Analysis on Hanger Location and Hanger Isolator of An Exhaust System with Powertrain

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Abstract-To improving the NVH (Noise, Vibration and Harshness) performance of vehicles, the NVH performance of exhaust system must be considered. As hanger is the main vibration transfer path and hanger isolator is the crucial vibration isolation components, it is necessary to study hanger location and stiffness of vibration isolation. About the study of exhaust system is executed in four steps. Firstly, according the free modal analysis of the exhaust system, the ADDOFD (Average Driving Degree of Freedom Displacement) is acquired, and the suitable location of hanger is confirmed soon. Secondly, studied the powertrain influences to the vibration performance of exhaust system, the constrained modal frequencies must avoid the engine idle resonant frequency. Thirdly, the dynamic analysis of the exhaust system is researched based on the second step, in which way 100000N·mm sinusoidal excitation is applied at the powertrain's barycenter in the crankshaft rotation direction, and then the response reaction force (less than 5N) of the hanger are gained. Finally, the relationship between the bending and torsional modes of exhaust system and the stiffness of the hanger isolator are inquired. It is of great engineering significance.

Keywords-NVH; ADDOFD; resonant frequency; reaction force; exhaust system

I. INTRODUCTION

When a vehicle runs, the exhaust system is excited by engine periodic dynamic vibration, and it affects fatigue life of component and reliability of the hanger. At the same time, periodic vibration through hanger and isolator transfer to the body influent NVH performance [1]. Therefore, it is necessary to make analysis and optimization on the vibration characteristics of the exhaust system. Among them, control passes to the body force [2] is one of the most important in the vibration control of exhaust system. Reasonable hanger location and stiffness of isolator are performed.

The study of the exhaust system is a systemic process. Most previous scholar do not consider the powertrain in the finite element model, nevertheless, as we know, the powertrain is of great importance on the modal analysis and dynamic analysis, especially at some concerned frequency ranges like bending modes and torsional modes. It can not guarantee the accuracy of model.

The author established the full finite element model for an exhaust system including powertrain using the shell element. Besides, the study of the exhaust system is executed in four steps, which are step and step. The result of ADDOFD [3] determines the optimal hanger location. Then the constrained modal analysis of exhaust system is researched to avoid the engine idle resonant frequency. After that, the dynamic analysis of the exhaust system is studied based on the optimal hanger location. In the end, the relationship between the bending and torsional modes of exhaust system and the stiffness of the hanger isolator are investigated, so that the reasonable stiffness can be guided to choose.

II. ADDOFD METHOD

In the past, the ADDOFD method is used in the modal test usually, hope will structure suspension point selection in small displacement amplitude. Therefore we need to pre-determine the optimal hanging position. As the same way, the vibration displacement of hanger should minimize.

The excitation is assumed to be single point drive. According to the modal analysis theory of multi-degree of freedom system, the response function between the response point l and exciting point p can be described as following [4]:

$$H_{lp}(\omega)=\sum_{r=l}^{N}\frac{\varphi_{lr}\varphi_{pr}}{M_r(\omega_r^2-\omega^2+j(2\zeta_r\omega\omega_r)} \tag{1}$$

Where φ_{lr} is the r order mode shape system of the l measuring point, M_r is the modal mass, ω_r is the frequency of the r order modal, ζ_r is damping ratio of the r order modal.

If the frequency of the exciting force is ω_r, $H_{lp}(\omega)$ can similar as following [5]:

$$H_{lp}(\omega)\approx\frac{\varphi_{lr}\varphi_{pr}}{j(2M_r\zeta_r\omega_r^2)} \tag{2}$$

For linear system, the amplitude of the displacement response is proportional to the amplitude of the frequency response function, which can be described as following [6]:

$$X(\omega_r)\propto H_{lp}(\omega_r)\approx\frac{\varphi_{lr}\varphi_{pr}}{j(2M_r\zeta_r\omega_r^2)} \tag{3}$$

Suppose further that the modal shape of the mass matrix is normalized, and the mode damping is approximately equal, as following [7]:

$$X(\omega_r) \propto \frac{\varphi_{lr}\varphi_{pr}}{\omega_r^2} \tag{4}$$

In order to predict the relative size of some degree of freedom's response displacement at excitation state, the ADDOFD value of the j degree of freedom can be depicted with the following equation [8]:

$$ADDOFD(j) = \sum_{r=1}^{m} \frac{\varphi_{jr}^2}{\omega_r^2} \tag{5}$$

Where m is number of modal order.

Lower values of ADDODF represent the lower vibration response displacement, and the node's position with lowest ADDODF values is usually selected as the hanger location.

