The effect of unsteady vortex behavior on noise characteristics in a centrifugal compressor

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Abstract. The effect of diffuser width on the noise level of a centrifugal compressor equipped with vaneless diffusers was investigated through experiments and numerical techniques. In this regard, two diffusers with different widths were installed in the compressor. In case the wider diffuser was used, there was the bandwidth (800 to 900 Hz) noise and pressure fluctuation which was closed to 1/2 BPF (700 Hz). This fluctuation occurred because the leakage vortices were of two different frequencies. Type 1 vortices are the ones that were discharged from the main blade and covered Passage 1. Type 2 vortices had the same length as Type 1, but they were discharged from the splitter blade. Type 3 vortices covered both Passages 1 and 2; these were discharged from the main blade. Type 3 vortices were generated predominantly at 270°, and their frequency was approximately 700 Hz at 1/2 BPF frequency. The former two types of vortices occurred occasionally, and their frequency was approximately 1400 Hz. Type 3 vortices grew with the wider diffuser because of the higher pressure-recovery, which caused the serious loading to the impeller blades. The broadband noise fluctuation occurred by the coexistence of the two frequency vortices.

1. Introduction

A centrifugal compressor consists of an impeller, a diffuser, and a volute, in general. The compressor utilizes the centrifugal force caused by the difference between radii of the impeller inlet and outlet for compression; thus, it is adopted in various applications because of the high pressure ratio and the small size. For example, it is used in chillers, aircrafts, and vehicles. In a centrifugal compressor, a vaneless diffuser is often employed because of its wider steady range compared to a vaned diffuser. To expand the application and improve the performance of a centrifugal compressor, it is required to reduce its noise. Previous studies [1, 2] have mainly focused on improving the performance of centrifugal compressor; however, currently, studies are being conducted to reduce the noise.

Several studies have been conducted on the noise characteristics of turbo machinery [3, 4]. Raitor and Neise [5] and Lee et al. [6] investigated the characteristics of inlet and outlet duct noises of impellers experimentally. According to their research, the blade passing frequency (BPF) causes the outlet duct noise, the instability intensity and circumferential flow fluctuations of the impeller during normal operation are closely related. Ohta et al. [7] reported that in a centrifugal blower, the BPF noise is not influenced by the cutoff geometries; however, it is largely influenced by the cutoff clearance.

Nevertheless, a few studies are available on the increase in broadband noise in centrifugal compressors. The objective of this study is to investigate the increase in the bandwidth of noise of a centrifugal compressor by revealing the relationship between the noise and unsteady behavior of vortex produced by the impeller.
2. Experimental apparatus and procedure

2.1. Experimental apparatus
In this study, a low-specific-speed centrifugal compressor was tested. The dimensions of the compressor are presented in Table 1, and the noise and pressure measurement systems are illustrated in Figures 1 and 2. As shown in Table 1, an open-shroud type impeller with seven main and splitter blades was tested. Its inlet and outlet diameters were 248 and 328 mm, respectively. Vaneless diffusers with two different widths (diffuser width, $B_{Dif} = 19.55$ and 26.14 mm, respectively) were used in the experiments.

Table 1. Dimensions of the tested compressor.

| Tested Centrifugal Compressor |  |
|-------------------------------|--|
| Rotational Speed $N$          | 6000 min$^{-1}$ |
| Mass Flow Rate $G$            | 1.64 kg/s |
| Pressure Ratio $P_5/P_1$      | 1.1 |

| Impeller |  |
|----------|--|
| Number of Blades (Main + Splitter) | 14 |
| Inlet Diameter $D_1$ | 248 mm |
| Outlet Diameter $D_2$ | 328 mm |
| Exit Blade Width $B_2$ | 26.14 mm |

| Diffuser |  |
|----------|--|
| Inlet Diameter $D_3$ | 360 mm |
| Outlet Diameter $D_4$ | 559 mm |
| Diffuser Width $B_{Dif}$ | 26.14 mm |

2.2. Measuring method
The rotational speed ($N$) of the compressor was set to 6000 min$^{-1}$. The operation point was set by using a butterfly valve installed at the outlet duct. The mass flow rates were calculated by using an orifice flow meter and a thermocouple installed at the outlet duct end. The pressure-rise was measured by using a differential pressure transducer mounted at the entrance of the outlet duct. The configuration of...
revolution and pressure measuring systems is shown in figure 2. The pressure transducers (Kulite XCQ-062) were installed at the diffuser exit (D.E.) on the shroud wall. As shown in figure 3, three different circumferential positions were chosen as the pressure measuring points on the shroud wall. The sound pressure level of the noise radiated from the compressor was measured by using a B&K 4133 condenser microphone located at a distance of 0.3 m away from the inlet bellmouth. All the analog signals were transferred to a computer equipped with a 24-bit/133 MHz A/D converter and a fast Fourier transform (FFT) analyzer after being amplified in a B&K NEXUS 2690A conditioning amplifier. The test compressor was installed in an anechoic chamber so that the background noise level is sufficiently lower than the compressor noise level.

