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Study of the Oscillations of a Composite Carding Drum with Toothed and Needle-Type Garnitures

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Abstract

This article theoretically examines the oscillatory behavior of a composite carding drum positioned in the discretization zone of pneumatic–mechanical spinning machines in order to improve technological efficiency. The study aims to analyze the motion characteristics of the drum and determine how the oscillations of its toothed and needle-type garnitures depend on key structural and operating parameters. Using a theoretical approach, the laws of motion and their solutions were derived through Lagrange's equations of the second kind. The analysis demonstrates the parametric dependence of oscillatory responses in the existing carding drum system, particularly with respect to the toothed and needle-type garnitures, and presents the resulting regularities in graphical form. These findings provide a scientific explanation of the mechanical behavior of the drum within the discretization zone and contribute to a more rigorous understanding of oscillation dynamics in pneumatic–mechanical spinning technology. The study offers a theoretical basis for technological improvement by informing the optimization of carding drum design and operating conditions to enhance system performance and efficiency.

Keywords: Carding Drum; Discretization Zone; Lagrange's Equations; Oscillation Dynamics; Pneumatic–Mechanical Spinning

Introduction

Improving the efficiency of the discretization carding process across the entire width of the fiber sliver, reducing fiber breakage in pneumatic–mechanical spinning machines, as well as improving maintenance conditions due to the serviceability of the carding drum, are important tasks. After a detailed study of various designs of carding drums, a new type of carding drum for pneumatic–mechanical spinning machines was proposed [1]. It consists of a cylinder with through longitudinal prismatic grooves of trapezoidal cross-section, in which identical composite elements are installed. Each composite element includes an outer plate connected to an inner plate by means of elastic rubber pads bonded together with adhesive. Alternating serrated teeth and needles are rigidly attached to the outer plate. Each prismatic composite element has four rows of serrated teeth and needles [2]. (see Fig. 1)

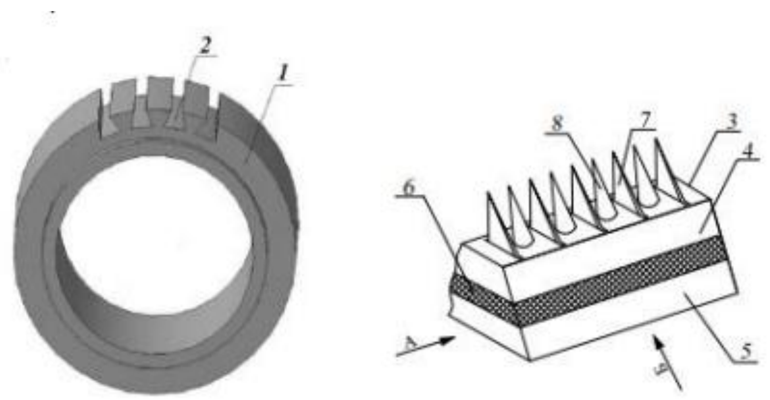


Fig. 1. Composite carding drum for pneumatic–mechanical spinning machines.

The proposed toothed garniture of the carding drum is mounted on its shaft through an elastic bushing. The garniture is divided into two identical parts. As a result, each part of the toothed garniture and the needles undergo oscillations in both rotational and vertical directions when interacting with the fibers. It should be noted that the amplitude of these oscillations during rotation is very small, yet they contribute to the straightening and parallel alignment of the fibers. Therefore, the analysis of vertical oscillations of the carding drum is of significant importance. It should also be noted that if the cross-sectional dimensions of the fiber sliver are (9.0 × 2.0) mm², its density in the middle part will be relatively high. During carding, the majority of fibers pass through the central area of the drum, while the number of fibers passing through the outer sections is somewhat smaller. Consequently, the amplitude of oscillations of the toothed garniture in the middle part of the drum will also be slightly higher compared to the amplitudes in the outer sections. Overall, it is important to dynamically ensure the equalization of oscillation amplitudes across all three parts of the toothed garniture. The theoretical studies are aimed precisely at achieving this goal.

