From transmission error measurement to Pulley-Belt slip determination in serpentine belt drives: influence of tensioner and belt characteristics

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Abstract—*Serpentine belt drives are often used in front end accessory drive of automotive engine. The accessories resistant torques are getting higher within new technological innovations as stater-alternator, and belt transmissions are always asked for higher capacity. Two kind of tensioners are used to maintain minimum tension that insure power transmission and minimize slip: dry friction or hydraulic tensioners. An experimental device and a specific transmission error measurement method have been used to evaluate the performances of a generic transmission by determining the pulley-belt slip for these two kinds of tensioner. A data acquisition technique using optical encoders and based on the angular sampling method is used with success for the first time on a non-synchronous belt transmission. Transmission error between pulleys, pulley/belt slip are deduced from pulley rotation angle measurements.Results obtained show that: the use of tensioner limits belt slip on pulleys, pulley-belt slip is reachable from transmission error measurement, belt non uniform characteristics are responsible of low frequency modulations of transmission error.*

Keywords: belt transmission, transmission error, pulley-belt slip, tensioner

I. Introduction

Multi ribbed (serpentine) belt transmission are often used in the industry machinery to transmit power. Another field of application concerns the Front End Accessory Drive of automotive engine, it recently asks for better performances in order to allow the development of new technological innovations as starter-alternator integration in the transmission. Resistant torques of the driven accessories are getting higher and design solutions have to be found in order to prevent pulley-belt slip and belt fatigue. Because of ,serpentine belt constitution based on cords and rubber, long belt spans and operating conditions (engine acyclism, fluctuating torques), the following phenomena occur during FEAD operation: longitudinal and transverse belt span vibrations, pulley-belt slip. In these transmissions, two kinds of tensioners can be used in order to reduce vibration and insure sufficient belt tension to transmit power and to avoid slipping. They differ in the way damping is generated: one is called dry friction tensioner, the other uses a hydraulic damper(figure 1). Both are equipped with a spring (compression or torsion) . Many works have conducted numerical analysis on the dynamics of serpentine belt drives [4], [1], [2], [3], [5], [10] , they aimed at determining belt tension fluctuations, pulley rotations, belt longitudinal and transverse vibrations. Pulley-belt slip has been investigated both numerically and experimentally by Gerbert [9] on v-belt in steady state operating conditions. Transmission error of belt drives has been studied for synchronous belt [12], [11]. The work presented here is an experimental analysis that aims at determining pulley-belt slip from the transmission error measurement in the case of a multi ribbed belt generic transmission. A test bench has been designed where pulley rotation angles are determined from optical encoder signals using a specific data acquisition system and process[7]. Angular sampling method is used for the transmission error analysis. Such a method is applied here for the first time on a non-discrete geometry.

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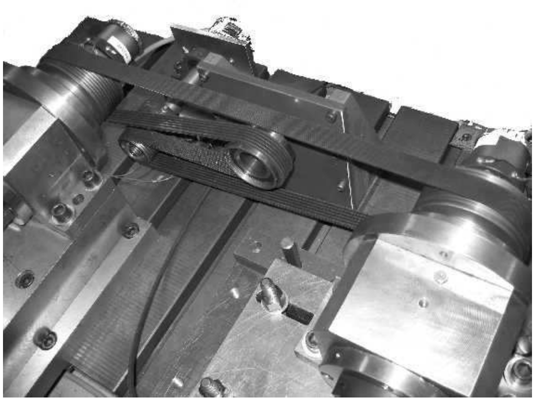
(a) (b)

Fig. 1. Tensioner types: dry friction(a) and hydraulic (b).

II. Experimental set up

A. Belt drive description

The studied transmission consists of four pulleys linked together by an automotive multi-ribbed belt, as shown in Fig. 2. The input shaft speed (from 0 to 2000 rpm) is controlled by a 60 kW electric motor . The driven shaft is connected to a hydraulic pump. The output pressure of the fluid is controlled to apply a mean torque on the driven pulley. Due to the rotation direction, the upper span is tight and the lower one is slack. Three configurations of transmission are studied depending on the type of pulley 4, Fig. 3 : idler pulley, dry friction tensioner or hydraulic tensioner. The objectives are to show the influence of the tensioners on the pulley-belt slip.



