International Journal of Fluid Machinery and Systems
Vol. 1, No. 1, October-December 2008

Original Paper

Matching Diffuser Vane with Return Vane Installed in Multistage Centrifugal Pump

**Daisuke Kawashima[1], Toshiaki Kanemoto[2],
Kazuyuki Sakoda[3] Akihiro Wada[3] and Takashi Hara[3]**

[1]Graduate, Graduate School of Engineering, Kyushu Institute of Technology
[2]Faculty of Engineering, Kyushu Institute of Technology
Sensui 1-1, Tobata, Kitakyushu, 804-8550, Japan
[3]Research & Development Department, Torishima Pump Manufacturing Co., Ltd.
Miyata 1-1-8, Takatsuki, 569-8660, Japan

Abstract

The effects of the diffuser vane on the performances of the multistage centrifugal pump were investigated experimentally, taking account of the interactions among the diffuser vane, the return vane, and the next stage impeller. It is very important to match well the diffuser vane with the return vane, for improving the hydraulic efficiency of the pump. The efficiency may be more improved by making the cross-sectional area of the channel from the diffuser vane outlet to the return vane inlet larger, as much as possible.

Keywords: Centrifugal pump, diffuser vane, return vane, multistage pump, performance, efficiency, hydraulic loss

1. Introduction

Multistage centrifugal pumps always play an important role in the construction of the future infrastructures for the sustainable developments. The improvement of the pump performances contributes to cope with the warming global environment, and the pumps with the fruitfully advanced technologies are under obligation to fulfil such contributions. To accomplish the obligations, the channel profile from the outlet of the impeller to the inlet of the next stage impeller must be optimized more and more, so as to bring the impeller with higher efficiency into full performance.

This serial research intends to optimize the channel profile equipped with the diffuser vane and the return vane, taking the interaction to the next stage impeller into consideration. In the previous paper, the desirable return vane and the impeller profiles were proposed [1]. Continuously, this paper discusses the effects of the diffuser vane profile on the multistage pump performances, taking account of matching the diffuser vane with the return vane and the next stage impeller. The rotor-stator interactions have been investigated experimentally and numerically to suppress the acoustic noises and the mechanical vibrations [2]-[5], and the flow conditions in the pump composed of the impeller and the stationary components have also been predicted numerically [6]-[8]. The pump profiles will be optimized by the advanced flow simulations in the near future. This paper may mediate between the present and the future technologies of the multistage pump design.

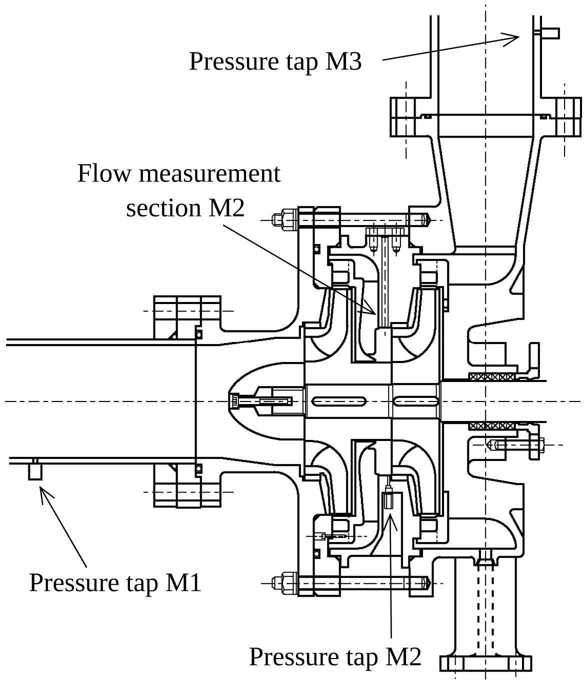


Fig. 1 Model multistage centrifugal pump

2. Preparation of Diffuser Vane

2.1 Model Pump

Figure 1 shows the model centrifugal pump supplied to the previous

Received June 19 2008; revised August 25 2008; accepted for publication September 12 2008: Review conducted by Prof. Jun Matsui.
Corresponding author: Toshiaki Kanemoto, Professor, turbo@tobata.isc.kyutech.ac.jp

