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The experimental analysis of the slip in the rubber belt CVT

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Abstract. This work deals with the analysis of the speed losses in CVT. The bench tests have been conducted on the scooter CVT equipped with the centrifugal regulation system. This solution is typical for this type of vehicles so the conducted experiments refer to real exploitation conditions. The slip has been defined on the base of the difference between speed ratios obtained from the angular speeds and the belt pitch radii. This approach corresponds with the Dittich model. The non-linear dependence between the slip and the transmitted torque has been obtained for the constant gear ratio. Also non-linear dependence between the slip and the gear ratio has been received for constant torque. The amount of slip value indicates that this is significant part of the total power losses as it has been described by Bertini. However it clashes with the Chen researches, where the slip corresponds with the marginal part of the overall losses.

1. Introduction

The advantages of the V-belt transmission cause this type of power train to be employed in machine drive systems from many years. The elaboration of the solutions which enable the step less gear ratio change (CVT) has contributed to the spread this type of transmission in vehicles – especially in the scooters and the snow vehicles.

A lot of works published recently concern the efficiency issue in the rubber belt continuously variable transmission. In this case the power loss mechanisms may be divided into two areas: torque losses and speed losses. The transmitted power (P) can be expressed as:

$$P = M\omega \quad (1)$$

where M is the torque transmitted by considered pulley while ω is its angular velocity. Consequently the transmission efficiency is represented as:

$$\eta = \frac{M_n \omega_n}{M_g \omega_g}, \quad (2)$$

where subscripts (_n) and (_g) denote driven and driving pulley respectively. The fact that both M and ω are variable over time, differentiation of the above equation leads to the relation:

$$dP = d(M\omega) = dM\omega + Md\omega \quad (3)$$



Results of (1) and (3) equations show that the power losses are the sum of the torque losses and speed losses:

$$\frac{dP}{P} = \frac{dM}{M} + \frac{d\omega}{\omega}. \quad (4)$$

The first category associated with the internal friction in the belt deals with bending and compression hysteresis of the belt, its wedge-in and wedge-out at entering and exit from the pulley. The second category associated with the external friction results from the belt extension and its flexural rigidity which leads to the wrap angle decreasing and the contact area deformation. According to the research results presented in work [2] the transmission efficiency primarily depend upon the torque losses. The belt slip corresponds with only a few percent of the total power loss. This dependence is illustrated in figure1 showing changes of the CVT efficiency, torque and speed losses against the load at the constant engine speed.

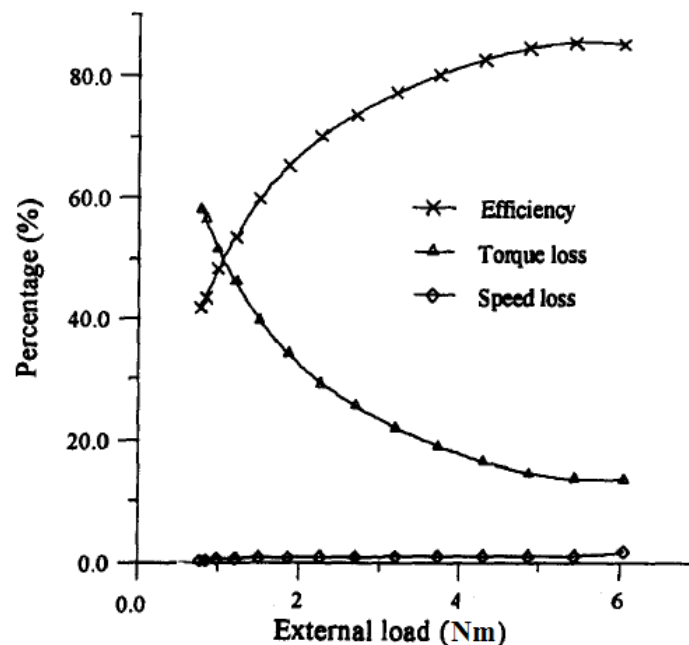


Figure 1. The CVT efficiency against load at constant engine speed (3600rpm) [2].

On the other hand the analytical power losses model in stationary conditions described in work [1] shows that the speed losses are essential for the CVT efficiency. Additionally this model has been confirmed experimentally.

