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Secondary Flow Due to the Tip Clearance at the Exit of Centrifugal Impellers

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ABSTRACT

The velocity distribution was measured at the exit of two different types of unshrouded centrifugal impellers under four different tip clearance conditions each; one with twenty radial blades and inducers and the other with sixteen backward-leaning blades. And the effect of tip clearance on input power was also measured. By increasing the tip clearance, the input power was hardly changed in the radial blade impeller and was reduced in the backward-leaning blade impeller. The velocity distribution normalized by the passage width between hub and shroud wall was hardly changed at the exit of the radial blade impeller by varying the tip clearance, on the other hand, the relative flow angle was reduced significantly and monotonously by an increase of tip clearance in the backward-leaning blade impeller. The change in input power due to the tip clearance was clearly related to the change of flow pattern at the exit of impeller due to the secondary flow, which is most likely caused by the component, normal to the blade, of the shear force to support the fluid in the clearance space against the pressure gradient in the meridional plane without blades.

NOMENCLATURE

- b = blade height
- c = tip clearance
- k = slip coefficient
- m = meridional length along the shroud
- p = pressure
- r = radius
- R = radius ratio refered to the impeller radius
- U_2 = peripheral speed of impeller outer diameter
- Vm = meridional component velocity normalized by U_2
- Vu = tangential component velocity normalized by U_2
- W = relative velocity normalized by U_2
- Wb = component of W along the blade
- Wn = component of W normal to the blade
- y = axial distance from the hub surface along the blade height
- y' = axial distance from the shroud wall surface
- Z = number of blades
- β = relative flow angle from circumference
- βb = blade angle
- η = impeller efficiency
- λ_2 = tip clearance ratio at impeller exit
- ρ = density of fluid
- τ = shear force due to meridional pressure gradient ; defined by equ.(2)
- ϕ = flow coefficient at impeller exit
- ϕn = secondary flow coefficient
- ψ = pressure coefficient of impeller

SUBSCRIPT

- 1 = impeller inlet
- 2 = impeller exit
- 3 = diffuser inlet
- s = static pressure
- t = total pressure
- PS = pressure surface
- SS = suction surface

INTRODUCTION

In recent high pressure ratio centrifugal compressors, blade height is short near the impeller exit due to the high pressure ratio and the high rotational speed. In such cases, the ratio of the tip clearance to the blade height is relatively large, and deterioration of the compressor performance due to the tip clearance is not negligible. Up to now many studies have been conducted experimentally and theoretically, and several models have been proposed to predict the tip clearance loss and the efficiency drop by Pfleiderer (1961), Eckert & Schnell (1961), Hesselgreaves (1969), Lakshminarayana (1970) and Senoo & Ishida(1986,1987). Engeda & Rautenberg(1987) studied also the effects of tip clearance on the efficiency drop and on the tangential component of the absolute velocity, on the basis of experimental investigations of five centrifugal pump impellers with backward-leaning blades, and they showed that the work factor at the optimal flow rate was hardly influenced by the tip clearance variations. Most researchers have disregarded the variation of input power due to a change of tip clearance, partly because it is difficult

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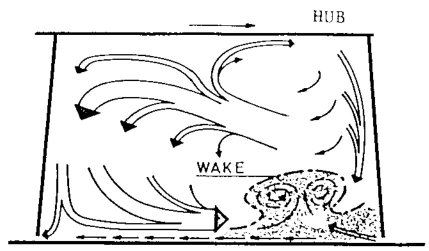

Fig. 1 Secondary flow pattern in the radial part of centrifugal impeller presented by Eckardt (1976)

Table 1 Principal particulars of test impellers

	R-impeller	B'-impeller
Exit diameter(mm)	210.8	510
Exit blade height(mm)	15	17
Exit blade angle(deg)	90	45
Inlet blade angle(deg)	34	28
Number of blades	20	16
Design flow coefficient	0.36	0.27
Specific speed	0.58	0.43

to get a high accuracy in the shaft torque measurement. In the literature (Senoo et al., 1986, 1987, Ishida et al., 1981), the authors have related the change of input power based on the tip clearance to the change of effective bloackage in the impeller channel.

