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An Engine Mount Design and Vibration Analysis

Mustafa Karagöz[1*] and Biran Tuncay[2]

0000-0002-2595-9002[1], 0000-0002-6149-6164[2],

[1.] *Mechanical Engineering Department, Faculty of Mechanical, Karabuk University, Karabük, 78050, Turkey*

[2] *Mechanical Engineering Department, Institute of Science, Karabuk University, 78050, Karabuk, 78050, Turkey*

Abstract

Vibration is an important parameter affecting performance in vehicles. Engine vibrations and vibrations coming from the road are the most important sources of vibration in vehicles. If the vibration is not damped, it will cause noise, fuel consumption and damage to the vehicles and it is transmitted to the passenger cabin, thus negatively affecting both the comfort of the driving and the health of the people in the vehicles. In the vehicles, engine mounts are used to damp vibration produced by the engine. However, in the recent years, there have been trends in the automotive industry to mitigate vehicles and reduce costs, while reducing emissions values and producing more powerful engines. Vehicles mitigating causes more vibration on the vehicle. Therefore improvement of existing systems is a necessity. In this study, in order to reduce the vibration caused by the engine, an engine mount system was created and vibration analysis of the engine mount system as made.

Keywords: Vibration, Engine mount, Vibration analysis

* Corresponding author
Mustafa KARAGÖZ
Mustafakaragoz@karabuk.edu.tr
Adress: *Mechanical Engineering Department, Faculty of Mechanical, Karabuk University, Karabük, 78050, Turkey*

Tel: +90 (370) 418 72 02
Fax: +90 (370) 418 93 45

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1. Introduction

Mechanical vibration is a type of repeated motion [1]. Vibration is ubiquitous condition. Although vibration is needed in vibrating machines such as sieves and road breaker, it is often undesirable. Because vibration causes damage in structures and humans, the effect increases if it continues continuously. People who are exposed to constant vibration and shock may be seen health problems [2]. Vibration causes various deformations in vibration systems and reduces. Their life expectancy [3]. This situation needs to be adjusted very well in devices that works with vibration.

User expectations in the automotive sector are increasing day by day. This expectation leads automotive manufacturers to work with customer focus. Although the user oriented operating situation is primarily taken into account in passenger vehicles, it is still tried to improve in commercial, military and almost all areas. In the light of these studies, it is tried to reduce the weight of vehicles and on the other hand to increase the comfort parameter constantly. However, the desire to reduce the weight of the vehicles has a bad impact on vibration control and noise vibration harshness (NVH) performance. Automotive manufacturers want to produce light vehicles and reduce emissions, reduce fuel and cost savings, as well as bring vibration isolation performance to the best ratings.

Two important sources of vibration in vehicles are the vibrations caused by engine vibrations and road irregularities. Engine vibrations are the most important source of vibration in vehicles. Engine vibrations are vibrations caused by gas pressure and unbalanced forces [4]. The force created by the combustion pressure is called as pressure, the gas pressure is applied to the piston at maximum work time because rises to be highest value at work time. Engine mount systems used in vehicles show good performance in vibration isolation. However, continuous change and improvement in the automotive sector make continuous improvement necessary in this regard. Balance forces caused by rotating mechanisms in engines and they have significant effect on vibration at moments [4]. The forces and moments that occur within the block by the movement of masses due to the explosion as a result of the combustion cycle and combustion process cause vibration and move in the vehicle body [5]. It is possible to study engine vibrations in three heads; the combustion forces arising from the combustion end explosion of the fuel air mixture cause the moment parallel to the crankshaft axis [6].

