A Study on Improvement of Vibration Performance of Air Damping Bush Using Fluid-Structure Interaction Analysis

Eui Soo Kim
Dept. of Safety Engineering, Korea National University of Transportation, Chungju-si, Chungbuk 27469, Korea
Email: es92kim@ut.ac.kr

***Abstract*— Because the three-point engine mount among the air damping engine mount has excellent performance, many studies have been carried out. However, studies on the high damping roll mount have been demanded because of the high contribution rate of the interior noise of the roll mount. In this study, the vibration characteristics for the basic design model of the engine mount according to each case were examined through the fluid-structure interaction analysis applying the trial & error method and the optimal design that can increase the damping effect is suggested. In order to examine the damping effect of the air damping engine mount, five design model including the initial model were reviewed. It was confirmed that the damping effect when the system stability is considered is the largest in final change model in which the stopper is removed from the reviewed design model. The optimum model of air damping bushing is proposed through this study to improve vibration performance.**

***Index Terms*—vibration performance, air damping bush, fluid-structure interaction analysis, optimum model**

I. Introduction

The engine mount is positioned between the vehicle body and the engine to support the engine and reduce noise and vibration transmitted from the engine to the vehicle interior through the vehicle body. In addition, resonance by engine mass and mount stiffness occurs due to excitation of road surface during driving, which causes excessive vibration of the engine [1]. In order to reduce this vibration, the mount must be designed so that the damping characteristics are largely exhibited at the resonance frequency. In order to satisfy these conditions, fluid sealing mounts are generally applied to vehicles. They have a high damping performance compared to general rubber mounts but they have complicated structure, which makes it difficult to manufacture and analyze. Therefore, recently, much research has been conducted on air damping engine mounts that are less costly and heavy in weight due to no fluid as an alternative to fluid sealing mounts [2]. Because the three-point engine mount among the air damping engine mount has excellent performance, many studies have been carried out. However, studies on the high damping roll mount have been demanded because of the high contribution rate of the interior noise of the roll mount. In this study, the vibration characteristics for the basic design model of the engine mount according to each case were examined through the fluid-structure interaction analysis applying the trial & error method and the optimal design that can increase the damping effect is suggested.

II. Parameter Model and Analysis

A. Applied Lumped Parameter Model

In the air damping engine mount presented in this research, the upper / lower chamber is fixed by the steel structure of the outer shell, and since it is symmetrical structure, it can be depicted as the following Figure 1. Here, P_1 and P_2 are the pressure of air in each chamber, and C is $C_1 = \Delta V_1 / \Delta P_1$, $C_2 = \Delta V_2 / \Delta P_2$ in the compliance of the chamber [3]. ΔV1 represents the volume change of the upper chamber, and ΔV_2 represents the volume change of the lower chamber. The volume of the upper chamber can be expressed as $V_1 = V_L At$, where V_L is the velocity acting on the wall of the chamber, A is the piston area of the face compressed, and t is time.

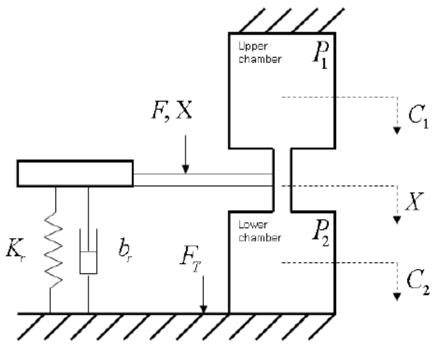

Figure 1. Parameter model of air mount

B. Concept of Fluid-Structure Coupled Analysis

Fluid-structure interaction (FSI) analysis applies physical strength to solid, and deformation of the solid due to this fluid force changes the area of the fluid. In the calculation region at the interaction part, interaction occurs along the boundary surface between the two regions by separating into fluid region and solid region

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having respective physical property values and boundary conditions [4]. Kinematic and dynamic conditions must be satisfied at the interface where both models can be coupled and predicted and reproduced simultaneously for multiple physical phenomena. The position of the fluid node at the fluid structure interface is determined by the kinematic condition and the displacement of the remaining fluid node is automatically determined by the program to maintain the initial mesh quality.

