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Dynamic Characteristic Simulation of Drum Washing Machine Rigid-Flexible Coupling Model

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Abstract

According to the software ANSYS, Rigid-Flexible coupling model of drum washing machine is built with parameterized characteristics. Based on ADAMS/View and ADAMS/Vibration platform, the simulation of the system in time domain and frequency domain is respectively achieved, and the effect of the key variables parameters (spring stiffness, damping coefficient) of vibration system is studied. Based on ADAMS/Durability module, the flexible body dynamic response is analyzed which obtains structure hot spots and provide a basis for optimal design of body structure.

Keywords:*Drum washing machine, Dynamic characteristic simulation, Rigid-Flexible coupling, Time and frequency domains, Flexible box*

1. Introduction

When drum washing machine is at work, the clothes are constantly lifted and thrown under the lifting reinforcement. Thereby, the clothes can be cleaned using detergent. Drum washing machine inevitably produces vibration and noise because of it's work principle [1]. Vibration has been one of the biggest problems of R & D washing machine. Drum washing machine entered the Chinese market relatively late. So in China, the ability of research and development mostly still remain in the imitation, analogy and experience stage. With the development of computer, virtual prototyping technology of ADAMS has been widely used. This provides advanced ideas and methods for vibration characteristics of drum washing machine [2]. In this paper, simulative study on rigid-flexible coupling parametric model of drum washing machine is made based on ADAMS. We obtain that the key parameter variables, such as spring stiffness and damping coefficients, have effects on dynamic and vibration characteristics. This provides a certain basis for the design and optimization of the drum washing machine.

2.Rigid-felexible Coupling Model

2.1 Box Body Flexible

The box body of drum washing machine is sheet metal parts. Micro deformation will happen in the work. To research this characteristic, this paper disperses the box body to small grids by finite element software ANSYS. After modal calculation, we obtain modal neutral file (MNF) and flexible box [3]. MNF contains the quality attributes and modal characteristics of the object model. Its grids achieve force transmission by sharing a common node. The internal stress and strain of the element can be calculated using

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the material properties. The method adopts modal superposition, so it can simulates element dynamic linear deformation [4].

Before the moda calculation based on finite element, the parameters of model should be predefined, as shown in Table 1. The unit type of the model are uesd Shell63 and Mass21. The Shell63 has bending capacity, and can be subjected to plane and normal load. It also has the stress of rigid and large deformation ability. So it is very suitable for sheet metal box mesh. Mass21 establishes the coupling nodes of the rigid unit and flexible unit. It simulates the transmission connecting force by using the small quality unit. The quality attributes and inertia properties are set to 1e-6.

Table 1. Parameters

Element type		Density (kg/m3)	Elasticity modulus (MPa)	Poissonr atio
Shell63	Mass21	7800	2e5	0.3

Rigid-flexible coupling nodes is the connection point of spring, damper and the box body. They are generated by the division of mass21 unit. These nodes to transfer force are Interface Nodes. They are also the key points in establishing the regid region. Markers will automatically form when importing ADAMS. According to the installation position of spring and damper on the box body, 4 Keypoints are established by coordinate. Then Mass21 element mesh it, as shown in Figure 1. Shell63 element mesh the box body grid. At last, the number of grids is 37389. The number of nodes is 36679.

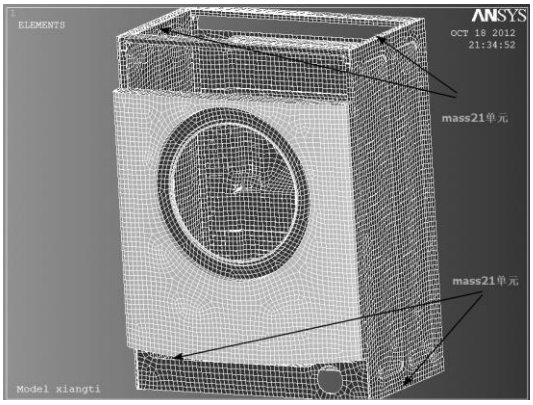


Figure 1. Mesh Box

In modal analysis, dynamic equations of the system can be transformed into an eigenvalue problem:

$$\left|[K]-\omega^2[M]\right|=0 \tag{1}$$

[M] is the total mass matrix of the structure; [K] is the total stiffness matrix.

By solving the equation (1), the matrix eigenvalue (frequency) and characteristic vector (vibration mode) can be solved. Due to the finite element, error becomes larger with the increase of the order. So the natural frequency of the first 6 order as a reference, as shown in Table 2. Rigid-flexible couple in ADAMS environment. The modal order formula derived from modal is:

$$Z=6\times Y+X \tag{2}$$

Z is the modal order of modal neutral file, Y is the number of rigid region node, X is the number of modal extraction.

