International Journal of Automotive Engineering, Vol. 8, No. 2, (2018), 2692-2699

Iran University of Science and Technology

International Journal of Automotive Engineering

Journal Homepage: ijae.iust.ac.ir

ISSN: 2008-9899

Designing a buffer-equipped hydraulic engine mount for mid-priced vehicles

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ARTICLE INFO

Article history:

Received : 13 Jan 2018

Accepted: 10 Jun 2018

Published:

Keywords: (Times New Roman 9 Bold)

Keyword 1 (Times New Roman 9 Regular)

Keyword 2

Keyword 3

Keyword 4

Keyword 5

A B S T R A C T

A combined hydraulic engine mount and buffer absorber are proposed in this study for use in the mid-priced vehicle. In some vehicle design projects, an engine is selected to use in new car design. To achieve a proper vibration, the engine mount can be redesigned with exorbitant costs and long-term research. The idea of using a buffer in the combination of the conventional engine mount is to suggest a solution with affordable price which can improve mount vibration specifications. As a case study, the engine of Renault L90 (Dacia Logan), which name is K4M engine, is selected to use in the national B class automotive platform design. This automotive platform has been designed at Automotive Engineering Research Center of Iran University of Science and Technology. The hydraulic engine mount is modeled in CATIA. Some tests are done to validate the simulation results. Both engine mount characteristics (with and without buffer) are imported to Adams/Vibration software to evaluate the vibration behavior of the engine mount system. CATIA software is used to determine the characteristics of the engine mounts. The results show that the use of buffer absorber reduces the stiffness of mount, which should be 2 to 3 times lower than engine's frequency excitation. In some directions, the buffer-equipped mount has a better modal energy and isolation characteristics.

1. Introduction

Mostly three or four engine mounts are used to install an internal combustion engine on the body of a vehicle. Always, two mounts bear the weight of the engine and reduce the transmitted vibrations. Meanwhile, the third engine mount (in the case of using three mounts) and the third and fourth engine mounts (in the case of four mounts) have the responsibility of reducing the torsional vibrations of the engine. The researchers carried out in modeling and simulation of the engine mount is of great importance for reducing the mutual vibrations between the engine and the chassis. In this section, some relevant research works are studied.

Kim and Singh in a comprehensive study identified and introduced the time-dependent parameters for a hydraulic mount. They have also proposed a model of these systems with variable portability and control of components as transient and adaptable engine mounts [1]. Golnaraghi and Nakhaie also assessed the mount systems used in the automotive industry. They presented both historical and too new types of research on engine mounts and showed that the optimum performance of the mounts with improvements

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features in frequency and domain of stiffness and depreciation is achievable. To attain this goal, passive, active and semi-active mechanisms have been developed. Moreover, they discussed the required different features of engine mounts for vibration isolation and displacement control of the engine and the chassis [2]. Geisberger and his colleagues' experiments on a sample system show that at high frequencies the behavior of the hydraulic mount is extremely nonlinear which do not match with their theoretical results at high frequencies. They have formed the equations of fluid momentum between the separator and the inertia track assuming a one-porosity's system in two stimulation modes of the small and large domain. Therefore, their model for justifying the simultaneous behavior of the components is inappropriate. They pointed out in their results that the new model is unable to describe the nonlinear behavior of the linear model at frequencies above 250 Hz. They considered that the isolation switching is the reason of this [3]. Adiguna and his colleagues in their recent research have examined the transient response of a hydraulic engine mount in both analytical and laboratory methods and justified the exact behavior of the isolation switching's mechanism. Their analytical results are very consistent with the experimental performance's results of the hydraulic mount. They did not investigate the stimulation frequency response, but by using the analytical method, they proved the nonlinear behavior of the linear system's performance at high frequencies [4]. Jeong and Singh also to determine the stiffness, position and setting angle of the engine mount using the method of complete separation of vibration modes of engine around the rotation torque axis [5]. Zavala and his colleagues with both experimental validation and also the simulation methods developed a diesel engine mount system to improve the vibration behavior of a vehicle [6]. Suye and his colleagues in research optimized the engine mounts based on the separation modes in the direction of the rotation torque axis. For this purpose, some dynamics and static analyzer software, such as Adams and Engineers/Insight have been used [7].

