Ryszard Przytulski

Technical University of Łódź
Department of Mechanics of Textile Machines
ul. Żeromskiego 116, 90-543 Łódź, Poland

Analysis of the Thread Guide Drive Mechanism

Abstract
An analysis of a complex tooth-lever mechanism to drive the thread guide (a device that forms cheese yarn packages) is presented in this paper. The described version of the mechanism design is equipped with elastic fenders that correct the positions of the thread guide lever. On the basis of a computer simulation of the mating of this device's basic links, a kinematic analysis of the thread guide motion was conducted. The influence of flexibility and the positions of the correction fenders on the parameters of thread guide motion has been considered in the analysis. The position of the initial point of the lever-fender contact with respect to the reverse position of the slide was optimised.

Key words: *tooth-lever mechanism, slide, computer simulation.*

Introduction

Among numerous devices for forming cheese yarn packages, a guide groove is commonly encountered in winding machines. The imprecise distribution of the yarn with this device is caused by the variable depth of the guide groove on the drum perimeter. Single-point mechanisms in which the thread guide is driven by a screw with a cross thread or by a cylinder with a notched helix are also known. Yet another, multi-point guide system is represented by a device that consists of a rail that moves with a reciprocating motion, enforced by a cam mechanism. Examples of yarn packages and selected mechanisms of their formation have been presented in [1]. The yarn distribution at the package base depends on the curvature radius of the guide grooves of the transition through reverse positions. The minimisation of the radius is limited by dynamic reasons, which results in the fact that the yarn is too densely placed at the package base. The problem of optimisation and the correct structure of yarn packages have been described in [2].

This paper presents a structural, operational and kinematic analysis of the design solution of the thread guide drive mechanism developed by the author. A physical model of the device with patented rigid fenders [3] was built on this basis. In the current theoretical considerations, this is a tooth-lever mechanism with elastic fenders that sharpen the yarn arrangement line shape at the package base.

Design, Operation and Equations of Motion of the Mechanism

The device for making cheese yarn packages is shown schematically in Figure 1. The body of the drive mechanism is a fixed central gear wheel (1) with internal teeth. The satellite (2) with a rolling radius, which is twice as short as the central wheel, co-operates with the central wheel. The satellite wheel is mounted on a crank shaft (3) supported in bearings along the geometrical axis of the central wheel. The crank shaft is a driving link of this mechanism. A journal that forms a rotational link together with slide (4) is fixed to the satellite wheel on the radius R_2. Also, a two-arm lever (5) of the thread guide is mounted on this journal in a rotational way. A guide bar (6) protects the slide against rotations. Two elastic fenders (7) are fixed on the body of the device. The thread guide lever is additionally supported by elastic elements (8) with respect to the slide. These springs ensure the perpendicular position of the thread guide lever with respect to the slide motion in its positions beyond the contact with the fenders. The position of the yarn guide point corresponds to these instantaneous positions of the slide. The displacement of the slide is proportional to the crank radius and to the sinus of the angle of its instantaneous position.

A turn of the crank enforces a planetary motion of the satellite wheel. The lever R_2 that is related to this wheel enforces a reciprocating motion of the slide in the guide bar. The basic equations of the instantaneous positions of links for the motion analysis of the mechanism shown in Figure 1 are as follows:

$$X_x = R_1 \cdot \sin \alpha_1 - R_2 \cdot \sin \alpha_2$$
$$U = X_x + \Delta U \qquad (1)$$
$$\Delta U = R_3 \cdot \sin \alpha_3$$

where:
$$\alpha_3 = 0 \text{ for } X_x <= R_5$$
$$\alpha_3 = \arctan [(X_x - R_5 - \Delta X_s) / R_6] \quad \text{for } X_x > R_5 \qquad (2)$$

From the moment of contact of the thread guide lever with one of the elastic fenders, the lever end performs a motion corrected by the turn around the suspension axis of this lever in the slide. Then, a motion of the thread guide occurs which allows a cheese package with a sharpened line shape of the yarn arrangement at its base to be obtained.

