THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47 St., New York, N.Y. 10017

91-GT-323

Investigation of the Prediction of Losses in Radial Vaneless Diffusers

HUASHU DOU
Lecturer and Candidate for Ph.D
Institute of Fluid Mechanics
Beijing University of Aeronautics and Astronautics
Beijing 100083
P. R. China

ABSTRACT

The non-dimensional width B and the inlet flow angle α_i are the important parameters influencing the losses and the stability of radial vaneless diffusers. Their effects on diffuser losses are analyzed in this paper.The portions of the various flow losses change with the variation of non-dimensional width parameter B and flow angle α_i .In a diffuser with small width B, the loss is primarily due to wall friction loss. In a diffuser with large B,the wall friction loss becomes a small part of the total loss,especially when α_i is large. Comparison with experimental data shows that it is better to calculate the performance parameters of radial vaneless diffusers by using Dou's method than by Senoo's method in the design of centrifugal compressors. Senoo's method is found to be only suitable for the conditions of small B values because it calculates simply the wall friction loss and the secondary flow loss and neglects diffusion loss.

NOMENCLATURE

b=diffuser width
b_2=flow passage width at impeller exit
$B=b/d_i$,width-to-diameter ratio,non-dimensional width parameter
C=mean velocity
c_{fu} =component of wall friction coefficient at main-flow direction
$Cp=(p_{se}-p_{si})/(p_{ti}-p_{si})$,static pressure recovery coefficient
D=diameter
F=function of $\bar{r}$,equation(3)
g=acceleration of gravity
h_f=wall friction loss
h_t=total flow loss
$K=h_t/h_f$,function of α_i ,B and inlet flow pattern
K_f=constant for wall friction coefficient, equation(1)
M_i=Mach number
p_s=static pressure
p_t=total pressure
$Reb=C_i b/\nu_i$,Reynolds number
$Rer_i=C_i r_i/\nu_i$,Reynolds number
r=radius
$\bar{r}=r/r_i$,radius ratio,dimensionless
α=flow angle,measured from the tangential direction
α_{opt}=optimum flow angle
β_{2A}=blade angle at impeller outlet,measured from the tangential direction
δ=boundary layer thickness
$\zeta=(p_{ti}-p_{te})/(p_{ti}-p_{si})$,total pressure loss coefficient
λ=constant for compressibility
ν=kinematic viscosity
ρ=fluid density
ϕ=variable coefficient in equation(3),function of α_i ,B and inlet flow pattern

Subscripts

2=impeller exit
e=diffuser exit
i=diffuser inlet

INTRODUCTION

A vaneless diffuser is often used to transform the high kinetic energy into static pressure energy in centrifugal compressors and pumps. The design of the diffuser has a substantial effect on the overall efficiency of the machine.The prediction of energy losses in the vaneless diffuser is of great importance in the process of engineering design. Therefore, a good method for estimating the losses and other performance parameters of the diffuser is desired.

A few theoretical models have been developed since Dean and Senoo's(1960) original work was published. Jansen(1964b) presented a

Presented at the International Gas Turbine and Aeroengine Congress and Exposition
Orlando, FL June 3-6, 1991

symmetrical model with respect to the width of the diffuser. Johnston and Dean(1966) proposed an analysis for calculation of the mixing loss and the wall friction loss in vaneless diffusers by an one-dimensional method. Senoo et al.(1977) developed a more sophisticated model based on the integration equation for a turbulent boundary-layer by selecting reasonable velocity profile expressions and wall friction coefficient. In their analysis, the asymmetry of velocity profile with respect to the width was taken into consideration.Bammert et al.(1978) satisfactorily calculated the changes of total pressure and static pressure along the diffuser radius for a tapered diffuser by using a simple wall friction and a dissipation coefficients even for the condition of extremely distorted inlet profile.

The present author(Dou,1989) proposed a semi-empirical method for estimating the energy losses in vaneless diffusers based on boundary-layer theory and experimental material with a simplified flow model.

