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Research Article

Abnormal Noise Source Identification and Control for Automobile Transmission in the Neutral Idle Condition

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Abstract: Aiming at the abnormal noise of a domestically-made automobile transmission in the neutral idle condition, seriously affecting the vehicle market competitiveness and the riding comfort ability for customers, the objective of this study to reduce the noise and vibration of the automobile transmission by accurately identifying the noise source of the transmission in the neutral idle condition. For this purpose, based on the working characteristics of the transmission, modal analysis of automobile transmission housing is formulated using 3D graphics software Pro/E together with Finite Element Method. In addition, the calculation of meshing frequency of gear pair is conducted also. Finally, through comparing model analysis results to the calculation results, it is indicated that the gear meshing impact noise of the third gear pair was identified as the noise resource of the automobile transmission in neutral idle condition, which will provide the theoretic basis to analyze its dynamic characteristics of the transmission as well as its improvement to reduce vibration and noise.

Keywords: Idle speed, mesh frequency, neutral position, transmission, vibration and noise

INTRODUCTION

Along with the rapid development of science technology and automobile industry, the increasingly stringent control requirements on vehicle noise were put forward by the national regulations, as well as car buyers have higher and higher demand for riding comfort ability, the vibration and noise has become one of the key indexes of vehicle performance. The noise of automotive power transmission system has a very large proportion in the whole automobile noise, furthermore automotive transmission is the key parts of automobile power transmission system, thereby reducing the noise of automobile transmission has very important significance for reducing the noise of the whole vehicle (Shi *et al*., 2010, 2011). The so-called neutral idle abnormal noise phenomenon, refer to a significant noise arising from the transmission housing as long as the transmission coupled to the clutch, when the vehicle is in a suspended state, the transmission hanging in the neutral position, the engine running at idle speed (about 700-800 rpm), which can be obviously distinguished from the sound with the clutch disengaged from the transmission (Feng, 2010). A domestically-made automobile, by using three shafts and five speed transmission, exists obvious noise in neutral idle condition, which can be clearly heard by car drivers and seriously affects the ride comfort. The abnormal noise greatly reduces the vehicle sound quality, makes it easy for customers to produce the illusion that the vehicle has a design and manufacturing quality problems, moreover has seriously affected the vehicle market competitiveness (Chu *et al*., 2009).

A large number of studies show that the main source of the high noise of transmission in the idle condition mainly includes two aspects: one is the gear meshing noise generated by mutually alternating meshing of each gear pair when the transmission was at work; two is the fluctuation of the engine output torque and rotational speed, causing the normal gear meshing relationship destroyed, thus excited vibration and then emitted the impact noise (He *et al*., 1999; Chu *et al*., 2005). Based on the comprehensive analysis of the noise radiation mechanism of the transmission housing, the study gives a finite element modal analysis to the whole transmission, studies on the inherent frequency and vibration condition of the transmission housing, as well as meshing frequency of every gear pair of the transmission in the neutral idle condition. Through comparative analysis between the inherent frequency of the transmission housing and meshing frequency of every gear pair, the noise resource in the neutral idle condition was quickly and accurately identified and the weak spots of transmission is intuitively analyzed, which has laid a solid foundation on the dynamics optimization design of the transmission, in order to ensure that certain inherent frequency of transmission should avoid the meshing frequency of every gear pair of the transmission in the neutral idle condition. The research results offer

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foundation for further establishing the noise control measure, give some rational improvement proposals for local structure of the transmission with severe vibration, which aims at reducing vibration and noise.

Noise source identification: The radiation noise of the transmission housing accounts for more than 90% of the whole transmission system noise, therefore, through simulating the real motivation acting on the transmission, the study on the noise radiation of the transmission housing under the action of the motivation is very meaningful. There are two general ways in which the transmission usually spreads the noise outward: one is the direct formation of the noise radiation due to the metal bump, which will produce reverberation in the transmission housing, where into, only a small portion of noise radiates out through the transmission housing wall, most of the remaining acoustic energy was dissipated by structural damping and other damping element; two is the structure vibration due to gear bump, which will be transferred through the gear, gear shaft, the bearing to the transmission housing and then produce the large vibration in the sensitive parts of the transmission housing to radiate the noise. The latter is a more effective spreading route and playa a leading role on noise radiation of the transmission (Okamura, 1996). Considering that the above transmission radiation noise comes mainly from the vibration excitation of the transmission housing due to the gear meshing impact, this study will use the modal analysis of the transmission housing to determine the gear pair which has significant contribution on the abnormal noise of the transmission in neutral idle condition.