In order to predict impeller efficiency more accurately and to design a vaned diffuser behind an unshrouded impeller properly, it is necessary to be informed of the tip clearance effect on the flow behavior in the impeller. It has already been indicated in the literature (Eckardt, 1976) that the flow bahavior in an unshrouded impeller is quite different from the one in a shrouded impeller. In cases of shrouded impellers where the shroud rotates with the impeller, there is a secondary flow along the shroud from the pressure side of a blade to the suction side of the adjacent blade. On the other hand, in cases of unshrouded impellers, where the shroud is a stationary casing and there is a narrow clearance between the shroud and the blade tip, the secondary flow along the shroud, from the pressure side of a blade to the suction side of the adjacent blade, is intercepted by the leakage flow through the clearance and also by the motion of the stationary shroud relative to the impeller blade. Fig.1 is a representative result presented by Eckardt (1976) measured in the impeller passage.

In the present work, the change of velocity distribution due to a change of tip clearance was measured precisely at the exit of two different types of unshrouded centrifugal impellers, and the variation of secondary flow induced by increasing the tip clearance was studied together with the effect of tip clearance on the input power.

TEST IMPELLERS AND MEASUREMENTS

The two investigated unshrouded centrifugal impellers, one with radial blades and the other with backward-leaning blades are identical to those shown in the literature (Senoo et al., 1986, Ishida et at., 1981). The R-impeller was designed for a turbocharger with twenty radial blades and inducers, and it had an exit diameter of 210.8 mm, the exit blade height of 15 mm, the specific speed of 0.58 and the design flow coefficient of 0.36. The B'-impeller was originally a shrouded conventional centrifugal impeller and it had sixteen backward-leaning blades with the exit angle of 45 deg; the exit diameter was 510 mm, the exit blade height was 17 mm, the specific speed was 0.43 and the design flow coefficient was 0.27. The front shroud was machined off to be an unshrouded impeller. The principal particulars of the two impellers are listed in Table 1.

The experimental setup consists of a suction plenum tank, a test impeller and a vaneless diffuser. Air was axisymmetrically discharged from the blower to atmosphere through a parallel wall vaneless diffuser with the exit radius ratio of about 1.8. The tip clearance of these impellers was changed by inserting shims at the exit position of the vaneless diffuser so that the axial clearance at the exit of impeller was varied stepwise from 0.1 to 3.4 mm for each impeller. In the present experiment, the R-impeller was operated at a constant speed of 4000 ± 4 rpm, and the B'-impeller at 2000 ± 2 rpm. The flow-rate was measured with an entrance flow nozzle which was located between the plenum and the suction pipe. Although the tip clearance was changed, the flow-rate was kept constant with an accuracy of 0.2% by controlling with a conical damper which was located at the entrance of the plenum chamber, and the input shaft torque was measured with an accuracy of 1.0%.

The velocity distribution between the hub and the shroud was measured at the position immediately downstream of the impeller by means of a wedge type two-hole yaw probe and a total pressure probe successively, which were made of 0.7 mm dia. tube, to minimise the disturbance on the flow. The two-dimensional periodic flow was measured at the same position using a specially designed single hot-wire probe. The 5 μm dia. hot-wire supported by two prongs of 0.4 mm dia. was placed parallel to the diffuser walls and they were rotated together with the holder. The surface of the holder was flush to the diffuser wall and only the prongs and the hot-wire were traversed lengthwise perpendicular to the diffuser walls.

EXPERIMENTAL RESULTS AND DISCUSSIONS

Changes in Efficiency and Input Power

Fig.2 shows an example of performance change due to increase of the tip clearance in the R-impeller; ψt and ψs are the total and static pressure coefficients at the exit of impeller respectively and η is the impeller efficiency. The abscissa is the tip clearance ratio at the exit of impeller, and the parameter is the flow coefficient ϕ . The tip clearance decreases the impeller efficiency showing a tendency similar to those reported in other studies; the decrement of efficiency is about 3 points for a change of 10 percent in the tip clearance ratio.