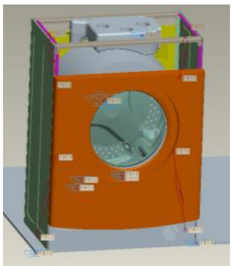
Table 2. The Natural Frequency of Flexible Body

Mode	frequency	Mode	frequency
1	42.297	4	49.664
2	48.583	5	60.620
3	48.936	6	73.769

2.2 Rigid Parametric Model

Through the professional software Mech/pro, virtual proto type model can be parametric. In the Mech/pro environment, we need to set: (1) component as a rigid body; (2) the acceleration of gravity direction; (3) connecting mode and constraints between parts; (4) spring, damping parameters; (5) unit of model of parts and assembly is mm.kg.s

The joint model in the Pro/E environment is as shown in Figure 2 (a). By setting the Interface, CMD file can be exported using ADAMS identification. In the ADAMS environment, the model is shown in Figure 2 (b). By setting the Mech/Pro Regenerate, model features in ADAMS can be refreshed.f gravity direction; (3) connecting mode and constraints between parts; (4) spring, damping parameters; (5) unit of model of parts and assembly is mm.kg.s



(a)

(b)

Figure 2. Models in Pro/E and Adams Environment

2.3 Rigid-flexible Coupling

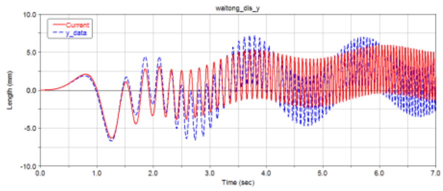
The rigid-flexible coupling mainly refers to the transformation of flexible bodies in the ADAMS environment. The rigid body will replace the previous modal neutral file. In order to ensure the accuracy of structure analysis, we need to set" align flex body CM with CM of current part". Then we can ensure that the center of mass between the two is a coincidence. After the rigid-flexible coupling, the model will be expired constraint. Then further constrain settings need.

3.Dynamic Analysis

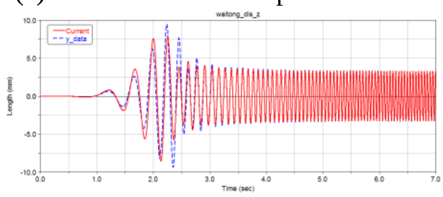
3.1 Time Domain Analysis

According to the washing machine industry standards, the eccentric mass is set as 0.3Kg. In the ADAMS/View environment, the cylinder speed characteristic is set as step (0, 0, 6,7200d). In 6s, the inner cylinder speed changes from 0 to 1200 r/min according to the interpolation function. After the static calculation, simulation runs. The dynamic characteristics of the system can be obtained through the establishment of response Measure [5]. This paper studies the transverse (Y direction) and vertical (Z) characteristics.

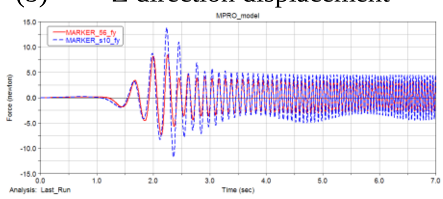
3.1.1 The Parametric Effects of Spring Stiffness: Keeping other parameters unchanged, the spring stiffness coefficient contrast in two cases, k=7.5 N/mm (solid line) and k=10 N/mm (dotted line). Figure 3 (a) and 3 (b) shows the cylinder centroid displacement variation chart. The figures show that when displacement amplitude is steady state, fluctuation amplitude in Y direction becomes bigger if the spring rigidity increases. It is basically unchanged in the Z direction. With the increase of spring stiffness coefficient, the natural frequency of the system becomes bigger and the vibration isolation effect becomes worse. Figure 3 (c) and 3 (d) shows the suspension point force contrast chart. The figures show that the stress increases obviously with the increase of spring stiffness coefficients, especially in the maximum force, namely resonance. The force of suspension point has a direct relationship with the stiffness of the spring. When spring stiffness increases, the stress becomes larger.



