Middle-East Journal of Scientific Research 16 (12): 1643-1647, 2013
ISSN 1990-9233

DOI: 10.5829/idosi.mejsr.2013.16.12.12028

Mathematical Model of Powerloaing Experimental Pad by Roll Spindles

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Abstract: As you know, getting adequate mathematical models and precise dynamic characteristics of each drive unit is very challenging. So this kind of problem is usually seen in a more simplified form for each case. In this regard, on the study of this drive unit, the following assumptions are made: power strip is under the influence of only one incident at its spindle and pad reverse rotation of the spindles is mainly exposed to the efforts of the district. On the basis of that we developed a mathematical model of the dynamics of loading; the article received engineering formulas to calculate the geometric, kinematic and dynamic parameters of vertical spindle cotton-harvesting apparatus (machine), their rollers and pads.

Key words: Vertical-spindle cotton-harvesting apparatus (machine) · Spindle · Spindle drive · Pads reverse rotation spindles · Traction capability pads

INTRODUCTION

In the technological process of vertical-spindle cotton-harvesting apparatus (machine) drive rollers of spindles alternately interact with straps of forward and reverse rotation. The movement of rollers consists of an instant in time phases, as acceleration, steady rotation, free rotation and braking.

As is known [1], the drive rollers cotton - harvesting machine, the raids atω_w to reverse rotation pad, is causing its counteraction. It consists of normal F_n and tangential F_f effort, passing on to the rolling circumference of the roller (Fig. 1).

Changing to these efforts consist of shock-dynamic nature, negatively influencing on durability of pads and rollers. To reduce strike force of rollers, it was requested (Fig. 2). The pad with increasing acceptability [2], containing elastic springand leash set at an angle φ.

Experimental mechanism of drive spindle consists of reverse rotation pad 1, with batteries link 2 and from direct-belt rotation 3. It differs from the serial, that mounting pads on the spindle drum [3], consisting of the guide bracket and guide bearing, replaced with the batteries link. One of the ends of the batteries link withes hinged with a frame pads and other – with the upper body of the drum.

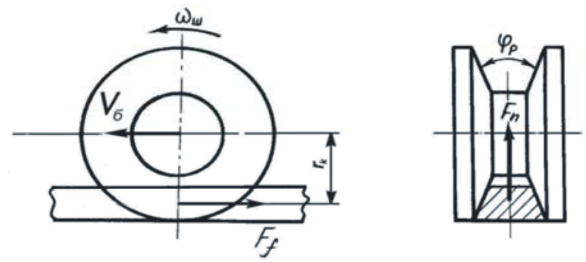

Fig. 1: Schematic of the forces acting on the rollers, *the raids to reverse rotation pad ($V_б$– linear speed of the spindle drum axis spindles. φ_p – groove angle roller)*

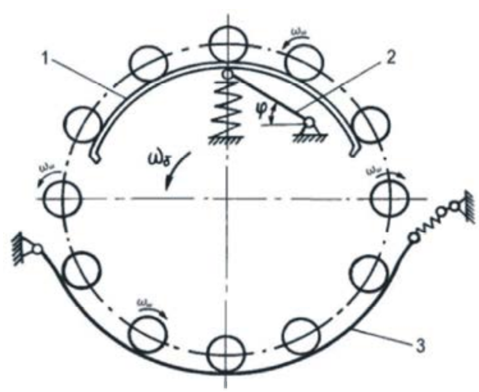

Fig. 2: The pilot scheme of experimental mechanism of drive spindle (1-reverse rotation pad;2 - *batteries link; 3-direct-belt rotation.)*

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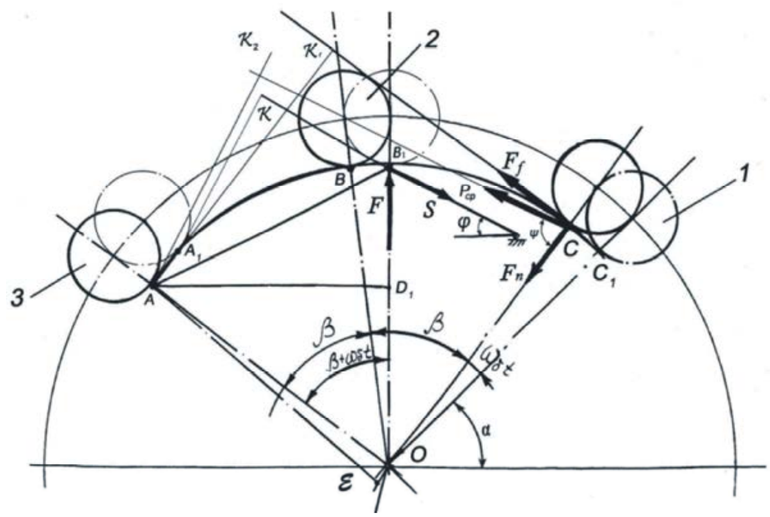

Fig. 3: Scheme forces acting on the reverse rotation pads of experimental mechanism of cotton –harvesting apparatus

Pads - structural framework of the actuate bend made of metal with a rubber gasket three brook straps, providing speed control roller spindle. Accordingly, the ratio between the roller and by the geometry of the curved material was obtained in the production process design [4].