The tangential component of the absolute velocity at the impeller exit was estimated from the measured shaft torque and the flow-rate subtracting the disk friction torque and the bearing torque. Fig.3 shows

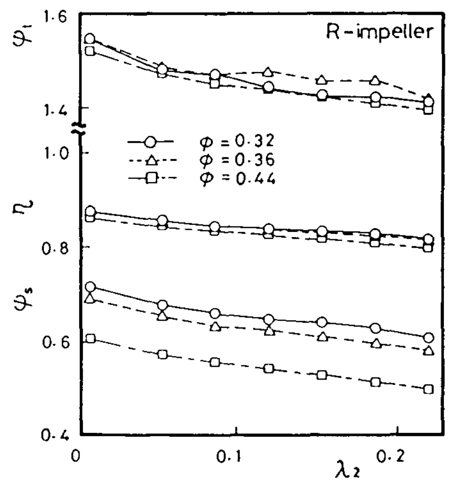

Fig. 2 Effect of tip clearance on the R-impeller performance (Uncertainties of Ψt, η and ψs are $\pm$ 0.019, $\pm$ 0.013 and $\pm$ 0.008 respectively)

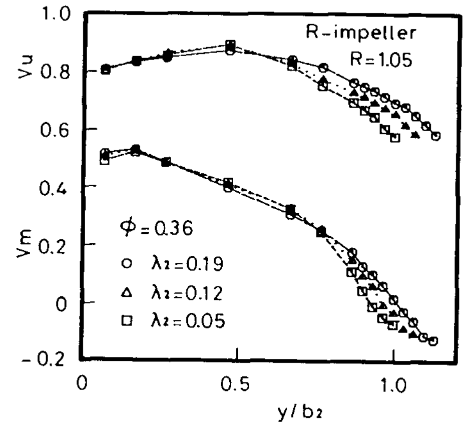

Fig. 4 Effect of tip clearance on the discharge velocity distribution; R-impeller, R=1.05, Φ =0.36 (Uncertainties of Vu and Vm are $\pm$ 0.006)

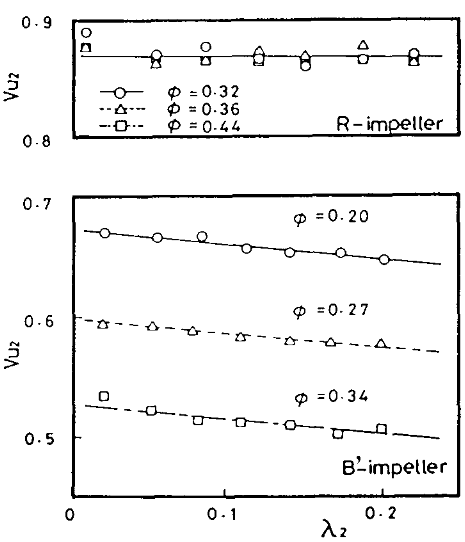

Fig. 3 Effect of tip clearance on input power (Uncertainties of Vu_2 and λ_2 are $\pm$ 0.007)

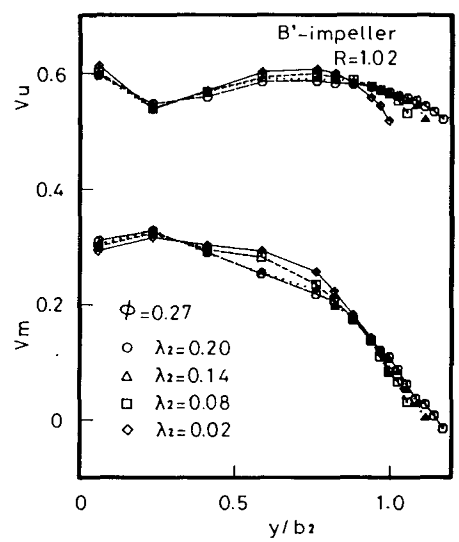

Fig. 5 Effect of tip clearance on the discharge velocity distribution; B'-impeller, R=1.02, Φ =0.27 (Uncertainties of Vu and Vm are $\pm$0.006)

the effect of tip clearance on the work factor, or the tangential component of velocity refered to the impeller tip speed. In the case of the R-impeller, work factors were hardly influenced within the accuracy of the torque measurement by increasing the tip clearance and also by varying the flow-rate. Only in the case of the minimum tip clearance ratio λ_2 = 0.007, the work factor was slightly different from other tip clearance conditions. It seems to be due to a relative misalignment between the impeller disk and the diffuser hub wall because the minimum tip clearance of 0.1 mm was attained by axially moving the impeller away from the hub.