The experiments by Yoshinaga et al.(1987) showed that the flow distortion at impeller exit increases with increasing specific speed of the stages(hence increasing diffuser width) and the performance of the vaneless diffuser for high specific speed centrifugal compressors is considerably reduced compared with the value predicted by Senoo's theory.

In the present paper the author intends to further show that the portion of each type of loss changes with the variation of inlet flow angle and the non-dimensional width. In engineering design of centrifugal compressors, the prediction of the diffuser performance must be carried out taking into account the variation of losses due to the change of width ratio as well as the flow angle. It is found by comparing with Yoshinaga's experimental data that it is better to calculate the performance parameters by using Dou's method than by Senoo's method in design, especially, when the width ratio B is large.

THE FLOW LOSSES IN RADIAL VANELESS DIFFUSERS

The flow discharged from a centrifugal impeller into a vaneless diffuser is highly distorted,three dimensional and turbulent. The behavior of this distorted flow has been fairly extensively studied experimentally(Senoo and Ishida(1975), Bammert et al.(1978), Inoue and Cumpsty(1984), Yoshinaga et al.(1987), and Maksoud and Johnson(1989)). The experimental results demonstrate that the kinetic flow energy at the impeller outlet enters the diffuser rather irregularly in both circumferential and axial directions. A number of experiments have shown that the non-uniformity of the flow with respect to the circumferential direction becomes uniform, steady and axisymmetric within the entry region of the diffuser($\bar{r}$<1.15-1.20) by mixing and reversible energy exchange.In this region,the so called "mixing loss" is generated because of the internal friction resulting from the shear stress distribution over the whole inlet cross-section of the diffuser passage.

The flow distortion in the axial direction does not decay as rapidly as the distortion in the circumferential flow direction. This profile could be expected to have a determining effect on the behavior of the flow in the vaneless diffuser(Senoo et al.(1977), Ris (1981), Inoue and Cumpsty(1984), and Yoshinaga et al.(1987)).

As is well known, the wall friction loss resulted from the formation of the boundary-layers on diffuser walls is the main source of flow losses. It is primarily related to width ratio B, flow angle α_i ,inlet Reynolds number and diffuser diameter ratio D_e/D_i .

For a parallel walled radial diffuser and incompressible flow, the mass averaged flow maintains a constant flow angle to the tangential direction,and the flow path traces a logarithmic spiral. Because of the long flow path within the diffuser, friction effects are important. Reducing the diffuser width increases the flow angle and shortens the flow path, but this also decreases the hydraulic diameter of the passage. Since the flow angle α_2 at impeller outlet is given for a given impeller geometry and mass flow coefficient, the wall friction loss is very large for diffusers of low specific speed impellers with narrow width.

Because of the effects of streamline curvature and pressure gradient, the boundary layers in vaneless diffusers are swept by secondary flow,and the velocity profile within the boundary layers is skewed. The degree of skewness in the diffuser entry region becomes severe with the fluid flow . At some radius position the wall limit streamline becomes tangential,and a so called "three dimensional separation" occurs.Thus, a reverse flow region can be present near the wall surface which only exists within the boundary layers, and does not penetrate into the outer main flow. When the flow angle is small,the wall boundary layers become extremely skewed with the rapid growth of the boundary layer thickness and merge with each other near to the diffuser inlet. For this case the thickness of the reverse flow zone occupies a large part of the diffuser width and the loss is increased.After the boundary layers on the walls have grown together the reverse flow zone shifts continuously between the two walls, which causes instability of the flow.

When the flow angle is very large, the stream-tube has a large diffusion along the streamline direction.The boundary layers grow rapidly due to deceleration of the main flow. The kinetic energy of the fluid in the boundary layers decreases rapidly, and the boundary layer separates at some radius in the diffuser. This type of separation is different from the above mentioned separation , the former is induced by secondary flow, and the latter is due to the momentum loss caused by the large diffusion. When the width is large,the flow non-uniformity at the diffuser inlet is severe,and the effect of diffusion is prominent. Therefore, the losses in a vaneless diffuser may be classified into mixing loss, wall friction loss, secondary flowloss, and diffusion loss.