In 2013, Fakhari and Ohadi at Amirkabir University of Technology researched a part of the national engine and especially on their holders with experimental validation. In this paper, a simulation of a six degree of freedom for the national engine on its holders (mounts) was presented. The simulation validation was performed by performing experimental tests. Dynamic equations are extracted by using Lagrange and Newton Euler methods. Also, to the dynamics of the body, wheels and the suspension system in the vibrational behavior of the engine be taken into account, a thirteen degree of freedom simulation was used for the engine and the vehicle. Simulations were performed using national engine's data of the Samand Vehicle. Experimental tests were also done on the roller brake legs of Samand which was equipped with the National engine. Finally, the validity of the dynamic simulation was presented, and the effect of considering the dynamics of the body, wheels and suspension system on the vibrational behavior of the engine is examined [8]. Reza Takini and his colleagues by extracting the state space equations using the Gradient bond method developed a linear model of hydraulic engine mount with two frequencies. Considering the nonlinear parameters in the governing equations, simulation in the time domain and then using the Variance method simulation in the frequency domain for the nonlinear of this hydraulic mount was done and the results were compared with the linear model [9]. Vertical stiffness and natural frequency are two main parameters in the design of the mounts. The engine mount's stiffness must be such that it does not have much displacement during the bump and does not disturb the car's performance. The first natural frequency according to the design requirements must be 2 or 3 times smaller than the stimulation frequency to prevent resonance phenomenon [10].

Since each engine has its weight, inertia momentums and operation rpm, it is also necessary to modify the mount by changing the engine. Regarding design costs and time-consuming of it, adding a buffer absorber with variable thicknesses and by regulating the stiffness and natural frequency types, it is possible that the engine mount matches with the new engine. In this paper, changing the stiffness, natural frequency and energy modes of a mount equipped with a buffer layer is studied.

In the next section, the specifications of the engine mounts and buffers are introduced. The experiment setup and test procedure for hydraulic engine mount's stiffness determination are discussed in Section 3. In this paper, the engine mount of L90's engine is examined. In Section 4, the dynamic modeling of the system in Adams/View software is explained. The analysis of the natural frequency of engine system and investigating the improvements are studied in Section 5.

2. Engine mounts and buffers' specification

The Renault L90 combustion engine has two rubber mounts and one simple hydraulic mount as shown in Figure 1 and Figure 2.

Figure 3 shows the used buffer absorber. The characteristics of this buffer are also given in TABLE 1.

Figure 1: hydraulic engine mount (L90)

Figure 2: Rubber mount (L90)

Figure 3: Buffer absorber

The main idea of this paper is the hybridization of a hydraulic engine mount with some excess rubber (buffer-equipped hydraulic engine mount) with better vibration specifications for a specific engine, without redesign and refabricating the hydraulic engine mount for a variation engine family. It is a kind of platform design for the economic production process.

The primary governing equation for a hybrid buffer hydraulic engine mount can be stated as follows [11-12].

$$F_T(t) = K_r x + B_r \dot{x} + A_p P_1 \tag{1}$$

were F_T is the transmitted force to the engine mount, B_r and K_r are dynamic damping and stiffness properties of the buffer absorber, respectively. A_p is the effective piston area, and P_1 is the pressure in the upper chamber of the hydraulic mount. In this paper, the parameters of the governing equations are determined by some