III. Pre-process of FSI Analysis

A. *Modeling and Material Property*

The object of this study is air damping bush (ADB) for damping vibration composed of one chamber at the top and two chambers at the bottom and air moves through the path of fluid during compression. In the centre part, the aluminium core connected to the shaft is surrounded by rubber and when it is excited, the core compresses the upper and lower air chambers and is designed to damping the vibration. In the proposed model, there is a space of 2 to 3 mm between the core part and the upper and lower chambers, and when the displacement is less than this value, the damping effect is expected to be small. First of all, in order to evaluate the vibration characteristic of this model, the transient analysis according to each excitation frequency is executed at the main interested frequency from10 to 50 Hz. Materials used for the structural model were largely made of aluminium, steel, rubber, and showed the material properties applied in Table I. Aluminium and steel use elastic material model, rubber use Mooney-Revlin material model and fluid model use general air properties. The lattice model for fluid-structure coupled analysis is as follows in Fig. 2 and Fig.

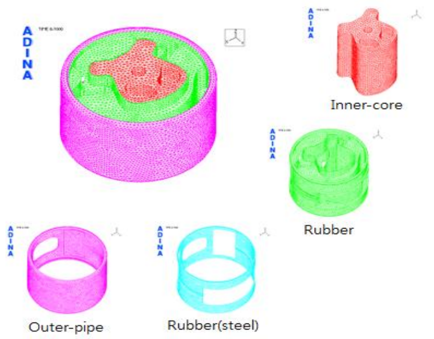

Figure 2. Structural analysis model of Air damping bush

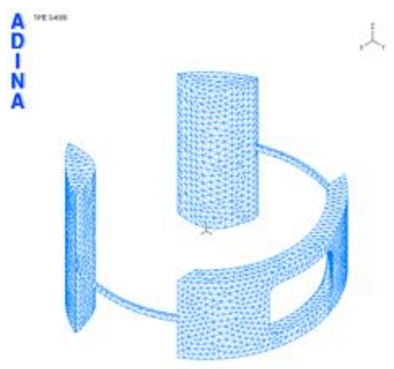

Figure 3. Structural analysis model of air chambers

3. 68,000 3D solid elements are used for the structural model and 6,500 3D fluid elements are used for the fluid model [5].

TABLE I. Material Properties of Structure and Fluid Model

Material	Property
Aluminium	P(2700 kg/m^3), E(7 GPa), ε(0.33)
Steel	P(7800 kg/m^3), E(200 GPa), ε(0.33)
Rubber	C10(0.0385 MPa), C20(0.0585 MPa), C30(0.019 MPa)
Ball	P(1.2 kg/m^3), E(7 GPa), μ(1.8e-5 Ns/m^3)

B. *Boundary and Constraint Conditions*

The loading condition of the structural model for analysing the vibration characteristic applied the time history load of the sine wave to the aluminium core and the value of displacement use 0.2-3 mm. As a boundary condition, the DOF (Degree of Freedom) of the outer part of the steel is all fixed, and the fluid-structure boundary condition is applied to the interface part with the fluid and the interface, and pressure boundary conditions were applied to a point of this interface. In this analysis, in order to confirm the result, loss angle and dynamic stiffness by frequency was calculated. The loss angle is calculated as follows by measuring the delay time, ϕ of the input ratio output. The dynamic stiffness is obtained by obtaining the force amplitude and the displacement amplitude from the analysis results and dividing the force amplitude by the displacement amplitude.

$$K_d = \sqrt{{K_s}^2 + {K_l}^2} \tag{1}$$

$$\emptyset = \mathrm{arctg}(K_l/K_s) \tag{2}$$

IV. Result of FSI Analysis

A. *Initial Design Model*

The initial model was designed with a space of 2 - 3 mm between the upper / lower air chamber and the core rubber parts. There is a stopper in the centre of the upper chamber, and the core comes into contact with a stopper when core is vibrated. In order to evaluate the vibration characteristics of the initial model, firstly, the force was measured while increasing the displacement and then the slip was calculated to calculate the static stiffness. After reviewing curve of the static stiffness, the space between the stopper of the upper chamber and the core rubber part is about 2 mm, so slop is changed after the contact occurred. The static stiffness measured at the position of displacement of 2 mm is 25.8 kg$_f$ / mm. The lost angle was calculated after initially with amplitude of 0.2 mm, but the delay time was not displayed as shown in Fig. 4, Fig. 5. The reason is that the amplitude is small compared to the space between the predicted structural models and there is no contact between the core and the phase / lower chamber, and there is no effect that the air is compressed. In order to solve this problem, we examined a method of filling the space between the core and the rubber with rubber so that contact from the beginning occurs. Virtually, it set that contact at the beginning of FSI analysis occurs by using the Glue condition.

