2002

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C14-4

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ABSTRACT

The contribution of the accumulator to the noise propagation was investigated through sound measuring experiments by checking the directivity of the noise. Also, experimental modal analysis results showed that some resonance modes of accumulator coincide with the highest peaks in sound spectrum. To investigate the reason of those resonance modes, the finite element analysis (FEA) is conducted. The last part of this paper described a design change of the attachment of the accumulator for noise reduction by the topology optimization that uses the density method and continuum sensitivity, since an empirical design of the present structure failed to constrain some resonance modes. Proposed design changes are applied and verified to eliminate aimed resonance modes in modal test and sound measurement experiment.

INTRODUCTION

Compressors are the main source of the noise and vibration of the air-conditioning unit or refrigerators because the compressor undergoes the severe compression process. Therefore, many researchers have been tried to reduce the noise and vibration of the compressor in various points of views. There are many noise sources of the compressor like gas pulsation, acoustic cavity, valve, and vibrating-structures of the compressor [1]. Regardless of the noise source, the fluctuating pressure of the fluid is given as follows [2],

$$\nabla^2 p = \frac{\rho_0}{B}\frac{\partial^2 \rho}{\partial t^2} = \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} \tag{1}$$

where ρ_0 is density and B is the adiabatic bulk modulus of the medium.

Equation (1) is the linearized, homogeneous acoustic wave equation with the pressure fluctuation, $p(\vec{x},t)$. Using velocity potential ϕ, and particle velocity u, the relation between the particle velocities and the pressure fluctuation is derived,

$$u = \nabla\phi = -\int \frac{1}{\rho_0} \vec{\nabla} p \, dt \qquad (2)$$

Actually, Equation (2) shows that the pressure fluctuation is directly proportional to the particle velocities imparted by the vibrating structure and that reduction of the vibration level of the structure will reduce the noise. Based on this basic concept, the research for reduction of the noise and vibration of a rotary compressor is conducted.

Figure 1: A rotary compressor

A rotary compressor assembly is shown in Figure 1. In Figure 1., the accumulator is attached to the main compressor with a standpipe and a strap. The standpipe is a passage of the refrigerant to be compressed. The strap fastens the accumulator to the compressor shell and reduces the vibration of the accumulator. Some researchers categorized the vibration motion of compressors in two ways as bending motions and rigid body motions [3]. Usually, the later has been known to have little contributions to noise generation, which is the case for vibration of the accumulator under 3kHz. However, using directivity of the noise propagation from the compressor, recent researches show that the accumulator is the main source of the noise at some frequency ranges [4-5].

In this paper, directivity of the sound radiated from the compressor assembly is measured and confirms the previous research results.

FE model of the rotary compressor composed of all the components is built for the analysis and topology optimization. Among various methods to reduce structure-borne noise propagated from a structure, design avoiding the structural resonance within some frequency range is the most preferable and efficient way [6]. In this research, *topology optimization* is used to get a new design for the attachment of the accumulator.

Topology optimization has been widely used to find the optimal design in various fields. Unlike shape and sizing optimization, topology optimization does not require an initial design [7]. Typically, the design starts from a block (a given space) of material formed by a large number of finite elements, and then topology optimization will remove unnecessary elements from the block. That kind of optimization method, especially in this research, is considered to be very efficient because the empirical design is failed to constrain some resonance modes contributing highly to the design.

EXPERIMENTAL ANALYSIS

Sound Measurement

Sound pressure radiated from the compressor is measured and given in Figure 2. In Figure 2, the most severe sound level frequency range is 600Hz~ 2kHz, which is targeted at this research for the sound pressure reduction.