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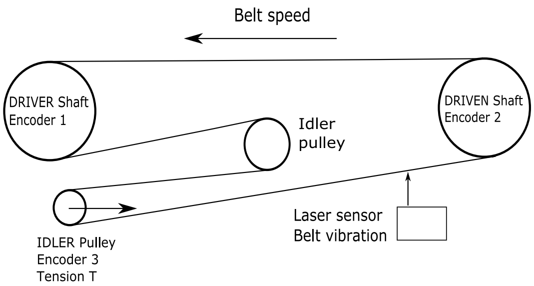

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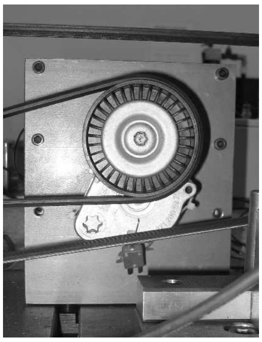
Fig. 2. Multi ribbed belt generic transmission set-up.



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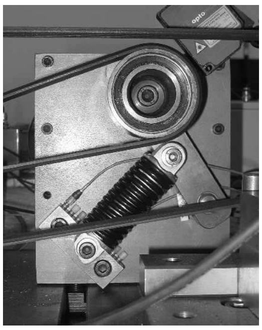


(b)

Fig. 3. Dry friction (a) and hydraulic (b) tensioner assemblies on the transmission.

B. Measurement devices

Angular positions are measured by optical encoders mounted on pulleys 1, 2 and 3 (respectively, 2048, 2048 and 2500 pulses/rev).Belt tension is measured by a piezo-electric sensor on the pulley 3 support, driving and driven torques are measured by means of strain gauges. The data acquisition system is custom made with a PXI frame including classical data acquisition boards and a four-channel counter board permitting the use of the pulse timing method [8]. Each optical encoder delivers a square signal (TTL) as it rotates. Between two rising edges of this signal, a counter records the number of pulses given by a high frequency clock ($80\ MHz$), see Fig.4. For each encoder, it is therefore possible to build a time vector that contains the times of occurrence of the TTL signal's rising edges. Hence, the total rotation angle of each shaft is determined and instantaneous rotation speed and acceleration are deduced. In this application, measurement is triggered on the reference encoder mounted on the driving shaft and analog signals are acquired at each instant of the reference encoder's rising edge. An important characteristic of this measurement principle is to separate resolution and precision. Resolution is given by the number of pulses per revolution, and the theoretical angular precision is proportional to the ratio between rotation speed and counter clock frequency. The grating quality of the optical encoder disk, as well as the electronic signal conditioning and processing may also affect the practical accuracy.

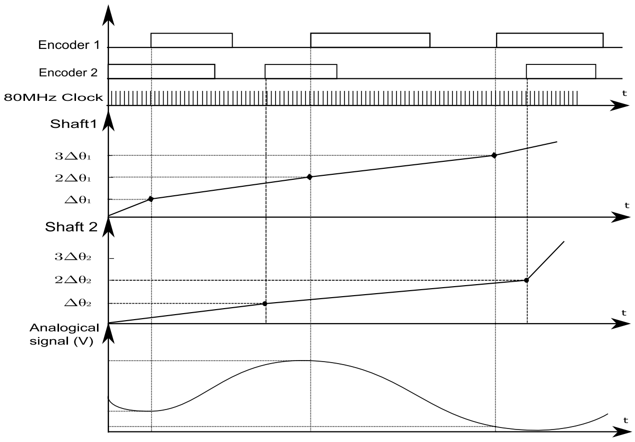


Fig. 4. Pulley angular rotations measurement principle

C. Angular sampling method

The data are re-sampled based on the angular rotation of a chosen encoder which is not necessarily the reference one. It consists in calculating the angular rotations of the other encoders at the times corresponding to the rising edges of the sampling encoder. Hence, if angular sampling is performed on encoder i, the angular positions of each of the slave encoders are computed from linear interpolation at the times corresponding to the encoder i rising edge locations, see Fig. 5. For the analog signals, the same method is applied and they are recorded at the angular frequency of the reference encoder. This method is called angular sampling and is detailed in [7]. It is mainly applied in rotating machines with synchronous transmission elements such as gears or timing belts. Its application to a transmission in the presence of belt slip is novel and provides important advantages as described below. This technique is especially useful for systems with variable speed because the position of the sampling points and the angular resolution remain exactly the same when the speed fluctuates.

