Analysis of the Influence of Machining Errors on the Dynamic Characteristics of the Dobby Modulator

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Received 26 February 2021; accepted for publication 21 June 2021

Abstract

In this paper, cam profile dynamic modeling and analysis of the cam slider modulator are presented. The nonlinear dynamic model is derived for the modulator equation with consideration of the flexible effect. The fourth order Runge-Kutta method is employed to solve the problem. The lumped parameter and Monte Carlo simulation methods are also utilized. Modulator roller vibration with regard to the cam profile machining errors are analyzed. In order to analyze the vibrational response at different speeds, VB.net simulation is performed. The results demonstrate that the disturbed cam profile vibrates smoothly and randomly when the dobby speed is less than 400 r/min. Furthermore, it is observed that the operating dobby speed should not exceed 900 r/min. For disturbed cam profile system, when the speed of the dobby is less than 400 r/min, small follower vibration response is observed. When dobby speed is greater than 200 r/min, significant impact on the follower vibration response is observed.

Key Words: Cam profile, Modulator, Dynamic model, Vibration response, Machining errors

1. Introduction

In textile machinery, the rotary dobby is currently one of the most advanced high-speed opening devices used in modern high-speed weaving machines [1-3]. The dobby mechanism divides the warp yarn into two layers, and forms a channel for the weft yarn to pass through. This process is achieved by converting the rotational main shaft motion of the loom into vertical motion of the heddle frame. Dobby must have good mechanical properties to adapt to the high-speed loom production. In recent years, a rotary dobby can be found in shuttleless looms such as rapier and air-jet looms. It is widely employed due to their compact structure, high motion accuracy, relatively simple control, and stable operation.

The modulator is the core part of the rotary dobby. During the weaving process, the heddle frame is lifted [4]. Hence, force and vibrational characteristics of the modulator have an important influence on the motion process of the heddle frame, which drives the warp yarn. Therefore, it is necessary to analyze the dynamic characteristics of the modulator, particularly the high-precision and high-speed characteristics of the conjugate cam-roller assembly [5]. Motion accuracy and dynamic characteristics influencing factors are relatively complicated. Therefore, dynamics and reliability of the modulator have to be investigated in detail.

Cam followers represent an important impact system type, which is widely used in various applications. Generally, a cam rotating with a constant speed provides the follower drive force. One of the most common examples is the valve mechanism of an internal combustion engine. In this mechanism, the cam rotation moves the engine valve through the follower, and the spring ensures the restoring force required to maintain contact between the components [6]. In high-speed cams, the inevitable camshaft fluctuations affect the accuracy of the follower [7, 8]. Rothbart [9] employed a shift cam and explicitly embedded its shifting speed as a design parameter. The vibrational response graph can be used to evaluate the influence of various parameters on the cam follower dynamics. These parameters include the cam speed, stiffness characteristics, and various cam profiles [10].

Complex behavior of impact mechanical equipment has become an important subject of the ongoing research [11]. As previously mentioned, one of the most common examples is an internal combustion engine valve mechanism, where contact loss between the cam and the follower occurs during high-speed rotation [12, 13]. This causes a reduction in the engine efficiency, fuel consumption, and emission performance [14]. Gatti [15] proposed a preliminary study on the follower vibration control when it is directly acted upon by the cam.
According to the above presented research, a certain foundation has already been laid for further analysis of the rotary dobby dynamics. In order to improve the production efficiency of modern looms, their speed is also continuously increased. Consequently, the machine vibration and stability are decreased when subjected to higher speeds, which shortens the service life of the loom. Therefore, in this paper, the principle of lumped parameter method and fourth-order Runge-Kutta method (among other methods), combined with the system dynamics theory, are employed to construct the modulator dynamic model. Furthermore, the numerical solution method is established, and dynamics calculation procedures are compiled. Modulator dynamic response analysis for the cam profile processing errors are conducted. Moreover, main influencing factors affecting the reliability and stability of the modulator are compared and analyzed. Lastly, theoretical basis and guidance for improving the efficiency and product quality of the rotary dobby are provided [16].

2. Dynamic modeling and analysis of dobby modulator

2.1 The principle of a rotary dobby with the cam-slider modulator

Schematic drawing of the cam-slider modulator is presented in Fig. 1. Gear (1) rotates clockwise with gear rotation speed \( \omega \). The conjugate cams (7 and 13) are fixed on the dobby box body. The cam swing arms (4 and 9) are hinged on the gear (1), and can rotate around its hinged joints, thus transmitting the uniform circular motion of the loom to the slider frame. The uniform circular motion of the loom, which is transmitted to the slider frame through the gear (1), the slider of the modulator is hinged at point \( C \), forms the dobby rotational motion of the main shaft \( O \).