III. Finite Element Model

The finite element model of the exhaust system includes powertrain model, catalytic converter, corrugated pipe, muffler, hanger, isolator, connecting flange and weld joint. The structure of exhaust system is as shown in Fig .1.

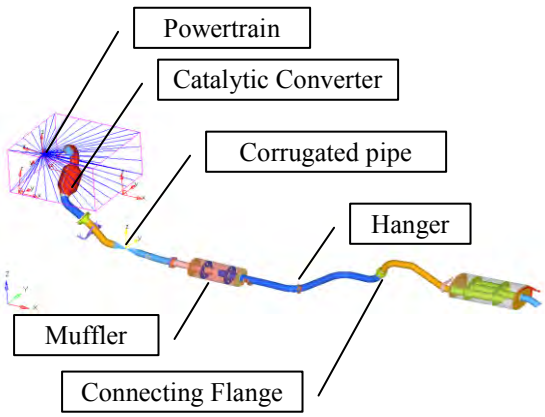

Figure 1. The finite element model of the exhaust system

A. [owertrain Model

Simplified the powertrain as rigid body, the PLOT element simulate powertrain's basic outline, and setting the mass and inertia parameters to the powertrain center in the powertrain local coordinate system. In addition, the engine mount is simplified to CBUSH element with corresponding stiffness parameter to simulate the vibration isolation characteristics. Using the rigid beam element connect the powertrain, the engine mount and the exhaust system. Powertrain model is as shown in Fig .2.

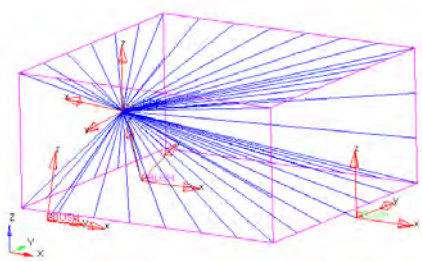

Figure 2. Simplified powertrain model

B. Catalytic Converter and Weld Joint

Using the shell element mesh the catalytic converter and adopting 4mm shell element with characteristic of the stannum signify the weld joint. The catalytic converter and weld joint finite element model are as shown in Fig .3.

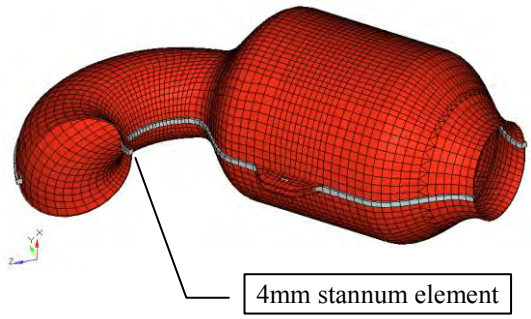

Figure 3. The catalytic converter and weld joint model

C. Corrugated Pipe

Corrugated pipe plays an extremely important role in the vibration transfer path from the engine to the exhaust system. The corrugated pipe model uses CBUSH element to simplify, according to the experimental data of the corrugated pipe to define its each direction stiffness. At the same time the rigid element couple nodes of each side. The corrugated pipe model is as shown in Fig .4.

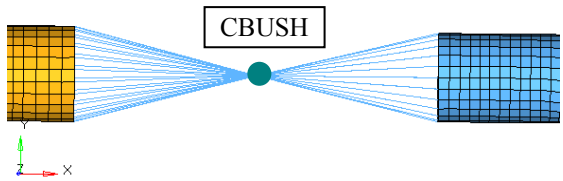

Figure 4. The corrugated pipe model

D. Connecting Flange

Connecting flange simplify as shell element, and using the CBAR element with radius of 5 mm to simulate bolt. The connecting flange model is as shown in Fig .5.

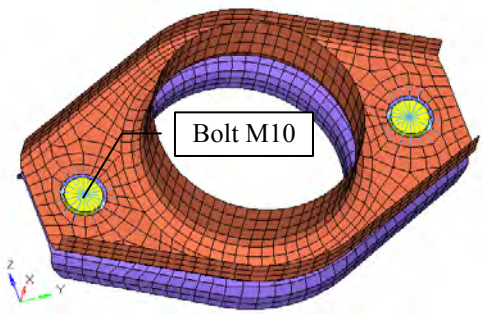

Figure 5. The connecting flange model

IV. Optimization of the Hanger Location

As shown in Fig .6, select 72 nodes in turn in the exhaust system, free modal of exhaust system is calculated within 200Hz. According to the (1), the ADDOFD values of the select nodes are analyzed, as shown in Fig .6