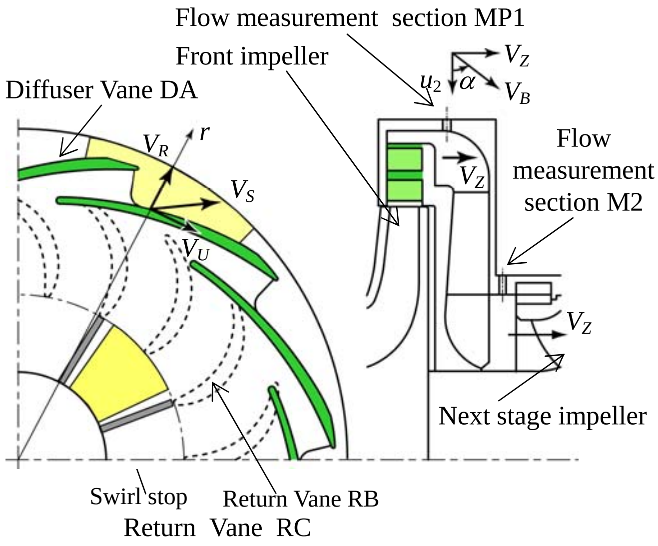


Fig. 2 Diffuser vane DA

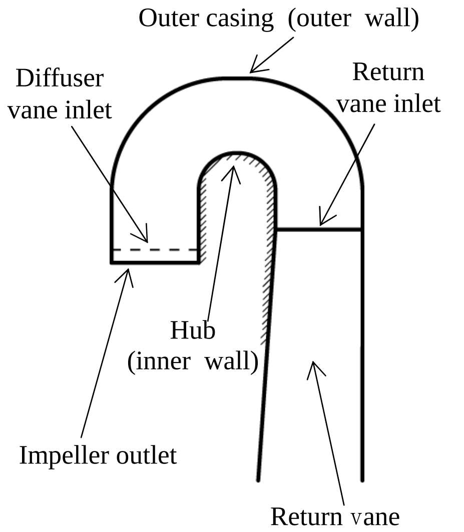


Fig. 3 Meridian profile of the U-turn channel

paper [1], which is composed of the suction pipe, the first stage impeller, the diffuser vane prepared below, the U-turn channel, Return vane RC, the annular channel, the second stage impeller, the diffuser vane, the discharge chamber and the discharge pipe. Both impellers are the same profile with 7 blades and are called Impeller B as was proposed in the previous paper [1], where the suction and the delivery diameters are 170 mm and 300 mm, while the specific speed per the stage is 170 (m, m^3/min, min^{-1}) at the design point with ϕ =0.105 [=Q/Au_2, Q: the discharge, A: the impeller outlet area, u_2: the peripheral velocity at the impeller outlet]. Return Vane RC is composed of Return Vane RB and the swirl stop as shown in Fig. 2, and gives the maximum pump efficiency [1]. The cross-sectional area of the discharge chamber just downstream of the second stage diffuser vane was temporarily kept constant in the tangential direction. The length of the annular channel downstream of the return vane, namely upstream of the second stage impeller, is longer than that of the prototype, to measure experimentally the flow conditions.

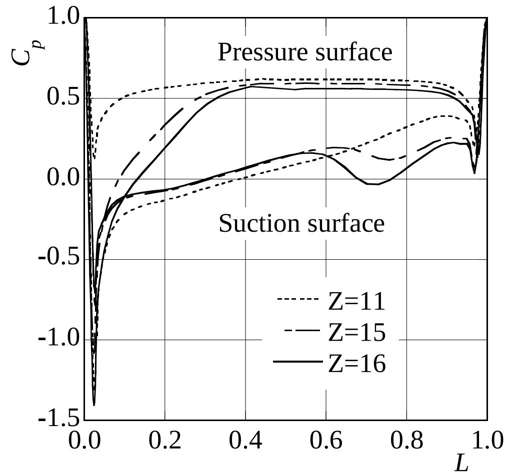


Fig. 4 Pressure on the diffuser vane surfaces

2.2 Diffuser Vanes

One of the model diffuser vanes, which was installed in the previous paper [1] and is called Diffuser Vane DA in this paper, is shown in Fig. 2, where the number of vanes is 11 and the trailing edge is in close to the outer casing wall. The flow is discharged not only from the narrow gap between the trailing edge and the outer casing wall but also from the opening on the end wall, and runs into the U-turn channel with sudden enlargement. The U-turn channel is formed with the concave type outer wall and the step type inner wall.

The cascade type two-dimensional diffuser vane was also prepared, after the U-turn channel was modified as follows. Several channel profiles with smoothly curved surfaces in the meridian view were proposed, in keeping the maximum radius of the outer casing wall and the channel widths at the diffuser and the return vanes, whose dimensions are the same as those of the channel with Diffuser Vane DA and Return Vane RC. The optimum channel profile supplied to the model pump is shown in Fig. 3, in which the pressure on the inner and the outer walls never changes suddenly in the flow direction from the outlet of the first stage impeller to the inlet of the return vane.

