Generative design of air compressor mounting rubber by using PACE software.

Kwanju Kim [1], Uyeup Park [2], Jun-Heon Lee [2]

(1) : Dept. of Mechanical System Design Engineering, Hongik Univ., 94,
Wausan-ro, Mapo-gu, Seoul, Republic of Korea
+82-23201643
E-mail : kwanju@hongik.ac.kr

(2) : Dept. of Mechanical Engineering,
Hongik Univ., 94,
Wausan-ro, Mapo-gu, Seoul, Republic of Korea
+82-23201113
E-mail : dnduq5065@naver.com,
heon11@gmail.com

Short Abstract: Urban rapid railway is one of the most popular transportation methods these days. It is equipped with large-capacity air compressors because they use pneumatic pressure to supply power for braking and for opening and closing doors. Passengers tend to complain about vibration and noise that occur during the operation of the air compressor. In this study, vibration reduction of air compressor was achieved through generative design of mounting rubber. First, the stiffness of mounting rubber has been derived through multi-dynamics analysis. Secondly, the shape of mounting rubber was chosen by topology optimization. Lastly, the proposed rubber mount was verified by experiments.

Key words: Generative design, Air compressor, Vibration, topology optimization, Mounting rubber.

1- Introduction

Railway vehicles are equipped with large-capacity air compressors because they use pneumatic pressure to supply power for braking and for opening and closing doors. Complaints from passengers are often raised from annoying vibration and noise generated during the operation of the air compressor. To solve such issues, it is necessary to either reduce the force unbalance occurred by the compressor or to minimize the transfer force of the air compressor excitation to the vehicle frame. In this study, optimum mounting rubber is derived, and vibration reduction is realized through experiments [R1]. This paper is organized as follows: the first section shows the contribution of the air compressor's excitation to the overall vibration and the acoustic amount inside the railway cabin when a railway vehicle is running. The second section derives optimal mounting rubber stiffness using multibody dynamics analysis program. The third section, in order to satisfy the proposed stiffness value, derives the optimal shape of mounting rubber using structural topology optimization. Production possibilities are also reviewed. Finally, in the last section, experiments are performed to measure vibration and compare baseline with proposed mounting rubber. Figure 1 is a flow chart of this study.

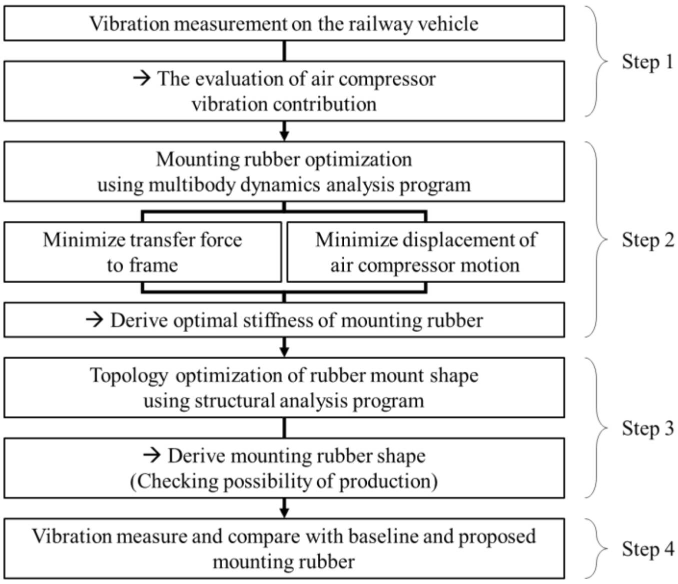

Figure 1: The flow chart of this study

2- NVH experiment inside railway vehicle cabin.

2.1 – Set-up of NVH experiment on railway vehicle interior.

The vibration and noise measurements on the cabin floor above the air compressor in an operating railroad vehicle have been carried out. The NVH contribution of an air compressor on the railway vehicle has been investigated. Four acceleration points on the floor and two sound pressure points located 1.5 m above the floor were measured. A photograph of the experiment scene is shown in Figure 2.