PREDICTION OF PERFORMANCE PARAMETERS FOR VANELESS DIFFUSERS

Senoo's method(Senoo et al., 1977) is usually considered to be more accurate than earlier models. It took into account the non-symmetry of inlet flow profile with respect to the width, explained the observed phenomenon of asymmetric and alternating flow separations on diffuser walls, and predicted the limits of rotating stall and stall for some vaneless diffusers(Senoo and Kinoshita (1977), Van den Braembussche(1980), and Kinoshita and Senoo(1985)).

Yoshinaga et al.(1987) made some experiments for eighteen diffusers with different widths at design conditions. They also made a comparison of measured ζ and Cp with those predicted by Senoo's theory,as shown in Fig.1. It may be seen from Fig.1 that the prediction agrees with experimental data for small width ratio B, but that the agreement is worse for large width ratio B. The authors attributed this poor agreement to the flow distortion, which increases with increasing width ratio B. It is also clear that the extent of agreement between theory and experiments depends on the selection of the value of the parameter K_f in the expression of wall friction coefficient, which limits the accuracy of Senoo's prediction method.

The component of wall friction coefficient in the main-flow direction in reference(Senoo et al., 1977) is derived from the two-dimensional wall friction coefficient, and is expressed as

$$c_{fu} = \lambda K_f \left(\frac{\delta U}{\nu}\right)^{-\frac{1}{4}} \qquad (1)$$

where the coefficient K_f is ordinarily 0.045. Senoo recommended K_f=0.051 through measuring the velocity distribution between the diffuser walls of a blower.

Based on an approximate simplified model, the present author(Dou,1989) presented a semi-empirical method for calculating the energy losses in vaneless diffusers. It is assumed that the fluid particle flows along a cylindrical surface formed by a logarithmic spiral.

The two-dimensional wall friction loss is derived as

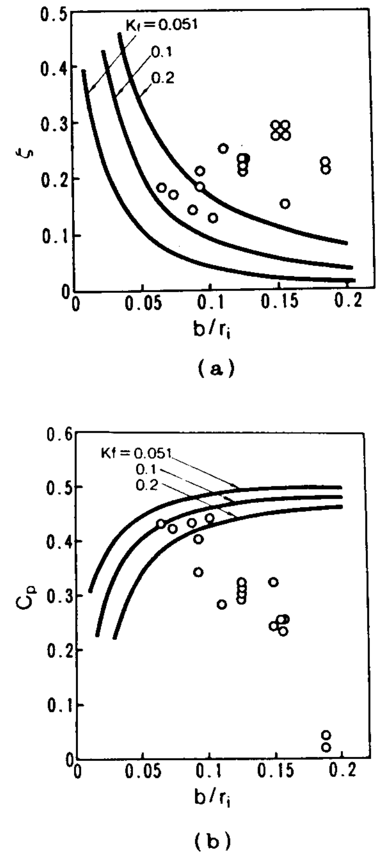
$$h_f = 0.07\left(\frac{R_{er_i}}{\sin\alpha_i}\right)^{-\frac{1}{5}} \frac{F(\bar{r})}{B\sin\alpha_i}\frac{C_i^2}{2g} \qquad (2)$$

where F is a gradually increasing function of radius ratio $\bar{r}$ (Fig.(2)) and is expressed as

$$F(\bar{r}) = \frac{2}{3}(1-\bar{r}^{-\frac{3}{2}}) + \frac{1}{20}(1-\bar{r}^{-4}) + \frac{6}{325}(1-\bar{r}^{-\frac{13}{2}}) + \frac{11}{1125}(1-\bar{r}^{-9}) + \cdots\cdots \qquad (3)$$

The total energy loss is expressed as

$$h_t = Kh_f = \phi F R_{eb}^{-\frac{1}{5}}\frac{C_i^2}{2g} \qquad (4)$$



(a) Total pressure loss coefficient
(b) Static pressure recovery coefficient

Fig.1 Effect of diffuser width on total pressure loss and static pressure recovery coefficients(Yoshinaga,1987)
O ——measured
—— ——Calculated by Senoo's method
(M_i =0.55, α_i=30°, r_e/r_i =1.44, uniform inlet flow)

