Effects of Diffuser Vane Geometry on Interaction Noise Generated from a Centrifugal Compressor

Yutaka OHTA Naho TAKEHARA Yasuhiko OKUTSU Eisuke OUTA
Department of Mechanical Engineering, Waseda University, 3-4-1 Okubo, Shinjuku, Tokyo 169-8555, Japan

The characteristics of interaction tone noise radiated from a centrifugal compressor with a vaned diffuser are discussed by experiments, including visualization techniques using the oil-film method. Research attention is paid to the leading edge geometries of the diffuser vanes that are deeply related to the generation mechanism of the interaction tone noise. The compressor-radiated noise can be reduced by several decibels by setting some clearances in both the hub and shroud surfaces of the diffuser wall along with some decline in the pressure-rise coefficient. Since the decline turned out to be caused by the flow impingement and also by the secondary flow within the diffuser passages, several new types of diffuser vane geometries which do not detract from both the performance and noise level are developed and utilized for the experiments. The presented diffuser vane geometries will offer a few basic guidelines for the diffuser vane design.

Keywords: centrifugal compressor, vaned diffuser, interaction noise, noise reduction.
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Introduction

Vaned diffusers are widely used for industrial centrifugal compressors because of their high pressure characteristics as compared with those of compressors with vaneless diffusers. However, one of the serious shortcomings of the vaned diffuser system is the extremely high noise level appearing as a discrete tone in the power spectrum of the radiated noise. Since the noise, which is called an interaction tone noise, is considered to be caused by direct impingement of the impeller-discharged flow on the diffuser vane, the leading edge geometries of the diffuser vane play an important role in discussing the noise generation mechanism. Despite the fact that the interaction is considered to be a clear and powerful noise source, surprisingly little research has been reported on the interaction tone noise[1 ~ 5]. Furthermore, the frequency of the interaction tone noise sometimes coincides with that of the blade-passing frequency components, abbreviated to BPF components, so the noise could dominate the overall noise level if sufficient care is not taken.

In the present paper, therefore, research attention is focused on the interaction tone noise and effects of diffuser vane geometries on the performance and noise level of the centrifugal compressor, which are investigated by experiments including flow visualization techniques. In the experiments, total pressure-rise coefficients and generated noise data are acquired precisely by making the flow coefficient into a parameter which pays attention to the height of the diffuse vane that is highly relevant to the interaction area of the tone noise. Axial positions of the diffuser vanes are adjusted by making some clearances in the hub and/or shroud surfaces of the diffuser parallel wall, and a flow structure within blade-to-blade passages is visualized by the oil-film method. Compressor-radiated tone noise can de reduced by several decibels by adjusting equal clearances in both the hub and shroud surfaces on the diffuser wall along with some decline in the pressure-rise coefficient. The decline turned out to be caused by the secondary flow which is a leakage getting through clearances and rotates within the diffuser passages. In order to reduce the interaction tone noise effectively while maintaining high compressor performance, a hooked diffuser vane is presented, by which the height of the vanes at only the 22% chord from the leading edge is contracted. Using the hooked vanes, pressure-rise characteristics can be improved by several percentage points with more than 5 dB of noise reduction attained at the maximum efficiency

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Yutaka OHTA: Professor

operation. Furthermore, a tapered diffuser vane which is an aerodynamically improved type of the hooked vane is designed and utilized for the experiments to verify the performance improvement and noise reduction.

Experimental Apparatus and Procedure

The tested compressor and noise-measuring system used in the experiments is shown in Fig.1. The compressor is a low-specific-speed centrifugal type and is designed based on a turbocharger for marine-use diesel engines. The inlet and outlet diameters of the unshrouded impeller are 248 and 328 mm, respectively. The number of main and splitter blades is 7 each. Essentially, two types of diffusers are used. One is a channel diffuser with log spiral at the leading edge, and the number of wedge-type vanes is 15. The vanes are located between two parallel diffuser walls 26.14 mm apart from each other. The other is a vaneless diffuser. Each diffuser has an identical meridional profile, as shown in Fig.1. By attaching different types of modified vanes on the vaneless diffuser, noise and performance measurements can be carried out under various conditions. The length of the inlet duct is 1.05 m, and a bellmouth is installed at the duct end. The exit of the diffuser is surrounded by a volute-type scroll collector with a large cross-sectional area. The specifications of the tested centrifugal compressor and dimensions of the impeller and vaned diffuser are listed in Table 1.

