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PERFORMANCE OF LOW SPECIFIC SPEED PUMPS

Junichi KUROKAWA[1], Takeshi YAMADA[2] and Hiroshi HIRAGA[1]

[1]Yokohama National University, Hodogaya, 240 Yokohama, JAPAN

[2]Tokyo Grass Company Ltd, JAPAN

ABSTRACT

An efficiency of a very low specific speed pump is very low. In order to reveal the reason and to improve efficiency, two kinds of experiments are performed together with the Q3D analysis. One is the pump performance test using 12 combinations of 6 impellers and 2 volutes, and the optimum combination and the matching performance of an impeller and a volute are determined. The other is the free impeller test using three impellers of different specific speed, and the reason of low efficiency is determined. It is mainly due to the disk friction and the formation of a reverse flow at the BEP in an impeller channel, which increases slip velocity and hydraulic losses.

NOTATION

b_2 ; impeller outlet width
d_2 ; impeller diameter
Ns ; specific speed[rpm,m^3/min,m]
v_{u2} ; outlet tangential velocity
β_2 ; impeller outlet angle
ϕ ; flow-rate coefficient(= $Q/\pi b_2 d_2 u_2$,
ψ ; head coefficient(= $2gH/u_2^2$,
τ ; shaft power coefficient(= $2T/\rho\pi b_2 d_2 u_2^3$,
b_3 ; volute width
k ; slip factor
u_2 ; tip speed
η ; efficiency
η_v ; volumetric eff.
Q ; flow-rate)
H ; pumping head)
T ; shaft power)

1. INTRODUCTION

An efficiency of a centrifugal pump decreases largely with a decrease in pump specific speed Ns, and in a very low Ns range a positive displacement-type has been widely used. But the demand for decreasing noise and vibration requires an adoption of turbo-type pumps to a very low Ns range.

An impeller itself has in general high efficiency in a wide flow-rate range, but the high efficiency range becomes very narrow when it is inserted in a volute. A pump efficiency is thus determined not only by an impeller but by a volute performance, and in a very low Ns impeller the influence of a volute is especially large. But little is known about the optimum combination of an impeller and a volute (Bowerman and Acosta, 1957 and Worster, 1963) and it is difficult to attain high efficiency in the range of Ns<100[rpm,m^3/min,m].

The present study is intended to reveal why the efficiency of a very low Ns pump is very low and to improve an efficiency, and two kinds of experiments are performed. One is the pump performance test using twelve combinations of six impellers and two volutes and the other is the free impeller test using three impellers of different Ns. The results are compared with the calculated results by Q3D method.

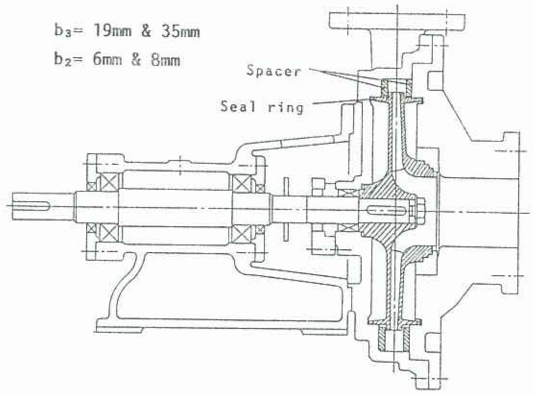

Fig.1 Volute pump used for matching performance test

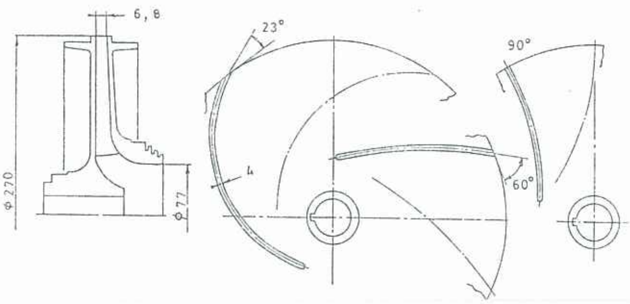

Fig.2 Impeller configurations tested

2. MATCHING PERFORMANCE BETWEEN IMPELLER AND VOLUTE

The matching performances of an impeller and a volute are measured using 12 combinations of impellers and volutes. Six impellers of different b_2 and β_2 shown in Fig.1 are combined with two volutes of different b_3 as shown in Fig. 2. They are designed in a conventional method (Stepanoff, 1957) under the condition of Ns=70 and 90[rpm,m^3/min,m]. The annular rings attached at the back of the impeller shrouds for the purpouse of a radial thrust test make the pump efficiency necessarily worse. The impeller speed tested is 1470 rpm and the corresponding tip speed Reynolds number $d_2u_2/\nu = 2.3\times10^6$.