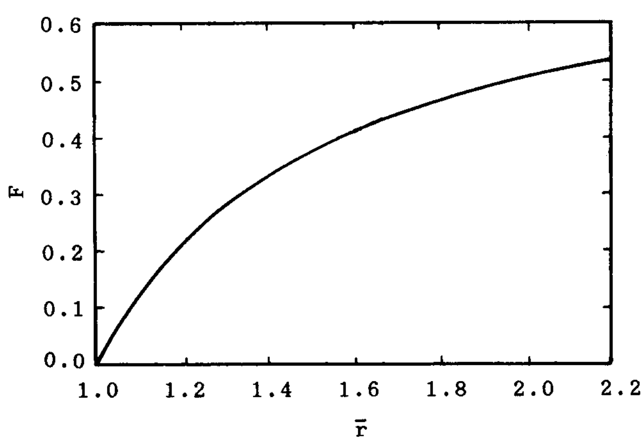


Fig.2 Function F($\bar{r}$) versus $\bar{r}$

where the parameters K and ϕ are both functions of α_i and B, where they have a relation

$$\phi=0.08K(B\sin\alpha_i)^{-\frac{4}{5}} \qquad (5)$$

In engineering design, for the ordinarily used range of α_i and B value, the value of ϕ may be taken from the semi-empirical curve as shown in Fig.3 or be calculated from the following approximate equations

$$\phi(z)=20.52-2.594z+0.136z^2-0.0020z^3 \qquad (6a)$$

for curve I;

$$\phi(z)=29.12-3.842z+0.217z^2-0.0039z^3 \qquad (6b)$$

for curve II, where $z=B\sin^2\alpha_i\times10^3$

Since the total loss may be written as

$$h_t=\zeta\frac{C_i^2}{2g} \qquad (7)$$

Comparing eq.(4) and eq.(7), we have

$$\zeta=\phi F R_{eb}^{-\frac{1}{5}} \qquad (8)$$

The following equation may be used to estimate the static pressure recovery coefficient at incompressible flow condition

$$Cp=1-\left(\frac{C_e}{C_i}\right)^2-\zeta\doteq1-\left(\frac{D_i}{D_e}\right)^2-\zeta \qquad (9)$$

The values calculated with the experimental data of eighteen stages (Yoshinaga et al.,1987) are shown in Fig.3. Although these data of all the eighteen stages were obtained at design condition, it may be seen that some stages were not at the best condition of the diffusers.

The comparison of ζ and Cp predicted by Dou's method with Yoshinaga's data are shown in Fig.4. Comparing Fig.4 with Fig.1, we find that for practical cases it is better to calculate the performance parameters of radial vaneless diffusers by Dou's method than by Senoo's method. This is because Senoo's method doses not involve the diffusion loss. When the width ratio B and α_i are both large, the loss caused by the large divergence of the flow passage is substantial.

The comparison of the experimental data by Rodgers with that predicted by Dou's method is shown in Fig.5. The predicted Cp is lower than the experimental data. This may be attributed to the effects of compressibility.Dou's method is only suitable for the conditions of low mach numbers because it is obtained from incompressible condition and low Mach number test data.

Neither Dou's method nor Senoo's method takes into account the "mixing loss",most of which is usually generated in the range of

