Effect of Support Angle of Automobile Engine Rubber Suspension on Dynamic Response

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Abstract: In this paper, combined with the research on automobile engine related fields, a three-dimensional solid model of the engine assembly was established according to the main structural characteristics of the actual engine by application of SDRC/Ideas9 software. The dynamic response law of the engine assembly was studied by changing the stiffness parameters, angle parameters and position parameters of the three suspension points of the engine, and it was found that the changes in the suspension parameters have caused the dynamic response of each support point to change. Of which, the change of the stiffness parameter has the greatest effect. After overall considering the modal responses, a suitable support position, support stiffness and angle are selected to achieve better vibration isolation.

1. Introduction

The directions of the three elastic spindles p, q and r on the rubber suspensions of the automobile engine are closely related to the angles of the rubber suspensions. Therefore, the changes in the rubber suspension angles will inevitably affect the dynamic response of each rubber suspension point. In this paper, the third set of data in Table 2 are still used as the stiffness with the use of five sets of angle parameters, to study the dynamic response of the engine assembly on three suspension points, to find out the changes in the resonance frequency and resonance peak of each set of values on the three suspension points, and thus to realize reasonable matching of the engine suspension parameters.

2. Support Angle of Rubber Pad

The three elastic spindles of the rubber pad area, q and respectively, and the included angles of them with the coordinate axis of the engine assembly system are shown in table 1.

Table1 Angle between the elastic spindles and the coordinate axis of engine assembly system

		θ_p (°)	θ_q (°)	θ_r (°)
Set 1	Support 1	0	35	35
	Support 2	0	-35	-35
	Support 3	0	0	0
Set 2	Support 1	0	40	40
	Support 2	0	-40	-40
	Support 3	0	0	0
Set 3	Support 1	0	45	45
	Support 2	0	-45	-45
	Support 3	0	0	0
Set 4	Support 1	0	50	50
	Support 2	0	-50	-50
	Support 3	0	0	0
Set 5	Support 1	0	55	55
	Support 2	0	-55	-55
	Support 3	0	0	0

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3. Dynamic Response of the Engine Assembly

Under the condition that the stiffness and support position of the rubber pad suspension of the engine assembly remain unchanged, the suspension support angle is changed to obtain the corresponding 6th order modal as shown in table 2. As can be seen from table 2, with the change of the suspension angle, the modal of the engine assembly moving along the axis Y, the axis Z and the axis X all shows an upward trend. The modals of rotation around the axis Z, the axis Y and the axis X are basically unchanged.

Table2 Order modal of the engine assembly at different suspension angle

	Modal（Hz）					
	Y	Z	X	R_Z	R_Y	R_X
Set 1	2.63	6.35	6.84	10.5	12.1	13.5
Set 2	2.64	6.37	6.86	10.5	12.1	13.4
Set 3	2.66	6.38	6.88	10.5	12.1	13.4
Set 4	2.68	6.40	6.90	10.5	12.2	13.3
Set 5	2.70	6.41	6.92	10.5	12.2	13.4

Apply a force of 0~30Hz 100N perpendicular onto the upper surface of the cylinder block of the engine assembly to obtain the dynamic response to each support point. Since the response to the 2nd support point is the same as that on the 1st support point, the displacement response to the 1st and 3rd support points of each set of data is only shown below, as shown in figures 1-10.