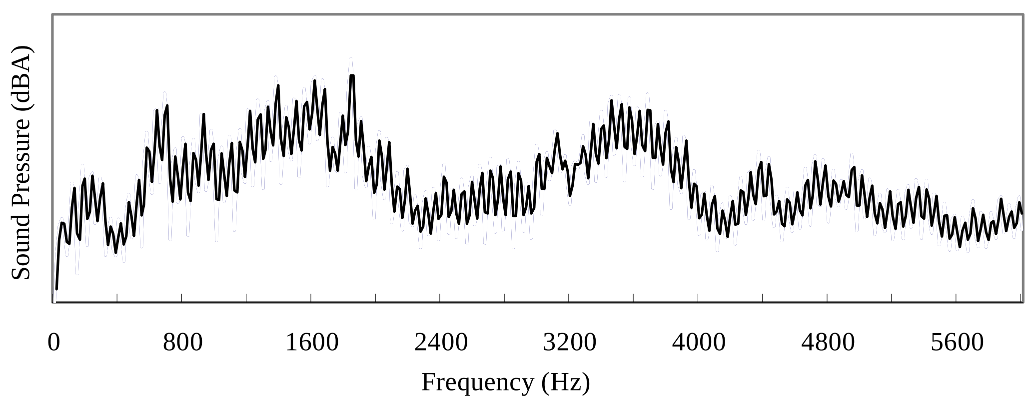


Figure 2: FRF of sound pressure level of the compressor

The sound level measured at 8 points around the compressor is shown in Figure 3.

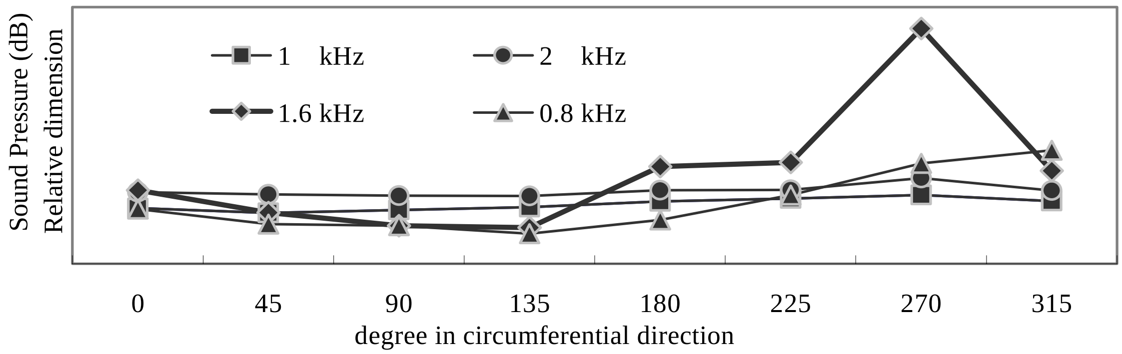
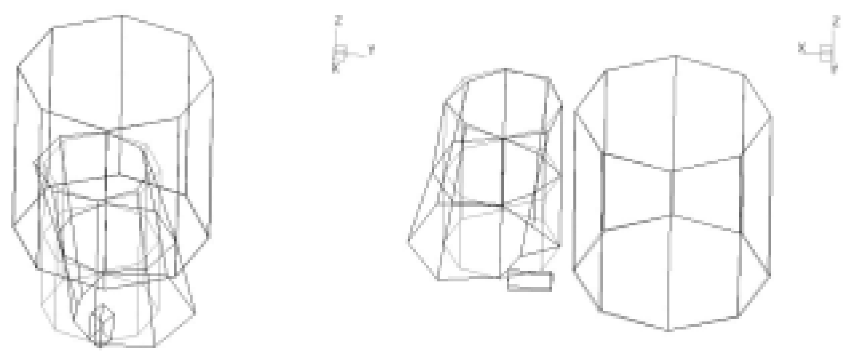


Figure 3: Sound pressure at 8 points around the compressor

The sound pressure level at 270° that falls at the accumulator part shows much higher sound level at 1600Hz center frequency on one-third-octave band analysis.

