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Performance of Very Low Specific Speed Centrifugal Pumps with Circular Casing*

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Abstract
Efficiency of a centrifugal pump is known to drop rapidly with a decrease of specific speed n_s in the range of $n_s \leq 100$ $[m, m^3/min, min^{-1}]$. However, below $n_s = 60$, the pump efficiency is not yet clear, and the spiral angle of a volute casing becomes too small to manufacture. To solve this problem, a circular casing is considered appropriate in the very low n_s range. The present study is aimed to reveal the relation between pump efficiency and a specific speed in the range of $n_s \leq 60$, when a circular casing is used. The results show that a circular casing gives higher efficiency than a spiral casing, and that radial thrust is considerably small in both casings compared with that of ordinary specific speed pump.

Key words : Turbomachinary, Fluid Machinery, Pump, Very Low Specific Speed, Matching Performance, Circular Casing

1. Introduction

Pump efficiency drops rapidly with a decrease of specific speed n_s in the range of $n_s \leq$ 100 $[m, m^3/min, min^{-1}]$ (Type number $K \leq 0.224$)(1). However, in the range of $n_s \leq 60$, pump characteristics are not clear yet. One of the present authors studied the pump performances of $n_s = 60$ and determined the effects of design parameters on pump performances(2)(3)(4). Moreover, very low specific speed pump tends to have unstable head curve near the shut off point in spite of impeller outlet angle $\beta_2 = 23°$(5). In order to improve unstable head curve, semi open impeller was applied and internal flow and performance of semi open impeller was also clarified(6). Recently, performance improvement based on optimum design method applied to semi open impeller of low specific speed was reported(7).

A spiral casing is used in these studies. However, in the range of $n_s \leq 60$, it is very difficult to attain the manufacturing accuracy, as the spiral angle of the casing becomes very small. Thus, a circular casing is considered suitable as a pump casing in the very low specific speed range to solve the problem of very small spiral angle. However, there are few studies on the flow in a circular casing and radial thrust will be caused even at the best efficiency point, because impeller outlet flow pattern is not uniform.

In the range of normal specific speed, the best efficiency point (BEP) of a centrifugal pump is largely influenced not only by impeller parameters but also by casing parameters. The BEP of a centrifugal pump is determined by the intersection of the head curves of both an impeller and a casing(8). However, the BEP of a very low specific speed pump comes to much higher discharge range than the designed(3)(4). Thus, a relationship between the designed specific speed and the best efficiency point is important in the low n_s range.

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The purpose of the present study is to reveal the pump performances and the matching performances between a casing and an impeller, when a circular casing is used. Two types of casing, a spiral casing and a circular casing, and five impellers are used to determine the matching performance. Moreover, pump performance in the wide range of $n_s \leq 60$ are clarified experimentally.

2. Nomenclature

A : sectional area $[m^2]$
b : channel width $[m]$
F_r : radial thrust $[N]$
H : pumping head $[m]$
L : shaft power $[W]$
n_s : specific speed $[m, m^3/min, min^{-1}]$
n_{sl} : local specific speed $[m, m^3/min, min^{-1}]$
p : pressure $[Pa]$
Q : flow rate $[m^3/s]$
r : radius $[mm]$
u_2 : impeller tip velocity $[m/s]$
Z : number of impeller blades
β : impeller angle $[°]$
ρ : density of fluid $[kg/m^3]$
ϕ : discharge coefficient $= Q/A_2u_2$
ψ : head coefficient $= 2gH/u_2^2$
τ : shaft power coefficient $= 2L/\rho A_2 u_2^3$
η : efficiency $= \rho g Q H/L$
c_p : pressure coefficienct $= 2(p - p_s)/(\rho u_2^2)$
c_F : radial thrust coefficient $= 2F_r/(\rho A_2 u_2^2)$

subscript :
$2, 3$: at impeller outlet, casing
s, v : at impeller suction, casing tongue
$D.P., M.P., BEP$: at design point, matching point, best efficiency point