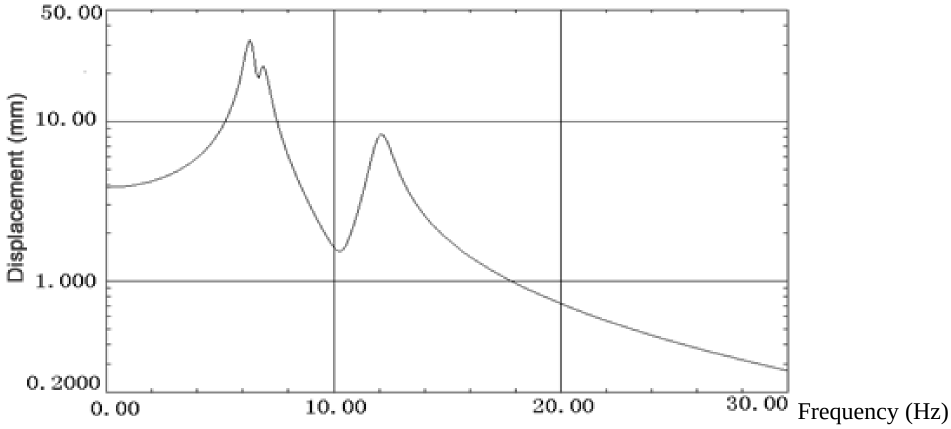

Fig.1 Displacement response to point 1 in set 1

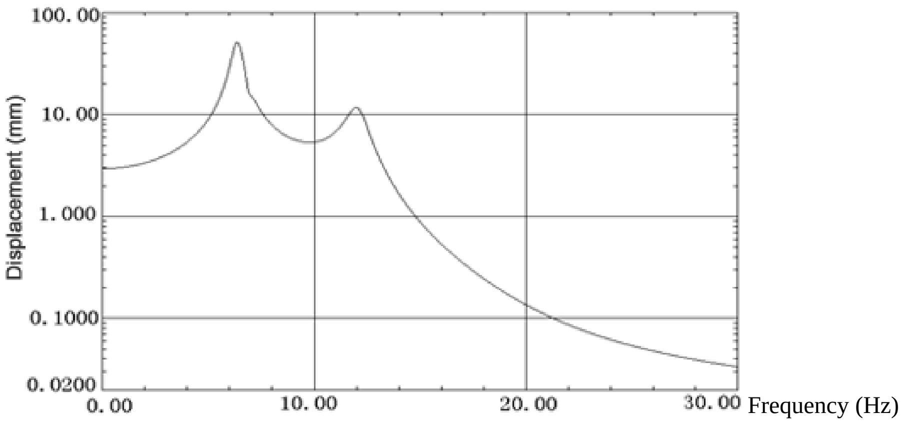

Fig.2 Displacement response to point 3 in set 1

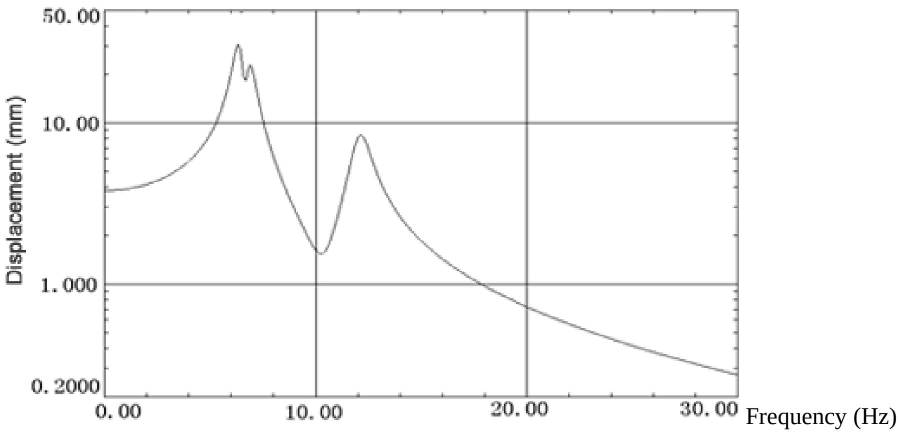

Fig.3 Displacement response to point 1 in set 2

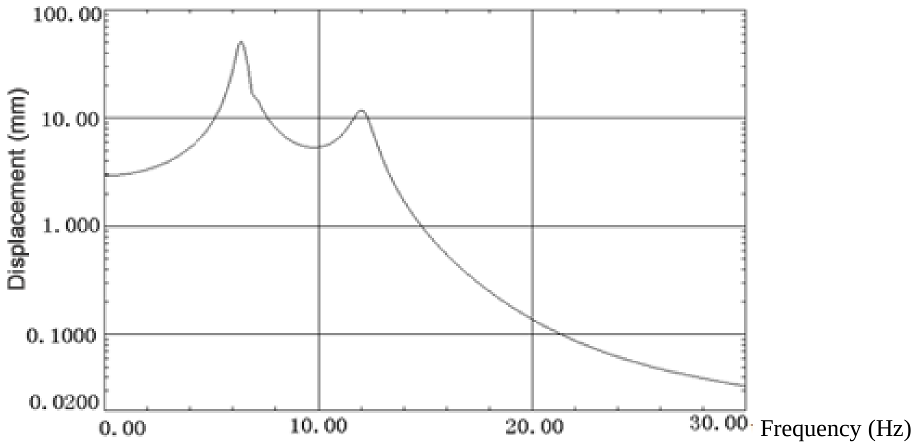

Fig.4 Displacement response to point 3 in set 2

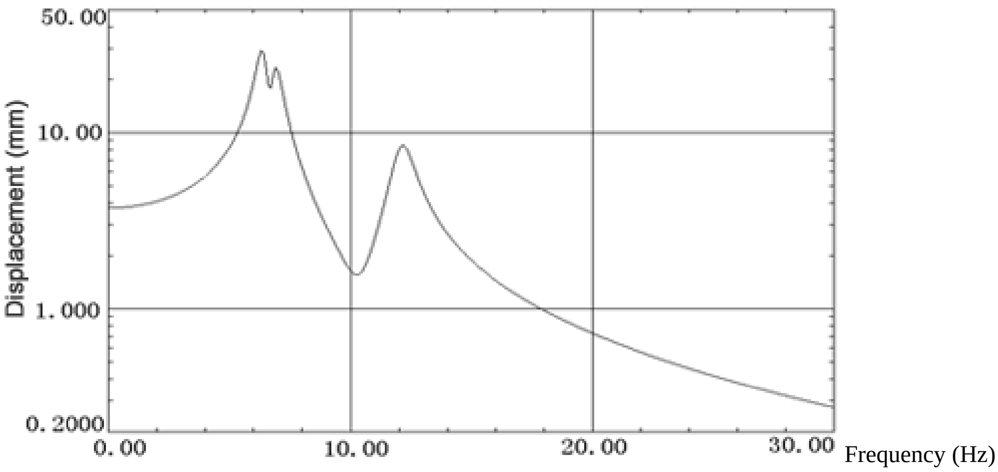

