Optimal Shape Design of an Air-Conditioner Compressor-Mounting Bracket in a Passenger Car

Doo-Ho Lee  
Dongeui University  
Jeong-Woo Chang  
Halla Climate Control Corp.  
Chan-Mook Kim  
Kookmin University

ABSTRACT

An air-conditioner compressor-mounting bracket is a structural component of an engine, on which bolts attach an air-conditioner compressor. In this paper, the shape of the air-conditioner compressor-mounting bracket of a passenger car is optimized using a finite element software, and the optimized bracket is manufactured and verified by tests. An objective function for the shape optimization of the bracket is the weight of the bracket. Two design constraints on the bracket are the first resonant frequency of the compressor assembly and the need not to fracture during the workbench durability test. The compressor assembly, which consists of a compressor, a bracket and connection bolts, is modeled using finite elements. The bracket is modeled by solid elements and the swash-plate-type compressor is represented by rigid masses and beam elements in order to consider the elastic effects of the compressor. For simulation of the dynamics stresses in the durability test, the large mass method is used. The design variable is the shape of the bracket including thickness profiles of the front and back surfaces of the bracket, radius of outer bolt holes, and side edge profiles. The coordinates of the FE nodes control the shape parameters. Optimal shapes of the bracket are obtained by using SOL200 of MSC/NASTRAN. The verification tests are conducted on the workbench and in a full vehicle. Test results show that the developed optimization procedure of the bracket are valid in the complex real world.

INTRODUCTION

Many peripheral pieces of equipment are attached on an engine block. Among them, the air-conditioner compressor is one of the largest components of the powertrain in a passenger car. An air-conditioner compressor-mounting bracket is a structural component of the engine, on which bolts attach the air-conditioner compressor. The compressor-mounting bracket supports the air-conditioner compressor and the dynamic characteristics of the compressor assembly mainly depend on that of the bracket. The compressor-mounting bracket experiences severe dynamic loads during the engine operation because the bracket is fixed by bolts directly to the engine block. From a designer's viewpoint, fatigue failure of the bracket should not occur during the car's life. Furthermore, resonant frequencies of the compressor assembly should not exist in the engine-operation range because vibrations of the compressor assembly can cause vibration and/or noise problems in the passenger compartment of the vehicle.

Nowadays, development of components using CAE tools such as the finite element method has been a common tool in early design stages [1,2]. In addition, the shape and topology optimization technique is becoming a useful tool. In this paper, the shape of the air-conditioner compressor-mounting bracket of a passenger car is optimized using commercial finite element software, and the optimized results are confirmed by tests.

ANALYSIS OF THE BRACKET

The bracket is bolted on the engine block and other bolts connect the air-conditioner compressor and the bracket. To operate the swash-plate-type compressor, the power of the engine is transmitted through belts, which connect the belt pulley of the compressor and crankshaft of the engine. In this study, the authors are interested in the design of the bracket and assume that the designs of the compressor and engine block are fixed.

There are two major quantities to be analyzed in order to design the compressor-mounting bracket. One is the resonant frequency of the compressor assembly and the other is the fatigue life of the bracket. First, the resonant frequency of the compressor assembly is very important
because the excitation frequency range of the engine is very wide and the resonance of the compressor assembly in the excitation range will cause many vibration and noise problems as well as fatigue fracture. To identify the resonant frequency of the compressor assembly in a supplier company, generally sine-sweep excitation tests are fulfilled with a specified excitation level. To calculate the resonant frequencies of a bracket in an early design stage, a finite element model is necessary. For the FE model, the engine block or excitation jig is assumed to be rigid and solid elements model the bracket in this study. The connection bolts are represented by equivalent elastic springs for each direction. The compressor consists of pistons, compression chambers, a rotary shaft, a clutch pad, an electromagnet, a belt pulley and a shell. Figure 1 shows a schematic diagram of the compressor assembly. The compressor operates by rotation of the belt pulley when the electromagnet pulls the clutch pad. To represent the dynamic characteristics of the compressor, a detail FE modeling or an experimental approach [3] is needed, however, this results in high computation costs in the design stage due to many design iterations. In this study a simplified compressor model is developed by identifying characteristics of the compressor assembly from tests. Figure 2 shows the response of a compressor assembly in the excitation test. From Figures 1 & 2 it is shown that the clutch pad and compressor axis affect very much the lowest resonant frequency of the compressor assembly. From the test results, the axis is modeled by beam elements and a concentrated mass represents the clutch pad at the end of the axis. The other structures of the compressor are considered as a rigid mass. Figure 3 shows a finite element model of the compressor assembly. The FE model of the compressor assembly is correlated with the results of the experimental modal analysis (EMA). The resonant frequencies of the compressor assembly identified by an impact hammer test and the FE model are tabulated in Table 1. In addition, Figure 4 shows the first mode shapes obtained from the test and the FE analysis, respectively. Table 1 and Figure 4 show that the developed FE model is valid for the compressor assembly.