Proposed Theory: Spring and leash in the process of interaction between the two bodies - the «roller-pad» creates elastic forces $\bar{F}$ and $\bar{S}$, contributing to extinction of the impact force on the roller pads. Powerful loading F_f *and* $F_n$linked dependence

$$F_f = fF_n, \tag{1}$$

where, $f' = \dfrac{f}{sin\dfrac{\varphi_p}{2}} = \dfrac{0,34}{sin\dfrac{38}{2}} \approx 1,0$

f' - reduced coefficient of friction;
f – 0, 34 -the coefficient of friction of steel on rubber;
$\varphi_p = 38°$ *– groove angle rollers.*

Then you can take$F_f = F_n$. (2)

The solution is designed to shear impact roller spindle to the elastic sphere (V-belts) in a half. By Hertz theory - called the mechanics of contact interaction, impact strength is used for the normal components of force and velocity and it is assumed that the friction coefficient is constant [5].

In turn, have a movie to a shoe similar impact. And here is the same effort as before, but they already have the opposite action (Fig. 3). In addition to a shoe back spindles are: F-pad pressing force to the drive rollers; S - flanged reaction force management.

Interactions remaining rollers in the zone pads neglected since their rotational resistance moment of their relatively small. To simplify the tasks and ignore the beating rollers and elastic deformations of the friction drive elements [6].

Mathematical Calculations: Equilibrium equations pads reverse rotation, the equations sum of the moments of forces about the center of rotation of the spindle drum and at the point of contact pads and rollers can be written as:

$$(S \cos\varphi) \cdot \cos\varphi - F_f = 0; \tag{3}$$

$$2S \cdot \sin\left(\frac{\beta+\omega_6 t}{2}\right) \cdot \sin\left(\varphi + \frac{\beta+\omega_6 t}{2}\right) - 2F_f \sin^2\beta + F_n \sin 2\beta - F\sin(\beta + \omega_{6t}) = 0 \quad (4)$$

After the corresponding transformations of equations (3) and (4) with (2) we get [3]

$$F_f = \frac{F\sin(\beta+\omega_6 t)\cdot\cos\varphi}{2\sin\left(\frac{\beta+\omega_6 t}{2}\right)\sin\left(\varphi+\frac{\beta+\omega_6 t}{2}\right)+\left(\sin 2\beta - 2\sin^2\beta\right)\cdot\cos\varphi} \quad (5)$$

Hereβ - angle arrangement of spindles on the periphery of the spindle drum;
$\omega_6 t$ - yaw angle of the roller on the raids to reverse rotation pad.

Expression (5) is equation circumferential force exerted by the strip on the axis of the reversible roller spindle vertically cotton-harvesting machine. For example, a machine with 8 drum spindle ($\beta = \frac{2\pi}{8} = 45°$) it has the form:

$$F_f = \frac{F\sin(45^\circ + \omega_6 t)\cdot\cos\varphi}{\cos\varphi - \cos\left(45^\circ + \omega_6 t + \varphi\right)}, \quad (6)$$

At the time the drive rollers cotton - harvesting machine, the raids to reverse rotation pad (t = 0) it is a circumferential force

$$F_{f(t=0)} = \frac{\frac{\sqrt{2}}{2}F\cdot\cos\varphi}{\cos\varphi - \cos(45^\circ+\varphi)} = \frac{\sqrt{2}F\cos\varphi}{2\left(\cos\varphi - \cos(45^\circ+\varphi)\right)}. \quad (7)$$

For serial cotton-picking machine with a 12-spindle drums have

$$F_f = \frac{F\cdot\sin(2\beta+\omega_6 t)\cdot\cos\varphi}{2\sin(\beta+0{,}5\omega_6 t)\sin(\varphi+\beta+0{,}5\omega_6 t)+(\sin 4\beta - 2\sin^2 2\beta)\cos\varphi}. \quad (8)$$

If $(\beta = 30°)$ this equation can be reduced to the form

$$F_f = \frac{F\cdot\sin\left(60^\circ+\omega_6 t\right)\cos\varphi}{\frac{\sqrt{3}}{2}\cos\varphi - \cos(60^\circ+\varphi+\omega_6 t)}, \quad (9)$$

Where t = 0

$$F_f = \frac{\sqrt{2}F\cdot\cos\varphi}{\sqrt{3}\cos\varphi - 2\cos(60^\circ+\varphi)} \quad (10)$$

As follows from equation (7), with $\varphi = 90°$, regardless of the value of *F,* holds $F_f = 0$. This means that the reverse rotation pads is not able to develop the roll axis collateral to peripheral force, it is unusable [7].