The causes of the slip in belt transmissions are not fully explained. The widely accepted rubber belt transmission model proposed by Dittrich [6], assumes the presence on each of the pulleys two cooperation areas: the undeveloped friction zone at the entrance to the bevel gear and the area of kinetic friction at the exit of the wheel. In this perspective, the belt slip corresponds with the developed friction, which is accompanied by a change of the belt tension. The differentiation of the forces in tight and slack belt side causes the unequal belt extension in these sectors, so the speed of the belt needs to be changed. In the other area where the belt tension remains constant slip does not occur. In turn, in the approach described in [3] slip is related to the deflection of the belt due to the shear. Gerbert [4] however has assumed that the difference in the belt speed at the entry and exit of the pulley results from the variable winding radiuses. As a reason of this belt behaviour he has pointed out its compressibility and flexural rigidity.

2. Test stand

The experiment was realized by the test bench included complete drivetrain assembly from the TGB 101S scooter with centrifugal regulator. In order to enable the CVT load in the whole range of the engine revs the centrifugal clutch was eliminated. The output bevel gear assembly was coupled with the water-supplied engine brake (dynamometer) which maximal absorbed power is 75kW. Additionally the steel flywheel coupled with the driven bevel gear too allows the simulation of the vehicle starting conditions. Figure 2 shows the test stand scheme.

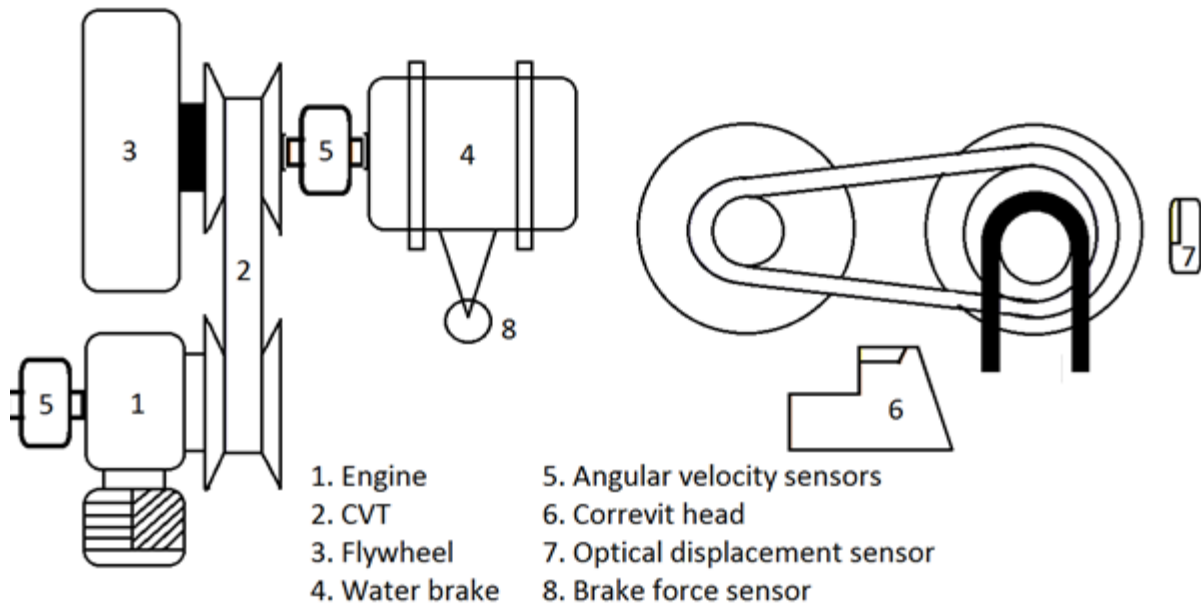


Figure 2. Test stand scheme

The engine torque results in the reaction force transmitted on the engine brake lever with force sensor at its end. The measurement allows for the resistant torque calculation. Furthermore the following quantities have been measured: the angular speed on both the driving and the driven pulleys, the belt velocity and the winding radius on the driven pulley. The belt velocity and winding radius were measured by the optical sensors. The signals from the sensors were recorded by the analog-to-digital transducer connected with the computer. The registration resolution were 100Hz. The driving pulley winding radius was calculated on the base of the belt length and the pulleys centers distance. The tests were conducted with the constant throttle position and variable load as well as the variable throttle position and the constant load.