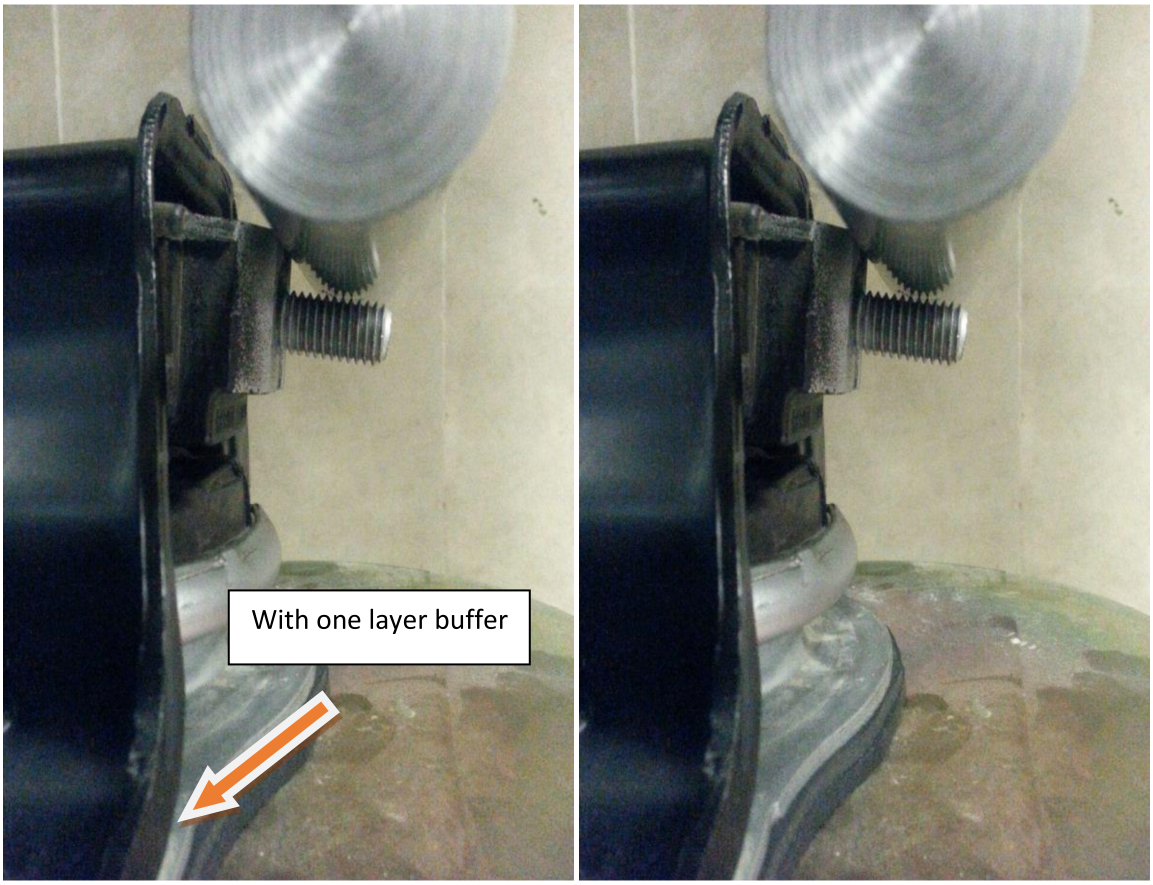

Figure 4: Load applied to the hydraulic mount with buffer and the mount without buffer

experimental processes which are discussed in the next sections.

TABLE 1: Buffer's Characteristics

Tensile Strength	Elastic Modulus	Poisson's Ratio	Density
$5e^{+006}$ N/m^2	$2e^{+007}$ N/m^2	0.48	930 kg/m^3

3. Hydraulic engine mount's stiffness determination for the combustion engine of L90

To determine the stiffness of the mount, a force-displacement curve is required. For this purpose, a tension-compression testing machine is used. First, the hydraulic engine mount by fittings is fixed to the lower jaw and then by the upper jaw the force is applied to the bolt which connects the mount to the body. It is a downward compressive force. The force applied to the hydraulic engine mount and also the upper jaw's displacement is measured by the device sensors. In this way, the force-displacement curve is derived, and the stiffness of mount is calculated from this curve. In this test, the force is applied gradually (in several steps) to the hydraulic mount, and in each level, the force and displacement values are extracted from the device. Then, this test is performed again by setting the buffer absorber in the bottom of the engine mount, and according to before, the stiffness of the mount equipped with the buffer is calculated. Figure 4 shows the test performed on both the first engine mount and also the mount fitted with the buffer absorber. Also, the force-displacement diagram of the engine mount's analysis with or without the buffer is shown in figures 5 and 6. For guarantee the results, any tests are repeated five times, and the results are the same.