3. Computational procedure
3.1. Governing equations
The simulations were conducted using the in-house computational fluid dynamics (CFD) code, which has been validated for various turbomachinery flows. This code solves the governing continuity equation, a three-dimensional compressible Navier-Stokes equation, an energy equation, and an ideal gas state equation. The convective flux was evaluated by using the flux difference splitting (FDS), which was extended to the third order by using the monotonic upwind scheme for conservative laws (MUSCL) interpolation. The viscous flux was determined as a second-order central difference by using Gauss’ theorem. The matrix free Gauss-Seidel (MFGS) implicit algorithm was employed for the time integration. The period of one rotor revolution was divided into 20,000 time steps. The detached eddy simulation (DES) approach, which is a hybrid scheme involving large eddy simulation (LES) and Reynolds-averaged Navier-Stokes equations (RANS), was employed in the turbulence modeling. The DES approach is based on the shear stress transport (SST) $k-\omega$ turbulence model. This model incorporates a dependency on the local turbulence length scale. The model constant ($C_{DES}$) was set to the value recommended by Strelets et al [8]. Both the Coriolis and centrifugal forces were considered as the inertial force terms in the relative coordinate system. This DES code was able to transform the RANS-mode in regions near the wall, and the LES-mode inside the fluid volume appropriately.

3.2. Computational domains
The computational domains applied in the numerical simulations are illustrated in figure 4. The grid system included 14 impeller passages, diffuser passages, and a volute. The computational domain was divided into four regions: moving impeller, stationary diffuser, moving tip clearance regions, and stationary volute. For the entire domain including the tip-clearance region, a mesh was generated by using the multiblock structured grid, which was generated by using AutoGrid5 and IGG ver9.1 (NUMECA International, Belgium). The impeller (including clearance region) and vaneless diffuser systems had 32.0 and 11.0 million cells, respectively. The volute region had 22.0 million cells. In total, the computational grid had 65.0 million cells. The grid had 100 cells in the passage along the height from the hub to the casing; among these, 24 were included in the clearance zone. The cell width at the

![Figure 4. Overview of the computational grid.](image-url)
walls was 0.1 μm, corresponding to a $y^+$ parameter, which was approximately equal to one along all solid surfaces. The inlet boundary was set at 2.1 times the impeller inlet diameter upstream from the impeller inlet. The outlet boundary was set at 7.0 times the casing diameter downstream from the axis of rotation.

3.3. Boundary conditions
At the inflow boundary, the total pressure and temperature were fixed, while at the outflow boundary, the mass flow rates were fixed. Across a sliding boundary that separates the moving impeller and stationary diffuser frames, the most recent data on one side were interpolated to obtain the data for the opposite side by using a sliding mesh for unsteady simulation. Nonslip and adiabatic conditions were adopted for the wall conditions.

4. Results and discussion
4.1. Performance and noise characteristics
The experimental and numerical results of the compressor performance are shown in figure 5. The numerical total pressure-rise characteristics were obtained from the unsteady DES and steady RANS analyses. The flow and total pressure rise coefficients are defined as follows:

$$\phi = \frac{Q}{\pi^2 D_z^2 B_z (N/60)}$$

$$\Psi_T = \frac{\Delta P_t}{\rho \pi^2 D_z^2 (N/60)^2 / 2}$$

The unsteady simulation was conducted at the design operating point ($\phi = 0.24$) and the steady simulations were conducted at four operating points from $\phi = 0.24$ (design point) to $\phi = 0.10$. Both the steady RANS analysis results and time-averaged results of the unsteady DES analysis were in good agreement with the measured results, which were obtained by using the two different widths of vaneless diffusers. To investigate the influence of diffuser width on the noise level, the compressor noise characteristics were measured by using a condenser microphone at the inlet bellmouth. The fast Fourier transformation results of the noise characteristics of the two diffusers with different widths are shown in figure 6. This figure shows that the noise level of the wider diffuser ($B_{Dif} = 26.14$ mm, called Diffuser A) around a frequency of 900 Hz was broadly higher than that of the narrower diffuser ($B_{Dif} = 19.55$ mm, called Diffuser B).

![Figure 5. Compressor performance.](image1)

![Figure 6. Noise characteristics.](image2)
4.2. Experimental and numerical investigation of the cause of the bandwidth noise increase

The pressure fluctuation level obtained from the measurement and analysis are shown in figure 7. In the experiment, the static pressure was measured at 1-D.E., 2-D.E., and 3-D.E. on the shroud wall. Furthermore, the magnitude of pressure fluctuation (900 Hz) was visualized at mid span from the numerical results. Both the experiment and CFD results indicated that Diffuser A led to a higher pressure fluctuation level in the diffuser passages compared to Diffuser B. This tendency was particularly evident in the regions before the cutoff (from 200° through 320°).