3. Design guide line of very low n_s pump

In the range of $n_s \leq 60$, the conventional design gives too small impeller outlet width b_2 and too small spiral angle γ of the casing to manufacture(1)(9). Consequently, new design guide line is needed in the range of $n_s \leq 60$. Specific speed is expressed as follows by using non-dimensional parameters;

$$n_s = n\frac{\sqrt{Q_{D.P.}}}{H_{D.P.}^{3/4}} = 1727\sqrt{\frac{b_2}{r_2}}\frac{\sqrt{\phi_{D.P.}}}{\psi_{D.P.}^{3/4}} \qquad (1)$$

Therefore, in order to design a very low n_s pump, there are only two methods. One is to make b_2/r_2 small and the other is to make $\phi_{D.P.}$ smaller and $\psi_{D.P.}$ higher. The former has the limitation of $b_2/r_2 = 0.02$ in manufacturing. Thus, the later is only the method to realize very low specific speed pump.

The pump design point ($D.P.$) is obtained by the intersection between an impeller head-curves ψ_{imp} and a casing head-curves ψ_{vol} as follows(5);

$$\psi_{imp} = 2(1 - k - \phi/\eta_v\epsilon_2 \tan\beta_2) \qquad (2)$$

$$\psi_{vol} = 4\pi b_2\phi/b_3 \ln(1 + A_v/b_3 r_v) \qquad (3)$$

where k is the slip factor(10), $\epsilon_2 = 1 - Zt_2/2\pi r_2 sin\beta_2$ (t_2; blade thickness) and η_v is the volumetric efficiency. Equation (3) is obtained under the assumption that the casing throat flow is a free vortex and the throat section is rectangle.

Figure 1 shows the constant specific speed lines plotted in the $\phi_{D.P.} - \psi_{D.P.}$ plane for the case of $Z = 6, b_2/r_2 = 0.04$ and $b_3/b_2 = 4.0$. Also, ψ_{imp} and ψ_{vol} are plotted in Fig. 1 together with the matching point ($M.P.$) by the symbol of •. ψ_{imp} is mainly influenced by impeller outlet angle β_2 and ψ_{vol} is mainly influenced by the casing throat sectional area A_v. Therefore, the β_2 value is varied from 30° to 90° and the A_v/A_2 value is selected to 0.05 and the half. ($A_v/A_2 = 0.025$)

Figure 1 reveals that $M.P.$ changes little as shown by ① ~ ③, even when β_2 is varied widely. However, $M.P.$ changes largely as shown by ① and ④, when A_v is varied. As a result, specific speed at $D.P.$ is changed largely from 60 to 40. After all, a very low n_s pump can

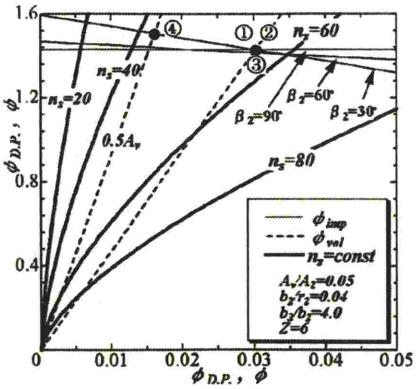


Fig. 1 Design chart of very low specific speed pump

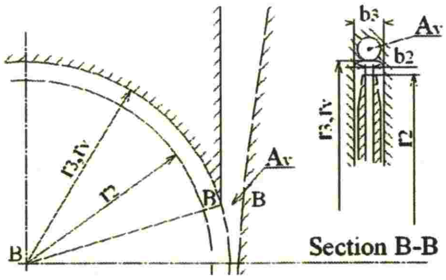


Fig. 2 Configuration of circular casing

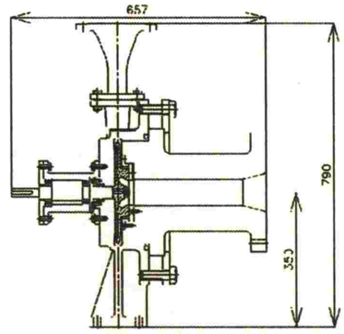


Fig. 3 Configuration of test centrifugal pump

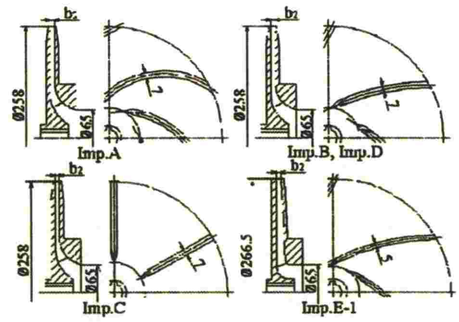


Fig. 4 Configuration of impellers tested

only be realized by reducing the sectional area of a volute tongue. And influence of impeller design parameters on specific speed is relatively small.