Modal analysis of the transmission housing: Automobile transmission is a multiple degree of freedom vibration system. A variety of exciting force acting on this system is the power and source to make the transmission produce complex vibration (Wang *et al*., 2007; Huang *et al*., 2011). Modal analysis is the base for vibration and noise analysis. Through the implementation of modal analysis on the transmission housing, the resonance phenomenon could be avoided in the structural design of the transmission housing as far as possible (Li, 2008b). In this study, we will take a certain type of transmission for example, firstly establishes the three-dimensional solid model of the transmission and then establishes the finite element model of the transmission to perform modal analysis.

Establishment of 3D model of transmission housing: The practical model of automobile transmission is quite complex, distributed with various reinforcing ribs, bosses, oil drain hole, corners and fillets and all kinds of bolt connection holes and so on. The gear box comprises five forward gears and one reverse gear. The gear box adopts a constant mesh gear transmission. The synchronizer, as gear shift mechanism, can make the input power be transmitted through different gear pairs, while the realization of the different transmission ratio.

Fig. 1: 3D assembly model of the transmission housing and the clutch

In order to exert the result of modal analysis to be close to the actual situation to the fullest extent, simplification of transmission structure should be reduced to as little as possible, to ensure that the calculation results have high precision, can more truly reflect the vibration characteristics of transmission structure. However, too small feature structure in the transmission housing has almost no effect on modal analysis, moreover, these small structures, such as small screw thread hole or sharp fillet, can generate thousands of nodes and elements when establishing the finite element model. Too small feature structure needs to divide into very small cell, thereby increasing the amount of data processing, can result from the inability of analysis. Therefore, simplification of transmission structure is necessary. Considering that the mass matrix and stiffness matrix of 3D solid model of transmission housing could not completely accord with the actual situation, we will, based on the equivalence principle, establish a simplified model of transmission housing and give full consideration to the factors that play a dominant role in the modal analysis of transmission housing. The powerful wildfire 4.0 Pro/E software is applied to establish the three-dimensional entity assembly model of the transmission housing and the clutch as shown in Fig. 1.

Establishment of FEA model of transmission housing: The finite element mesh generation is the key technique of the Computer Assistance Engineering analysis (CAE). A large number results of test research indicates that the meshing density is a key step in Finite Element analysis, which has great influence on computing time and accuracy of Finite Element analysis, so both aspects must be weighed correctly while determining the meshing density. Generally speaking, the smaller the meshing density is, the higher the calculation accuracy is, but the longer the computing time is proportionately and otherwise, it will make a big difference. Under ideal operation condition, the needed meshing density may occur when the following conditions occur: the results of finite element

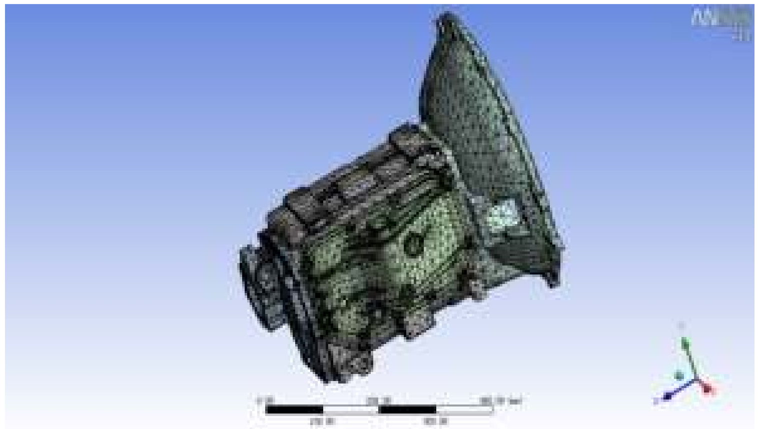

Fig. 2: Meshing the finite element model of the transmission housing and the clutch

analysis don't vary with the increase of the numbers of the mesh (Li *et al*., 2008a; Li *et al*., 2012). Based on such consideration above, the author gives a path breaking study of the model analysis of transmission housing in different meshing densities, which focuses on comparing all analysis results with different meshing densities.