On the other hand, in the case of the B'-impeller, work factors were decreased by increasing the tip clearance in the all flow-rates. It is noticed that the decreasing rates due to the tip clearance were almost equal in the present three flow-rates, that was, $dVu_2/d\lambda_2 \fallingdotseq 0.13$. A similar trend is reported in the literature (Ishida et al., 1981).

Comparison of Velocity Distributions with respect to the Impeller Blade Height

Figs.4 and 5 show the hub-to-shroud distributions of the meridional and tangential components of the absolute velocity measured at the radial positions of R=1.05 for the R-impeller and R=1.02 for the B'-impeller respectively. The abscissa is the axial distance y from the hub along the blade height, which

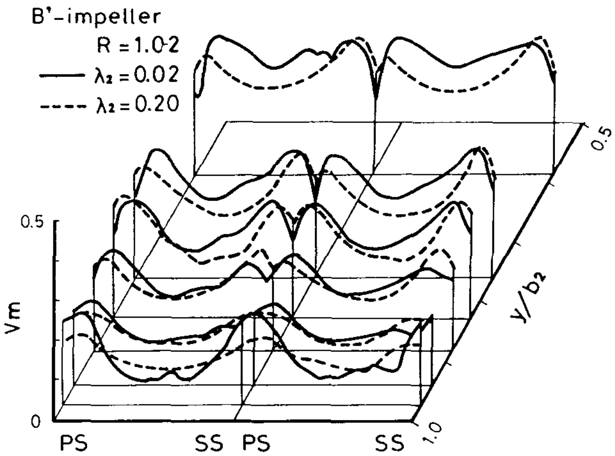

Fig. 6 Effect of tip clearance on the blade-to-blade distribution of the meridional component velocity ; B'-impeller, ϕ =0.27 (Uncertainty of Vm is ± 0.006)

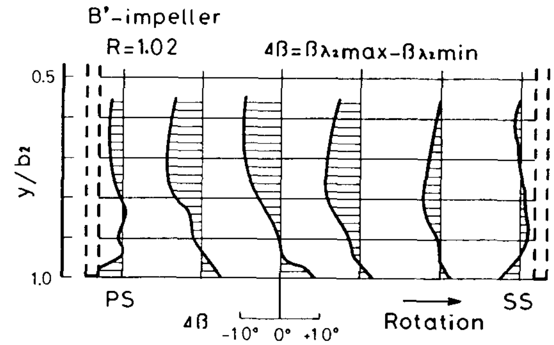

Fig. 7 Secondary flow pattern presented in the form of $\Delta\beta$ -distribution ; B'-impeller, ϕ =0.27 (Uncertainty of $\Delta\beta$ is ± 1.0 deg)

is normalized by the exit blade height b_2 . The change in velocity distributions is partly similar in the both impellers, that is, the tangential and meridional components of the velocity are decreased near the middle of blade height by increasing the tip clearance, and on the contrary, both Vu and Vm are increased in the blade tip region by an increase of the tip clearance. The increase of tip clearance affects not only the flow near the blade tip but also the flow in the whole region between hub and shroud.

The mass averaged tangential component of the absolute velocity at the impeller exit was evaluated from the following equation;

$$Vu_2 = \frac{\int_0^{1+\lambda_2} R^2\ Vm\ Vu\ d(y/b_2)}{\int_0^{1+\lambda_2} R\ Vm\ d(y/b_2)} \tag{1}$$

The value of Vu_2 evaluated by equ.(1) agreed well with the value evaluated from the measured shaft torque shown in Fig.3.

Fig.6 shows the blade-to-blade distributions of the meridional component velocity Vm, which were measured at the radial position R=1.02 for the B'-impeller by means of a hot-wire probe. The data in the diagram were the values averaged over 100 revolutions and, furthermore, averaged by superposing every two pitches of 16 blades; the scale of blade height is magnified about 12 times the scale of blade spacing. The measuring area is about a half of the blade height in the tip side. The solid lines designates the minimum tip clearance condition λ_2 = 0.02 and the broken lines the maximum tip clearance condition λ_2 = 0.20.