3. The belt slip

3.1 Slip definition

The torque load transmitted by the CVT causes the differentiation of the forces in the belt. Because of the belt compliance the strain of the tight and slack side will be different. In the effect of the mass conservation rule a tight side belt velocity has to be higher than slack side velocity in a single circuit. Therefore the belt slip can be defined as:

$$S = \frac{v_1 - v_2}{v_1}, \quad (5)$$

where v_1 and v_2 refers to the velocities in tight and slack belt side (figure 3). In the basic theory of belt drives (among others in the Dittrich work [6]), the wrap angle can be divided into the reserve angle and

the active (effective) angle (θ_{ga} and θ_{na} in figure 3). The change of the belt tension, and therefore slip occur in the active part of the wrap arc. In the inactive zone (at the entrance), where there is static friction, the belt velocity equals the circumferential speed of the pulley on the belt radius, hence the equation (5) can be written as:

$$S = \frac{\omega_g r_g - \omega_n r_n}{\omega_g r_g}, \quad (6)$$

where ω_g and ω_n are angular velocities whereas r_g and r_n are winding radiuses for driving ($_g$) and driven pulley ($_n$). Owing to the fact that the radius of the belt cooperation may vary along the arc of wrap, the slip determination by equation (6) requires defining winding radiuses in the area where the belt enters at the pulley.

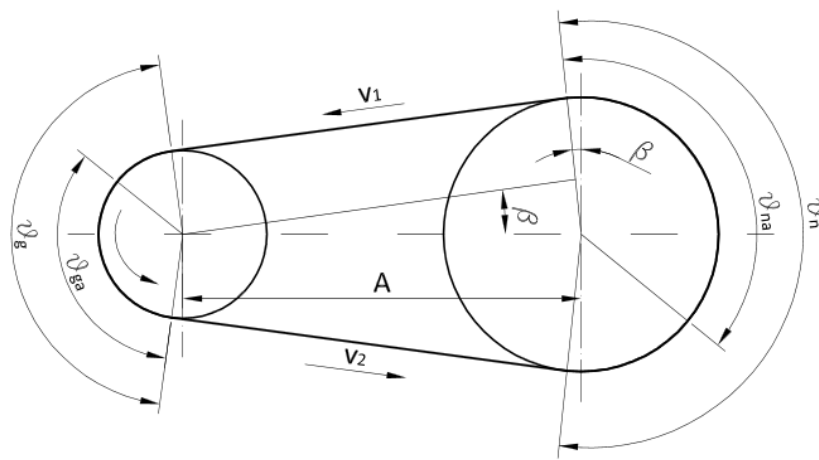


Figure 3. Belt transmission scheme.

The transmission ratio can be specified by the angular velocities ratio or the winding radiuses ratio as follows:

$$i_n = \frac{\omega_g}{\omega_n}, \quad (7)$$

$$i_r = \frac{r_n}{r_g}. \quad (8)$$

Using the expressions (7) and (8) in the equation (6) the belt slip can be finally defined as:

$$S = \frac{i_n - i_r}{i_n}. \quad (9)$$

On the other hand the belt support layer made of a fabric cord prevents excessive belt extension. The experimental research presented in paper [1] shows that the currently utilized CVT rubber belts feature a very high tensile strength. The longitudinal stiffness of the belt reaches a value about 8kN/mm. For the transmission which is the subject of the present research, where the transmitted torque does not exceed 10Nm, it can be assumed that the speed of the belt in a single circuit does not change. Furthermore, widely employed in the literature models of the V-shaped rubber belt transmission [5] indicate the slip lack on the driving pulley. This belt behaviour is explained by the occurrence of the self-locking phenomenon. As a result, the winding radius and the belt tension on input pulley does not change. Accordingly, the total slip may be determined based on the difference in belt speed and the circumferential velocity of the driven wheel in the radius of the belt:

$$S' = \frac{v_l - \omega_n r_n}{v_l}, \quad (10)$$

where v_l is the measured belt speed.

