Optimization of Vibration Damping for the Power Assembly Suspension System Based on Ant Colony Algorithm

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***Abstract* - In this paper, modelling and simulation calculation were carried out on the suspension system of a MPV (Multi-Purpose Vehicle) via ADAMS software. Multiple simulation calculations were carried out by revising the model parameters repeatedly, and finally a model consistent with the data of idling vibration experiment was obtained. On the basis of the revised model, a mathematical model for the optimization design of vibration damping for the power suspension system of MPV was built, with the inherent frequency constraints of the suspension system, the vibration displacements and the turning angle constraints of the power assembly system along X axle and Y axle as the constraint conditions, the rigidities of the suspension components in the 3 main directions as well as the installation angles of the left and right front suspensions of the engine as the design variables, the 6-degree-of-freedom decoupling of the suspension system as the objective function. The optimization design of vibration damping for the suspension system of the engine was completed by means of ant colony algorithm. Simulation analysis was carried out on the dynamic response of the optimized model in terms of time domain and frequency domain, and relevant data and curves were obtained. The simulation results show that the optimized model can control the highest order inherent frequency of the suspension system within the range required by the design, and can make the decoupling rate along Z axle and axle reach over 90%. Meanwhile, the transmissibility of the optimized suspension system was calculated, which concludes that the optimized transmissibility is reduced. The results before and after the optimization indicate that the optimized suspension system has good vibration damping effect.**

Keywords - Engine suspension system; Ant colony algorithm; Vibration mode decoupling; Optimization of vibration damping

I. INTRODUCTION

With the development of science and technology, people have an increasing high requirement on the ride comfort of automobiles. The vibration and noise occurring during the automobiles' starting, running, acceleration and braking directly affect the passengers' ride comfort. Engine is a main vibration source on the automobile. Its vibration is transmitted to the car frame or car body by the engine through the suspension system. Moreover, the vibration caused by road disturbance force and the vibration from the engine can be mutually affected through the suspension. Therefore, the design of the engine's vibration isolation system is of great importance to the vibration and noise reduction of an automobile, and is very important for improving the automobiles' ride comfort.

With several decades' development, the production technology of Hydraulically Damped Mount is increasingly mature. In 1920s, people found that rubber materials which had good elasticity and damping characteristics could absorb the mechanical energy from vibration by its internal damping [1]. By making use of the characteristics of the rubber materials, designers bonded the rubber materials and metal components by vulcanization and made rubber suspension components to isolate the vibration energy of the engine [2]. After rubber suspension, the US firstly put forward the idea of combining hydraulic vibration damper and traditional rubber suspension, and Hydraulically Damped Mount (HDM) occurred [3]. Hydraulically Damped Mount overcomes the defects of rubber suspension in terms of dynamic characteristics. It has large damping and high rigidity under the excitation with low frequency and large amplitude, and has small damping and low rigidity under the excitation with high frequency and small amplitude [4-5].

In this paper, a mathematical model for the optimization design of vibration damping for the power suspension system of MPV was built, with the inherent frequency constraints of the suspension system, the vibration displacements and the turning angle constraints of the power assembly system along X axle and Y axle as the constraint conditions, the rigidities of the suspension components in the 3 main directions as well as the installation angles of the left and right front suspensions of the engine as the design variables, the 6-degree-of-freedom decoupling of the suspension system as the objective function. The optimization design of vibration damping for the suspension system of the engine was completed by means of ant colony algorithm. Simulation analysis was carried out on the dynamic response of the optimized model in terms of time domain and frequency domain, and relevant data and curves were obtained. The simulation results show that the optimized model can control the highest order inherent frequency of the suspension system within the range required by the design. Meanwhile, the transmissibility of the optimized suspension system was calculated, which concludes that the optimized transmissibility is reduced. The comparison on results before and after the optimization indicates that the optimized suspension system has achieved the objective of realizing optimization design of vibration damping.

II. MODEL BUILDING AND VERIFICATION

A. Building the Model of the Power Assembly System

A three-dimensional model for the power assembly suspension system was built via ADAMS/View. The power assembly system was simplified on the basis of the dynamic vibration model of the power assembly suspension system. In ADAMS software, the characteristics of the rigid body are only related to its mass, mass center location, inertial matrix and constraint relation etc. Therefore, when building the dynamic model, with some of its original parts and components ignored, the dynamic model of the power assembly suspension system was simplified as shown in Fig.1.