The diffuser vane installed in the above channel was designed using numerical simulation by means of the singularity method in the potential flow field [9], so as to give the flow angle 15 degrees at the return vane inlet, which is measured from the tangential direction and is the vane inlet angle of Return Vane RC. The diffuser vane profile was also designed so as to suppress the separation of the boundary layer flow as much as possible [10]. The domain of the numerical simulation was set from the outlet section of the first stage impeller to the inlet section of the return vane, and the impeller outlet flow was predicted by the commercial code SCRYU/Tetra, at the normal discharge coefficient ϕ =0.105 (design point). The shock-free flow angle at the diffuser vane inlet, that is 8.5 degrees measured from the tangential direction, is slightly larger than that of Diffuser Vane DA with 8.2 degrees, and the trailing edge was set at the maximum radius of the hub cone, namely the inner wall, to make the vane chord longer.

The optimum profile of the diffuser vane, whose pressure distributions along the vane surfaces not only never drop suddenly near the leading edge but also recover moderately towards the trailing edge, was supplied to the pump model, where the separation points are on the rounded surfaces of the trailing edge. To get the optimum cascade profile formed with the diffuser vane designed just above, the effect of the vane number Z on the pressure distributions along the vane surfaces was predicted by the above singularity method and the pressures on the middle stream surface of the channel are shown in Fig.4, where L is the

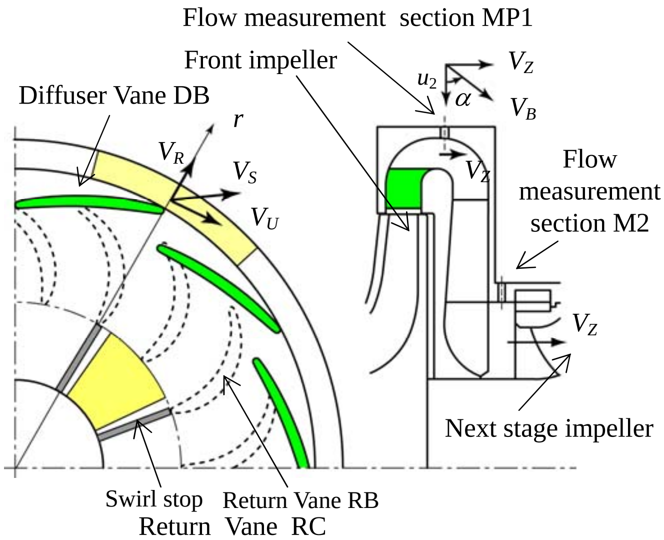


Fig. 5 Diffuser Vane DB

dimensionless distance measured from the leading edge and C_P is the pressure coefficient based on the flow at the impeller outlet. Equipping with 11 vanes, the pressure drop on the pressure surface in close to the leading edge is obviously small, and the downstream pressure on the suction surface is scarcely affected by the neighboring vane because of the small solidity. On the contrary, the downstream pressure on the suction surface conspicuously drops while equipping with 15, and 16 vanes. With the increase of the vane number, the flow through the cascade becomes faster due to the blockage effect and increases the friction loss, in general.

Based on the above numerical simulations and discussions, the cascade composed of 11 diffuser vanes is supplied to the first stage of the model pump and is called Diffuser Vane DB. Figure 5 shows the first stage profile equipping with Diffuser Vane DB, where the second stage equips conveniently with Diffuser Vane DA as it is.

3. Pump Performances

The effect of the diffuser vane profile on the performances of the first stage at the rotational speed n=750 min^{-1} is shown in Fig. 6, where the stage was divided by the same procedure described in the previous paper [1], ψ is the head coefficient [=$H/(u_2^2/2g)$, H: the total head, g: the gravitational acceleration], ν is the shaft power coefficient [=$P/(\rho A u_2^3/2)$, P: the shaft power without the mechanical loss power such as the bearings], η^*_h is the hydraulic efficiency divided by the maximum hydraulic efficiency of the two-stage model pump equipped with Diffuser Vane DA and Return Vane RA [=$\eta_h/\eta_{hA13\max}$, $\eta_h=\rho gQH/P$]. Besides, the subscripts 12 and 13 stand for the values of the first stage and the two-stage model pumps. The head of the model with Diffuser Vane DB is lower than that with Diffuser Vane DA at the higher discharge than ϕ =0.025, but the operation point giving the maximum efficiency goes to the higher discharge. The comparison between both heads suggests that the hydraulic losses of the model with Diffuser Vane DB can not be disregarded at the higher discharge but has a tendency to decrease relatively at the extremely low and high discharges as compared with that with Diffuser Vane DA. Besides, the discharge giving the maximum efficiency is smaller than ϕ =0.105 at the design point.

