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INFLUENCE OF GEOMETRICAL AND FUNCTIONAL PARAMETERS ON THE BENDING VIBRATIONS OF TEXTILE SPINDLES

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ABSTRACT
This paper continues the investigations carried out by the authors on bending vibrations of long textile machines. The type of spindle under investigation is SKF, the same as in [2]. A dynamic model has been adopted and Lagrange's formulae were used for finding the equations of vibration motion. The amplitude of vibrations over a wide range of working speeds were calculated by means of a program written in Microsoft FORTRAN, based on Runge-Kutta numerical method. The influence of some geometrical and functional parameters (stiffness, unbalance, and damping) upon textile spindle operation was analyzed.
Keywords : vibration, textile spindle.

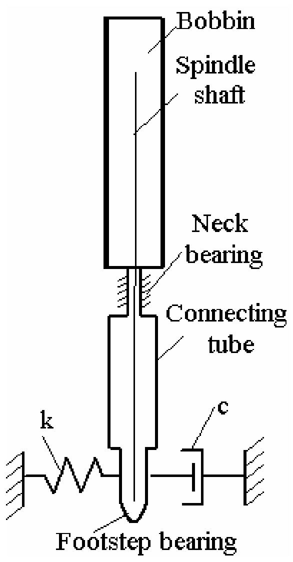
1. INTRODUCTION

The production and quality of yarns processed on classic spinning frames and twisting machines are influenced by factors dependent on textile spindles. The influence of spindles becomes even stronger in case of machines of over 400 work stations and working speeds of over 20000 rpm. For providing a reasonable limited tension in the yarn and number of rupture as well as a corresponding quality of the yarn, one need to provide correct and secure operation of the spinning ensemble, the spindle is a part of. This implies a low level of vibration, noise and low energy consumption.
The present paper is a sequel of paper [2] and a part of researches carried out by the authors on vibrations of spinning and winding mechanism in spinning machines with a great number of working stations.
The goal of investigations is finding a viable solution to improve spindle performances. For this, one needs to establish the optimum relationships required between the geometrical characteristics of this ensemble, masses and mechanical properties of components, as well as the damping, so that the magnitude of the vibration amplitude and noise level be minimum.
This paper aims at examining the influence of the following constructive and functional factors upon the vibratory behavior of spindle: the degree of unbalance, stiffness of lower bearing and damping degree.

2. DETERMINING THE AMPLITUDE OF BENDING VIBRATIONS

The spindle under investigation is of SKF-type, lightly loaded, rigidly mounted on the frame. It is belt driven, having the lower bearing elastically connected, with capabilities of hydraulic damping of vibrations (fig.1).
The dynamic model adopted is that of [3] as well as the simplifying hypotheses. These are:

- spindle shaft, the bobbin and connecting tube are considered to be rigid, centered and dynamically balanced;
- the elastic and damping forces are on a level plane with radial directions;



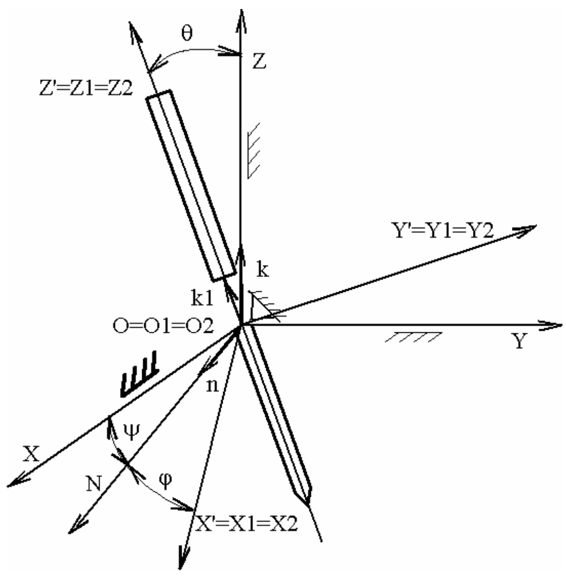


Figure 1. Schema of textile spindle

Figure 2. Dynamic model for the textile spindle (three degrees of freedom)

- the elastic force complies with Hooke's law and the damping force is of a viscous type;
- the disequilibrium mass is considered to be concentrated in a point;
- internal and external friction forces are neglected and the belt is considered simply as driving torque.