In this study, the material parameters used in calculation are as follows: the material is grey cast iron HT200, material density is 7.2×10^3kg/m^3, elasticity module E = 110 GPa, Poisson's ratio μ = 0.28 . Three-dimensional entity model from Pro/E software can be imported directly into FEA software, all kinds of property from entity model in the Pro/E can be inherited entirely, which includes initial position parameter and mass property of all component assembly of the transmission housing and the clutch. The mesh generation of finite element model of the transmission housing and the clutch has a total of node 155364, element 268289, as shown in Fig. 2.

Solution of finite element model of transmission housing: In the structure dynamics problem, structural inherent frequency and vibration mode are the basis of dynamic analysis. In the case of undamped free vibration, structural inherent frequency and vibration mode can be converted to Eigen value and eigenvector problem. Free vibration of N degree of freedom undamped system can be expressed as:

$$[m]\left\{\ddot{q}(l)\right\}+[K]\{q(l)\}=\{0\} \tag{1}$$

Due to the free vibration of elastic body can be decomposed into a series of simple harmonic vibrations. So the solution of Eq. (1) can be assumed to be:

$$\{q(l)\}=\{u\}\cos(\omega t-\varphi) \tag{2}$$

In the equations: ω-real number, the frequency of simple harmonic motion; φ- arbitrary constant.

The Eq. (2) is substituted into Eq. (1) and then get:

$$[K]\{u\}-\omega^2[m]\{u\}=0 \tag{3}$$

This is N variables homogeneous linear algebraic equations about {u}.

Necessary and sufficient condition of the equations with non-zero solution is that its determinant of the coefficients equal to zero, that is:

$$\left|k_{ij}-\omega^2 m_{ij}\right|=0 \tag{4}$$

This equation is known as the system frequency equation, the determinant is called the characteristic determinant.

The determinant can be unfolded to get the n order algebraic expression about ω^2: $\omega^{2n}+\alpha_1\omega^{2(n-2)}+\ldots+\alpha_{n-1}\omega_2+\alpha_n=0$.

It is assumed that the mass matrix and stiffness matrix is positive definite real symmetric matrix, it can be demonstrated in mathematics, n roots of the frequency equation $k_{ij}-\omega^2 m_{ij}| = 0$ are all positive real root, which correspond to the n natural frequencies of the system. There into, if the roots are not equal, that is to say, there are no repeated roots, the natural frequencies are arranged in ascending order $\omega^2_1<\omega^2_2<\ldots<\omega^2_n$, each ω_r (r = 1, 2,…, n) is respectively substituted into Eq. (3) to obtain the corresponding {u(r)} and this is the system modal vector or mode vector.

Modal analysis is used to determine the vibration characteristics of the structure or parts of the machine (inherent frequency and modal shape), the inherent frequency and modal shape are important parameters in the design of structure under the dynamic loads (Zhai *et al*., 2006). Considering that modal analysis is a linear analysis, any non-linear characteristics even definition also will be ignored. Modal analysis is determined by the inherent characteristics of the system and is independent on the external load, it is not necessary to set the boundary and loading conditions. Generally,

Table 1: Previous 8 orders modal frequencies of the transmission housing and the clutch

Modal order	1	2	3	4	5	6	7	8
Frequency (Hz)	303.49	327.18	533.56	572.36	675.88	759.17	880.78	946.81

zero displacement constraints are only applied to modal analysis.

Modal analysis results of transmission housing: Based on the modal analysis of transmission, the natural frequencies and mode shapes of the transmission housing are studied and transmission vibration sensitive parts are also theoretically analyzed and calculated. In finite element analysis, in order to simulate test conditions of transmission, zero displacement constraints are attached in the input shaft cover connecting section to go in for calculation of the transmission housing constrained mode. In general, the main reason of causing engine resonance is the lower order frequency of transmission housing. Therefore, in the use of ANSYS solution and extension mode, it is enough to expands and extract the former 14 order frequencies of transmission. When conducting modal analysis of the transmission housing in the boundary condition, the first 6 order modes are close to zero; this is so-called rigid body mode. Therefore, the truly meaningful mode should be started from seventh order mode. In this study, ANSYS workbench is used to calculate the first 14 order modes, remove the first 6 order rigid body mode and extract the first 8 order non-zero modes as shown in Table 1.

Meshing frequency of gear pair:

Excitation of gear meshing to produce noise: In automobile transmission system, the transmission is the second most important component part only to the engine, directly relates to the automobile manipulation of fun and ride comfort. The transmission is mounted between the clutch and the transmission shaft, adjusts the power and speed of the engine, then transfers to a drive shaft, plays an important role in the allocation function.

