Performances of centrifugal pumps of very low specific speed

JUNICHI KUROKAWA, *Professor, Dept. of Mechanical Engineering, Yokohama National University, 156 Tokiwadai, Hodogaya-ku, Yokohama, 240 Japan*

KAZUNARI MATSUMOTO, *Graduate Student, Dept. of Mechanical Engineering, Yokohama National University, 156 Tokiwadai, Hodogaya-ku, Yokohama, 240 Japan*

JYUN MATSUI, *Associate Professor, Dept. of Mechanical Engineering, Yokohama National University, 156 Tokiwadai, Hodogaya-ku, Yokohama, 240 Japan*

TAKAYA KITAHORA, *Assistant, Dept. of Mechanical Engineering, Yokohama National University, 156 Tokiwadai, Hodogaya-ku, Yokohama, 240 Japan*

ABSTRACT

Pump efficiency drops rapidly with a decrease in specific speed N_s in the very low N_s range. In order to improve a pump efficiency and to establish the optimum design method of a very low specific speed pump in the range of N_s<100 [*rpm, m^3/min, m*], eight kinds of centrifugal impellers were tested in the same volute casing designed at N_s - 60. Three impellers have the same outlet angle of β_2=22.5° with a different N_s ranging from 80 to 200, and the other five have radial vanes (β_2=90°) from N_s =50 to 80.

The results showed that the conventional design is not suitable for a very low N_s pump and that the maximum efficiency is largely influenced by the volute design parameters, such as a volute tongue, a olute width and a gap between a impeller exit and a volute tongue, while the maximum efficiency differs little by the design of impeller. The BEP of a very low *Ns* pump agrees with the desined discharge of a volute and is far from that of an impeller.

1 Introduction

Efficiency of a centrifugal pump decreases rapidly with a decrease in specific speed *Ns*. In the very low *Ns* range, such as *Ns*<70 [*rpm, m^3/min, m*], pump efficiency becomes extremely low and reciprocating pumps have long been used in thevery low *Ns* range. However, not only because of vibration and noise problems but also because of a recent trend toward small-size and high speed, the application of turbo-pumps to such a low *Ns* range is strongly required.

As for the performance characteristics of centrifugal pumps, many studied have long been performed and the optimum design method are established for the range of *Ns*>100, such as Pfleiderer and Petermann(1972)and Stepanoff(1957). However, little is known about the performance characteristics of a very low *Ns* pump, as there are few studies, and it is still difficult to attain high efficiency by a conventional design.

Kurokawa *et al.*(1992, 1996) made extensive studies using seven impellers in a symmetrical channel and revealed that the low efficiency of a very low *Ns* impeller is mainly caused by large disk friction, and that the hydraulic loss is not very large. It is then of key importance to make an impeller diameter as small as possible to reduce disk friction and to make an impeller exit angle as large as possible to compensate the head drop due to diameter reduction in order to attain high efficiency in the very low *Ns* range.

One more serious problem in a very low *Ns* impeller is that the best efficiency point (BEP) comes to much higher discharge than the designed. As Worster(1987) pointed out, BEP is determined by the intersection of the head curves of both an impeller and a volute. It is then necessary to determine the motching performances between an impeller and a volute in order to establish the optimum design method of a very low *Ns* pump.

The present study is aimed to reveal the motching performances, and eight impellers of different specific speed are tested in the same volute casing designed at *Ns* =60. The influence of volute design parameters on the performances of a very low *Ns* pump are also determined experimentally.