Objects and Methods of Research

The parts of the toothed garniture of the discretizing drum have identical dimensions. Therefore, their calculation schemes are also identical. The overall calculation scheme is shown in Fig. 2. To obtain the differential equation describing the oscillations of the toothed garniture, Lagrange's equation of the second kind was used [3, 4].

$$\frac{d}{dt}\left(\frac{\partial \tau}{\partial \dot{q}i}\right) - \frac{\partial \tau}{\partial qi} + \frac{\partial \Phi}{\partial \dot{q}i} + \frac{\partial \Pi}{\partial qi} = Q(qi) \quad (1)$$

Here, $q_i, Q(q_i)$ – are the generalized coordinates and generalized forces, respectively.

Т,П – **Kinetic and potential energies**; Φ – **Rayleigh dissipation function** [5].

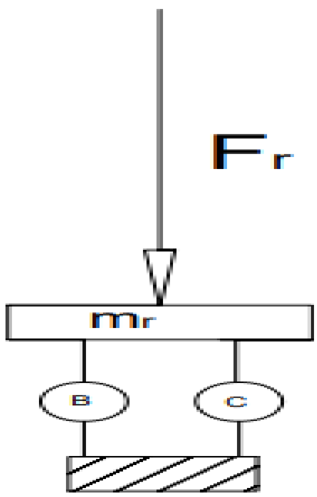


Fig. 2. The calculation (analytical) model describing the vertical vibrations of the comb drum's toothed element.

Based on the analytical model, the potential and kinetic energies, as well as the dissipation function of the elements of the toothed assembly, are determined by the following expressions [6]:

$$\Pi = \frac{1}{2}\mathrm{cy}^2; \mathrm{T} = \frac{1}{2}m_q; \Phi = \frac{1}{2}b(\frac{dy}{dt})^2 (2)$$

Here, $C\ and\ B$ are the stiffness and dissipation coefficients corresponding to the linear deformation of the rubber bushing located beneath the comb elements; $y$is the vertical displacement; $m_{\text{г}}$-is the mass of the toothed assembly element. Based on the obtained expressions (2), the derivatives with respect to the generalized coordinate were calculated [7].

$$\frac{\partial \Pi}{\partial y} = cy;\ \frac{\partial \Phi}{\partial y} = b\frac{dy}{dt};\frac{\partial \mathrm{T}}{\partial \dot{y}} = m_q\dot{y};\frac{\partial \mathrm{T}}{\partial y} = 0;$$

he time derivatives will have the following form [7,8]:

$$\frac{d}{dt}\left(\frac{\partial \mathrm{T}}{\partial \dot{y}}\right) = m\frac{d^2y}{dt^2}; (3)$$

The generalized force [8]:

$$F_l = F_1 + F_0 sin\omega t \mp \delta F_1 (4)$$

By substituting expressions (3) and (2) into (1), a differential equation was obtained that describes the vertical vibrations of the toothed and needle assemblies of the combing drum [8,9,10].

$$m_q\frac{d^2y}{dt^2} + cy + b\frac{dy}{dt} = F_1 + F_0 sin\omega t \mp \delta F_1 (5)$$

The resulting expression (5), based on the solution describing the free vibrations of the toothed and needle assembly elements [8], has the following form:

$$y = (E_1 sin\omega_0 t + E_2 cos\omega_0 t) (6)$$

$$\text{Where } \omega_0 = \sqrt{p_0^2 - n^2}; p_0 = \sqrt{\frac{c}{m_q}}; n = \frac{b}{2m_q}$$

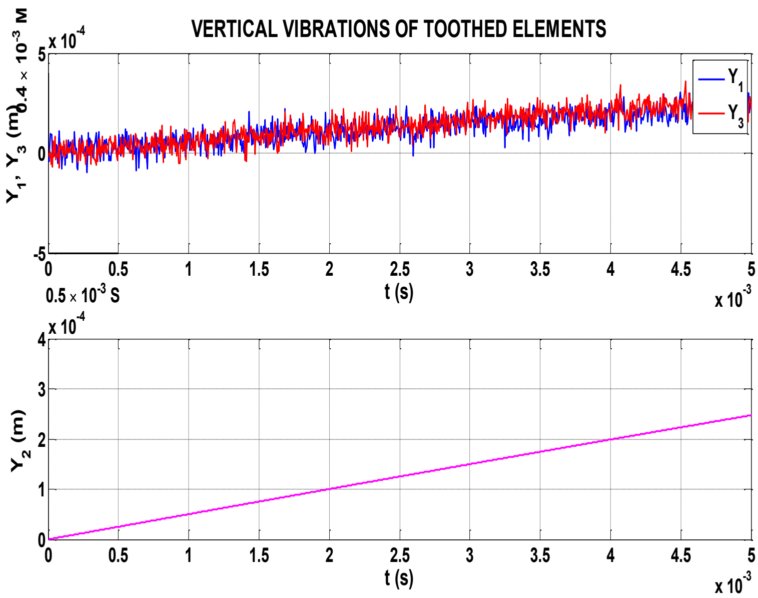
For $F_1 = 0 \ in \ \delta F_1 = 0$ this solution variant, according to [9], the expression for the forced vibrations has the following form:

$$y = \frac{F_0 sin\left[\omega t - arctg(\frac{2n\omega}{p_0^2 - \omega^2})\right]}{m_q\sqrt{(p_0^2-\omega^2)^2+4\pi-\omega^2}} \quad (7)$$

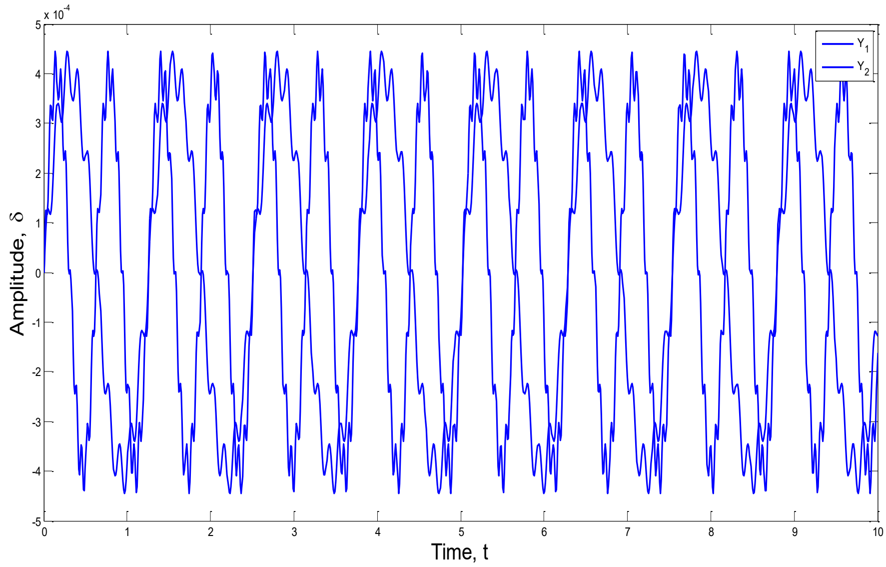
However, the general solution of equation (5) was implemented on a computer using the Runge–Kutta method and a random-number generator [5]. To determine the vibration patterns of the gear transmissions of the discretizing drum, the numerical solution obtained from expression (7) was carried out separately for each transmission. The calculations were performed at the following parameter values:

$$m_2 = (1.2 \div 1.8) \times 10^{-2} kg; n_\partial = (6.0 \div 7.5) \times 10^{-3} rev/min$$

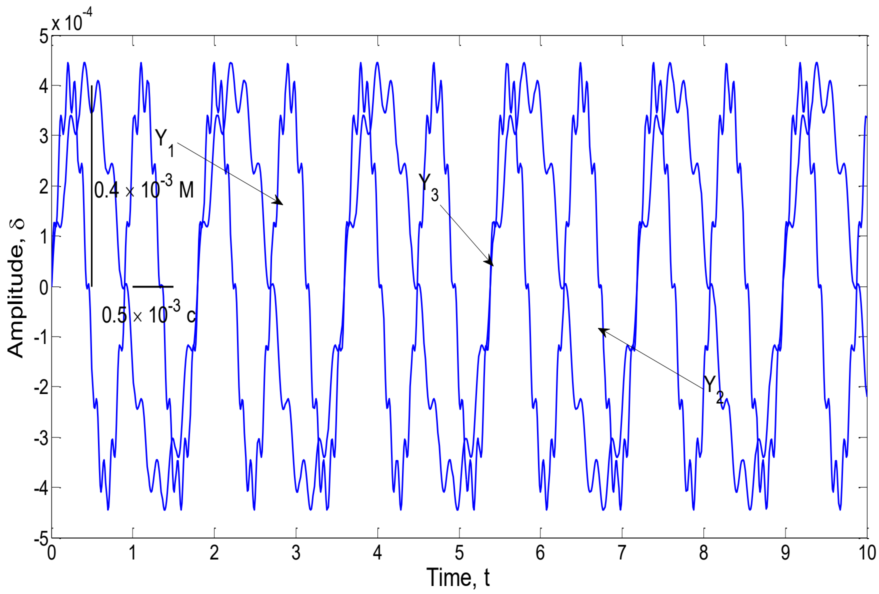
$$F_1 = (1.2 \div 2.4)cN, F_0 = (2.3 \div 4.5)cN, \delta F_1 = (0.25 \div 0.65)cN, c = (0.08 \div 0.35) \times \frac{10^3 N}{m}, b = \frac{(1.3 \div 2.5)Ns}{m};$$



