Enhancement of Vibration Characteristics of an Air Filter Box Utilizing Numerical Analysis

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Abstract: Air intake system (AIS) that prepares clean air for engine aspiration, plays an important role in the performance of the internal combustion engines. It contributes in different aspects of engine's attributes such as fuel consumption, engine power, pollution, noise, and vibration. Powertrain system is known as one the main sources of the noise and vibration in the vehicle. So, enhancement of the engine noise, vibration, and harshness (NVH) characteristics provides passenger's comfort. AIS is one of the components that generates significant noise and vibration due to its connection to the engine and passing transient air flow. Therefore, tuning this system decreases vehicle sound and vibration level noticeably. In this paper, in the first step modal analysis of a concept filter box is conducted and some modifications are proposed to enhance the vibration performance of this system. At the next stage, frequency response function (FRF) analysis is performed to survey sensitivity of the air box structure caused by these modifications and finally suitable air filter box is chosen. To this end, finite element method (FEM) is applied.

Keywords: Air intake System, Engine Vibration, Finite Element Method.

1. INTRODUCTION

Noise, vibration, and harshness (NVH) are among the most important attributes in development process of the vehicle. Different studies are conducted to improve the noise and vibration level of the vehicle [1-4]. In recent decades, numerical method is known as a powerful instrument to simulate vibration and acoustic phenomenon. Numerical simulations not only reduce the time and cost of the development process, but also they prepare a good perspective to find NVH solutions. There are many vibro and areo-acoustic studies using numerical simulations [5-10]. The reduction of the intake system noise can significantly improve vehicle interior acoustics and pass-by noise. There are some studies in order to investigate air intake performance in aspect of noise and vibration view. Siano and D'Agostino [11] have performed a numerical simulation using finite element method on air intake to find a geometry with optimum acoustic performance. Their simulations had a good agreement with experimental test. They achieved an air intake geometry with appropriate acoustic performance which had no significant volume change in comparison to the original one. Batooei and Beigmoradi [12] have studied the effect of resonator size on the air intake noise. They have used computational fluid dynamics (CFD) to investigate the flow pattern variations into air intake caused by changing resonator size. Then they proposed the appropriate dimensions of the resonator. Bourdon et al. [13] employed finite element method to study the radiated noise of the air filter box and verified their simulations by experimental tests. They evaluated different propagation paths of noise separately in their tests. Das et al. [14] designed and optimized air intake snorkel to meet attributes of the temperature, noise, and water ingress. They used CFD method to simulate the temperature and water patterns around the vehicle. They also utilized 1D simulation to predict the transferred noise from snorkel. Arnault et al. [15] explained the process of optimal design of the air intake system. They introduced a methodology to cope with the limitations of acoustic emissivity, pressure drop, temperature, performance, and packaging. Arunachalam et al. [16] perused the effect of AIS elements such as resonators, manifold and air filter to tune up noise characteristics utilizing 3D analysis. They improved the acoustic performance by adding the resonator and changing the snorkel shape. Ghodake and Haque [17] predicted the pulsed and flow noise of AIS by coupling 1D and 3D simulations. They used the engine pulsations from GT-Power 1D software as an input boundary condition to the ANSYS FLUENT 3D software. Then they solved transient 3D simulation to derive pressure signals. Gosain et al. [18] surveyed the development of an intake system by considering some attributes such as the NVH, fuel efficiency, torque, power output, and cost. They applied experimental modal analysis and finite element method to this end. In none of the above mentioned works structural vibration of air intake is investigated. Due to the connection of the air intake to the engine, some of the engine vibrations are transferred to the air intake system. Not only these vibrations can be one of the sources of the noise, but they also can lead to the failure of the structure and its connected components. In this paper, vibration characteristics of air filter box for different structures is investigated. For this matter, commercial finite element code is applied.

2. MODEL DESCRIPTION

In this work, vibration analysis of a filter box, which is designed for a MPFI 1650cc engine, is investigated. Figure 1 shows the primary concept of the filter box. In the concept design phase, different aspects of the attributes such as packaging, engine performance, and NVH are considered and according to these targets primary shape of the filter box is created.

Figure 1. Air filter box

This box is made of polypropylene with 20% talc. Table 1 shows the material characteristics of the filter box.