Fig.5 Displacement response to point 3 in set 3

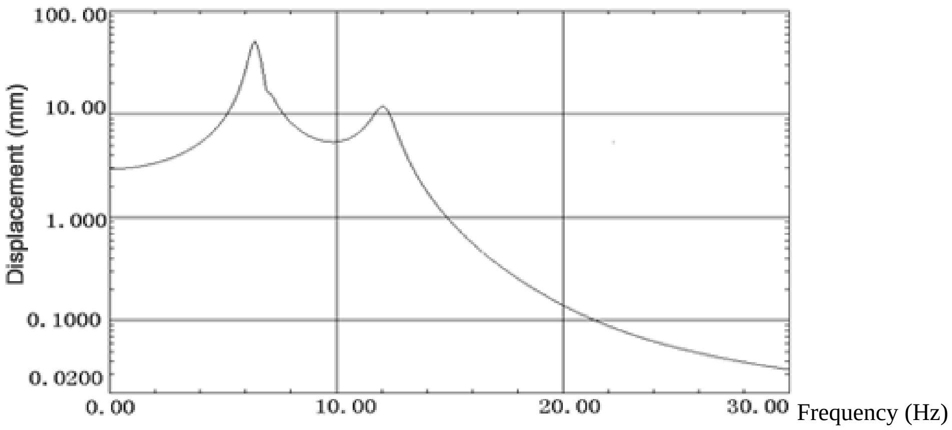

Fig.6 Displacement response to point 3 in set 3

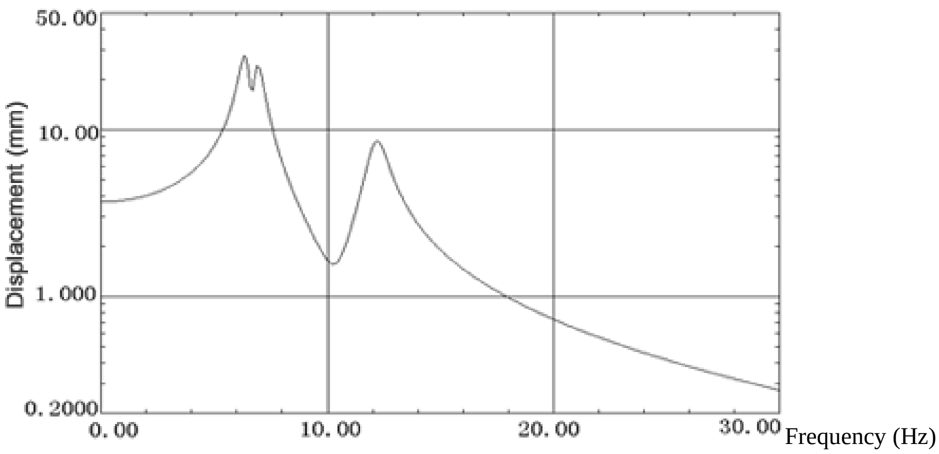

Fig.7 Displacement response to point 1 in set 4

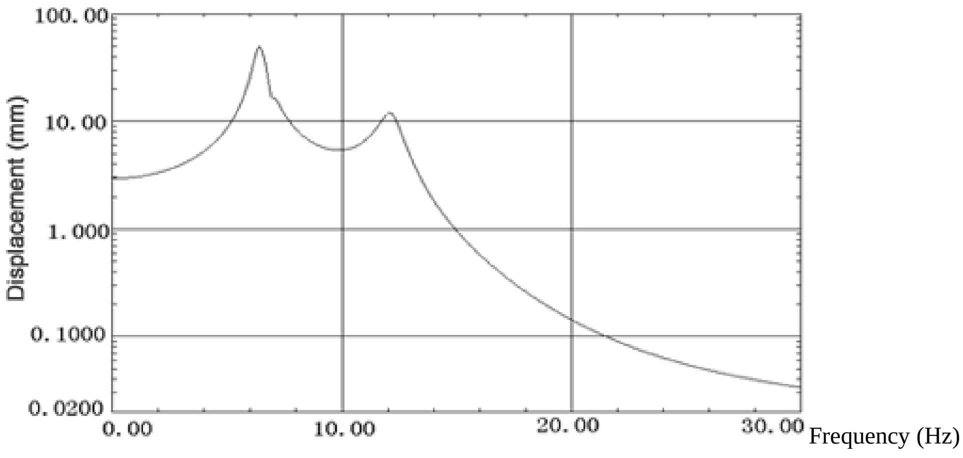

Fig.8 Displacement response to point 3 in set 4

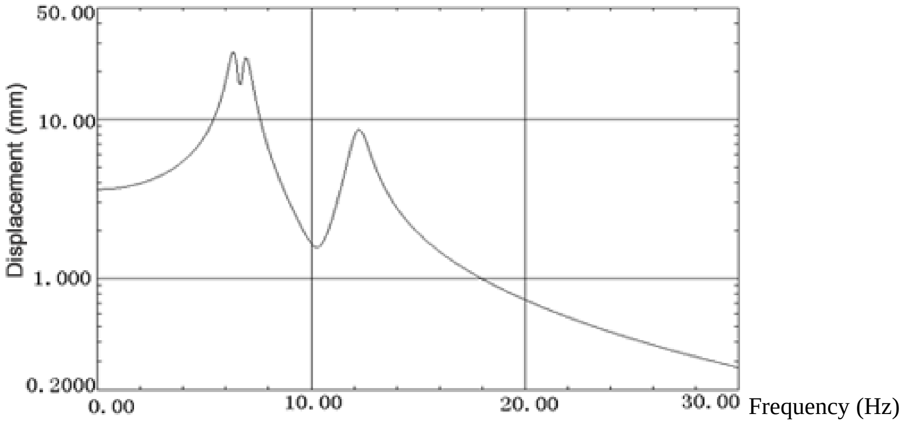