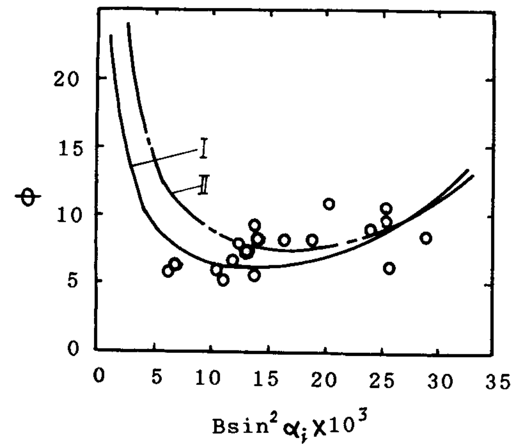
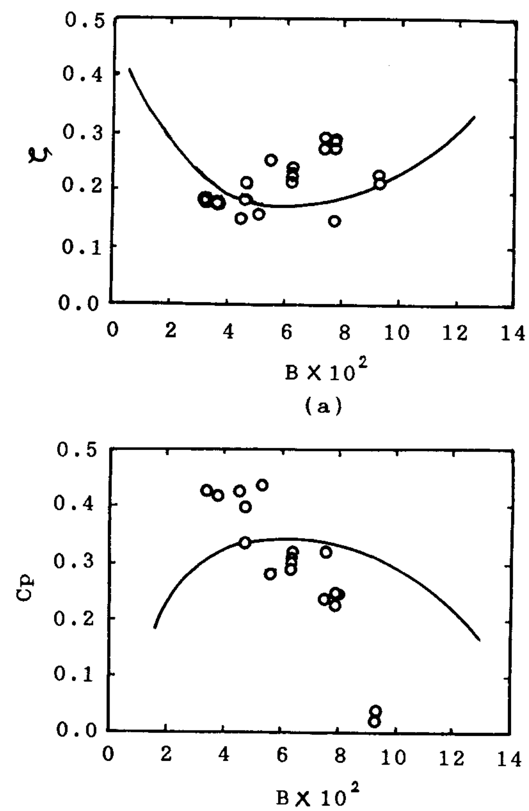


Fig.3 Variation of ϕ versus $B\sin^2\alpha_i\times10^3$
——— Diffusers followed by scrolls
—·— Diffusers followed by return channels
o —— Experimental data by Yoshinaga (1987)



(a) Total pressure loss coefficient
(b) Static pressure recovery coefficient

Fig.4 Comparison of performance parameters predicted by Dou's method with Yoshinaga's experimental data
——— Predicted(at $\alpha_i=30^\circ$)
o —— Yoshinaga's experimental data

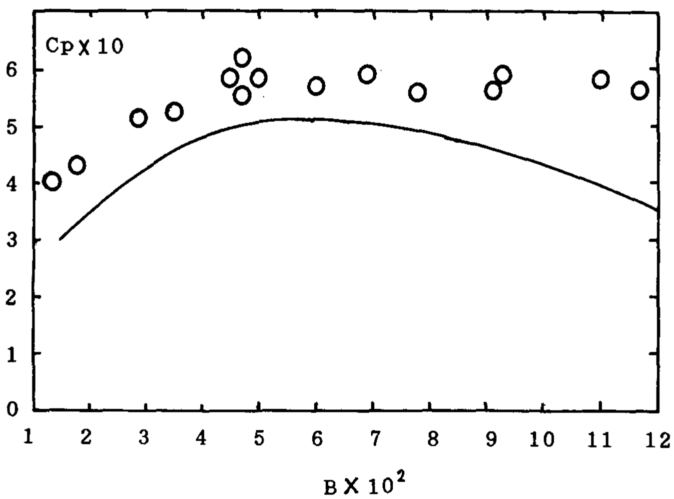


Fig.5 Comparison of Cp predicted by Dou's method with Rodgers' experimental data ($De/Di=1.71$, $Reb=10^5$—10^6, $Mu=0.6$—1.2, at $\alpha_i=30°$)

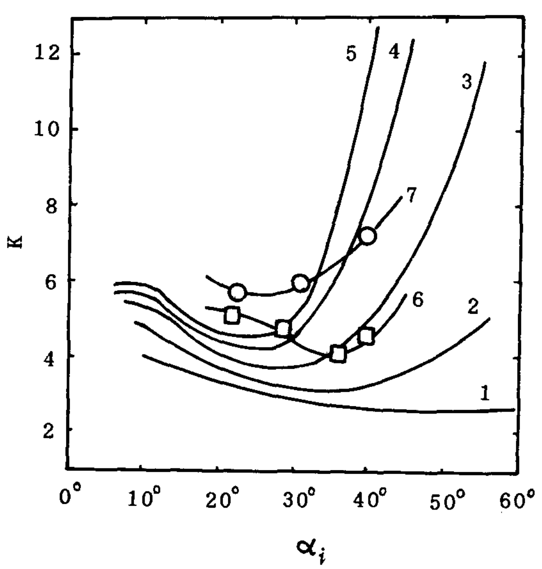


Fig.6 Variation of K versus mean inlet flow angle

References	number	B	r_e/r_i	β_{2A}
Den(1960)	1 2 3 4 5	0.0187 0.0312 0.0468 0.0608 0.0697	1.583	45°
Johnston & Dean (1966)	6	0.0313	1.80	90°
Dou(1989)	7	0.0601	1.45	40°