Table 1. Material characteristics of polypropylene with 20% talc

Properties	Unit	Value
Density	Kg/m^3	1050
Melt Flow Rate	g/10min @Load 2.16 Kg Temperature 230^{0}c	10
Tensile Strength	MPa	300
Flexural Modulus	MPa	27,000

Finite element analysis is conducted on the primary concept to derive mode shapes and frequency values of the filter box structure. To this end, shell elements are applied to mesh the structure of the box. There are some criteria for meshing the model shown in Table 2. CQUAD4 & CTRIA3 are applied as element types. The element size of the filter box is chosen 4mm.

Table 2. Element criteria for meshing

criteria	value
Triangle element (%)	10
skewness	45
warping	10
Aspect ratio	3

Upper and lower box are fixed to each other from the considered fixing points using rigid body element of form 2 (RBE2). Constrained modal analysis of the structure is conducted to extract the frequency mode shapes. All degrees of freedom of the fixing points of the box to the body considered to be zero. To this end, single point constraint (SPC) is applied in fixing points. To obtain natural frequency modes SPC is omitted from the model to simulate free-free condition. In order to perform the modal analysis Lanczos algorithm is applied to extract the resonance mode shapes and frequencies. To reduce the time and cost of the simulations the first ten frequencies are considered for vibration evaluation of the filter box. Figure 2 shows the finite element model which is used for numerical simulations.

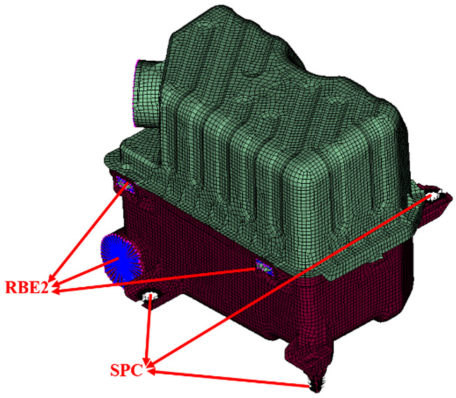

Figure 2. Finite element model of the filter box

3. RESULTS AND DISCUSSION

3.1. Primary assessment of the base model

Modal analysis of the filter box is done to achieve ten primary mode shapes and frequencies of the concept structure.

As it is shown in Figure 3, the first frequency value of the filter box is about 87Hz. Moreover, for the first three mode shapes, the upper box has the most displacements in the air box system. In the fourth mode, both upper and lower box have the same vibration. In the fifth mode, the lower box has more contribution to vibration than the upper box. In the sixth mode, the upper box, and in the seventh mode the lower box has the most vibration. In the three later modes, both upper and lower boxes have resonance.