By increasing the tip clearance, the meridional velocity component is decreased significantly in the middle of blade height, and what's more, the velocity decrement is larger near the pressure side than the suction side. It seems that the low momentum fluid is carried from the suction side shroud corner towards the pressure side mid-blade height of the adjacent blade. It can be also seen that the blade wake has disappeared near the blade tip. The trailing vortex induced by the leakage through the tip clearance may be responsible for smoothing the circumferential variation of velocity near the tip.

Fig.7 shows the secondary flow pattern, which is represented in the form of $\Delta\beta$ -distributions, in the same measuring area as Fig.6, where $\Delta\beta = \beta(\lambda_{2max}) - \beta(\lambda_{2min})$. The secondary flow which moves from the presssure side of a blade towards the suction side of the adjacent blade along the shroud, and the other secondary flow which crosses from the suction side shroud corner towards the middle of the adjacent pressure surface, are combined with the tip leakage flow. This secondary flow pattern is similar to those presented by Eckardt(1976) and Farge, Johnson & Maksoud (1988). The observed secondary flow in this experiment seems to be inherent in unshrouded centrifugal impellers, and it is strengthened by an increase of the tip leakage flow.

Comparison of Velocity Distributions with respect to the Passage Height

In the preceding section, the secondary flow pattern induced by the tip clearance was shown and studied relative to the impeller blade as is seen in most studies. From the flow pattern with respect to the impeller blade height, the changes in the flow pattern and the input power due to the tip clearance were different from each other for the two different types of impellers. In this section, the velocity distribution between the hub and the shroud was reexamined from the view point with respect to the passage height between hub and shroud wall.

Figs.8 and 9 show the hub-to-shroud distributions of the relative flow angle β and the relavive velocity W, which were measured at the radial position R=1.05 for the R-impeller and R=1.02 for the B'-impeller respectively. The abscissa is the axial distance y' from the shroud wall surface refered to the diffuser inlet width b_3; where $y'=b_3 - y$, and $b_3 = b_2 + c_2$, and the hatched boundaries designate the locations of the blade tip at each tip clearance condition.

It is noticed in the figures that, in the case of the R-impeller, the relative flow angle is hardly changed by varying the tip clearacne and the flow-rate in the whole region between the hub and the shroud. On the other hand, in the case of the B'-impeller, the relative flow angle is significantly and monotonously reduced by increasing the tip clearance in almost all

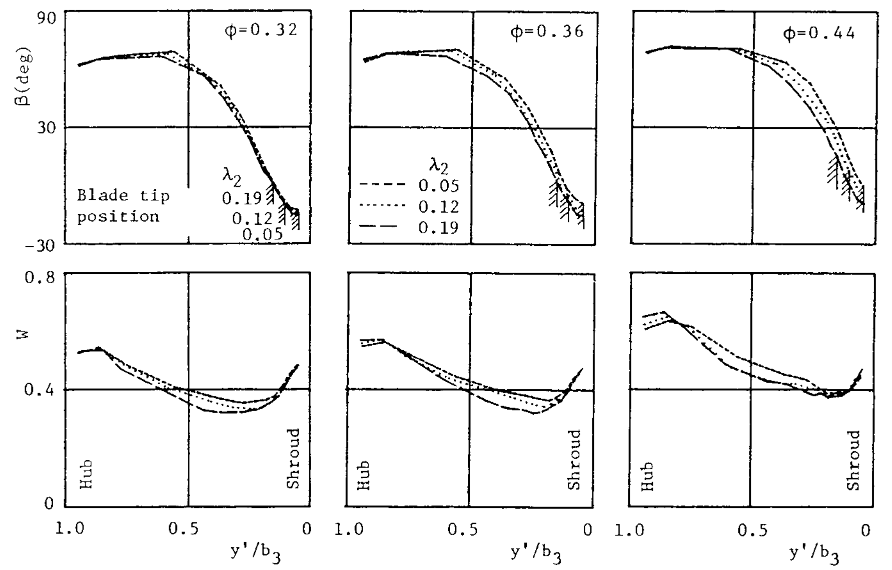

Fig. 8 Hub-to-shroud distributions of the relative flow angle and the relative velocity refered to the shroud wall surface; R-impeller, R=1.05 (Uncertainties of β and W are ± 1.0 deg and ± 0.006)

