A Study of Aerodynamic Performance of Diffusers for Centrifugal Compressors*

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The performance of vaneless diffusers and the flow distribution discharged from the impellers of various specific speed centrifugal compressors were studied. The experimental results of eighteen different stages showed, that (1) the impeller exit flow distortion in axial direction increased with an increase in specific speed, and (2) the pressure recovery of vaneless diffusers for high specific speed compressors was extremely low compared with the value expected by an ideal two dimensional analysis. New type diffusers with half guide vanes on the shroud side wall are proposed in order to improve the pressure recovery of the diffuser under the distorted inlet flow condition. The results show that the pressure recovery of diffuser under distorted inlet flow is considerably improved by half guide vanes, and the best height of the guide vanes is a little less than one half the diffuser width.

Key Words : Fluid Machine, Centrifugal Compressor, Vaneless Diffuser, Guide Vane, Pressure Recovery Coefficient, Pressure Loss Coefficient

1. Introduction

Vaneless diffusers are widely used in industrial centrifugal compressors which are required in a wide flow range. Many experimental and theoretical studies have been made on radial vaneless diffusers (1), (2), but, it seems that a clear account is not given for an effect of a highly distorted inlet flow on the diffuser performance. The three-dimensionally distorted flow discharged from centrifugal impellers is expected to have a significant effect on the diffuser performance (3).

Recently, Senoo et al. (4), (5) studied a distorted flow in vaneless diffusers precisely, and made clear the distinction of equalization mechanism between in circumferential direction and in axial direction. They showed that the flow distortion in axial direction did not easily disappear unlike in the circumferential flow distortion. Furthermore, the development of axial distortion with respect to radius in the diffuser was theoretically analyzed and good agreement with experiments was demonstrated.

The present work is planned to show the influence of distorted inlet velocity distribution in axial direction on the aerodynamic performance of vaneless diffusers. Eighteen different specific speed centrifugal compressors were tested, and the experimental results were compared

with the results of calculation by the theoretical method (6).

The results show that, (1) the impeller exit flow distortion increases with an increase of the specific speed of the stages, and (2) the pressure recovery of a vaneless diffuser for a high specific speed stage is extremely low compared with the value expected by an ideal two-dimensional analysis. Then, a new type diffuser with small guide vanes set on the shroud side wall is proposed in order to improve the pressure recovery of the diffuser for high specific speed centrifugal compressors. The present paper discusses the effect of the guide vanes on equalization of the inlet flow distortion, and also the influence of vane height on the pressure recovery of the diffuser.

2. Nomenclature

b : diffuser width
 B_f : velocity distortion coefficient, $\overline{C_{m\theta}}/\overline{C_m}$
 C : absolute velocity
 C_f : wall friction coefficient
 $\overline{C_m}$: area averaged mean radial velocity, $\int \rho C_m dA / \int \rho dA$
 $\overline{C_{m\theta}}$: mass averaged mean radial velocity, $\int \rho C_m^2 dA / \int \rho C_m dA$
 C_p : pressure recovery coefficient, $(P_{se} - P_{si}) / (P_{ti} - P_{si})$
 g : gravitational acceleration parameter
 h : height of guide vanes
 Had : adiabatic head
 M : Mach number
 K_f : constant of wall friction coefficient, Eq. (1)
 N : rpm
 n_s : specific speed, $NQ^{1/2}/Had^{3/4}$
 P : pressure

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Q : flow rate
 r : radius
 u : tip speed of impeller
 w : relative velocity
 y : length from shroud side wall
 α : flow angle as measured, from circumferential direction
 β : vane angle as measured, from circumferential direction
 ζ : pressure loss coefficient, $(P_{ti} - P_{te}) / (P_{ti} - P_{si})$
 η : efficiency
 η_{ad} : adiabatic efficiency
 φ : flow coefficient, $Q / (\pi r_2^2 u_2)$
 ψ : pressure coefficient, $Had / (u_2^2 / g)$