As for the head coefficient ψ_{23} of the second stage, ψ_{23} of the model with Diffuser Vane DB installed in the first stage is slightly higher than that with Diffuser Vane DA (see Fig. 7), nevertheless the second stage is assembled with the same components as the model with Diffuser Vane DA. That is, Diffuser Vane DB makes the head of the next stage

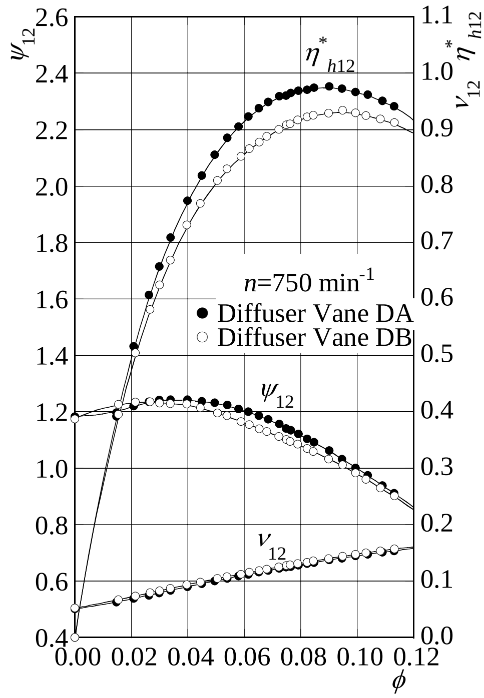


Fig.6 Performances of the first stage (M1-M2)

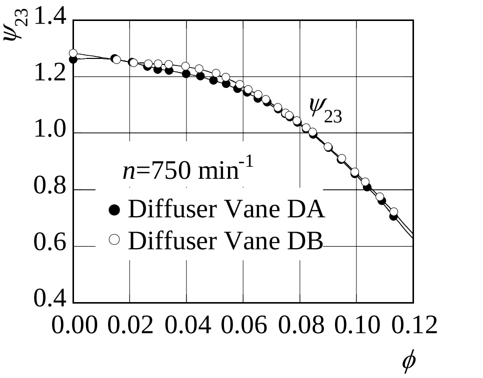


Fig. 7 Pump head of the second stage (M2-M3)

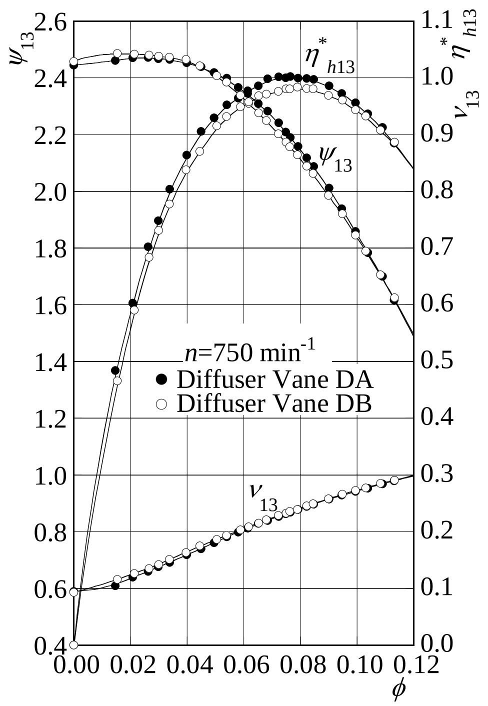
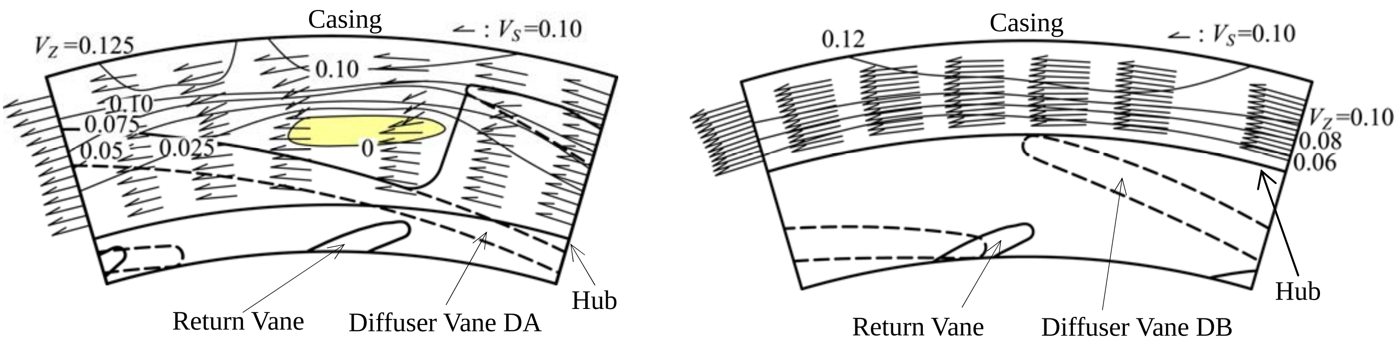


