Theoretical calculation and experimental analysis of the rigid body modes of powertrain mounting system

YONGJUN JIN, JIANWU ZHANG, XIQIANG GUAN
School of Mechanical Engineering
Shanghai Jiao Tong University
800 Dong Chuan Road, Shanghai, 200240
China
13816691189@139.com   http://www.sjtu.edu.cn

Abstract: -The performance of dynamic property of the mount is influenced by multiple factors and strongly depends on the working conditions. This means that the modal parameters of powertrain mounting system would make changes under different operating conditions. A novel approach to simulate the actual working condition is proposed in the testing of dynamic stiffness. Then the mechanical model of powertrain mounting system based on dynamic stiffness is established in this paper. In order to examine the rationality and accuracy of the computational model based on dynamic stiffness; experimental modal analysis is performed by multiple means and methods in this paper. Through the contrast analysis, advantages and disadvantages of these methods are illustrated and it is shown that using the method of Operational Modal Analysis could obtain more accurate and more reliable results. Based on the experimental and evaluation results, it is shown that there is smaller relative error and higher fitting degree between the calculation results based on dynamic stiffness and the results obtained from operational modal analysis. Moreover, the proposed method also enjoys satisfactory consistency with the actual working condition.

Key-Words: - Powertrain mounting system; Dynamic stiffness; Static stiffness; Operational Modal Analysis; Rigid body modes; Experimental modal analysis; Impact hammer testing

1 Introduction

Other than road-tire excitation, powertrain is one of the major sources of vibration in the vehicle. It transmits vibration energy caused mostly by the unbalanced engine disturbances to the passenger compartment through powertrain mounting system. The vehicle powertrain mounting system, generally, consists of a powertrain and several mounts connected to the vehicle structure [1]. Besides supporting the powertrain weight, the major function of powertrain mounts is to isolate the unbalanced engine disturbance force from the vehicle structure. Accurate understanding of the dynamic properties of powertrain mounting system is a main reference of vehicle vibration controlling and NVH enhancement [2]. Thus, establishing a feasible mechanical model of powertrain mounting system or accurately acquiring its dynamic properties through experiments play an important role in system design and optimization [3]. The main purpose of this paper is to present a comprehensive theoretical analysis and experimental research on the dynamic properties of powertrain mounting system.

In some studies, the stiffness matrix in the model of powertrain mounting system is constructed based on static stiffness of mounts [4]. However, in most actual conditions, the mount is working under certain frequency and amplitude of excitation, which is closely related to the dynamic stiffness [5]. Therefore, computational results from the model based on static stiffness are inconsistent with the actual situation. Taking the idle speed condition of engine as an example in this paper, a novel approach to simulate the actual working condition is proposed in the process of dynamic stiffness testing. Then the mechanical model of powertrain mounting system based on dynamic stiffness is established. By comparison the computational results respectively from the model based on static stiffness and dynamic stiffness, the differences between two methods are demonstrated.

In order to examine the rationality and the accuracy of the computational model based on dynamic stiffness, multiple means and methods are used to obtain the natural frequencies of mounting system through experimental modal analysis in this paper. First, the traditional hammer impact testing is carried out. In the process of the experiment, it's found that applying different mass of hammer could cause quite different results. This is due to the nonlinear characteristics of the stiffness of mounts. To avoiding such deficiency of traditional hammer

impact method, operational modal analysis is adopted to identify the modal parameters in the idle speed condition of engine.

Furthermore, the modal parameters, which calculated from mechanical model based on static stiffness and that based on dynamic stiffness, are compared respectively with the identification results obtained under different experimental modal analysis method. Through the contrast analysis, advantages and disadvantages of these methods are illustrated clearly.

2 Theoretical analysis and calculation

In this section, the equations of motion of powertrain mounting system are presented. An approach to establish the mechanical model based on dynamic stiffness and based on static stiffness is discussed.

2.1 Mechanical model of vehicle powertrain mounting systems

Modes are characterized as either rigid body or flexible body modes. Up to six rigid body modes, three translational modes and three rotational modes could exist in all structures [6]. If the structure bounces on some soft springs, its motion approximates a rigid body mode [7].

Because the powertrain's natural frequencies of the flexible modes are much higher than that of the mounting system, it can be modeled as a rigid body which has six degrees of freedom: three translational motions and three rotational motions [8]. The powertrain is supported on vehicle frame by some mounts. Each mount can be represented by three mutually perpendicular sets of spring and viscous damper in each of the three principal directions, which defined are as u , v and w . With the three-point mounting system as an example, the simplified mechanical model is shown in Figure 1.