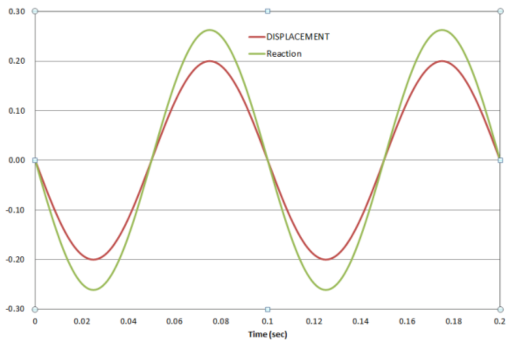

Figure 4. Displacement load curves(10Hz)

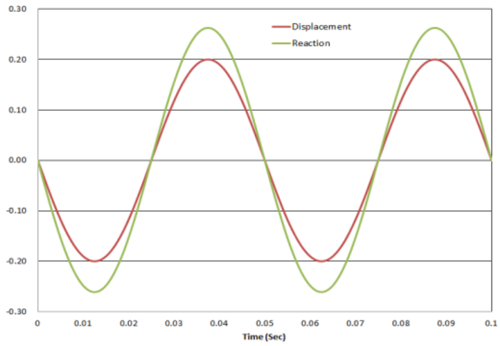

Figure 5. Displacement load curves(20Hz)

B. *Design Change Model*

The space between the upper / lower chamber and the core part was made to vibrate region around the stopper in the upper portion and region close to the chamber in the lower portion. Because the volume of the air chamber does not change, compression is not expected to be large when the amplitude of the air chamber is small, so we accomplish FSI analysis by increasing the amplitude to ± 3 mm. We can see that there is a loss time compared to the initial model, but when converting at the loss angle, the value is not so large. In other words, we can know that the damping effect is not so large by partially filling the space with rubber as shown in Fig. 6. In order to confirm the damping effect corresponding to the cross sectional area of the flow path, the 2nd change model which is changed the cross sectional area of the flow path is compared with the 1st change model. Analysis was performed by changing the cross-sectional area of 2 mm x 1 mm flow channel of the primary model to 0.5 mm x 0.5mm. The reason for reducing the cross section area is because we expected that the damping effect will increase as the speed increases. Unlike expectations, rather than increasing by about 2 degrees at 40 Hz, we got the overall decrease as shown in Fig. 7. According to above results, the compression of the air in the upper chamber hardly occurred and the damping effect did not occur. The reason is that the air on both sides of the stopper will not be compressed normally even if the amplitude in the part that immediately comes into contact with the core is increased by the stopper. So we removed the stopper and modified it with core attached and designed so that enough air can be compressed. The third model applied the same ± 3 mm displacement as the second model and applied the Glue condition at the same position and made it to be excited when the contact occurred. As a result, we can see that the loss angle has greatly increased compared with the first and second models as shown in Fig. 8, Fig. 9. However, the deformation of the upper chamber was large, and when the results of the overall analysis time were confirmed, the upper chamber showed rocking to the extent that it could be identified with the naked eye. This may require supplementation as actual stress may concentrate on certain places and damage the system. Finally, In order to reduce the excessive deformation of the upper chamber, FSI analysis was performed with the space between the upper chamber and the core filled with rubber. We calculated loss angle and dynamic stiffness at 30, 40 and 50 Hz with displacement of ± 3 mm. The results are the same as in Fig. 10, Fig. 11. As a result, there is no significant difference in the dynamic stiffness value compared with the third-order model, but the loss angle is greatly reduced. Although the third-order model has a large damping effect, the deformation of the rubber is so severe. In conclusion, the final change model is the best design because the rubber might be damaged in the case of the 3rd change model.