Modal test

Impact tests are conducted to detect the natural frequencies of the compressor assembly and sub-structure. The first shell bending of detached accumulator falls at 3300Hz which is far beyond the concerned frequency range, 600~ 2000Hz. That result represents that rigid body motions of accumulator under 2kHz contribute highly to the directivity of sound pressure level at 1600Hz on octave band analysis in Figure 2. For the compressor assembly, two kinds of modal tests, without the strap and strap-installed, are conducted to confirm the role of the present strap. Both of the modal tests show that under 2kHz the main resonance modes are the rigid body motions of the accumulators, which are caused by deflection of the standpipe. These results are given and compared with FE analysis results in Table 1. Among them, resonance modes at 945Hz and 1746Hz are given in Figure 4 for strap-installed assembly. In Figure 4, the pivot point of the modes is the strap, which is intended to reduce the vibration motion of the accumulator. Therefore, the present strap doesn't constrain or reduce those vibration motions. And those resonant frequencies correspond with the highest sound level frequencies at Figure 3.



(a) 945Hz (b) 1746Hz

Figure 4: Mode shapes of the rigid body motion of the accumulator in modal test

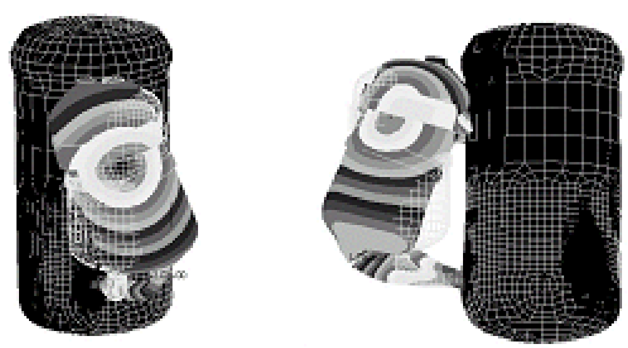
FINITE ELEMENT ANALYSIS

Finite element model of the compressor assembly including whole inner parts such as the shaft, stator, rotor, and bearings is built for the FE analysis and design change using CAE.

Table 1 shows the comparison of resonance modes between the modal test and FE analysis results. Figure 5 shows the mode shapes at 963Hz and 1742Hz, respectively, for the strap installed case. Considering the mode shapes in Figure 4 and Figure 5 and the frequency change after installation of the strap, it can be inferred that the present strap doesn't work for those modes. Furthermore, those resonance modes are located at the highest sound level frequency range. Design change for the attachment of the accumulator to compressor using topology optimization is conducted in this research.

Table 1: Modal analysis comparison between Experiment and FE analysis

Strap un-installed assembly				Strap installed assembly			
Mode number	Test (Hz)	FE analysis (Hz)	Error (%)	Mode number	Test (Hz)	FE Analysis (Hz)	Error (%)
1st	42	43	2	1st	231	234	1
2nd	52	53	2	2nd	390	375	4
3rd	379	361	5	3rd	504	448	11
4th	390	397	2	4th	526	483	8
5th	426	406	7	5th	945	963	2
6th	472	471	1	6th	1428	1382	3
7th	528	484	8	7th	1558	1673	7
8th	908	933	2	8th	1746	1742	1
9th	1548	1553	1				
10th	1744	1702	2				



