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TRANSMISSION ERROR IN SYNCHRONOUS BELT DRIVES

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ABSTRACT. *The purpose of this paper is to analyze the transmission error in a synchronous belt drives. First, theoretical aspects are presented, detailing the kinematics model of a belt drive and the factors which affect the transmission error. An experimental device and a specific transmission error measurement method have been used. Results show the similitude of obtained values in both cases, theoretical and experimental.*

KEYWORDS. *Synchronous, timing, belt, drive, transmission error*

NOMENCLATURE

Symbol	Description
i, $i_{\max}$, i, $i_{\min}$	Speed ratio
Δr	Variation of wrapping radius
$\Delta\varphi$	Synchronism error

1. INTRODUCTION

Several methods of power transmission available for today's machinery are known, two of which include belt and chain drives. Both belt and chain drives possess a unique characteristic: they can be used in transmissions with large shaft center distances. Synchronous belt drives have significant advantages when com-pared to chain drives. Some examples are excellent elongation behavior and good damping characteristics under load.

Synchronous belts especially provide ideal solutions for power transmission, conveying, and linear drive applications because they offer high accuracy, lower maintenance and can accommodate many different product-conveying requirements. Weld-on profiles, backing materials, and custom machining modifications are just some of the options available for these applications. The flexibility of belts allows them to absorb shock loads, but the related compliance plays an important role with respect to large amplitude vibrations, which represent a key concern, especially for applications in automotive industry (Callegari et al, 2003).

Synchronous belts are frequently used in combustion engines (about one-third of industrial motors) for driving the timing/injection system as well as various accessories. Irregularities of crankshaft's rotary speed and alternating driving torques of timing/injection components lead to excitation of belt-system's natural frequencies. As a consequence, resonance effects will appear within the operating speed range causing significant rise in system's forces, lateral vibrations, wear, and noise. On the other hand, the difference between the polygonal shape of the pulleys and an ideal cylindrical shape initiates various effects in drive behavior.

Like any multi-axis timing function, synchronous belt drives require highly accurate positioning, or registration. Registration is the difference in angular position between two sprockets and can be classified as static or dynamic. Static registration concerns how accurately a drive moves from its initial to secondary position, and is determined primarily by backlash. Dynamic registration, on the other hand, is a measure of accuracy over an entire cycle and is subject to belt elongation, backlash, and tooth deflection. Both types of positioning accuracy must be considered when selecting belt drives (Dittmer, 2006).

2. KINEMATIC MODEL OF BELT DRIVE

Synchronous belt drive is obtained by gearing teeth on the front of the belt with corresponding teeth on wheels and driven pulleys (Fig. 1). The transmitted load is carried by cordage resistance inserted in the belt, made of fiber glass or steel and virtually inextensible under the transmitted force.

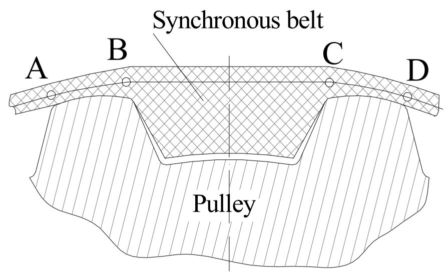

Fig. 1. Positions of the tension member (cordage) under meshing

During running belt, the neutral zone, where is the cordage, is not placed on a perfectly cylindrical surface. Flat areas between belt teeth (arcs AB and CD in Fig. 1) remain in direct contact with the teeth peaks of pulley and forces the cordage to adopt a path along a concentric circle arc of the wheel and belt radius equal with extra thickness range wheel coating on the inside of the belt.

In normal working conditions, no teeth of belt notes that would bend or twist, they occupying the direction of a string of circle, the teeth of the wheel, according to Fig. 1 (Child et al, 1997), (Firbank, 1977).