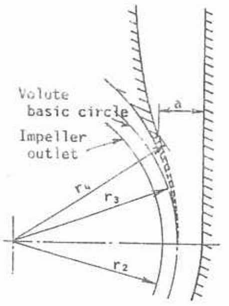

Fig.3 Dimension of volute tongue

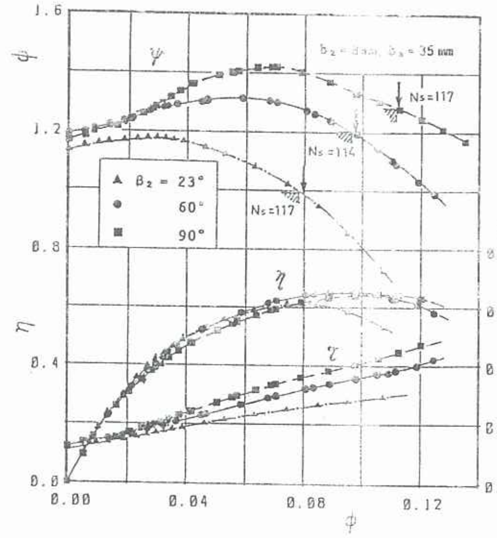

Fig.4 Performance curves (b_2=8mm, b_3=35mm)

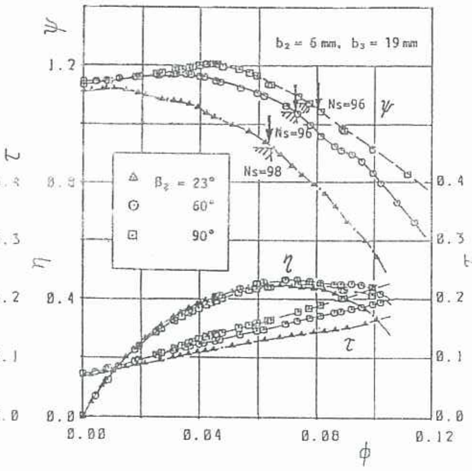

Fig.5 Performance curves for b_2=6mm, b_3=19mm

2.1 Determination of Matching Point

The best efficiency point (BEP) of a pump is so determined that the impeller outlet flow fits for the volute configuration, that is, the intersection point between the impeller head curve ψ_{imp} and the volute performance curve ψ_{vol} (Worster, 1963). In a very low Ns impeller the ψ_{vol}-ϕ curve has a sharp gradient and its small change induces large change in the BEP, and then for the accurate determination of the BEP a more accurate expression of ψ_{vol} than that of Worster is required.

The impeller outlet flow characteristics are given as

$$\psi_{imp} = 2v_{u2}/u_2 = 2(1 - k - \phi\cot\beta_2/\eta_v\varepsilon_2) \quad \text{...............(1)}$$

where $\varepsilon_2 = 1 - zt_2/\pi d_2\sin\beta_2$ (z;vane number, t_2;vane thickness).

The flow at the volute tongue shown in Fig.3 is approximated to a free vortex at the BEP and the velocity integration in the throat section yields the volute performance as follows.

$$\psi_{vol} = 2v_{u2}/u_2 = 4\pi b_2\phi/b_3\ln(1+a/r_4) \quad \text{...................(2)}$$

Hence the intersection of two curves (1) and (2) is given as

$$\phi_{BEP} = (1-k)/\{\cot\beta_2/\eta_v\varepsilon_2 + 2\pi b_2/b_3\ln(1+a/r_4)\} \quad \text{.....(3)}$$

2.2 Experimental Results and Discussions

The performance curves of the three impellers of different β_2 are illustrated in Fig.4 for the case of b_2 = 8mm and b_3 = 35 mm and in Fig.5 for the case of b_2 = 6mm and b_3 = 19mm. It is recognized that the efficiency curves and the shut-off heads differ little for large difference in β_2, though the head curves differ largely. The measured BEPs shown by the notation are compared with the calculated ones from Eq.(3) shown by ↑ and they are in very good agreement. The Ns value at the initial design were 70 for Fig.4 and 90 for Fig.5, but the measured results are about 100 and 120 respectively, that is, about (30-40)% higher than the designed, which results that it is difficult to get a very low Ns impeller by a conventional design.