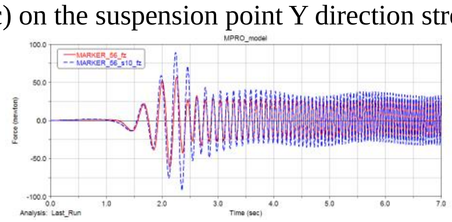
(a) Y direction displacement



(b) Z direction displacement



(c) on the suspension point Y direction stress



(d) on the suspension point Z direction stress

Figure 3. Dynamic Contrast in Different Spring Stiffness

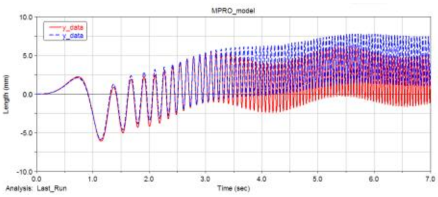
3.1.2 The Parametric Effects of Damping Coefficient

Damper has energy dissipation function in the vibration isolation system. Mechanical energy is converted into heat. Damper can also buffer high speed amplitude. Calculation formula of damping force is:

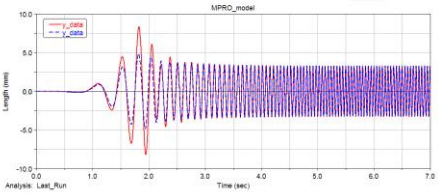
$$F = c \cdot \dot{x} \tag{3}$$

C is damping coefficient, X is vector displacement.

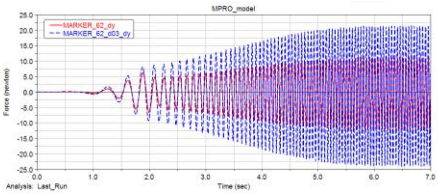
Similar to the methods of suspension spring, damper coefficients contrast in two cases, c=0.15 N/ (mm • s) (solid line) and c=0.25 N/ (mm • s) (dashed line) in time domain. Figure 4 (a) and 4 (b) shows the cylinder centroid displacement variation chart. The amplitude of fluctuation in the Y direction becomes smaller when damper coefficient becomes bigger. It is basically unchanged in the Z direction. This suggests that the effect of vibration isolation in displacement enhances with the increase of damping coefficient. Especially in the resonance of the Z direction, damper reduces the vibration displacement in the resonance. Figure 4 (c) and 4 (d) shows that damping coefficient has an influence on the suspension point force. Damper force increases will cause the suspension point force increasing. The relative value is larger; the box body force is more uneven. If the damping coefficient is too large, the machine on the ground will be unstable. This may even cause the drum washing machine “walking”.



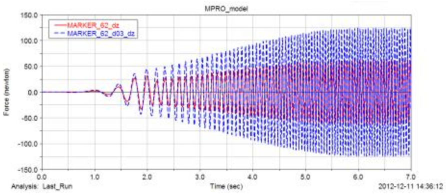
(a) Y direction displacement



(b) Z direction displacement



(c) the lower hanging point Y direction stress



(d) the lower hanging point Z direction stress

Figure 4. Dynamic Contrast in Different Damping Coefficient

3.2 Frequency Domain Analysis

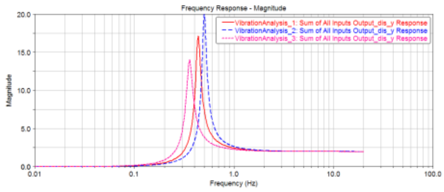
Time domain characteristics can directly reflect the dynamic characteristics of the system. But it doesn’t show the vibration easily [6]. The frequency domain analysis can solve this problem. The ADAMS/Vibration module is the frequency domain analysis tool. It can simulate forced vibration conveniently and intuitively [6]. In the

ADAMS/Vibration environment, through the simulation of on the washing machine vibration system, we can get a variety of vibration characteristics. They are the system natural frequency, eigenvalue calculation, and vibration mode animation display and spectrum response curve.

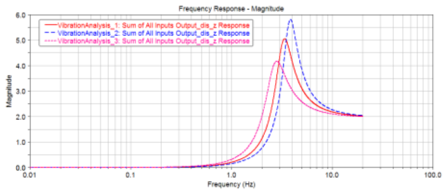
3.2.1 Input and Output Channel: In ADAMS/Vibration, the input channels define the position and direction of the vibration. Analog ADAMS object is the Marker markers. Stimuli motivate system vibration. Analog ADAMS object is the SFORCE. Through the establishment of input channels and incentive, we can simulate the drum washing machine vibration in the conventional load. It is also the source simulation. Input channel and the incentive define as following: eccentric mass is 0.3Kg, eccentricity is 235mm and excitation is in Y, Z direction. Input Marker selects centroid of the inner cylinder. Two input channels are leading and Lagging. Under the two channels and motivation action, we can simulate the vibration of eccentric mass 0.3Kg.

In order to study the forced vibration frequency response with the drum washing machine displacement and box body force, output channel are: two connection point Force of box and the outer cylinder, outer suspension displacement point in Y, Z direction. Set the excitation frequency is 0.01Hz~20Hz and step 2000.