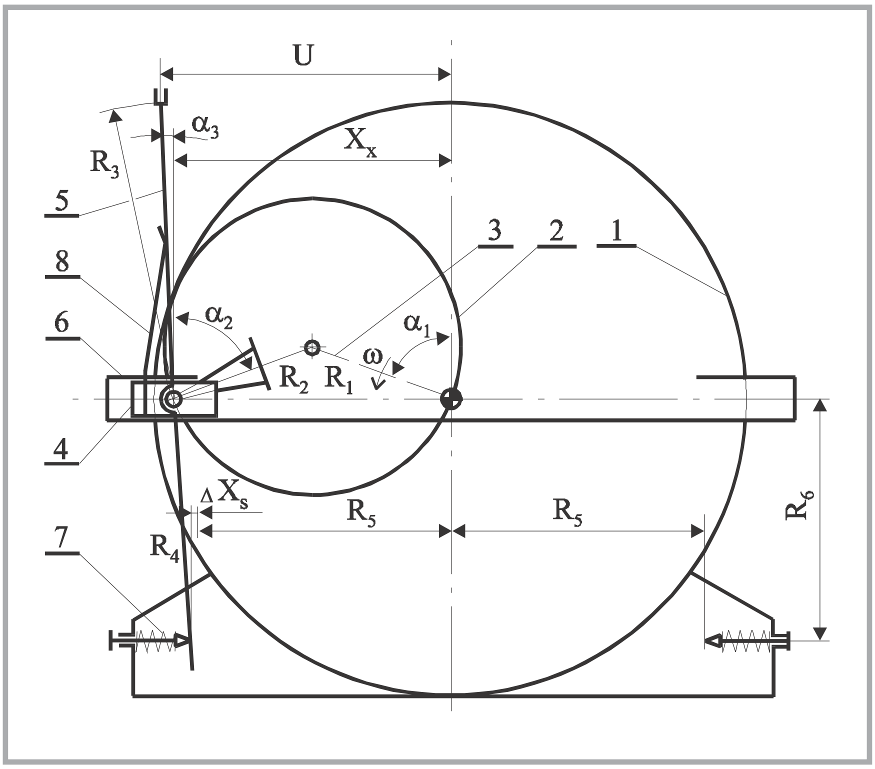

Figure 1. Schematic view of basic elements of the mechanism.