Define a Cartesian coordinate system $G_0 - XYZ$ as system coordinate, the origin of the coordinate system G_0 is in static equilibrium position of powertrain's center of gravity. X axis points to the backward direction of vehicle, Y axis is parallel to the direction of engine crankshaft, and Z axis is vertically upward. Moving coordinate system $G_0 - xyz$ moves with the powertrain [9]. The generalized coordinates of the system is defined as three translational coordinates x, y, z relative to X, Y, Z , and three rotational coordinates $\theta_x, \theta_y, \theta_z$ around X, Y, Z . It can be written as

$$[q] = [x \quad y \quad z \quad \theta_x \quad \theta_y \quad \theta_z]^T . \qquad (1)$$
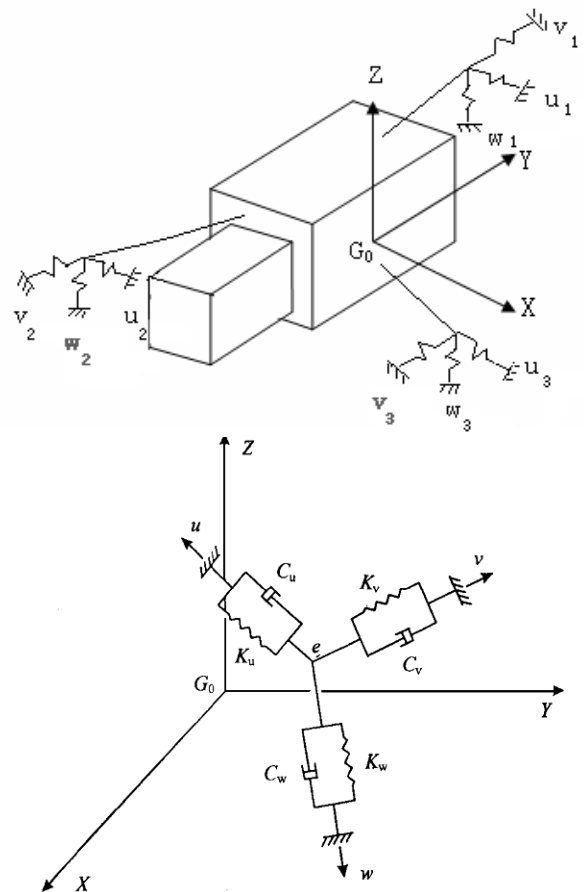

Fig.1 Mechanical model of mounting system

The dynamic equations of powertrain mounting system can be written as the matrix form:

$$[M][\ddot{q}] + [C][\dot{q}] + [K][q] = [F], \qquad (2)$$

where $[M]$ is mass matrix of the system, $[C]$ is damping matrix of the system, $[K]$ is stiffness matrix of the system and $[F]$ is a 6×1 vector of excitation forces and moments.

Ignoring damping and external force, the differential equations of the system's free vibration can be obtained. Analysis equation of system's inherent characteristics could be presented as

$$[M][\ddot{q}] + [K][q] = 0 . \qquad （3）$$

Mass matrix of the system $[M]$ can be written as

$$[M] = \begin{bmatrix} m & 0 & 0 & 0 & 0 & 0 \\ 0 & m & 0 & 0 & 0 & 0 \\ 0 & 0 & m & 0 & 0 & 0 \\ 0 & 0 & 0 & J_{xx} & J_{xy} & -J_{zx} \\ 0 & 0 & 0 & -J_{xy} & J_{yy} & -J_{yz} \\ 0 & 0 & 0 & -J_{zx} & -J_{yz} & J_{zz} \end{bmatrix} .$$

where m is the total mass of the powertrain, J_{ij} is the powertrain's moment of inertia around each axis. Stiffness matrix of the system $[K]$ can be written as

$$[K] = \sum_{i=1}^{n} [B_i]^T [T_i]^T [k_i][T_i][B_i],$$

where $[k_i]$ is stiffness matrix of mount i:

$$[k_i] = \begin{bmatrix} k_{u_i} & 0 & 0 \\ 0 & k_{v_i} & 0 \\ 0 & 0 & k_{w_i} \end{bmatrix}$$

$[B_i]$ refers to the position matrix of mount i:

$$[B_i] = \begin{bmatrix} 1 & 0 & 0 & 0 & z_i & -y_i \\ 0 & 1 & 0 & -z_i & 0 & x_i \\ 0 & 0 & 1 & y_i & -x_i & 0 \end{bmatrix}$$

where x_i, y_i, z_i is the coordinate of mount i.