![Fig. 1 Cam-slider modulator operating principle.](image)

2.2 Dynamic modeling of the cam-slider modulator

Dobby modulator movement is transmitted to the heddle frame through the link. Dynamic performance of the modulator directly affects the shedding performance of the loom. Currently, for kinematics and dynamics analysis of the rotary dobby, elastic deformation effect of the mechanism components on the heddle frame motion performance is not considered. With the development of dobby for lightweight and high-speed applications, the inertial forces and system flexibility have sharply increased. Therefore, the elastic deformation of components has an increased impact on the overall system performance. The flexible body in the system has an important effect on the motion of the entire system. During modulator operation, its vibration response has a significant impact on the overall dynamics of the loom system. Therefore, the elastic characteristics of the modulator must be considered.

![Fig. 2 Dynamic models of the modulator.](image)
when investigating overall dynamic characteristics of the dobby. By studying the influence of the modulator components, flexible deformation effect on the opening performance is investigated. Moreover, theoretical basis for solving the problems of low operational stability and poor reliability is established. In mechanical systems, flexible components have a significant impact on the movement of the entire system. To accurately simulate motion of the entire mechanical system, the influence of the flexible body on the system motion characteristics has to be considered and its vibrational characteristics analyzed.

Dynamic model of the cam slider modulator is shown in Fig. 2 (a). The lumped parameter method is used to simplify the mechanism into two masses, three dampers, and three springs. In the dynamic model of the cam slider modulator, $m_1$ represents the combined concentrated mass of cam roller 1, swing arm 1, link 1, and half of the main shaft link. Parameter $m_2$ represents the combined concentrated mass of cam roller 2 and swing arm 2. Additional springs and damping elements are simplified as spring coefficients $k_1$, $k_2$ and $k_s$. The viscous damping coefficients $b_1$, $b_2$, and $b_3$ are used to tie masses $m_1$ and $m_2$ to the contact point. The aforementioned is depicted in Fig. 2 (b).

Separated bodies of the cam slider modulator followers 1 and 2 are considered for the force analysis. Followers 1 and 2 are simultaneously subjected to the elastic restoring force and the damping force in the vertical direction. The force analysis is shown in Fig. 2 (c).

According to Newton’s second law and under assumption of negligible friction resistance between the cam and the roller, modulator differential motion equation can be derived:

$$
\begin{align*}
\{ k_1[S(t) - y_1] + b_1[\dot{S}(t) - \dot{y}_1] - k_2(y_1 - y_2) - b_2(\dot{y}_1 - \dot{y}_2) &= m_1\ddot{y}_1 \\
 k_3(y_1 - y_2) + b_3(\dot{y}_1 - \dot{y}_2) - k_4[y_2 - H(t)] - b_4[\ddot{y}_2 - \dot{H}(t)] &= m_2\ddot{y}_2 
\end{align*}
$$

(1)

$$
S(t) = S_i(t) + \Delta S(t) \tag{2}
$$

$$
H(t) = H_i(t) + \Delta H(t) \tag{3}
$$

$$
\dot{t} = \frac{\dot{\theta}}{\omega} \tag{4}
$$

where:

$S(t)$ - Actual displacement of the contact point A between the main cam and the roller.

$S_i(t)$ - Ideal displacement of the contact point A between the main cam and the roller.

$\Delta S(t)$ - Displacement variation of the contact point A between the main cam and the roller.

$H(t)$ - Actual displacement of the contact point B between the auxiliary cam and the roller.

$H_i(t)$ - Ideal displacement of the contact point B between the auxiliary cam and the roller.

$\Delta H(t)$ - Displacement variation of the contact point B between the auxiliary cam and the roller.

$y_1$ - Actual movement displacement of the mass $m_1$ of the follower.

$y_2$ - Actual movement displacement of the mass $m_2$ of the follower.

$k_1$, $k_2$, and $k_s$ - Spring stiffness.

$\theta$ - Gear rotation angle, $\theta \in [0^\circ, 360^\circ]$.

$\omega$ - Gear running speed.

$t$ - Gear running time.

In an ideal situation, for a relatively low rotational speed of the gear, the contact point between the cam surface and the roller never separates. We accurately study the static dynamic behavior of the modulator through theory, and the displacement characteristic curve $S_i(t)$ calculated by the contact between the cam and the roller under the ideal fit condition. The $S_i(t)$ displacement curve is shown in Fig. 3. Follower mass $m_1$ meets the expected displacement characteristic curve $S_i(t)$, while follower mass $m_2$ meets the expected displacement characteristic curve $H_i(t)$. Assuming there is no assembly errors and the two rollers are in close contact with the conjugate cam profile: $S_i(t) = H_i(t)$, $\Delta S(t) = \Delta H(t) = 0$.