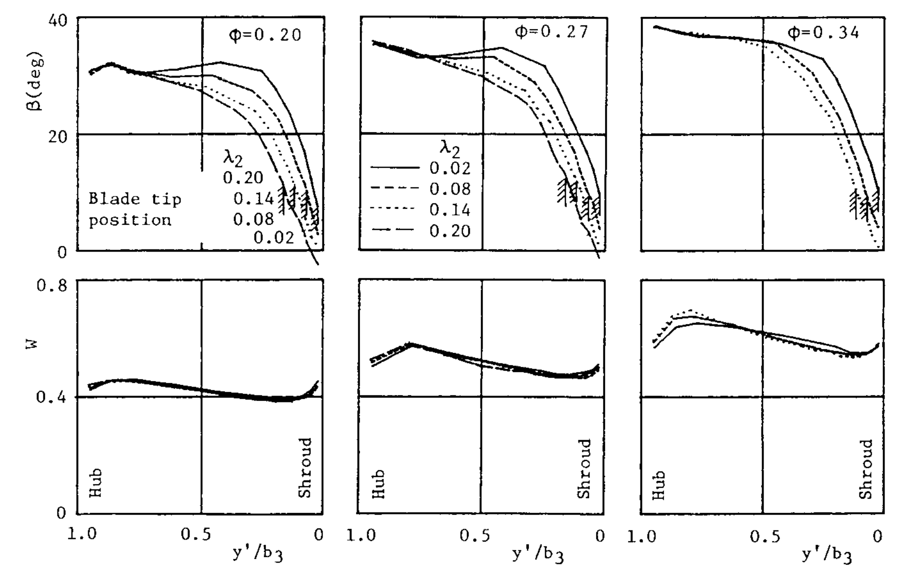

Fig. 9 Hub-to-shroud distributions of the relative flow angle and the relative velocity refered to the shroud wall surface; B'-impeller, R=1.02 (Uncertainties of β and W are ± 1.0 deg and ± 0.006)

of the exit section except near the hub side. The change of relative flow angle in the two impellers seems to be directly related to the change of input power shown in Fig.3. Regarding the relative velocity distribution, there is little change by varying the tip clearance in both impellers. A small change in relative flow angle in the R-impeller seems to be attributed to the inducer with backward-leaning blades.

In the case of the R-impeller, the fact that the relative flow angle was hardly changed inspite of the large tip clearance, means that the fluid in the tip clearance was supported by the fluid in the impeller passage. This was achieved by the more work of the blade especially near the tip, therefore, the relative flow angle was increased at the blade tip as shown in Fig.8 by the hatched boundary. Judging from this fact, the significant reduction of relative flow angle in the case of the B'-impeller is, therefore, not due to an increase of the leakage flow through the tip clearance but due to another causes.

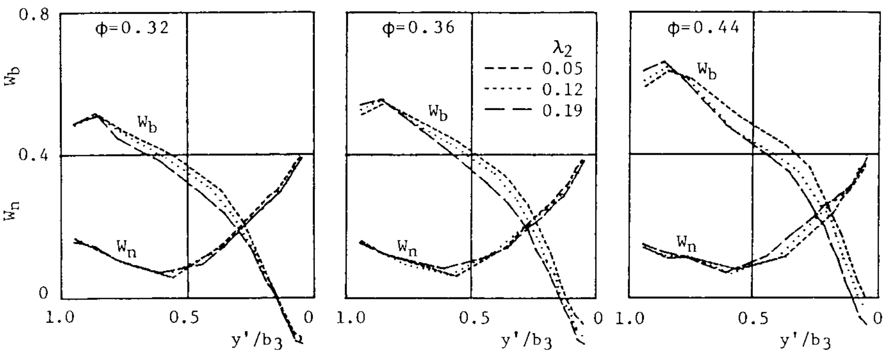

Fig.10 Velocity distributions of the component along blade and the component normal to blade; R-impeller, R=1.0 (Uncertainties of Wb and Wn are ± 0.006)