And at φ=0 block turns into a serial devoid of compliance at all. This means that the serial reverse rotation pads are a special case a new master. Of particular interest may cause a case where there is a$F_f = F$. Then equation (7)

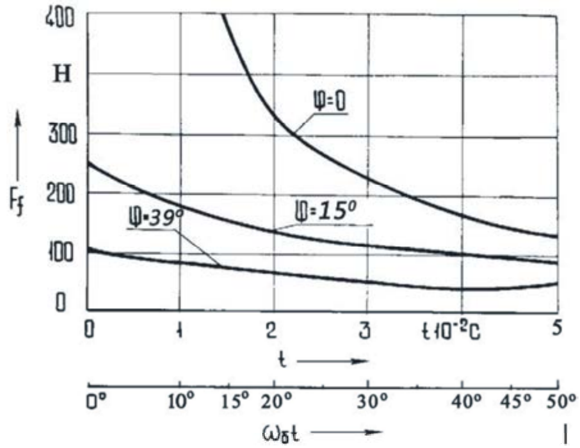

Fig. 4: Change in the district efforts on the roller (F_f) depending on the angle of the drive disc level and angle- φ drum- $\omega_б t$

$$\varphi = arctg\left(\frac{\sqrt{3}+\sqrt{2}-2}{\sqrt{2}}\right) \qquad (11)$$

Hence, φ = 30° reverse rotation pads has a normal suppleness in which at the time of the reverse spindle circumferential force on the axis of the drive roller equal to the force pressing the pads.

When φ> 30° occurs F_f<F yielding pads increases. But there may be a way to reduced traction pads, increased coefficient of slipping clips on straps. This can lead to overheating and rapid failure of the belts [8].

When *φ<30 °* and *F_f> F* yielding pads down and its ability to increase traction.

Graphical Results: The regularity of district efforts to roll axis as developed in an effort to block clamping it to the drive rollers are equal to *F = 100N,* depending on the angle- φ install the flanged managers and yaw angle of the roller on the raids to reverse rotation pad -$\omega_б t$, are shown in Fig. 4. It follows that in the early clash of the roller on the pad peripheral force F_f reaches its maximum value, which depends on the angle of the drive disc level - φ varies widely [9]. When φ = 30° its maximum value equal to the force pressing the pad, i.e. F_f = F.

The results are presented in diagrams.

For preliminary calculations assume the following limits the angle of the drive disc link:*15° = φ = 30°,* at which holds

$$F_f = (1, 0...2, 41) \cdot F, \qquad (12)$$

Note that (12) hold for *f′ =1, 0.*

Low coefficient of friction in the technical systems without lubrication can have a significant impact on the development of many components of the machine and to define its exact dynamic characteristics [10].

In fact, this only occurs when the geometric sliding in a pair of friction is very small, that occur during steady state operation. At the time of reverse roller slide is large enough and the rate of speed of its roller circumferential speeds commensurate spindle drum and in which the friction coefficient should be substantially smaller [11].

CONCLUSIONS

The resulting mathematical model of the power to determine the loading forces driving this load then set their functional connections.

An analytical expression of the efforts of the district on the axis of the drive roller spindle during its interaction with the reverse rotation of the pads, which is dependent on the design parameters of the interacting elements, clamping force block and the angle of the drive disc its management.

The experiments revealed that the most probable values of the reduced coefficient of friction *f′* 'is within 0, 5…0, 9.

Given *f′* = 0,5…0,9, when $\omega_б t = 0$, *β* - 45° and *β-30°* from equation (7) and (10) to obtain a small-sized and mass cotton-harvesting apparatus respectively

$$F_f = \frac{F}{0{,}189 + tg\varphi};\; F_f = \frac{F}{0{,}517 + tg\varphi}.$$

Received these engineering formulas to calculate the geometric, kinematic and dynamic parameters of the spindles are the basis for the calculation of vertical-spindle cotton-harvesting machine, their rollers and pads.

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