The main advantages of performing angular sampling here are:
- sampling points are exactly located in reference to the geometry of the rotating machine, even when speed varies. It permits to compare several measurement results based on the same sampling conditions,
- by choosing the encoder 3 as reference, and assuming that no slip occurs between belt and the idler pulley since no torque is being transmitted, the sampling points are attached to the belt.

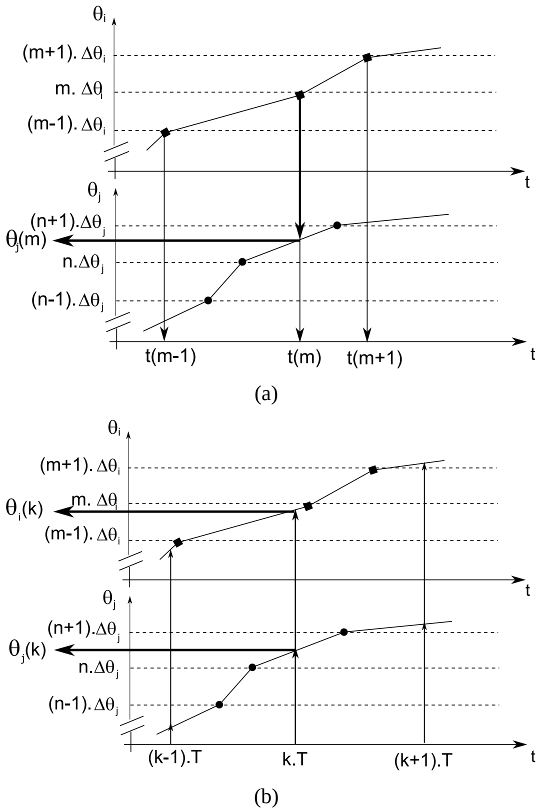


Fig. 5. Angular re-sampling method(a) and Time re-sampling method (b).

For standard analysis, it is necessary to get the measurements as a function of one single time vector with equally spaced intervals. This requires a time re-sampling of the data using linear interpolation as shown in Fig. 5(b).

III. Pulley - belt slip analysis

A. From transmission error to slip rate

This angular sampling method has already been used for many synchronous transmission studies (gearbox, timing belt drive) but never for non synchronous transmissions such as serpentine multi-ribbed belt drives. The transmission error ϵ is defined as the angular rotation difference between shaft i and shaft j,

$$\epsilon = \theta_i - \eta \cdot \theta_j \tag{1}$$

where η and $\theta_{i,j}$ are respectively the transmission ratio and the angular positions of shaft $i,\ j$.

In the case of non-synchronous belt drive systems, some creep occurs between the belt and the pulleys due to the power transmission by friction [9]. Indeed, the creep corresponds to the relative slip between the belt and the driven pulley as the belt elongates on the pulley contact arc as its tension increases. Here, we consider the transmission error between pulleys 3 and 2. Pulley 3 is not loaded and pulley 2 has the hydraulic pump resistant torque applied. The rotation of pulley 3 is not totally transmitted to pulley 2 due to the belt stretching on pulley 2 which causes a delay. Therefore, the mean value of the transmission error is not zero as it is for a synchronous drive, but rather always increases (Fig.6). In our application, analysis permits decomposition of the observed transmission error as the sum of a linear function of time representing the transmission error due to pulley belt creep ϵ_{creep}, and the residual transmission error ϵ_{res} due to belt span elasticity characteristics as in synchronous transmission.

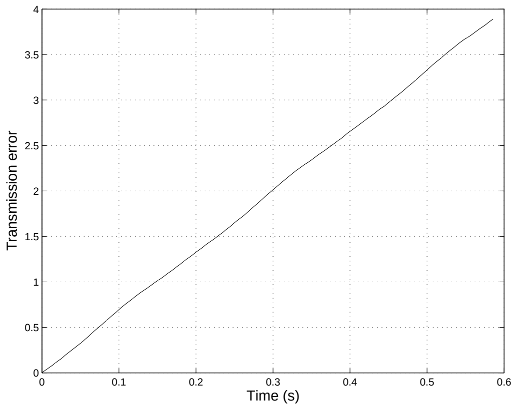
$$\epsilon = \theta_3 - \eta \cdot \theta_2 = \epsilon_{res} + \epsilon_{creep} \tag{2}$$

where ϵ_{creep} is identified from ϵ as a linear regression of time assuming a constant mean rotation speed Eq. 3. Hence the slope a represents the slip rate and is expressed in (rad/s). Removing the linear part ϵ_{creep} from the transmission error ϵ yields the zero-mean periodic residual transmission error ϵ_{res} (Fig. 6). The pulley-belt slip has been quantified using the slip rate factor for the three transmission possible configurations where pulley can be fixed (idler), a dry friction or hydraulic tensioner and for several operating conditions of speed, torque, and initial belt tension.