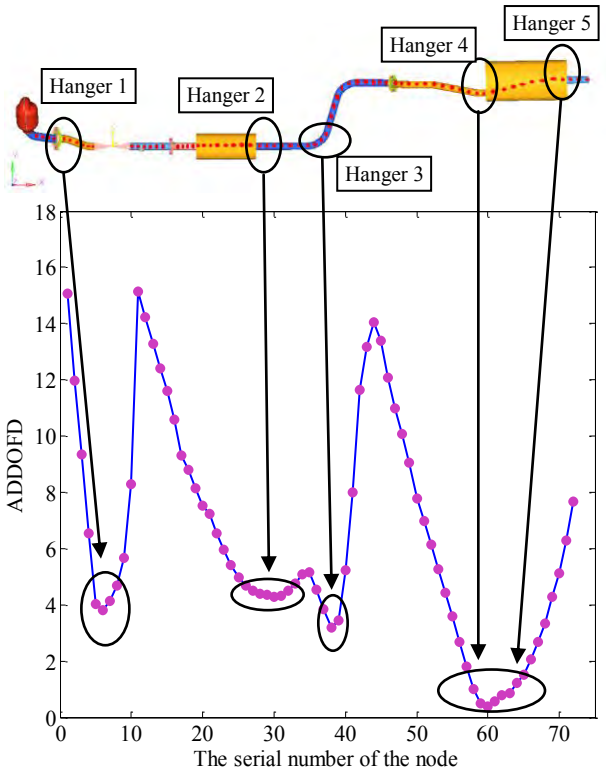

Figure 6. The ADDOFD values of the nodes

Considering the actual structural of the exhaust system, floor and chassis assemble space. The final optimal hanger location is shown in the Fig .6. After the hanger location has been confirmed, the constrained modal analysis of the exhaust system with the powertrain would be studied.

V. CONSTRAINED MODAL ANALYSIS OF THE EXHAUST SYSTEM WITH POWERTRAIN

Constrained modal analysis is the key to the exhaust system dynamic calculation. Exhaust system is connected to the engine and car body, so the exhaust system modal frequencies must avoid engine idle resonant frequency. Therefore, constrained modal analysis can test the reasonability of the selected hanger location. All exhaust system constraint modal frequencies within 200Hz are as shown in TABLE I.

TABLE I. EXHAUST SYSTEM CONSTRAINT MODAL FREQUENCY

Mode No.	*Frequency/Hz*	*Mode No*	*Frequency/Hz*
1	5.64	12	59.73
2	7.64	13	66.47
3	8.53	14	96.99
4	10.64	15	108.30
5	14.88	16	129.50
6	17.72	17	135.88
7	18.07	18	144.80
8	34.38	19	146.43
9	40.81	20	167.63
10	42.19	21	184.31
11	49.25	22	198.45

According to the cold and warm-up engine idle speed, calculate excitation frequency range 22Hz to 28Hz [9], and TABLE I shows that the natural frequency of the exhaust system avoids the engine idle excitation frequency. It is conclusion that hanger location is suitable. There are two modes closest the engine idle excitation frequency, and these include transverse bending mode and torsional mode as shown in Fig .7 and Fig .8.

Figure 7. Transverse bending mode

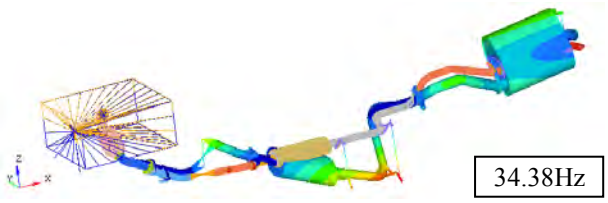

Figure 8. Torsional mode

Fig .9 is contrast of the modal frequency between the exhaust system with and without powertrain. It is obvious that there is some influence in frequency within some concerned frequency ranges. These are differences mainly occur nearby the mode of pipe connected to the engine, and the biggest difference reached 5Hz. In consequence, the powertrain is an indispensable part in the dynamic analysis of the exhaust system.

