Alternate Blade Stall and Rotating Stall in a Vaned Diffuser*

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Flow instability in a vaned diffuser with an even number of blades was examined experimentally and analytically. In the experiments, an alternate blade stall, an asymmetric stall, and two types of rotating stalls (backward/forward rotating stall) were observed depending on the impeller/diffuser clearance. For narrow clearance with strong impeller/diffuser interaction, the alternate blade stall and backward rotating stall mainly occurred. With increasing the clearance, the forward rotating stall also occurred, and the onset of rotating stall shifted toward the higher flow rate corresponding to the pressure performance in the vaned diffuser. Simple 2D stability analysis showed that the impeller/diffuser clearance affects the speed and direction of the stall propagation, and the slope of the diffuser pressure performance vs. flow rate curve affects fundamentally the onset of the flow instability within the diffuser.

Key Words: Rotating Stall, Alternate Blade Stall, Asymmetric Stall, Vaned Diffuser, Diffuser Pressure Performance, Impeller/Diffuser Clearance

1. Introduction

For low specific speed centrifugal pumps with a vaned diffuser, dents in the performance curve are occasionally observed⁽¹⁾. These dents are often attributed to the flow instabilities in the vaned diffuser⁽²⁾. With the aim of improvement of the performance curve instabilities, Hergt and Benner⁽³⁾, and Eisele et al.⁽⁴⁾ made experimental observations of the internal flow in vaned diffusers at a low flow rate.

On the other hand, it is well known that "alternate blade flow instabilities" occur in the rotor with an

even number of blades, such as alternate blade cavitation in an inducer⁽⁵⁾ and in a propeller turbine⁽⁶⁾, or alternate blade stall in a centrifugal impeller⁽⁷⁾. However, as far as the authors know, alternate blade flow instabilities in stationary blades (i.e., diffuser) have not been reported except for the flow instability of the branched diffuser reported by Ehrich⁽⁸⁾. In addition, there are few reports about the rotating stall in pump vaned diffusers^{(9)–(11)}, although there are several reports concerning the flow structure in vaned diffusers^{(12)–(14)}.

In this study, the alternate blade stall and rotating stall within a vaned diffuser were investigated experimentally by using a model of a boiler feed pump with a vaned diffuser. To examine the effect of the clearance between the impeller and diffuser on these flow instabilities, a systematic experiment was conducted by increasing the impeller/diffuser clearance step by step. We discuss how that the diffuser pressure performance affects primarily the onset of the alternate stall and the rotating stall through the influence of the impeller/diffuser clearance. In addition, the flow instabilities observed in the experiment are compared with the results of 2D linear stability analysis.

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2. Nomenclature

- b : vane height
 D : diameter
 D_3/D_2 : ratio of inlet diameter of diffuser to outlet diameter of impeller
 j : imaginary unit
 N : number of stall cells
 P_{xy} : location of pressure tap, 'x' represents the radial location, and 'y' represents the diffuser channel
 p : pressure
 Δp : pressure fluctuation
 Q : flow rate
 T : period of rotating stall propagation
 u : peripheral speed of impeller
 v : velocity
 ϕ : flow coefficient $= \frac{Q}{\pi D_2 b_2 u_2}$
 Ψ : impeller inlet total to delivery tank pressure coefficient $= \frac{p_d - p_{1t}}{\rho u_2^2}$
 Ψ_s : impeller inlet total to impeller outlet static pressure coefficient $= \frac{p_2 - p_{1t}}{\rho u_2^2}$
 Ψ_d : diffuser outlet static to diffuser inlet total pressure coefficient $= \frac{p_{3t} - p_d}{\rho v_{\theta 3}^2}$
 $\Delta \Psi$: coefficient of pressure fluctuation $= \frac{\Delta p}{\rho u_2^2}$
 Ω : impeller angular velocity
 Ω_s : rotating stall angular velocity
 Ω_s/Ω : propagation speed ratio of rotating stall
- Subscripts**
- 1: impeller inlet
 - 2: impeller outlet
 - 3: diffuser inlet
 - 4: diffuser outlet
 - d: delivery tank
 - r: radial direction
 - t: total pressure
 - θ : tangential direction