Fig.9 Displacement response to point 1 in set 5

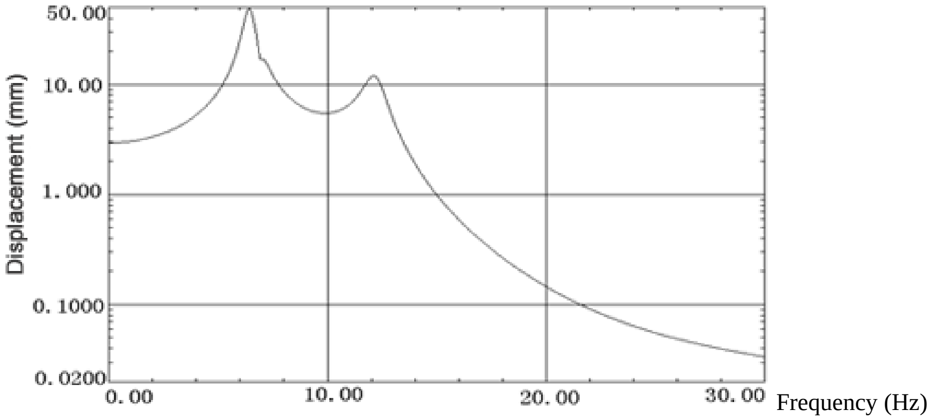

Fig.10 Displacement response to point 3 in set 5

4. Analysis on Dynamic Response Results after Changing the Rubber Suspension Support Angle

Fig. 1, Fig. 3, Fig. 5, Fig. 7, and Fig. 9 show the peak displacement of the first support point and the corresponding frequency value, and the results are shown in table 3.