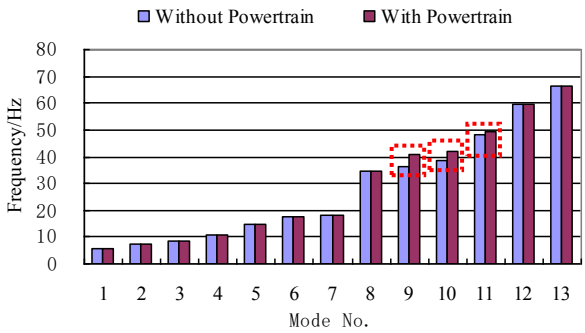

Figure 9. Modal frequency with/without powertrain

VI. DYNAMIC RESPONSE FORCE ANALYSIS OF THE EXHAUST SYSTEM WITH POWERTRAIN

Dynamic analysis of the exhaust system mainly concerns the dynamic load and structural damping and inertia. Since hangers are connected with the vehicle floor, the structure and dynamic stiffness [10] of hangers can guarantee the vibration isolation performance. Importantly, ideal hangers can separate the low vibration frequencies of the exhaust system effectively. It adopts finite element model with powertrain of the exhaust system as shown in Fig .1. The 100000N•mm sinusoidal excitation is applied at the powertrain's barycenter in the crankshaft rotation direction, and then the response reaction force of the hanger are gained, which is transferred to the vehicle floor. The response force of each hanger is as shown in Fig .10. Where Hanger 1-1 is "-Y" direction side of the Hanger 1, and Hanger 1-2 is "Y" direction side of the Hanger 1 as

shown in Fig .1. It is clear that the maximum value of the hangers' response force is about 4.25N, less than 5.00N of requirement limit.

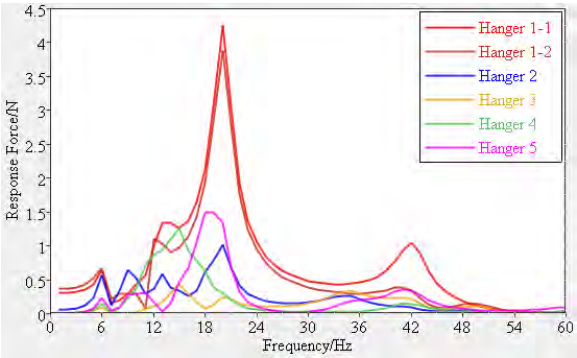

Figure 10. The response force of each hanger

VII. INVESTIGATION OF THE ISOLATOR'S STIFFNESS

Vibration energy of the exhaust system is mainly through the global bending and torsional vibration transfer to the vehicle body. And that the stiffness of the hanger isolator has significant influence on the first bending mode and the first torsional mode of the exhaust system especially. These modes belong to the low frequency range from 30Hz to 50Hz. The shapes of first transverse bending mode and first torsional mode are as shown in Fig .7 and Fig .8 respectively. In order to research the relationship between the first bending mode and the first torsional mode of the exhaust system and the Z direction stiffness of the hanger isolators in the global coordinate system. There are six conditions with different isolators' stiffness: 30N/mm, 50N/mm, 100N/mm, 150N/mm, 200N/mm and 300N/mm. The modal analysis results of these cases are as shown in Fig .11.

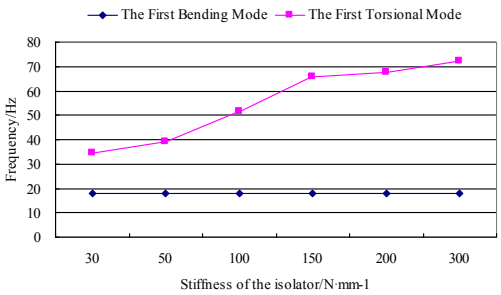

Figure 11. The modal frequency at different isolator stiffness

In Fig .11, it is obvious that the Z direction stiffness of the hanger isolator has weak influence on the first transverse bending mode and has strong effect on the torsional mode of the exhaust system. More importantly, the targets of the two modes avoid the excitation frequency of the engine idle speed. As a consequence, the reasonable isolator's stiffness can be selected according to the contrastive cases.

VIII. CONCLUSION

Hanger location analysis and hanger isolator of the exhaust system are of great engineering significance for the NVH performance. This paper presents a finite element simulation technique for the optimal hanger location of a vehicle exhaust system. The investigation is executed in four steps. According the free modal analysis of the exhaust system, the ADDOFD of each node is acquired, and the suitable location of hanger is confirmed soon. Based on the optimal hanger location, the constrained modal analysis with the powertrain of the exhaust system is calculated, and the results show that the natural frequency of the exhaust system avoids the engine idle excitation frequency, and it is conclusion that hanger location is suitable. Moreover, there is some influence in frequency within some concerned frequency ranges. Then the dynamic analysis and the response force of hangers are acquired, the maximum value is 4.25N, less than 5.00N of requirement limit, located in Hanger 1. In the end, the relationship between the first transverse bending mode, the first torsional mode of exhaust system and the stiffness of the hanger isolator are inquired, and the results shown that the Z direction stiffness of the isolator has weak influence on the first transverse bending mode and has strong effect on the torsional mode of the exhaust system.

ACKNOWLEDGMENT

This research was supported by "the Fundamental Research Funds for the Central Universities (WUT: 2014-VII-004)". This financial support is gratefully acknowledged.

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