A throat sectional area becomes very small with the decrease of specific speed and the effect of roughness in throat becomes relatively large. The circular casing is suitable in the range of very low specific speed, because it is possible to be machined precisely.

For the circular shape of the throat section shown in Fig. 2, ψ_{vol} is gives as follows under the assumption that the casing flow is a free vortex and the velocity in the casing throat section is uniform;

$$\psi_{vol} = 2\frac{v_{u2}}{u_2} = 4\frac{\pi b_2 r_v}{A_v}\phi \qquad (4)$$

4. Experiments

4.1. Experimental Apparatus

Figure 3 shows the test centrifugal pump when circular casing is used. Casings tested as a circular and a spiral are shown in Table 1, where r_3 denotes both the inner radius in the circular casing and the volute basic circle radius in the spiral casing. Both casings are designed for n_s = 60. A casing radius r_3 of a circular casing is determined that the total area in the casing channel becomes same as that of spiral casing. The form of throat in both casings are different, such as rectangle in a spiral casing and circle in a circular casing. Both casings have rectangle volute channel and surface roughness is $R_{max} = 25\mu m$. The roughness at the discharge throat is different in both casing, such as casting in a spiral casing and machined precisely in a circular casing.

In order to determine the matching performance for a circular casing, five impellers are tested as shown in Fig. 3 and Table 2. The impeller has different β_2 and Z. Four impellers

Table 1 Dimensions of casings tested

	Circular casing	Spiral casing
b_3	20.0	20.0
r_3	138.5	133.3
γ	-	0.68
A_v	196.5	201.1

b_3, r_3 : mm A_v :mm^2

Table 2 Dimensions of impellers tested

Imp. No.	β_2	$b_{2design}$	$b_{2meas.}$	r_2	Z
A	30	5.0	5.0 ~5.8	129	6
B	60	5.0	4.7 ~5.7	129	6
C	90	5.0	4.6 ~6.0	129	6
D	60	5.0	5.6 ~6.8	129	4
E-2	60	6.2	6.2	129	6

$b_{2design}$, $b_{2meas.}$, r_2 : mm

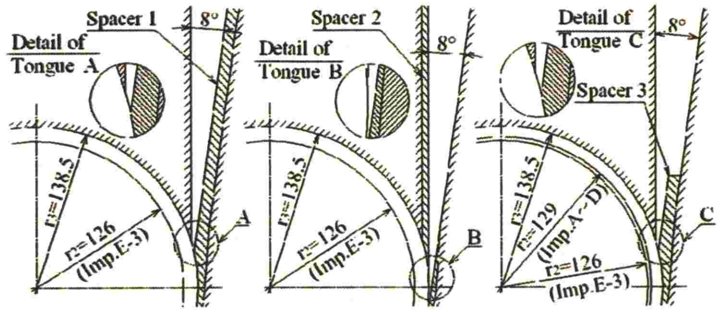


Fig. 5 Configuration of casing throat with spacer

(Imp. A, B, C and D) are made by casting using stainless steel, and the manufacturing accuracy of these impellers is very low as shown in Table 2, because the impeller outlet width is very small compared with the impeller radius. All impeller is designed by conventional design method(1). In order to reveal the influence of dimensional scatter of an impeller on pump performance, the new impeller (Imp. E) is machined precisely. The test Reynolds number $Re = u_2 r_2/\nu$ is $(2.6 \sim 2.9) \times 10^6$.

As the pump performance of a circular casing is not known. The influence of impeller outlet radius r_2 on pump performance is also measured by cutting the outer radius of Imp. E as shown in Table 3.