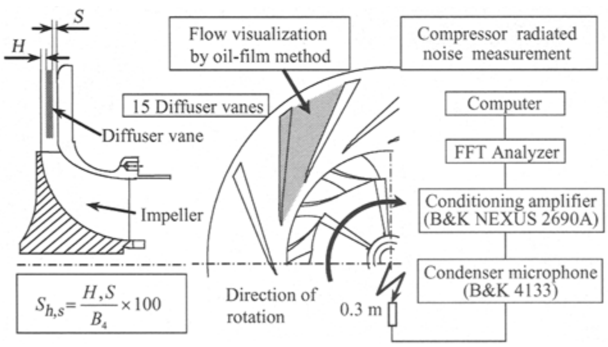


Fig.1 Experimental apparatus and measuring system

At the outset of the noise research, the compressor rotational speed is limited to 6000 r/min, where the wavelength of the fundamental BPF noise is considered to be long enough to presume that the dominant noise travels as a plane wave. The operation point is set by a butterfly valve installed at the outlet duct and the flow coefficient is changed in the restricted range between 0.11 and 0.28. The mass flow rate can be calculated by using an orifice flow meter and a thermo-couple installed at the outlet duct. The compressor is installed in an anechoic chamber, and the background noise level is sufficiently lower than the compressor noise level. A silencer is located in the outlet duct to block the wave reflection from the duct end.

The sound pressure level of the compressor-radiated noise is measured using a B&K 4133 condenser microphone at a location 0.3 m apart from the inlet bellmouth. Each of the analogue signals is transferred to a computer boarded with a 24-bit/133 MHz A/D converter and an FFT analyzer after amplifying with a B&K NEXUS 2690A conditioning amplifier. The analysis range of frequency is up to 10 kHz with a resolution of 3.125 Hz. The average number is 64 for each spectrum. Moreover, in order to investigate the flow field at the time of changing the diffuser vane geometry, the flow structure in the vicinity of the vane surface is visualized by using an oil film which contains mixed titanium dioxide with light oil and oleic acid. The colored oil-film method using mixed blue and red dyes with oil is especially adopted for the visualization of the detailed flow structure within the diffuser passage.

Table 1 Dimensions of tested compressor.

Tested centrifugal compressor		
Rotational speed	N	7000 r/min
Mass flow rate	G	1.64 kg/s
Pressure ratio	P_5/P_0	1.1
Impeller		
Number of blades (main + splitter)	Z	14 (7 + 7)
Inlet diameter	D_1	248 mm
Outlet diameter	D_2	328 mm
Exit blade width	B_2	26.14 mm
Diffuser		
Blade shape		Wedge
Number of vanes	V	15
Leading edge diameter	D_3	360 mm
Trailing edge diameter	D_4	559 mm
Diffuser width	B_4	19.55 mm

Experimental Results and Discussion

Characteristics of compressor tone noise

Typical power spectra of compressor-radiated noise with vaned and vaneless diffusers are presented in Fig.2 with the main results of this paper shown by Types A, B and C. Since waveforms of these results are very similar and are overlaid on the figure, it is very difficult to discriminate each other. Therefore, the peak noise levels of the fundamental blade-passing frequency noise, which are our most concern, are indicated by small arrows with A, B and C in the figure.

The fundamental frequency and harmonics of blade-passing frequency components appear most remarkably in the power spectra and dominate the overall noise level. The discrepancy level of the radiated noise between cases of vaned and vaneless diffusers is about 20 dB in almost all frequency ranges up to 10 kHz. Especially, the sound pressure of the 1st BPF noise, in the case of the vaned diffuser shown by a small arrow, exhibits an extremely high level as compared to that for the vaneless diffuser.

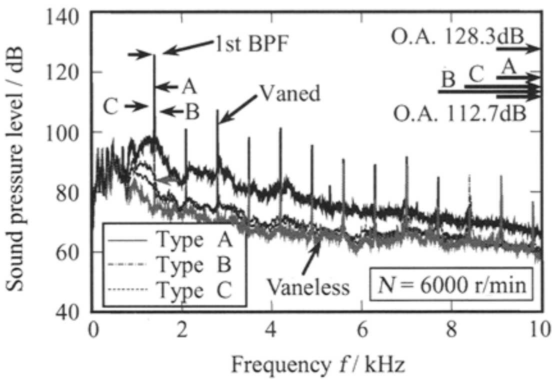


Fig.2 Typical power spectra of compressor noiise

According to previous research on axial flow compressor noise, the BPF noise is considered to consist of rotor noise and interaction noise[6]. The former is mainly caused by periodic pressure patterns induced by rotor rotation, and the latter is due to the direct impingement of the rotor-discharged wake on diffuser vanes[1]. The contribution of the rotor noise to the overall noise level is considered to be very slight, and almost all the discrete tones are generated by rotor-stator interaction. The m number of the pressure pattern, the so-called lobe pattern, generated by rotor-stator interaction, can be expressed as[7]:

$$m = nZ + kV, \quad n = 1,2,\cdots, \quad k = \cdots -1,0,1,\cdots \tag{1}$$

Similarly, in the case of the tested compressor, the interaction between impeller blades and diffuser vanes may be a powerful noise source, as one can recognize from the power spectrum shown in Fig.2. The expression for the smallest value of m in Eq. (1) gives -1 ($n=1$, $k=-1$), which is interpreted as a 1-lobe pattern, rotating in the opposite direction to that of the impeller, its speed being Z times the impeller rotational speed. Therefore, the frequency of the interaction tone noise coincides with that of the BPF noise.

In the experiments, the height of the diffuser vanes, which seldom attracted any attention previously, is chosen as a parameter, and the reduction methods of the interaction tone noise are examined. The overall noise level and the total pressure coefficient are measured precisely, changing the flow coefficient. The flow and total pressure coefficients used in the experiments are defined as follows:

$$\phi = \frac{Q}{\pi^2 D_2^2 B_2 N}, \quad \psi_T = \frac{P_T}{\rho \pi^2 D_2^2 N^2 / 2} \tag{2}$$

Where, N = rotational speed of the compressor (r/min), Q = volume flow rate (m^3/s), P_T = total pressure-rise (Pa), ρ = air density (kg/m^3).

Effects of diffuser vane height

The experimental results of the overall noise level and the compressor performance curves at the time of changing the diffuser vane height are indicated in Fig.3. The axial positions of the diffuser vanes are adjusted by making clearances in the hub and/or shroud surfaces of the diffuser wall. In the notation, 0% means the usual vaned diffuser without clearances, and $S_h = 5\%$, $S_s = 5\%$ denote vaned diffusers which have 5% clearances in the hub and the shroud surfaces of the diffuser wall, respectively.

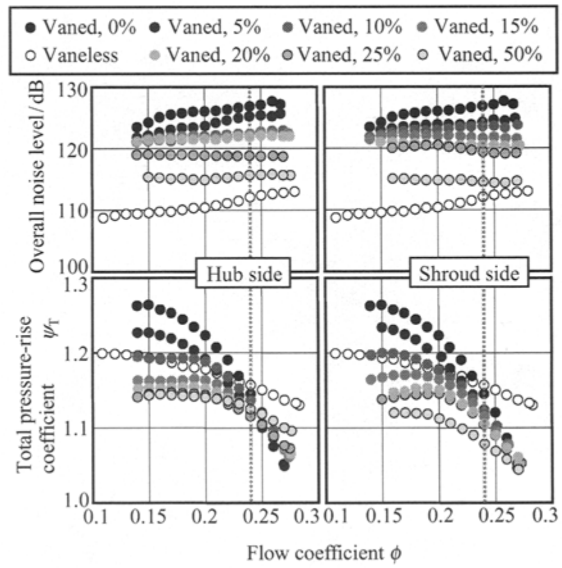


Fig.3 Effects of diffuser vane clearances on performance and noise of compressor (N = 6000 r/min)

When the vaned diffuser is used, compared with a vaneless diffuser, while high pressure-rise characteristics can be obtained, the operation flow range becomes narrow and the noise level also increases. In particular, if the flow coefficient ϕ is 0.24, the maximum efficiency point of the vaned diffuser operation, the overall noise level increases by about 15 dB. The following features are acquired from the results of this experiment:

(1) Both the overall noise level and the total pressure-rise coefficient declined with an increase of the vane clearances S_h and S_s. This tendency does not change in the cases of hub-side clearance S_h and shroud-side clearance S_s.

(2) At the time of $S_h = 5\%$, a total pressure-rise higher than the case of the vaneless diffuser installation is acquired, especially in a low-flow operation region. Moreover, a maximum noise reduction of 2.5 dB is possible. However, the performance decrement is remarkable close to the maximum efficiency point, and the operation flow range turns out to be very narrow under the influence of a single-sided clearance.

(3) The total pressure-rise in the range of $S_h, S_s = 15\%$ to 50% decays in the case of the vaneless diffuser installation. This is considered to be due to installed diffuser vanes not operating effectively but only acting as passage resistance.