The manner how the synchronous belts go in and out of contact is studied by considering wrapping and unwrapping of the belt with fixed wheel. Regarding the native motion between the belt and the wheel the movements are equivalent. Referring Fig. 1, the following movements can be observed during the unwrapping process: (1) rotation round point A; (2) unwrapping along arc AB; (3) rotation round point B; (4) rotation round point C; (5) unwrapping along arc CD; (6) rotation round point D.

In order to clearly reveal the relevant geometric features, Firbank has proposed a simply model for a drive between pulleys showing three and four teeth respectively.

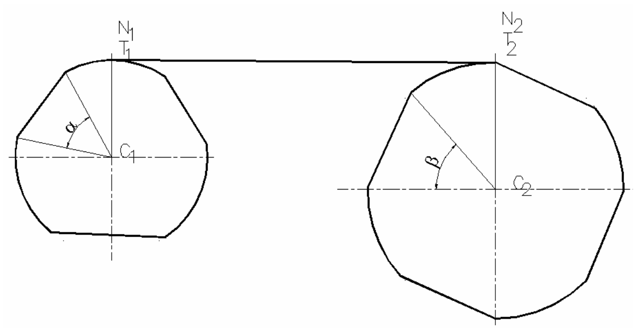

Fig. 2. Belt drive model with three and four teeth pulleys

Examination of Fig. 2 shows that the transmitted motion between these pulleys at any instant depends on their size, shape and relative position. For this position of pulleys, the angular speed ratio can be write as

$$\frac{\omega_2}{\omega_1} = \frac{C_1N_1}{C_2N_2} \quad , \tag{1}$$

where ω_1 and ω_2 are the angular speed of the polygons. For a given configuration (pulley teeth number, pitch value, belt), this ratio doesn't remain constant, but varies in a cyclic manner, with the minimum value when the segments C_1N_1 and C_2N_2 occur together the maximum or the minimum values. To provide this, it is necessary for the span length N_1N_2 to be a multiple n of the tooth pitch p_b:

$$(C_1C_2)^2 = n^2 p_b{}^2 + \left[R_1 \cos(\alpha/2) - R_2 \cos(\beta/2)\right]^2 , \tag{2}$$

where R_1 and R_2 are the radius of the two pulleys (Firbank, 1977).

3. POLYGONAL EFFECT

The belt running manner on the wheel causes a variation of the instantaneous wrapping radius, so tangential speed (Dreucean, 1998), (Firbank, 1977), (Nitulescu, 1999).

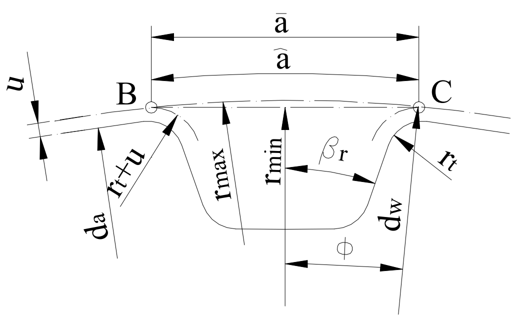

Fig. 3. Variation of the instantaneous radius

Extreme values of radius $r_{\max}$ and $r_{\min}$ can be written as follows (Fig. 3):

$$r_{\max} = \left(\frac{d_a}{2} + u\right); \qquad r_{\min} = \left(\frac{d_a}{2} + u\right)\cos\phi , \tag{3}$$

where d_a is the external diameter of the belt wheel; u - the distance from the fiber to the foot tooth; Φ - central angle equivalent segment BC / 2 (see Fig. 3). These relationships can be applied only through synchronous belts drive with the gap between the bottom tooth wheel and belt tooth (RPP, HTD, HPR, XL, L, H, XH). For a belt drive, speed ratio can be calculated, neglecting the variation of pulleys diameter, so

$$i = \frac{n_1}{n_2} = \frac{\omega_1}{\omega_2} = \frac{z_2}{z_1} \tag{4}$$

with the following extremes values (Butnariu, 2009), (Nitulescu, 1999):

