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Dynamic Balance Optimisation for the Middle Shaft System of a Cotton Comber Based on ADAMS

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Abstract

For the vibration problem of a cotton comber, a model of the middle shaft system of a headstock was built using SOLIDWORKS software. It was found that the vibration was mainly caused by the shifting of the centroid. Then a dynamic optimisation model was established according to the dynamic balancing theory using the revolving method. The installment of the middle shaft system was optimised using ADAMS software with the aim of minimising its counterforce. The optimal value of the flange plate and timing regulation plan are given correspondingly. The results show that the noise and vibration of the comber CM500 are decreased significantly.

Key words: cotton comber, dynamic balance design, parameterization design, ADAMS.

Introduction

The cotton comber is one of the most important types of production equipment in cotton spinning. Short fibres, neps and impurities can be removed and cotton fibers can be straightened and paralleled by using a comber [1]. Therefore research on the comber has been present in recent years, and many interesting results have been established [2 - 6]. Especially it is shown that the middle shaft system of the headstock is the main component producing noise and vibration because of the uneven distribution of quality due to uneven material quality, blank defects, and errors in installation [2]. Therefore it is important to optimise the dynamic balance of the middle shaft system of the headstock in order to improve the overall performance of the cotton comber.

Balance is a technological process of rotating machinery in manufacturing, commissioning and maintenance, in which the inertial axis of the rotation system approaches the axis of rotation by changing the mass distribution of the rotor [7, 8]. Motivated by all these research works, this paper attempted to optimise the dynamic balance of the comber by using computer-aided tools - SolidWorks and ADAMS. First a model of the middle shaft system of a headstock was built using SolidWorks software. Then a dynamic optimisation model was established according to dynamic balancing theory using the revolving method, and the installment of the middle shaft system was optimized using ADAMS software. The result shows the effectiveness of the method.

Model built using 3-D software: SolidWorks

The intermediate shaft system is mainly composed of some disc-type parts, such as the crankshaft, flange 138 tooth wheel and timing adjustment plate. A model of the quality attribute of the component can be built using 3-D modelling software: SolidWorks, which is an effective computer-aided tool for 3-D modelling [9]. It is found that the intermediate shaft system has a certain degree of eccentricity, and a centrifugal force will be produced during work, which is the major factor that causes the vibration and noise of the comber. From the composition of the intermediate shaft system and analysis results, we can conclude that the intermediate shaft system is dynamically imbalanced and needs to be optimised.

Dynamic balance optimisation

When rigid rotary parts rotate around a fixed axis, and if the mass distribution of each component is in the same plane, a centrifugal force is produced due to the inertia-force, which constitutes a force system intersected in the rotation center. If the resultant force of the force system is not equal to zero, this is called static

imbalance. If the mass distribution of each component is in a different plane, the force system composed of the centrifugal force is not intersected in the same plane, but there is a space force system instead. In this case, if the resultant force or moment of couple of the force system is not equal to zero, this is called dynamic imbalance. From the composition of the intermediate shaft system and analysis results, we can conclude that the intermediate shaft system is dynamically imbalanced.

The model of the middle shaft system of a cotton comber is shown in **Figure 1**, which mainly contains a crankshaft, flange 138, tooth wheels and timing adjustment plane.

Definition of variables

According to mechanical dynamic balancing theory, firstly planes are made in the flange 138, tooth wheels and timing adjustment plate, respectively. Then a dynamic model is established according to the changing mass center positions in the plane. The dynamic balance model is shown in **Figure 2** (see page 44). The distance between the plane of the tooth wheel plate and flange plate is 31.5 mm, and that between the flange and timing adjustment plane - 84.0 mm.