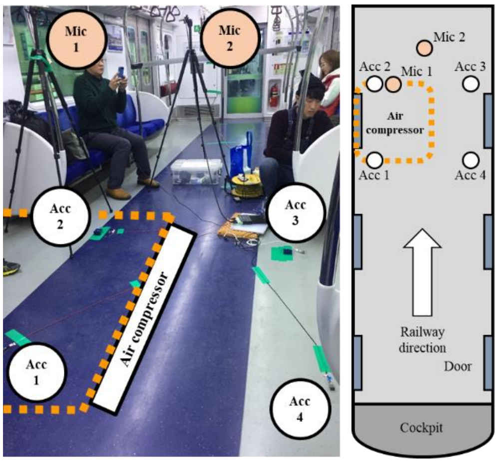

Figure 2: Vibration and noise experiments carried out in running railway vehicles.

2.2 – Results of NVH experiment on railway vehicle interior.

Figure 3 shows the acceleration magnitude at the location of accelerometer 1 in the frequency domain depending on the air compressor operation. The magnitude of the acceleration during the compressor operation is 20 times larger than when the compressor stopped operating at 29.5 Hz and 80 times larger at 176 Hz [KP1].

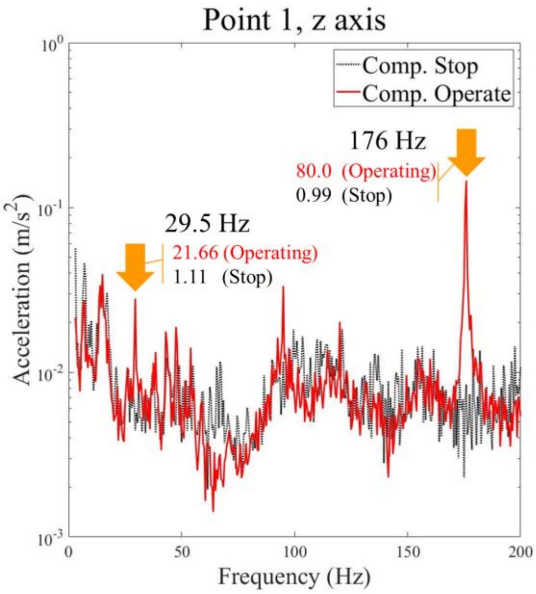

Figure 3: The acceleration magnitude at the location of accelerometer 1 in the frequency domain depending on air compressor operation. (Grey line: The compressor is activated. Red line: The compressor stopped.)

3- Derivation of optimal mounting rubber stiffness using multibody dynamics analysis program.

3.1 – Air compressor model in multibody dynamics program.

Figure 4 shows the air compressor model in MSC Adams. The crankshaft is rotating at 1750 rpm and the stiffness values of baseline mounting rubber are k_x=203 kN/m, k_y=203 kN/m and k_z=613 kN/m. Design of Experiment method was used to derive the optimal stiffness value of mounting rubber in order to minimize the transferring force to the frame and to minimize the motion of the air compressor at the same time. Full factorial technique was used in the DOE method. The design variable was stiffness of the mounting rubber, and the range was from 20 kN/m to 1020 kN/m [K1].

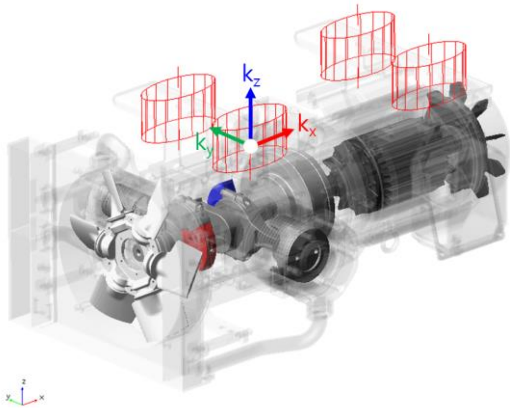

Figure 4: The air compressor model in multibody dynamics program, MSC Adams.