As part of the examination the attempt of the non-contact belt speed measure was conducted by the Correvit L head. The real results were obtained only for the transmission ratio close to 1. The effective measurement in the whole range of the ratios was not possible because of the change in the belt slope relative to the head and occurred belt vibrations. In result the determination of the slip by equation (10) was not feasible for all the range.

Non-contact measurement of the winding radius does not give information about the location of the neutral axis of the belt. Due to the relatively small winding radii (45 mm) the consideration of this amendment is essential to determine the actual slip. In view of the above assumptions, the value of the distance between the back of the belt and the neutral axis can be determined to obtain no difference between the measured speed of the belt and the circumferential speed of the driving pulley on the diameter of the belt. For the analysed powertrain an average value of this distance, specified for the proper operation of the Correvit head, is about 3mm.

Ultimately the belt slip determined on the basis the equation (9) arises from the difference in the transmission ratios specified by the angular velocity and the winding radiuses taking into account the neutral axis of the belt.

3.2 Analysis of the research results

The samples test results for the two methods of transmission load are presented below. Using them the influence of the load torque (M_n) and the transmission ratio (i_r) on the belt slip is analysed.

The first case concerns the increasing load at a constant, full throttle. As it is shown in figure 4 and 5 for the unload transmission working with minimal ratio the belt slip is about 9%. After more or less 5s. the engine speed decreasing is initiated due to the increase of load. In the next second the transmission ratio remains unchanged what is accompanied by temporary slip value peak to approx. 13%. The reason for this phenomenon is inertia of the centrifugal regulator. Instantaneous reduction of the engine angular speed entails the reduction of the driving pulley axial force. Increasing of the axial force and the traction ability appears as an effect of the change of the ramp area where the rollers act. This is associated with the transmission ratio growth. The load increase accompanied by the ratio shift leads to the belt slip rise. Figure 6 shows the changes of the slip against the driven pulley torque in the range of 9th to 21th s., when the transmission ratio is maintained at the constant value 2,7. This relationship can be approximated by a second-order polynomial with the coefficient of determination about 0,58. The slip value increases with the load torque reaching the maximum of 18% at the torque about 6Nm. This tendency arises from increasing of the active wrap angle on the driven pulley. The greater load implies the slip reduction, which may be associated with the increased axial force of the torque regulator.

The second case deals with a rapid throttle opening at a fixed load. As it is seen in figure 7 and 8 for the loaded transmission working with the maximum ratio the belt slip is about 15%. Decreasing of the ratio due to the engine speed grow then tails reduction of the slip to the value about 12%. Figure 9 shows the changes of the slip against the transmission ratio in the range of 14th to 30th s., when the load torque is maintained at the constant value about 1Nm. As it was before this dependence is approximated by a second-order polynomial with the coefficient of determination about 0,64. As it can be seen the ratio increasing to the value about 1,6 causes the slip reduction to the minimum value 10%. Further the ratio growth results in the slip increasing. The maximum value in this case is 15% for the ratio 2,5. The slip growth associated with reduction of the driving pulley wrap angle (increasing ratio) follows from additional reduction of the wrap angle on account of the belt flexural rigidity. It is astonishing in this case that the minimum slip corresponds with the ratio about 1,5. The slip increase with the smaller ratios can be explained by reducing of the driven pulley wrap angle due to the belt flexural rigidity.

The experiment results clearly show a substantial influence of the transmission ratio and the transferred torque to the power loss caused by the belt slip. Thus, it confirms the test results on the