3. Test Apparatus

Figure 1 shows the test pump that consists of a seven bladed centrifugal impeller and an eight vaned diffuser connected with a crossover and return vane. This is a model for one stage of a multistage boiler feed pump. The design flow coefficient is $\phi=0.077$. The outlet diameter of the impeller is $D_2=150.0$ mm, and the inlet diameter of the diffuser is $D_3=155.2$ mm; i.e., the ratio of diameters is $D_3/D_2=1.035$. In order to examine the effects of the impeller/diffuser clearance, the impeller diameter is trimmed in sequence from D_2

Table 1 Principal dimensions of test impeller and diffuser

	Impeller	Diffuser
Inlet diameter (mm)	$D_1=62.6$	$D_3=155.2$
Outlet diameter (mm)	$D_2=150.0, 147.8, 144.4, 141.1$	$D_{\max}=196.8$ (Maximum diameter)
Channel height (mm)	$b_2=14.3$ (Outlet)	$b_3=15.4$ (Inlet)
Vane angle (deg)	$\beta_2=22.5$	$\alpha_3=11.5$
Number of vanes	7	8

Fig. 1 Sketch of test pump, impeller with seven blades, and diffuser with eight vanes

$=150.0$ mm to 147.8 mm, 144.4 mm, and 141.1 mm. Air was used as the working fluid, and the shaft rotational speed was 3 000 rpm. The principle dimensions of the impeller and diffuser are shown in Table 1.

Figure 2 shows the test facility. The loop is an open type with a delivery tank (0.0743 m^3). The Helmholtz resonance frequency of the test facility could be calculated as about 10 Hz. The flow rate is measured with the flow nozzle (JIS Z 8762) at the discharge pipe. Pressure taps in each diffuser channel are located at impeller inlet P_2 ($P_{21} \sim P_{28}$: $D=154.0$ mm), diffuser inlet P_3 ($P_{31} \sim P_{38}$: $D=165.8$ mm), and return vane outlet P_4 ($P_{41} \sim P_{48}$: $D=125.0$ mm), as shown in Fig. 2. The subscripts of the pressure tap ' P_{xy} ' represent the radial location 'x' ($=1 \sim 3$), and the diffuser channel 'y' ($=1 \sim 8$). These pressure taps penetrate through the front casing of the diffuser. In addition, pressure tap P_d is located at the delivery tank to measure the pump discharge pressure. Steady pressures are measured by using the manometer to obtain the pressure performance. Moreover, unsteady pressures are measured by using the pressure sensors. To examine the onset of the various stalls within the diffuser, the cross spectra of the pressure fluctuations are analyzed in each channel. Furthermore, the oil film method for flow visualization was performed to investigate the flow structure in the diffuser.

4. Alternate Blade Stall and Backward Rotating Stall

For the case of the impeller with the maximum

Fig. 2 Sketch of test facility. Pressure taps P_2 at impeller outlet, P_3 at diffuser inlet, and P_4 at return-vane outlet are located on the casing wall in each diffuser channel.

diameter $D_2=150.0$ mm ($D_3/D_2=1.035$), the pressure performances Ψ , and Ψ_s plotted against the flow rate ϕ are shown in Fig. 3(a). There is a small dent in the pressure performance Ψ at $\phi=0.06$ (about 80% of the design flow rate). In the flow range of $\phi=0.04\sim0.06$, the impeller static pressure performance curves Ψ_s are separated in two lines; i.e., the pressures at the diffuser inlet become higher in Channels 1, 3, 5, and 7 and lower in Channels 2, 4, 6, and 8. For the typical flow rate $\phi=0.054$, the pressures at P_{2y} ($y=1\sim8$) are shown in Fig. 4(b). This figure represents the deviation from the averaged pressure of eight channels. The pressures in Channels 1, 3, 5, and 7 are higher than those in Channels 2, 4, 6, and 8, then the high-low pressure pattern in two adjacent channels repeats in the circumferential direction. This pressure pattern is stable and does not propagate at all. We term this phenomenon "alternate blade stall (A.B.S.)". We should note here that the pressures at diffuser outlet P_{4y} are equal in each channel even if the alternate blade stall is observed at the diffuser inlet P_{2y} .