The motion of spindle shaft- bobbin and connection tube subassembly is assimilated with the rigid of the fix point (the neck bearing) and three degrees of freedom. Its position is related with a fix reference system Oxyz with the origin in the neck bearing.

For establishing the equations of vibration motion, Lagrange's formulae from analytical mechanics are used, the generalized coordinates being Euler's angles (θ, ψ, φ). These formulae applied to the moving system are of the form:

$$\begin{cases} \dfrac{d}{dt}\left(\dfrac{\partial Ec}{\partial \dot{\theta}}\right) - \dfrac{\partial Ec}{\partial \theta} = -\dfrac{\partial V}{\partial \theta} + Q_{\theta}^{dis} + Q_{\theta}^{m} \\ \dfrac{d}{dt}\left(\dfrac{\partial Ec}{\partial \dot{\psi}}\right) - \dfrac{\partial Ec}{\partial \psi} = -\dfrac{\partial V}{\partial \psi} + Q_{\psi}^{dis} + Q_{\psi}^{m} \\ \dfrac{d}{dt}\left(\dfrac{\partial Ec}{\partial \dot{\varphi}}\right) - \dfrac{\partial Ec}{\partial \varphi} = -\dfrac{\partial V}{\partial \varphi} + Q_{\varphi}^{dis} + Q_{\varphi}^{m} \end{cases} \tag{1}$$

After the calculation of kinetic energy Ec, potential energy V and of generalized forces Q, the equations of system (1) of non-linear differential equations of second order with three unknowns can be constructed. This turns into a system of four first order differential equations through changes of variables:

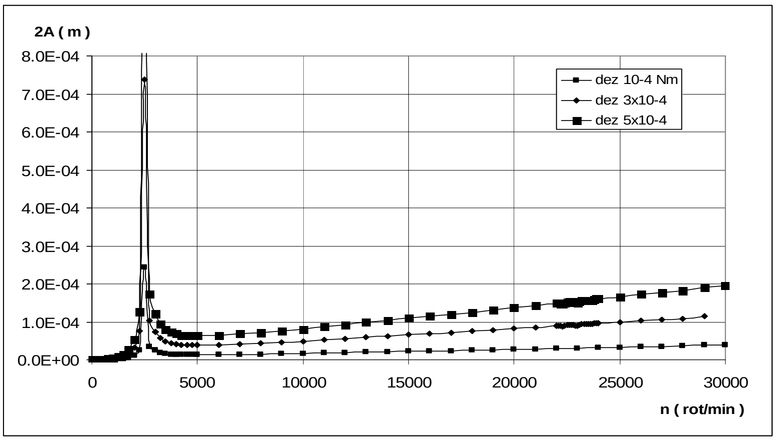
$$\begin{cases} \dot{y}_1 = y_3 \\ \dot{y}_2 = y_4 \\ \dot{y}_3 = \dfrac{f_2(y_1,\varphi)\cdot f_6(y_1,\varphi,y_3,y_4,\dot{\varphi},\ddot{\varphi}) - f_3(\theta,\varphi,\dot{\theta},\dot{\psi},\dot{\varphi},\ddot{\varphi})\cdot f_5(y_1,\varphi)}{f_1(\varphi)\cdot f_5(y_1,\varphi) - f_2(y_1,\varphi)\cdot f_4(y_1,\varphi)} \\ \dot{y}_4 = \dfrac{f_3(\theta,\varphi,\dot{\theta},\dot{\psi},\dot{\varphi},\ddot{\varphi})\cdot f_4(y_1,\varphi) - f_1(\varphi)\cdot f_6(y_1,\varphi,y_3,y_4,\dot{\varphi},\ddot{\varphi})}{f_1(\varphi)\cdot f_5(y_1,\varphi) - f_2(y_1,\varphi)\cdot f_4(y_1,\varphi)} \end{cases} \tag{2}$$

which can be solved by numerical methods.
The calculation of vibration amplitudes induced by spindle flexion was made by means of program written in Microsoft FORTRAN language, based on the Runge-Kutta numerical method with four accuracy degree.