The constant full throttle and increasing load

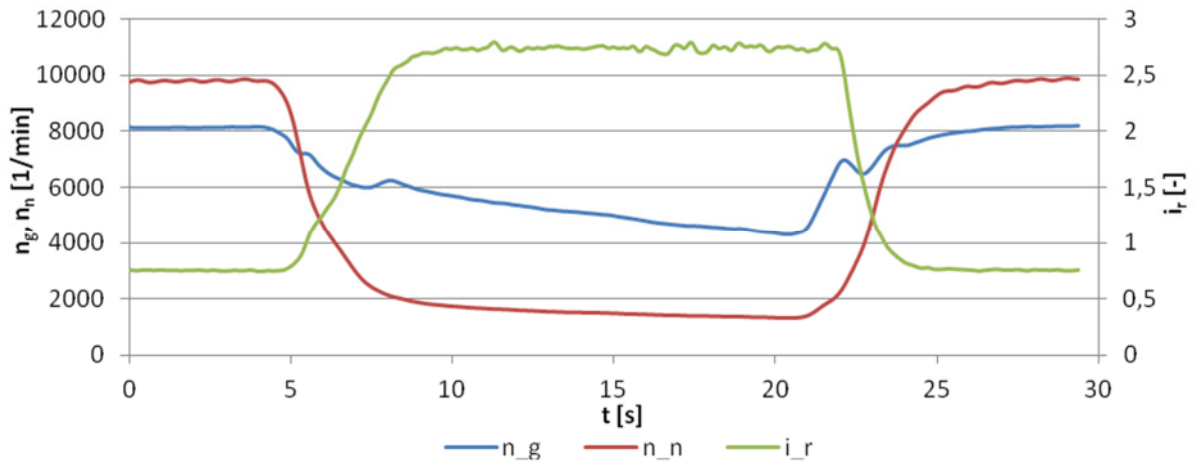


Figure 4. Change of the pulley angular speed (n) and transmission ratio (i_r) against time; subscript(g) – driving pulley, subscript (n) – driven pulley.

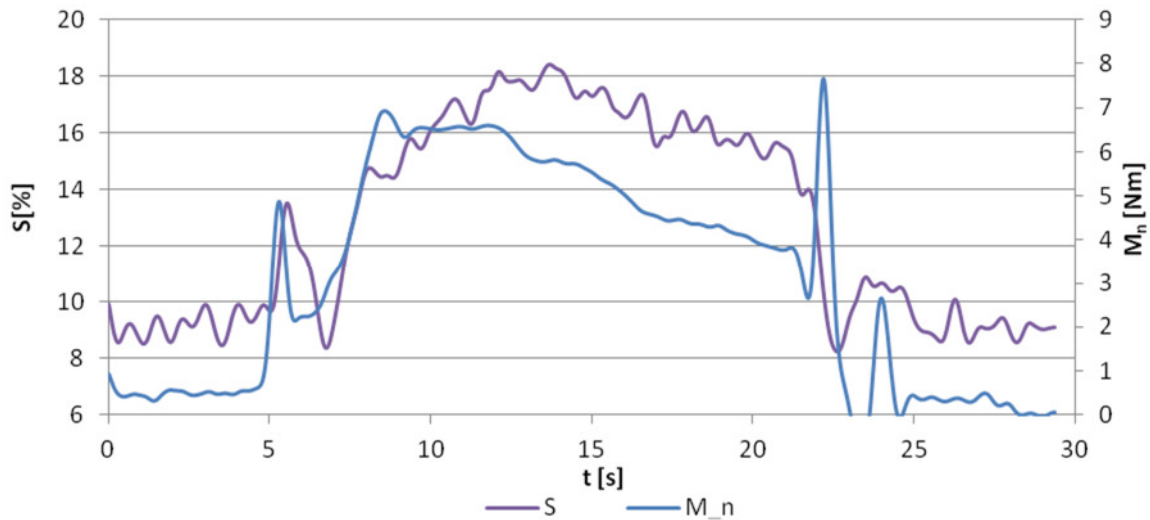


Figure 5. Change of the belt slip and driven pulley torque against time.