Internal forces and moments caused by elements making vargel movement, such as piston and connecting rod. The piston moves between the top dead center (TDC) and the bottom dead center (BDC) [6]. While doing this tis movement, while changing the direction in dead spots, it provides the continuity of the movement with the force causes by its weight, this is the inertia force. Inertial forces and movements act parallel to the piston axis perpendicular to the crankshaft axis. Another source of vibration is the vibrations caused by friction and pumping of the systems. These vibrations can be neglected because they are too few compared to other engine vibrations. Vibration causes mechanical waves in systems. These mechanical mounts are described as sound [7]. The spread and speed of sound in space is directly linked to ambient conditions. Noise is a term used to express discomfort caused by this sound, thought to be unwanted sound [7]. Not all sound occurs qualifies as noise, this is directly related people's discomfort with the sound [8]. The intensity and magnitude of the sound are expressed by decibels. Noise above certain decibels can cause health problems in humans. Noise in vehicles can be studied in two different headings. Noise due to air and noise due to structure. Examples of airborne noise are vehicles are exposed to various resistors when travelling, some of which are: air resistance, acceleration resistance and road resistance. Air resistance causes energy fluctuations on the surface of vehicles, wheels and other equipment, leading to noise [9]. The noise caused by the structure is the noise caused by vibrations that occur during the operation of the powertrain [9]. Engine vibrations can be propagated in two ways after they occur, the first as a transfer to the elements in contact with the engine and powertrain, the second as a sound that exists at certain levels when vibrations are prevented from spreading. This is why damping is needed in systems. In order to prevent the vibrations caused by the engine from passing into the vehicle body, the engine mount is connected between the vehicle body and the engine [10]. Four types engine mounts are used in vehicle; rubber, hydraulic, semi-active and active. Hydraulic mounts perform better in shocks than rubber mounts. [11]. Elektro rhelogical (ER) and manyeto reolojik (MR) fluids are used in semi-active and active mounts [12]. Semi-active mounts are more reliable than active mounts [13]. Rubber mounts are the most widely used type due to lack of maintenance and proper cost [14]. Damping is the stopping of movement due to the reduction of energy of vibration movement. The purpose of vibration damping is always to end the vibrations and prevent the transfer of sound and vibration. However, it may not always be possible to end the vibrations, in this case it is tried to reduce as much as possible. The principle of vibration damping is based on placing a damper between the source causing the vibration and the system where the vibration will be transferred. Thus, after the engine vibrations begin to form, it is prevented or reduced as much as possible to travel to the body and the powertrain and into the car cab. In order to reduce the amount of noise in vehicles, insulation materials that dampen the sound are used in the engine section [15].

The aim of this study is to introduce the MSC. Adams engine mount system model in the program, vibration analysis, location analysis and optimization by performing different damping ratio, ports and stiffness values the performance of the product to be manufactured testing and analysis, natural frequency, kinetic energy to achieve the desired levels of values, damping values for transmissivity to be reduced as much as possible and ideal for improving the performance of the product features is the presence. Engine mount system characteristics are shaped according to the performance of the entire engine mount system. Therefore, when designing the engine system, the mount system is taken together. The design of the engine mount system also includes hardness values, damping rates and positioning of the mounts.

2. Materials and Methods

The engine and mount system were modeled using four mounts in the MSC. Adams software. The engine was considered the solid model with six degrees of freedom. The center of gravity of the engine is considered the midpoint of the geometric structure. As shown in Fig. 1, chassis model was created so that the engine can be connected from the bottom. The center of chassis is considered the middle point of the geometric structure. Table 1. shows the features of the engine and chassis.

Table 1. Test engine properties

Engine weight	300 kg
Moment of inertia (Ix, Iy ve Iz)	2.0315 kgm², 2.0315 kgm², 8.1260 kgm²

As shown in Fig. 2, the motor has the freedom to shift in the direction of all three axes and to make rotational motion around these axes.