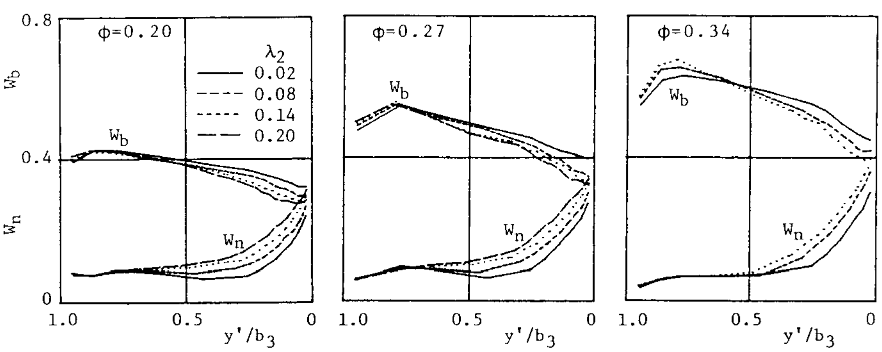

Fig.11 Velocity distributions of the component along blade and the component normal to blade ; B'-impeller, R=1.0(Uncertainties of Wb and Wn are ± 0.006)

Secondary Flow Normal to the Blade

According to the theory presented by the authors (Senoo et al., 1986, 1987), the tip clearance loss mainly consists of two kinds of loss; one is the drag due to the leakage flow through the tip clearance and the other is the pressure loss to support the fluid in the thin annular clearance space between the shroud and the blade tip against the pressure gradient in the meridional plane without blades. Regarding the tip leakage loss, the situation is identical for the radial blade impeller and the backward-leaning blade impeller, but regarding the second loss, it is different between the two types of impellers. The shear force to support the meridional pressure gradient has the component normal to the blade in the case of backward-leaning blade impeller. On the other hand, there is no normal component of the shear force in the radial blade impeller. This is the only difference between the two impellers. That is, the secondary flow in the B'-impeller passage section normal to the blade must be increased by the normal component of the shear force.

The meridian component of the shear force along the surface of revolution was estimated by the following equation in the literature (Senoo et al., 1986);

$$\tau = c\ \{dp/dm - (dp/dm)_c\} \tag{2}$$

where dp/dm is the meridional pressure gradient along the shroud induced by an impeller, (dp/dm)c is the pressure gradient which can be supported by the centrifugal force in the annular space. (dp/dm)c is estimated by assuming that the circumferential velocity of fluid in the annular space is equal to that in the impeller. The component of the shear force normal to the blade is integrated using the equation(3) from the inlet to the exit of impeller in order to correlate with the secondary flow rate normal to the blade at the exit of impeller, which is evaluated by the equation(4).

$$Fn = \frac{\int_{m2}^{m1} 2\pi r\ \tau\ \cos\beta_b\ dm}{2\pi r_2\ b_2\ (\rho/2)\ U_2^2} \tag{3}$$

$$\phi n = (b_3/b_2)\int_0^1 Wn\ d(y'/b_3) \tag{4}$$

Figs.10 and 11 show the velocity distributions of the component along the blade Wb and the component normal to the balde Wn at the exit of impeller for the R-impeller and the B'-impeller respectively. These components of the velocity at the impeller exit were reduced from the measured data by using the relations

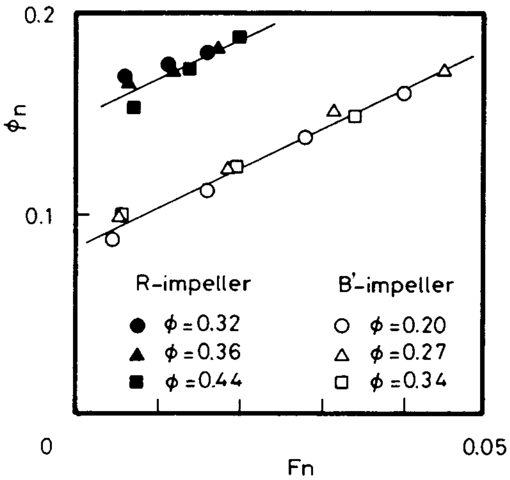

Fig.12 Effect of tip clearance on the secondary flow rate (Uncertainty of ϕn is $\pm$ 0.009)