In order to separate the influences of b_2 and b_3, the performances for the case of the different b_2 and the same b_3 are compared in Fig.6 and those for the different b_3 and the same b_2 in Fig.7. From these figures the followings are recognized;

(1) The decrease in b_2 causes a little change in the ψ-ϕ curve, an increase in the ϕ value at the BEP, and an uniform increase in the τ-ϕ curve, which results in an uniform drop of the η curve, but the Ns value at the BEP changes little.

(2) The decrease in b_3 causes a large drop in the ψ-ϕ curve, remarkable in a large ϕ range, and takes the BEP to a very low

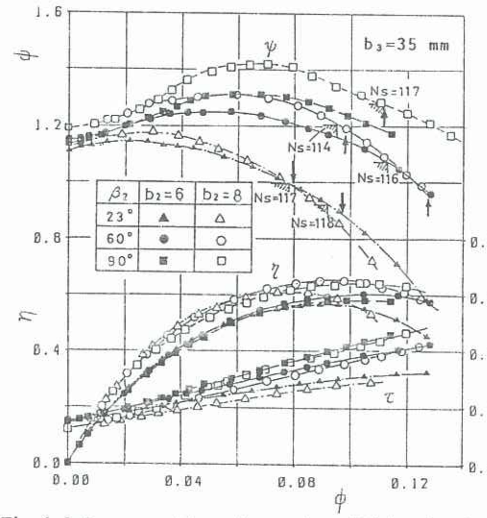

Fig.6 Influence of impeller outlet width b_2 (b_3=35mm)

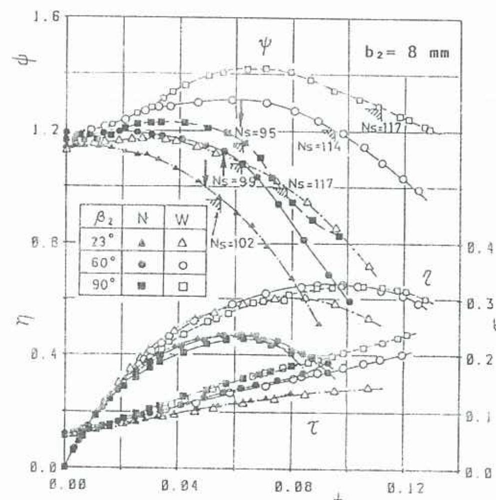

Fig.7 Influence of volute width b_3 (b_2 =8mm)

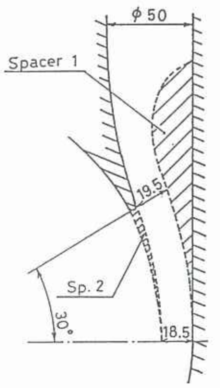

Fig.8 Spacer inserted in volute throat

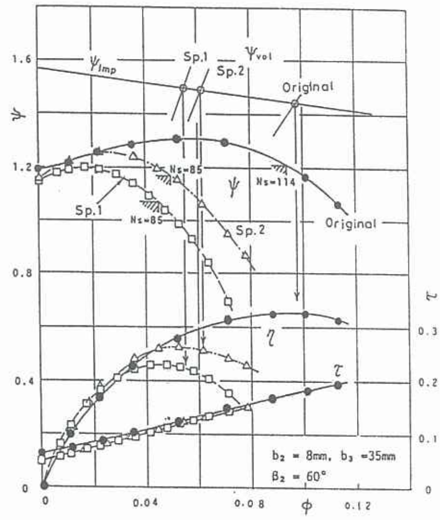

Fig.9 Effect of spacer

flow-rate range, which causes a large decrease in the efficiency curve, and the Ns value at the BEP decreases largely.

From the above it is then suggested that the combination of an impeller of a larger b_2 and a volute of a smaller b_3 gives better efficiency than that of a smaller b_2 and a larger b_3 for the design of a very low Ns impeller. In a conventional design a very low Ns impeller has a very small b_2 and a large β_2, but it is revealed that a small b_2 and a large β_2 are not recommended from the viewpoint of low efficiency and unstable head curve.

As the volute width b_3 represents a volute throat area, and its influence is proved remarkable in the above, the throat area must be a key parameter to the performance of a very low Ns pump. The throat area is controlled by inserting the spacer shown in Fig.8 and the change of the performance curves is illustrated in Fig.9. The followings are recognized;

(1) A very low Ns value is attained at the BEP by inserting a spacer, and the shut-off head changes little by the spacer.

(2) The maximum efficiency attained by use of the Spacer 2 is higher than that attained by use of a small b_2 or a small b_3, which suggests that it is more profitable to control only the throat dimensions than to adjust the volute whole dimensions.