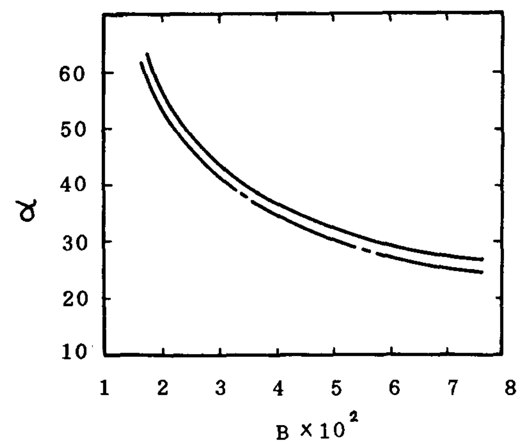


Fig.7 The flow angle minimizing the total loss and the flow angle minimizing K versus B(from Den's data)
—— α_{opt} ,the flow angle minimizing ζ
—·— The flow angle minimizing K

$D/D_2=1.00$-1.06.The evaluation of "mixing loss" has been given by Johnston and Dean(1966). Bammert et al.(1978) estimated this part of the loss by using a "dissipation factor" which was defined in the range of $D/D_2=1.00$~1.16 and decreased with the radius ratio.

EFFECT OF WIDTH RATIO B AND FLOW ANGLE α_i ON THE LOSSES

The value of K in equation(4) calculated with data cited in reference(Dou,1989) is shown in Fig.6, and the results are very interesting. It is found that there exists a flow angle which minimizes the magnitude of K for a given width ratio B. At this flow angle the wall friction loss h_f is the largest portion in the total loss. When the flow angle decreases from this α_i value,the parameter K increases. This is due to the secondary loss rising as a result of boundary layer skew and the occurence of reverse flow. When the flow angle increases from this α_i value, the parameter K increases strongly, which is because the diffusion loss becomes larger owing to the non-uniformity of flow and the possible separation.

For the large width B, K varies over a large range with the flow angle α_i because the losses other than the wall friction loss are large for both small and large α_i . For the small width B, the K curve is flat and low as a larger portion of the losses are due to wall friction loss. Since the divergent degree is reduced for small width B,the uniformity of flow is strengthened so that the diffusion and secondary loss are decreased.The wall friction loss then becomes the primary loss and the other losses are of little importance because of the smaller hydraulic diameter.It is also clear that the variation of B has an intense effect on the value of K when the flow angle α_i is large.In other words,the magnitude of B has a strong influence on the diffusion loss, especially for large flow angles.

Fig.7 shows the inlet flow angle minimizing K ,which is near the optimum flow angle α_{opt} for the minimum of loss coefficient.

The above discussion may be used to interpret the fact that Senoo's theory differs from experimental data for large width B in Fig.1. Senoo's method calculates simply the wall friction loss and the secondary loss, and does not include the diffusion loss. Although the parameter K_f is changed from 0.045 to 0.051, the loss predicted is far less than the real loss for the cases of large width B. Therefore, this prediction method is only suitable for the condition of small width B, and not for the diffusers with large width B.

The value of K calculated with Yoshinaga's data and that predicted by Dou's method are shown in Fig.8. Good agreement is demonstrated. Yoshinaga's data varies more steeply than Dou's prediction does when B increases. This may be attributed to that the distortion of the inlet flow field with the increasing width by Yoshinaga is more severe than those data used by Dou.For different impeller blade geometries, the distortion of the discharged flow at impeller outlet might vary in a different manner with increasing width. The assumption of uniform inlet flow(Dou,1989) is only approximate.