Subscripts

2 : impeller exit
 m : meridional component
 e : diffuser exit
 u : circumferential component
 i : diffuser inlet
 ref : reference condition
 des : design condition
 $-$: mean value

3. Test Rig and Instrumentation

Experiments were made with a centrifugal compressor test rig, consisting of a single stage compressor, a torque meter, a d-c motor with built-in gears, an air cooler, and flow piping. The closed loop of air piping had variable suction pressure and temperature to control the inlet conditions of the compressor. Figure 1 shows a cross-sectional view of a single stage test rig which simulates the flow

Fig. 1 Sectional view of the model compressor rig

Fig. 2 Velocity distributions at selected impeller exits

passage of a multi stage centrifugal compressor. A vaneless diffuser, a return channel, and a collector casing were set downstream of an impeller.

Eighteen different stages were tested in this experiment; the impellers had the same diameter (300mm), the vaneless diffusers with parallel walls had the same radius ratio ($r/r_2 = 1.6$), and the specific speed of them was varied from 250 to 500 (rpm, m, m³/min). Tip circumferential speed of the impellers was varied from 270 m/s to 300 m/s, the machine Reynold's numbers were between 2×10^5 and 5.6×10^5 , and the Mach number of the diffuser inlet flow was about 0.55.

The total and static pressures at the diffuser inlet ($r_1/r_2 = 1.067$), and at the diffuser exit ($r_e/r_2 = 1.533$), were measured at three circumferential positions with cobra type yaw probes consisting of 1 mm dia. stainless steel tubes. The pressure recovery factors and the pressure loss coefficients of the vaneless diffusers were rated on the mass averaged values of the time mean measured pressure. The performance of the compressor stage was evaluated on the basis of the total-to-total rating at the inlet and the exit of the stage as shown in Fig. 1. The flow rate was controlled with a valve, and was measured with an orifice type flow meter. The main dimensions of the eighteen test diffusers, such as diffuser width ratios b/r_1 and inlet flow angles at design flow rate, are listed in Table 1 together with the pressure recovery coefficients C_p and the pressure loss coefficients.

4. Aerodynamic Performance of Vaneless Diffusers with Various Values of Specific Speed

4.1 Velocity distribution at an impeller exit

The flow discharged from a centrifugal compressor impeller is extremely distorted. Then the highly distorted flow is expected to have a significant effect on the performance of the diffuser. A few

Table 1 Diffuser geometries, inlet flow distortion and performances

Diffuser	b/r_1	α_1 (deg)	C_p	ζ	B_1
1	0.066	26.6	0.43	0.18	1.03
2	0.074	24.0	0.42	0.17	1.02
3	0.089	30.7	0.43	0.14	1.04
4	0.094	28.2	0.40	0.18	1.02
5	0.094	30.4	0.34	0.21	1.05
6	0.103	31.4	0.44	0.15	1.07
7	0.112	30.7	0.28	0.25	1.10
8	0.126	27.6	0.30	0.21	1.04
9	0.126	27.3	0.32	0.22	1.02
10	0.126	33.6	0.29	0.23	1.04
11	0.126	31.1	0.31	0.23	1.09
12	0.150	35.5	0.24	0.29	1.14
13	0.150	25.9	0.32	0.27	1.10
14	0.156	31.4	0.25	0.29	1.09
15	0.156	34.7	0.25	0.27	1.14
16	0.156	35.4	0.23	0.14	1.09
17	0.187	34.3	0.04	0.21	1.23
18	0.187	30.8	0.02	0.22	1.22

$r_1/r_2 = 1.067$
 $r_e/r_1 = 1.44$ for all cases

examples of the measured velocity distributions at the impeller exit ($r/r_2 = 1.067$) are presented in Fig.2. Figures 2(a), (b) and (c) show the impeller exit velocity distributions of the No.4, No.7, and No.18 compressors in Table 1 respectively, and the specific speed of each stage was 350, 400 or 500 (rpm, m, m^3/min). The measured velocity distributions shown in Fig.2 indicate that the flow at the impeller exit, or at the diffuser inlet, is highly distorted in axial direction with an increase of the specific speed of impellers. The extremely distorted exit flows similar to the results shown in Fig.2 were sometimes observed with highly loaded centrifugal compressors (3).

