus

4. Experimental results and discussions

4.1 Pump performance

Experiments were made on pumps with various combinations of one of the four impellers and one of the two delivery casings. The test pumps are designated by combining the name of the impeller with that of the casing. For example, AV pump means a pump with impeller A and casing V.

The performances of test pumps with the volute casing and those of ones with the circular casing are shown in Fig.7 and Fig.8 respectively. In Fig.8 the performance of the pump with a twin-vane impeller is also shown to see the effect of the number of vanes. The following characteristics were observed from these figures.

(1) Head coefficient curves for all combinations have a steep slope due to the small number of vanes and the small blade exit-angle.

(2) The effect of the subtended angle on the head coefficient is small.

(3) There is little difference between the head coefficients of pumps with the volute casing and those of pumps with the circular casing. The result shows that the geometry of delivery casing has small effect on performances of pumps.

4.2 Radial thrust

4.2.1 Effect of subtended angle of vane

Fig.9 shows the relationship between the normalized radial thrust and the flow coefficient for the pumps with the volute casing. In the figure, the term "maxima" means the maximum magnitude of radial thrust during a revolution of the impeller, and the term "mean values" shows the mean magnitude of the thrust vector. Since the influence of volute casing on radial thrust is minimized at the best efficiency point, as discussed later, the effect of the subtended angle of vane should be examined at that point. There is little difference in the magnitude of radial thrust of test pumps near the best efficiency point. Therefore, the subtended angle of vane appears to have small effect on radial thrust in the range between 320 deg and 400 deg. This is attributable to the weak effect of the subtended angle on the head coefficient.

Fig.6 Device for measuring radial thrust

Fig.7 Performances of AV and CV pumps

Fig.8 Performances of AC and DC pumps

4.2.2 Effect of geometry of delivery casing

Fig.10 shows the radial thrust for the test pumps with circular casing. The following features are observed comparing Fig.10 with Fig.9 for the volute casing.

(1) In the case of the test pump with the volute casing the radial thrust is maximum at shutoff, decreases with an increasing flow coefficient and reaches a constant value near the best efficiency point. On the other hand, the radial thrust tends to increase gradually with an increase in the flow coefficient in test pumps with the circular casing.

(2) The difference between the maxima and mean values during a revolution, which corresponds to the magnitude of fluctuation in the radial thrust, is maximum at shutoff in the pumps with the volute casing, and it is maximum in the large capacity region in the pumps with the circular casing.

Fig.9 Radial thrust (Volute casing)

Fig.10 Radial thrust (Circular casing)

The above result (1) could be explained as follows. The radial thrust of usual pumps with a multi-vane impeller varies with the geometry of the delivery casing as shown in Fig.11.⁽²⁾ Such characteristics may appear in the pumps with a single-vane impeller. But the radial thrust does not vanish at the best efficiency point in the case of a single-vane impeller with a volute casing. It does not vanish at shutoff in the case of a pump with a circular casing. The radial thrust curves of pumps with a single-vane impeller are similar to the curves in Fig. 11. According to these results, in the case of pumps with a single-vane impeller, it can be seen that the radial thrust consists of the force caused by the delivery casing and that caused by the impeller itself.

To examine the radial thrust caused by an impeller with a single-vane the pressure distribution on the single-vane was calculated by singularity method. The radial thrust was obtained by integrating the pressure distribution on the vane through the whole blade. The resultant radial thrust F_r shown in Fig.12 and its components in rectangular coordinates are expressed by the following equations.

Fig.11 Radial thrust in usual centrifugal pumps (2)

Fig.12 Forces acting on an impeller vane

$$F_r = \sqrt{F_{rx}^2 + F_{ry}^2} \quad (1)$$

$$\frac{F_{rx}}{(\rho/2)u_2^2 b_2 r_2} = \int_0^{\theta_2} \frac{p_+ - p_-}{(\rho/2)u_2^2} \cos(\beta + \theta) \frac{br d\theta}{b_2 r_2 \cos \beta} \quad (2)$$

$$\frac{F_{ry}}{(\rho/2)u_2^2 b_2 r_2} = \int_0^{\theta_2} \frac{p_+ - p_-}{(\rho/2)u_2^2} \sin(\beta + \theta) \frac{br d\theta}{b_2 r_2 \cos \beta} \quad (3)$$

The static pressures on the vane p_+ and p_- were calculated by the analytical method of reference (3). The theoretical results obtained for impeller A and impeller C are shown in Fig.10. The calculated values of radial thrust near the best efficiency point are approximately equal to the mean values of experimental results. This result proves that the radial thrust of pumps with a single-vane impeller can be estimated by this analysis provided that the effect of a delivery casing is small. The reason why the calculated values exceed the empirical ones in the low capacity region could be explained as follows. The theory deals with a two-dimensional potential flow. But the actual flow in the impeller was three-dimensional in the low capacity region and a two-dimensional calculation could only give an approximate result.