On the contrary, the pressure pattern is uniform at lower and higher flow rates than the range of $\phi=0.04\sim0.06$, as shown in Figs. 4(a), (c). When we decreased the flow rate to less than $\phi=0.04$, periodic pressure fluctuation with small amplitude was observed at the diffuser inlet P_{3y} . From the cross spectrum analysis of the pressure fluctuations in two

Fig. 3 Pressure performance curves for $D_3/D_2=1.035$. (a) Total pressure performance Ψ , and impeller pressure performance Ψ_s , (b) Diffuser pressure performance Ψ_d

Fig. 4 Pressure pattern of P_2 for $D_3/D_2=1.035$ at (a) $\phi=0.032$, (b) $\phi=0.054$, (c) $\phi=0.065$. Radial axis represents the deviation from the averaged value of eight channels. Pressure pattern in (b) shows the alternate blade stall, showing the higher pressure in Channels 1, 3, 5, and 7, and lower pressure in Channels 2, 4, 6, and 8

adjacent channels, it can be found that this pressure fluctuation is caused by "backward rotating stall (B.R.S.)"⁽¹⁵⁾. The rotating stall propagates in the opposite direction of the impeller rotation, the number of stall cells is $N=3$, and the propagation speed ratio is considerably small, $\Omega_s/\Omega=-0.008$. On decreasing the flow rate further, the amplitude of the component caused by the rotating stall decreases, although overall pressure fluctuation increases. Then, the rotating stall could not be specified at a lower flow rate than $\phi=0.03$.

Diffuse pressure performance Ψ_d is shown in Fig. 3(b). Ψ_d is calculated by the following equation:

$$\psi_d = \frac{p_{3i} - p_4}{\rho v_{\theta 2}^2} \quad (1)$$

Under the assumption that the total pressure at diffuser inlet P_3 equals that at impeller outlet P_2 , we obtain the following equation:

$$\psi_d \approx \frac{p_2 + \frac{\rho}{2}(v_{r2}^2 + v_{\theta 2}^2) - p_4}{\rho \left(\frac{D_3}{D_2} v_{\theta 2} \right)^2} \quad (2)$$

To estimate ψ_d , the value of $v_{\theta 2}$ is calculated from the singularity method, and the others are taken from the experimental results. From the linear stability analysis, Yoshida et al.⁽⁹⁾ reported that the condition, under which the slope of the diffuser pressure performance ψ_d vs. flow rate ϕ curve is flat, represented the boundary between the stable and unstable region of the rotating stall. The range where the slope of ψ_d - ϕ curve is negative represents the unstable region of the diffuser rotating stall. Considering the averaged value of ψ_d in eight channels shown in Fig. 3(b), the slope of the ψ_d - ϕ curve is almost flat around the onset of the alternate blade stall, and it is negative at the occurrence region of the backward rotating stall. These experimental results correspond to the above analysis qualitatively.

On the other hand, Hergt and Starke⁽²⁾ observed that the dent of the pressure performance curve in a centrifugal pump with vaned diffuser occurs at 70% of the design flow. He reported that this is caused by the diffuser reverse flow near the front shroud, the flow at the diffuser inlet becomes nonuniform in the span-wise direction due to the reverse flow. For the purpose of investigating the span-wise extension of the present alternate blade stall, additional pressure taps are formed at the back casing of the diffuser. However, measured results are the same as that measured at the front casing. Therefore, we might conclude that the alternate blade stall in the present experiments is almost a two-dimensional phenomenon in the span-wise direction.

Under the condition of the alternate blade stall, the pressures in Channels 1, 3, 5, and 7 are not always higher than the others, the opposite pressure pattern was often observed; i.e., the pressures in Channels 2, 4, 6, and 8 are higher than the others. From this result, it was found that the alternate blade stall is not caused by the manufacturing geometric error of the diffuser vane. But the alternate blade stall is primarily caused by the flow instability of the diffuser.

5. Flow Pattern under Alternate Blade Stall

For further investigation of the flow structure of the alternate blade stall, thirty-two additional pressure taps were formed in Channels 1 and 2. The

Fig. 5 Locations of the pressure taps in Channels 1 and 2. There are 18 pressure taps in each channel, respectively.