3.2 – Results of DOE to minimize the transfer force and the motion of the air compressor.

Figure 5 is a graph of the transfer force with respect to the mounting rubber stiffness k_x, (= k_y) and k_z. The transfer force has an adverse effect on ride comfort, so it should be as small as possible. Therefore, the optimal stiffness value of the mounting rubber should be within the pink-dotted border.

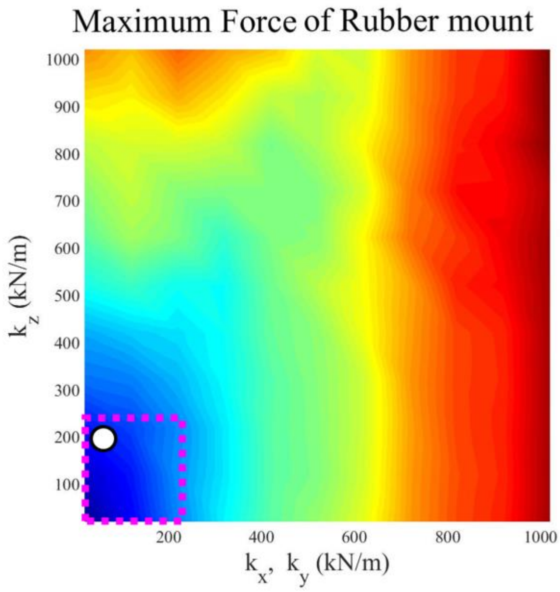

Figure 5: Transfer force from the air compressor to frame with respect to mounting rubber stiffness.

Figure 6 shows the maximum motion of air compressor with respect to the mounting rubber stiffness. The larger the magnitude of motion, the worse the operation and the durability of the mounting rubber. Therefore, the selected stiffness value of the mounting rubber in terms of motion should be within the pink dotted border.

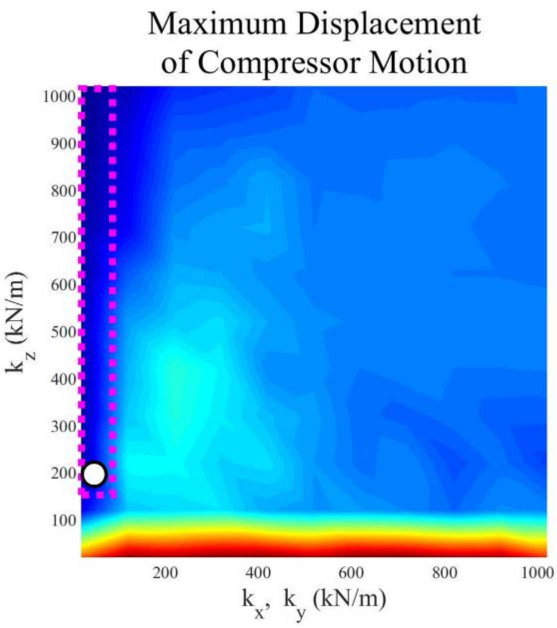

Figure 6: The motion of the air compressor motion with respect to mounting rubber stiffness.

The stiffness value which overlaps the range of the pink-dotted ranges of Figure 5 and Figure 6 is the most suitable stiffness value for the rubber mount. Therefore, the values of k_x, k_y will be selected to 60 kN/m, and k_z 200 kN/m which is indicated by a white circle in Figure 5 and Figure 6.

4- Derivation of optimal shape of the mounting rubber using structural topology optimization analysis program.

4.1 – First topology optimization of the baseline mounting rubber.

The shape of the baseline mounting rubber is a solid cylinder as shown in the left hand side of Figure 7. In order to find the proper shape to meet the suggested stiffness value viz k_x, k_y=60 kN/m, k_z=200 kN/m, topology optimization routine in ANSYS has been executed. The volume of the mounting rubber was set to reduce up to 50% to satisfy the target value. The right hand side of Figure 7 shows the suggested shape of the mounting rubber. However, it tends to be an impractical and unrealistic shape to manufacture. Hence, it is necessary to contemplate on a mounting rubber with a more practical shape.