$[T_i]$ is angle matrix between the elastic principal axis of mount and system coordinates, which can be written as

$$[T_i] = \begin{bmatrix} \cos\alpha_{u_i} & \cos\beta_{u_i} & \cos\gamma_{u_i} \\ \cos\alpha_{v_i} & \cos\beta_{v_i} & \cos\gamma_{v_i} \\ \cos\alpha_{w_i} & \cos\beta_{w_i} & \cos\gamma_{w_i} \end{bmatrix}.$$

For equation (3), assuming the theoretical solution is

$$[q] = [X_i]\sin(\omega_i t + \alpha). \tag{4}$$

Take equation (4) into (3), and the following equation can be obtained as

$$[K][X_i] = \omega_i^2 [M][X_i]. \tag{5}$$

Then

$$[K] - \omega_i^2 [M] = 0. \tag{6}$$

And the natural frequency of the system is

$$f_i = \omega_i / 2\pi. \tag{7}$$

The circular frequency of the system is the eigenvalues of the matrix $[M]^{-1}[K]$. And the corresponding shape of vibration is the eigenvector of the matrix $[M]^{-1}[K]$. Thus, the natural frequency and the shape of powertrain mounting system can be obtained by solving eigenvalues and eigenvectors of the matrix $[M]^{-1}[K]$.

2.2 Calculations of modal parameters based on the static stiffness of the mounts

The performance of powertrain mounting system depends on the stiffness characteristic of the mount, which is used to construct the stiffness matrix in the mechanical model. Static stiffness is the ratio between the static load variation and the displacement variation, which could be calculated by the equation:

$$k = \frac{\Delta F}{\Delta S},$$

where ΔF and ΔS can be tested by using Electronic universal testing machine. The schematic diagram of testing is shown in Figure 2.
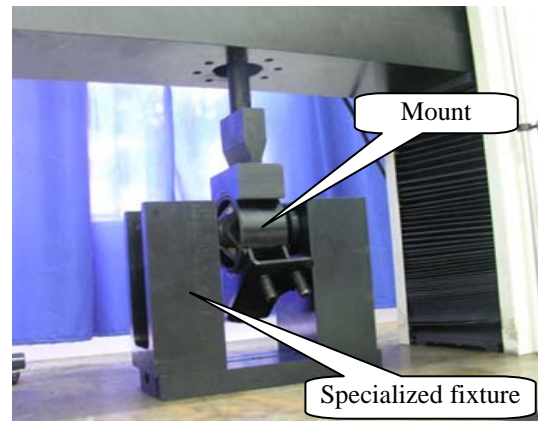

Fig.2 Static stiffness testing of the mount

The distinction is made between the static and the dynamic stiffness on the basis of whether the applied load has enough acceleration in comparison to the structure's natural frequency. A static load is one which varies sufficiently slowly. Static stiffness can be obtained in this process.

Take static stiffness values of each mount into the equation (6), and the natural frequencies of powertrain mounting system can be obtained. The results calculated are listed in Table 1.

Table 1 Calculated modal parameters of mounting system based on the static stiffness

Modes order	Natural frequency (Hz)
1	6.06
2	8.17
3	8.59
4	12.71
5	16.07
6	20.41

2.3 Calculations of modal parameters based on the dynamic stiffness of the mounts

A dynamic load is one which changes with time fairly quickly in comparison to the structure's natural frequency. In general, dynamic stiffness depends on three factors: preload, excitation

frequency and the amplitude of dynamic load [10]. For powertrain mounting system supported by three mounts, the value of preload equals to the total mass of powertrain, which is distributed to each mount at vertical direction. This is a space force problem, which can be easily calculated. The excitation frequency and amplitude of dynamic load are determined by the corresponding working conditions [11]. The value of excitation frequency and amplitude of dynamic load at certain operating condition can be obtained through vibration measurement. Then exerting the load of equivalent frequency and amplitude on the mount using vibration testing equipment, the value of dynamic stiffness can be acquired properly. This simulation of testing condition is consistent with the actual situation.