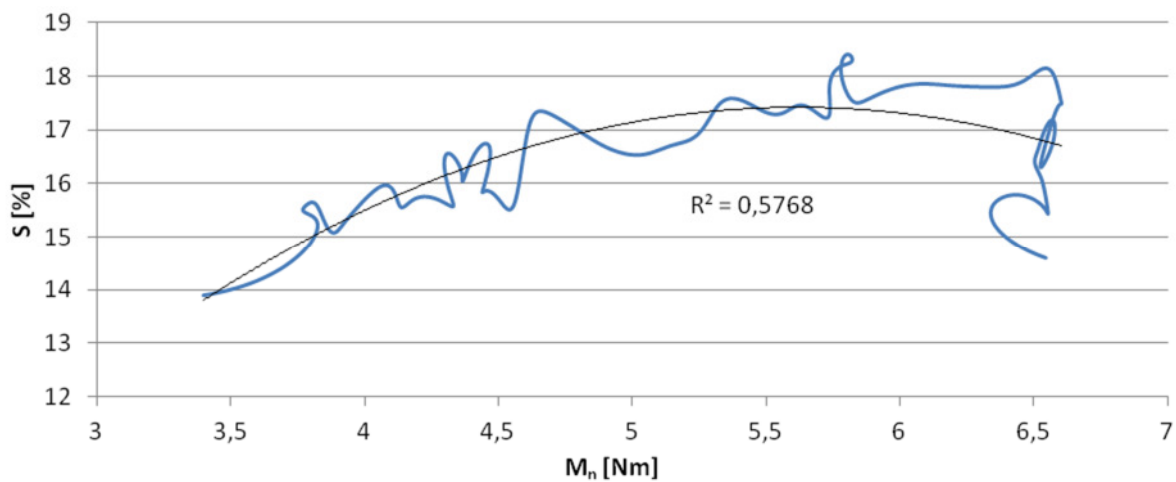


Figure 6. Belt slip against driven pulley torque for $i_r = 2.7$ (range 9th-21th s.).

The constant load and rapid throttle opening

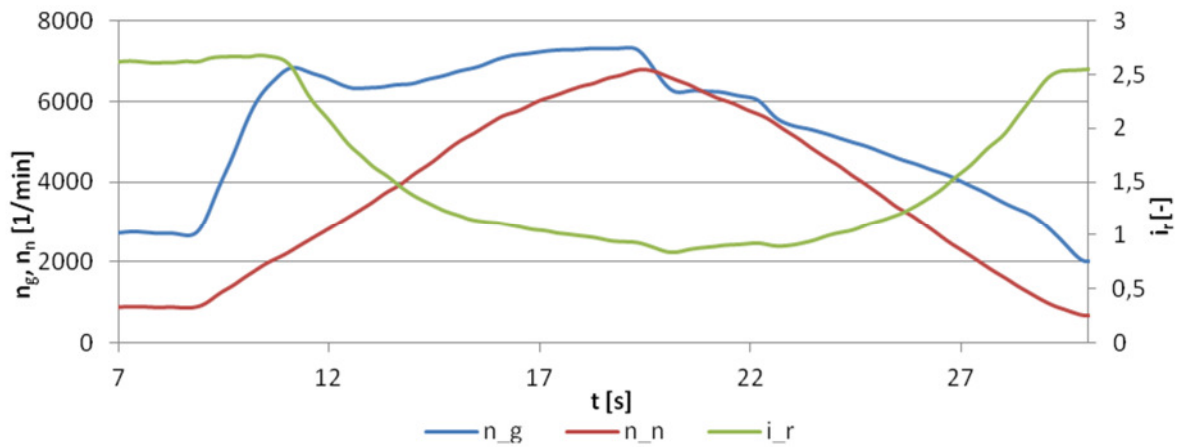


Figure 7. Change of the pulley angular speed (n) and transmission ratio (i_r) against time; subscript(g) – driving pulley, subscript(n) – driven pulley.