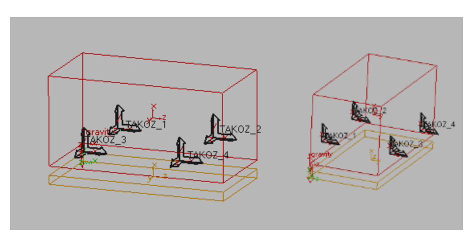

Fig. 1. Engine mount system model

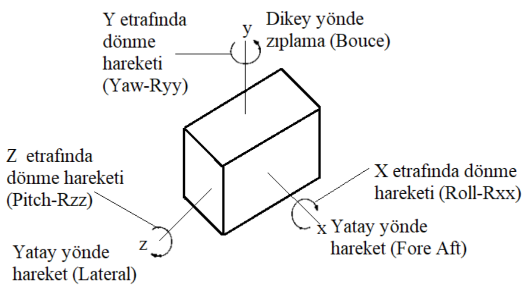

Fig. 2. Engine freedoms

The movement of six different modes in the vertical direction is more prominent in terms of vibration analysis in bounce (vertical) engines. The engine mounts are modeled with the spring coefficient with three-way rigidity and the three-way force with the damping coefficient. In light and gasoline-powered vehicles, three mounts are generally used to connect the engine to the body, and in heavy and diesel-powered vehicles, four mounts are generally used to connect the engine to the body [16]. There are various connection methods of the engine mounts depending on the requirements of the system how to make a connection is decided. In this study, a connection pattern was made on an axis near the center of gravity, unlike normal.

In this study, natural frequency analysis, modal analysis, position analysis and transmissivity analysis were made. Modal analysis is performed to see the values of the system in 6 modes, such as natural frequency and kinetic energy. Natural frequency analysis covers the calculation of natural frequencies for the six modes of the system according to mount stiffness values and attempts to be kept at certain intervals. In the sixth mode, the frequency range of which the human body is most sensitive on the vertical axis is between 4-8 Hz, for the first and second mode this value is between 1-2 Hz. Therefore, different frequencies have been tried to be obtained from these modes.

Location analysis covers the kinetic energy distribution that occurs on the axes in the six modes of the system, depending on the elastic locations of the motor mounts. Positioning the engine mounts to the correct locations is an important parameter in terms of the life of the engine mounts, the distribution of kinetic energy reached on the axes, and accordingly keeping the forces and moments formed on the axes at the desired levels. To be able to provide a good position optimization, the kinetic energy distribution for the movements occurring in six modes is higher than 85 percent and the kinetic energy generated by the movements occurring within the modes is to be discretized. In the another vibration analysis, by creating input channels and output channels, different damping and stiffness coefficients between the values of specific frequency ratios of the system with damping, by analysing forced vibration of the system designed, the performance, the transmissivity values of natural frequency and resonance analysis was made.

The parameters of the designed engine mount system are seen in Table 2, 3 and 4. These values in tables were used in the modal analysis. The results of the analysis was shown in the next to section.

Table 2. First design variables

	Design variables	Location (x, y, z) Stiffness (kx, ky ,kz)
Mount location [mm]	Engine mount1	(-50 ,0, 0)
	Engine mount2	(1050, 0, 0)
	Engine mount3	(-50, 0, 600)
	Engine mount4	(1100, 0, 600)
Mount stiffness [N/mm]	Engine mount1	(80, 120, 80)
	Engine mount2	(80, 120, 80)
	Engine mount3	(80, 120, 80)
	Engine mount4	(80, 120, 80)

Table 3. Second design variables

	Design variables	Location (x, y, z) Stiffness (kx, ky ,kz)
Mount location [mm]	Engine mount1	(0 ,50, 0)
	Engine mount2	(1000, 0, 0)
	Engine mount3	(50, 0, 550)
	Engine mount4	(1050, 0, 550)
Mount stiffness [N/mm]	Engine mount1	(220, 260, 220)
	Engine mount2	(220, 260, 220)
	Engine mount3	(220, 260, 220)
	Engine mount4	(220, 260, 220)

Table 4. Third design variables

	Design variables	Location (x, y, z) Stiffness (kx, ky ,kz)
Mount location [mm]	Engine mount1	(200, 100, 0)
	Engine mount2	(750, 100, 0)
	Engine mount3	(200, 100, 500)
	Engine mount4	(750, 100, 500)
Mount stiffness [N/mm]	Engine mount1	(220, 260, 220)
	Engine mount2	(220, 260, 220)
	Engine mount3	(220, 260, 220)
	Engine mount4	(220, 260, 220)

3. Results and Discussion

In the Table 5 shows the results of the first modal analysis. According to the results of the first modal analysis, the natural frequencies were slightly lower and the sixth mode was around the peak of the range of 4-8 Hz, where the human body is sensitive [17]. According to the results of the analysis, the natural frequency values should be increased by optimization. Natural frequency values are directly related to mount stiffness. By optimization process, the ideal mount stiffness values were obtained for the system.