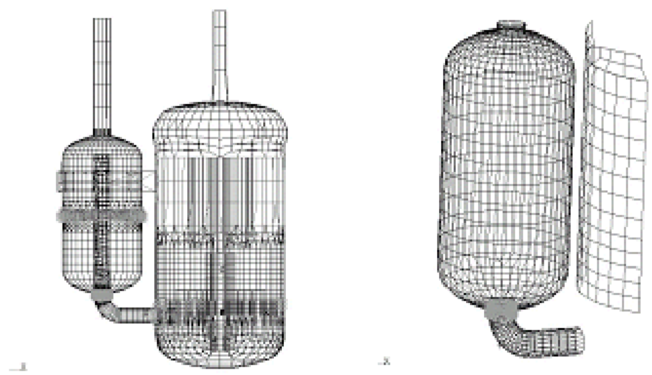
(a) 963 Hz (b) 1742 Hz

Figure 5: Mode shapes of the rigid body motion of the accumulator in FE analysis

TOPOLOGY OPTIMIZATION

Optimization problem

FE models are constructed for the modal analysis and topology optimization, respectively. For the former case, analysis results are deeply affected by the element size. Thus, through the convergence test, the minimum available element size which can guarantees the accuracy is used. But for topology optimization, the optimization procedure needs many structural analyses and sensitivity analyses as well. Therefore, the analysis model is reduced to a design model because the size of the analysis model is too large to be used for the optimization. In design model, design domain is modeled in detail and the other part is modeled with coarse meshes. Figure 6 shows the analysis model and design model, respectively. In the design model, compressor shell part is regarded as rigid wall. Normal mode analysis of the design model shows that reduction of the model and assumption of the rigid wall of compressor shell doesn't make much difference of the target modes. Table 2 shows the comparison of model size, and Table 3 gives the modal analysis results.



(a) Analysis model (b) Design model for optimization
Figure 6: Analysis and Design models

Table 2: Comparison of model size

	Analysis model	Design model
Element number	9191	2179
Node number	10117	2216
Degree of Freedom	39755	8970

Table 3: Comparison of normal mode analysis results

Mode number (Original model)	Analysis model (Without strap)	Design model
1^{st}	42Hz	55Hz
2^{nd}	52Hz	64Hz
3^{rd}	379Hz	
4^{th}	390Hz	
5^{th}	426Hz	
6^{th}	472Hz	
7^{th}	528Hz	
8^{th}	908Hz	1000Hz
9^{th}	1548Hz	
10^{th}	1744Hz	1840Hz

Design variable domain

In topology optimization, design parameters are set to a given domain of design variables, in this research as solid elements. Optimization results are deeply affected by configuration of design variable domain, design variable numbers, and size.

For the problem given, maximizing a target frequency is tried as follows,

maximize *target frequency*
subject to *given amount of design variables* (3)

bounds on *design variables* $\mathbf{x}^L \leq \mathbf{x} \leq \mathbf{x}^U$

where $\mathbf{x}$ is a set of design variables and $\mathbf{x}^L$, $\mathbf{x}^U$ are lower and upper side limits of design variables, respectively.

Result of the optimization

Using only 10% of the original design variable, the target frequency of 1744Hz is shifted to beyond 2500Hz. Figure 7 shows the result of the topology optimization. Figure 8 shows the topology optimization result from the top and bottom view.

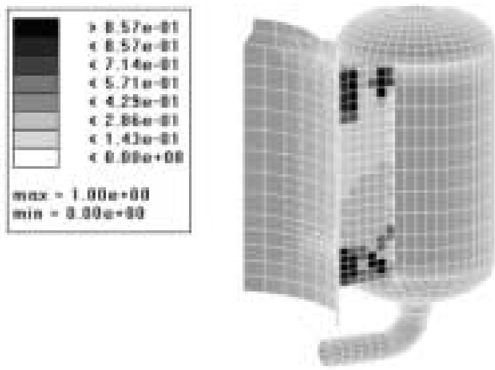
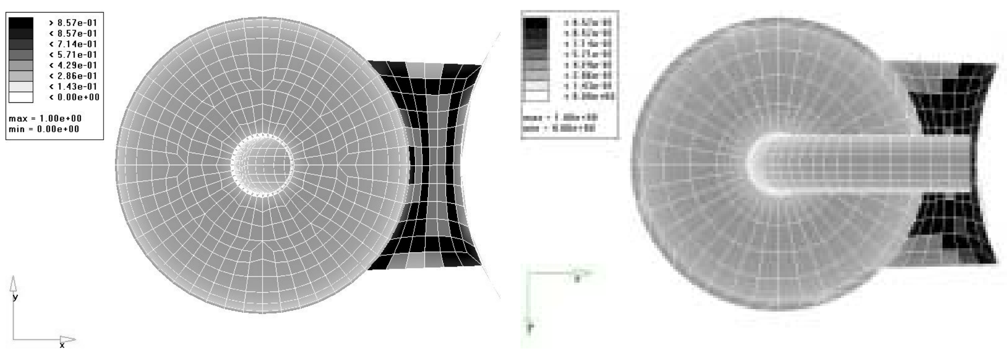