Fig. 8 Model pump performances (M1-M3)



(a) Diffuser Vane DA (b) Diffuser Vane DB

Fig. 9 Flow conditions at Section MP1 (ϕ=0.105)

higher though the vane makes the head of the first stage lower. Resultantly, the effect of the diffuser vane profile installed in the first stage on the head of the two-stage model pump, ψ_{13}, becomes smaller than that on the head of the first stage, and the effect on the efficiency is also small, as shown in Fig. 8. These reasons will be discussed latter in detail.

4. Flow Discharged from Diffuser Vane

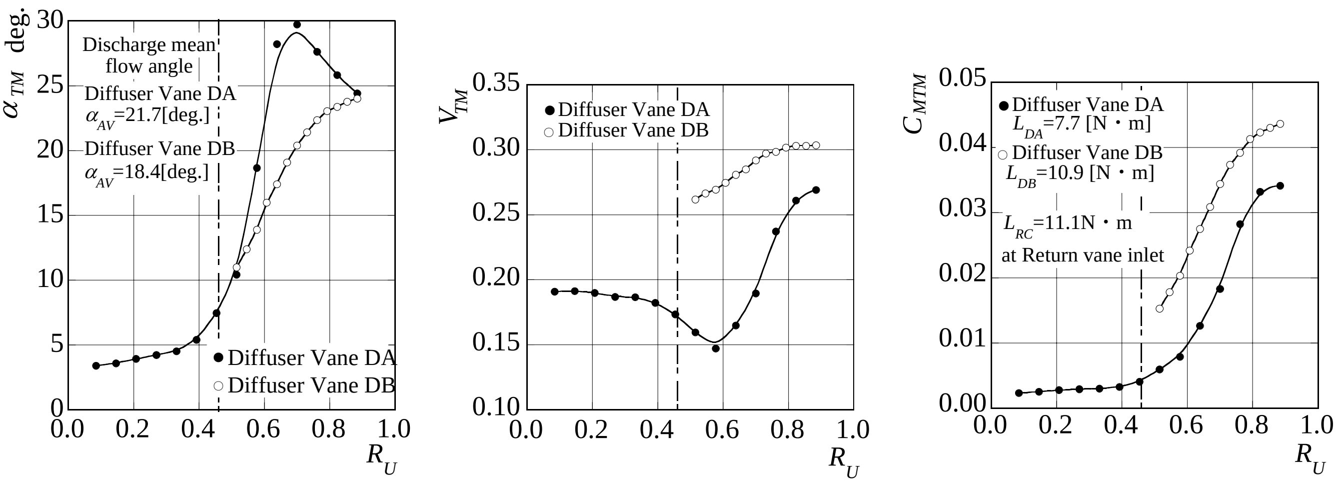
4.1 Flow Conditions

Figure 9 shows the flow conditions at Section MP1 (on colored section in Figs. 2 and 5) while keeping the discharge constant at ϕ=0.105 (design point), where the velocities were measured by the appropriated 5-hole Pitot tube in the steady state conditions [1], the axial velocity component $V_Z = (v_z/u_2)$ is shown by the iso-velocity lines, the velocity component parallel to the cross section $V_S = (v_s/u_2)$ is shown by the velocity vectors, and the diffuser and the return vanes are drawn with the dashed and the full lines. The flow is scarcely discharged from the opening of the end wall in Diffuser Vane DA (V_Z=0), but most of the flow is discharged from the narrow gap between the casing wall and the trailing edge, as shown in Fig. 9(a). That is, the flow discharged from Diffuser Vane DA may enlarge suddenly in the downstream U-turn channel, and the flow at the smaller radius on Section MP is occupied with the swirling velocity component V_S. On the contrary, the flow discharged from Diffuser Vane DB is in axis-symmetric and uniform conditions on the cross section, but the flow velocity is very high owing to the narrow width, as shown in Fig 9(b).