We express the distortion of the inlet velocity distribution by the following velocity distortion coefficient,

$$B_f = \frac{\int \rho C_m^2 dA / \int \rho C_m dA}{\int \rho C_m dA / \int \rho dA}$$

where ρ is the density, C_m is the radial component of velocity and A is the area at the diffuser inlet. The coefficient B_f is the ratio of the mass averaged radial component to the mean radial velocity.

These values are included in Table 1 and they are related to the diffuser width ratio b/r_i as presented in Fig.3: b/r_i increases with an increase of the specific speed. Figure 3 shows that the velocity distortion coefficient B_f increases with an increase of the diffuser width ratio b/r_i , and the axial distribution of velocities is quite distorted in a high specific speed compressor.

4.2 Performance of vaneless diffusers

The pressure recovery factor C_p and the loss coefficient ζ were used to indicate the performance of the test diffusers. Pressure was measured at the inlet ($r_i/r_2 = 1.067$) and the exit ($r_e/r_2 = 1.533$) of the diffuser with use of cobra type yaw probes. Then the radius ratio of the test diffusers was held constant, $r_e/r_i = 1.44$. The pressure recovery coefficients C_p and the loss coefficients ζ of the eighteen test diffusers measured at the design flow rate are listed in Table 1.

Fig.3 Variation of velocity distortion coefficient with respect to diffuser width ratio

When the inlet velocity distribution is uniform, it is generally known that the pressure recovery of vaneless diffusers, which have the same radius ratio r_e/r_i and the same inlet flow angle α_i , improves with an increase of the diffuser width b/r_i (2). In Fig.4 the full lines indicate the pressure recovery coefficient and the loss coefficient, which are predicted based on the theory shown in ref. (5) assuming a uniform inlet condition. The parameter K_f in Fig.4 is a constant of the wall friction coefficient expressed as

$$C_f = \lambda K_f (U\delta/\nu)^{-1/2} \quad (1)$$

where K_f is usually 0.045, λ is a constant for compressibility, U is free stream velocity, δ is boundary layer thickness, and ν is kinematic viscosity. According to ref. (5), $K_f = 0.051$ was recommended for vaneless diffusers which were tested with a blower impeller.

In Fig.4 the experimental data in Table 1 are included. They show pressure recovery coefficients which are lower than the predicted lines especially in the wide diffusers. The measured loss coefficients

(a) Pressure recovery coefficient

(b) Pressure loss coefficient

Fig.4 Effects of diffuser width on pressure recovery and pressure loss coefficients

are also large compared with the calculated value assuming a uniform inlet condition. Many data of pressure recovery coefficients and loss coefficients in Table 1 were verified at the inlet flow angle not equal to 30 deg, which was used for the prediction. Since not only the diffuser width ratio but also the inlet flow angle was varied in the eighteen different stages, the experimented data in Fig.4 are not considered to show the influence of the diffuser width only. But according to the example calculations the pressure recovery coefficient was not sensitive to the inlet flow angle from 24 deg to 36 deg in the case of a high specific speed compressor. Only $\pm 5\%$ difference was estimated as compared with the pressure recovery coefficient of the 30 deg inlet flow angle.

Then the difference between the experimental results and the calculated results in Fig.4 could be due to the other effect such as the influence of the inlet flow distortion increasing with the diffuser width ratio shown in Fig.3.

5. Effect of the Inlet Flow Distortion in Axial Direction on Diffuser Performance

As mentioned in the last section, compressors with a large specific speed have a pressure recovery in the vaneless diffuser considerably lower than expected. In the following, through a flow analysis(5),(7) on an axially distorted flow, the influence of the distorted inlet velocity distribution on the diffuser performance will be discussed.