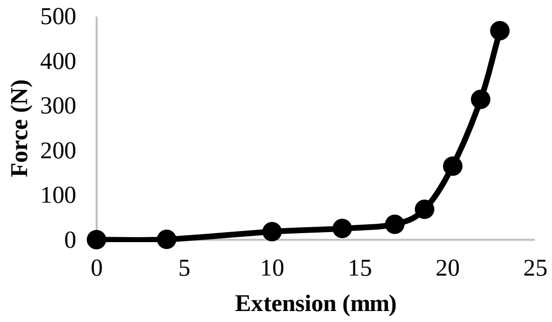

Figure 5: Output diagram of the hydraulic mount without the buffer absorber

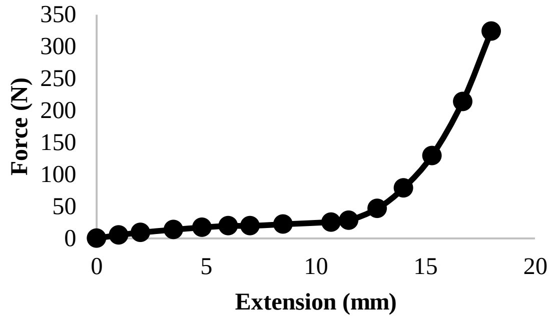

Figure 6: Output diagram of the hydraulic mount equipped with the buffer absorber

Given that the engine mount's stiffness is the same as the force-displacement curve's slope; the stiffness for both cases is calculated by the equations (1) and (2).

The stiffness of the mount without the buffer:

$$\frac{300-100}{22-19}=\frac{200}{3}=66.66\,{}^{kN}\!/_{m} \tag{2}$$

The stiffness of the mount with one layer buffer:

$$\frac{300-100}{18.3-14.8}=\frac{200}{3.5}=57.14\,{}^{kN}\!/_{m} \tag{3}$$

Correspond to the final results, by adding the buffer absorber the engine mount's stiffness in decreased by 14.2%.

4. Dynamic modeling in Adams/View software

Using the information shown in TABLE 2 and TABLE 3 and also the stiffness obtained from the experimental method, a dynamic model for the mount of the L90's combustion engine was created in Adams/View software. Figure 7 represents an illustration of the model. In this model, two elastomeric engine mounts and as well as one hydraulic mount were combustion engine of L90. The purpose of the performed modeling is to investigate the energy modes and the natural frequency. The setting location of the engine mounts is usually determined by the axial position called the rotation torque axis. If the torque is applied around the axis of engine's crankshaft and also the engine is not placed on any bearing; the engine's rotation would be around its axis. The position of this axis depends on the mass center position and engine's mass tensor moment of inertia. If the mounts are placed on this axis, the applied force to them would be less, and the engine would have a better vibration behavior. Therefore, it is always tried to place them on this axis as much as possible. Of course, because of location's limitations, it is not still possible. In this paper, to carry out the corresponding analyzes, as shown in Figure 8 the x-axis is in the longitudinal direction, the y-axis is in the transverse direction and the z-axis is in the vertical direction of the vehicle.

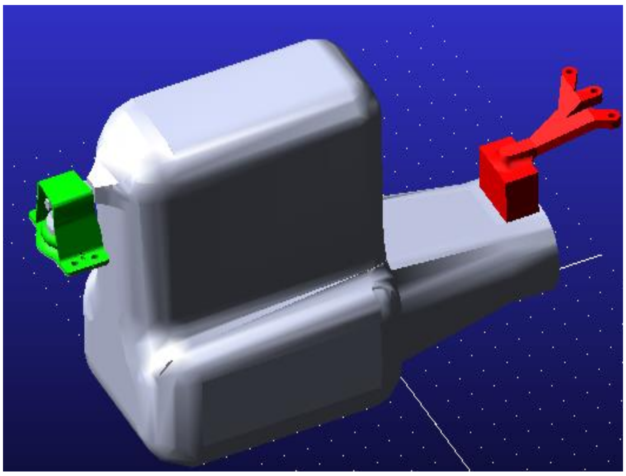

Figure 7: Illustration of the mounts' position (L90) in Adams software

The governing equations of the engine mount system are as follow [10].

$$M\,\ddot{x}_c = F_x \tag{4}$$

$$M\,\ddot{y}_c = F_y \tag{5}$$

$$M\,\ddot{z}_c = F_z \tag{6}$$

$$I_{xx}\ddot{\alpha}_c - I_{xy}\ddot{\beta}_c - I_{xz}\ddot{\gamma}_c = M_x \tag{7}$$

$$I_{xy}\ddot{\alpha}_c - I_{yy}\ddot{\beta}_c - I_{yz}\ddot{\gamma}_c = M_y \tag{8}$$

$$I_{zx}\ddot{\alpha}_c - I_{zy}\ddot{\beta}_c - I_{zz}\ddot{\gamma}_c = M_z \tag{9}$$

were x_c, y_c and z_c are the engine system center of gravity locations; α_c, β_c and γ_c are the rotational angular accelerations around the x-axis, y-axis, and z-axis, respectively; F_x, F_y, and F_z are the net forces in the three primary directions and M_x, M_y and M_z are the net torques around the x-axis, y-axis, and z-axis, respectively.