![Fig. 3 $S_i(t)$ displacement curve.](image)

### 2.3 Dynamic analysis of the cam-slider modulator

Mass $m_1$ is considered as an example. The mass is placed in an ideal state. Displacement motion equation of $m_1$ in the modulator is then:

$$
y = F(U, q_1, \ldots, q_n) \tag{5}
$$

where:

$y$ - Roller follower displacement of ideal cam.

$U$ - Displacement of ideal cam drive.

$q_1, \ldots, q_n$ - Theoretical parameters of each component in the rotating speed change mechanism.

In the actual state, due to various modulator errors and parameter changes, the movement displacement $U$ of the driving part cam should be increased by a certain amount $U + \Delta U$. The actual parameter $q$ of each component can be represented by $q_i + \Delta q_i$. In the actual state, the modulator displacement motion equation of $m_1$ is as follows:

$$
y = F(U + \Delta U, q_1 + \Delta q_1, q_2 + \Delta q_2, \ldots, q_n + \Delta q_n) \tag{6}
$$

This formula can be expanded according to the Taylor series expansion. Because the change items $\Delta U$ and $\Delta q_i$ are both relatively
minor, derivation of both second order and high-order terms can be neglected. Only the first-order terms in the series containing $\Delta U$ and $\Delta q_i$ are retained. The aforementioned formula can be approximately expressed as:

$$y_1 = F(U, q_1, q_2, \ldots, q_n) + \sum_{i=1}^{n} \left( \frac{\partial y}{\partial q_i} \right) \Delta q_i$$

(7)

$$= y + \sum_{i=1}^{n} \left( \frac{\partial y}{\partial q_i} \right) \Delta q_i$$

(8)

Follower displacement errors of the mechanism is equal to $\Delta y_1 = y_1 - y$. According to Eqs. (6) and (7):

$$\Delta y_1 = y_1 - y = \left( \frac{\partial y_1}{\partial U} \right) \Delta U + \sum_{i=1}^{n} \left( \frac{\partial y_1}{\partial q_i} \right) \Delta q_i$$

(9)

Assuming negligible effect of the active modulator part displacement errors, when $\Delta U = 0$, the Eq. (8) can be simplified as:

$$\Delta y_1 = \sum_{i=1}^{n} \left( \frac{\partial y_1}{\partial q_i} \right) \Delta q_i$$

In the above equations, the partial derivatives $\frac{\partial y}{\partial q_i}$ and $\frac{\partial y}{\partial q_i}$ represent displacement errors functions. These functions are taken as the partial derivatives of ideal values $U$ and $q_i$, respectively. When the original errors are known, in order to obtain the value of the displacement errors, it is necessary to solve the partial derivative $\frac{\partial y_1}{\partial q_i}$.

3. Cam profiles machining errors

3.1 Random machining errors of cam profiles

The modulator requires a gapless contact between the cam and the roller. This demands higher requirements for the conjugate cam mechanisms processing. Due to its structural characteristics, the factors involved in the conjugate cam mechanism machining process are relatively complex. Therefore, machining errors influencing factors are also very abundant, resulting in an inevitable cam profile machining errors. Therefore, in this section, modulator dynamic response analysis of the cam profile machining errors are considered.

First, influence of random machining errors on the follower vibration characteristics is analyzed. Then, interval and magnitude of random machining errors are set according to Table 1.

| Serial number | Error interval / ℃ | Error size / (mm) |
|---------------|--------------------|------------------|
| 1             | (43.2 – 45]        | 0.01 * $S_i(t)$  |
| 2             | (45 – 50.4]        | -0.01 * $S_i(t)$ |
| 3             | (129.6 – 135]      | 0.01 * $S_i(t)$  |
| 4             | (135 – 140.4]      | -0.01 * $S_i(t)$ |

Since the introduction of simulated random machining errors causes deviations in the cam profile shape, actual position of the follower is also modified when cam profile disturbance is introduced.

3.2 System machining errors of cam profiles

In addition, based on the Monte Carlo method, dynamic response analysis of the cam profile system errors modulator is carried out. Moreover, the cam profile change model caused by the system machining errors is established, as shown in Fig. 4.

