NOISE REDUCTION IN CENTRIFUGAL COMPRESSORS BY USING QUARTER WAVELENGTH RESONATOR

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ABSTRACT
Centrifugal compressors generate very strong noise, which is typically dominated by the blade passing frequency and its higher harmonics. The high level of noise is not only very disturbing to the worker in factory but also people living close to the installation site. The common solution for reducing the noise level is to use external noise control measures such as sound enclosures and insulations of the compressors. The objective of this paper is to recommend noise control method that is to install quarter wavelength resonators, to reduce noise radiation from centrifugal compressors. In order to achieve this objective, (1) the optimal design parameter study for quarter wavelength resonators installed at the scroll exits, and (2) the experimental verification for designed quarter wavelength resonators are conducted. By installing the quarter wavelength resonators, reducing overall and BPF component sound pressure level up to 3.5dB and 8.3dB can be achieved.

INTRODUCTION
Air compressors are widely used to increase potential energy in pressurized air. Among several types of compressors, a centrifugal compressor is utilized to supply compressed air for pneumatic tools in manufacturing factories and other applications.

In order to define noise generation mechanism and reduce radiation noise of compressor, numerous studies have been performed. An experimental study is described to explore the dominant sound generation mechanisms of the spectral components governing the overall noise level of centrifugal compressors [1]. Numerical studies are performed to understand the generation mechanism of sound and to develop a prediction method for analysis and optimization of aerodynamic noise in a centrifugal compressor [2]–[4]. Improvement of aerodynamic performance and noise reduction of centrifugal compressor are studied by experiments and visualization techniques [5], [6].

Most of previous work focused on estimating radiation noise from a compressor or optimizing impellers and diffusers to reduce the noise generation (i.e., controlling noise sources). By using these results, it is hard to apply to exist systems and cost a lot since it require to change the current impellers or diffusers. Therefore, it will be only applicable to next generation compressors. For this reason, in this research, it is focused on controlling noise method which install quarter-wavelength resonators for reducing the radiation noise without changing exist compressor core systems. Quarter wavelength resonators can be very effective means to reduce a targeted frequency. This is great advantage for reducing the noise of centrifugal compressor because centrifugal compressors generate very strong noise, which is dominated by the blade passing frequency. This can be achieved by designing the resonators so that their maximum sound attenuation occurs around the blade passing frequency.

The objective of this paper is to recommend noise control method that is to install quarter wavelength resonators, to reduce noise radiation from centrifugal compressors. In order to achieve this objective, (1) the optimal design parameter study for the quarter wavelength resonators installed at the scroll exits, and (2) the experimental verification for designed quarter wavelength resonator are conducted

QUARTER WAVELENGTH RESONATOR
A quarter wavelength resonator that can be used to reduce tonal noise components is usually installed as a side branch in a duct system. The quarter wavelength resonator with a constant cross-sectional area and a rigid end is shown in Figure 1. An acoustic resonance occurs when the depth of the resonator (l) is one quarter of the wavelength at the excitation frequency (f) under the speed of sound (Co).

\[ f = \frac{C_0}{4l} \]  

(1)
The acoustic wave reflected from the rigid end of the resonator is out-of-phase with the incident wave to the resonator, generating a standing wave whose the length is corresponding to the quarter wavelength, change the acoustic impedance, and thus reduce the amplitude of the transmitted excitation waves.

Figure 1. Typical configuration of quarter-wavelength resonator installed on duct

OPTIMAL DESIGN

In order to obtain the optimal design parameters of the quarter wavelength resonators, a 3-D finite element model is built in the commercial software package, COMSOL Multiphysics – Acoustics Module. Figure 2 is a schematic of COMSOL model.

Figure 2. Schematic of COMSOL Model

Geometry is imported into COMSOL model. Then, the entire domain is set as air of which the pressure and density are corresponding to the specific operation condition to be analysed. At the inlet boundary, a plane wave excitation at the BPF is applied. Non-reflecting condition is set at the outlet boundary. The maximum meshing size is set as on eighth of the excitation wavelength at the BPF.

(1) Design conditions

It is known that centrifugal compressors generate very strong noise, which is dominated by the blade passing frequency (BPF). Therefore, for the design of the quarter wavelength resonators to reduce the noise radiation from compressors, design conditions are used to identify the BPF of the compressor. Table 1 presents the operation conditions of compressor test rig of Hanwha Techwin.