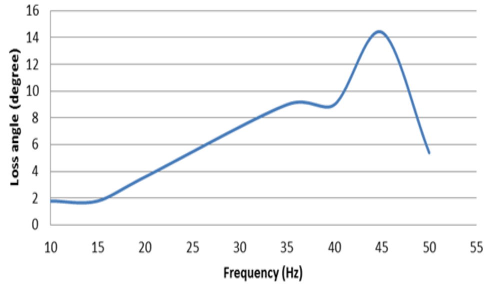

Figure 6. Loss Angle by Frequency at 1ST design change

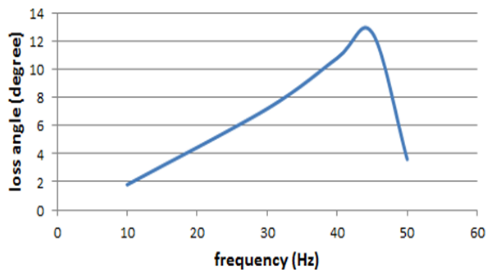

Figure 7. Loss Angle by Frequency at 2nd design change

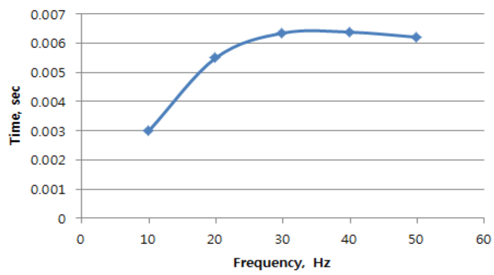

Figure 8. Loss time of the 3rd design change

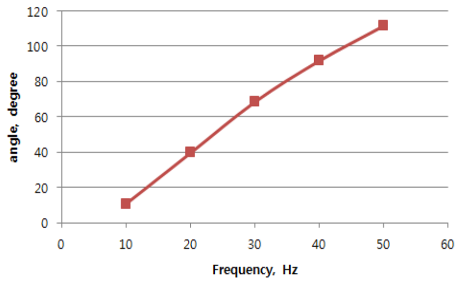

Figure 9. Loss angle of the 3rd design change

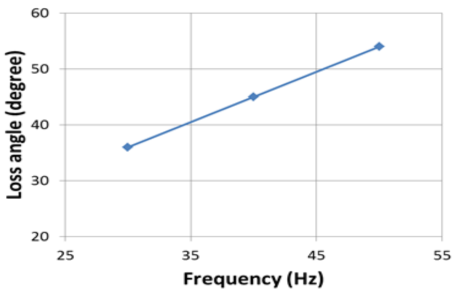

Figure 10. Loss time and angle of final model

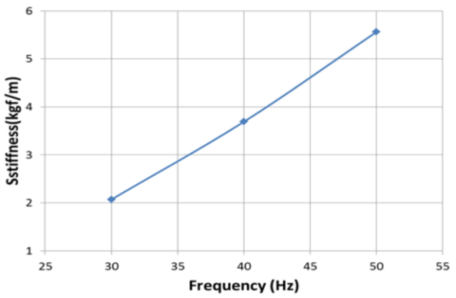

Figure 11. Dynamic stiffness of final model

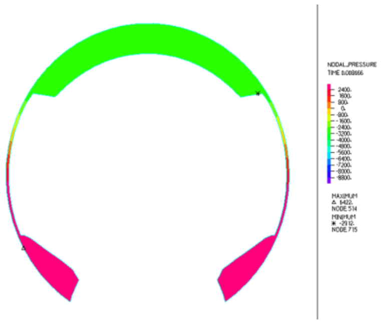

Figure 12. Fluid pressure of air chambers(lower)

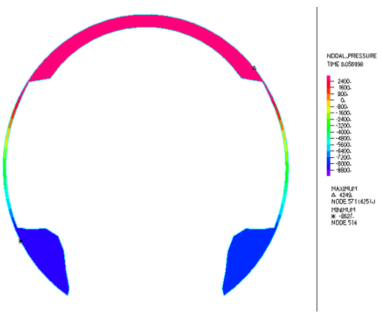

Figure 13. Fluid pressure of air chambers(upper)

V. Conclusion

In order to examine the damping effect of the air damping engine mount, five design model including the initial model were reviewed and the results are summarized in the follows. The initial model was designed with a space of 2 - 3 mm between the upper and lower air chamber and the core rubber parts. The lost angle was calculated after initially with amplitude of 0.2 mm, but the delay time was not displayed. In secondary design change model, in order to confirm the damping effect corresponding to the cross sectional area of the flow path, the 2nd change model which is changed the cross sectional area of the flow path is compared with the 1st change model. The third model applied the same ± 3 mm displacement as the second model and applied the Glue condition at the same position and made it to be excited when the contact occurred. Finally, In order to reduce the excessive deformation of the upper chamber, FSI analysis was performed with the space between the upper chamber and the core filled with rubber. It was confirmed that the attenuation effect when the system stability is considered is the largest in the fourth order model in which the stopper is removed from the reviewed design model. By removing this stopper, it is judged that the area compressed by the core increases.

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Eui-soo. KIM was born in 1974. He received his doctor's degree at Pusan National University, Currently teaching at the Korea National University of Transportation, Associate Professor