The visualization results by the oil-film method at the time of setting some clearances in the hub side wall of the diffuser are shown in Fig.4. In the figure, the arrows on vanes express the inlet velocity vector entering into the vaned diffuser which are calculated from the impeller-discharged flow angle in consideration of the slip factor[8].

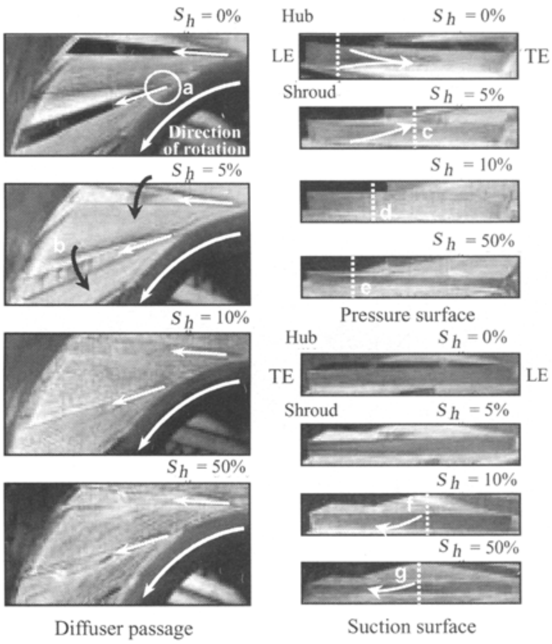


Fig.4 Visualization of diffuser flow field by oil-film method (Clearance on the Hub side, $\phi = 0.28$)

In the case of the vaned diffuser ($S_h = 0\%$), the angle of the impeller-discharged flow vector is almost in accordance with the diffuser inlet angle, and the flow separation has taken place from the leading edge, as indicated by **a** in the figure, in the pressure surface of the diffuser vane. In cases of $S_h = 5,10,50\%$, the difference between the diffuser inlet angle and the velocity flow vector is noticeable, and the separation region becomes small. Moreover, in the case of $S_h = 50\%$, the direction of a streamline on the diffuser passage and that on the clearance part mostly agree well, the circulation flow which enters directly into the clearance part from the impeller becoming dominant. The flow which enters into the clearance and leaks to adjacent passages can also be recognized from the visualization result of $S_h = 5\%$, shown by **b** in the figure. In the case of the vaned diffuser ($S_h = 0\%$), on the pressure surface, the separation flow which goes toward the mid-span comes about from both the hub and shroud surfaces of the diffuser. On the other hand, in cases of $S_h = 5,10,50\%$, separation flow is generated only from the shroud side of the diffuser and leaks to the next passages in the position in which it arrives at the hub side, as indicated by **c**, **d** and **e**. This tendency is recognized also in the suction surface of the diffuser vane, and the leakage flow shown by **f** and **g** in the figure can be observed.

As mentioned above, it was qualitatively confirmed from the visualization result by the oil-film method that the leakage flow between diffuser passages and the circulation flows in the diffuser are the main causes of the compressor performance decrement. Moreover, when clearance is provided only in the hub or shroud side of the diffuser vane, a non-uniform flow field is generated.

Effects of two-sided clearance

In order to cancel the non-uniform flow field generated by single-sided clearance of the diffuser vane, the influence of two-sided clearance is discussed.

The measurement results of the overall noise level and the compressor performance curves in cases the sum of the clearance width are 10 and 20% are shown in Fig.5. In this figure, the clearance width of both sides of a hub and a shroud is totaled, and is compared with the result of a single-sided clearance which has the same width. By setting the clearance equally on both sides, the compressor performance is improved by several percentage points, especially in a low-flow operation region. However, the decrement of the compressor performance near the best efficiency point $\phi = 0.24$ is remarkably large even in this case. With respect to the overall noise level, since the interaction area between the impeller-discharged flow and the diffuser vane is equal in this case, not so much change is recognized. Only about 2 dB of noise reduction is attained in a low-flow operation region where the compressor performance is improving. Moreover, as compared with the overall noise level at the time of vaned diffuser installation, noise reduction of about 3 to 5 dB is attained in the wide range of the flow coefficient, and a change in the diffuser vane height is found to be an effective way to reduce the interaction noise level.