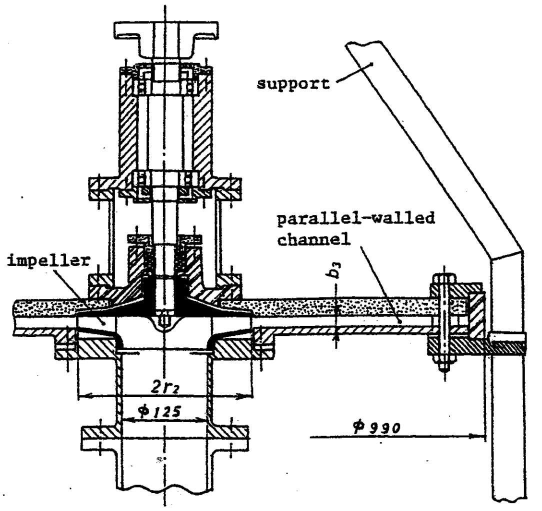


Fig. 1 Free impeller test stand

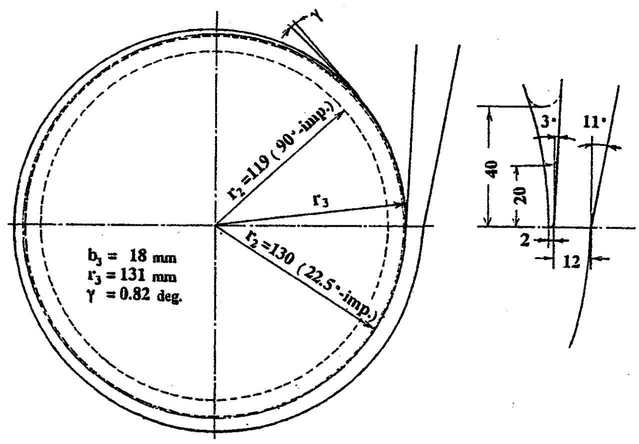
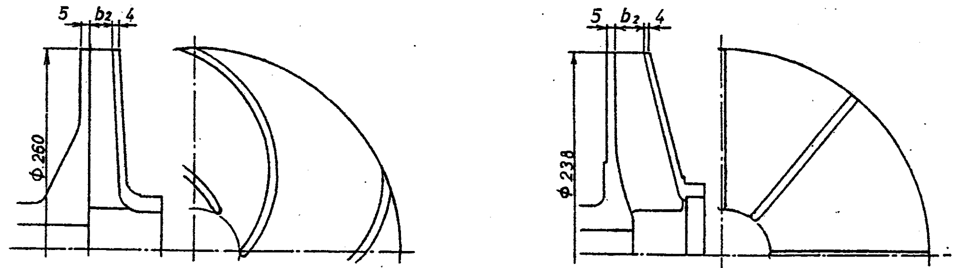
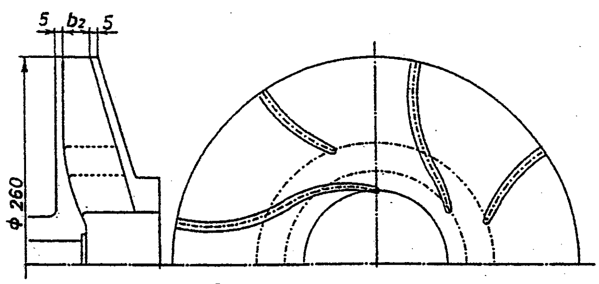


Fig. 2 Configurations of volute



(a) $\beta_2 = 22.5°$: Imp. A, B, C

(b) $\beta_2 = 90°$ (radial vane) : Imp. D, E, F



(c) $\beta_2 = 90°$ (intermediate vane) : Imp. G, H

Imp.	β_2	r_2	b_2	z	N_s
A	22.5°	130	16	5	200
B	22.5°	130	8	5	127
C	22.5°	130	4	5	84
D	90°(radial)	119	18	8	90
E	90°(radial)	119	12	8	70
F	90°(radial)	119	6	8	50
G	90°(interm.)	130	18	8	60
H	90°(interm.)	130	10	8	60

(mm)