$$F_L = (12 + 2.3 \sin \omega t \pm 0.25)cN,, c = 0.25 \times \frac{10^3 N}{m}, b = \frac{2.5Ns}{m}; n_\partial = 6 \times \frac{10^{-3} rev}{min};$$



$$F_L = (18 + 3.35 \sin \omega t \pm 0.45)cN,, c = 0.25 \times \frac{10^3 N}{m}, b = \frac{2.5Ns}{m}; n_\partial = 6 \times \frac{10^{-3} rev}{min};$$



$$F_L = (24 + 4.35 \sin \omega t \pm 0.65)cN,, c = 0.25 \times \frac{10^3 N}{m}, b = \frac{2.1 \times Ns}{m}; n_\partial = 6 \times \frac{10^3 rev}{min};$$

Figure 3. Laws of vertical vibrations of the recommended composite discrete drum gear sets as functions of the technological and drive rotational frequencies.

Discussion

In the obtained patterns, the vertical displacement behaviors of all three gear sets of the discrete drum were analyzed separately (Figure 3, graphs y, y_2, y_3). Figure 3 shows the dependence of the vertical vibration laws of the recommended composite discrete drum gear sets on the technological and drive rotational frequencies. Analysis of the obtained patterns revealed that the vibration amplitude of the gear set in the middle of the discrete drum is higher than that of the peripheral gear sets. The main reason for this is the larger technological resistance acting on the middle gear set.

This is because the density of pile fibers is higher in the central zone. It should be noted that the vibration amplitudes y_1, y_2, y_3 also depend on the linear stiffness coefficients of the rubber bushings in which the gear sets are installed. As seen from the laws shown in Figure 3., an increase in technological resistance leads to a corresponding increase in the vibration amplitudes of the gear sets.

Based on the processing of the obtained patterns, graphs showing the dependence of the vertical vibration envelopes of the gear sets on their parameters were constructed. Analysis of these graphs revealed that as the drive rotational frequency increases from $4.5 \cdot 10^3$ rpm to $7.2 \cdot 10^3$ rev/min , the vibration envelopes of the two peripheral gear sets increase in a nearly linear manner $F_1 = (12 + 2.3 \sin \omega t \pm 0.25)cN$, from $0.130 \cdot 10^{-3}$ m to $0.615 \cdot 10^{-3}$ m. The trends of their changes are almost identical, differing only in the phase shifts of the vibrations.

Conclusion

Analysis of the graphs shows that when the damper stiffness increases from $0.35 \cdot 10^3$ N/m to $0.65 \cdot 10^3$ N/m and $F_1 = 12.0\ cN$, the values decrease from $0.62 \cdot 10^{-3}$ m to $0.132 \cdot 10^{-3}$ m. Similarly, when $F_1 = 18\ cN$, the values decrease linearly from $0.93 \cdot 10^{-3}$ m to $0.26 \cdot 10^{-3}$ m . For the central gear set, the vibration envelope decreases from m to m when F1 = 12.0 cN. When $F_1 = 18\ cN$, the values decrease in a nonlinear manner from m to m. Therefore, to ensure that the values remain within the m range, it is advisable to select higher stiffness values for the generalized damper, in N/m. Figure 1 presents graphs showing the dependence of the vibration envelopes of the recommended discrete drum gear sets on their parameters. Analysis of these graphs indicates that the vertical vibration envelope of the gear set in the central zone increases relative to the peripheral gear sets as both the technological resistance and the drive rotational frequency increase.

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