$$i_{\max} = \frac{\omega_{\max}}{\omega_{\min}}, \quad i_{\min} = \frac{\omega_{\min}}{\omega_{\max}}$$

$$i_{\max} = \frac{r_{2\max}}{r_{1\min}} = \frac{\dfrac{d_{a2}}{2} + u}{\left(\dfrac{d_{a1}}{2} + u\right)\cos\phi_1}, \qquad i_{\min} = \frac{r_{2\min}}{r_{1\max}} = \frac{\left(\dfrac{d_{a2}}{2} + u\right)\cos\phi_2}{\dfrac{d_{a1}}{2} + u}. \tag{5}$$

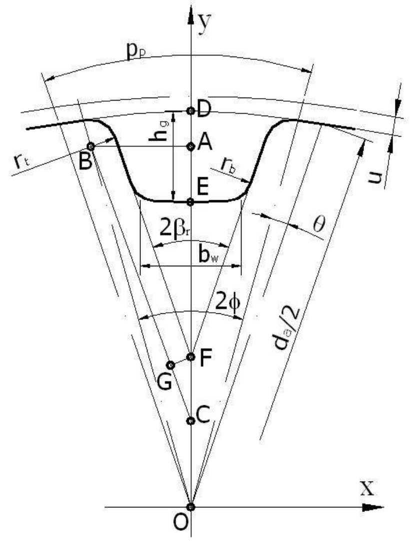

Fig. 4. Geometrical schema of pulley tooth

In Fig. 4, considering the coordinates of point B (XB and YB), the following relationship can determines the angle Φ

$$\phi = arctg \frac{|x_B|}{y_B},$$

where

$$y_B = \frac{btg^2\beta_r + \sqrt{b^2 tg^4\beta_r - (1+tg^2\beta_r)(b^2 tg^2\beta_r - a}}{1+tg^2\beta_r} \quad (6)$$

$$x_B = -(y_B - b)tg\beta_r,$$

and the variation of wrapping radius is (Butnariu, 2009):

$$\Delta r = r_{\max} - r_{\min} = \left(\frac{d_a}{2} + u\right)(1 - \cos\phi). \quad (7)$$

4. TRANSMISSION PRECISION

Synchronous transmission of movement is the most important role that of synchronous belts. Synchronous belts drives are used frequently in recent years, machines and mechanisms require synchronous movement of high-precision (robotics, office supplies, and internal combustion engines). An important role in the transmission of movement through a synchronous belt has belt geometry and wheels, but especially loading transmission. Pitch deviation between belt and pulley exercise an influence far less significant than gear drive due to the large number of belt teeth that are simultaneously in contact, and a flexibility of teeth. Transmission precision of movement is characterized by deviation angle of rotation $\Delta\varphi$ (for belts drive called synchronism error). This is a difference between angular position of driving and driven wheels. Because of the multitude of factors that influence the synchronism error, it is difficult to define a mathematical description of it. The main causes of error of synchronism are operating conditions and geometric features. A summary of these factors is shown in Fig. 5 (Butnariu, 2009), (Nitulescu, 1999).

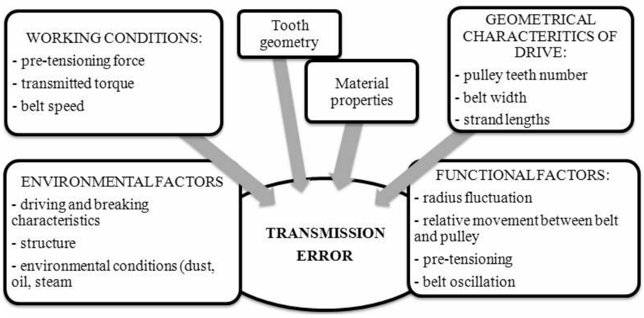

Fig. 5. The main causes of transmission error

In literature, the synchronism error it has been studied taking into account some causes, but neglecting others, depending on the perceived importance. Typically, in most cases, the following factors were taken into account: (i) variations of rolling radius, defined as the distance between the rotation axis and cordage median line resistance at the entrance and exit of the branch gearing assets; (ii) elongation strap on tight span; (iii) relative motion between the belt and pulley (Play, 1994).