4.2. Experimental method

In order to change n_s widely in the range of $n_s \leq 60$, the throat sectional area A_v is changed by use of a spacer inserted into a casing throat as shown in Fig. 5. The measured A_v - value is shown in Table 4 for three different specific speed. As shown later, the efficiency of a circular casing is higher than that of a spiral casing, and throat sectional area is reduced only in a circular casing.

5. Results and Discussions

5.1. Comparison of circular casing with spiral casing

The comparison of performance curves between a circular casing and a spiral casing is shown at $n_s = 60$ in Fig. 6. It is clearly seen that the efficiency of a circular casing is higher than that of a spiral casing in all flow range and η_{max} of a circular casing is 7% higher than that of a spiral casing. But head curve of a circular casing has unstable part at low flow rate. In other impellers, the same tendency was obtained. Considering that shaft power τ is same in both casings and impeller outlet pressure distribution is almost same (shown in Fig. 7), the difference of the efficiency is mainly caused by the hydraulic loss in the volute casing and throat channel. In both casings, the maximum efficiency η_{max} is attained at $n_s = 60$. The

Table 3 Dimension of impeller E for impeller cut test

Imp. No.	b_2	r_2	r_3/r_2
E-1	6.0	132.3	1.05
E-2	6.2	129	1.07
E-3	6.3	126	1.10

b_2, r_2 : mm

Table 4 Variation of throat sectional area

No.	A_v-I	A_v-II	A_v-III
Sp. 1	23.2	45.9	98.0
Sp. 2	29.6	45.9	96.2
Sp. 3	23.2	28.6	-

A_v : mm^2

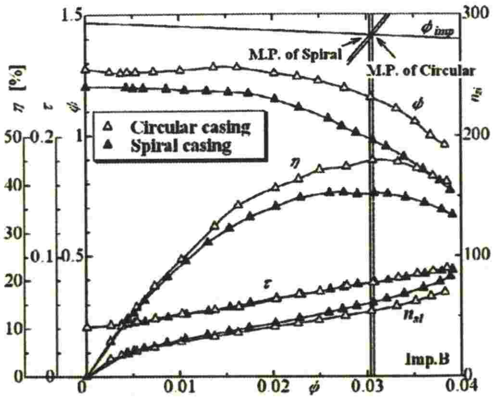
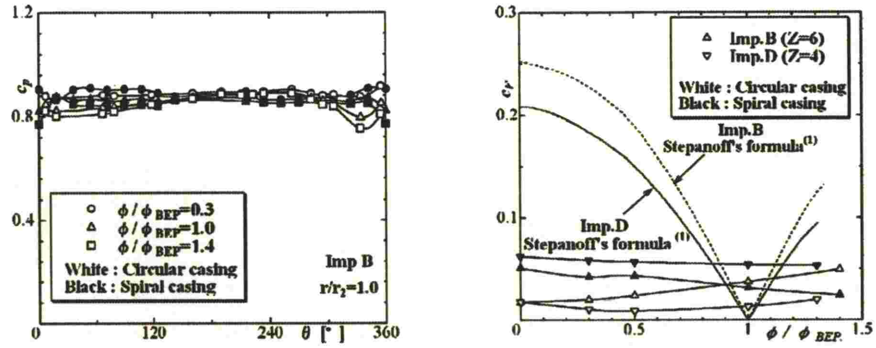


Fig. 6 Influence of casing type on pump peformances



(a) Pressure distribution

(b) Radial thrust

Fig. 7 Influence of casing type on casing wall static pressure and radial thrust

calculated *M.P.* of both casings are in good agreement with the measured ϕ_{BEP}.

When a circular casing is used, the serious problem is radial thrust based on non-uniform impeller outlet flow. In order to clarify the radial thrust, the pressure distribution around an impeller is measured. Figure 7 (a) shows the comparison of pressure distribution in both casings and Fig. 7(b) also shows radial thrust coefficient calculated from the pressure distribution. Figure 7 (b) reveals that the radial thrust is considerably small in both casings compared with that of an ordinary n_s pump, which is because the pressure distribution around an impeller outlet is almost uniform as shown in Fig. 7(a). In Fig. 7(b), the Stepanoff formula(1) is also plotted, and it is seen that the Stepanoff formula is not applicable in the very low n_s range. As a result, the radial thrust is remarkably small compared with that of an ordinary specific speed pump.