The calculation was performed under the following two different conditions: (1) various inlet boundary layer thicknesses were assumed, and the velocity distri-

Fig.5 Velocity triangles at the impeller exit

Fig.6 Velocity distributions assumed for calculation

bution was uniform across the diffuser depth, (2) various velocity gradients across the diffuser depth were assumed, but the flow was assumed to be uniform in circumferential direction in both cases. The calculated result of a large inlet boundary layer thickness showed a minor effect on the diffuser performance as reported formerly(7). Then the influence of the axial distribution of inlet velocities was studied. Figure 5 shows the velocity triangles across the diffuser width of the No.18 compressor in Table 1. The velocity triangles across the diffuser width in Fig.5 show that the circumferential velocity component was fairly uniform across the width and the inlet distortion was mainly induced by the distorted distribution of the radial velocity components. The velocity distributions marked with the symbols ① through ⑤ in Fig.6 were modeled after the velocity triangles in Fig.5, and only the distribution of the radial components of velocity was varied. Then the calculation was

(a) Pressure recovery coefficient

(b) Pressure loss coefficient

Fig.7 Effects of velocity distortion coefficient on pressure recovery and pressure loss coefficients

performed at various values of velocity gradient across the width keeping the mean flow angle equal to 30 deg as shown in Fig.6, and the inlet Mach number equal to 0.55. The diffuser width was 0.187 times the inlet radius.

The calculated pressure recovery coefficient C_p and the total pressure loss coefficient ζ are presented in Fig.7 against B_f and the experimental data are marked by circles. It should be noted that many of the experiments were not performed at the inlet flow angle of 30 deg and the diffuser width was not equal to 0.187 ri as indicated in Table 1. Furthermore, the real velocity distributions were not always similar to the ones used in the flow analysis even though the velocity distortion coefficient was identical. Therefore, these experimental data can only approximately be compared with the prediction. But both the theoretical

prediction and the experimental data clearly demonstrate that the pressure recovery was considerably deteriorated by a distortion of the inlet velocity distribution.

6. Improvement of Diffuser Performance by Means of Half Guide Vanes on the Shroud Side Wall

In high specific speed compressors, the pressure recovery coefficient of the vaneless diffuser is considerably low as shown in Fig.4, because the flow is distorted along the diffuser width due to a large curvature of the shroud.

A new type diffuser with half guide vanes set on the shroud side wall, was tested in order to improve the pressure

Fig.8 Half guide vanes set on the shroud side wall of a diffuser

Fig.9 Stage performance values of a compressor with different diffusers

Fig.10 Effect of guide vane height on compressor efficiency

Fig.11 Effect of guide vane height on pressure recovery coefficient

recovery of the diffuser for high specific speed compressors. The effect of similar guide vanes for the conical diffuser for the mixed flow blowers has been reported⁽⁸⁾. The flow in the conical diffuser is substantially unstable because of the curvature of the inner wall, and it was shown that the guide vanes were effective to prevent the skewed boundary layer distortion on the convex inner wall. The present work is intended to investigate the effect of the half guide vanes on the equalization of the highly distorted inlet flow discharged from high specific speed centrifugal impellers, and on the improvement of the pressure recovery of the radial diffuser and the stage efficiency.

6.1 Half guide vanes

The guide vanes of a new concept were applied to the No.15 compressor of

Table 1. The specific speed was 500 (rpm, m, m³/min) and the impeller had 20 blades, which were inclined 60 deg to the circumference at the exit. There were 20 guide vanes which had a vane angle of 35 deg all the way through and a constant thickness of 2.3 mm, and they were set at $1.02 r_2$ to $1.6 r_2$ as shown in Fig.8, and the vane height was changed in several steps: $h/b = 0.05, 0.1, 0.2, 0.4, 0.5, 0.6$, and 1.0 .