<table>
<thead>
<tr>
<th>Mode No</th>
<th>EMA(Hz)</th>
<th>FEA(Hz)</th>
<th>Mode Shape</th>
<th>Error [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>239</td>
<td>234</td>
<td>Lateral mode</td>
<td>2.1</td>
</tr>
<tr>
<td>2</td>
<td>-</td>
<td>255</td>
<td>Vertical mode</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>328</td>
<td>318</td>
<td>Lateral mode</td>
<td>3.0</td>
</tr>
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</table>

Next, to identify the fatigue life of the bracket, a supplier company fulfills durability tests on a testbed. The durability tests consist of several excitation tests. A shaker excites the compressor assembly in three directions. The specification of the durability test regulates the time period and input acceleration level of the excitations as well as the excitation patterns. There are two excitation patterns in the durability specification: one is a sine-sweep over the frequency range and...
another is a fixed frequency excitation at the resonant frequencies of the compressor assembly. Generally the fatigue damages are accumulated mostly in the resonant-point excitations. To estimate the fatigue life of the bracket in the early design stage, dynamic stress analyses are necessary for the durability test. For the dynamic stress analysis, we used the same FE model of the resonant frequency problem and applied the large mass method [4] in commercial software, MSC/NASTRAN. For the purpose of verifying the stress analysis model, stresses of an existing bracket are calculated and compared with those of a measured one during the durability test. Figure 5 shows a test set-up to measure strains using the strain gauge. Figure 6 shows a comparison result of a selected point. In Figure 6, a measured strain is compared with the calculated strains at every node located within a hemisphere the diameter of which is equal to the length of the strain gauge because the measured strain represents an averaged value due to the finite magnitude of the strain gauge. It is shown in Figure 6 that the FE model represents the stress state of the bracket during the durability test.

SHAPE OPTIMIZATION OF THE BRACKET

The air-conditioner mounting bracket should have infinite fatigue life as well as light mass for fuel economy and minimum manufacturing cost. To insure the fatigue life a company has not only durability test specifications on the workbench but also a guide for the lower bound of the resonant frequency of the compressor assembly. To optimize the shape of a bracket, an optimization problem should be defined in standard form as follows:

\[
\text{Find the shape } \Omega \text{ such that}
\]

\[
\begin{align*}
\text{Minimize} & \quad f(\Omega) \\
\text{Subject to} & \quad g_i(\Omega) \leq 0, i = 1, \ldots, n
\end{align*}
\]  

Here, \( f \) is the objective function and \( g_i \) is i-th inequality constraint.

The authors select the weight of the bracket as an objective function because the material cost occupies a large portion of the total cost of production especially at the component level. In addition, minimization of the mass enhances fuel economy of the passenger car. The weight of the bracket can be calculated by summation of element weights as:

\[
f = \int_{\Omega} \rho dV = \sum_{i=1}^{NE} \rho V_{Ei}
\]  

where \( V_{Ei} \) is the volume of i-th element, \( \rho \) the density, and \( NE \) is the number of elements.

As a constraint of the shape optimization problem, the lowest resonant frequency of the compressor assembly should exceed a prescribed value. This constraint can be expressed as:

\[
g_i = \frac{\omega_1}{\omega_0} - 1 \leq 0
\]  

where \( \omega_1 \) is the first resonant frequency and \( \omega_0 \) is the prescribed value. Another constraint in the problem is that it must not fracture during the durability workbench test. Fatigue life can be estimated by accumulating damage using the Miner’s rule [5]. However, it is noted that in the durability test, most of the damage come from the resonant point excitation, which means the fatigue life of the overall test can be approximated by that of the resonant point excitation test. For a conservative design, the authors impose the infinite-life constraint on the resonant excitation. As a result, the constraints on the fatigue life is converted to stress constraints such as:

Figure 5. A test set-up to measure dynamic strains

Figure 6. A calculated strain compared to the measured one

Figure 7. Initial design of the bracket for shape optimization
\[ g_2 = \frac{\sigma_j}{\sigma_{\infty}} - 1 \leq 0, \quad j = 1, \cdots, m \]  

where \( \sigma_j \) is the Von-Mises stress of \( j \)-th node, \( \sigma_{\infty} \) is the fatigue strength and \( m \) is the total number of nodes.

A current-design (CD) bracket for a passenger car is selected as a target bracket to be optimized. The air-conditioner compressor-mounting bracket is equipped on an inline four-cylinder engine and is made by casting aluminum. Similar to the current design, an initial shape for the optimization is selected as shown in Figure 7. The bracket is modeled by 2393 solid elements and 3621 nodes. The design variable of the problem is the shape of the bracket. Here, the shape of the upper and lower surfaces, the radius of the outer bolthole 6, and the shape of the Side2 surface in Figure 7 are selected as the design variables. The shapes selected as the design variables are controlled by the position of the nodes. For the upper, lower, and the Side2 surfaces, every node on the surfaces controls the surface shapes by movements perpendicular to each surface. Internal FE nodes move proportionally to the surface movements of the design variable nodes. The number of the design variables is 1072 for the bracket problem.