Set of random numbers that satisfy a normal distribution is used to simulate the systematic machining errors of the cam profile. In the normal direction of the cam, a systematic machining errors with a tolerance zone of -0.01 mm to +0.01 mm is generated. Parameter $\Delta \theta$ is equal to 18°, while 20 groups of systematic machining errors conforming to normal distribution are continuously generated on the entire cam profile.

Due to the introduction of the simulated system processing errors that cause deviations in the shape of the cam profile, the actual position of the follower also changes with the introduced cam profile disturbance. The cam profile coordinates can be expressed as a function of the gear rotation angle $\theta$. The coordinates can also be expressed via follower theoretical displacement curve $S_i(t)$. Following the introduction of the simulated system machining errors disturbance $\Delta S_i(t)$ into the cam profile, the actual displacement of the follower compared with its theoretical displacement is calculated during the 20 gear cycles.

4. Vibrational response and numerical solution of the modulator

4.1 Dynamic parameters of the modulator

In Table 2, modulator dynamic parameter values are listed. Parameter values are comprehensively considered using experience, experimental conclusion, and digital prototype simulation methods.
### 4.2 Numerical solution based on the fourth-order Runge-Kutta method

In order to obtain vibrational characteristics the modulator system of masses \( m_1 \) and \( m_2 \), the fourth-order Runge-Kutta method based on the nonlinear vibration theory is employed to numerically solve the system. According to Eqs. (1)-(3), the actual motion characteristic curves \( y_1 \) and \( y_2 \) of mass blocks \( m_1 \) and \( m_2 \) are obtained. Furthermore, dynamic characteristics of the mass block and the follower system are investigated. In Fig. 5, the schematic diagram of the calculation process is shown.

The fourth-order Runge-Kutta method is based on the following equation:

\[
y_{i+1} = y_i + \left( \frac{h}{6} f_1 + \frac{1}{2} f_2 + \frac{1}{2} f_3 + f_4 \right)
\]

If value of \( y = y_i \) at \( \theta \) is known, the value of \( y = y_{i+1} \) at \( \theta_{i+1} \) can be determined. Then:

\[
h = \theta_{i+1} - \theta_i
\]

Taylor series expansion of Eqs. (1)-(9) is carried out:

\[
y_{i+1} = y_i + \frac{\partial^2 y}{\partial \theta^2}_{\theta_i} h^2 + \frac{1}{2} \frac{\partial^2 y}{\partial \theta^2}_{\theta_i} h^2 + \frac{1}{3!} \frac{\partial^3 y}{\partial \theta^3}_{\theta_i} h^3 + \frac{1}{4!} \frac{\partial^4 y}{\partial \theta^4}_{\theta_i} h^4
\]

Given that \( \frac{\partial y}{\partial \theta} = f(\theta, y) \):

\[
y_{i+1} = y_i + f(\theta_i, y_i) h + \frac{1}{2} f'(\theta_i, y_i) h^2 + \frac{1}{3} f''(\theta_i, y_i) h^3
\]

According to Eqs. (12) and (13), a common solution is used:

\[
y_{i+1} = y_i + \frac{1}{6} (k_1 + 2k_2 + 2k_3) h
\]

where:

\[
k_1 = f(\theta_i, y_i)
\]

\[
k_2 = f \left( \theta_i + \frac{1}{2} h, y_i + \frac{1}{2} k_1 h \right)
\]

\[
k_3 = f \left( \theta_i + \frac{1}{2} h, y_i + \frac{1}{2} k_2 h \right)
\]

### 5. Comparison of vibration results

Dooby speed used in this paper is approximately 450 r/min, which represents a commonly employed speed in practical applications. The following vibrational response diagram presented

![Fig. 5 Schematic diagram of the calculation process.](image1)

![Fig. 6 Comparison of \( y_1 \) and \( y_2 \) displacements with \( \omega = 450 \) r/min.](image2)
in Fig. 6 uses the gear speed as the research speed, with the ratio of the gear speed to the speed of the dobby being 1:2. Higher dobby speeds are not conducive to dynamic behavior study of the modulator system. Therefore, in this paper, dynamic analysis for dobby speeds less than 900 r/min is conducted. The input gear rotation speed $\omega$ of the gear is kept between 50 r/min and 450 r/min, with 50 r/min intervals. The actual displacement characteristic curves $y_1$ and $y_2$ of mass followers $m_1$ and $m_2$ are obtained, respectively. Due to space limitations, only displacement deviation of $y_1$ and $y_2$ for the gear speed of 450 r/min is shown in Fig. 6.

According to the above presented research results, basically consistent result curves obtained by analyzing the follower mass blocks $m_1$ and $m_2$ are observed. Therefore, when studying modulator dynamic response, analysis and calculation of the follower mass $m_1$ (hereinafter referred to as the "follower") are shown.

