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Flows and Pressure Recovery in a Rotating Diffuser* (Effects of Inlet Velocity Distributions)

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The flow patterns and pressure recoveries in a rotating two-dimensional diffuser were studied for the case where uneven flows with a shear type velocity distribution were introduced into it.

If the entering flow has higher velocities near the lower pressure side of the diffuser, the pressure recovery is improved by rotation of the diffuser and the recovery coefficient becomes higher than that observed in the stationary state of the diffuser. The improvement will be caused by a uniformization of flow velocities in the rotating duct due to a vigorous secondary flow development. The reverse results prevail in the case where the flow enters the diffuser with higher velocities near the pressure side of the diffuser, the secondary flow being less vigorous in this case.

Key Words: Fluid Mechanics, Rotating Channel, Shear Flow, Coriolis Force, Pressure Recovery, Separation, Secondary Flow

1. Introduction

In order to obtain detailed data about the flow patterns in radial impellers of centrifugal pumps and compressors, many studies have been made by using impellers of real fluid machines and also a single channel having comparatively simple configurations(1),(2). Rothe and Johnston(3) measured the pressure recovery in rotating straight diffusers having a rectangular cross section of various area ratios and found that the flow separation caused by the channel rotation deteriorated the pressure recovery in the diffusers. The present authors have also studied the pressure recovery and flow patterns in rotating conical diffusers with various opening angles(4),(5) and demonstrated that the pressure recovery coefficient decreases with an increase of the spin ratio of the channel and that the flow separation begins on the wall of the lower pressure. Moore(6),(7) studied both experimentally and theoretically the flow patterns inside a rotating diffuser with a simple form. In all of the above studies, the flows entering the rotating channel were in a uniform state or a fully developed state and the results deduced from them can not always be applied to the flow in the fluid machines because the uniformity of the inlet velocity profile

to the impeller passages of centrifugal pumps and compressors is destroyed for the flow rate lower than the normal. This problem is very important in practice but remains unclarified because of its complexity in nature.

To solve this problem, in this study, changes of the velocity profile and the pressure recovery in a two-dimensional straight diffuser are clarified experimentally when uneven flows with shear flow type velocity distributions are introduced into a diffuser rotating about an axis perpendicular to the channel.

2. Nomenclature

A : Area of diffuser section
 AR : Area ratio of inlet and exit sections = A_2/A_1
 C_p : Mean value of C_{pw} at section S(9)
 C_{pi} : Coefficient of pressure recovery for inviscid fluid (Eq. (18))
 C_{pim} : Value of C_{pi} at section onsetting separation
 C_{pw} : Coefficient of pressure recovery assumed from wall static pressure (Eq. (10))
 E : Flux of total energy (Eq. (22))
 E_k : Flux of kinetic energy (Eq. (20))
 E_p : Flux of pressure energy (Eq. (21))
 L : Axial distance measured from diffuser inlet
 N : Spin ratio = $\omega W_1 / V_{m1}$
 p : Reduced static pressure
 p_w : Wall pressure
 Re : Reynolds number = $V_{m1} W_1 / \nu$
 Ri : Richardson number (Eq. (25))
 u : Velocity vector
 V_{m1} : Mean flow velocity at section S(1)
 V_x, V_y, V_z : Velocity components in x , y and z directions, respectively
 x, y, z : Orthogonal coordinates in diffuser (Fig. 1(b))

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- W : Width of diffuser section
 η_p : Coefficient of pressure energy recovery (Eq. (23))
 ζ : Coefficient of total energy loss (Eq. (24))
 ω : Angular velocity of rotating diffuser
 ω : Vectorial representation of angular velocity
 λ : Dimensionless value of velocity gradient at section S(0) (Eq. (5))
 λ_0 : Prescribed value of λ at the inlet section
 λ_m : Mean value of λ calculated from measured velocity distribution at section S(0)
 ξ_z : Vorticity component in z direction
 Subscripts
 1 : Inlet section (Section S(0))
 2 : Exit section (Section S(9))