<table>
<thead>
<tr>
<th>Mass Flow Rate [kg/s]</th>
<th>Mean Flow [m/s]</th>
<th>Impeller Exit Pressure [kPa]</th>
<th>Temperature [K]</th>
<th>1st BPF [Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.26</td>
<td>278.8</td>
<td>189.1</td>
<td>264.7</td>
<td>377.1</td>
</tr>
</tbody>
</table>

(2) Numerical Optimization

The design parameters of a single quarter wavelength resonator are diameter (d) and depth (l) of each resonator hole. In addition, the geometric design parameters of multiple resonator holes are axial hole spacing (ΔL), the circumferential angular hole spacing (Δθ), and the number of axial hole layers (N). Figure 3 shows the aforementioned design parameters.

Figure 3. Quarter-wavelength resonators Design parameters

In order to obtain the optimal design parameters, finite element analysis model is built in a commercial software package, COMSOL Multiphysics–Acoustics Module, and transmission coefficient (TC) is used to indicate the transmitted sound power reduction in a Decibel (dB) scale: i.e.,

\[
TC = 10 \times \log \left( \frac{I_{Trans}}{I_{Incident}} \right) \quad (2)
\]

Where, \( I_{Trans} \) represents the spatially-averaged sound intensity of the transmitted wave and \( I_{Incident} \) is the spatially-averaged, incident sound intensity under the given operation condition.

A. Resonator depth and diameter

The first step of optimization procedure involves obtaining the optimal combination of the resonator’s depth and diameter. For the compressor operation condition described in Table 1, Figure 4 shows the predicted TC results as a function of the single resonator’s depth and diameter. The white line in Figure 4 indicates effective quarter wavelength curve. The blue area is maximum value of TC.

Here, the resonator depth(l) is chosen as 6.5mm, and the diameter(d) as 7mm. This selection is based on the shape of the blue area in Figure 4 where the absolute value of the TC reaches the maximum and the range of the optimal depth is getting narrower with a smaller diameter. Thus, the diameter should not be too small to avoid the acoustic
performance degradation induced by any small machining errors in the hole dimensions.

**Figure 4. Transmission coefficients[dB] of single resonator hole**

**B. Circumferential Spacing**

After the selection of the optimal resonator depth and diameter, the circumferential spacing (Δθ) is found for a single-layered holes. These holes are assumed to be equally-spaced in the circumferential direction. Figure 5 shows the numerical results of the TC as the function of the circumferential angular hole spacing with a single layer (N=1). From the Figure 5, it is shown that the TC reaches the maximum value at angular hole spacing (Δθ) 18° for operation condition.

**Figure 5. Transmission coefficients with respect to circumferential spacing**

**Table 2. Optimal axial spacing and number of axial layers**

<table>
<thead>
<tr>
<th># of Axial hole layers (N)</th>
<th>Operation Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>5 layers</td>
<td>Max. TC</td>
</tr>
<tr>
<td>6 layers</td>
<td>6.92</td>
</tr>
<tr>
<td>7 layers</td>
<td>9.85</td>
</tr>
<tr>
<td>5 layers</td>
<td>38.5</td>
</tr>
<tr>
<td>6 layers</td>
<td>39.0</td>
</tr>
<tr>
<td>7 layers</td>
<td>39.0</td>
</tr>
</tbody>
</table>

From the Figure 6 and Table 2, the simulation results show the maximum noise reduction can be achieved at the same axial spacing (ΔL) about 39mm regardless of the number of the axial hole layers. Therefore, once the axial spacing is determined, the optimal number of axial hole layers can be determined by noise reduction requirement as well as the total axial space limitation. In this study, the number of the axial hole layers (N) is selected of 6 layers.

For the operation conditions (i.e., the first BPF is 11,864Hz and the sound speed in the duct is 408.3m/s), the identified design parameters are 7mm in each hole diameter (d=7mm), 6.5mm in each hole depth (l=6.5mm), 18° in the circumferential angular spacing between two adjacent holes (Δθ=18°), and 39mm in the axial spacing (ΔL=39mm) to achieve the maximum noise reduction.

**EXPERIMENT**

After the optimal design of quarter wavelength resonators, prototype hardware was manufactured for experimental testing. The manufactured prototype is shown in Figure 7.

The noise reduction performance of the real resonators can vary due to the manufacturing tolerances. The tolerances can result in negative effects on the real noise reduction performance. However, for the hole diameter/depth, and circumferential angular spacing (Δθ), the performance variation due to the manufacturing tolerances can be neglected, since the performance variations are relatively small for their small dimensional changes, which can be seen from Figure 4 and Figure 5. For axial spacing (ΔL), the tolerance of ±0.5mm have approximately 25~60% variation on the noise reduction performance. For example, from Figure 6, for the 6 layer case, the best noise reduction can be
achieved at $\Delta L = 39.0\text{mm}$, which leads to 9.8dB noise reduction. However, at $\Delta L = 38.5\text{mm}$, the noise reduction is 7.4dB, and for $\Delta L = 39.5\text{mm}$, it is 4.0dB. The reason for the large variation of TC in small axial spacing ($\Delta L$) differences is because the wavelength under the given operation condition is short.