(3) The calculated BEPs from Eq.(3) are in good agreement with the measured values even for such an irregular throat.

However, the maximum efficiency for the case of Ns<100 is very low and is about 20% lower for the case of Ns=85 than that for Ns=115. To reveal this reason it is necessary to know an impeller performance alone of a very low Ns.

3. FREE IMPELLER PERFORMANCE

In order to determine the performance of a very low Ns impeller alone, the free impeller test is performed using three impellers, Ns=84, 127 and 200 shown in Fig.10. The test stand is a vertical axis type as shown in Fig.11 and the velocity and the pressure distributions were measured by three-hole Pitot probes at several sections behind the impeller outlet.

3.1 Overall Performance of Free Impeller

The performance curves are shown in Fig.12. The comparison of curves reveals that low efficiency of the lowest Ns impeller is due to a large shaft power. The head curves reveal that the difference of hydraulic loss in an impeller channel is

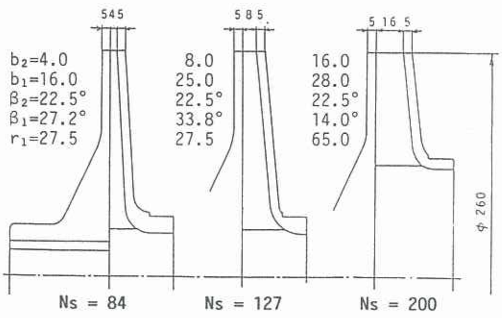

Fig.10 Impeller dimensions used for free impeller test

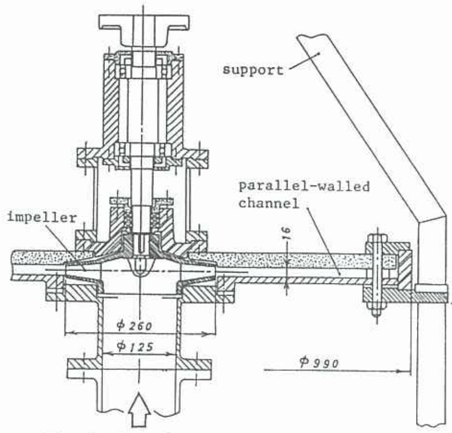

Fig.11 Free impeller test stand

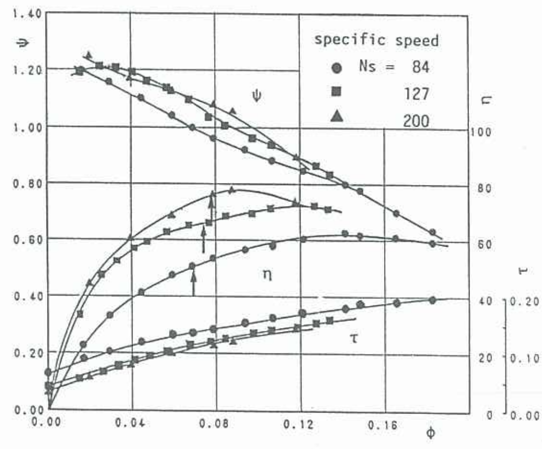

Fig.12 Performance curves of impeller alone

comparatively small in the three impellers, which implies that the impeller hydraulic loss per unit discharge (unit width) differ little. It is also recognized that the measured BEP comes to a much higher flow-rate range than the designed (shown by ↑) and that the high efficiency range is very wide in the lowest Ns impeller, though the maximum value is low.

In order to reveal the reason of low efficiency of a very low Ns impeller, the net power supplied to the fluid is calculated by subtracting the disk friction τ_f estimated by Kurokawa (1988) from τ and is shown in Fig.13. The net power is nearly equal in two impellers, and accordingly the theoretical head also becomes nearly equal. It is then concluded that the low efficiency of a very low Ns impeller is mainly due to a large disk friction and that the impeller hydraulic loss differs little from that of a relatively high Ns impeller.

3.2 Slip Factor and Impeller Hydraulic Loss

The flow velocity at an impeller outlet is usually estimated by use of a slip factor defined as $k=(v_{u2} - v_{u2})/u_2$. The slip factor obtained by the measured velocity at $r/r_2=1.1$(Kurokawa 1985) is compared with those by the measured shaft power and by the Wiesner formula (1967) in Fig.14. A slip factor is usually treated as a constant for the flow-rate variation (see the curve of Ns=127), but in the case of Ns=84 both the measurements and the Q3D calculation reveal that it varies largely with ϕ, and that the Wiesner formula gives very high values for the case of Ns=84. It is to be remarked that the slip factor becomes very small in the large flow-rate range in the case of Ns=84 resulting high pumping head and high efficiency (see Fig.13).