3. Equipment and Method of Experiment

The outlines of the experimental equipment and method are almost the same

as used in the previous study(4). Figure 1(a) shows the detailed dimensions and measuring sections of the diffuser. The channel is composed of a two-dimensional straight diffuser, the opening angle of which is 12° and it has a rectangular cross section. The area ratio of the exit section to the inlet one, AR , is equal to 2.6. Wall static pressures were measured at the sections S(-2), to S(10), and the velocities were also measured by traversing a three hole yaw meter in y direction along the lines ①-①, ②-②, and ⑤-⑤ and also in z direction along the lines a-a and b-b shown in Fig. 1(b) at the sections of S(0), S(4), S(7), and S(9). To get the prescribed shear flow pattern at the diffuser inlet, grids were installed at the section S(-3). The grids were made of thin wires, the intervals of which were decided by the method of Owen et al. (8) and the velocity gradients of $\lambda_0 = 0.35$ and 0.65 were used at the inlet section. Experiments were performed at the Reynolds number of $Re = 4.6 \times 10^4$ and the spin ratios of $N = 0, 0.06, 0.08$ and 0.12, respectively.

4 Equation Used to Express Experimental Results

4.1 Equations of motion and dimensionless value of velocity gradient

The equations of continuity and momentum for an incompressible and steady flow are written as,

$$\nabla \cdot \mathbf{u} = 0 \quad \dots\dots\dots (1)$$

$$(2\omega + \frac{\xi}{2}) \times \mathbf{u} = -\nabla(p + \mathbf{u} \cdot \mathbf{u}/2) + \nu \nabla^2 \mathbf{u} \quad \dots\dots\dots (2)$$

where $\mathbf{u}$ and ω denote the flow vector and angular vector of the rotating pipe, and ξ is the vorticity vector equal to $\nabla \times \mathbf{u}$. From Eq. (2) the equation of vorticity is obtained as follows

$$(\mathbf{u} \cdot \nabla) \xi - [(2\omega + \frac{\xi}{2}) \cdot \nabla] \mathbf{u} = \nu \nabla^2 \xi \quad \dots\dots\dots (3)$$

In a two dimensional flow, all derivatives with respect to z vanish and Eq. (3) can be written in the following form

$$V_x(\partial \xi_z / \partial x) + V_y(\partial \xi_z / \partial y) = \nu(\partial^2 \xi_z / \partial x^2 + \partial^2 \xi_z / \partial y^2) \quad \dots\dots\dots (4)$$

This equation shows that ξ_z , the vorticity component in z direction, does not depend upon the angular velocity ω , and is the same as that in the stationary state of

(a) Dimensions and Measuring sections

(b) Traversing directions of probe

Fig. 1 Rotating diffuser

Fig. 2 Rotating coordinates and shear type of flow in two-dimensional diffuser

the channel.

The velocity gradient at the inlet section (dV_x/dy) as shown in Fig. 2 can be made dimensionless as follows,

$$\lambda = (dV_x/dy)(W_1/V_{m1}) \quad (5)$$

The sign of λ is positive if the velocity increases toward the higher pressure side as shown in Fig. 2 and negative if it decreases. In an inviscid flow each term in the right hand side of Eq. (2) vanishes and ξ_z reduces to -2ω . Provided that only x component of velocities exists in the inlet section, the gradient of the main velocity is given by

$$\partial V_x / \partial y = -2\omega \quad (6)$$

Substitution of Eq. (6) into Eq. (5) provides the dimensionless velocity gradient λ_p as

$$\lambda_p = -2\omega W_1 / V_{m1} \quad (7)$$

Now, the spin ratio of the diffuser flow, N , is defined by

$$N = \omega W_1 / V_{m1} \quad (8)$$

With use of Eqs. (7) and (8), λ_p is expressed by

$$\lambda_p = -2N \quad (9)$$

The velocity gradient in a real flow can not be linear across the inlet section due to the effects of the wall friction and secondary flow, and as the value of λ in the inlet section, a mean value λ_m calculated by the least squares method is used.

4.2 Coefficient of pressure recovery

If the flow in the rotating diffuser is two-dimensional, the term representing the effect of system rotation disappears in the equation of ξ_z as is seen in Eq. (4) and both of the flow pattern and the pressure recovery in the rotating diffuser remain unchanged from those in the stationary state.