In typical idle speed condition of the engine, for example, acceleration response signals can be measured by accelerometer fixed on the side of mount [12]. The excitation frequency on the mount can be obtained through using the Fast Fourier Transform (FFT) of acceleration response signals [13]. And the amplitude of dynamic load can be got by twice integration of acceleration response signals. A sample of frequency spectrum response in the vertical direction from left mount under idle speed condition can be seen in Figure 3. The diagram shows that the vibration energy of left mount is mainly concentrated in 27Hz. It means that the dominant frequency of excitation is 27Hz. So the input signals of 27Hz can be excited on the mount in the dynamic stiffness testing.
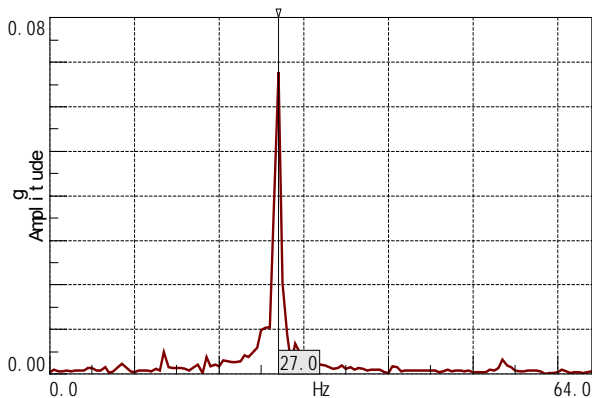

Fig. 3 Frequency spectrum of left mount in the idle speed condition

Using the approach described above, the preload, dominant excitation frequency and amplitude of input signals can be obtained. Then the dynamic stiffness of all mounts at all directions can be measured by vibration testing machine [14]. The schematic diagram of dynamic testing is shown in Figure 4.
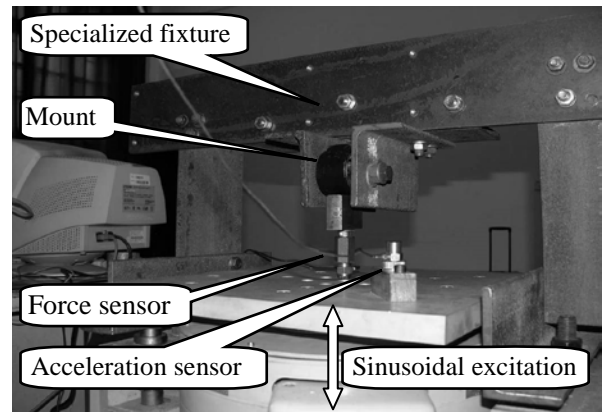

Fig. 4 Dynamic stiffness testing of the mount

Take dynamic stiffness values of each mount acquired through the approach above into the equation (6), the natural frequencies of powertrain mounting system in the idle speed condition can be obtained. The results calculated are listed in Table 2.

Table 2 Calculated modal parameters of mounting system based on the dynamic stiffness

Modes order	Natural frequency (Hz)
1	8.05
2	11.85
3	12.58
4	13.16
5	15.57
6	16.52

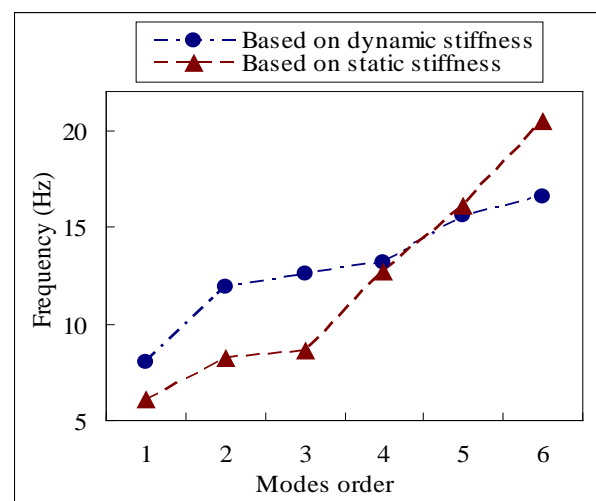## 2.4 Comparison between two groups of calculation results

Fig.5 Comparison between two groups of calculation results

Natural frequencies calculated from the mechanical model based on static stiffness are compared with the results based on dynamic stiffness, shown in Figure 5. Obvious differences between the two groups of results could be found out in the graph. The relative error of the third order frequency reached 46.4%, which indicates that calculation model based on static stiffness is not consistent with the actual working condition. As for the method based on dynamic stiffness, it's required to be further verified by the experimental modal analysis.

3 Experimental modal analysis of powertrain mounting system

In this section, the experimental modal analysis is used to identify the modal parameters of powertrain mounting system. Through contrastive analysis of the experiment and the theoretical calculation, the rationality of calculation model based on dynamic stiffness can be verified.