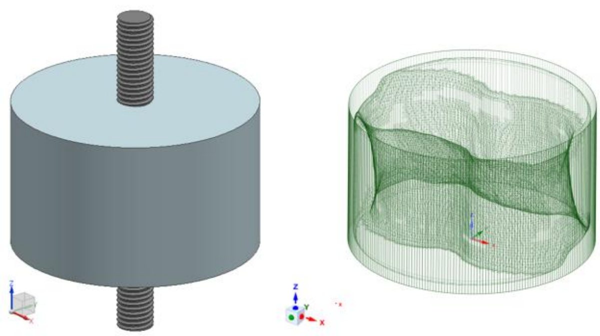

Figure 7: The shape of the baseline mounting rubber(left), The suggested design shpae of the mounting rubber(right)

4.2 – Second topology optimization of the mounting rubber with changing how to assemble bolts.

The force generated from the air compressor is directly transmitted to the frame through the rubber as depicted in the left hand side of Figure 8. However, in the mounting rubber with the modified assembly method, the force from the compressor can be transmitted to the frame indirectly and has more freedom to control it, which is shown in the right hand side of Figure 8.

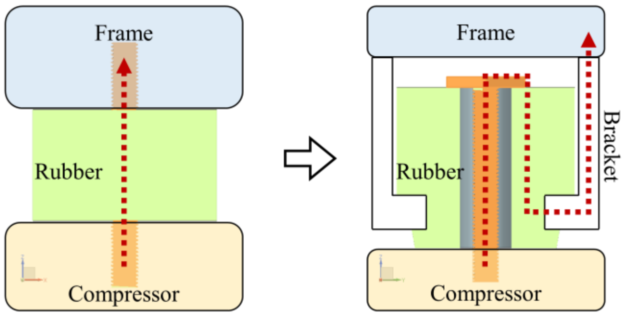

Figure 8: The orginal mounting rubber location(left) and the modified one(right) (Red dotted line : the path of transfer force)

The left hand side of Figure 9 is the mounting rubber with a modified assembly method. Topology optimization for this mounting rubber, and the optimal shape of the rubber is shown in the right hand side of Figure 9.

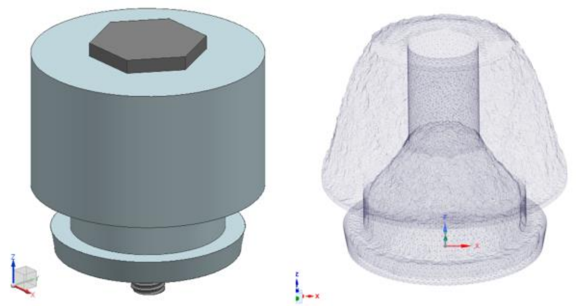

Figure 9: The mounting rubber with modified assembly method(left), Shape of proposed mounting rubber suggested by the second topology optimization(right)

5- The comparison of dynamic behaviour of the baseline and the proposed mounting rubber by the vibration jig test.

Experiments were carried out to compare the transfer force of a compressor equipped with a conventional rubber and that of the proposed rubber. Figure 10 shows the scene of performing the vibration experiment, and Figure 11 shows an enlarged view of the attached rubber.

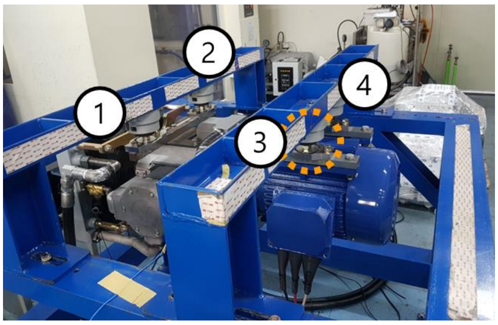

Figure 10: The jig test of the air compressor mounted at four points on the frame. The orange-dotted circle shows the proposed mounting rubber.