4.2 Evaluation of Hydraulic Losses

The hydraulic losses were discussed using the flow conditions in Fig. 9, because the head of the model with Diffuser Vane DB is lower than that with Diffuser Vane DA although the flow discharged from the Diffuser Vane DB is more uniform. As for the channel from the outlet of the diffuser vane to the inlet of the return vane, it may be necessary to pay attention to the friction losses for the channel equipped with Diffuser Vane DB and to the mixing losses for the channel equipped with Diffuser Vane DA. The friction loss is in proportion to $v^{7/4}$ (v: the flow velocity) in Blasius formula and the mixing loss is in proportion to v^2 in the sudden enlargement. That is, the more the velocity increases, the more the mixing loss becomes large, in comparison with the friction loss. Then, the model with Diffuser Vane DA has the tendency to increase the hydraulic loss, namely the mixing loss, with the increase of the discharge and that can be reconfirmed indeed by ψ_{12} in Fig. 6.

The mixing loss head of Diffuser Vane DA is h_W = 0.01 m at ϕ =0.105, where $h_W = (1-A_1/A_2)^2 v_z^2/2g$, v_z is the axial velocity component, A_2 is the cross-sectional area at MP1, and A_1 is the area after subtracting the area with the velocity less than v_z=0 from A_2. On the contrary, the friction loss head h_F = 0.15 m from Section MP1 to the return vane inlet of the channel with Diffuser Vane DB is three times larger than h_F = 0.05 m of the channel with Diffuser Vane DA, where ϕ =0.105, $h_F = \lambda(l/D_e)v_m^2/(2g)$, l =



(a) Flow angles (b) Absolute velocities (c) Angular momentum

Fig. 10 Mean flow conditions at Section MP1

$l_m/\sin[(\alpha_1+\alpha_2)/2]$, $D_e = (d_{e1}+d_{e2})/2$, $v_m^2 = (v_1^2+v_2^2)/2$, λ is the friction coefficient of the circular pipe given by Blasius formula, l_m is the stream distance along the middle meridian line from Section MP1 to the return vane inlet, α is the mean flow angle measured from the tangential direction with the velocity u_2 (subscripts 1, 2: the values at HP1 and the return vane inlet), and $v = v_z/\sin\alpha$, d_e is the hydraulic diameter (= $4A/S$; A, S: the cross-sectional area and the length of the wetted perimeter perpendicular to the flow direction). The larger friction loss in the channel with Diffuser Vane DB is caused by the markedly faster velocity reconfirmed in Fig. 9(b).

Above hydraulic losses suggest that it is necessary, for the channel with Diffuser Vane DB, to decrease the friction loss. That is to say, it can be expected to improve the efficiency more and more by making the channel width downstream of the diffuser vane enlarged so as to decrease the flow velocity.

4.3 Matching with Return Vane

The flows of Fig. 9 were averaged in the tangential direction at the same radius and are shown in Fig. 10, where α_{TM} is the flow angle measured from the tangential direction, V_{TM} is the absolute velocity [= $(V_Z^2+V_S^2)^{1/2}$], C_{MTM} is the angular momentum coefficient (=$V_{ZTM}V_{UTM}$ r/r_0, r/r_0: the radial ratio divided by the impeller outlet radius), R_U is the dimensionless width divided by the channel with Diffuser Vane DA at Section MP1, and the dotted-and-dashed line give the inner wall position of the channel with Diffuser Vane DB at Section MP1. The flows in the channel with Diffuser Vane DA are remarkably distorted in the radial direction [see Fig. 10(a)(b)] as discussed in Fig. 9, and the flow angel α_{AV} averaged in the cross section of the channel with Diffuser Vane DB is smaller than that with Diffuser Vane DA [see Fig. 10(a)]. It is not only the smaller flow angle but also the narrow width that make the velocity in the channel with Diffuser Vane DB faster [see Fig. 10(b)], and that causes the increase of the friction loss.

Judging from the flow angle α_{AV}, Diffuser Vane DB is suitable for Return Vane RC with 15 degrees of the vane inlet angle. Figure 10(c) shows the distribution of the angular momentum coefficient C_{MTM} , the mean angular momentum on Section MP1 (L_{DA}: the value of the channel with Diffuser Vane DA, L_{DB}: the value of the channel with Diffuser Vane DB), and the angular momentum expected at the return vane inlet L_{RC} which is estimated in the shock-free and uniform flow conditions at the inlet cross section. Assuming that the angular momentum is kept constant in the stream direction, the return vane installed in the channel with Diffuser Vane DA has the negative incidence angle because the angular momentum L_{DA} is less than L_{RC}. On the contrary, L_{DB} is almost the same as L_{RC}, that is, Diffuser Vane DB is suited to Return Vane RC.