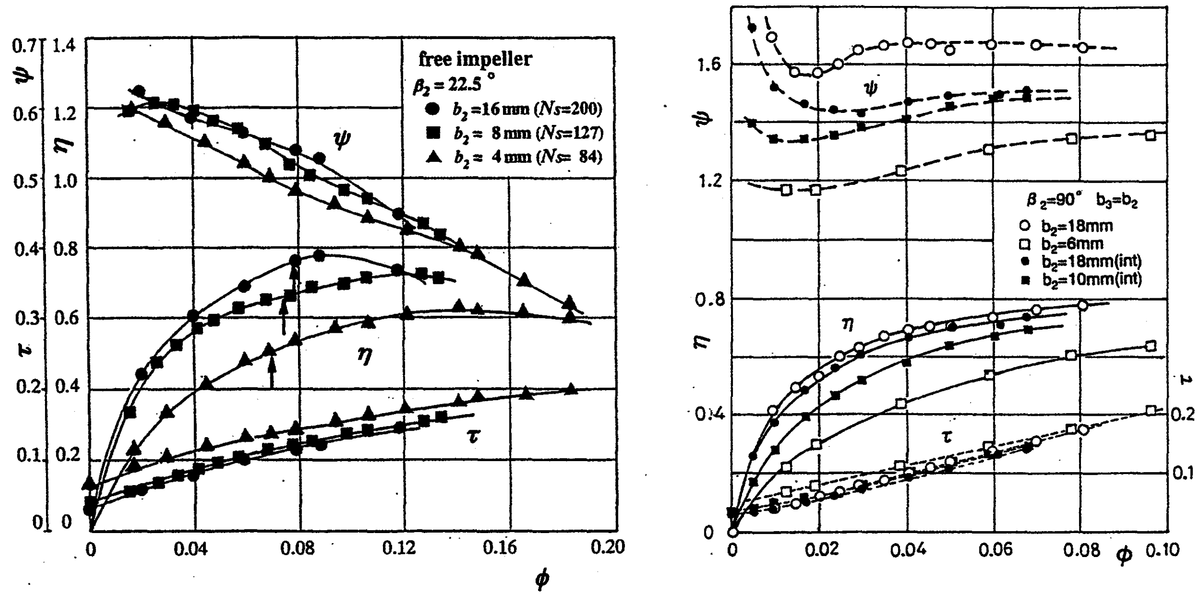
Fig. 3 Configurations and dimensions of impellers tested

2 Test Apparatus

In order to determine the performance characteristics of a very low Ns pump, the test stand of a vertical axis type shown in Fig. 1 is used. Both the free impeller test and the volute pump test can be conducted by removing or inserting a spiral wall into the parallel-walled channel around an impeller. In the free impeller test the total head H is obtained by mass-averaging the total pressure distribution in the parallel-walled channel measured by a three-hole Pitot probe at the radius r=143 mm. In the volute pump test the volute wall of a logarithmic spiral designed under Ns=60 is inserted as shown in Fig. 2. The test Reynolds number $Re=u_2 r_2/\nu$ is $(2.3\sim3.5)\times10^6$.

In order to develop a high efficiency pump in the very low Ns range, eight impellers are tested. Three impellers are designed by a conventional method [Stepanoff] with the outlet angle of $\beta_2 = 22.5°$ and with different outlet width b_2 as shown in Fig. 3(a). The other five impellers have the outlet angle of $\beta_2 = 90°$, three of them have radial vanes, $\beta_1 = \beta_2 = 90°$, as shown in Fig. 3(b), and two of them have intermediate vanes with shockless entry as shown in Fig. 3(c). The specific speed and main dimensions of eight impellers are listed in the figure.

Comparison of the performance characteristics of 3 impellers of $\beta_2 = 22.5°$ should reveal the influence of disk friction, since the main difference among them is the impeller outlet width. Comparison of the performance curves of 5 impellers of $\beta_2 = 90°$ should reveal the new design concept suitable for the very low Ns range.



(a) 22.5°-impeller (Imp. A, B, C) (b) 90°-impeller(Imp. D, F, G, H)

Fig. 4 Comparison of performance curves of free impellers

3 Results and Discussions

3. 1 *Free impeller performances*

The performance characteristics of an impeller alone is compared in Fig. 4, in which those of 22.5°-impeller are compared in Fig. 4(a) and 90°-impeller in Fig. 4(b). The allow in the figure indicates the designed discharge. From Fig. 4(a), the maximum efficiency decreases largely with a decrease in Ns as was expected, but the difference in head-capacity curves is unexpectedly small, which proves that the impeller hydraulic loss per unit impeller width differs little. It is also recognized from Fig. 4(a) that the measured BEP comes to a much higher discharge range than the designed and that low efficiency of the lowest Ns impeller is due to large shaft power. It is then concluded that the low efficiency of a very low Ns impeller is mainly due to large disk friction and that the impeller hydraulic loss differs little from that of a relatively high Ns impeller.