5.2. Influence of impeller design parameters

Figures 8 and 9 show the influence of impeller design parameters β_2 and Z on pump performance curves, respectively.

Figures 8 and 9 reveal that the influence of the design parameters of impeller on η_{max} is very small($\pm 1\%$), though the tendency of dependence of performance curves on impeller design parameter is almost same as that in the ordinary n_s range. For example, unstable head characteristics increases with increasing β_2.

Figure 9 reveals that head and shaft power of $Z = 6$ is higher than that of $Z = 4$. This might be caused by the increase of theoretical head based on the decrease of slip factor.

The influence of manufacturing tolerance and internal surface roughness of impeller can be clarified by the comparison of Imp. B with Imp. E-2. The differences of η_{max} and performance curves between Imp. B and Imp. E-2 are seen to be small. It suggests that influence of

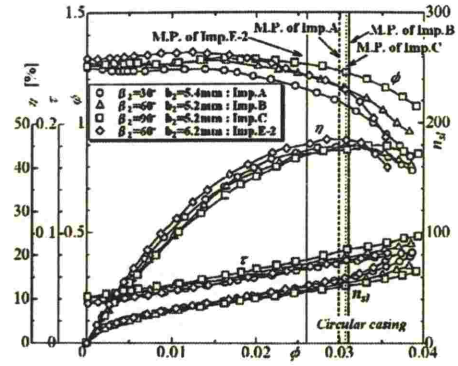


Fig. 8 Influence of impeller outlet angle

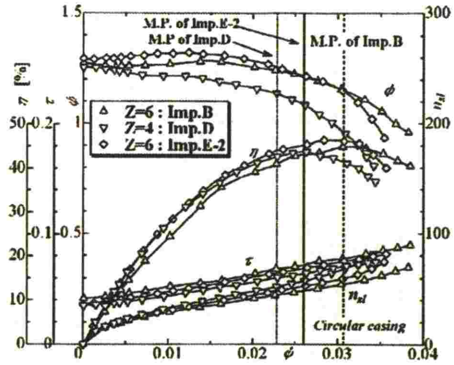
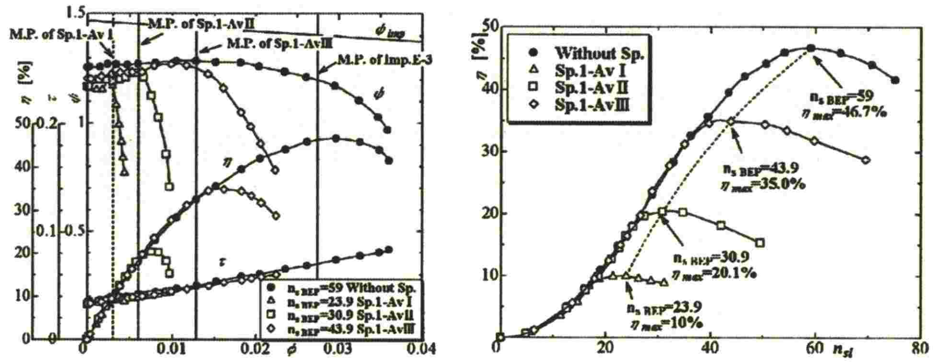


Fig. 9 Influence of impeller number



(a) Performance curves

(b) Efficiency vs. local specific speed

Fig. 10 Influence of volute throat sectional area A_v (Sp. 1)

impeller hydraulic loss on pump performance is not large.

Though not shown in the Figures, the influence of impeller design parameters on pump performance in a spiral casing was almost same as that in a circular casing. These results reveals that the influence of manufacturing tolerance and the impeller design parameters on pump performance is very small in the range of very low specific speed.