For calculating the natural frequencies and modes, the equations become as follow [10].

$$([K] - \omega^2[M])\Phi = \{0\} \tag{10}$$

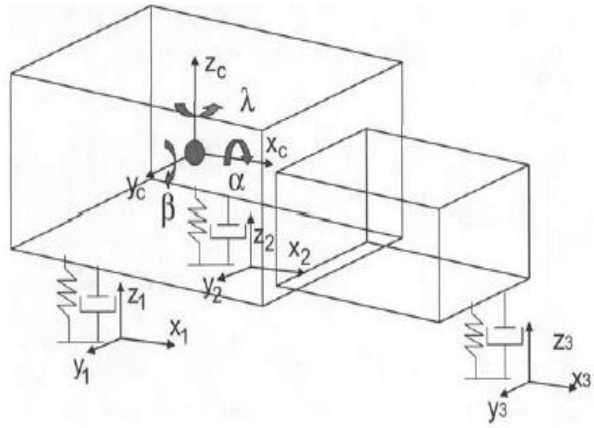

Figure 8: The engine mount system [10]

were $[K]$ is the stiffness matrix of the engine mount system and $[M]$ is the mass matrix of the system. In this situation, Φ is the essential mode matrix of the mount system, and ω is the natural frequency matrix.

In this paper, the natural frequencies and modes are determined by ADAMS/Vibration, which are reported in the next section.

TABLE 2: Location of mounts (L90)

	Mass center of the power transfer system	Location of the right mount	Location of the left mount	Location of the bottom mount
X[m]	0.842	0.844	0.798	0.961
Y[m]	0.0625	0.47	-0.387	0.198
Z[m]	0.5096	0.751	0.538	0.297

TABLE 3: Parameters of L90

215 kg	$\boldsymbol{M}$	**Engine mass**
4.235 kgm^2	$\boldsymbol{I_{xx}}$	**Moment of inertia**
9.371 kgm^2	$\boldsymbol{I_{yy}}$	**Moment of inertia**
9.371 kgm^2	$\boldsymbol{I_{zz}}$	**Moment of inertia**
8.288 kgm^2	$\boldsymbol{I_{xy} = I_{yx}}$	**Moment of inertia**
0.908 kgm^2	$\boldsymbol{I_{yz} = I_{zy}}$	**Moment of inertia**
0.948 kgm^2	$\boldsymbol{I_{xz} = I_{zx}}$	**Moment of inertia**
3.716 kgm^2	$\boldsymbol{I_{11}}$	**Main moment of inertia**
9.942 kgm^2	$\boldsymbol{I_{22}}$	**Main moment of inertia**
8.229 kgm^2	$\boldsymbol{I_{33}}$	**Main moment of inertia**

In TABLE 5, the energy modes of the mount without and equipped with a buffer absorber are visible. As can be seen, it is better than the energy mode of an engine mount be as close as possible to 100 [10]. Moreover, by adding a buffer, energy modes in four degrees of freedom are improved and in two directions are reduced. One of the design requirements of an engine mount is that neither of the energy modes must be less than 85 [10], which according to the results, this requirement has been satisfied.

TABLE 5: Values of energy mode's improvements in the ADAMS software

	X	Y	Z	RXX	RYY	RZZ
Without buffer	86.83	90.89	93.81	90.77	90.95	98.95
With buffer	88.80	91.22	92.04	91.79	92.55	98.37
Values of energy mode's improvements	2.2%	0.36%	-1.9%	1.1%	1.7%	-0.6%

As an illustrative result, a bump test for a sedan vehicle is performed in CarSim software, with and without buffer for the engine mount. Figure 9 shows the pitch angle of the engine without (Vehicle #1) and with (Vehicle #2) buffer mount. As seen in this figure, the pitch angle of the modified engine is better than of the conventional engine (about 31%).