In Fig. 4(b) the head-capacity curves of 90°-impeller have unstable characteristics with positive slope as was expected, and comparison of ●(shockless entry) with ○ reveals that the impeller shock loss is negligibly small. Compared with the 22.5°-impeller, the ψ-values are much higher over all discharge range, which suggests that the 90°-impeller has very low Ns. In the efficiency curves it is recognized that the maximum efficiency of 90°-impeller is not low and is comparable to that of 22.5°-impeller of an ordinary specific speed. However, in the case of b_2=6mm, the maximum efficiency is low and the BEP comes to a very high discharge range.

In order to develop a design concept suitable for the very low Ns range, the relations between efficiency η and specific speed Ns are plotted in Fig. 5. It is apparently shown that the 90°-impeller with the largest blade exit width of b_2=18mm takes the highest efficiency over the whole N_s range. It is also recognized that the 22.5°-impeller with the smallest blade exit width of b_2 = 4 mm takes the lowest efficiency over all Ns range. Since the latter was designed to fit for Ns =84 by a conventional method, it should attain high efficiency in the low Ns range, but the larger exit width of b_2 = 8 mm attains much higher efficiency. The conventional design is thus not suitable for the very low Ns range, and too small exit width should be avoided.

It is thus concluded that the 90°-impeller with relatively large impeller exit width should be used for a very low Ns impeller and should be improved to avoid unstable characteristics. The second choice might be the use of 22.5°-impeller with large exit width. In either case, the maximum efficiency of the impeller comes to a much higher discharge range than the designed, even if the impeller is designed to have a very low specific speed.

In order to reduce disk friction it is recommended to increase the fluid rotational speed in the impeller back space. According to Kurokawa *et al.* (1988), the radial inward leakage increases the fluid rotation in the back space largely, which suggests that the increase in leakage decreases the disk friction. To confirm this, the seal gap of the Imp. F is increased from 0.4 mm to 2.4 mm and the performance curves are compared in Fig. 6. It

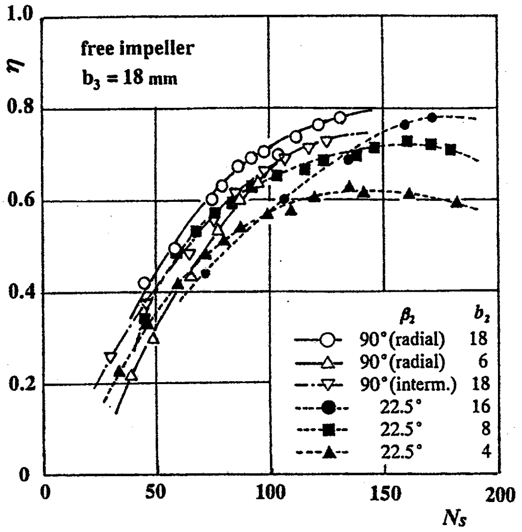


Fig. 5 Efficiency η vs. specific speed Ns of impeller

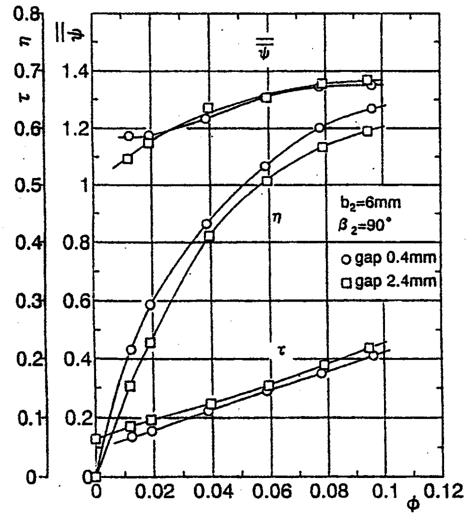


Fig. 6 Influence of seal gap on impeller performances

is recognized in Fig. 6 that the efficiency curve changes little, though the leakage should increase to about 6 times. In an ordinary impeller such a rapid increase in seal gap causes considerable drop of efficiency, but in the very low N_s impeller the increase in shaft power caused by the increase of leakage is compensated by the decrease in disk friction power. This suggests that there exists the optimum seal gap in a very low N_s impeller.