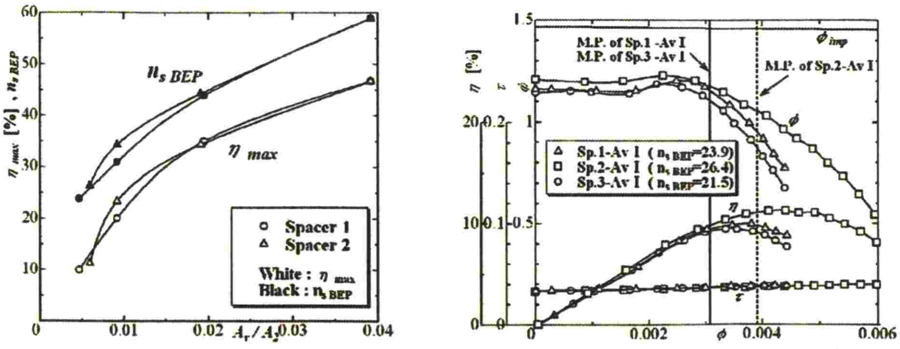
5.3. Performances in the wide range of n_s

Figure 10(a) shows the comparison of pump performance curves when throat sectional area A_v is widely changed by use of spacer 1(Sp. 1), and Fig. 10(b) shows η against local specific speed n_{sl}. The test impeller is Imp. E-2, and test casing is circular casing of $r_3/r_2 = 1.10$.

Figure 10(a) reveals that ϕ_{BEP} and n_{sBEP} decrease largely with the decrease of A_v as was expected, and rapid drop of head is very large when the throat sectional area is reduced. It is also recognized in Fig. 10(a) that the shaft power-capacity curves is almost same, which proves that the impeller internal flow pattern differs little for a large change of n_s. It is also recognized that the head curve becomes more unstable with the decrease of n_s near the shut-off point. The calculated *M.P.* is in good agreement with the measured ϕ_{BEP}.

Figure 10(b) reveals that all the $\eta - n_{sl}$ curves for different n_s come to one curve in the low flow range of each pump. It also reveals that the η_{max} of each pump is about 6% lower than the no spacer case. Therefore, η_{max} of each A_v (dotted line) is lower than η of $n_s = 60$ pump. As a result, higher efficiency of a very low specific speed pump is attainable by a partial operation of a higher n_s pump rather than by the *BEP* operation of low n_s pump.

These results also clarify that the reduction of throat sectional area makes specific speed change widely. In this case, it is important to determine the optimum spacer position and configuration. The values of n_{sBEP} and η_{max} are plotted against A_v/A_2 in Figure 11(a), when



(a) A_v/A_2 vs. η_{max}, n_{sBEP} (b) Influence of spacer type ($n_s \simeq 25$)

Fig. 11 Influence of spacer on pump performance

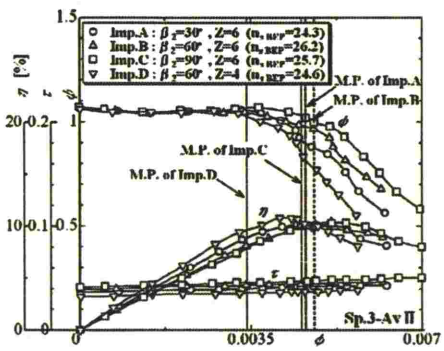


Fig. 12 Influence of impeller design parameter (Sp. 3)

Sp. 1 and Sp. 2 are used. Figure 11(a) reveals that n_{sBEP} and η_{max} decrease largely with the decrease of A_v/A_2, and drops rapidly in the range of $A_v/A_2 \leq 0.01(n_s \leq 40)$. Comparison of Sp. 1 with Sp. 2 in Fig. 11(a) reveals that the effect of spacer position is unexpectedly small.

Figure 11(b) shows the comparison of pump performance curves when three different types of spacer are used. Figure 11(b) reveals that η_{max} is very low, but the difference of η_{max} is small. Here, Sp. 2 gives higher efficiency which is due to higher specific speed, and not due to spacer configuration. Comparison of Sp. 1 with Sp. 3 reveals that the difference of spacer type does not influence on η_{max} and n_{sBEP}.

In addition, there are no studies about pump performance data in the extremely low specific speed range. The influence of impeller design parameters on pump performance at the specific speed 25 is shown in Fig. 12.