3.1 Experimental modal analysis based on the impact hammer testing

Impact testing is the most classical modal testing method used today. According to the theory of experimental modal analysis, the modal parameters of the system can be identified through the curve fitting of the Frequency Response Function (FRF), which is the ratio between a response (output acceleration) signal and a reference (input force) signal expressed in frequency domain [15]. Impulse force can be provided by the impact hammer which has a force transducer attached to its head to measure the input force. Acceleration response signals can be measured using accelerometers fixed on the surface of the system [16]. The process of impact hammer testing is depicted in Figure 6 and Figure 7.
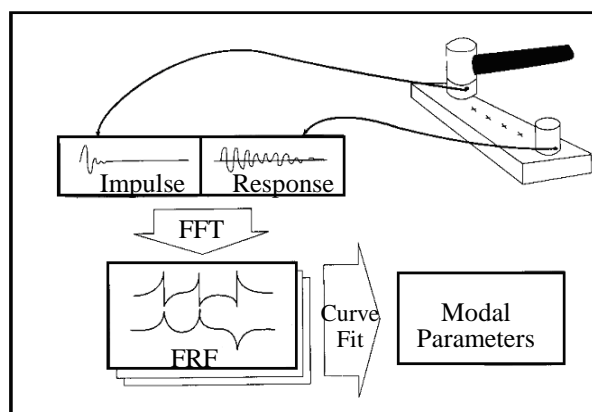

Fig. 6 Method of experimental modal analysis

Different sized hammers can provide different impact forces. Here, the test is performed respectively using two kinds of impact hammers with different sizes. The magnitude of impact force produced by big hammer is about ten times of the small hammer.
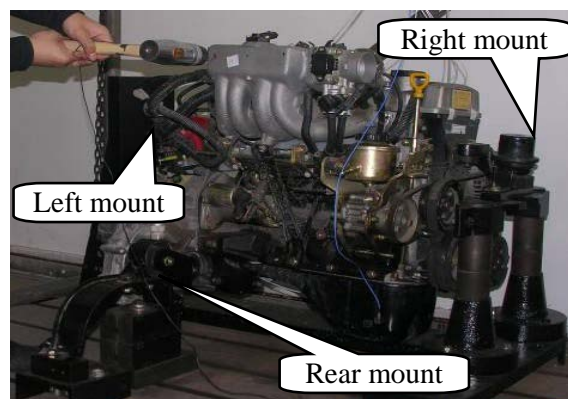

Fig. 7 Impact hammer testing

Fig. 8 Different size of impact hammer

Table 3 lists the experimental results identified through these two impact testings.

Table 3 Experimental modal results of mounting system based on the impact hammer testing

Modes order	Natural frequency (Hz)		Relative error
	Small hammer	Big hammer	
1	6.91	6.95	0.6%
2	7.74	8.23	6.3%
3	8.36	9.60	14.8%
4	10.40	13.07	25.7%
5	13.46	16.12	19.8%
6	17.24	19.24	11.6%

According to the characteristics of the Frequency Response Function, the magnitude of impact force should not influence the results in experimental modal analysis. Table 4 shows that there are obvious

differences between the modal parameters obtained from the testing using small hammer and those using big hammer. This phenomenon is due to the non-linear relationship between the dynamic stiffness of the mount and the frequency and amplitude of the excitation force. The stiffness performance of the mount varies under different impact forces.

These experiments indicate that the modal parameters will vary with working conditions due to the change of dynamic stiffness values of mounts. Therefore, It is unable to obtain accurate modal parameters of mounting system under actual working conditions using traditional experimental method based on impact hammer testing.

3.2 Operational modal analysis

Traditional Experimental modal identification method and procedure such as impact hammer testing is based on forced excitation laboratory tests during which Frequency Response Functions are measured. However, the real loading conditions to which the mounting system is subjected often differ considerably from those used in laboratory testing. The method of operational modal analysis is to extract modal frequencies, damping and mode shapes from data taken under operating conditions [17]. This means that environmental and its natural excitation influences on system behavior, such as preload of mounts and dynamic-load-induced stiffening, will be taken into account [18]. The identification results under the actual working status will more accurately reflect the actual dynamic characteristics of the mounts [19].
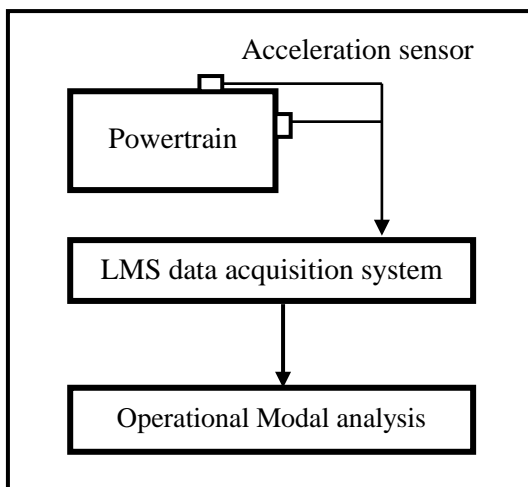

Fig. 9 Testing and analysis procedures of operational modal analysis

Attach the accelerometers on the surface of the powertrain, Measure the vibration response signals of the powertrain mounting system in idle speed condition and the modal parameters can be obtained by using identification methods of operational modal analysis. Testing and analysis procedures are shown in Figure 9, Figure 10 and Figure 11.