The visualization result of the diffuser internal flow by the oil-film method in case the sum total of clearance width is 10% is shown in Fig.6. From the result of $S_h = 10\%$, disturbance cannot be recognized in the

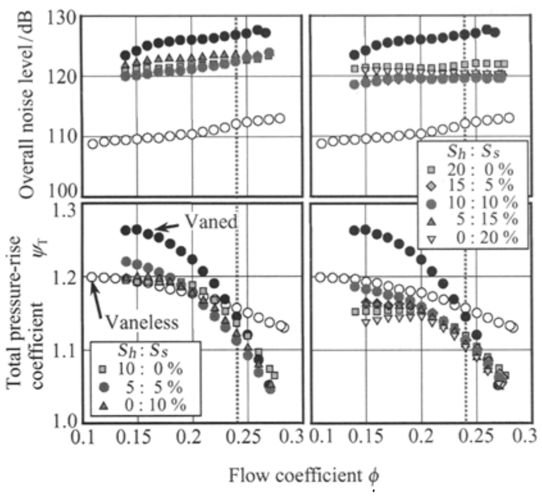


Fig.5 Effects of two-sided clearances on performance and noise of compressor (N = 6000 r/min)

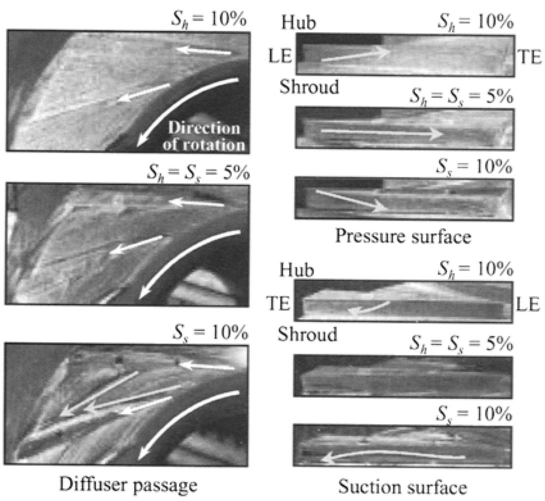


Fig.6 Visualization of diffuser flow field by oil-film method (Two-sided clearances, $\phi = 0.28$)

streamline in the diffuser passages between vanes. However, in the case of $S_s = 10\%$, a complicated secondary flow as well as the circulation flow which passes through the clearance exists. These flows are not the two-dimensional property that exists in the diffuser meridian plane —a non-uniform flow in the vane-height direction is also included. Accordingly, when the clearance is equally installed in the hub and the shroud side of the diffuser, the influence of the non-uniform flow referred to above can be reduced, and the compressor performance is improved.

Development of a hooked diffuser vane

As a result of setting some clearances in the hub and shroud side of the diffuser and performing measurements of the compressor performance and generated noise, it is thought that the causes of the performance decrement by the increase of the clearance width are due to two kinds of flow components shown below. One is the leakage flow which passes through the clearance and flows into the adjacent diffuser passage, and the other is the flow which rotates in the clearance part independently. The decline of the compressor performance caused by these secondary flows can be prevented to a certain extent by setting the clearance equally in both the hub and shroud surface of the diffuser. On the other hand, the sound pressure of the interaction tone noise can be controlled over the wide range of the flow coefficient by decreasing the interaction area, i.e., effective noise source, on the diffuser vane surface.

Consequently, in order to suppress a decline of the compressor performance by the secondary flow to the minimum extent and to reduce the overall noise level effectively, a new type of diffuser vane, called a hooked diffuser vane shown in Fig.7, is developed as the first-step trial. As shown in the figure, the height of the vane from the leading edge to the 22% chord is contracted, and the clearance is equally installed on both sides. The compressor performance curve and the overall noise level on the occasion of changing the diffuser vane height B_H at the leading edge are shown in Fig.8. The symbols A, B, C and D in the figure correspond to the visualization results shown in Fig. 9, respectively.

According to the experimental result, in cases of $B_H = 60\%$ or more, total pressure-rise characteristics higher than those of the vaneless diffuser installation are acquired, and the overall noise level is also decreased significantly. However, if the clearance width is enlarged too much, i.e., if B_H is less than 50%, the operation flow range will become narrow and the generated noise will also increase conversely. This is because, as one can see from the visualization result shown in Fig.9, the fluid loss generated by the flow impingement to the corner part of the hooked diffuser vane (shown by A in Fig.7) increases with the enlargement of the clearance. The flow is then curved greatly to the suction-surface of the diffuser vane and extends the separation region as a result (indicated by c and d). Moreover, the noise level also becomes remarkable by interaction with the corner