![Figure 7. Pressure fluctuation level obtained from experiment and analysis.](image)

Figure 8 shows the contour of mass flow fluctuations at the impeller exit within the full blade impeller passages for Diffusers A and B. According to this figure, the mass flow decreases before each impeller passage reaches the tongue. Especially, the magnitude of decrease in mass flow for Diffuser A was larger than that for Diffuser B. The span-wise distribution of the radial velocity at the impeller exit is shown in figure 9. A large wake area I and small wake area II were formed around the shroud side of impeller exit across the volute tongue. Based on the results, it is evident that a larger wake, as seen for Diffuser A, led to a substantial reduction in the mass flow at the impeller exit, before each passage reached the cutoff.

![Figure 8. Mass flow fluctuation at the impeller exit.](image)
To clarify the difference between the internal flow of the main and splitter blades, the impeller passages were assigned as shown in figure 10(a): Passage 1 is located between the suction surface side of main blade and the pressure surface side of splitter blade, and Passage 2 is located between the suction surface side of splitter blade and the pressure surface side of main blade. At the impeller exit, the mass flow of Passage 1 was always lower than that of Passage 2 for both diffusers as shown in figure 10(b). In addition, the magnitude of mass flow fluctuation at Passage 1 was larger than that at Passage 2. The distributions of radial velocity at the impeller exit (90° and 270° from the cutoff) for Passages 1 and 2 are shown in figure 11. Reverse flow was observed at the impeller exit in Passage 1, and it was more pronounced at 270°. A wider reverse flow was observed for Diffuser A when compared with Diffuser B near the shroud side. Figures 10 and 11 indicate that such low mass flow through Passage 1 was caused by the leakage vortices formed near the shroud side, thereby resulting in a reverse flow. A wider diffuser inlet area demanded a higher pressure recovery, thereby increasing the blade loading, which stimulated the growth of leakage vortices.

**Figure 9.** Radial velocity distribution at the impeller exit.

**Figure 10.** (a) Passage number, and (b) mass flow fluctuations in Passages 1 and 2.

**Figure 11.** Radial velocity distribution of Passages 1 and 2 at the two circumferential positions.
The instantaneous vortical structures formed in the impeller exit area at 90° and 270° for Diffuser A are illustrated in Figure 12. According to the results, massive separation vortices that cover the two blades (main and splitter) occurred at 270°, and they were discharged from the main blade. In contrast, large vortices did not occur at 90°, instead small-scale vortices were intermittently discharged from the main and splitter blades. This comparison revealed that the massive separation vortices at 270° led to the reverse flow and pressure fluctuations in the impeller exit area.

It is assumed that the existence of three major types of vortices contributed to the increase in the bandwidth of noise and pressure fluctuations from 800 Hz to 900 Hz. Type 1 vortices are the ones that were discharged from the main blade and covered Passage 1. Type 2 vortices had the same length as Type 1, but they were discharged from the splitter blade. Type 3 vortices covered both Passages 1 and 2; these were discharged from the main blade and these were generated predominantly at 270°. The vortices of Type 1 and 2 attach and rotate with the each blade so the rotating speed (frequency) is almost same as the impeller rotating speed (1stBPF : 1400 Hz). The vortices of Type 3 attach to the blade as same as Type 1 and 2, however those cover the two impeller passages (attach to the main blade and cover both the passages of main blade and splitter blade). The rotating speed of the vortices of Type 3 are same as the impeller rotating speed, but those emit from half numbers of the impeller blades so the frequency of fluctuation caused by Type 3 vortices are also approximately one-half of the frequency of 1stBPF fluctuation (1/2BPF : 700 Hz). The former two types of vortices are smaller than that of Type 3 vortices. Consequently, this frequency difference caused the broadband-frequency noise enlargement nearby 1/2 BPF.

Figure 13(b) shows the instantaneous vortical structures formed at the impeller exit area at 270° for Diffuser B. The massive vortical structures were not observed for Diffuser B; however, minor leakage vortices could be observed. In this case, because of the low blade loading and narrow diffuser width, the growth of leakage vortices was suppressed. Thus, the noise level of Diffuser B from 800–900 Hz was lower than that of Diffuser A.
5. Conclusion
The effect of unsteady vortex behavior on noise characteristics of a centrifugal compressor with a vaneless diffuser was investigated through experiments and CFD analysis.

The results can be summarized as follows:

(1) A bandwidth fluctuation (from 800 Hz to 900 Hz) similar to the noise raising was observed in the pressure measurement of experiment and in the result of CFD analysis for Diffuser A (wider diffuser, 26.14 mm). Specifically, this fluctuation was large between $270^\circ$ and the tongue. However, this tendency was not seen for Diffuser B (narrower diffuser, 19.55 mm).

(2) The wake and reverse flow increased near the shroud side of impeller exit before the impeller passages reached the volute tongue; this phenomenon instigated the reduction in mass flow. In the case of Diffuser A, there was a larger reverse flow and a substantial decrease in mass flow when compared with Diffuser B.

(3) Decrease in mass flow was caused by the leakage vortices around the impeller exit, and these vortices were predominantly generated for Diffuser A. Because a wider diffuser passage demands a larger pressure recovery, it resulted in an increased blade loading. Moreover, there were three types of vortices and their frequency difference led to the increase in bandwidth of noise.

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