3. 2 *Volute pump performances*

When a spiral wall is inserted around a very low Ns impeller, the pump performance curves change remarkably as shown in Fig. 7, where the efficiency curve of the free impeller G is reproduced by a chain-dotted line for the comparison with ○. Comparison of Fig. 7 with Fig. 4 reveals that the spiral wall causes rapid drop of efficiency due to large hydraulic loss of a volute channel and hat the BEP agrees with the volute designed discharge. This shows that the volute design influences much more than impeller design on the pump performances. It is also recognized that the maximum efficiency is the lowest in the 22.5°-impeller of a wide exit width b_2 (high Ns), though it was the highest in the free impeller test. The smaller b_2 impeller attains higher efficiency, though it attained much lower efficiency in the free impeller test. It is also recognized that the performance instability with positive slope of head curve appears in the low discharge range of almost all impellers.

In order to discuss about a high efficiency pump in the very low Ns range, the efficiency curves in Fig. 7 are replotted against N_s in Fig. 8. Compared with Fig. 4 of a free impeller, the BEPs of almost all pumps agree with the volute designed Ns, and the difference of maximum efficiency is small except for the high Ns impeller shown by ●. It is then concluded that the maximum efficiency of a very low N_s pump is mainly determined by a volute design and the influence of an impeller design is rather small except for a high Ns impeller.

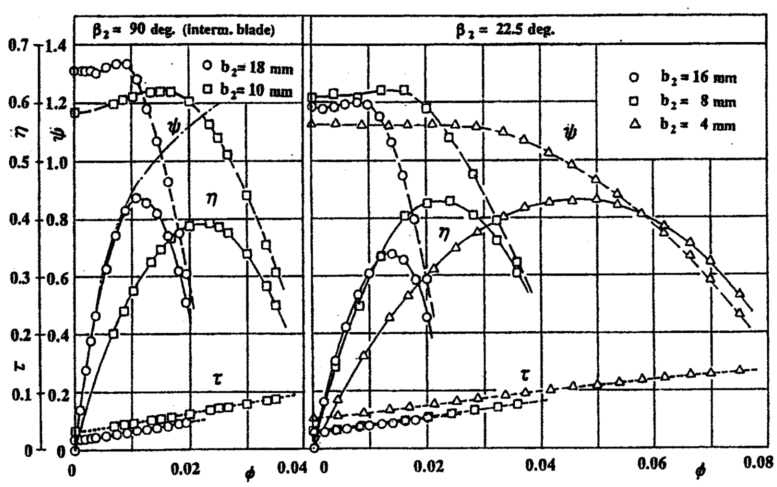


Fig. 7 Comparison of pump performance curves (r_2=130mm, b_3=18mm) (volute tongue 20mm cut)