6.2 Diffuser performance and the height of guide vanes

The characteristic curves of the compressor stage are presented in Fig.9. A considerable improvement was achieved when the vane height was about one half of the diffuser depth. On the other hand, the stage performance of the diffuser with full vanes ($h/b = 1.0$) was not found satisfactory especially at high flow rate.

Fig.12 Velocity distributions at the inlet and exit of diffusers with and without half guide vanes

The relationship between the compressor efficiency and the vane height is shown in Fig.10. In Fig.10 the efficiencies are shown in relative values when the efficiency of the vaneless diffuser stage is equal to unity. Figure 10 shows that the stage efficiency near the design flow rate reaches a maximum value when the guide vanes have a height between $0.4b$ and $0.5b$.

Figure 11 shows that the best pressure recovery coefficient was achieved near a height of $0.4b$. The pressure recovery coefficients of the diffusers shown in Fig.11 were measured at the center of a pitch of the adjacent guide vanes with use of three holes yaw probes. The pressure recovery coefficients of the diffuser with half guide vanes increased with an increase of the height of vanes until the height of vanes reaches about one half the diffuser width. When the height of guide vanes was larger than one half the diffuser width, the pressure recovery coefficient decreased with an increase of the height of vanes, and showed various curves with different flow rates.

6.3 Velocity distribution at the diffuser inlet and exit

The half guide vanes at the diffuser inlet are effective to improve the pressure recovery in an extremely distorted flow, as shown in Fig.11. In the following, the relationship between the velocity distribution across the diffuser width and the diffuser pressure recovery is discussed. The velocity distributions were measured across the diffuser width at the middle of the vane pitch.

Figures 12(a) to 12(h) show the velocity and the flow angle distributions in axial direction at the diffuser inlet

($r_1/r_2 = 1.067$) and the exit ($r_e/r_2 = 1.533$) at 0.92 times the design flow rate, where the stage efficiency showed a maximum value. In Fig.12 the solid lines indicate the inlet velocity distributions and the dotted lines indicate the exit velocity distributions. It was clear that by increasing the height of the guide vanes the flow distortion could be reduced, and that when the height of guide vanes reaches one half the diffuser width the velocity distribution became considerably uniform as shown in Figs. 12(e) and 12(f). These figures clearly show that the improvement of the pressure recovery shown in Fig.11 is due to the effect of the flow equalization by the half guide vanes.

In higher guide vanes the velocity distributions at the diffuser exit, shown in Figs. 12(g) and 12(h), indicate different patterns from the ones of the diffuser with smaller height guide vanes; the velocity near the hub side wall increased on the contrary with the inlet distribution, and the velocity distribution was not equalized sufficiently. Therefore, the pressure recovery of the diffuser with higher guide vanes decreased as shown in Fig. 11.

The velocity distributions at the diffuser exit in several types of diffusers are presented in Fig.13 at various flow coefficients. It was also shown that the velocity distribution changes greatly when the height of the guide vanes is beyond about one half the diffuser width. The experimental results prove that the maximum pressure recovery is related closely to the uniformity of the axial velocity distribution attained by the half guide vanes.

Fig.13 Velocity distributions at the diffuser exit for different flow rates

7. Conclusions

The performance of the vaneless diffusers and the flow distribution discharged from the impellers of various specific speed centrifugal compressors were studied. And an experiment to improve the diffuser performance in an extremely distorted inlet flow in axial direction was carried out, applying the half guide vanes on the shroud side wall of the diffuser.

Conclusions are as follows;

(1) The performance of the diffuser for high specific speed centrifugal compressors is considerably reduced compared with the value expected by a uniform two dimensional analysis.

(2) A highly distorted flow discharged from high specific speed centrifugal impellers has a significant effect on the diffuser performance. The result of a flow analysis shows also the same effect.

(3) The pressure recovery of diffuser under distorted inlet flow is considerably improved by half guide vanes. The best height of the guide vanes is a little less than one half the diffuser width.

(4) The half guide vanes on the shroud side wall have the effect to make the axially distorted inlet flow uniform.

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