Using the measured values of wall pressures and correcting an increase in pressure due to the centrifugal force caused by the system rotation, the pressure recovery coefficient C_{pw} in the diffuser can be expressed by

$$C_{pw} = 2(p - p_1) / \rho V_{m1}^2 \quad (10)$$

where p_1 and p are the wall pressures at the section of S(0) and at a section in the diffuser, respectively. In an inviscid flow Bernoulli's equation can be applied, and the relationship between the two sections ① and ② along the stream tube (m-m, n-n) shown in Fig. 2 becomes

$$p_{1t} + \rho(V_{x1t}^2 + V_{y1t}^2)/2 = p_t + \rho(V_{xt}^2 + V_{yt}^2)/2 \quad (11)$$

where the subscripts t and l denote the values in a section in the stream tube and at the inlet section ①, respectively. Provided that the secondary velocities are negligible with respect to the main flow velocity, Eq. (11) can be simplified as

$$p_t - p_{1t} = \rho(V_{x1t}^2 - V_{xt}^2)/2 \quad (12)$$

Substitution of Eq. (12) into Eq. (10)

gives a coefficient of pressure recovery in an inviscid flow, C_{pi} , as follows

$$C_{pi} = 2(p_t - p_{1t}) / \rho V_{m1}^2 = (V_{x1t}/V_{m1})^2 \{1 - (V_{xt}/V_{x1t})^2\} \quad (13)$$

Putting the areas of the two sections ① and ② of the stream tube as dA_{1t} and dA_t , respectively, the continuity equation becomes

$$V_{x1t} dA_{1t} = V_{xt} dA_t \quad (14)$$

From Eqs. (13) and (14),

$$dA_t = \{V_{x1t} / (V_{m1} \sqrt{(V_{x1t}/V_{m1})^2 - C_{pi}})\} dA_{1t} \quad (15)$$

Integration of Eq. (15) gives

$$\begin{aligned} \frac{A}{A_1} &= \frac{1}{A_1} \int_{A_1} \frac{(V_{x1t}/V_{m1})}{\sqrt{(V_{x1t}/V_{m1})^2 - C_{pi}}} dA_{1t} \\ &= \frac{1}{W_1} \int_{-W_1/2}^{W_1/2} \frac{(V_{x1t}/V_{m1})}{\sqrt{(V_{x1t}/V_{m1})^2 - C_{pi}}} dy \quad (16) \end{aligned}$$

When the inlet flow has a constant velocity gradient λ as shown in Fig. 2, the dimensionless velocity V_{x1t}/V_{m1} can be expressed by

$$V_{x1t}/V_{m1} = 1 - \lambda(y/W_1) \quad (17)$$

From Eqs. (16) and (17), the coefficient C_{pi} in an inviscid flow is given as

$$C_{pi} = 1 - 1/\beta^2 - (\lambda^2/4)(\beta^2 - 1) \quad (18)$$

where β denotes the area ratio of a certain section to the diffuser inlet and is equal to A/A_1 . A legitimate solution of Eq. (16) can be obtained only when $((V_{x1t}/V_{m1})^2 - C_{pi})$ is positive. V_{x1t}/V_{m1} has its minimum equal to $(1 - \lambda/2)$ on the wall when a shear flow expressed by Eq. (17) is introduced into the diffuser, and the maximum value of C_{pi} is given by $(1 - \lambda/2)^2$. Thus, the area ratio of the section where C_{pi} reaches the maximum can be expressed by use of Eqs. (18) as

$$A/A_1 = \sqrt{2/\lambda} \quad (19)$$

In the section at which the cross sectional area exceeds this limit, the flow will separate from the wall and a reverse flow appears in the downstream region.

4.3 Coefficients of pressure energy recovery and energy loss

The fluxes of the kinetic, pressure and the total energy, respectively, are given by the following equations,

$$E_k = \int_A \{\rho(V_x^2 + V_y^2 + V_z^2)/2\} dA \quad (20)$$

$$E_p = \int_A \{(p - p_0)V_x/E_0\} dA \quad (21)$$

$$E = E_k + E_p \quad (22)$$

where $E_0 (= \rho V_{m1}^3 A_1/2)$ denotes the kinetic energy flux which would be obtained if the flow entered uniformly with a velocity corresponding to the mean flow velocity. The coefficients of the pressure energy recovery η_p and the total energy loss are given, respectively, by

$$\eta_p = (E_{p2} - E_{p1}) / (E_{k1} - E_{k2}) \quad (23)$$

$$\zeta = (E_1 - E_2) / E \quad (24)$$

4.4 Richardson number

Turbulent intensities in the flow in a rotating channel may be suppressed or intensified according to the flow conditions by the effects of the Coriolis force (2),(10),(11) and gradient Richardson number Ri defined by the following equation can be used as the criterion for flows in the rotating system.

$$Ri = 2\omega(\partial V_x/\partial y - 2\omega)/(\partial V_x/\partial y)^2 \quad (25)$$

In an inviscid flow the above expression reduces to

$$Ri = 2N(\lambda - \lambda_p)/\lambda^2 \quad (26)$$

When $Ri > 0 (\lambda > \lambda_p)$, the flow is stabilized and when $Ri < 0 (\lambda < \lambda_p)$, the flow is destabilized.