5. EXPERIMENTAL DETERMINATION OF THE SYNCHRONISM ERROR

For practical determination of the synchronous error, presented below, a test stand from the Department of Product Design and Robotics, Transilvania University of Brasov was used. This stand is designed so that it can be used for testing different types of transmission: by belts, gears, chain, friction and couplings for testing (Fig.6). The stand is composed primarily of two identical modules that can operate independently, as follows:

I. module consisting of SIEMENS asynchronous motor, 3 kW of power, which has torque output measured with a torsion transducer Hottinger T20 (driver unit);

II. a second module, the same, which can act as brakes, which is identical to first module and can act as a brake as well as measure transmitted torque;

III. automation, control and registration system - a personal computer equipped with controllers for different sensors and transducers.

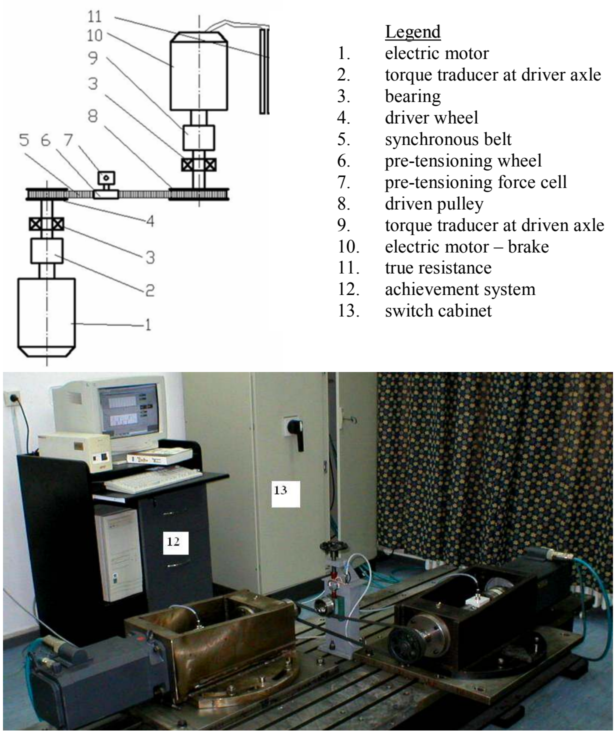

Fig. 6. General view of the test device

A synchronous belt drive used at FIAT 1300 cm^3 engines was positioned between the shafts of the test stand. This is composed by: (1) L type synchronous belt, trapezoidal tooth profile, pitch 9,525 mm, number of teeth 144, width 18 mm, code 540 L 075; (2) driver wheel - number of teeth 21, pitch 9,525 mm; (3) driven wheel - number of teeth 42, pitch 9,525 mm; (4) pre-tensioning device.

The following data was measured: (i) instantaneous speed of rotation of the driver and driven shaft; (ii) couple for driver and driven shaft; (iii) wheel position from a starting position; (iv) pre-tensioning force. The parameters which were varied during testing are: (a) pre-tensioning force; (b) motor torque; (c) braking torque; (d) rotation speed of driver shaft. Experimental measures may be made by modifying the parameters: motor torque, braking torque, engine speed, tensioning force. In literature (Monternot, 1998), (Nitulescu, 1999), it is noted that the synchronism error depends on belt pitch, a section of cordage and variation cordage rigidity. The mounted drive, used in FIAT engines, runs between 600 and 4000 rpm, driving the camshaft and the water pump. The maximum power transmitted by this type of belt is 1 kW, at 6000 rpm, which corresponds to a maximum belt tension of 3000 N. Under these assumptions, the limits for the test parameters are established to: speed: 0 ... 3000 rpm; braking couple: 0 ... 90 Nm; pre-tensioning force 50 ... 250 N.