Fig. 10 Test environment and real operating status of powertrain mounting system

Fig. 11 Online acquisition and analysis

The experimental modal parameters of powertrain mounting system identified by operational modal analysis at the idle speed condition are listed in Table 4. Evaluation of the results will be given in the next section.

Table 4 Experimental modal results of mounting system based on operational modal analysis

Mode order	Natural frequency (Hz)
1	8.50
2	12.34
3	13.38
4	13.85
5	16.92
6	17.76

3.3 Results comparison of the three experimental modal analyses

Experimental modal analyses of powertrain mounting system have been performed described above using three different test methods and means. Thus, three groups of natural frequency of the mounting system have been obtained already. The comparison of these experimental identification results is shown in Figure 12.
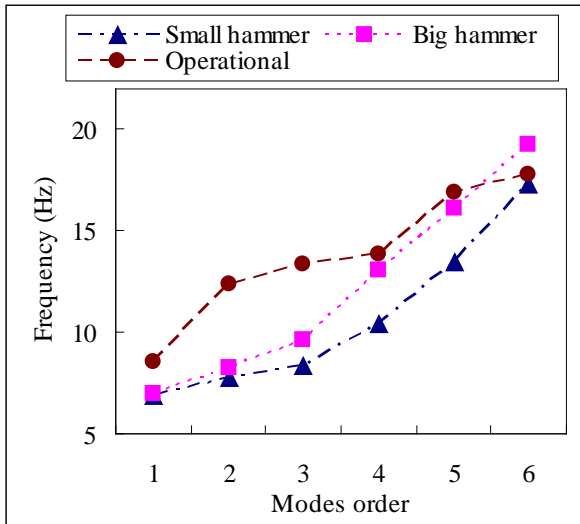
Fig. 12 Results comparison of the three experimental modal analyses

The graph shows that there are obvious differences among the three groups of results. In theory, the natural frequency of a linear system is the inherent property of the system itself, and should not vary with the working conditions or the excitation force. The essential difference between the three experiments is the exciting force imposed on the mounting system, which causes the dynamic characteristics of mounts to change accordingly. Thus the inherent dynamic characteristics of the entire mounting system also vary with the different conditions. The result of contrastive analysis verified that the modal parameters of powertrain mounting system will change with different working conditions from the perspective of experimental analysis.

4 Comparison between theoretical calculation and experimental analysis

In Figure 13, Natural frequencies, which calculated from the theoretical model respectively based on static stiffness and that based on dynamic stiffness, are compared with the experimental results identified using the method of operational modal analysis.
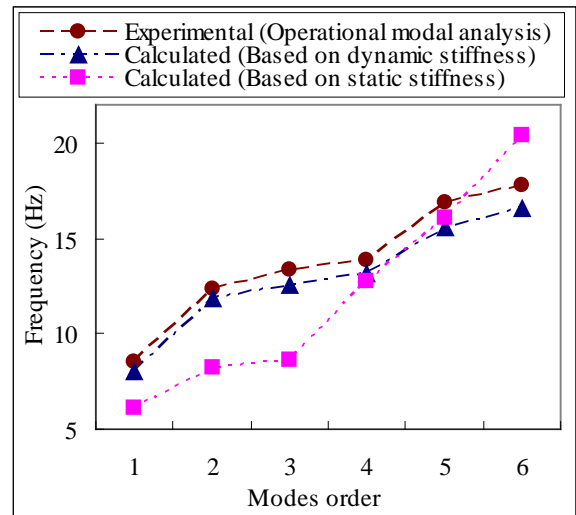
Fig. 13 Results Comparison between theoretical calculation and experimental analysis