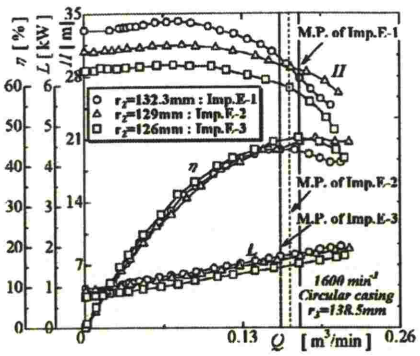
In ordinary, impeller outlet angle β_2 makes head curve change largely. However, head curves from shut off point to BEP are same for every impeller in spite of various β_2 and Z. This is because hydraulic loss at casing tongue is very large. Consequently, the efficiency is almost same in all impellers (±0.5%). The pressure distribution and the radial thrust calculated by pressure are also same result as that in $n_s = 60$, though not shown in figures.

These results suggest that the influence of impeller design parameters on pump performance in the very low specific speed range is very small.

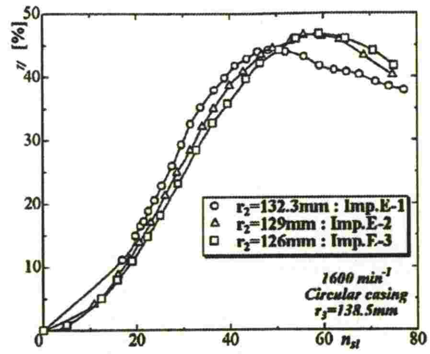
5.4. Optimum design dimension of circular casing and impeller cut test

In the former section, it was made clear that the pump efficiency changes largely with the change of specific speed in the range of very low specific speed. As the throat dimensions of a casing makes specific speed widely change, the best efficiency changes widely as shown in Fig. 10(a). Here, the influence of another dimension of a casing is studied below.

According to Eqs. (2) and (3), the decrease of r_v gives larger $\phi_{M.P.}$ and smaller $\psi_{M.P.}$ for the same impeller, resulting in increase of efficiency due to increase of n_s. On the contrary,



(a) Performance curves



(b) Efficiency vs. local specific speed

Fig. 13 Influence of impeller cut on pump performance

the decrease of r_2 for the same casing results in the increase of efficiency due to increase of n_s defined by Eq. (1). The above discussion seems to be contradictory when $\Delta r = r_3 - r_2$ is considered; efficiency increases when Δr become smaller for the same impeller, but the efficiency increases when Δr become larger for the same casing.

In order to confirm the above inference, impeller radius cut test is performed using Imp. E and the circular casing. Figure 13(a) shows performance curves in dimensional form, and Figure 13(b) also shows the relationship between local specific speed and efficiency. Figure 13(a) shows that power become decrease when impeller outlet radius becomes small, but the best efficiency is improved. This is because the best efficiency point moves to high specific speed, and the inference is confirmed by Fig. 13(a).

As a result, the optimization of a circular casing can be made by increasing the specific speed at the *BEP*. Here, it is interesting that the efficiency of large r_2 case is higher than that of small r_2 case in the range of $n_s \leq 50$. For example, the efficiency of Imp. E-1 is 7% higher than that of Imp. E-3 at $n_s = 30$. This implies that the performance improvement should be attained by the optimization of impeller first and then casing should be optimized.

6. Conclusions

The pump performances and the influence of design parameters are made clear in the very low n_s range ($n_s \leq 60$). The main conclusions obtained are summarized as follows;

(1) The pump maximum efficiency with a circular casing is about 7% higher than that with a spiral casing at $n_s = 60$. Therefore, a circular casing is suitable in the very low specific speed range.

(2) The best efficiency is shifted to low flow rate along the same curve, when the specific speed is decreased by reducing the throat sectional area.

(3) Radial thrust is remarkably small compared with that of ordinary specific speed pump. This is because a pressure distribution around an impeller is almost uniform at all flow range.

(4) Influence of manufacturing tolerance and impeller design parameters on pump performance is very small.

(5) The best efficiency rapidly decreased with the decrease of specific speed. Especially, the amount of decrease is especially sensitive in the range of specific speed $n_s \leq 40$.

(6) Reductions of casing inner radius and cut of impeller outlet radius makes efficiency improve. However, specific speed at the best efficiency point also increases.

(7) The best efficiency point is well predicted by the equation shown in this study.

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