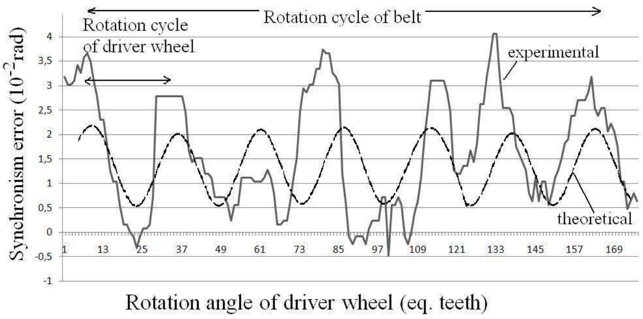

Fig. 7. Variation of the kinematic transmission error

The synchronism error is presented graphically in Fig. 7, for a particular case of a simple transmission consisting of two toothed wheel, tensioning device and synchronous belt and the following parameters: pre-tensioning force F_t = 90 N, driver wheel speed n_1 = 300 rot/min, motor couple M_m = 22 Nm, braking couple M_b = 0 Nm. If the operating parameters are changed, a periodic variation of phase difference $\Delta\varphi$ regardless of the contact conditions (useful force, the pre-tensioning force). This change represents the effect of imperfections in belt manufacturing. On the other hand, it is visible that the synchronism error can be clearly distinguished depending on the full revolution of the driver wheel, caused by imperfections of transmission: wheels eccentricity, shaft deformation or backlash in bearings. This error is repeated periodically at an interval equal to the full revolution of the belt.

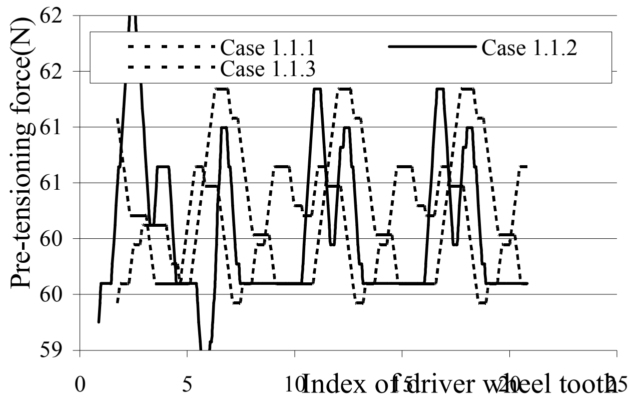

Fig. 8. Variation of pre-tensioning force depending of transmission torque

According to the graph of Fig. 8, there is variation in passive branch tension force in the operation. Transmission loading cases are: pre-tensioning force: 60 N; driver wheel speed: 30 rot/min; transmission torque: 5,5 Nm (case 1.1.1), 11 Nm (case 1.1.2), 22 Nm (case 1.1.3). This may be due to several factors: uneven movement of the belt required gear teeth, unevenness of the outer layer of the belt (extradited) and elasticity trees. The trend is to decrease the pre-tensioning force of passive branch with increasing torque.

6. CONCLUSION

The experimental determinations were used to establish the dependence of synchronism error on the forces of the belt drive (pre-tensioning force, torque).

There is a difference between the theoretical and experimental results (Fig. 7). This is explained by the precision and the rigidity of the equipment used for tests.

The results of the research have confirmed that the synchronism error depends on: fabrication error of elements of the belt drive; pitch deviation between belt and wheel; kinematic characteristics of the belt drive; system forces of the belt drive.

Further research is required in this area, to investigate conditions to improve belt drives and reduce possible errors, focusing on the use of specialized software.

The experimental stand will still be used to identify new ways of investigating the behavior of synchronous belt drives.

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