Obviously, there is smaller relative error and higher fitting degree between the calculation results based on dynamic stiffness of mounts and the results obtained from operational modal analysis, and the relative error between them of every order frequency is less than 8% as shown in Figure 14. While, there is large error between the calculation results based on static stiffness and the results obtained from operational modal analysis, for example, the relative error between them of the third order frequency is even more than 55%.
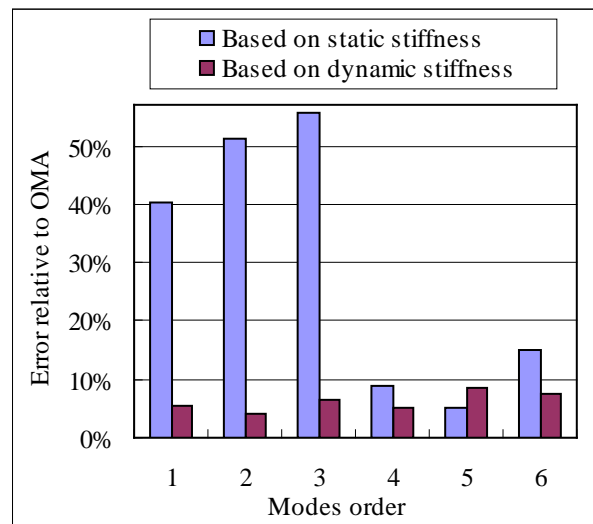
Fig. 14 Error of theoretical calculation relative to operational modal analysis

In the testing of dynamic stiffness, frequency and amplitude of excitation force applied on the mount is measured under the idle speed condition, and the operational modal analysis is performed at the same condition. So it could conclude that the mechanical

model proposed based on such dynamic stiffness can simulate the dynamic behavior of the mounting system under the actual working condition more accurately.

5 Conclusions

Dynamic stiffness of the mount has a strong nonlinear nature. The performance of dynamic property of the mount is influenced by multiple factors and strongly depends on the working conditions. This means that the modal parameters of powertrain mounting system could make changes under the different operating conditions. [20-27]

In this paper, a novel approach to simulate the actual working condition is proposed in the testing of dynamic stiffness. Based on the experimental and evaluation results, it is shown that stiffness matrix in the mechanical model of powertrain mounting system should be constructed based on dynamic stiffness of mounts in the process of theoretical calculations. The input parameters required to test the dynamic stiffness, such as temperature, preload, frequency and amplitude of excitation force applied on the mount, should also be measured under the actual working conditions corresponding to the study goal.

Aspect of the experimental modal analysis of powertrain mounting system, using the method of traditional impact hammer testing may produce large error and the identification results of modal parameters are not reliable, while identification using the method of Operational Modal Analysis could obtain more accurate and more reliable results which are also consistent with the actual working condition.

Acknowledgement

References:
1. [1] Yunhe Yu, Nagi G Naganathan and Rao V. Dukkipati, *A literature review of automotive vehicle engine mounting systems*, Mechanism and Machine Theory, Vol. 36, No. 1, 2001, pp. 123-142.
2. [2] Yao Li, Jianwu Zhang and Xiqiang Guan, *Estimation of Vehicle Parameters and Road Friction Using Steering Torque and Wheel Speeds*, WSEAS Trans. on SYSTEMS, Vol. 11, No. 1, 2012, pp. 1-11.
3. [3] Vazquez Alvarez, J.J. Ocampo Hidalgo, C. Aviles Cruz and A. Ferreyra Ramirez, *Vehicle mathematical model reduction considering the brake system dynamics*, WSEAS Trans. on SYSTEMS, Vol. 9, No. 2, 2010, pp. 156-166.
4. [4] Karen, Idris, Kaya Necmettin, Oztürk, Ibrahim, *Optimisation of vehicle engine mount system using simulation-based design approach*, International Journal of Vehicle Design, Vol. 52, No. 1, 2010, pp. 38-53.
5. [5] Shangguan Wen-Bin and Chen, Da-Ming, *A method for calculating the displacements of a powertrain and mounts in a powertrain mounting system including torque struts under quasi-static loads*, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, Vol. 226, No. 5, 2012, pp. 634-647.
6. [6] Huan-Jung Lin and Tain-Sou Tsay, *Modeling Identification and Simulation of Bank to Turn Unmanned Aerial Vehicle*, WSEAS Trans. on SYSTEMS, Vol. 10, No. 4, 2011, pp. 91-103.
7. [7] Jae-Yeol Park and Rajendra Singh, *Role of spectrally varying mount properties in influencing coupling between powertrain motions under torque excitation*, Journal of Sound and Vibration, Vol. 329, No. 14, 2010, pp. 2895-2914.
8. [8] Ali El Hafidi, Bruno Martin and Alexandre Loredo, *Vibration reduction on city buses: Determination of optimal position of engine mounts*, Mechanical Systems and Signal Processing, Vol. 24, No. 7, 2010, pp. 2198-2209.
9. [9] Zhang Xin and Zhang Jianwu, *Performance Analysis of Limited Bandwidth Active Suspension with Preview Based on a Discrete Time Model*, WSEAS Trans. on SYSTEMS, Vol. 9, No. 8, 2010, pp. 834-843.
10. [10] Yun-qing Zhang and Wen-Bin Shangguan, *A novel approach for lower frequency performance design of hydraulic engine mounts*, Computers & Structures, Vol. 84, No. 8, 2006, pp. 572-584.
11. [11] Park Jae-Yeol and Singh, Rajendra, *Analysis of power train motions given a combination of active and passive isolators*, Noise Control Engineering Journal, Vol. 57, No. 3, 2009, pp. 232-243.