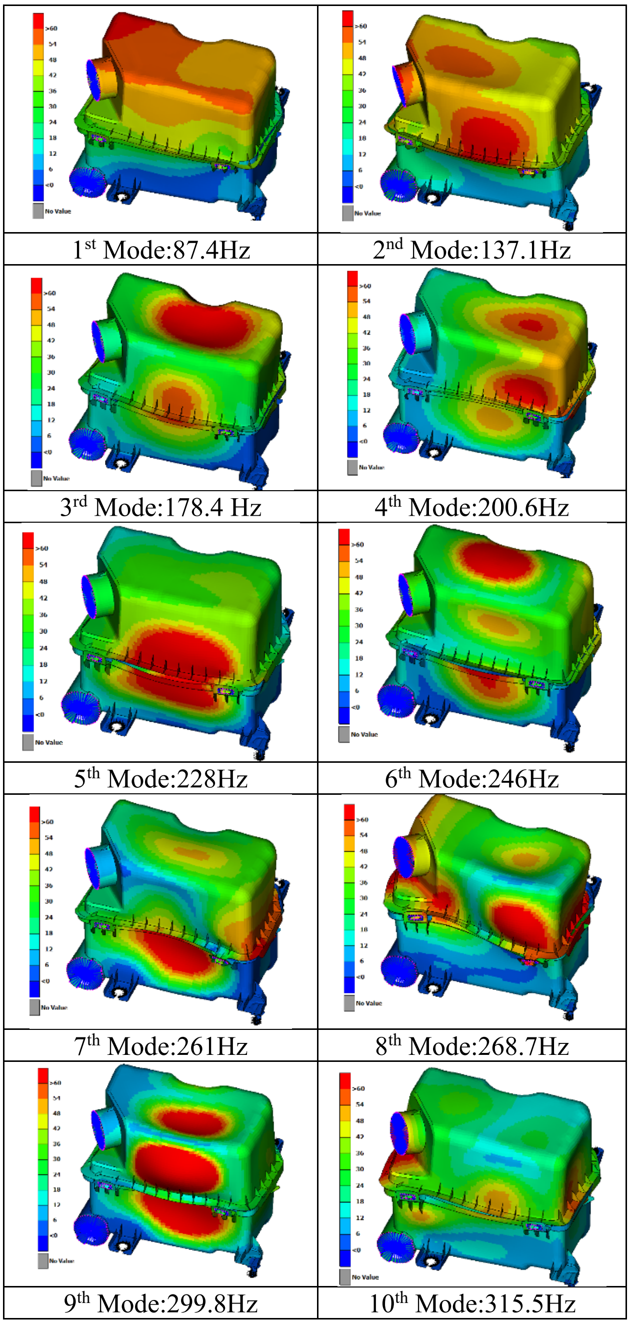

Figure 3. Frequency and mode shapes of the filter box

3.2. Experimental validation

To investigate the accuracy of the numerical results, the experimental tests are conducted on the basic model. For this purpose, filter box structure is suspended by elastic rope to provide the free-free condition (Figure 4) and then experimental modal analysis is performed to find out the resonance frequency values. To carry out the free-free modal analysis, roving hammer test is the most common type of impact test to determine modal characteristics. An accelerometer is fixed at a single degree of freedom (DOF), and the structure is impacted as many DOFs as desired to define the natural frequency mode. Using a FFT analyzer, frequency response functions (FRFs) are computed, at a time between each impact of DOF and the fixed response of DOF to find out frequency modes. The charge type hammer (Type 8202, B&K) is used to excite the filter box. To cover the required frequency ranges, a rubber tip has been utilized. Excitation points are located near the accelerometers. Six measurement points are selected on the center of the filter box walls and the tri-directional accelerometers (type 356 A11, PCB) are attached to these locations. In order to reduce the noises in the measurements, the results of each stage have been obtained by averaging five measurements of the same kind. Here, the most general parameter estimation technique, called time domain multi degree of freedom (MDOF) Analysis, has been employed. It provides a complete and accurate modal model which is achieved by multiplying a single input to the output frequency response functions. It uses global estimators; means it analyzes all the recorded data simultaneously in order to estimate the characteristics of the structure. With this approach, a unique estimation of the pole values (natural frequencies) has been obtained. To gather data from sensors, a Multi-channel data acquisition system (Difa Measuring System, Type ScadasII) is applied. Moreover, LMS CADA-X/ Modal Analysis is used to compute the resonance frequency.

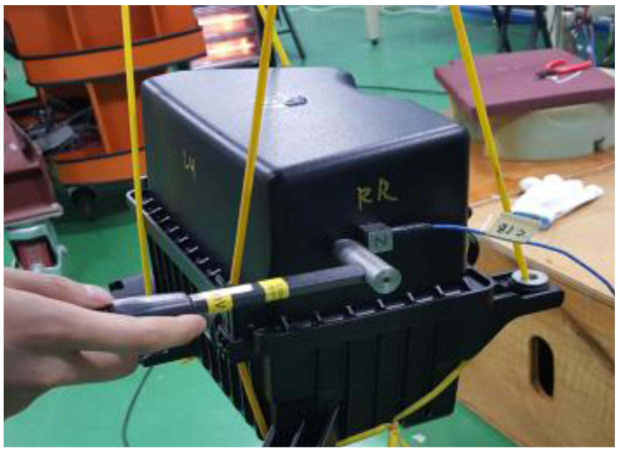

Figure 4. Setup test for free-free modal analysis

The results of the free-free resonance frequency values are depicted in Table 3. As the first six modes are rigid body with frequency about zero, they are omitted from Table 3. As shown in Table 3, numerical results have close agreement with experimental tests. The maximum difference between the experimental and numerical results is about 4.6%, which is acceptable in the aspect of the engineering view.