5. Effects on Next Stage Impeller

It was confirmed previously in Chapter 3 that the diffuser vane profile installed in the first stage affects not only the first stage but also the second stage performances. The separation points of the boundary layer flow along the return vane surface were predicted [10] and are shown in Fig. 11, where the uniform flow conditions at the return vane inlet were estimated by applying the angular momentums L_{DA}, L_{DB} given in Fig. 10(c) to the vane inlet (the flow angles in the channel with Diffuser Vanes DA and DB are 21.1 degrees and 15.3 degrees measured in the tangential direction, against 15 degrees of the return vane inlet angle).

In the channel with Diffuser Vane DA, the boundary layer on the pressure surface separates near the leading edge and the layer on the suction surface separates at the latter half of the surface, due to the negative incidence angle. Resultantly, the flow through the return vane cascade is put under the control of the flow along the suction

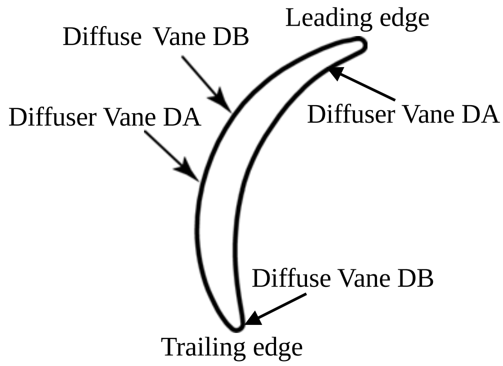
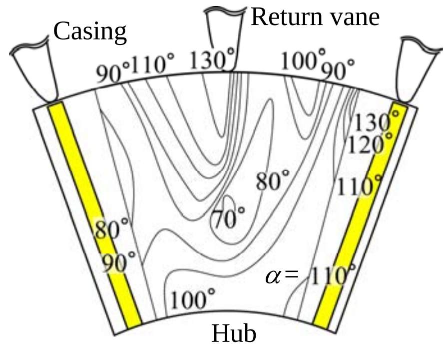
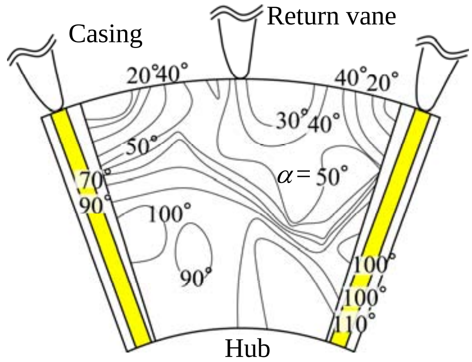


Fig. 11 Separation points on the return vane surface



(a) Diffuser Vane DA



(b) Diffuser Vane DB

Fig. 12 Flow angles at Section M2 (ϕ=0.105)

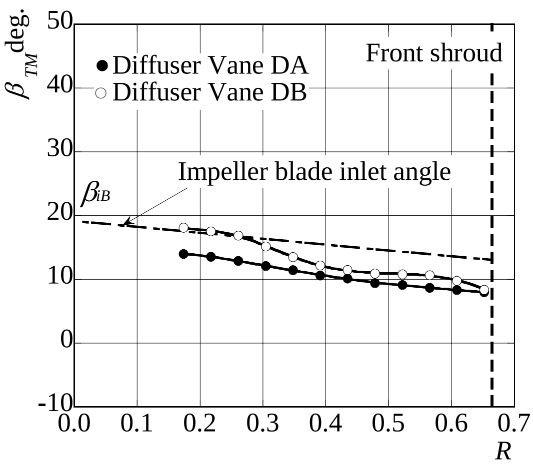


Fig. 13 Relative flow angles at Section M2 (ϕ=0.105)

surface, and that can be confirmed with the downstream flow conditions. Figure 12 shows the flow angle α measured from the tangential direction at Section M2 (see Figs. 2 and 5) while ϕ=0.105 (design point), where the blades and the front shroud of the second stage impeller were eliminated, and the angle of the flow running in the rotational direction is less than α =90 degrees. The flow angle near the casing wall is partially larger than 101.3 degrees of the return vane outlet angle [see Fig. 12(a)], due to not only the above flow along the suction surface but also the effect of the secondary flow running from the pressure to the suction surfaces on the end wall and the meridian curvature just at the downstream of the return vane.