4.2.3 Effect of the number of vanes

The relationship between the radial thrust and the flow coefficient in pumps with a single-vane impeller and those in ones with a twin-vane impeller is shown in Fig.13. The performance of a single-vane impeller is different from that of a twin-vane impeller as shown in Fig.8. Since the radial thrust varies in proportion to the total head, to see the effect of the number of vane, data of the radial thrust of the twin-vane impeller were modified by multiplying them by the ratio of total head, $H_{\text{single}}/H_{\text{twin}}$. The dash-dot line in Fig.13 indicates the modified values of radial thrust corresponding to DC pump. The effect of the number of vanes was examined by comparing the solid line of AC pump with the dash-dot line of DC pump. It is confirmed from these results that an increase in the number of vanes reduces the radial thrust considerably.

Fig.13 Radial thrust in AC and DC pumps

4.2.4 Lissajous figures of radial thrust

Fig.14 shows the Lissajous figures which indicate the change in the magnitude and the direction of radial thrust vector during one revolution of AV pump. The circumferential position of the impeller changes counterclockwise by 45 deg as the point number increases by one. Fig.15 shows the relative positions of impeller and delivery casing. Fig.16 and Fig.17 show the Lissajous figures of radial thrust of AC and DC pumps respectively. In Figs.14 and 16, the Lissajous figures of radial thrust obtained theoretically are also shown. The Lissajous figures are convex closed curves and the radial thrust vector rotates synchronously with the impeller rotation. In the case of the circular casing, the centers of the closed curves remain close to the center of rotation regardless of the variation in the flow coefficient. In the case of the volute casing, it can be seen that the center of figures moves with a change in the flow coefficient, and the magnitude and the direction of the radial thrust vector change greatly during one revolution of the impeller. The reason why the center of the figures at shutoff is in the second quadrant could be explained as follows. The typical pressure distributions on the volute casing wall are shown in Fig.18 for three flow coefficients. At shutoff the pressure reaches maximum near the exit. Then the radial thrust caused by the pressure distribution on the casing wall acts in the direction of the second quadrant. Thus, the centers of the Lissajous figures are in the second quadrant. The Lissajous figures obtained by Eqs.(1) to (3) agree fairly well with the experimental results at the best efficiency points. The closed curve of Lissajous figures corresponds to

Fig.14 Radial thrust in AV pump

Fig. 15 Relative positions of impeller and delivery casing

the rotation of radial thrust vector of the single-vane impeller. On the other hand, the movement of the center of closed curves shows the effect of the geometries of delivery casing. Then it is confirmed by the Lissajous figures that the radial thrust in the centrifugal pump with a single-vane impeller consists of the force caused by a delivery casing and that caused by an impeller itself. The Lissajous figures of the twin-vane impeller show two closed curves which indicate the effect of the number of vanes.

Fig. 16 Radial thrust in AC pump

Fig. 17 Radial thrust in DC pump

5. Conclusions

The experimental and theoretical studies on the radial thrust in the centrifugal pumps with a single-vane impeller were performed, and the following conclusions were obtained.

- (1) It was confirmed that the radial thrust consisted of the force caused by a delivery casing and that caused by a single-vane impeller itself.
- (2) The geometry of the delivery casing has a great influence on radial thrust. It was made clear that the magnitude of radial thrust for the circular casing was less than that for the volute casing.
- (3) The subtended angle of vane has little effect on the magnitude of radial thrust within the experimental range of the subtended angles.
- (4) It was found that the radial thrust caused by a single-vane impeller could be estimated by a two-dimensional potential flow analysis based on the singularity method.

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Fig. 18 Pressure distribution on the volute casing wall⁽⁴⁾

References

- (1) Agostinelli, A., et al., Trans. ASME, Ser.A, -1 (1960-2), 120.
 - (2) For example, Biheller, H.J., Trans. ASME, Ser.A, (1965-7), 319 ; Stepanoff, A.J., Centrifugal and Axial Flow Pumps, (1967), 116, John Wiley.
 - (3) Matsuoka, Y., Trans. Japan Soc. Mech. Engrs. (in Japanese), Vol.29, No.198 (1963-2), 270.
 - (4) Stepanoff, A.J., Centrifugal and Axial Flow Pumps, (1967), 121, John Wiley.
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