5. Experimental Results and Discussions

5.1 Velocity distributions

Figures 3(a), (b) and (c) show the velocity profiles measured along the line ③-③ (shown in Fig. 1(b)) at the inlet section S(0), when a uniform flow ($\lambda_0 = 0$, without a grid) and shear type flows ($\lambda_0 = -0.65$ and 0.65 by use of grids), respectively, are introduced into the diffuser. When the inlet condition is $\lambda_0 = 0$, the distributions of the main flow V_x are almost uniform for all spin ratios, N , and when the inlet condition is

$\lambda_0 = -0.65$ (Fig. 3(b)), the distribution of main flow velocities shows a shear type flow across the channel except in the wall region. The velocity curves shift further upward as the spin ratio increases. For reference, the velocity profile with $\lambda_0 = -0.65$ corresponding to an inviscid flow is also shown, which lies almost parallel to the measured profiles for $N = 0$ and 0.06 . When λ_0 is positive and equal to 0.65 , and when the diffuser is stationary, the gradient of the measured velocity profile has almost the same value as λ_0 at the inlet section, but the profile becomes concave in the central portion of the section when the diffuser is rotated. The gradient becomes larger and the curves themselves shift downward as the spin ratio N increases. The regions in which the Richardson numbers Ri defined by Eq.(25) are positive and negative are shown in Figs.3(b) and (c). The negative value of Ri prevails in the greater part of the section when $\lambda_0 = -0.65$, and the positive value prevails in the section when $\lambda_0 = 0.65$.

Figures 4(a) and (b) show the equi-velocity curves and the secondary flow vectors in the section S(0) for the inlet conditions of $\lambda_0 = -0.65$ and 0.65 , respectively, when $N = 0.06$. When $\lambda_0 = -0.65$, V_x/V_{m1} has the largest value near the lower pressure side of the channel on the center line ③-③. This will be due to a shift of the higher velocity region in

Fig. 3 Velocity profiles on ③-③ of section S(0)

Fig. 4 Equi-velocity curves and distribution of secondary flow for $N = 0.06$

course of flow along the diffuser. Higher velocity region observed near the lower pressure side of the channel downstream of the grid will be moved toward the channel center by the secondary flow. From the picture of the secondary flow component in z direction, generation of a secondary flow (parallel to y axis) toward the higher pressure side in the channel center can be expected. The secondary flow increases with N , and a noticeable upward shift of the velocity profile can be seen in Fig. 3(b). When the inlet condition is $\lambda_0 = 0.65$ (Fig. 4(b)), the larger velocity region is stretched toward both sides by secondary flow V_z . In this case fluids with lower velocities existing in the lower pressure region are carried into the central region of the section and there occurs a decrease in V_x/V_{m1} as shown in Fig. 3(c). The dotted lines in Figs. 4(a) and (b) denote the boundary on which $Ri = 0$. Positive values of Ri prevail in the region lying upside of this line and the turbulence will be suppressed there by the Coriolis force. An examination of the results in Figs. 4(a) and (b) shows that negative values of Ri are predominant when $\lambda_0 = -0.65$ and positive values of Ri prevail when $\lambda_0 = 0.65$.

Figure 5 shows the changes in the velocity profile along the channel for different values of N when a shear type flow of $\lambda_0 = -0.65$ is introduced. When $N = 0$, the gradient of the velocity curve given at the inlet section, S(0), is preserved almost unchanged up to the downstream sections. When the diffuser is rotated with $N = 0.06$, the inlet velocity profile is also preserved up to the section S(4), but after this section the profile is deformed gradually and at the

Fig. 5 Changes of velocity profiles along channel for $\lambda_0 = -0.65$ (measured on ③-③ in each section)

section S(9) quite a different profile with an opposite velocity gradient is seen to be brought about. When the spin ratio N is increased to 0.12, the secondary flow is also intensified and a large change in velocity profile is seen between the sections S(0) and S(4).