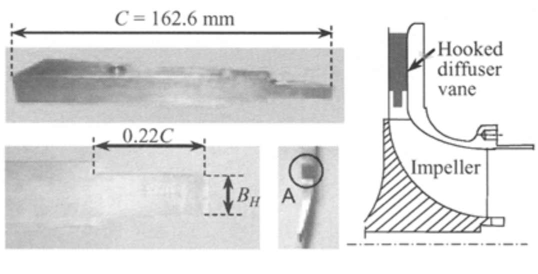


Fig.7 Hooked diffuser vane

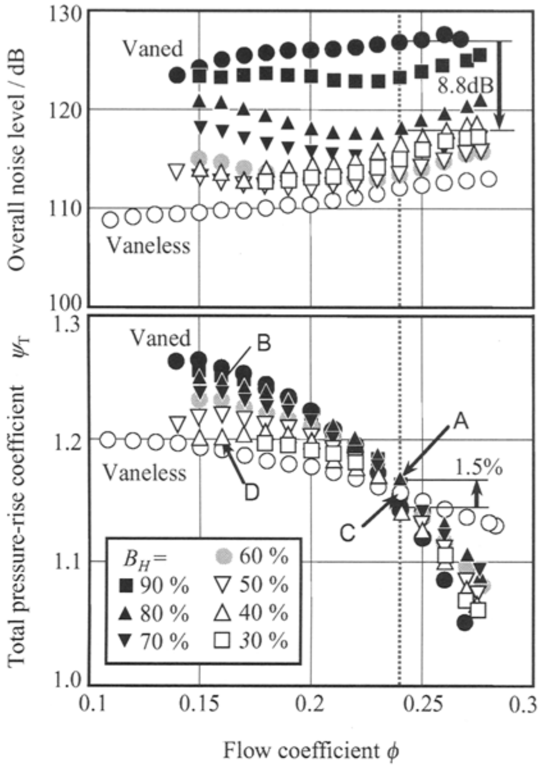


Fig.8 Effects of hooked diffuser vanes on performance and noise of compressor (V = 6000 r/min)

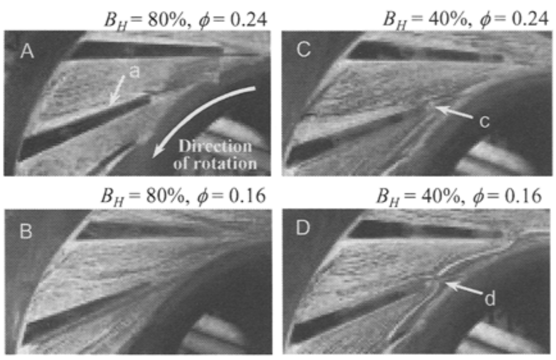


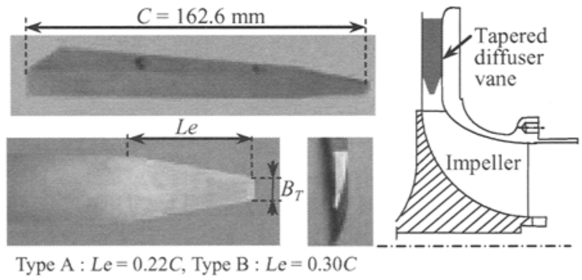
Fig.9 Visualization of diffuser flow field by oil-film method (Hooked vanes)

part A and the impeller-discharged flow.

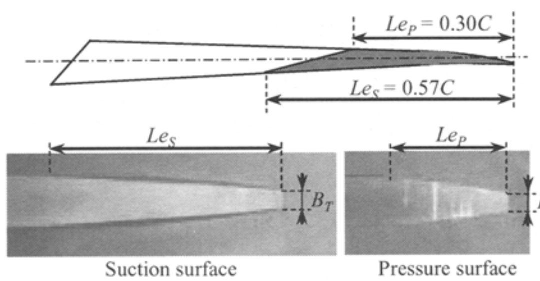
When the clearance width is set narrow (B_H = 80, 90%), the character of the performance curve changes to expand the compressor operation flow range, and shows pressure-rise characteristics higher than those of the vaned diffuser installation near the best efficiency point of ϕ = 0.24. As a result, in the case of B_H = 80% , about 1.5 percent of compressor performance improvement and noise reduction of about 9 dB are attained. The result of the oil-film method also shows that the flow conditions are favorable in general, although a small separation region is observed in the pressure-side surface of the vane (indicated by a in Fig.9). Moreover, also from the visualization result in the low flow region (ϕ = 0.16) shown by B in the figure, the flow field is almost favorable and it is proven that the high pressure-rise characteristic is acquired. As stated above, the compressor noise level can be reduced effectively by using the hooked diffuser vanes, thus suppressing the performance decrement to the minimum.