The transmission is a kind of multi gear pair transmission machinery; it is inevitable to produce different degrees of running noise in any mechanical drive. Different from the single gear transmission mechanism, the transmission noise is much more complex, which is the irregular random combinations of many sound waves with different frequencies and intensity. The signal of sound waves will be distributed over the entire frequency range. Hence, as far as the automobile is concerned, the transmission is also a passive vibration noise source and excitation source.

Calculation of meshing frequency of gear pair: In general, 5 gear and constant mesh gear pair of the transmission are in motion in neutral idle condition. Some studies show that the abnormal noise source of the transmission in neutral idle condition are mainly for meshing gear pairs when the processing and assembly precisions meet the requirement, so it is necessary to calculate the meshing frequency of every gear pairs (Liang *et al*., 2006).

The gear pair meshing frequency calculation formula can be expressed as:

$$f = nz/60 \tag{5}$$

Axial rotation frequency formula can be expressed as:

$$f_a = n/60 \tag{6}$$

There in, n is the speed of axis (r/min).

The so-called neutral idle abnormal noise phenomenon, refer to a significant noise arising from the transmission housing as long as the transmission coupled to the clutch, when the vehicle is in a suspended state, the transmission hanging in the neutral position, the engine running at idle speed (about 700-800 rpm), which can be obviously distinguished from the sound with the clutch disengaged from the transmission.

The speed range of the transmission input shaft in the neutral idle condition is about 700-800 rpm; this study respectively calculates meshing frequency of the transmission gear pair in three speed points that is 700 rpm, 750 rpm and 800 rpm. When the transmission input shaft speed in the neutral idle condition is 700 rpm, input shaft rotation frequency is 11.667 Hz. So the corresponding meshing frequency of each gear pair as shown in Table 2.

When the transmission input shaft speed in the neutral idle condition is 750 rpm, input shaft rotation frequency is 12.5 Hz. So the corresponding meshing frequency of each gear pair as shown in Table 3.

When the transmission input shaft speed in the neutral idle condition is 800 rpm, input shaft rotation frequency is 13.33 Hz. So the corresponding meshing frequency of each gear pair as shown in Table 4.

By solving the finite element model of the transmission housing and the clutch, the first and second order modal frequencies of the transmission housing and the clutch are 303.48 and 327.18 Hz respectively, at the same time, through the calculation of the gear pair meshing frequency in the neutral idle condition, the meshing frequencies of the third gear pair are 278.409, 298.295 and 318.1818 Hz, respectively close to the first and second order modal frequencies of the transmission housing and the clutch, which will cause the resonance phenomenon and then generate the abnormal noise.

Table 2: Meshing frequency of each gear pair with the input shaft speed 700 rpm

Gears	First	Twice	Third	Fourth	Fifth
Number of teeth	14	22	35	44	54
Mesh frequency /Hz	111.3636	175	278.409	350	429.5454

Table 3: Meshing frequency of each gear pair with the input shaft speed 750 rpm

Gears	First	Twice	Third	Fourth	Fifth
Number of teeth	14	22	35	44	54
Mesh frequency/Hz	119.318	187	298.295	375	460.2273

Table 4: Meshing frequency of each gear pair with the input shaft speed 800 rpm

Gears	First	Twice	Third	Fourth	Fifth
Number of teeth	14	22	35	44	54
Mesh frequency/Hz	127.2727	200	318.1818	400	490.9091

After comprehensive evaluating the transmission mode analysis results, combined with the operation principle of transmission, finally, the gear meshing impact noise of the third gear pair was identified as the noise resource of the automobile transmission in neutral idle condition.

CONCLUSION

In this study, in response to the neutral idle abnormal noise phenomenon arising from a domestically-made automobile, based on the establishment of the finite element model of the transmission housing and its modal analysis, the natural frequency of the transmission housing was estimated and analysis results can be generated to find transmission vibration sensitive parts. In addition, the calculation of meshing frequency of gear pair is conducted also. Through comparing model analysis results to the calculation results, it is indicated that the gear meshing impact noise of the third gear pair was identified as the noise resource of the automobile transmission in neutral idle condition, which has theoretical significance value to some extent for the dynamics optimization design of the transmission structure and provides the theoretic basis to analyze its dynamic characteristics of the transmission as well as its improvement to reduce vibration and noise.

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