5.1 Vibrational response of the random machining errors

In order to understand the vibrational response of the cam slider follower, the corresponding vibration response diagram is derived for different values of $\omega$, as shown in Fig. 7.

The vibration amplitude is the vibration range between the expected displacement $S_1(t)$ of the follower and the actual displacement $y_1$. As an example, 1000 gear cycles are investigated in this paper. Following stable modulator operation, the follower vibration response diagram during 20 gear cycles is calculated.

According to the ideal vibration analysis of the cam profile in Fig. 7, the vibration amplitude of the cam profile for $\omega \leq 200$ r/min is decreased, while the impact on the follower vibration results in the most stable case. In the range of $200 < \omega \leq 450$ r/min, the desired profile (DP) vibration amplitude is increased, which has a greater impact on the vibration produced by the follower.

According to the cam profile random machining errors analysis in Fig. 7, the cam profile vibration of the disturbed profile random (DPR) has a great influence on the dynamic behavior of the follower of modulator. For the gear rotation speed lower than $\omega \leq 200$ r/min, which the vibration amplitude of the DPR has a great influence on the follower. In the range of $200 < \omega \leq 300$ r/min, DPR produces obvious vibration. Lastly, in the gear rotation speed

![Fig. 7 Follower vibrational response in 20 cycles.](image)

![Fig. 8 Vibration of the follower in 20 cycles (Correspond to Fig. 7).](image)
range of $300 < \omega \leq 450 \text{ r/min}$, DPR produces greater vibration and fluctuation.

According to the presented data, the lifting motion of the modulator, the impact damage of the mechanism cannot be caused by excessive vibration. For this type of mechanism, optimal gear running speed is $\omega \leq 200 \text{ r/min}$. It should not, however, exceed 450 r/min. The above analysis is consistent with engineering practice, which can verify that the established dynamic model has reference and guiding significance for engineering applications.

Due to space limitations, Fig. 8 (a) to (d) only show the vibration of the modulator follower for the gear rotation speed of 100 r/min, 200 r/min, 300 r/min, and 450 r/min. Here, vibration of the follower in 20 cycles.

According to data presented in Fig. 8, the DPR has the greatest impact on the dynamic behavior of the modulator follower.

### 5.2 Vibrational response of the system machining errors

Based on the Monte Carlo method, the dynamic response analysis of disturbed cam profile system (DPS) modulator is performed. According to Fig. 9, when the gear is running at low gear rotation speed ($\omega < 200 \text{ r/min}$), the vibrational response of the follower is relatively small under DPS. However, when the gear is running at high rotation speed ($\omega \geq 200 \text{ r/min}$), DPS has a greater impact on the vibration response of the follower. With an increase in the gear rotation speed, the amplitude of the DPS machining errors gradually increases as well.

DPS is compared with the DP. Follower vibration for gear rotation angle of 20 cycles (as shown in Fig. 10) is solved. It can be clearly seen that the system machining errors is greater than the desired profile when the gear operates at low speeds. The vibration of the follower is increased. However, when the gear is running at high speeds, the DPS and the DP will both produce greater follower vibration.

Fig. 10 (a) to (d) show the vibration of the modulator follower for the gear rotation speed of 100 r/min, 200 r/min, 300 r/min, and 450 r/min. Here, vibration of the follower in 20 cycles.

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**Fig. 9** Follower vibrational response in 20 cycles.

**Fig. 10** Vibration of the follower in 20 cycles (Correspond to Fig. 9).
6. Conclusions

Aiming at the dynamic problem of the dobby modulator, its dynamic model is constructed and numerical solution method is established. Starting from some of the key parameters that affect its dynamic performance, a detailed dynamic calculation is carried out for the cam slider modulator. The main factors affecting the vibrational characteristics and dynamic behavior of the modulator are analyzed and studied.

Due to various inevitable modulator errors, the errors size reflects the mechanism dynamic response size, and affects the entire machine efficiency and as well as the product quality. Therefore, modulator dynamic response research method with consideration of DPS is constructed. Furthermore, the Monte Carlo method is used to analyze DPS modulator dynamic response. Moreover, the cam caused by the DPS is established.

Via conducted machining errors analyses, it can be concluded that the DPR and the DPS have a significant impact on the dynamic characteristics of the follower. This, in turn, causes strong vibration, noise, and wear of the moving parts of the system. Consequently, the service life of the system is reduced. Therefore, random and systematic machining errors of the cam pair surface should be controlled to improve the vibration stability and reliability of the system.

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