It can be seen from table 3 that with the change of the engine assembly mount Angle, the resonance peak of the 1st support point and the corresponding resonance frequency value change. The first resonant frequency value is very close to the modal frequency value moving along the axis (see table 2), and there is synchronous fluctuation. The second resonant frequency value is close to the modal frequency value moving along the shaft and has synchronous fluctuation. The third resonant frequency value is close to the modal frequency value of the rotation around the shaft.

Table3 Dynamic response peak at the first support point of the engine assembly

	Dynamic response					
	Frequency 1（Hz）	Peak 1 (mm)	Frequency 2 (Hz)	Peak 2 (mm)	Frequency 3 (Hz)	Peak 3(mm)
Set 1	6.30	32.5	6.90	22.3	12.2	8.17
Set 2	6.30	30.8	6.90	22.9	12.2	8.33
Set 3	6.30	29.2	6.90	23.5	12.2	8.50
Set 4	6.30	27.6	6.90	23.9	12.2	8.56
Set 5	6.30	26.1	6.90	24.2	12.2	8.61

FIG. 2, FIG. 4, FIG. 6, FIG. 8 and FIG. 10 are the displacement peak and corresponding frequency value of the third support point and are the resonance peak and resonance frequency values of the third support point. The results are shown in table 4.

It can be seen from table 4 that with the change in the suspension angle of the engine assembly, the resonance peak and the corresponding resonance frequency value of the third support point have changed. The first resonant frequency value is very close to the modal frequency value of the engine assembly moving along the axis Z (see table 2), and there is synchronous fluctuation; at this point, the vibration is mainly manifested by the movement of the engine assembly along the axis Z.The second resonant frequency value is close to the modal frequency value of the engine assembly moving along the axis X, and there is synchronous fluctuation; the vibration is mainly manifested by the movement of the engine assembly along the axis X, of which the values of set 1, set 2, set 4, and set 5 are empty; mainly because it is too close to the frequency of the first resonance peak, it is covered up. The third resonant frequency value is close to the modal frequency value of the engine assembly rotating around the axis Y; at this point, the vibration is mainly manifested by the rotation of the engine assembly around the axis Y. With the change of the suspension angle, the modals of the engine assembly moving along the axis Y, along the axis z and along the axis X all show an upward trend. The modals of the engine assembly rotating around the axis Z, the axis Y and the axis X are basically unchanged.

Table4 Dynamic response peak at the third support point of the engine assembly

	Dynamic response					
	Frequency 1（Hz）	Peak 1(mm)	Frequency 2(Hz)	Peak 2(mm)	Frequency 3(Hz)	Peak 3(mm)
Set 1	6.30	50.9	——	——	12.0	11.8
Set 2	6.40	51.0	——	——	12.0	11.9
Set 3	6.40	50.7	7.10	15.7	12.1	11.8
Set 4	6.40	50.1	——	——	12.1	12.0
Set 5	6.40	49.4	——	——	12.2	11.9

5. Summary

The impact of changing the support angle of the rubber suspension on the dynamic response of the engine assembly was discussed in the paper, and the following conclusions are drawn:

With the change of the suspension angle, the modals of the engine assembly moving along the axis Y, the axis Z and the axis X all show an upward trend, indicating that the engine's lateral movement, up and down movement, and forward and backward movement all show an upward trend at this time. The modals of the engine assembly rotating around each coordinate axis are basically unchanged, indicating that the horizontal torsion, pitch, and roll motion of the engine are weakened.

With the change of the suspension angle, the resonance peak value and corresponding resonance frequency value both vary, but their basic rules are the same: the first resonance frequency value is very close to the modal frequency value of the engine assembly moving along the axis Z, and there is synchronous fluctuation. At this time, the vibration is mainly manifested by the up-down vibration of the engine assembly; the second resonant frequency value is close to the modal frequency value of the engine assembly moving along the axis X and there is synchronous fluctuation; the vibration is mainly manifested by the forward and backward movement of the engine assembly. The third resonance frequency values is close to the modal frequency value of the engine assembly rotating around the axis Y, and the vibration is mainly manifested by the pitching motion of the engine assembly; the results are the same as the changed stiffness and location, so that the research and optimization design of the engine suspension system has a clear focus object, thus providing an idea for effective and reasonable vibration reduction measures.

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