Fig. 6 Pressure contour in Channels 1 and 2 for $D_3/D_2 = 1.035$ at (a) $\phi = 0.063$, (b) $\phi = 0.053$, (c) $\phi = 0.032$. The pressure at the diffuser inlet in Channel 1 is higher than that in Channel 2 at (b) $\phi = 0.053$, showing the alternate blade stall.

coordinates of the static pressure taps were formed by the cross of six radial grid lines and three stream-wise grid lines in the blade-to-blade passages of Channels 1 and 2, respectively, as shown in Fig. 5.

Figure 6 (b) shows the pressure contour in the channels under the condition of alternate blade stall at

Fig. 7 Oil surface-flow on the casing wall of Channels 1 and 2 for $D_3/D_2=1.035$ at $\phi=0.053$, corresponding to Fig. 6 (b). Two arrows in the figure show the flow direction observed with tufts.

$\phi=0.053$. For comparison, Figs. 6 (a), (c) show the pressure contour without the alternate blade stall at a higher flow rate ($\phi=0.063$), and lower flow rate ($\phi=0.032$). In case of a higher flow rate $\phi=0.063$, all contour lines near the throat are almost perpendicular to the suction side, and parallel to each other. Then, the pressure recovery is normal in both channels. On the contrary, contour lines are crowded near the leading edge in case of a lower flow rate $\phi=0.032$. The pressure recovery on the pressure side at the leading edge is extreme. This is because the blade load of the diffuser vane near the leading edge becomes high caused by large incidence. However, the flow patterns in both channels are identical.

In the case of $\phi=0.053$ under the alternate blade stall condition, both characteristics mentioned above are observed simultaneously in Channels 1 and 2. The contour lines are almost parallel to each other in Channel 2, which is similar to that at $\phi=0.063$. However, the contour lines are bent around the throat in Channel 1, which is similar to that at $\phi=0.032$.

Figure 7 shows the result of flow visualization on the front casing wall obtained by the oil film method under the alternate blade stall condition ($\phi=0.053$); the arrows in the figure represent the flow direction observed by using the tuft. In this picture, a different flow pattern can be observed by comparing the two channels. Reversed streamline is observed around the throat in Channel 1, and no oil surface-flow pattern exists downstream of the throat. This region corresponds to the higher pressure region in Fig. 6 (b), and the tuft points toward the impeller to avoid the reverse region. On the contrary, the tuft in Channel 2 points toward the diffuser inlet, and an oil surface-flow pattern extends behind the throat.

From these results the following can be considered. When we decrease the flow rate, the incidence to the diffuser vane increases, and the diffuser leading

Fig. 8 Clearance between the impeller and diffuser. Gap B is varied by trim of the impeller, although Gap A is kept constant.

edge stall starts to occur, but not all vanes stall at once, and at least one of them starts to stall before the others. In the channel with stall, the flow reverses to the impeller and turns to the next diffuser channel through the leading edge of the diffuser vane. Therefore, the pressure around the leading edge on the suction side decreases in the next channel. In the next channel, the diffuser effect becomes larger than that in the adjacent channel because the flow rate in the next channel is larger than that in the adjacent one due to the "leakage flow" through the impeller/diffuser clearance. Similar leakage flow was observed during the developed states of the rotating stall in the vaned diffuser⁽¹⁰⁾. In case of an even number of blades, this paired nonuniform flow pattern spreads circumferentially to all channels, and therefore the alternate blade stall pattern appears clearly.

From these experimental results, it was found that the flow instabilities in the vaned diffuser, such as the alternate blade stall or rotating stall, occur when the slope of the diffuser pressure performance Ψ_d vs. flow rate ϕ curve is flat or negative. When we increase the flow rate through a diffuser channel with constant circumferential velocity in the range with negative slope of Ψ_d - ϕ , the diffuser pressure recovery increases, and the diffuser inlet pressure decreases if the diffuser outlet pressure is kept constant. Then, the flow toward the channel inlet increases further. This is the principal reason why the flow instabilities in the vaned diffuser occur depending on the slope of the diffuser pressure performance Ψ_d vs. flow rate ϕ curve.