By using SolidWorks 3D software, the centroid of the mass and deviation position of the center rotation of the tooth wheel, flange and timing adjustment plane can be found, respectively. The mass center plane is shown in **Figure 3** (see page 44). It is easy to see that the angle between their mass centers and the positive Z axis is, respectively:

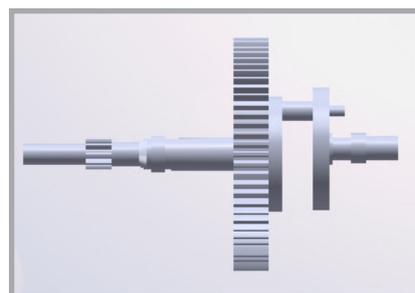


Figure 1. Model of a middle shaft system.

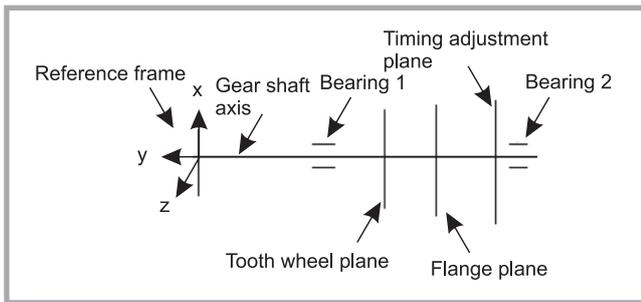


Figure 2. Dynamic balance model.

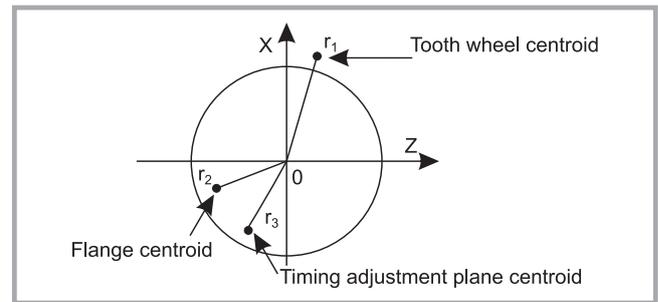


Figure 3. Position of centroid plane.