Table 3 shows the result comparison between the design and manufactured quarter wavelength resonators.

<table>
<thead>
<tr>
<th>Design Parameter</th>
<th>Design</th>
<th>Manufacture (Average)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hole diameter ($d$)</td>
<td>7mm</td>
<td>6.85mm</td>
</tr>
<tr>
<td>Hole depth ($l$)</td>
<td>6.5mm</td>
<td>6.5mm</td>
</tr>
<tr>
<td>Circumferential angular spacing ($\Delta \theta$)</td>
<td>18°</td>
<td>18°</td>
</tr>
<tr>
<td>Axial spacing ($\Delta L$)</td>
<td>39mm</td>
<td>38.8mm</td>
</tr>
<tr>
<td># of layers ($N$)</td>
<td>6</td>
<td>6</td>
</tr>
</tbody>
</table>

Quarter wavelength resonators are installed at the downstream of the single stage compressor. Figure 8 shows the installation with and without resonators.

In order to minimize the fluctuation effect of radiated sound from the compressor, 5 minutes averaged data are used during measurements. The bandwidth is set as 20 kHz, with 75% overlap. The number of FFT is 6400, and the number of average is 4797. The measurements are conducted at three measuring points around the compressor facilities. The locations of the measuring points can be seen in Figure 9.

One microphone, positioned 1m away from the suction pipe, was measured the sound pressure level on the suction side. Two microphones were positioned 1m away from the compressor casing to measure the sound pressure level near the compressor housing and discharge side.

Figure 7. Manufacturing quarter wavelength resonator

Table 3 shows the result comparison between the design and manufactured quarter wavelength resonators.

Table 3. Result comparison between design and manufacturing

RESULTS AND DISCUSSION

In order to apply the optimal design studies and determine the noise reduction for a centrifugal compressor, measurements were carried out around the compressor facilities.

Acoustic data were first acquired on compressor setup without resonators installed. Then, resonators were installed and tests were repeated. Comparison of sound pressure level could be made at each point. In addition to acoustic data, efficiency and head coefficient were measured at every test. The performance measurements were used to determine whether the resonators had any impact on the compressor aerodynamics or not.

Table 4 shows the information of the driving condition with and without resonators.

Table 4. Parameters of driving points
The experimental results are shown in Table 5 and Figure 10. At each measuring point, the overall averaged sound pressure levels and the sound pressure levels of BPF component are obtained with and without resonator. The average overall noise reduction is about 2.5dB, and the highest is 3.5dB. The compressor noise, particularly the noise component in the frequency band around the blade passing frequency, is significantly reduced. The average BPF noise reduction is about 5.5dB, and the highest is 8.3dB. However, noise reduction at point 3 is smaller than other two points due to the near field effect of point 3 adjacent to the wall.

Table 5. Experimental results at driving points

<table>
<thead>
<tr>
<th>Sound Pressure Level [dB]</th>
<th>Without Resonator</th>
<th>With Resonator</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall SPL</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Point 1</td>
<td>103.6</td>
<td>100.5</td>
<td>3.1</td>
</tr>
<tr>
<td>Point 2</td>
<td>110.1</td>
<td>106.6</td>
<td>3.5</td>
</tr>
<tr>
<td>Point 3</td>
<td>108.4</td>
<td>108.0</td>
<td>0.4</td>
</tr>
<tr>
<td>BPF SPL</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Point 1</td>
<td>99.0</td>
<td>90.7</td>
<td>8.3</td>
</tr>
<tr>
<td>Point 2</td>
<td>106.3</td>
<td>99.8</td>
<td>6.5</td>
</tr>
<tr>
<td>Point 3</td>
<td>100.3</td>
<td>99.2</td>
<td>1.1</td>
</tr>
</tbody>
</table>

From these figures, it is clear that no appreciative differences can be seen in ranges. It can be confirmed that quarter wavelength resonators have no adverse effect on the performance.

CONCLUSIONS

In order to reduce noise radiation from the compressor, two major tasks have been conducted: (1) determination of the numerical optimization design for quarter wavelength resonators applied real compressor, (2) the experimental verification for the quarter wavelength resonators. By applying the quarter wavelength resonators, overall sound pressure level (SPL) and SPL of BPF component can be reduced up to 3.5dB and 8.3dB, depending on the operation conditions and measurement positions. In addition to that quarter wavelength resonators have no adverse effect on the performance for entire flow range. Therefore, it is expected that quarter wavelength resonators are effective in reducing noise for centrifugal compressor and will be used widely for all applications where noise reduction designs are considered.

REFERENCES