6. Effects of Impeller/Diffuser Clearance

6.1 Impeller trim

To examine the effects of impeller/diffuser clearance on the diffuser flow instability, a test impeller is trimmed as shown in Fig. 8. The Gap B, the clearance

Fig. 9 Pressure performance curves Ψ , Ψ_s , and Ψ_d showing the occurrence regions of diffuser stalls for various impeller/diffuser clearances, (a) $D_3/D_2=1.050$, (b) $D_3/D_2=1.075$, (c) $D_3/D_2=1.100$

Fig. 10 Pressure pattern of P_2 for $D_3/D_2=1.050$ at $\phi=0.045$, showing the asymmetric stall with higher pressure in Channel 3 and lower pressure in Channels 2 and 4

between the diffuser inlet and impeller outlet, changes from 2.6 mm to 3.7, 5.4 and 7.1 mm in sequence. Then, the ratios of D_3/D_2 are 1.050, 1.075, and 1.100, respectively. However, the Gap A, the clearance between the diffuser casing and impeller shroud, is kept constant at 0.5 mm, as shown in Fig. 8, because Guelich et al.⁽¹⁾ reported that Gap A is sensitive to the flow instability particularly at a low flow rate. The pressure performance curves showing the occurrence regions of various types of diffuser stall are shown in Figs. 9(a)~(c).

6.2 In the case of $D_3/D_2=1.050$ (Fig. 9(a))

The alternate blade stall occurs at $\phi=0.045\sim0.06$. This region of alternate blade stall slightly decreases compared with that in the case of $D_3/D_2=1.035$. In the flow range at $\phi=0.040\sim0.045$, the pressure at P_{2y} becomes nonuniform as shown in Fig. 10. This pattern

is asymmetric, however stable. As we can distinguish this pressure pattern from that of the alternate blade stall, we term this state "asymmetric stall (A.S.)". In this situation, a low-high-low pressure pattern can be found only in P_{22} , P_{23} , and P_{24} . When we decrease the flow rate from the asymmetric stall condition, the backward rotating stall with three stall cells and propagation speed ratio $\Omega_s/\Omega=-0.008$ is observed. However, the pressure fluctuation caused by the rotating stall disappears at less than $\phi=0.035$.

6.3 In the case of $D_3/D_2=1.075$ (Fig. 9(b))

In this case, the alternate blade stall is not observed in the entire range of flow rate. In spite of the alternate blade stall, asymmetric stall is observed at $\phi=0.050$ where the slope of the diffuser pressure performance curve Ψ_d is almost flat. In addition, the rotating stall with larger amplitude of the pressure fluctuation is observed at a flow rate lower than $\phi=0.050$. Unsteady pressure patterns at P_{2y} are shown in Fig. 11. From these figures, it was found that the pressure pattern rotates in the opposite direction to the impeller rotation, the number of stall cells is $N=2$, and the propagation speed ratio is $\Omega_s/\Omega=-0.007$. When we decrease the flow rate to less than $\phi=0.020$, the forward rotating stall (F.R.S.) that propagates in the same direction as the impeller rotation is observed. Unfortunately, the propagation speed ratio and the number of stall cells could not be specified due to the unsteadiness of the pressure fluctuation.

6.4 In the case of $D_3/D_2=1.100$ (Fig. 9(c))

Finally, we increase the impeller/diffuser clearance further to $D_3/D_2=1.100$. The rotating stalls are observed in a wide flow range. The cross spectra

Fig. 11 Unsteady pressure pattern of P_2 for $D_3/D_2=1.075$ at $\phi=0.050$. The pressure pattern with two cells rotates in the opposite direction of impeller rotation, showing the backward rotating stall

Fig. 12 Cross spectra between the pressure fluctuations at P_{21} and P_{22} , for $D_3/D_2=1.100$ at various flow rates. The components with low frequency 1~3 Hz are caused by the forward/backward rotating stall. The component with 11 Hz is caused by the acoustic resonance in the test facility with the Helmholtz resonance frequency.