On the contrary, the flow near the casing wall is under-turning while equipping with Diffuser Vane DB as shown in Fig. 12(b), because the flow thorough the return vane cascade is put under the control of the flow along the pressure surface as understood with the separation points in Fig. 11. Such flow conditions, however, scarcely affect the work of the second stage impeller because the under-turning region is almost at the front shroud position (occupying about 1/3 of the width, see Figs. 2 and 5).

The flows of Fig. 12 were averaged in the tangential direction at the same radius and the relative flow angles β_{TM} to the next stage impeller inlet are shown in Fig. 13, where β_{TM} is measured from the tangential direction and R is the dimensionless radius divided by the channel width at Section M2. The relative flow angle in the channel with Diffuser Vane DB is better than that with Diffuser Vane DA. That may contribute to make the head of the next stage higher, and can make the pump efficiency relatively higher with the increase of the stage.

6. Concluding Remarks

The effects of the diffuser vane profile on the multistage pump performances were investigated experimentally, taking account of matching the diffuser vane with the return vane and the next stage impeller. This investigation suggested that it is very important to match the diffuser vane with the return vane for improving the pump efficiency as the multistage. It is also desired to enlarge the channel width downstream of the diffuser vane so as to make the flow velocity slower, for more improving of the efficiency.

Acknowledgments

The authors wish to thank to The 25th (2005) Harada Memorial Foundation in Japan, which co-sponsors some parts of this research.

References

[1] Miyano, M., Kanemoto, T., Kawashima, D., Wada, A., Hara, T. and Sakoda, K., 2008, “Return Vane Installed in Multistage Centrifugal Pump”, International Journal of Fluid Machinery and Systems, Vol. 1, No. 1, pp. 57-63.
[2] Tsugawa, T, 2007, “The Relation between Solidity and the Other Shape Factors for Complete Optimum Meridian Profile of Impeller with Guide Vane”, Proceedings of the 5th Joint ASME/JSME Fluid Engineering Conference, CD-ROM:FEDSM2007-37344.
[3] Sano, T., Kobayashi, K., Miyagawa, K., Yamamoto, Y. and Mitsui N., 2007, “Development of High Performance Multistage Centrifugal Pump”, Proceedings of the 9th Asian International Conference on Fluid Machinery, CD-ROM:AICFM9-282.
[4] Sugimura, K., Obayashi S. and Jeong, S., 2007, “Multi-objective Design Exploration of a Centrifugal Impeller Accompanied with a Vaned Diffuse”, Proceedings of the 5th Joint ASME/JSME Fluid Engineering Conference, CD-ROM:FEDSM2007-37502.
[5] Pavesi., G., Cavazzini, G., Dupont, P., Coudert, S., Caignaert, G., Bois, G. and Ardizzon, G., 2006, “Analysis of Rotor-Stator Interactions Effects within the Vaned Diffuser of a Radial Flow Pump”, Proceedings of the 23rd IAHR Symposium, CD-ROM:F206.pdf.
[6] Yamanishi, N., Kato, C., Okita, K. and Motomura T., 2007, “Large Eddy Simulation of Unsteady Flow in the LE-7A Liquid Hydrogen Pump”, Proceedings of the 43th AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit, AIAA 2007-5515.
[7] Huang, S., 2006, “A Numerical Study on the Impeller-diffuser Interactions os a Multi-stage Centrifugal Pump”, Proceedings of the 23rd IAHR Symposium, CD-ROM:F172.pdf.
[8] Berten, S., Farhat, M., Dupont P. and Avellan F., 2007, “Rotor-Stotor Interaction Induced Pressure Fluctuations: CFD and Hydroacoustic Simulations in the Stationary Components on a Multistage Centrifugal Pump”, Proceedings of the 5th Joint ASME/JSME Fluid Engineering Conference, CD-ROM:FEDSM2007-37549.
[9] Toyokura, T., Kanemoto, T. and Hatta, M., 1986, “Study on Circular Cascades for Return Channel of Centrifugal Turbomachinery (1st Report: Inverse Method and Cascade Design)”, Bulletin of the JSME, Vol. 29, No. 255, pp.2885-2892.
[10] Kanemoto, T., 1986, “Study on Circular Cascades for Return Channels of Centrifugal Turbomachinery (3rd Report, Discussion on Separated Flow)”, Bulletin of the JSME, Vol. 29, No. 255, pp. 2901-2908.