The shape optimization problem is solved by SOL200 of MSC/NASTRAN. Here, two cases are examined which have identical formulation except the prescribed resonant-frequency constraint value, \( \omega_1 \). Table 1 summarizes the results. It is noted that CASE 2 does not satisfy the resonant frequency constraint at the initial

<table>
<thead>
<tr>
<th></th>
<th>Initial</th>
<th>CASE1</th>
<th>CASE2</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \omega_1 ) (Hz)</td>
<td>241.7</td>
<td>241.6</td>
<td>249.1</td>
</tr>
<tr>
<td>( \omega_2 ) (Hz)</td>
<td>-</td>
<td>240</td>
<td>250</td>
</tr>
<tr>
<td>( f ) (Kg)</td>
<td>1.04</td>
<td>0.617</td>
<td>0.915</td>
</tr>
<tr>
<td>No. of Iter.</td>
<td>-</td>
<td>28</td>
<td>5</td>
</tr>
</tbody>
</table>

Figure 8. The shape optimization results of the bracket

Figure 9. Stress contours of the bracket

Figure 10. History of the object function for the case 1
design. Figure 8 shows the optimized shapes for each case. Figure 9 shows stress contours of the initial and the optimized shapes for CASE 1. From the figures, it is shown that the highly stressed parts of the initial design are reinforced during the design iterations, and conversely the lower stressed parts are shrunken. Figure 10 shows a history of the objective function during design iterations for CASE 1.

The shape optimization problem of the bracket is a very large one, which has one thousand design variables and one modal analysis and three frequency response analyses. For an example, determining the results for CASE 1 took 9480 sec of CPU time in HP X2000 Win2000 workstation with one 1.7 GHz Pentium 4 processor, but more than two days in real elapsed time. Introducing control parameters of the surface patches as design variables will reduce the solving time of the optimization problem. However, this may increase the total engineering time. Therefore, some trade-off is needed in selecting the design variables.

**PROTOTYPING AND VERIFICATIONS**

Based on the optimization results of the previous section, a prototype bracket is manufactured and tested. The optimized shape of CASE 1 is selected as a prototype because CASE 1 has a lighter weight than the current product but has a similar resonant frequency.

![Figure 11. A simplified bracket for the manufacturing](image)

Considering a manufacturing stage, however, the optimized surfaces of the bracket have too many curvatures, which results in higher manufacturing costs because of the high machining cost and shorter life of die casting mold. To reduce the manufacturing cost, the optimized shape of the bracket is simplified with flat surfaces when possible. In addition, shapes around bolt holes are modified in order to enhance the efficiency of the assembly work. Figure 11 shows the final shape of the bracket.

![Figure 12. Results of impact hammer test of the compressor assembly](image)

The simplified bracket is prototyped by an aluminum sand casting method. The weight of the final product is 0.730 kg, which is a 30% decrease compared to the current bracket. Dynamic characteristics of the prototype bracket are tested by an impact hammer test for compressor assembly and compared in Figure 12. The resonant frequencies of the prototype bracket are slightly lower than the current-design one in Figure 12. However, considering the differences of manufacturing procedure of the two brackets, it should be mentioned that the prototype bracket has almost equivalent dynamic characteristics to the current-design bracket.

![Figure 13. The second harmonic components of acceleration to crankshaft revolution on the bracket during operation](image)

In terms of the durability, the prototype brackets have passed the standard durability test on a test bench without failure. Figure 13 shows acceleration level in z-direction on the bracket of a vehicle equipped with the prototype bracket and the current-design one, respectively. As shown in Figure 13, two brackets are under the very similar loading conditions during operation.
To confirm performances of the prototype bracket, vehicle tests on noise and vibration are fulfilled. From the vehicle tests, we can see that two brackets have equivalent performance in vibration and noise level for each operational mode. As an example, Figure 14 shows acceleration levels on the prototype bracket during operation compared to those on the MP bracket.

CONCLUSION

The shape of the air-conditioner compressor-mounting bracket of a passenger car is optimized using a finite element software package, and the optimized results are verified by tests. The cost function is the weight of the bracket. The first resonant frequency and durability are considered as the design constraints. The constraints on durability were transformed into the stress constraints. The bracket is modeled by solid elements and the swash-plate-type compressor is represented by rigid masses and beam elements in order to consider the elastic effects of the compressor. The design variable is the shape of the bracket such as the thickness profiles of the front and back surfaces of the bracket, radius of the outer bolt holes and, the side edge profiles. The optimal shapes of the bracket are obtained by using MSC/NASTRAN.

The optimized bracket is manufactured and verification tests are fulfilled. The shape of the bracket is simplified to take manufacturing costs into consideration. The verification tests are conducted on the workbench and in the full vehicle. Test results show that the developed optimization procedure of the bracket are valid in the complex real world.

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REFERENCES


Figure 14. Acceleration levels on the bracket during operation

(a) At idle condition

(b) 80 km/h on flat road

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