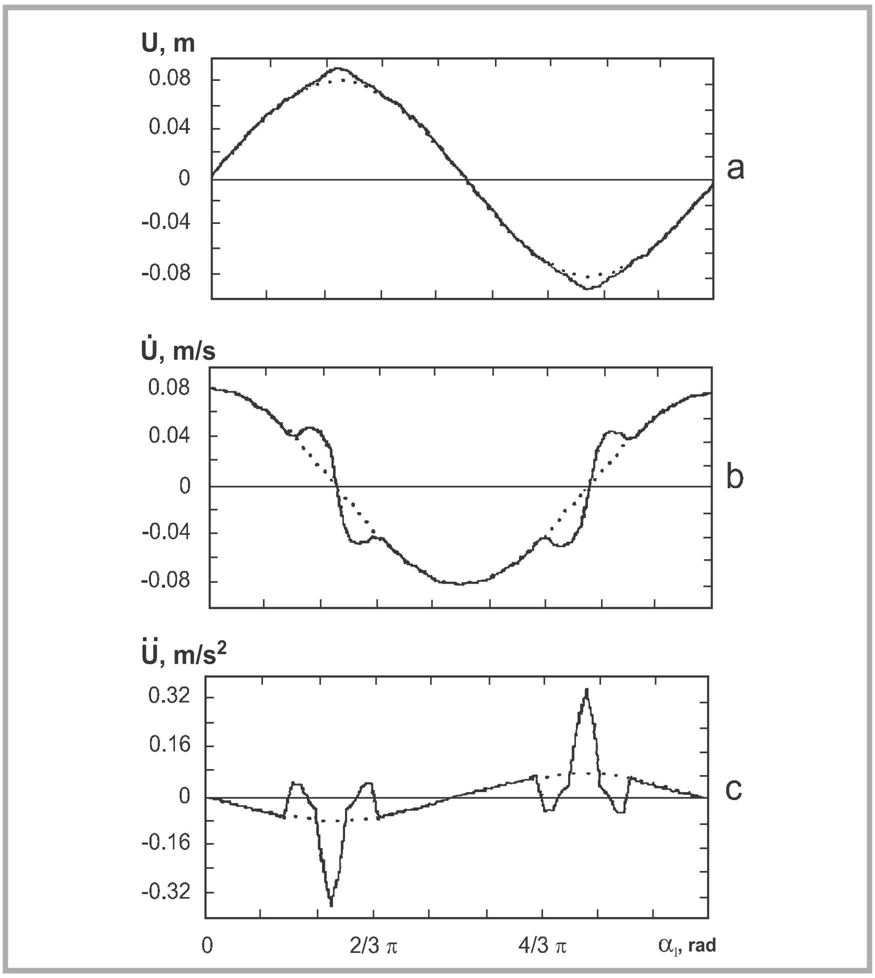

Figure 2. Parameters of the thread guide motion in one cycle of the mechanism motion: a - path, b - velocity, c - acceleration.

Parameters of the Computer Simulation of the Thread Guide Motion

The aim of the computer simulation was to check the shape of the yarn arrangement outline in the vicinity of the reverse positions of the thread guide which could be obtained with elastic fenders. The comparative investigations of the thread guide mechanism motion, with and without fenders, were carried out on the assumption of the unit quantity of the angular velocity ω of the driving crank. The geometrical parameters of the system of mechanism links correspond to the length of 0.160 m of the slide pitch of the physical model built. In constructing the central wheel element with the inner teeth outline, the technological conditions imposed this length of pitch. The doubled length of the cranks R_1+R_2 corresponds to the slide pitch. The remaining structural parameters of the mechanism are as follows: the lever elements $R_3=R_4=0.08$ m, and the region of the lever-fender contact point optimisation under analysis was limited by the segments $R_5=0.07\text{-}0.08$ m, $R_6=0.06\text{-}0.07$ m. It was assumed that the deflection ΔX_s of the fender elastic elements was a function of the angle α_1 of the main shaft crank R_1 position:

$$\Delta X_s = X_s \cdot k \cdot \sin^2 6\alpha_1 \qquad k = (n_i - n_o)/n \tag{3}$$

where:

X_s - the maximum deflection of the elastic element,

n_o - the first point in the contact region of the thread guide with the fender,

n_i - the next contact point in the given iteration step,

n - the number of computational ranges of this region.

In the computations, the Newton-Brent iterative method for solving non-linear equation systems was applied [4-7].

A plot of the thread guide path, its velocity and acceleration along the slide motion direction for $R_5=0.072$ m, $R_6=0.065$ m and $X_s=0.003$ m are shown in Figure 2. The figure presents a change in kinematic parameters as a function of the angle α_1 of the crank turn R_1 for one cycle of the mechanism motion. Dotted lines represent the plots for mechanisms without fenders.

Conclusions

The sharpened line shape of the yarn arrangement on the package obtained by means of this design solution of the mechanism differs significantly from the sinusoidal plot in the part of the slide path in the neighbourhood of its reverse positions in mechanisms without fenders. This advantageous change takes place when the thread guide lever co-operates with elastic fenders. At the instant of this contact, both an instantaneous increase in the velocity of the thread guide and a fault on the acceleration plot related to this change occur. However, an increase in the acceleration of thread guide reverse positions is inevitable.

It can be assumed that a change in these kinematic parameters will exert an influence on the force with which the thread guide affects the yarn being formed. When the dynamic analysis is performed, a comparison of kinematic parameters of the thread guide motion simulation with industrial devices used as well as a complete evaluation of this design solution suitability will be possible. This analysis must account for forces and momenta of inertia, as well as forces of elastic strains of elastic elements, and forces of damping of the thread being wound. With an increase in the winding cycle frequency, a change in dynamic loads can result in a deformation of the thread guide trajectory.

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For more information please contact:
Prof. Dr Alenka Majcen Le Marechal, Tel.: +386(0)22207910
Dr Vanja Kokol, Tel.: +386(0)22207896
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