Table 5. First modal analysis results

	Frekans (Hz)
1.Mod	2.08552
2.Mod	2.45493
3.Mod	3.16255
4.Mod	5.29956
5.Mod	5.53253
6.Mod	6.36977

In the Table 6 shows the results of the first modal analysis. According to the results of the first position analysis, kinetic energy values were very low than the target, the motion values in some modes were very close, and there is no complete decomposition of the values formed between the movements. According to the results of the analysis, it is necessary to increase the kinetic energy values to the targeted values by optimization and to separate the values between the movements. Kinetic energy values are directly related to the position of the engine mounts. By optimization process, the ideal mounts positions were found for the targeted ranges.

Table 6. First modal analysis results

	1.Mod	2.Mod	3.Mod	4.Mod	5.Mod	6.Mod
Fore Aft	0.03	27.70	5.58	69.19	0.04	0.05
Bounce	1.03	0.05	0.01	0.22	17.45	81.38
Lateral	35.89	0.05	0.02	0.01	56.51	7.99
Roll	62.92	0.12	0.04	0.00	25.92	10.40
Yaw	0.01	29.59	72.49	0.88	0.06	0.00
Pitch	0.12	42.49	21.86	32.70	0.03	0.18
Total	100	100	100	100	100	100

According to the second analysis in Table 7, the lowest frequency value of the system was found 3 Hz. The values critical to the human body in the sixth mode have been diverged. Natural frequency values have reached acceptable levels. An analysis was done to achieve better levels after optimization.

Table 7. Second modal analysis results

	Frekans (Hz)
1.Mod	3.07208
2.Mod	4.07069
3.Mod	4.68441
4.Mod	8.78104
5.Mod	9.11816
6.Mod	9.37689

As a result of the second analysis, when looking in Table 8, kinetic energy levels in some modes have increased to the desired values, but it is not possible to say this for all values. On the other hand, some modes are not fully discretized. Location optimization will be done to obtain the elastic Center location values necessary to reach the targeted values.

Table 8. Second modal analysis results

	1.Mod	2.Mod	3.Mod	4.Mod	5.Mod	6.Mod
Fore Aft	0.16	1.65	21.86	76.34	0.01	0.14
Bounce	0.22	0.01	0.19	0.40	3.72	95.43
Lateral	30.32	0.44	0.49	0.03	67.01	1.76
Roll	68.13	0.04	0.58	0.00	29.01	2.20
Yaw	0.21	93.47	5.78	0.25	0.24	0.00
Pitch	0.97	4.39	71.15	22.97	0.01	0.46
Total	100	100	100	100	100	100

In the Table 9 shows the results of modal analysis after optimization. After optimization, the kinetic energy values increased at the target values for each mode according to the position analysis results. Results of post-optimization analysis can be seen to be at acceptable levels.

Table 9. Third modal analysis results

	1.Mod	2.Mod	3.Mod	4.Mod	5.Mod	6.Mod
Fore Aft	4.50	0.00	0.00	95.48	0.00	0.02
Bounce	0.11	0.00	0.00	0.04	0.00	99.85
Lateral	0.00	0.20	14.69	0.00	85.13	0.00
Roll	0.00	0.03	85.19	0.00	14.75	0.00
Yaw	0.00	99.77	0.12	0.00	0.12	0.00
Pitch	95.39	0.00	0.00	4.47	0.00	0.00
Total	100	100	100	100	100	100

In the Table 10 shows the results of modal analysis after optimization. According to the results of post optimization modal analysis, natural frequencies increased to targeted values and the sixth mode moved away from the range of 4-8 Hz, where the human body is sensitive. Results of post-optimization analysis are at acceptable levels.

Table 10. Third modal analysis results

	Frekans (Hz)
1.Mod	3.40846
2.Mod	4.59224
3.Mod	5.20060
4.Mod	9.90501
5.Mod	10.2796
6.Mod	10.4032

In this analysis, the first Damping ratios corresponding to the Damping coefficients are respectively ζ=0.01/0.03/0.05/0.07 in Table 11.