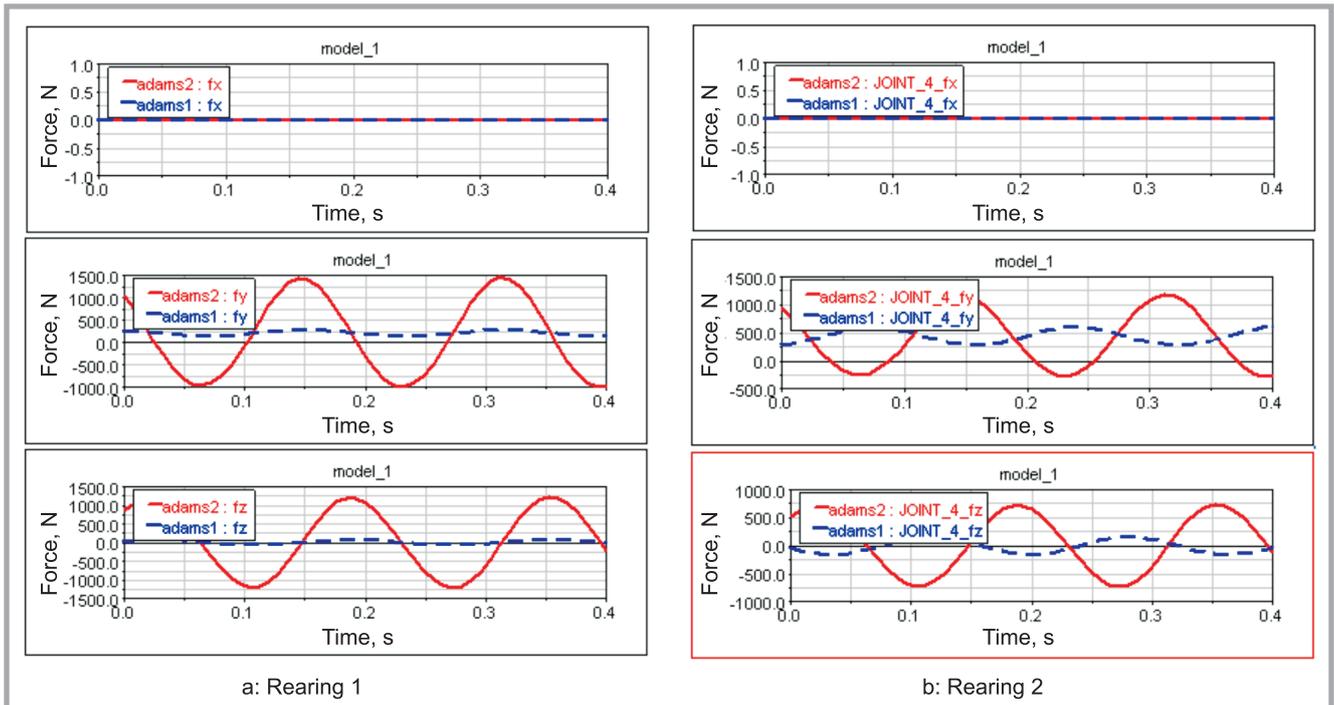


Figure 4. Counterforce curve.

$$\theta_1 = 82.9^\circ \quad \theta_2 = -164.3^\circ \quad \theta_3 = -105.6^\circ \quad (1)$$

The distance between their mass centers and rotation center O is, respectively:

$$r_1 = 16.2 \quad r_2 = 9.1 \quad r_3 = 9.6 \quad (2)$$

The moment of inertia around the Y axis of the moment of inertia can be calculated as:

$$I_1 = 0.408 \quad I_2 = 0.204 \quad I_3 = 0.203 \quad (2)$$

Based on the analysis above, we choose the angle between the centroid of the flange and the timing adjustment positive

Z axis θ_2, θ_3 as the design variables. It is easy to see that the ranges θ_2, θ_3 are

$$-180^\circ \leq \theta_2 \leq 180^\circ, \quad -180^\circ \leq \theta_3 \leq 180^\circ \quad (4)$$

Establishment of the objective function and results analysis

imbalance of the rigid cursor is the main source of vibration of rotating machinery, which will bring deflection and internal stress, causing vibration correspondingly. During the balancing optimisation, we should minimise the force on the bearings, i.e. minimise the reaction force on

bearings. Therefore we choose the counterforce on the two bearings as the objective function in this paper.

Importing the 3-D model shown in *Figure 1* into ADAMS software [10] and adding the constraints of each component according to the actual work, the curves of the counterforce on the two bearings can be obtained at a speed of 500 r.p.m., which is shown in *Figure 4*. Correspondingly we can get the optimal values of θ_2, θ_3 as follows:

$$\theta_2 = -159.7^\circ \quad \theta_3 = -101.4^\circ \quad (5)$$

In *Figure 4*, the red real line is the initial counterforce on the corresponding bearing before optimisation; the blue dotted line is the counterforce after the corresponding optimisation.

The counterforce value for bearings 1 and 2 before and after optimisation are

Table 1. Counterforce value before and after optimisation.

Optimisation			Counterforce, N		
			Fx	Fy	Fz
Bearing 1	Maximum amplitude	before optimisation	0	1457.05	1229.19
		after optimisation	0	274.04	72.48
Bearing 2	Maximum amplitude	before optimisation	0	1191.41	736.96
		after optimisation	0	604.73	160.85

shown in **Table 1**. From **Table 1**, we can see that the counterforce on bearings can be reduced significantly after optimisation, which indicates that the method presented in this paper is effective.

■ Conclusion

In this paper, the dynamic balance optimisation for the middle shaft system of the cotton comber CM500 is mainly analysed using ADAMS. The optimal centroid of the flange and timing adjustment plate are presented. The results show that the counterforce is reduced by 81.2% and 49.3% in the Y-direction, and 94.1% and 78.3% in the Z-direction for bearings 1 and 2, respectively, which indicates that the method presented in this paper is effective.

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