$$\epsilon_{creep} = a * t + b \tag{3}$$

B. Results

The pulley-belt slip is observed via the slip rate that has been determined for several tests. Experiments were done for two different initial belt tensions combined with two driving speeds, and several resistant torques applied by the hydraulic pump connected to pulley 2.
It can be seen in figure 7 that the pulley-belt slip increases with the torque for the three transmission configurations. The use of a tensioner instead of an idler for pulley 4 reduces the slip rate of 50 percent. The hydraulic tensioner seems to be a little more efficient than the dry friction one, but their effects are almost identical.



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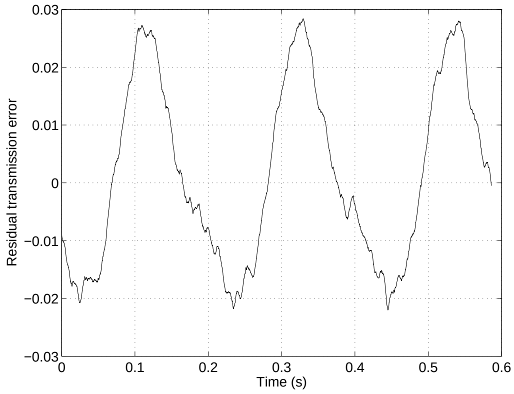

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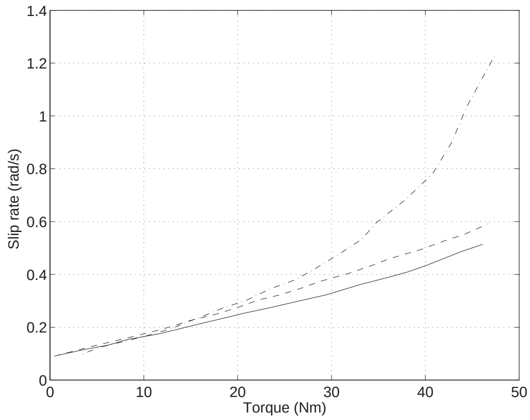
Fig. 6. Total (a) and residual (b) transmission error versus time.



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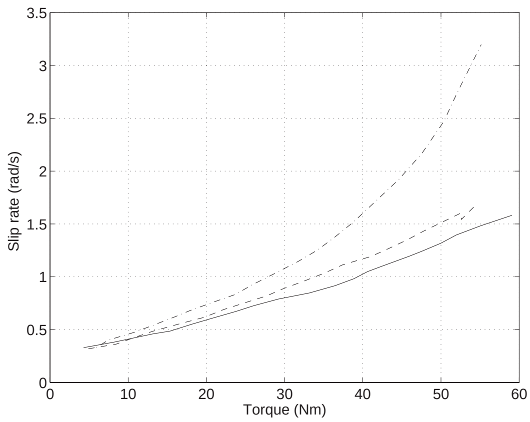


(b)

Fig. 7. Slip rate when pulley 4 is an idler (-.-), dry friction (–) or hydraulic (-) tensioner, at driving speeds (a) 280 rpm and (b) 840 rpm.

The belt setting or initial tension permits the power transmission, in practical applications it is adjusted to avoid pulley-belt slip. The contact conditions at the belt-pulley interface and therefore the pulley-belt slip depend on this initial tension, the torque transmitted and also the belt velocity. The experimental results presented on figure 8 illustrate that dependency. When a tensioner replaces pulley 4, the initial tension has no more influence on the pulley-belt slip and the slip rate increases with the torque.