EFFECT OF DIFFUSER WIDTH ON THE OVERALL PERFORMANCE STABILITY OF STAGES

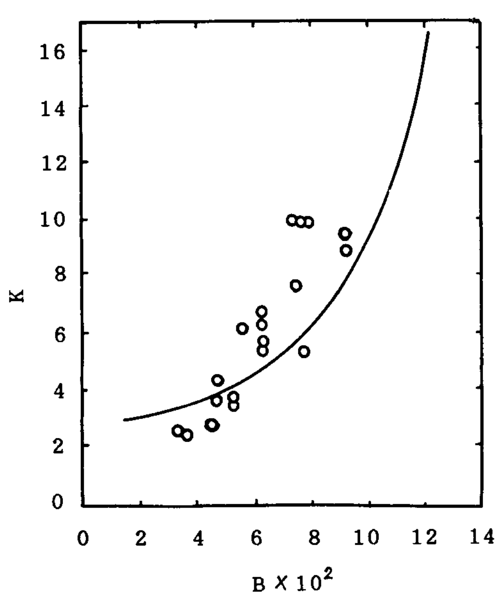


Fig.8 Variation of K versus non-dimensional width B

——— Predicted by Dou's method with $\alpha_i = 30°$

O ——Calculated from Yoshinaga's data(1987)

This paper does not add anything to our knowledge of rotating stall in vaneless diffusers, about which there exist two contradictory conclusions(Jansen(1964a), Senoo and Kinoshita(1977)) in terms of the influence of the width ratio B.

Surge is a system instability depending on compressor and throttle characteristic and inlet and outlet geometry. Investigations have shown that the slope of the overall stage pressure ratio is an indication of the onset of instability.Surge will occur when the slope of the overall pressure rise curve exceed a given positive value(Greitzer, 1981; Van den Braembussche,1985). The compressor overall pressure rise is the sum of impeller and diffuser pressure rise. The vaneless diffuser usually has a positive slope at small flow angle. At smaller flow angles the positive slope of the diffuser pressure rise curve increases and if it is not compensated by the negative impeller pressure rise slope surge will occur. Therefore, the characteristic of pressure recovery of diffusers has a decisive impact on the stability of the overall performance of the machine. Dou(1990) states that an inlet flow angle of 14° is the onset of flow instability for diffusers with usual widths(B=0.035-0.060), which agrees with considerable data.

According to available data, for diffusers of low specific speed impellers (B<0.04-0.05),the narrower the diffuser width, the more unstable the static pressure recovery. This is because the wall friction loss is dominant for small width diffusers.The smaller the width, the large the loss due to the smaller hydraulic diameter. This accords with Rodgers' extensive investigation(Rodgers, 1984).

CONCLUSIONS

The following conclusions may be summarized:

1. The wall friction loss is the primary loss in vaneless diffusers with small width ratio B.For diffusers with large width ratio B, the wall friction loss is a smaller part of the total loss.

2. When the flow angle α_i is small, the loss in the diffuser results primarily from the wall friction loss and the secondary loss. When the flow angle α_i is large, the loss in the diffuser is mainly composed of the wall friction loss and the diffusion loss. For a given width ratio B, there exists a flow angle at which the largest part of the losses are due to wall friction. This flow angle is near the optimum flow angle which minimizes the loss coefficient.

3. Senoo's method involves only the wall friction loss and the secondary loss, and does not include the diffusion loss.This method is only suitable for cases of narrow diffusers at small and moderate flow angle. Dou's method is a semi-empirical method, but it includes all the possible loss sources.

4. For practical cases occurring in compressor design it is more accurate to evaluate the performance parameters of vaneless diffusers by using Dou's method than

by Senoo's method.

The author is grateful to Dr.M.V.Casey and the reviewers for their favourable comments on the original manuscript.