Table 3. Comparison between experimental and numerical results

Mode number	Numerical	Experiment
7th	154 Hz	161 Hz
8th	217 Hz	225 Hz
9th	235 Hz	246 Hz
10th	247 Hz	258 Hz

3.3. Enhancement of the filter box design

In detail design phase, the filter box is modified to improve the vibration characteristics without considerable effect on the other attributes such as volumetric efficiency. For this matter, four proposals are suggested in this work. In the first and second proposals stiffeners are added to the upper and lower filter box, respectively. In the third proposal a filter box with full stiffeners is studied and in the last proposal beads are added to the filter box. Results of the first ten resonance frequencies in the constrained modal conditions for base case and four filter box proposals are illustrated in Table 4.

Table 4. Resonance frequencies for base case and four proposals

Mode Number	Base Case (Hz)	Upper Box with Stiffeners (Hz)	Lower Box with Stiffeners (Hz)	Box with Stiffeners (Hz)	Box with Beads (Hz)
1st	87.4	82.4	89	84.2	87.5
2nd	137.1	130.8	142.8	136.1	138.8
3rd	174.8	175.9	176	176.8	176.7
4th	200.6	200.6	208	209.1	205.3
5th	228	228.4	245.9	256.1	233.3
6th	246	260	259.7	267.6	253
7th	261	266.9	269.1	283.5	263.1
8th	268.7	278.6	295.3	297.3	273.1
9th	299.8	314.6	309.5	339.8	318.4
10th	315.5	338.2	333.6	362.2	326.2

Figure 5 shows the first mode shape of the four proposals in the constrained condition simulation.

According to the results of Figure 5, even though adding stiffeners to the upper box decreases the displacements of this part, it also decreases the frequency value of air box system which has negative effect on powertrain NVH.

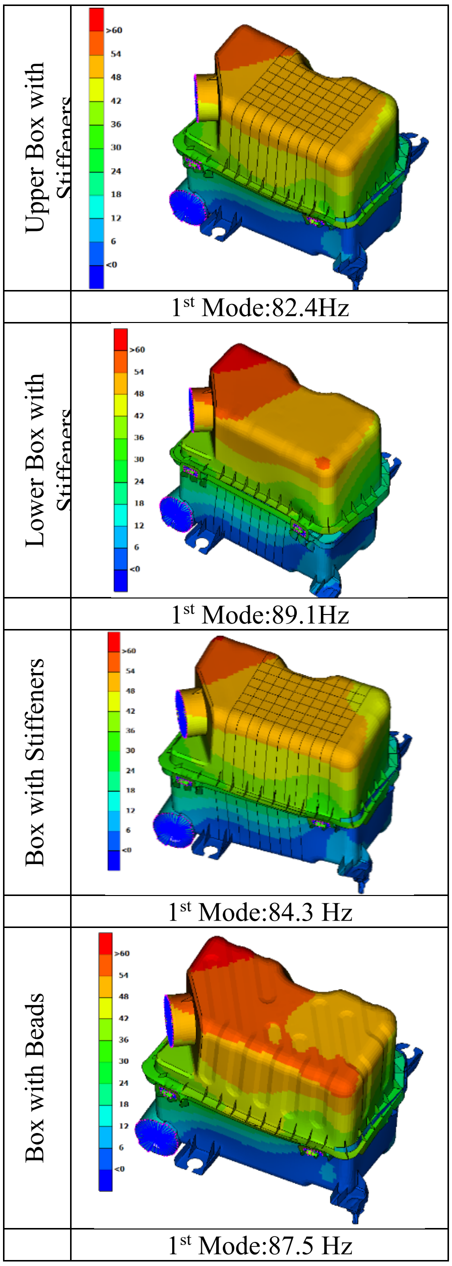

Figure 5. First mode shape of the proposed filter boxes

Decreasing the frequency value of the filter box causes the system frequency approaches to the engine working orders. It can be concluded that the effect of mass due to adding stiffeners outweighs the effect of the stiffness. For the lower box, adding stiffeners leads to the increasing of the resonance frequency about 2Hz. Adding stiffeners to the lower and upper parts simultaneously, decreases the first frequency of the resonance about 3Hz, which shows that the effect of the stiffener mass dominates the effect of stiffness. In this case, displacement of the parts due to the resonance is significantly lower than the base case. By creating beads on the upper and lower parts, as shown in Figure 5, the resonance frequency does not change in comparison to the base case. Moreover, It seems that displacement of the parts is as much as the base model.