Proposition of tapered diffuser vanes

A new style of tapered diffuser vane is developed as a trial for reducing the loss induced by the impeller-discharged flow impingement to the corner part of the hooked diffuser vanes, and for improving the compressor performance more than the hooked vane installation. They are types A and B (two-dimensional tapered vanes), and the three-dimensional type C, as indicated in Fig.10. As for the shape of the 2-D tapered vanes (A and B), the vane height from the leading edge to the 22% and 30% chord respectively constitutes the tapered shape. On the other hand, the shape of the 3-D tapered vane (C) is asymmetric, by which the range up to the 30% chord from the leading edge on the pressure surface and up to the 57% chord on the suction surface has the tapered shape. This is based on the absolute discharge angle calculated from the velocity triangle at the impeller exit. The diffuser vane height B_T at the leading edge is experimented with by making it change in the 30 to 90% range, like the case of the hooked diffuser vanes. Only the results of B_T = 40% cases, which are considered to be the most effective for both the compressor performance and the overall noise level, are shown in Fig.11. The



(a) 2-D tapered diffuser vanes type A and B



(b) 3-D tapered diffuser vane type C

Fig.10 2-D and 3-D tapered diffuser vanes results obtained from this figure can be summarized as follows:

(1) If any type of tapered vane is used, in the whole of the experimented range of the flow coefficient, the total pressure-rise characteristic is acquired much more than in the case of the hooked vane installation and the overall noise level is also reduced. In particular, in the high flow region near the best efficiency point ($\phi = 0.24$), a pressure-rise considerably higher than that of the vaned diffuser installation is acquired.

(2) By extending the tapered portion from the leading edge from the 22% chord (type A) to the 30% chord (type B), the overall noise level can be reduced by about 3 to 5 dB in all flow ranges. However, the increment of the fluid loss resulting from an enlargement of the tapered portion reduces the compressor performance to some extent. This performance decrement is remarkable, especially in the low-flow region.

(3) The effect of the 3-D tapered vane (type C) is extremely outstanding. The improvement of the compressor performance up to 2.6% is attained at the best efficiency point by suppressing the performance decrement in the low-flow region to the minimum extent, and the amount of attenuation of the overall noise level also reaches 14.2 dB.

The visualization results using the colored oil-film method at the time of changing the flow coefficient are shown in Fig.12. Here, the colored oil film are utilized

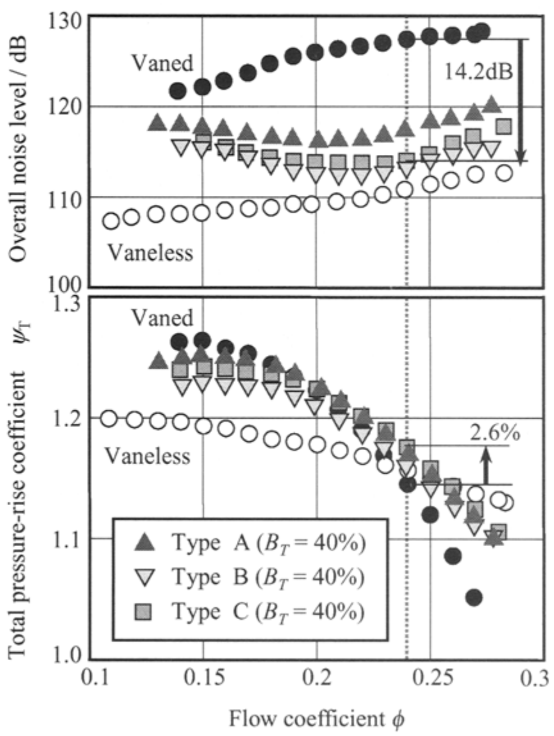


Fig.11 Effects of tapered diffuser vanes on performance and noise of compressor (N = 6000 r/min)