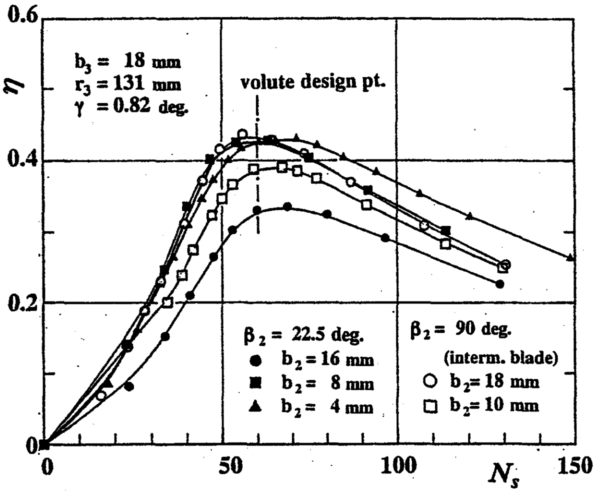


Fig. 8 Pump efficiency η against Ns

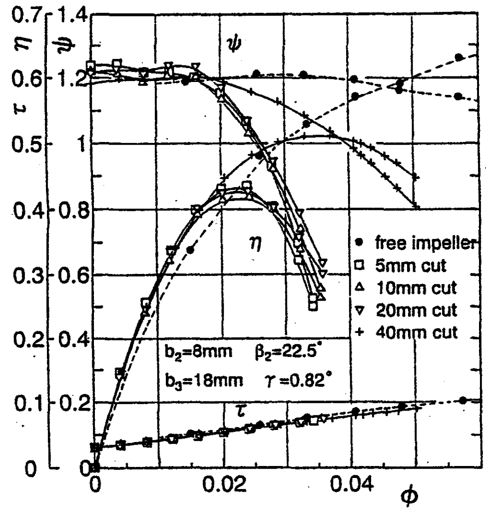


Fig. 9 Effect of cut of volute tongue

3.3 *Effect of volute design parameters*

In order to determine a high efficiency pump, the influence of volute design parameters such as a volute tongue, a volute width b_3 and a gap between an impeller exit and a volute tongue on the pump performance characteristics are studied using the impellers B, D and G.

When the volute tongue is cut short by 5, 10, 20 and 40 mms as shown in Fig. 2, the pump performance curves change as illustrated in Fig. 9. In a pump of an ordinary N_s the effects of cutting the volute tongue is almost equivalent with the increase in the volute exit area, and the discharge at BEP usually increases. However, in a very low Ns pump the cutting of a volute tongue causes a small change in the maximum efficiency until 20 mm cut without changing the BEP discharge and causes a sudden and rapid increase in the maximum efficiency at 40 mm cut with an increase in the BEP discharge. The optimum value is not yet determined, but a considerable increase in efficiency is expected by cutting the volute tongue more.

When the volute channel width is varied with keeping the volute design discharge same, the pump performance curves change as illustrated by ○, △ and ▽ in Fig. 10, in which the case of larger volute designed discharge is also plotted by □. The narrower volute width than the impeller exit width (b_3~b_2/2, indicated by △) produces better efficiency than the equal width case(○), but too narrow case (b_3=b_2/3, ▽) decreases a pump efficiency. This might be due to strong reverse flow re-entering from a volute channel into an impeller. This reverse flow decreases with a decrease in the volute width but too narrow volute channel creates large friction loss instead and decreases the pumping head and efficiency as shown in Fig. 10. When a volute is designed to have higher