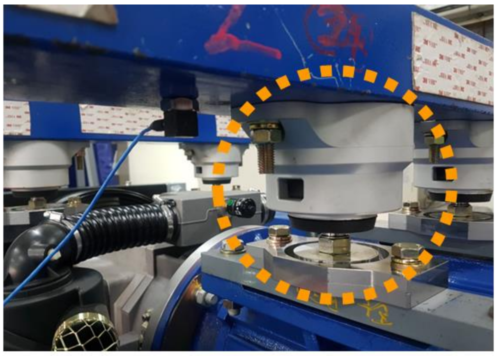
Figure 11: The picture of enlarged mounting rubber on jig with the force sensor on the bottom.

The result of acceleration measurement at point 2 is shown in Figure 12. The acceleration peaks appear at 29.5 Hz, driving frequency of the compressor, and its harmonic components. The acceleration magnitudes of the proposed mounting rubber decreased by more than 2 digits compared with those of baseline one. The peaks at 90 Hz and 120 Hz are bending modes of the frame. So there are no magnitude changes in both cases. The overall magnitudes of acceleration values are shown in Table 1. It can be seen that the acceleration value has decreased by 30 % on average.

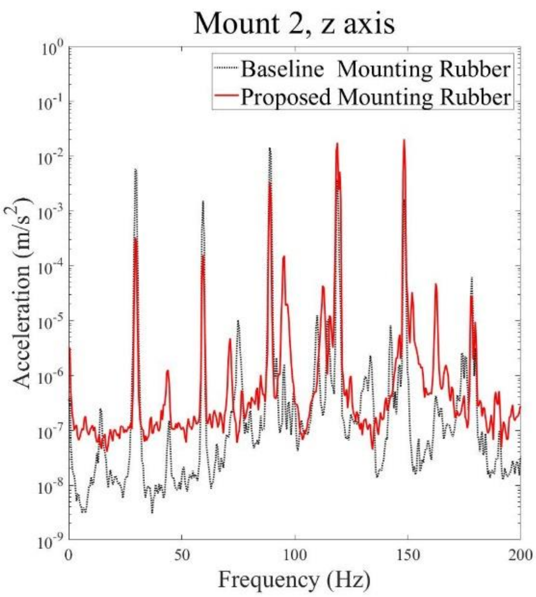

Figure 12: Acceleration magnitudes in frequency domain at point 2 (Black dotted line : The baseline mounting rubber case, Red line : The proposed mounting rubber case).

	Baseline (m/s^2)	Proposed (m/s^2)	Reduction rate (%)
Point 1	0.148	0.107	27.5
Point 2	0.235	0.127	45.9
Point 3	0.148	0.119	19.9
Point 4	0.160	0.134	16.5
Average	0.173	0.122	29.6

Table 1: The overall acceleration magnitudes of the baseline and proposed mounting rubber.

6- Conclusion

In this study, the vibration contribution of the air compressor was investigated through experiments in running vehicle. It was confirmed that vibration reduction was needed via the mounting rubber design. In order to find out the optimal dynamic behaviour mounting rubber, first, a conventional air compressor-mounting rubber system was modelled through the multibody dynamics program, MSC Adams, and the optimal stiffness of mounting rubber was derived. Secondly, the shape of mounting rubber was founded through topology optimization to satisfy the stiffness value. Finally, it was confirmed by experiment that the vibration magnitude of the proposed mounting rubber was reduced by 30 % on average compared to baseline one.

7- Acknowledgements

This research was supported by 2018 Hongik University Research Fund and a grant(18RTRP-B122846-04) from Railroad Technology Research Program funded by Ministry of Land, Infrastructure and Transport of Korean government.

8- References

[K1] Kim C., MSC Adams for multibody dynamic analysis. Jinsaem, pp. 356-366, 2016.

[KP1] Kim K., Park U., Prediction of transfer force of railway air compressor using transfer path analysis. In Journal of the Korean Society for Railway, 22(2): 101-108, 2019.

[R1] Randall R., Frequency analysis, Larsen & Son A/S, pp. 271-304, 1987.