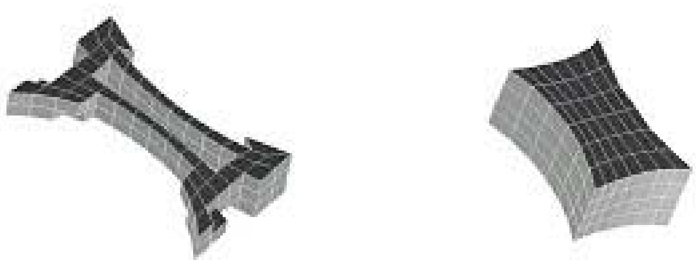
Figure 7 : Topology optimization for the given design variable domain



(a) Top view (b) Bottom view

Figure 8: Top and bottom view of the topology results

From figures 7 and 8, new design of accumulator attachment to rotary compressor is suggested by topology optimization as shown in Figure 9.



(a) Upper part (b) Lower part

Figure 9: Final design of accumulator brackets

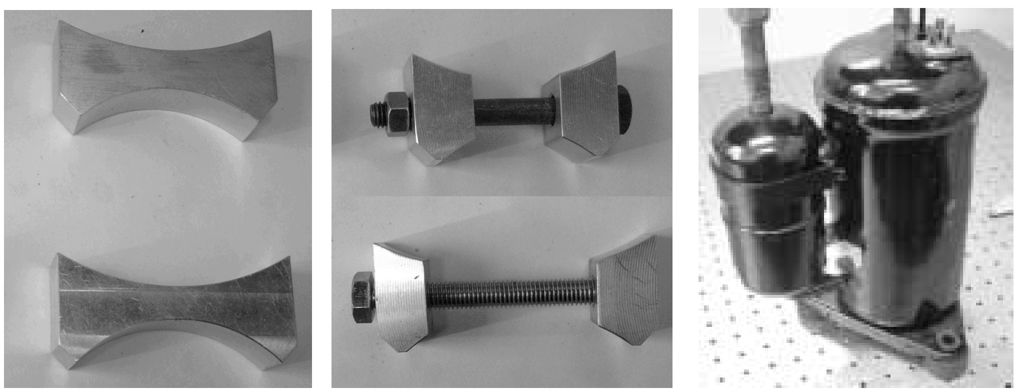
In Table 4, modal analyses of three models are compared. These models are the design model, the model with optimized upper and lower brackets, and the model with optimized upper bracket. Comparison of the modal analysis results shows that stiffness increase of the upper strap alone is not sufficient for the constraint of the target modes. Therefore, a block shaped lower bracket should be added to shift the target resonance modes.

Table 4. Comparison of the normal mode analysis

Mode number	Design model	Upper & Lower	Upper only
1st mode	55Hz	833Hz	721Hz
2nd mode	64Hz		
3rd mode	1000Hz	2179Hz	1288Hz
4th mode	1840Hz	2375Hz	1606Hz

VERIFICATION WITH EXPERIMENT

Figure 10, (a) shows blocks made following the optimization results. But that configuration made it difficult to install the block into the compressor assembly. Thus, considering assembling process modified block, figure 10, (b) is developed. Figure 10, (c) shows block installed compressor assembly.