B. Comparison Between Experiment and Simulation

The work of straight six-cylinder four-stroke engine is mainly affected by mass torque, and the excitation torque includes the multiple components of 3rd order of angular velocity such as 3rd order component, 6th order component and 9th order component etc. High order components have a small influence on the vibration process; it mainly affects the automobiles' comfort through noise. Moreover, with the increase of order, the coefficient will decrease significantly, and the influence degree will reduce accordingly. Therefore, the influence of 3rd order components on the entire system was mainly discussed when analyzing straight six-cylinder four-stroke engine [6].

The excitation frequencies of the engine caused by different rotating speeds under idling status as calculated by this theory are as shown in Table 1. The vibration frequencies on the left and right suspensions of the engine obtained from experiment are as shown in Table 2 and Table 3.

Comparison was carried out on the frequencies obtained from theoretical analysis and the measured results in the vibration experiment under different idling speeds to confirm that the two are highly consistent. This shows that the analysis on the vibration excitation of the engine is correct. This proves that the simulation is consistent with reality and the theory and is correct to some extent.

III. OPTIMIZATION AND ANALYSIS ON THE OPTIMIZATION RESULTS

A. Principle of Ant Colony Algorithm

Featured by high flexibility, strong robustness, distributed calculation and self-organizing, ant colony algorithm is applicable to solving the problems in many fields which have higher requirements on dynamic performance and fault tolerance in the nature. In the following paragraph, the model of ant colony algorithm will be introduced by TSP problems in n cities.

When solving the TSP problem of n cities with a swarm of intelligent ants, what's different from general condition is that they will leave some hormones on the path they have passed while moving forward. The number of the hormones left is in direct proportion to the time that they have spent in passing the path. When selecting the next path, they will select according to the concentration of the hormones on the paths to be selected. At t time, the probability that intelligent ant k in city i selects the city j to be arrived is:

TABLE 1. THE EXCITATION FREQUENCIES OF THE ENGINE CAUSED BY DIFFERENT ROTATING SPEEDS UNDER IDLING STATUS AS CALCULATED BY THE THEORY

Rotating speed (rpm)	600	650	700	750	800
Excitation frequency (Hz)	31.20	31.80	33.00	35.60	39.81

TABLE 2. THE MEASURED VIBRATION FREQUENCIES ON THE LEFT SUSPENSION OF THE ENGINE OBTAINED FROM EXPERIMENT

Rotating speed (rpm)	600	650	700	750	800
Excitation frequency (Hz)	30.89	31.38	33.19	34.24	40.12

TABLE 3. THE MEASURED VIBRATION FREQUENCIES ON THE RIGHT SUSPENSION OF THE ENGINE OBTAINED FROM EXPERIMENT

Rotating speed (rpm)	600	650	700	750	800
Excitation frequency (Hz)	30.55	31.04	33.29	34.14	40.00

$$p_{ij}^{k}(t)=\begin{cases}\dfrac{[\tau_{ij}(t)^{\alpha}\eta_{ij}(t)]^{\beta}}{\sum[\tau_{ij}(t)^{\alpha}\eta_{ij}(t)]^{\beta}} & if \quad j\in J_k(i)\\ 0 & if \quad j\notin J_k(i)\end{cases} \tag{1}$$

In Equation (1), $J_k(i)$ indicates the set of all the possible paths selected by intelligent ant k in city i, $\tau_{ij}(t)$ is the number of hormones on the path connecting point i and point j at t time. η_{ij} is the heuristic information transferred from city i to city j; the heuristic information can be provided by specific problem. For instance, make $\eta_{ij}=1/d_{ij}$,d_{ij} indicates the distance between city i and city *j*.α and $\beta$are parameters for adjusting the influence degree of$\tau_{ij}(t)$ and η_{ij} on decision-making.

In addition, evaporative factor ρ is often introduced in ant colony algorithm to eliminate the influence of beginning status. This is because that the beginning status is given randomly, and the hormone track at the beginning is random to a great extent and doesn't have guidance significance.

$$\tau_{ij}(t+1) = (1-\rho)\tau_{ij}(t) + \sum_{i=1}^{m} \tau_{ij}(t) \tag{2}$$

where $\sum_{i=1}^{m} \tau_{ij}(t) = \sum_{k=1}^{m} \Delta\tau_{ij}(t)$ is the number of the total hormones left by *m* ants on the path from *i* to *j* during the visiting process from time *t* to time *(t+n)*.