After modal analysis, frequency response function (FRF) analysis is done to evaluate the structural sensitivity of the box caused by the excitation force from the engine. This analysis can provide an appropriate estimation from each modification which is done to enhance vibration characteristics. To this end, unit dynamic forces in three directions are simultaneously imposed to the connecting side to the engine and the responses are achieved on the entrance side. In this research, acceleration is chosen as the response parameter. As it is depicted in Figure 6, acceleration in x direction for a fully stiffed model is overally smaller than other cases in different frequency except 315Hz. In the next rank, a model including beads has smaller vibration response in x direction in comparison to the three other cases.

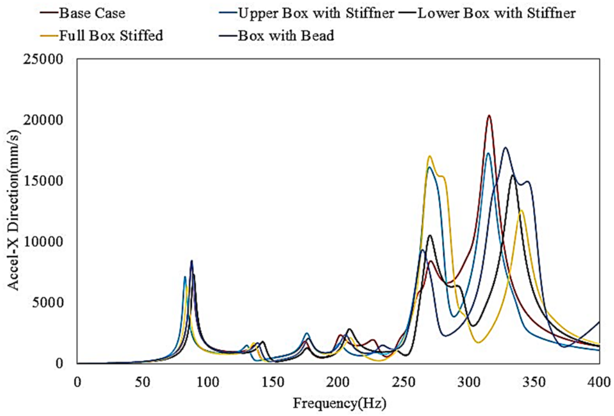

Figure 6. Response frequency in X direction

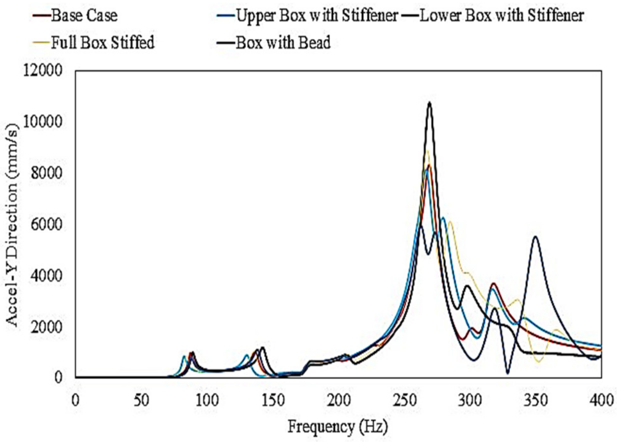

Figure 7. Response frequency in Y direction

According to Figure 7, the acceleration of the response point for the model with beads has a lower vibration level compared to the other models in the frequency range except for about 350Hz. This figure shows that none of these modifications have a significant effect on y direction responses for the low range frequencies.

Regarding to Figure 8, acceleration response in z direction for fully beaded box in the high range frequency is located in a more reasonable range in comparison with other case studies. So, by compromising all these cases it can be concluded that a fully beaded filter box is a suitable solution for using along with this engine.

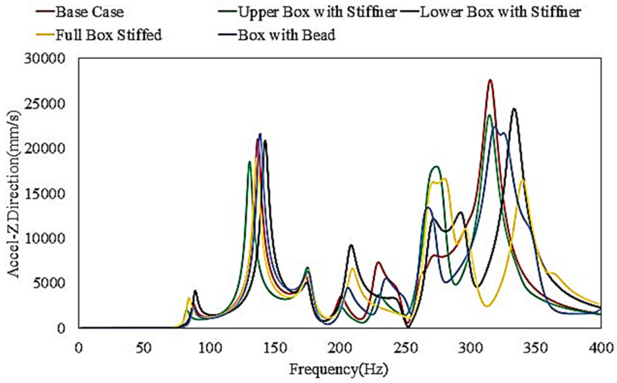

Figure 8. Response frequency in Z direction

4. CONCLUSIONS

In this paper the vibration characteristics of an air filter box is investigated. Then some modifications are proposed to enhance the NVH attribute of the powertrain system. After all simulations, it is concluded that modifications usually influence the mid to higher range of the frequencies. At the end, beaded model is chosen as the suitable option with considering vibration feature.

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