The intensity of the secondary flow caused by the channel rotation is considered to depend on the distribution of the Richardson numbers in the diffuser sections, and turbulent velocity fluctuations and the secondary flows are activated in the region of $Ri < 0$ while they are suppressed in the region of $Ri > 0$. The regions of positive and negative Richardson numbers are shown in Fig. 5 when $N = 0.12$. In the section S(0) the region of $Ri < 0$ prevails in the greater part, and the secondary flow is intensified, resulting in a gradual deformation of velocity profile along the channel as is seen in the section S(4). But in this section the region of $Ri > 0$ covers a considerable part of it, resulting in a flow stabilization and a comparatively small change in the velocity

(a) $\lambda_0 = 0$

(b) $\lambda_0 = -0.65$

(c) $\lambda_0 = 0.65$

Fig. 6 Velocity profiles in section S(9)

profile in the downstream sections.

To see the effect of the inlet condition on the flow at the exit, the velocity profiles at the section of $S(9)$ for $\lambda_0 = 0, -0.65$ and 0.65 are compared in Figs. 6(a), (b) and (c), when the spin ratios N are 0, 0.08 and 0.12. In case of a uniform inlet flow ($\lambda_0 = 0$) and $N = 0$, a nearly uniform velocity profile is seen to be maintained. When $N = 0.06$ and 0.12 , the velocity profiles, however, shift slightly toward the higher pressure side, and the boundary layer is thickened in the lower pressure side. When $\lambda_0 = -0.65$, the velocity gradient given at the initial section is maintained almost unchanged up to this section $S(9)$ in the stationary state, but in the rotating states of $N = 0.06$ and 0.12 , large changes in the velocity gradient are caused in the diffuser, showing a remarkably flattened velocity profile at the exit section. When larger velocity fluids shift toward the pressure side of wall as in the case of $\lambda_0 = 0.65$, this uneven profile is kept unchanged up to the downstream sections irrespective of the spin ratio of the diffuser. In this section, $S(9)$, the velocity in the lower pressure region is too small to be measured by the probe and assumed profiles are shown by dotted lines. An observation of flow pattern near the wall was also made in this study but the results are not cited here.

5.2 Coefficient of pressure recovery

The changes in coefficients of the pressure recovery, C_{pw} , along the channel are shown in Figs. 7(a), (b) and (c), when the inlet conditions of flow are $\lambda_0 = 0, -0.65$ and 0.65 , respectively. For reference, the changes of C_{pi} obtained by Eq. (18) for an inviscid flow are also shown by dotted lines. When a uniform flow ($\lambda_0 = 0$) is introduced into the diffuser, the pressure recovery in the downstream sections is slightly decreased both on the higher and lower pressure walls in the rotating state. When the inlet condition is $\lambda_0 = -0.65$, the pressure recoveries are largely improved by the rotation of the diffuser and the value of C_{pi} in downstream sections exceeds that of C_{pim} for an inviscid flow. This increase in C_{pw} is due to a uniformization of velocity distribution in the downstream sections caused by a secondary flow. A small difference observed in the changes of the pressure recovery between the values on the higher and lower pressure walls will be caused by a deviation of the velocity profile from the two-dimensional flow pattern.

When the inlet condition is $\lambda_0 = 0.65$, the channel rotation deteriorates the pressure recovery in the diffuser and it is saturated at the section of $L/W_1 = 3$, not far from the channel inlet. An occurrence of a reverse flow in a downstream section was confirmed by visualizing the flow with use of an oil surface flow method.

The wall pressures at the section of $S(1)$, just upstream of the diffuser inlet,

exhibits a minimum value for each curve of C_{pw} as is seen in Figs. 7. Thus, as a base of pressure in the diffuser in Eq. (10), the wall pressure at the inlet section, P_1 , is replaced by that at the section of $S(0)$.