In order to examine the above-described behavior of a slip factor for the case of the Ns=84 impeller, the velocity diagrams obtained by the Q3D calculation are illustrated in Fig. 15. A large reverse flow region is formed even at the designed flow-rate($\phi/\phi_0=1.0$), and it disappears at the flow-rate of about 300% of the designed. This reverse flow makes the slip velocity vary largely with the flow-rate.

As described above, the slip factor of a very low Ns impeller varies largely with ϕ, which causes non-linear variation of the theoretical head against the flow-rate, while it varies linearly in an ordinal impeller. The hydraulic loss in an impeller channel is calculated by subtracting a measured head from a theoretical one, and is compared in Fig.16 for the case of Ns=84. The hydraulic losses obtained from the shaft power measurements, the velocity measurements, the Q3D calculation and the Wiesner formula are compared. There are several differences among them but the shaft power measurements, the velocity measurements and the Q3D analysis give nearly the same tendency that the hydraulic loss takes the minimum around the BEP, but the Wiesner formula gives very different behavior showing a monotonous decrease with the flow-rate. Hence, in the design of a very low Ns impeller the more accurate values of a slip factor is required taking the variation of a slip factor with ϕ into consideration than the conventional formula.

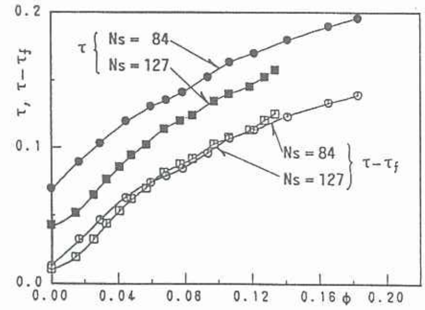

Fig.13 Comparison of shaft power τ and net power (τ-τf)

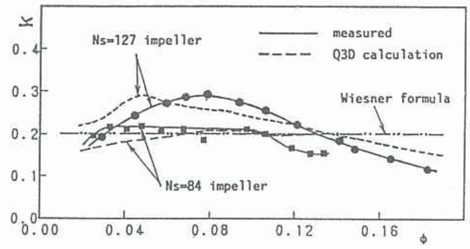

Fig.14 Comparison of slip factor in two impellers

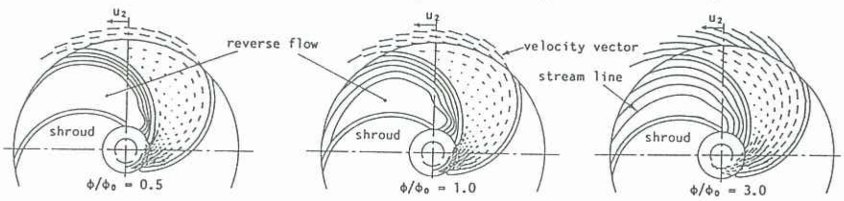

Fig.15 Q3D calculation (Ns=84)

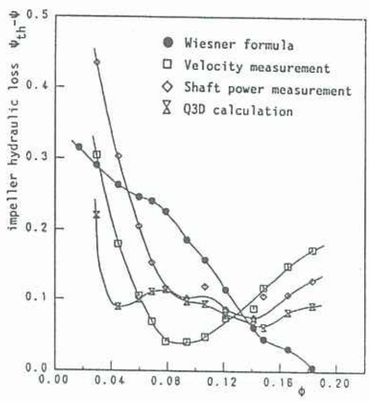

Fig.16 Hydraulic loss in impeller channel

4. CONCLUSIONS

(1) In a very low Ns impeller the rate of disk friction is considerably large, which makes an impeller efficiency low and a large reverse flow region is formed in an impeller channel even at the BEP, which makes a slip factor relatively large and induces hydraulic loss. But the impeller hydraulic loss is not very large compared with an ordinal impeller. In a large flow-rate range a slip factor and a reverse flow region become small, which makes a high efficiency range very wide.

(2) It is difficult to design a very low Ns impeller with good efficiency by a conventional method, as the slip factor changes largely with the flow-rate. It is recommended to design the volute width b_3 narrow and the impeller width b_2 wide to attain high efficiency, and adoption of large β_2 is not desirable. It is more profitable to control only the volute throat by inserting a spacer as proposed here. The decrease in b_2 causes a uniform drop of the η curve and the decrease in b_3 a sharp drop.

(3)The best efficiency point of a pump is accurately determined by Eq.(3), which is also applicable for an irregular throat configuration.

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