between two pressure fluctuations measured in P_{21} and P_{22} for various flow rates are shown in Fig. 12. From the cross spectrum analysis, it was found that the components with frequency 1~3 Hz are caused by the rotating stalls. There is a phase difference between two pressure fluctuations at different peripheral locations, P_{21} and P_{22} . In addition, the component with frequency 11 Hz is also observed at $\phi=0.016\sim0.024$. This frequency is close to the Helmholtz resonance frequency (10 Hz) of the test facility, and there is no phase difference at each location at the diffuser inlet. Therefore, it is reasonable to consider that the pressure fluctuation with 11 Hz is caused by the acoustic resonance of the test facility. In Fig. 12, the amplitude of the pressure fluctuation caused by rotating stalls suddenly changes at $\phi=0.03$, where the pressure performance of the pump discharge Ψ drops as shown in Fig. 9(c). At a flow rate higher than $\phi=0.03$, the backward rotating stall with a small amplitude of

Table 2 Various stalls in the vaned diffuser depending on the impeller/diffuser clearance D_3/D_2 (number of stall cells N , propagation speed ratio Ω_s/Ω)

	$\phi < 0.03$	$\phi > 0.03$
$D_3/D_2=1.035$	N/A	A.B.S. ($N=4$, $\Omega_s/\Omega=0$) B.R.S. ($N=3$, $\Omega_s/\Omega=-0.008$)
$D_3/D_2=1.050$	N/A	A.B.S. ($N=4$, $\Omega_s/\Omega=0$) A.S. ($N=1$, $\Omega_s/\Omega=0$) B.R.S. ($N=3$, $\Omega_s/\Omega=-0.008$)
$D_3/D_2=1.075$	F.R.S. (N , Ω_s/Ω : Unknown)	B.R.S. ($N=2$, $\Omega_s/\Omega=-0.007$)
$D_3/D_2=1.100$	F.R.S. ($N=2$ or 3 , $\Omega_s/\Omega=0.008\sim0.02$)	B.R.S. ($N=3$, $\Omega_s/\Omega=-0.008$)

pressure fluctuation occurs, and the number of stall cells is $N=3$ and the propagation speed ratio $\Omega_s/\Omega=-0.008$. On the contrary, at a flow rate lower than $\phi=0.03$, the forward rotating stall with large amplitude of pressure fluctuation occurs. The number of stall cells and the propagation speed ratio vary depending on the flow rate. The number of stall cells is $N=2$ or 3 , and the propagation speed ratio $\Omega_s/\Omega=0.008\sim0.02$ in the present experiment.

6.5 Summary of the experiments

Table 2 shows the summary of the various diffuser stalls observed in the present experiments. From these results, we could conclude that the alternate blade stall occurs in the region where the slope of the diffuser pressure performance Ψ_d vs. flow rate ϕ curve is flat for the case with small impeller/diffuser clearance. The backward rotating stall is observed in all cases of a higher flow rate than $\phi=0.03$, in which the slope of $\Psi_d-\phi$ curve is slightly negative. The amplitude of the pressure fluctuation caused by the backward rotating stall is relatively small. In addition, the forward rotating stall with a large amplitude of pressure fluctuation is observed at a lower flow rate, in which the negative slope of the $\Psi_d-\phi$ curve is considerably steep. Moreover, the pressure performance Ψ at the pump discharge drops suddenly, when the forward rotating stall starts to occur.

Fig. 13 Comparison of linear stability analytical results with experimental ones for various impeller/diffuser clearances, (a) $D_3/D_2=1.035$, (b) $D_3/D_2=1.075$, (c) $D_3/D_2=1.100$

7. Analytical Method

In the present experiment, various kinds of flow instabilities in the vaned diffuser are observed; i.e., the forward/backward rotating stall, alternate blade stall, and asymmetric stall. To obtain a better understanding of these stalls, 2D linear stability analysis is performed. An actuator disk theory is applied, which was proposed by Tsujimoto et al.⁽¹⁶⁾ for the analysis of the rotordynamic fluid force on a whirling impeller in a vaned diffuser. The flow is assumed to be 2D, inviscid, and incompressible.

We assume the flow disturbance at the impeller inlet/outlet, diffuser inlet/outlet, and in the vaneless region between the impeller and diffuser as a circumferential propagating velocity pattern, that is $\Delta v \propto \exp\{j(\Omega_s t - N\theta)\}$, where Ω_s is the complex rotating stall angular velocity, and N is the number of stall cells.