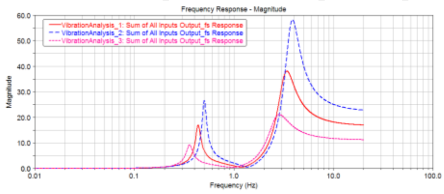
3.2.2 The Parametric Effects of Spring Stiffness: In this paper, we analyze three kinds of spring stiffness in the frequency domain. The results are shown in Fig.5. The spring stiffness are: k=5 N/mm(Analysis_1), k=7.5 N/mm (Analysis_2) and k=10 N/mm (Analysis_3). With the increase of the spring stiffness, natural frequency of the system increases. The displacement and resonant peak stress also increase, as shown in Figure 5 (a) and 5 (b). Vibration characteristic curve shift translationally and magnify with the spring stiffness increasing. If spring stiffness increases, the natural frequency will increase and vibration will magnify.



(a) Y direction displacement response spectrum



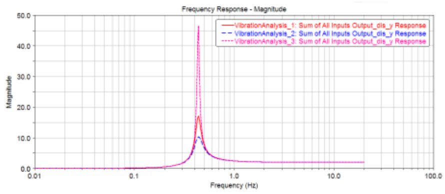
(b) Z direction displacement response spectrum



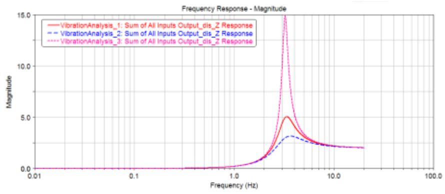
(c) the spring force spectrum response

Figure 5. Comparison of the Response of Different Spring Stiffness

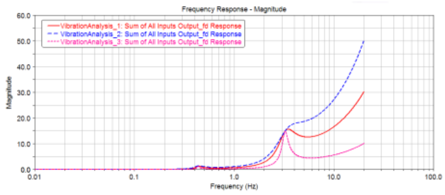
3.2.3 The Parametric Effects of Damping Coefficient: The damper coefficients are c=0.15 N/ (mm • s) (Analysis_1), c=0.25 N/ (mm •s) (Analysis_2) and c=0.05 N/ (mm • s) (Analysis_3). Figure 6 shows the results of comparing them. As we can see, the natural frequency has nothing to do with the damping. But with the increase of the damping coefficient, the amplitude of displacement at the resonance point decreases. Also the damping force increases. Therefore, if the damping coefficient is too small, it will not damp vibration. If the damping coefficient is too large, it will increase the instability of the model on the ground. This is consistent with the characteristics of time domain in Figure 4.



(a) Y direction displacement response spectrum



(b) Z direction displacement response spectrum



(c) the damping force response spectrum

Figure 6. Comparison of Response in Different Damping Coefficient

4. Analysis of Flexible Body Dynamic Response

From Table 2, we can see that natural frequency value of first order is 42.297Hz. It is relatively close to the highest frequency of 20Hz. With the demand of the highest dehydration speed increasing [7], it means that the excitation frequency will be more close to the resonance frequency. So, the next design needs to further improve the stiffness through some approaches.

In the post-processing module, transient response of flexible body running is shown in Figure 7. With the gradual increase of speed, the eccentric force increases. The spring force and damping force also increase. The flexible body stress characteristics can be studied in the instantaneous dynamic response

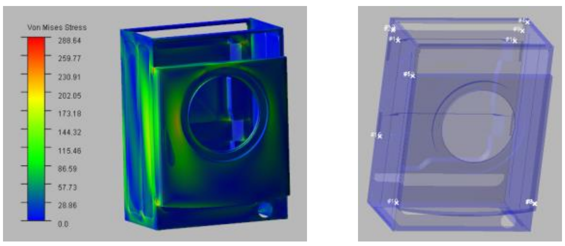


Figure 7. The Transient Response and Hot Spots Position

Application of Durability module in ADAMS, we can easily get the position of the system’s hot spots during running. In Figure 7, we select the first top 15 hot spots. These positions of hot spots are the maximum stress of whole operation condition. It can be seen that the maximum stress generally locates in the end node among the parts. In actual production, the node connectivity has key influence on the whole structure.

5. Conclusion

In this paper, we model the virtual prototype of drum washing machine. We study the spring stiffness and damping coefficient's influence on system vibration in time and frequency domain. The conclusions are as follows:

(1) Through the shell63 and mass21 unit, we realize the box body flexibility. MNF file is derived from modal calculation. Box hanging point is transferred outside Interface Node. It is also the key point of the establishment of rigid coupling region.

(2)With the increase of the spring stiffness coefficient, resonance vibration increases. Natural frequency increases, so the effect of vibration isolation of variation. The damping coefficient increases, so the amplitude of displacement resonance point decreases. The damping force increases, and so natural frequency invariant.

(3) The lowest resonance frequency of box body is 42.297Hz. We calculate the first 15 hot spots dynamically. The position of the connectivity has a key effect on the stability of the structure of box.

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