In order to evaluate the pressure recovery in the rotating diffuser, the pressure coefficients C_{pw} both on the higher and lower pressure walls are averaged and using this mean value, C_p , the experimental results are discussed. In Fig. 8, values of C_p are plotted against the inlet velocity gradient λ_m , which is calculated from the measured velocity distribution. In a stationary case of $N = 0$ the curve of C_p is approximately symmetric with respect to the line

(a) $\lambda_0 = 0$

(b) $\lambda_0 = -0.65$

(c) $\lambda_0 = 0.65$

Fig. 7 Changes of C_{pw} along channel

$\lambda_m = 0$ but in a rotating case of $N > 0$ the symmetry of the curves is lost and C_p in the region of $\lambda_m > 0$ is reduced below that observed in the stationary state, but C_p in the range of $\lambda_m > 0$ is increased by the channel rotation. In Fig. 8 the curve of C_{pi} obtained for an inviscid fluid by Eq. (18) is also shown by a dotted line. At the exit section where the area ratio AR is 2.6, C_p attains its critical value $C_{pi} = (1 - 1/AR^2)^2$, when the inlet condition is $|\lambda| = \lambda_c = (2/AR^2)$. In this case the flow velocity on the wall at the lower velocity side of the diffuser is reduced to zero. The critical points are indicated by a symbol of $\bullet$ in the figure. With a further increase in $|\lambda_m|$ a reverse flow occurs in the diffuser and no additional pressure recovery can be realized in the further downstream sections. The experimental results show that the value of C_p is larger than that assumed in an inviscid fluid (C_{pi}) when $\lambda_m < -0.4$.

5.3 Recovery of pressure energy and loss of energy

In a rotating diffuser the pressure and velocity can not be constant across a section due to the effect of the Coriolis force and the secondary flow motion. In order to examine the pressure energy recovery and loss of energy, it is necessary to calculate the fluxes of pressure, kinetic and total energies across the inlet and outlet sections of the diffuser with use of the measured data. The results are shown in Fig. 9, where E_p and E_k are dimensionless pressure and kinetic energy fluxes both for the inlet and exit sections. In the stationary state of $N = 0$, the pressure in the inlet section is equal to p_1 , irrespective of the inlet condition λ_m , and $E_{p1} = 0$. If the diffuser is rotated, E_{p1} exhibits a small deviation from zero, which will be due to the effect of the pressure gradient caused by the Coriolis force. The value of E_{k1} becomes minimum when $\lambda_m = 0$. In the downstream section S(9) the value of E_{p2} is maximum when the flow is uniform at the inlet section and when $N = 0$. The value decreases as the inlet velocity gradient $|\lambda_m|$ increases.

Fig. 8 Relation between C_p and λ_m

When the channel is rotated with $N = 0.06$ and when $\lambda_m < 0$, E_{p2} increases with a decrease in λ_m and it exceeds the value in the stationary state, when λ_m is below -0.25 approximately. When $N = 0$ the value of E_{k2} is minimum in case of a uniform flow inlet but the minimum point of E_{k2} in case of $N = 0.06$ shifts slightly toward the negative side of λ_m .

Figure 10 shows the coefficients of pressure energy recovery η_p and energy loss ζ , calculated, respectively, by Eqs. (23) and (24). The value of η_p observed at the rotating condition of $N = 0.12$ becomes greater than that in the stationary state when the approaching flow has larger velocities in the lower pressure side ($\lambda_m < 0$). The opposite results are obtained if the flow is introduced into the rotating channel with larger velocities in the higher pressure side ($\lambda_m > 0$), because the low energy fluids are driven to the lower pressure side and they flow downstream along the side. The curves $\eta_p - \lambda_m$ exhibit the same tendency with the curves $C_p - \lambda_m$ (Fig. 8).

6. Conclusions

Two-dimensional straight diffusers with an opening angle of 12° , an aspect ratio of the inlet section and area ratios

Fig. 9 Changes in E_p and E_k of inlet and exit sections with λ_m ($N = 0$ and 0.06)

Fig. 10 Relations of η_p and ζ vs. λ_m

2:1 and 1:2.6, respectively, are rotated about the axis perpendicular to the channel, and the effects of uneven inlet velocity distributions are investigated experimentally. The principal results obtained are as follows:

(1) When a flow having a larger velocity at the lower pressure side is introduced into the diffuser, the velocity distribution is flattened in course of flow by a secondary flow caused by the Coriolis force and the pressure recovery is improved by the rotation.

(2) When the inlet flow has a larger velocity at the higher pressure side, an occurrence of the secondary flow is greatly suppressed and the uneven velocity profile produced at the entrance is maintained up to the diffuser exit and the pressure recovery is deteriorated by the system rotation.

(3) The coefficient of the pressure energy recovery shows almost the same tendency as the coefficient of the pressure recovery C_p .

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