Table 11. First design variables

Mount stiffness [N/mm]	(80, 120, 80)
Damping coefficient [kg/s]	0.1/0.2/0.3/0.45

In the Fig. 3 and 4 show changes in first transmissivity graphs before and after optimization. As shown in the blue color Chart before optimization, damping of the engine mounts starts at 12 Hz. At the first pig, the transmissivity ratio 17 and the natural frequency 5 Hz was found. At the second pig the natural frequency 9 Hz and transmissivity ratio of 3 was achievement. At the third pig, the frequency value was 10 Hz and the transmissivity rate was 5.5. In the Fig. 4 shows changes in transmissivity graphs depending on the Damping coefficients taken after optimization. In the red Chart in Fig. 4, when the Damping coefficient was taken 0.2, the transmissivity rate at the first pig is 9, the value at the second pig point was 2 and the value at the third pig was reduced to 3. When the Damping coefficient 0.3 was taken, respectively, the transmissivity rates were reduced to 6 at the first pig, 1.8 at the second pig and 2.3 at the third pig point.

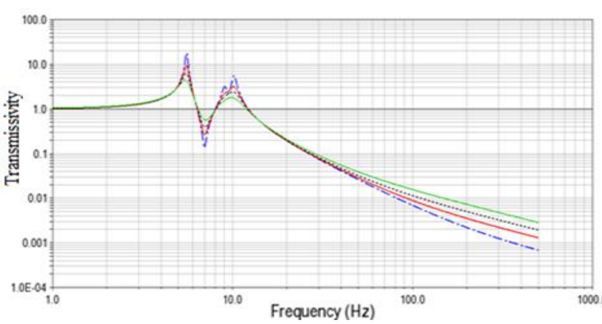

Fig. 3. First vibration transmissivity graphic

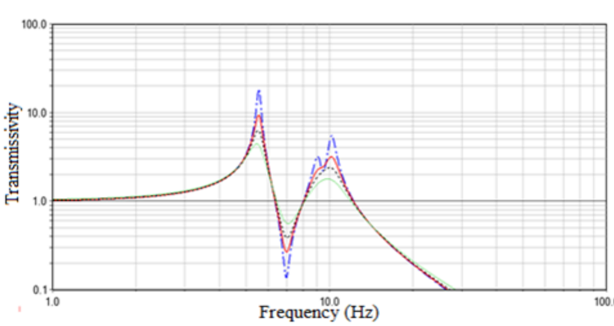

Fig. 4. First vibration transmissivity graphic

When the Damping coefficient was 0.45, the transmissivity ratio was 4 at the first pig, 1.5 at the second pig and 1.7 at the third pig. A decrease in transmissivity and better damping was observed due to changing the Damping coefficient. Due to the change in the damping rate, there was no change in the frequency at which the mount began damping. The sideways movement of the graphics, that is, the frequency values that it starts damping, varies according to the mount stiffness values.

According to the graphs obtained by vibration analysis, The Force and moment values transmitted to the vehicle body are pig at one or two frequency values. The values where these pig points are seen are the natural frequencies of the system in the corresponding direction transmissivity values rise at natural frequency points. The frequency ranges in which pig values are seen in the graphs are resonance regions.

In this analysis, the second Damping ratios corresponding to the Damping coefficients are respectively ζ=0.01/0.03/0.04/0.05 in the Table 12.

Table 12. Second design variables

Mount stiffness [N/mm]	(220, 260, 220)
Damping coefficient [kg/s]	0.2/0.35/0.46

In the Fig. 5 and 6 show changes in second transmissivity graphs before and after optimization. The damping of the engine mounts starts at 18 Hz as shown in the blue color Chart before optimization. At the first pig, transmissivity ratio 36.4 and the natural frequency 8 Hz was found. Respectively, in the second pig frequency 14 Hz transmissivity ratio was 9.49. At the third pig, the frequency value was 16 Hz and the transmissivity rate was 2.47. Fig. 5 and 6 show changes in transmissivity graphs depending on the Damping coefficients taken after optimization. Fig. 5 Damping coefficient when the Damping coefficient was 0.2, the transmissivity rate at the first pig was 11.8, the value at the second pig was 5 and the value at the third pig was reduced to 2. When the Damping coefficient 0.35 was taken in the green color Chart in Fig. 5, respectively, the transmissivity rates were reduced to 4 at the first pig and 2.4 at the second pig. When the Damping coefficient 0.46 was taken in Figure 5, the permeability rates were reduced to 1.5 at the first pig and 1.7 at the second pig, respectively. Due to the change in the damping rate, there was no change in the frequency at which the wedges began damping. The sideways movement of the graphics, that is, the frequency values that it starts damping, varies according to the mount hardness values.