B. Optimization of the Power Assembly Suspension System

B1. Objective Function

In actual optimization process, the optimization objectives are the 2 main directions where the engine generates vibrations to make the decoupling in the vertical direction (Z axle direction in the coordinate system) and surrounding crankshaft direction (X axle direction) reach a larger value (over 90%). The decoupling of the suspension system in θ_x direction is larger than 90%, while the decoupling rates in Z axle direction and θ_y direction are both less than 80%. On the other hand, the highest order inherent frequency of the suspension system is $1/\sqrt{2}$ times more than the excitation frequency under idling condition (n=600r/min). To enable the system to reach good vibration isolation effect under this idling condition, the 6th order inherent frequency of the suspension system was optimized to make it less than 21Hz. Taking comprehensive consideration, energy decoupling was deemed as the optimization objective in this paper to carry out further optimization on the suspension system. Formula (1) is the general theoretical expression for the energy decoupling objective function of the suspension system:

$$J = \sum_{i=1}^{6} \omega_i (100 - DIP_{ii}(\bar{\mathrm{x}})) \tag{3}$$

In the equation, i is the order of the inherent frequency of the suspension system, ω_i is the weighting factor corresponding to the *i*th order frequency of the suspension system, DIP_{ii} is the percentage of the vibration energy in the dominant direction of the *i*th order inherent frequency of the corresponding suspension system, $\bar{x}$ is the vector variable of the optimization design.

B2. Design Variables

According to the vibration isolation theory of the power assembly suspension system, the dynamic characteristics of the power assembly system are related to the mass parameters of the power assembly as well as the damping, rigidity, installation position and angle factors etc. of the suspension components [7]. In this paper, the optimization design was carried out with the rigidities in the 3 main directions of the suspension components as well as the installation angles of the left and right front suspensions of the engine as the design variables. The variation range of the optimization of the rigidity of the suspension components is 60%, the variation range of the installation angle of the suspension is 30°～60°, and then the expression for calculating the rigidity is:

$$K = \frac{2(p_1 - p_2)}{X_1 + X_1' - X_2 - X_2'} \tag{4}$$

In the equation, p_1 and p_2 are the loading or unloading force at certain time, X_1 and X_1\`as well as $X_2$ and $X_2$\`are the corresponding loading deformation and unloading deformation of the suspension components at p_1 and p_2.

B3. Constraint Conditions

During the optimization process, the value ranges of each order modal frequency were taken as the constraint conditions to make the optimization results basically comply with the frequency distribution of the vibration isolation theory [8-9].

The upper limit of the inherent frequency of the suspension system should be 21Hz. In addition, the inherent frequency in vertical direction should avoid $4 \sim 8Hz$, the frequency range that human body is sensitive to. Through comprehensive consideration, the upper limit of the inherent frequency of the suspension system should be 5Hz [10-11]. The general equation for solving the inherent frequency of the power assembly suspension system is:

$$([K] - \omega^2[M]) = 0 \tag{5}$$

Its solution ω is the value of each inherent frequency, $[K]$ is the rigidity matrix, and $[M]$ is mass matrix [12].

C. Analysis on the Optimization Results

C1. Analysis on the Inherent Characteristics of the Power Assembly Suspension System

Optimization analysis was carried out on the suspension system. The optimized suspension parameters were obtained through multiple calculations and adjustments. The installation angle changed to 35° after optimization from previous 45°, the rigidities of the 6 suspensions in 3 directions are as shown in Table 4.

The 6-order intrinsic modes of the power assembly suspension system and their decoupling rates after optimization are as shown in Table 5. Compared with the data before optimization, the minimum frequency of the optimized suspension system is 5.18Hz which is larger than 5Hz, the lower limit of the inherent frequency. The inherent frequency of the highest order of the suspension system reduces to 18.31Hz.

Judging from the decoupling of each modality of the suppression system, the decoupling of the optimized suspension system in Z axle direction and θ_y direction reach 91.56% and 91.23% respectively, which increase by 25.61% and 34.38% respectively compared with that before optimization. The decoupling rates in other directions all reach 90%. Therefore, judging from the overall condition, the suspension system has basically realized decoupling and achieved the objective of optimizing decoupling.

C2. Simulation and Analysis on the Dynamic Response of the Optimized Suspension System

In this section, simulation calculation was carried out on the optimized power assembly system in terms of time domain. In time domain simulation, analysis was carried out on the performance of the suspension system by means of ADAMS/View module.