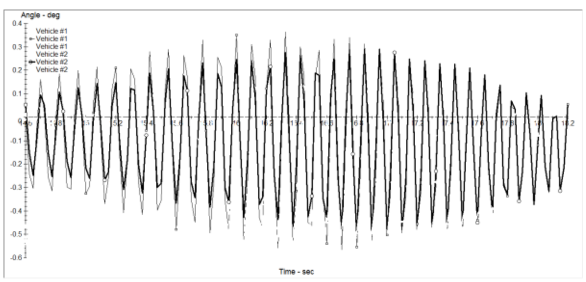

Figure 9: The pitch angle of the engine without (Vehicle #1) and with (Vehicle #2) buffer mount

5. Conclusions

The primary aim of this paper is to advise a modification in the vertical stiffness, natural frequency, and energy modes of the hydraulic engine mount. For this purpose, a layer of buffer absorber was added to the mount. The feature of this change is to reduce the engine mount's design costs. At first, the characteristics of the mount were determined using the catalogs and experimental tests. Then, the engine mount's model was created and analyzed in Adams/view software. The results of experimental tests and simulations of the mount equipped with a buffer show that the vertical stiffness was decreased by 14%. Furthermore, the first natural frequency was reduced by 3% and got away from the stimulation frequency. Also, if the number of energy modes be closer to 100 and not be less than 85, the engine mount would have a better performance. The value of four energy modes has been improved. Moreover, the values of two reduced energy modes are in the acceptable range of design.

Acknowledgements

The authors of this research work express their gratitude and appreciation to the Institute of Automotive Engineering at the Iran University of Science and Technology for the information support and test facilities provided to them.

References

[1]. G. kim, R. Singh, "Nonlinear analysis in automotive hydraulic engine mounts", *ASME Journal of Dynamic Systems Measurements and Control*, 115:482-487, (1993).

[2]. F. Golnaraghi, R.N. Jazar, "Development and analysis of a simplified nonlinear model of a hydraulic engine mount", *Journal of Vibration and Control*, (2000).

[3]. A. Geisberger, A. Khajepour, F. Golnaraghi, "Nonlinear modeling of hydraulic

engine mount, theory and experiment", *Journal of Sound and Vibration,* (2002).

[4]. H. Adiguna, M. Tiwari, R. Singh, D. Hrovat, "Transient response of a hydraulic engine mount", *Journal of Sound and Vibration,* (2003).

[5]. Jeong, T. & Singh, R., "Analytical methods of decoupling the automotive engine torque roll axis", *Journal of Sound and Vibration,* 234, pp. 85-114, (2000).

[6]. P.A.G. Zavala, M.G. Pinto, R. Pavanello, J. Vaqueiro, "Powertrain mounting development based on the computational stimuli and experimental verification method", *SAE Technical Paper*, 2001-01-1509, (2001).

[7]. J. Sui, C.H. Hoppe, J. Hirshey, "Poertrain mounting design principles to achieve optimum vibration isolation with demonstration tools", *SAE Technical Paper*, 2003-0101476, (2003).

[8]. V. Fakhari, A.R. Ohadi Hamedani,"Dynamic modeling of national engine on mounts together with experimental validation", *The Journal of Engine Research*, (2014).

[9]. E. Piltan, R. Tikani, "Nonlinear modeling and sensitivity analysis of a double-notch hydraulic engine mount", *Amirkabir Journal of Mechanical Engineering,* 48(4), pp. 347-354, (2017).

[10]. R. V. Dukkipati, "Road vehicle dynamics", *SAE International*, pp. 231-254, (2008).

[11]. T. Feyzi, M. Esfahanian, R. Tikani, S. Ziaei-Rad, "Simulation of the dynamic behavior of the magneto-rheological engine mount for automotive applications", *International Journal of Automotive Engineering*, 1(2), pp. 1-8, (2011).

[12]. R. Singh, G. Kim, P.V. Ravindra, "Linear analysis of automotive hydraulic-mechanical mounts emphasis on decoupler characteristics", *Journal of Sound and Vibration*, 158, pp. 219–243, (1992).