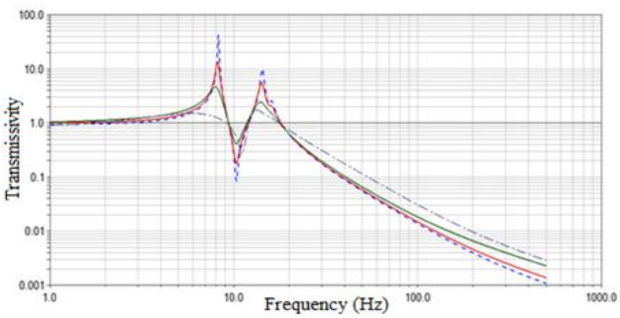

Fig. 5. Second vibration transmissivity graphic

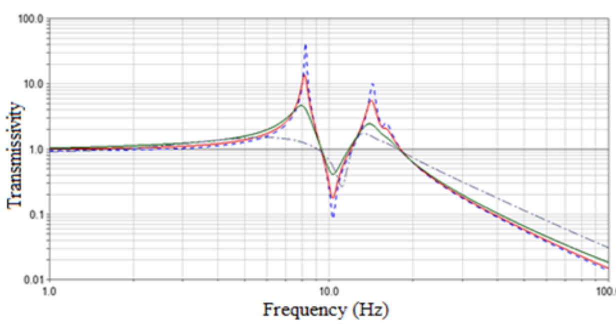

Fig. 6. Second vibration transmissivity graphic

In this analysis, the third damping ratios corresponding to the Damping coefficients are respectively ζ=0.01/0.02/0.03 in the Table 13.

Table 13. Third design variables

Mount stiffness [N/mm]	(280, 320, 280)
Damping coefficient [kg/s]	0.1/0.2/0.7

In the Fig. 7 and 8 show changes in third transmissivity graphs before and after optimization. As shown in the blue color Chart before optimization, damping of the engine mounts starts at 20 Hz. At the first pig, transmissivity ratio 27 and the natural frequency 9 Hz was found. The natural frequency at the second pig was 17 Hz transmissivity ratio of 12. At the third pig, the frequency value was 17 Hz and the transmissivity rate was 2.3. In the Fig. 6 shows changes in transmissivity graphs depending on the Damping coefficients taken after optimization. In the red Chart Figure 6, when the Damping coefficient was taken 0.2, the transmissivity rate at the first pig was 13, the value at the second pig was 6 and the value at the third pig was reduced to 1.9. When the Damping coefficient was taken 0.7, respectively, the permeability rates were reduced to 4 at the first pig and 2 at the second pig. A decrease in transmissivity and better damping was observed due to changing the Damping coefficient. Due to the change in the damping rate, there was no change in the frequency at which the mount began damping. The sideways movement of the graphics, that is, the frequency values that it starts damping, varies according to the mount stiffness values.