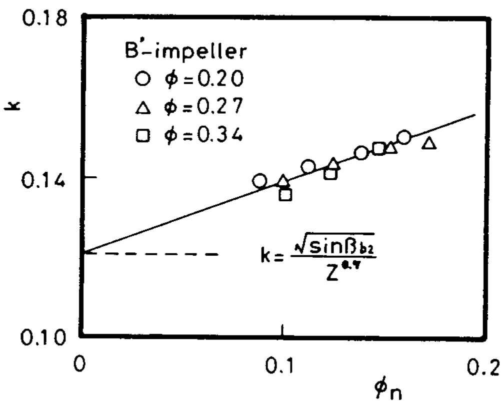

Fig.13 Effect of secondary flow on slip coefficient (Uncertainty of k is $\pm$ 0.007)

of the continuity and the conservation of angular momentum. As is seen in the figures, the normal component of the velocity is significantly increased in the case of the B'-impeller but it is hardly changed in the case of the R-impeller. Furthermore, the normal component under the minimum tip clearance condition shown in the figures is based mainly on the slip near the exit of impeller and partly on the tip leakage flow.

Fig.12 shows the correlation between the secondary flow coefficient ϕn and the normal component of the shear force due to the meridional pressure gradient Fn for the two impellers. The figure shows a good correlation for the respective impellers. It is noticed that the secondary flow is linearly increased by the force Fn , and what's more, the increasing rate is almost identical in both impellers. The difference of ϕn at Fn = 0 between the two impellers is based on the difference of slip near the exit of impeller. The secondary flow is, therefore, significantly induced in the backward-leaning blade impeller because the driving force Fn varies largely by an increase of the tip clearance. It is insignificant in the radial blade impeller because of a small change in the driving force which is originated only in the inducer portion with backward-leaning blade.

Fig.13 shows the correlation between the slip coefficient and the secondary flow coefficient for the B'-impeller. The experimental slip coefficient was calculated by the following equation;

$$k = 1 - \phi \cot\beta b_2 - Vu_2 \qquad (5)$$

where Vu_2 is the experimental value evaluated by equ.(1). A good correlation is observed between them. The k-value at $\phi n = 0$, which is obtained by the extrapolation of the fitted straight line, is extremely close to the value evaluated by the Wiesner's empirical equation; $k = \sqrt{\sin\beta b_2}/Z^{0.7}$ (Wiesner, 1967).

As the effect of tip clearance on the secondary flow rate was nearly equal at the three flow rates of the B'-impeller, the rate of reduction in the input power was almost identical at the three flow-rates shown in Fig.3. Furthermore, in the case of the R-impeller, the variation of slip coefficient due to the tip clearance was so small that the correlation was not clear.

CONCLUSIONS

The effects of tip clearance on the flow pattern at the exit of impeller and on the input power were examined experimentally using two entirely different types of impellers. The principal conclusions are as follows;

(1) The input power was hardly changed in the radial blades impeller and was reduced in the backward-leaning blades impeller by the increase of tip clearance.

(2) From the flow pattern refered to the impeller blades, a secondary flow different from that in shrouded impellers is induced at the impeller exit by the tip clearance and the stationary shroud. The observed pattern at the exit of impeller was quite similar to those observed inside of centrifugal impeller passages. But, the difference of the flow pattern between the two impellers could not be made clear by this view point.

(3) The flow pattern refered to the shroud wall was hardly changed at the exit of the radial blade impeller by increasing the tip clearance. On the other hand, in the case of backward-leaning blade impeller, the relative flow angle was reduced significantly and monotonously by an increase of the tip clearance. The change in input power due to the tip clearance was clearly related to the change of the secondary flow normal to the blade.

(4) The secondary flow due to the tip clearance is most likely caused by the component, which is normal to the blade, of the shear force to support the fluid in the thin annular clearance space between the shroud and the blade tip against the pressure gradient in the meridional plane without blades.

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