12.   [12] Lee Jun Hwa and Singh, Rajendra, *Nonlinear frequency responses of quarter vehicle models with amplitude-sensitive engine mounts*, Journal of Sound and Vibration, Vol. 313, No. 5, 2008, pp. 784-805.

13.   [13] Ranglin Fan and Zhenhua Lu, *Fixed points on the nonlinear dynamic properties of hydraulic engine mounts and parameter identification method: Experiment and theory*, Journal of Sound and Vibration, Vol. 305, No. 4, 2007, pp. 703-727.

14.   [14] Christof Devriendt, Gert De Sitter, Steve Vanlanduit and Patrick Guillaume, *Operational modal analysis in the presence of harmonic excitations by the use of transmissibility measurements*, Mechanical Systems and Signal Processing, Vol. 23, No. 3, 2009, pp. 621-635.

15.   [15] Van Der Auweraer H., P. Guillaume, P. Verboven and S. Vanlanduit, *Application of a fast-stabilizing frequency domain parameter estimation method*. ASME Journal of Dynamic Systems, Measurement, and Control, Vol. 123, No. 4, 2001, pp. 651-658.

16.   [16] Tian Ran Lin, Nabil H. Farag and Jie Pan, *Evaluation of frequency dependent rubber mount stiffness and damping by impact test*, Applied Acoustics, Vol. 66, No. 7, 2005, pp. 829~844.

17.   [17] Peeters B, Van der Auweraer H, Guillaume P and Leuridan J, *The PolyMAX frequency domain method: a new standard for modal parameter estimation*, Shock And Vibration, Vol. 11, No. 4, 2004, pp.395-409.

18.   [18] F. Javier Cara, Jaime Carpio, Jesús Juan and Enrique Alarcón, *An approach to operational modal analysis using the expectation maximization algorithm*, Mechanical Systems and Signal Processing, Vol. 31, 2012, pp. 109-129.

19.   [19] Edwin Reynders and Guido De Roeck, *Reference-based combined deterministic stochastic subspace identification for experimental and operational modal analysis*, Mechanical Systems and Signal Processing, Vol. 22, No. 3, 2008, pp. 617~637.

20.   [20] F. Neri, *Agent Based Modeling Under Partial and Full Knowledge Learning Settings to Simulate Financial Market*, AI Communi- cations, 25 (4), 2012, IOS Press, pp. 295-305.

21.   [21] Mohammad Ali Pakzad, Sara Pakzad, *Stability Map of Fractional Order Time Delay Systems*, WSEAS Trans. on SYSTEMS, Issue 10, Vol. 11, October 2012, pp. 541-550.

22.   [22] Mohammad Ali Pakzad, *Kalman Filter Design for Time Delay Systems*, WSEAS Trans. on SYSTEMS, *Issue* 10, Vol. 11, October 2012, pp. 551-560.

23.   [23] Roman Prokop, Jiri Korbel, Radek Matusu, *Autotuning Principles for Time-delay Systems*, WSEAS Trans. on SYSTEMS, Issue 10, Vol. 11, October 2012, pp. 561-570.

24.   [24] Libor Pekar, *A Ring for Description and Control of Time-Delay Systems*, WSEAS Trans. on *SYSTEMS*, Issue 10, Vol. 11, October 2012, pp. 571-585.

25.   [25] Petr Dostal, Vladimir Bobal, Zdenek Babik, *Control of Unstable and Integrating Time Delay Systems Using Time Delay Approximations*, WSEAS Trans. on SYSTEMS, Issue 10, Vol. 11, October 2012, pp. 586-595.

26.   [26] Vladimir Bobal, Petr Chalupa, Marek Kubalcik, Petr Dostal, *Identification and Self-tuning Control of Time delay Systems*, WSEAS Trans. on SYSTEMS, Issue 10, Vol. 11, October 2012, pp. 596-606.

27.   [27] Marek Kubalcik, Vladimir Bobal, *Predictive Control of Higher Order Systems Approximated by Lower Order Time-Delay Models*, WSEAS Trans. on SYSTEMS, Issue 10, Vol. 11, October 2012, pp. 607-616.

28.   .

29.