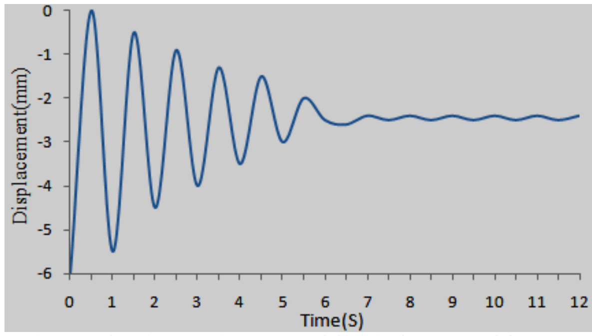

Fig. (2). The Time-varying Curve of the Displacement of the Mass Center of the Optimized Power Assembly along Z Axle Direction Under Idling Condition

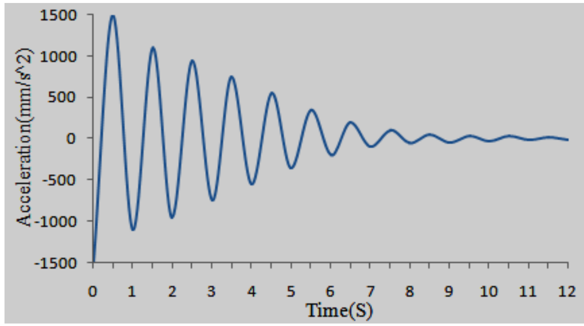

Fig. (3). The Time-varying Curve of the Acceleration of the Mass centre of the Optimized Power Assembly along Z Axle Direction Under Idling Condition

TABLE 4. THE RIGIDITIES OF THE OPTIMIZED SUSPENSIONS IN EACH DIRECTION (N/MM)

Direction	X axle	Y axle	Z axle
Front suspension of the engine (left)	150.88	237.42	938.81
Front suspension of the engine (right)	185.32	229.15	892.13
Rear suspension of the engine (left)	256.05	335.56	881..45
Rear suspension of the engine (right)	315.63	329.39	879.85
Gear box suspension (left)	179.44	243.91	476.15
Gear box suspension (right)	149.76	223.16	501.01

TABLE 5. THE 6-ORDER INTRINSIC MODES OF THE POWER ASSEMBLY SUSPENSION SYSTEM AND THEIR DECOUPLING RATES AFTER OPTIMIZATION

6-degree-of-freedom decoupling	Fore/Aft	Lateral	Bounce	Roll	Pitch	Yaw
Frequency (Hz)	5.18	5.74	7.61	18.31	11.85	9.94
Fore/Aft	95.45	0.05	0.01	0.01	1.78	0.12
Lateral	0.05	91.03	0.42	0.51	0.15	0.24
Bounce	0.01	0.22	89.76	0.34	1.68	2.45
Roll	0.02	1.01	0.28	95.15	0.41	0.65
Pitch	1.78	0.02	1.46	0.68	91.86	0.17
Yaw	0.18	1.51	7.85	0.89	0.13	91.23

Fig.2 Fig.3 are the time-varying curves of the optimized suspension system under the idling condition of 750r/min. According to the contrastive analysis on it and the un-optimized suspension system, the maximum displacement of the mass centre of the un-optimized power assembly in Z axle direction is -7.894mm, and the maximum displacement after optimization is -6.0mm; the RMS value of the accelerated velocity of the mass centre of the un-optimized power assembly is 1651.6434 mm/sec2, and the RMS value after optimization is 1495.06 mm/sec2. According to the aforesaid data, the displacement and the accelerated velocity at the mass centre of the optimized suspension system decrease to some extent, they are still in the allowed design range, which proves that the overall vibration isolation performance of the system is improved to some extent.

IV. CONCLUSION

(1) Modelling and simulation calculation are carried out on the suspension system of a MPV (Multi-Purpose Vehicle) via ADAMS software. Multiple simulation calculations are carried out by revising the model parameters repeatedly, and finally a model consistent with the data of idling vibration experiment is obtained.

(2) A mathematical model for the optimization design of vibration damping for the power suspension system of MPV is built, with the inherent frequency constraints of the suspension system, the vibration displacements and the turning angle constraints of the power assembly system along X axle and Y axle as the constraint conditions, the rigidities of the suspension components in the 3 main directions as well as the installation angles of the left and right front suspensions of the engine as the design variables, the 6-degree-of-freedom decoupling of the suspension system as the objective function.

(3) The optimization design of vibration damping for the suspension system of the engine is completed by means of ant colony algorithm. Simulation analysis is carried out on the dynamic response of the optimized model in terms of time domain. Comparison on it with the previous scheme indicates the vibration damping effect after optimization.

CONFLICT OF INTEREST

The authors confirm that this article content has no conflicts of interest.

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