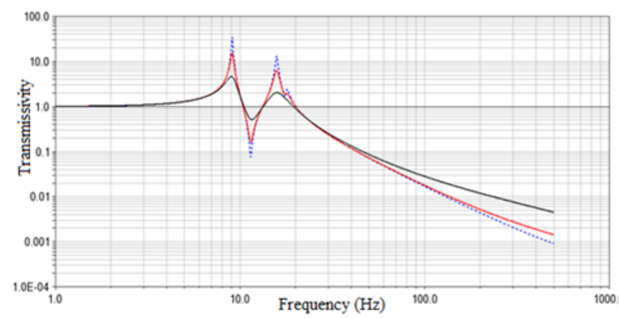

Fig. 7. Third vibration transmissivity graphic

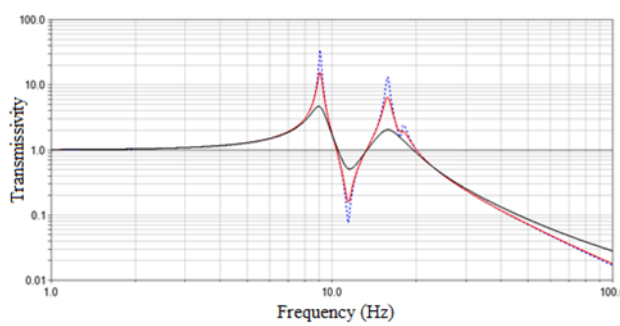

Fig. 8. Third vibration transmissivity graphic

4. Conclusions

In this study, modeling, analysis and optimization studies were performed in order to determine the optimum engine mount parameters. In this study, 3 different modal, natural frequency and location analysis tests were performed between horizontal spring constants 80/220/280 N/mm and vertical spring constants 120/260/320 N/mm.1 Hz and 500 Hz frequency ranges, transmissivity and damping analysis were made by creating input and output channels. As a result of the analysis, the smallest natural frequency in the first mode increased from 2 Hz to 3.4 Hz. In particular, the natural frequency value in the vertical direction increased from 6.3 Hz to 10.4 Hz, resulting in the targeted natural frequency values. After position optimization, the target values of 85 percent and above were reached for each mode and acceptable separations were made between the six modes. According to damping and stiffness coefficients, in the analysis and optimizations, rate of transmissivity and damping were improvement and in resonance regions decrease was made. In the light of these parameters, the system properties that will be used can be decided by making the necessary evaluations.

References

[1] Rao, S.S. (2011). Mechanical vibrations 5nd ed, *Prentice Hall*, 1-15.

[2] Kelly, G. (2011). Mechanical vibrations theory and applications SI ed, *Randall Adams*, 1-6.

[3] Çay, C.İ. (2006). A study for vibration damping equipment of operator's seat of agricultural tractors, *Department of Agricultural Machinery*, 30-35.

[4] Lichty, L. (1967). Combustion engine process, *McGraw Hill*, 20-30.

[5] Schmitt, R.V. and Charles, J.L. (1976). Design of elastomeric vibration isolation mounting system of internal combustion engines, *SAE Technical Paper Series*, 760431, 10-14.

[6] Doughty, S. (1988). Fundamentals of IC engine torsional vibration, *Energy Sources Technology Conference and Exhibition*, 10-14.

[7] Becker, S. and Yu, S. (1999). New venture gear, *Objective Noise Rating of gear whine*, SAE 1720, 20-25.

[8] Arı, Ş. (2011). Investigation of the effects of combustion chamber design on diesel engine noise. *Mechanical Engineering*, 3-4.

[9] Güney, A. (1989). Taşıt titreşimleri ve irdelenmesi, *Mechanical Engineering*, 5-10.

[10] Engel, E. (2013). Predicting the design relevant loads in the engine mount system at an early stage of the development process, *Industrial Engineering and Development*, 6-11.

[11] Mansour, H. (2010). Design and development of active and semi-active engine mounts, *Simon Fraser University Mechatronic System Engineering*, 9-13.

[12] Pell, D., Stanway, R. and Bullough, W.A. (1996). Dynamic modelling of an ER vibration damper for vehicle suspension applications, *Journal of Smart Meterials and Structures*, 5, 597-606.

[13] Muller, M., Eckel, H., Leibach, M. and Bors, W. (1996). Reduction of noise and vibrations by approximate engine mount systems and active absorbers. *SAE Paper Series*, 960184, 50-54.

[14] Alkhatib, F. (2013). Techniques for engine mount modelling and optimizasyon, *Mechanical Engineering*, 2-12.

[15] Ağbulut, Ü., Karagöz, M., Sarıdemir, S. and Öztürk, A. (2020). Impact of various metal-oxide based nanoparticles and biodiesel blends on the combustion, performance, emission, vibration and noise characteristics of a CI engine. *Fuel*. (No. 2020-06- 117521), 1-5.

[16] Heisler, H. (2002). Advanced vehicle technology 2ed, *Butterworth Heinemann*, 15-17.

[17] Özkan, M.O. (2016). Çalışanların bütün vücut titreşimine maruziyet risklerinden korunmalarına ilişkin uygulama rehberi. *Ministry of Family, Labor and Social Services*, 24-25.