(a) single-piece blocks (b) bolted blocks (c) block installed assembly

Figure 10: Pressure comparisons according to the block installation

Figure 11 and Figure 12 show FRF of the impact tests and sound pressure respectively. For the case of block-inserted compressor magnitude in impact test and sound pressure are decreased at around 1.7kHz

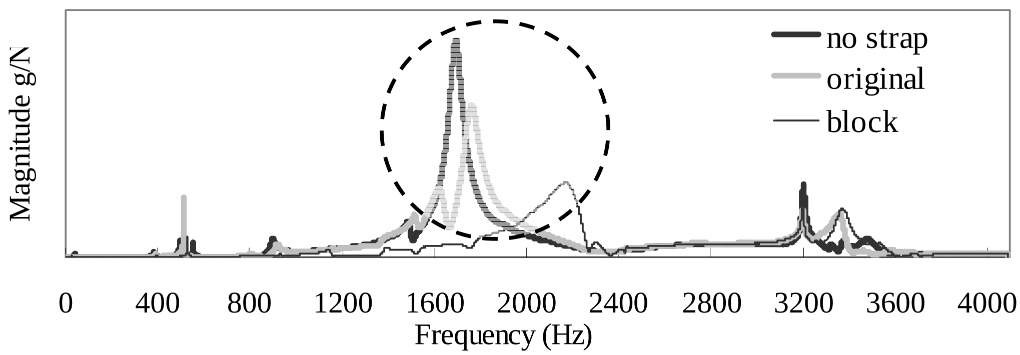


Figure 11: Comparison of the FRF of the modal test for original and block-inserted model

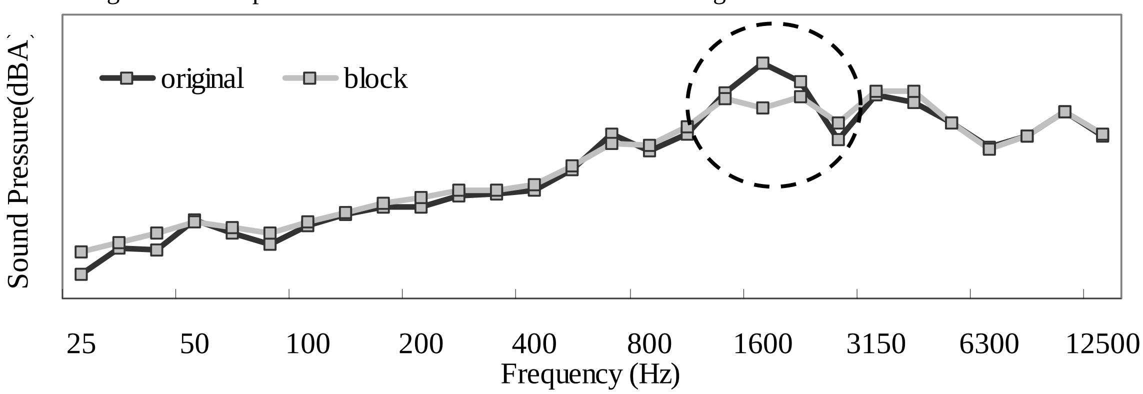


Figure 12: Sound Pressure comparisons according to the block installation

CONCLUSIONS

Sound measurement test is carried out to find target frequency range, which contributes highly to the noise propagation. And sound spectrum indicates that the accumulator is a significant component in objective frequency range.

Experimental modal analysis is used to disclose the dynamic characteristics of the rotary compressor. One of the resonance modes corresponds to the highest peak in sound spectrum, which implies that constraint of that mode will reduce the noise level.

Finite element model with whole component of the compressor assembly is constructed and used for the modal analysis.

Using topology optimization result, a block-shaped component is inserted between the accumulator and the compressor. Sound measurement test and modal test with the block inserted verifies the topology optimization results. Sound spectrum shows reduced noise level at the target frequency range. Topology optimization results show that the empirical design for the present structure fail to constrain some resonance modes and a new component is required.

ACKNOWLEDGMENT

This work was supported by RRC-HECS, CNU under grant (R12-1998007000-0) and by Carrier Co.

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