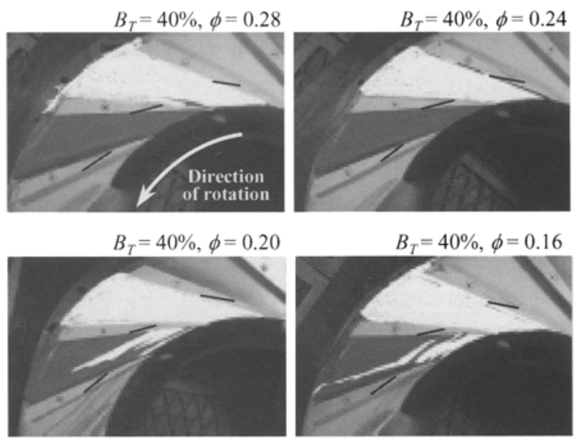


Fig.12 Visualization of diffuser flow Field by colored oil-film method (3-D tapered vanes Type C)

for the pressure and suction side of the diffuser vane passages, respectively. Moreover, the thin black line in the figure indicates the chordwise position where the tapered part from the leading edge ends. The features of the flow structure in the diffuser passages as shown below are obtained.

(1) The dark color oil film applied to the suction side of the diffuser vane passage passes through the clearance installed near the leading edge and reaches the pressure side passage in a high-flow region (ϕ = 0.28). In connection with this leakage flow, a small separation region is observed on the pressure surface of the vane.

(2) In the best efficiency point (ϕ = 0.24), there seems to be no flow which passes through the clearance part of the vane, and the flow within the diffuser passage streams along with the diffuser vane for the most part. This may be one of the proofs of the compressor having acquired high pressure-rise characteristics.

(3) In the low-flow regions, ϕ = 0.20 and 0.16, the influence of the leakage flow which rotates in the clearance part of the diffuser vane comes out strongly, and the light color oil film applied on the pressure side of the diffuser passage streams into the suction side. Moreover, the separation region on the pressure-side of the diffuser vane observed in the case of the hooked vane installation is also reduced. Generally, in such a low-flow region, since the absolute discharge angle at the impeller exit becomes small, the flow impinges on the diffuser vane and generates the large loss, called a shock loss. However, in the case of this experiment, the shock loss turns out to be reduced under the influence of the diffuser vane clearance installed near the leading edge, and then the high pressure-rise characteristic is acquired

The new style diffuser vane which possesses a 3-D shape and shortens the tapered part is expected to be manufactured and to perform quality assessment, as a future plan. Moreover, it will be necessary to conduct various experiments by which the chord length of the vane and solidity of the diffuser are changed.

Power spectra of the compressor noise

The power spectra of the compressor-radiated noise at the time of installing tapered vanes (types A, B, and C) are shown in Fig.2. In the case of the tested compressor, the sound pressure level of the interaction tone noise shows an extremely high level and dominates the overall noise characteristics, since the generating frequency is 1400 Hz (Z=14, V=15) and is in agreement with that of the fundamental BPF noise. By installation of the tapered diffuser vane, a considerable reduction of 10 to 20 dB is possible for the sound pressure level of the interaction tone noise, and the effectiveness of our method is confirmed. Moreover, especially when tapered vanes B and C are installed, not only the sound pressure level of the discrete component but also that of the broadband noise can be reduced by more than 10 dB in all frequency ranges up to 10 kHz. It can be presumed that the disturbed flow generated by the impeller-discharged flow impingement to the diffuser vane has attenuated effectively under the influence of the vane clearance. However, much more detailed measurements or CFD studies on the diffuser flow field are still required.

Conclusions

The compressor performance as well as the radiated noise level was experimentally investigated by changing various kinds of diffuser vane geometries in order to reduce the interaction tone noise which is generated by the interaction between the impeller-discharged flow and the diffuser vanes. The presented diffuser vane geometries will supply some basic guidelines for diffuser vane design. The findings can be summarized as follows:

(1) By setting some clearances in the axial direction of the diffuser vane and making the interaction area small, the discrete interaction noise can certainly be reduced. The compressor will become more efficient and have low noise when the clearance is equally installed in the hub and shroud surface of the diffuser passage. The decline of the compressor performance by enlargement of the clearance results from the secondary flow which rotates in the clearance part as well as from the leakage flow to the adjacent diffuser passage.

(2) The hooked type diffuser vane manufactured in order to suppress the secondary flow within the diffuser passage was able to improve the compressor performance at the best efficiency point, and was also able to reduce the noise level effectively. Furthermore, the tapered diffuser vanes which were processed near the leading edge into tapered form indicated much greater characteristics in both compressor performance and noise level. The improvement of the compressor performance up to 2.6%, and the attenuation of the overall noise level (14.2 dB) were attained at the best efficiency point. The presented tapered diffuser vane turned out to be effective not only in discrete tone noise reduction, such as BPF components, but also in broadband noise reduction.

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