3. THE STUDY OF THE INFLUENCE OF SOME PARAMETERS ON THE VIBRATION AMPLITUDES

The influence of the disequilibrium, of the footstep bearing rigidity and of damping coefficient upon induced vibrations, by respected running program and results processing were investigated. For considering the disequilibrium degree, mass and eccentricity values were used so that recommended admissible disequilibrium moments result in [3[.
The stiffness of the footstep bearing was entered in the calculation through the value of the elastic constant. Its accurate determination by calculation is difficult due to the bearing construction (it contains the end thrust bearing, a helical spring, a spiral spring, a helical canal tube). Thus, a wide range of values have been considered in conformity with the indications from [4]. For damping coefficient values recommended or calculated according to formulas from [1] were used.
Thus, the following graphs have been drawn by means of program EXCEL:

- amplitude of vibration within the range 0–30000 rpm, for various degrees of disequilibrium (fig.3);
- amplitude of vibration within the same range, for various damping coefficients (fig.4);
- amplitude of vibration within the same range, for various values of elastic constant of footstep bearing (fig.5);



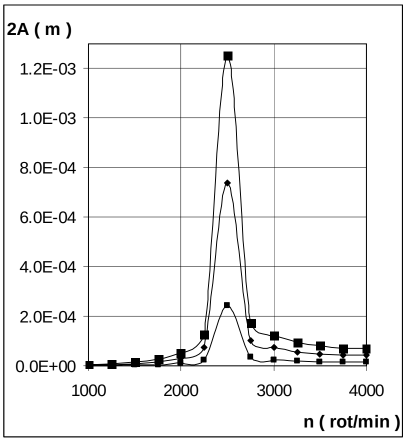
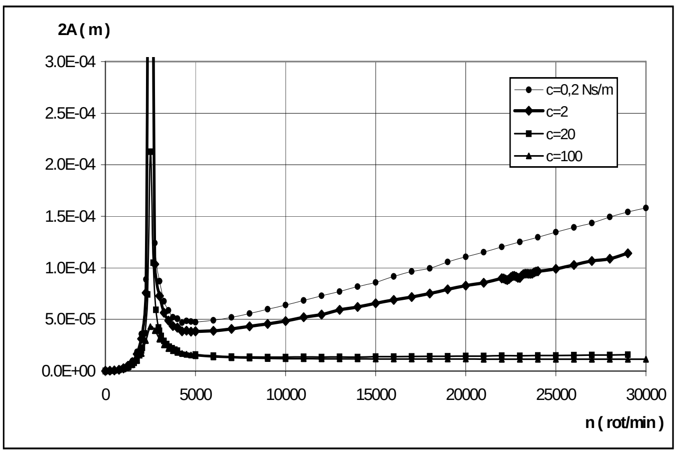


Figure 3a. Influence of unbalance on the amplitude (c=2 Ns/m; k=1.5x10^5 N/m)

Figure 3b. Detail



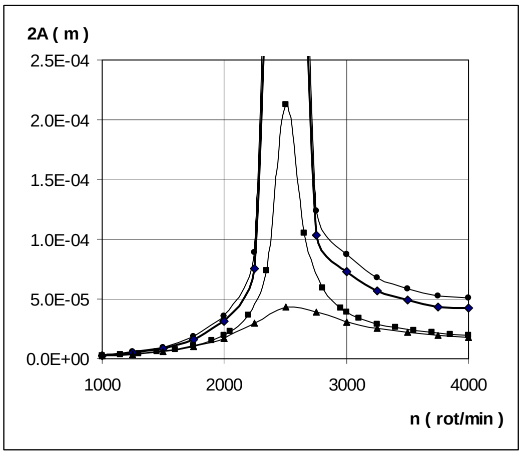


Figure 4a. Influence of damping on the amplitude (dez=5x10^{-4} Nm; k=15x10^4 N/m)

Figure 4b. Detail

4. CONCLUSIONS

The amplitude of spindle bending vibrations varies proportionally with the value of the moment created by unbalance mass thus:

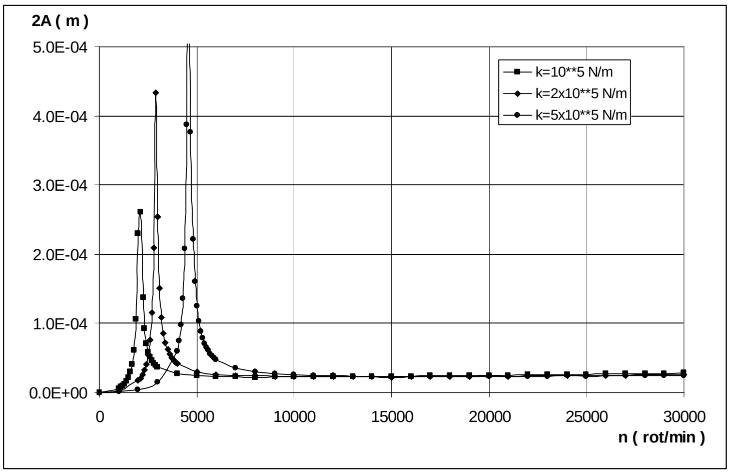


Figure 5a. Influence of stiffness on the amplitude (dez=5x10^{-4} Nm; c=20 Ns/m)

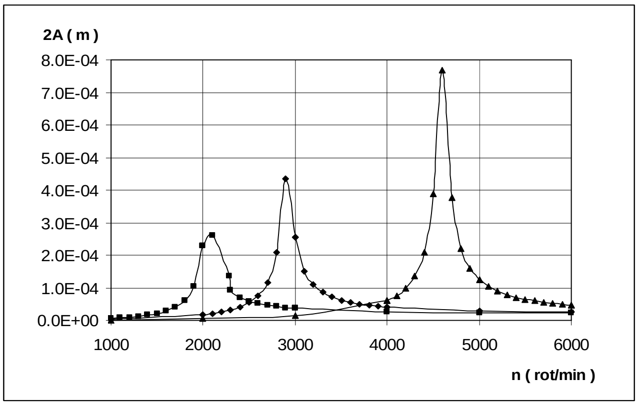


Figure 5b. Detail

- on trebling the unbalance moment an increase of 200% in amplitude is recorded;
- for a certain stiffness of the bearing and an adopted damping coefficient, at an unbalance of 10^{-4} Nm, an amplitude at peak of 0,25 mm resulted, at a critical speed of 25 rpm, and much lower in the rest (behind 0,15 mm at 20000 rpm).

The amplitude of spindle bending vibrations varies inversely proportional with the damping force:

- for c = 20 Ns/m the amplitude at a speed of 2500 rpm is 0,21 mm and 10 times lower in the rest;
- for a 10 times fall of the damping coefficient (c = 2), the amplitude of bending vibrations increases 3,5 times.

The increase of stiffness of footstep bearing displaces the critical zones towards higher speeds, causing the amplitudes increase within the critical speeds of rotation, thus:

- doubling stiffness as compared with reference values k=10^5 N/m brings about the increase of the prime critical speed of 38%, and an increase of 1,7 times of the correspondent amplitude;
- the increase of stiffness of 5 times brings about an increase of prime critical speed of 119 %, and 3 times the corresponding amplitude;
- outside the critical zones, the influence of stiffness is minimum.

In short, we can conclude that the amplitude of spindle bearing vibrations varies proportionally with the value of the moment produced by the unbalancing mass and inversely proportional with damping force. The magnitude of bearing stiffness displaces the critical zones towards higher speeds.

Researches should be continued for finding optimal constructive solutions, appropriate to each type of material manufactured, so that the value of vibrations amplitude be minimum for each working station and for the entire machine.

4. REFERENCES

[1] Ghiolțean, L.M. Contribuții la studiul dinamic şi de vibrații al maşinilor textile, Teză de doctorat, Facultatea de Mecanică, Cluj-Napoca, 2003.

[2] Ghiolțean, L., Tripa, M., Study of free bending vibrations for textile machines spindles, 10 th International Research/Expert Conference „Trends in the Development of Machinery and Associated Technology”, TMT 2006, Proceedings, Barcelona-Lloret de Mar, Spain, 11-15 September, 2006, ISBN 9958-617-30-7.

[3] Hanganu, V., Hanganu, L., Maşini pentru filatură, Curs litografiat, Institutul Politehnic Iaşi, 1992.

[4] Sepetilnikova, V.A., Osnovi balansiiravocinoi tehniki, Tom 1, Masinostroenie, Moskva, 1975.

[5] Ursu-Fischer, N., Vibrațiile sistemelor mecanice, Editura Casa carții de ştiință, Cluj-Napoca, 1998.