To simplify the analysis, impeller and diffuser pressure performance curves are assumed as the quadratic functions of the flow rate, if total pressure loss coefficients are estimated by using the experimental results. Friction losses of the impeller and diffuser are assumed to be proportional to the square of the flow rate, and shock losses at the impeller/diffuser inlet are assumed to be generated due to the deviation of the flow angle from zero incidence⁽¹⁶⁾. In this

analysis, it is necessary that the slope of the pressure performance fits the experimental one as far as possible. To obtain the theoretical Euler head, slip factor is calculated by the singularity method with a finite number of blades. The pressure performance curves used for the present analysis are shown in Fig. 13 for comparison with experimental ones.

From these assumptions, the amplitudes of the unsteady velocities can be analyzed. When the amplitude of the disturbance is divergent in time (i.e., $\text{Imag}(\Omega_s) < 0$), we consider this to the rotating stall onset.

8. Comparison with Experimental Results

Pressure performance curves used in the analysis and the experiment are compared in the lower part of Fig. 13. Experimental values of eight channels are averaged, and fitted by quadratic curves as shown with dotted lines for reference. For the analysis, solid lines were applied. The minimum point of the diffuser pressure performance curve shifts to a higher flow rate on increasing the impeller/diffuser clearance. Therefore, it is possible to say that the unstable region of the diffuser rotating stall extends toward the higher flow rate.

The occurrence regions of the flow instabilities and number of stall cells in the analysis and experiment are compared in the top part of Fig. 13. In this

figure, analytical results show only the neutral stability points. The flow range lower than the neutral stability points represents the unstable region of the rotating stall in the analysis.

In the case of the smaller impeller/diffuser clearance with strong interaction, a backward rotating stall is also calculated in the analysis, although the amount of the propagation speed ratio is relatively larger than that in the experiment. In the case of $D_3/D_2=1.035$, rotating stalls in the analysis are all backward-propagation stalls, but the propagation direction of the rotating stalls changes from backward to forward on increasing the impeller/diffuser clearance. However, the propagation speed of the forward rotating stall is very low. This is because the interaction between impeller and diffuser decreases on increasing the clearance, and the diffuser rotating stall intrinsically with low propagation speed occurs easily. In the experiment, forward rotating stall starts to occur with increasing the clearance; this tendency agrees well with the analytical results.

All of the stalls observed in this experiment have a low propagation speed ratio. The impeller rotating stall with a high propagation speed ratio⁽⁹⁾ is not observed at all. This is because the slope of the impeller pressure performance Ψ_s vs. the flow rate ϕ curve is negative in the whole flow range. And the propagation speed considered in the present experiment is significantly smaller than that in the analysis, particularly in case of $D_3/D_2=1.035$. One of the reasons for this overestimation might be that the inertia length of the diffuser channel in the experiment is longer than that in the analysis, because the crossover and return vane are ignored in the present analysis.

9. Concluding Remarks

On increasing the clearance between the impeller and diffuser, the diffuser pressure performance Ψ_d changes and has a significant influence on the onset of various diffuser stalls, such as the backward/forward rotating stall, alternate blade stall, and asymmetric stall. The alternate blade stall is mainly observed in the region where the slope of the Ψ_d - ϕ curve is flat for the case with strong interaction between the impeller and the diffuser; i.e., D_3/D_2 is small. Backward rotating stall with a relatively small amplitude of the pressure fluctuation occurs at the flow rate larger than $\phi=0.03$, where the slope of Ψ_d - ϕ curve is slightly negative. For the case with weak impeller/diffuser interaction (i.e., larger D_3/D_2), the forward rotating stall with a large amplitude of pressure fluctuation occurs in the region with a steep negative slope of Ψ_d - ϕ curve. As a whole, diffuser flow instabilities are initiated around the flat slope region of Ψ_d - ϕ , and

develop in the negative slope region.

As a conclusion, the clearance between impeller and diffuser affects the diffuser pressure performance, and the slope of the diffuser pressure performance Ψ_d vs. flow rate ϕ curve influences the flow instability within the vaned diffuser. Not only the forward rotating stall that is well known, but also the other stalls, i.e., the alternate blade stall, asymmetric stall, and backward rotating stall, could occur depending on the diffuser pressure performance.

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