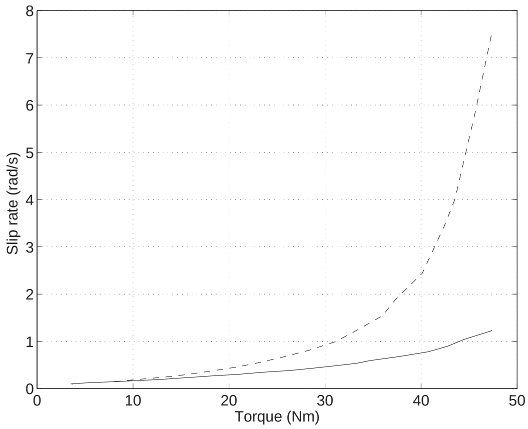
C. *Transmission error related to non uniform belt characteristics*

The low modulation observed on the dynamic transmission error (Fig. 6) corresponds to the belt traveling frequency and demonstrates there are non-uniform belt characteristics. In order to check this non-uniformity, a belt has been cut in ten equal parts. Each part has been tested to determine longitudinal rigidity modulus k and damping C. Each belt sample is clamped at one end and has a mass m suspended at the other, see Fig. 10. This system is excited via a shock hammer. The free response is recorded via an accelerometer and post-processed to obtain belt longitudinal stiffness and damping. Longitudinal damping coefficients of belt samples, normed by the maximum measured value, are plotted versus belt sample number in Fig. 11. Non negligible variation is observed for C while local stiffness and therefore EA is almost constant.

This irregularity, probably due to the manufacturing process (printing, cord winding, cutting), generates transmission error fluctuations at the belt revolution frequency.

IV. Conclusion

This study has presented an original way to determine pulley-belt slip from the transmission error measurement in a serpentine belt drive. The measurement process is based on the use of optical encoders coupled with the angular sampling method, its application to a non synchronous transmissions is novel. The transmission error in such system can be decomposed in two components:
- the creep transmission error which is increasing linearly and that represents the pulley-belt creep due to torque transmission, its derivative named the slip rate has been used to quantify the pulley-belt slip,
- a residual zero mean periodic transmission error which



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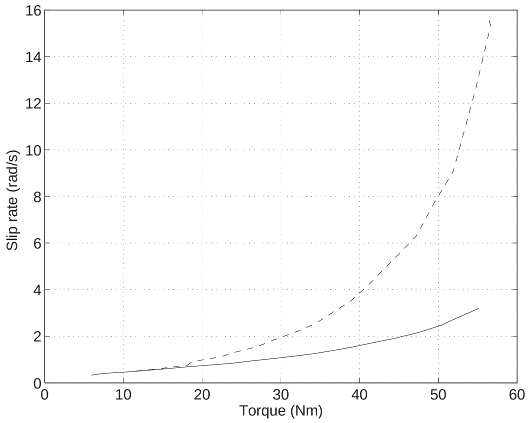


(b)

Fig. 8. Slip rate, pulley 4 fixed, initial tension 111 N(–) and 274 N (-), driving speed (a) 280 rpm and (b) 840 rpm.

main harmonic corresponds to the belt revolution frequency and is the consequence of longitudinal belt characteristic irregularity.

The pulley belt-slip observed in the studied multi ribbed belt serpentine drive is shown to be largely influenced by the presence of a tensioner that reduces it, it also increases with the transmitted torque . Large differences have been observed experimentally between the used of a tensioner instead of an idler. No significant difference is observed between the uses of dry friction or hydraulic tensioners regarding the slip rate.

References

[1] Fan G.W., Kraver T.C. and Shah J.J, Complex modal analysis of a flat belt pulley system with belt damping and coulomb-damped tensioner, *Journal of Mechanical Design*, ASME, vol. 118, 306-311, 1996.

[2] Leamy M.J., Perkins N.C., Nonlinear Periodic Response of Engine Accessory Drives With Dry Friction Tensioners *Journal of Vibration and Acoustics*, ASME, vol. 120, 909-916, 1998.

[3] Leamy M.J. and Wasfy T.M., Analysis of belt-drive mechanics using a creeprate- dependent friction law *Journal of Applied Mechanics*, ASME, vol. 69, 763-771, 2002.