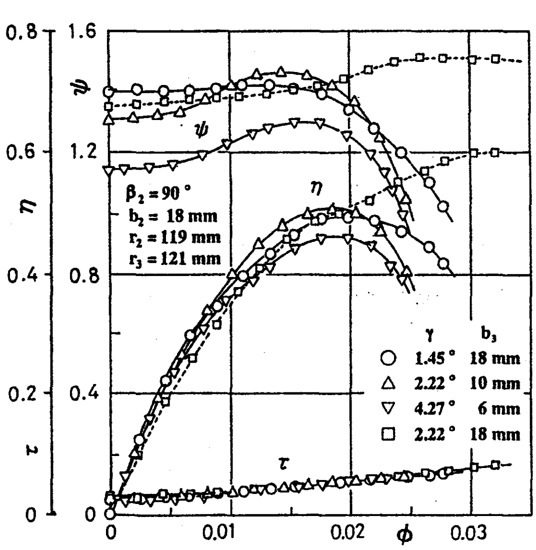


Fig. 10 Effect of volute width

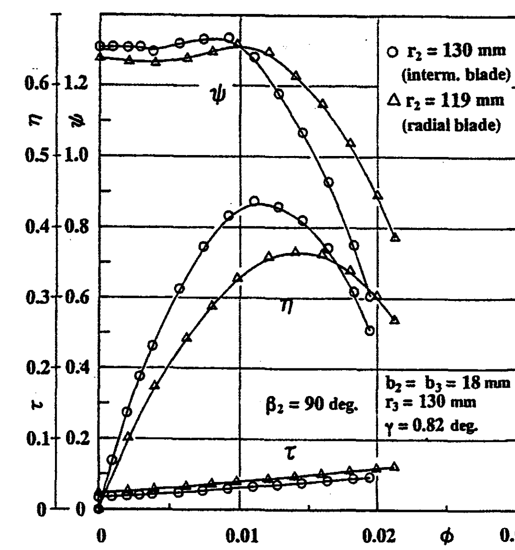


Fig. 11 Effect of gap between impeller and volute tongue

Ns, the BEP discharge increases much as shown by □, but the maximum efficiency does not rise in the low discharge range. It is thus concluded that there exists the optimum volute width determined by the balance between the increase of shaft power due to reverse flow and the decrease of pumping head due to friction loss. When the gap between a volute tongue and an impeller exit is increased, the pump efficiency decreases largely. The performance curves of two impellers of different diameter, r_2=119 mm (Imp. D) and 130 mm (Imp. G), inserted in the same volute are compared in Fig. 11. Though the free impeller test revealed that the smaller diameter attained higher efficiency as shown in Fig. 4(b), the pump test reveals that the larger diameter attains much higher efficiency and the BEP comes to a smaller discharge range. In this case, the pressure fluctuation at the tip of volute tongue was not large, though it increases considerably with a decrease in the gap in an ordinary impeller. Thus a very small gap is recommended to improve pump efficiency in a very low *Ns* pump.

4 Conclusions

(1) In order to design a high efficiency pump in the very low *Ns* range, the optimization of volute design parameters is of key importance, and a conventional design is not suitable. The volute design influences the maximum efficiency of a pump much, while an impeller design influences little as far as the impeller exit width is not too large. A too large impeller exit width should be avoided.
(2) The BEP of a very low *Ns* pump agrees with the volute design point, and is little influenced by the impeller design. The BEP of a very low *Ns* impeller alone comes to much higher discharge range than the designed.
(3) There exists the optimum value in cutting the volute tongue, and the pump efficiency rises much with an increase in the volute tongue cut. There also exists the optimum value in the volute channel width, and as for the gap between an impeller exit and a volute tongue the pump efficiency rises with a decrease in the gap. The pressure fluctuation is not large even if the gap is very small.
(4) In the low discharge range the performance instability appears in almost all of the pumps tested.
(5) Large gap of impeller seal causes relatively small decrease in pump efficiency.

Acknowledgment

The authors express their sincere gratitude to Mr. T. Tsutsui of Nikkiso Co. Ltd. for the financial supported and to Mr. W. Yao for conducting the measurements.

Notations

b ; channel width (mm)
N_s ; specific speed [rpm, m^3/min, m]
r ; radius (mm)
U_2 ; impeller tip speed (m/s)
z ; impeller number
β_2 ; impeller exit angle (deg.)
η ; pump efficiency
ϕ ; discharge coefficient ($= Q/2\pi b_2 r_2 U_2$)
ψ ; head coefficient ($= 2gH/U_2^2$)
τ ; shaft power coefficient($= T/\rho\pi b_2 r_2 U_2^3$)
Subscripts 2 ; at impeller exit
3 ; at volute inlet circle

References

KUROKAWA, J., YAMADA, T. and HIRAGA, H.(1992), Performances of Low Specific Speed Pumps, *Proc. of 11th Austarlasian Fluid Mechanics Conf.* (Tasumania, Australia), Vol. 1, pp.861-864.

KUROKAWA, J., KITAHORA, T. and TSUTSUI, T.(1996), Performances of Centrifugal Impeller of Very Low Specific Speed, *Proc. of 1st Int. Sympo. on Fluid Eng.* (Beijing, China), Vol. 1, pp.276-282.

KUROKAWA, J., SAKUMA, M.(1988), Flow in a Narrow Gap Along an Enclosed Rotating Disk with Through-flow, *JSME International Jr.*, Ser. 2, Vol. 31, No. 2, pp. 243-251.

PFLEIDERER / PETERMANN (1972), Stromungsmaschinen, 4 Auflage, Springer-Verlag, S. 128-216.

STEPANOFF, A. J. (1957), Centrifugal and Axial Flow Pumps (2nd ed.), *John Wiley and Sons*, pp.69-89.

WORSTER, R. C. (1963), The Flow in Volutes and Its Effect on Centrifugal Pump Performance, *Proc. Inst. Mech. Eng.*, Vol. 177, No. 31, p. 843.