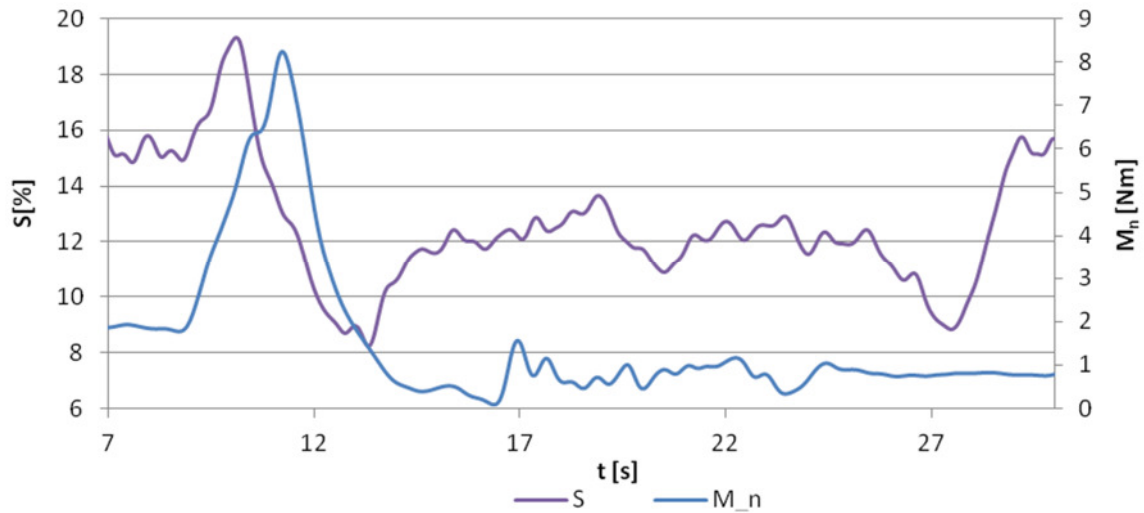


Figure 8. Change of the belt slip and driven pulley torque against time.

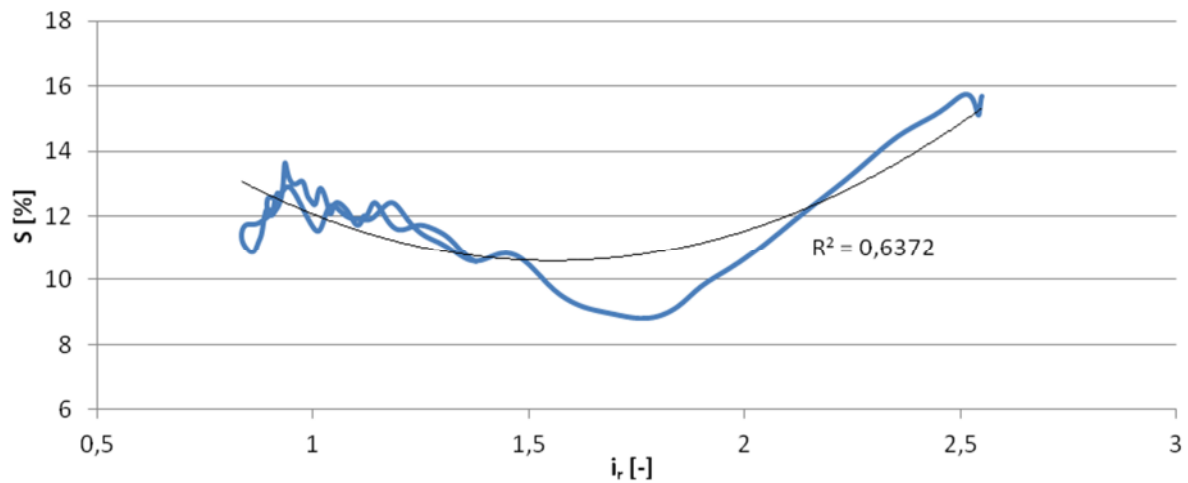


Figure 9. Belt slip against transmission ratio for $M_n \approx 1\text{Nm}$ (range 14th-30th s.).

influence of the load on the speed losses presented in [1], except for the fact that obtained relationship is nonlinear. The power loss resulting from the belt slip doesn't exceed 200W. As part of further research, it seems necessary to measure the belt temperature and to present dependences shown for a wider range of transmission ratios and loads. In each examined case, the slip is between 8% and 18%. Consequently it is a key factor in the efficiency of the CVT. That is why it does not confirm the marginal significance of the slip to the transmitted power, as Chen [2] suggested. Difficulties associated with non-contact measurement of the belt speed allow to determine the slip by the equation (10) only for the transmission ratio close to 1. Within these ranges the slip obtained by equations (9) and (10) reaches a similar value, which confirms the assumed position of the belt neutral axis.

4. Conclusions

1. The belt slip determined in the examined range of parameters reaches values from 8 to 18%, which demonstrates that the speed losses are the main or one of the main factors of the power losses.
2. The momentary belt slip value is the result of the momentary transmission conditions such as the torque and the transmission ratio. The slip appeared to be nonlinearly related to the load torque at the constant ratio. A similar trend takes place for the variable transmission ratio at the fixed torque.
3. Defining the belt neutral axis is significant for the accurate slip determination. Omitting this factor leads to the slip value overestimation.

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