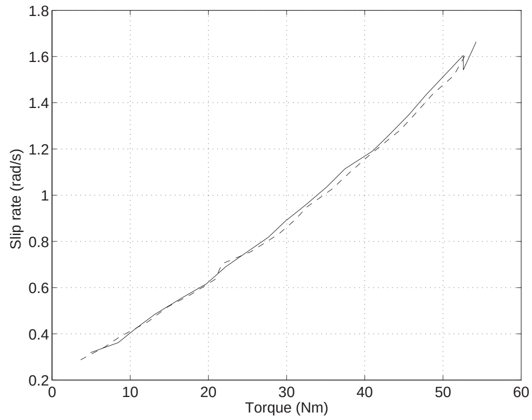
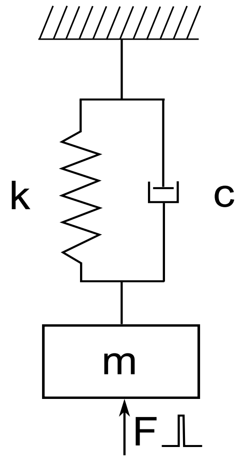
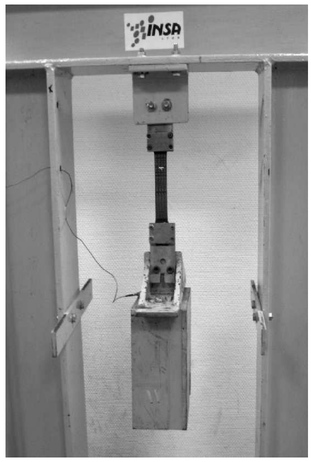


Fig. 9. Slip rate, with dry friction tensioner, Initial tension = 380 N (-.-) and 510 N (-) at 840 rpm.





(a) (b)

Fig. 10. Experimental set-up for the local belt characteristics identification.

[4] Cheng G. and Zu J.W. Nonstick and stick-slip motion of a coulomb-damped belt drive system subjected to multifrequency excitations, *Journal of Applied Mechanics*, ASME, vol. 70, pages 871-884, 2003.

[5] Ulsoy A.G., Whitesell J.E. and Hooven M.D., Design of belt-tensioner system for dynamic stability, *Journal of Vibration, Acoustics, Stress and Reliability in Design*, ASME, vol. 107, 282-290, 1985.

[6] Houser D.R. and Blankenshio G.W., Methods for measuring transmission error under load and at operating speeds, *SAE Transactions*, vol. 98, nb 6, 1367-1374 , 1989.

[7] Remond D. and Mahfoud J. From transmission error measurements to angular sampling in rotating machines with discrete geometry, *Shock and Vibration*, IOS Press., Vol. 9, 1-13, 2005.

[8] Remond D., Practical Performances of High-speed Measurement of Gear Transmission Error or Torsional Vibrations with Optical Encoders, *Journal Measurement Science and Technology*, vol. 3, 347-353, 1998.

[9] Gerbert G., Traction Belt Mechanics, MVD, Chalmers 1999, 600p.

[10] Kong L. and Parker R., Mechanics of serpentine belt drives with tensioner assemblies and belt bending stiffness, *Journal of Mechanical Design*, ASME, vol. 127, 957-966, 2005.

[11] Makita K., Kagotani M., Ueda H. and Koyama T., Transmission Error in Synchronous Belt Drives With Idler (Influence of Thickness

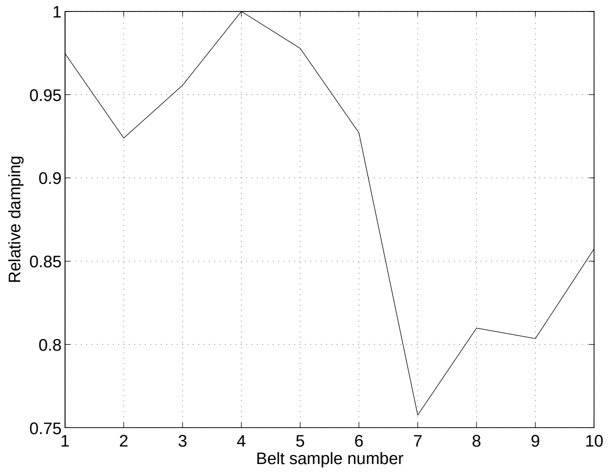


Fig. 11. dimensionless damping evolution as a function of belt sample number C_i/C_{max}.

Error of Belt Back Face Under No Load Conditions), *Journal of Mechanical Design*, ASME, vol. 126, 148-155, 2004.

[12] Makita K., Kagotani M., Ueda H. and Koyama T., Influence of Idler on Transmission Error in Synchronous Belt Drives (Under Transmission Force), *Journal of Mechanical Design*, ASME, vol. 125, 404-410, 2003.