REFERENCES

Bammert,K.,et al.,1978,"Vaneless Diffuser Flow with Extremely Distorted Inlet Profile," ASME Paper, 78-GT-47

Dean,R.C.Jr.,and Senoo,Y.,1960, "Rotating Wakes in vaneless Diffusers," ASME Journal of Basic Engineering,Vol.82,No.3,pp.563-574

Den,G.N.,1960, "The Effects of Relative width of Fixed Part on the work of Centrifugal Stages with Vaneless Diffusers,"Power-Machine-building,(in Russian),No.11,pp.20-24

Dou,H.S.,1989, "A Method of Predicting The Energy losses in Vaneless Diffusers of Centrifugal Compressors,"ASME Paper,89-GT-158

Dou,H.S.,1990, "On The Problems of Design for Radial Vaneless Diffusers with Convergent Walls," In: Proceedings of the 1st International Symposium on Experimental and Computational Aerothermodynamics of Internal Flows, World Publishing Corporation , Beijing, pp.595-601

Greitzer,E.M., 1981, "The Stability of Pumping Systems--The 1980 Freeman Scholar Lecture, ASME Journal of Fluid Engineering, Vol.103,No.2,pp.193-242

Inoue,M.,and Cumpsty,N.A., 1984, "Experimental Study of Centrifugal Impeller Discharge Flow in Vaneless and Vaned Diffusers," ASME Journal of Engineering for Gas Turbines and Power,"Vol.106,No.2,pp.455-467

Jansen,W.,1964a, "Rotating Stall in a Radial Vaneless Diffusers, " ASME Journal of Basic Engineering,Vol.86,No.4,pp.750-758

Jansen,W.,1964b, "Steady Fluid Flow in a Radial Vaneless Diffuser, " ASME Journal of Basic Engineering,Vol.86,No.3,pp.607-619

Johnston, J.P., and Dean,R.C.Jr., 1966, "Losses in Vaneless Diffusers of Centrifugal Compressors and Pumps," ASME Journal of Engineering for Power,Vol.88,No.1,pp.49-62

Kinoshita,Y.,and Senoo,Y.,1985, "Rotating Stall Induced in Vaneless Diffusers of Very Low Specific Speed Centrifugal Blowers," ASME Journal of Engineering for Gas Turbine and Power,Vol.107,No.2,pp.514-521

Maksoud,T.M.A., and Johnson,M.W., 1989, "Stress Tensor Measurements within The Vaneless Diffuser of a Centrifugal Compressor," Journal of Mechanical Engineering Science,Vol.203,No.C1,pp.51-59

Ris,B.F.,1981, The Centrifugal Compressive Machinery,(in Russian), 3rd Edition, Machine-building Leningrad Branch,Leningrad,pp.162-175

Rodgers,C.,1984,"Static Pressure Recovery Characteristics of Some Radial Vaneless Diffusers," Canadian Aeronautics and Space Journal,Vol.30,No.1,pp.42-54

Senoo,Y.,and Ishida,M.,1975, "Behavior of Severely Asymmetric Flow in a Vaneless Diffuser," ASME Journal of Engineering for Power,Vol.97,No.3,pp.375-387

Senoo,Y.,et al., 1977, "Asymmetric Flow in Vaneless Diffusers of Centrifugal Blowers," ASME Journal of Fluid Engineering, Vol.99, No.1,pp.104-114

Senoo, Y., and Kinoshita, Y., 1977, "Influence of Inlet Flow Conditions and Geometries of Centrifugal Vaneless Diffusers on Critical Flow Angles for Reverse Flow," ASME Journal of Fluid Engineering,Vol.99,No.1, pp.98-103

Van den Braembussche, R. et al., 1980, "Rotating Non-uniform Flow in Radial Compressors," AGARD CP-282,paper 12.

Van den Braembussche,R.,1985, "Design and Optimization of Centrifugal Compressors," In: Thermodynamics and Fluid Mechanics of Turbomachinery,Ucer,A.S.et al.,eds., Vol.II, Martinus Nijhoff Publishers, Dordrecht, pp.829-885

Yoshinaga,Y.,et al., 1987, "A Study of Performance Improvement for High Specific Speed Centrifugal Compressors by Using Diffusers with Half Guide Vanes, " ASME Journal of Fluid Engineering, vol.109,No.4, pp.359-367