The analysis of an influence of rubber V-belt physical properties on CVT efficiency

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Abstract. This work deals with the analysis of the CVT efficiency for three different rubber V-belts. The bench tests have been conducted on the complete light two-wheeled vehicle drive system so the experiments refer to real exploitation conditions. The research has been performed for fixed gear ratios. The overall efficiency and belt slip have been specified. The longitudinal belt velocity has been measured by the contactless measurement head. The basic belts physical properties have been estimated and significant dissimilarities of their construction details, used materials and stiffness parameters have been revealed. The explicit differences of overall efficiency and belt slip have been observed between tested belts. The V-belt features limiting specified type of losses contributes to increasing other losses at the same time. Furthermore the gear ratio essential influence on the belt slip has been noticeable. 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 research, 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. Two-wheeler and snowmobile are major sections of automotive industry using CVTs. In this application a rubber dry belt is commonly used. Dry belts are usually used because a high friction coefficient is created between a belt and pulleys so that clamping force can be much smaller than it is in lubricated variants. Unfortunately, the problem which appears in non-lubricated belt pulley contact points results from the lack of cooling of this contact, which causes limits of the torque capacity of this type of variator. At present, the automotive market offers wide range of rubber V-belts. The different quality and price of these parts are connected with miscellaneous material and structural properties. The main goal of this work is to show the differences of mentioned belts features and their influence on the transmission efficiency. The belts comparing requires to determine their properties in first step. However the methodology of testing of the rubber strips is not normalized. The examples of this kind of research are included in work [1].
A lot of works published recently concern the efficiency issue in the rubber belt continuously variable transmission [6, 8, 9, 10]. 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:

$$ P = M \omega $$

(1)

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

$$ \eta = \frac{M_n \omega_n}{M_g \omega_g} $$

(2)

where subscripts ($n$) and ($g$) denote driven and driving pulley respectively. The fact that both $M$ and $\omega$ are variable over time, differentiation of the above equation leads to the relation:

$$ dP = d\omega M + dM \omega $$

(3)

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

$$ \frac{dP}{P} = \frac{dM}{M} + \frac{d\omega}{\omega} $$

(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 [3] the transmission efficiency primarily depend upon the torque losses. The belt slip corresponds with only a few percent of the total power loss. On the other hand the analytical power losses model in stationary conditions described in work [2] 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. That’s why the slip measurement entails significant difficulties. Non-contact measurement of the winding radius does not give information about the location of the neutral axis of the belt. This neutral axis could also to change along a wrap arc. Additionally the belt and pulley cooperation line is a spiral section rather than a circle arc. On this way the determination of the circumferential speed of the pulley on the belt radius as a constant value for whole wrap arc is practically impossible.

2. Test stand

The experiment was realized by the test bench contained complete drivetrain assembly from the TGB 101S scooter with blocked centrifugal regulator. The tests were performed for four constant gear ratios. 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 eddy current brake. Figure 1 shows the test stand view.

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 radiuses on the driving and the driven pulleys. 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 tests were conducted with the constant full throttle position and the powertrain angular velocity characteristics were performed. The testes were performed in quasi-static conditions. The same research was carried out for each one from three belts. The examined belts were varied in terms of their quality and price.
3. **The belts characteristic**

The examined belts are shown in figure 2, where A is the Dayco belt, B is the Malossi belt and C is the Tecnium belt. The belts are the same size 765x17,5 (length x width in mm). In each case general construction is similar. The strips consists of a rubber core with vulcanized ropes of the cord and a fabric cover. The first difference deals with the cogs shape. As it is seen in figure 2, the A and B cogs are definitely more circular than the C cogs which are rather trapezoidal. The width of the B and C cog is the same but A cog is approximately 1mm narrower. The compositions of the rubber mixtures have not been tested however the differences are noticeable at the level of organoleptic examination. The belts cross sections are shown in figure 3. The special attention should be paid to the fact that the thickness of fabric cover layer over the cord is much bigger for the C belt. It can results in increased flexural rigidity. This thesis is confirmed by figure 2. As it is seen, the C belt the most keeps the shape after tearing.

In figure 4 the results of the friction coefficient measurement between the surface of belt and belt pulley were presented. The measurement were conducted for aluminium pulley (driving) as well as
steel pulley (driven). Mean values were shown in the figure. To quantify the amount of variation of data values, the standard deviation of the sample was applied. As can be noticed, in case of all the studied belts, differences in the value of friction coefficient occur, depending on the place from which the sample was taken. Therefore, for the same belt differences in the friction coefficient occur on its perimeter. Due to the fact that this area of the belt interacts with the surface of the belt pulley, fragments of the cord will also slide on the surface of belt pulley. The sliding friction during the slip of the cord ropes is different than during the slip of the rubber surface.

![Figure 4. Coefficients of friction (COF) for considered belts](image)

The figure 5 contains the results of tensile tests of the strips on the back side. It can be interpreted as the characteristic of the cord stiffness. The differences in stiffness, seen in the figure are caused by arrangement and the initial tension of the cord ropes in particular belts. The tests were conducted for the loads corresponding with real CVT work conditions [4,5,7]. In this range the characteristics have a linear course.

![Figure 5. The belts cord stiffness](image)
The results of tensile tests of the strips on the ‘cogs side’ are shown in figure 6. The examination were performed in conical pulleys. The wrap diameter was the same on both pulleys and referred to its minimal work value. The load range was the same as previously and the characteristics have also a linear course. In this attempt the susceptibility of the rubber core was a key factor. The noticeable differences result mainly from varied rubber mixtures.

![Figure 6. The belts ‘cogs side’ stiffness](image)

Figure 7 shows the belts transverse stiffness. The tests were conducted for single cogs of each belt. The research was performed in conical pulleys where radial motion of the cog was blocked. Each presented characteristic is the average for four cogs. As it is seen the transverse stiffness is definitely varied for considered belts. The B belt characteristic is strictly linear. The maximum displacement for C belt is similar but in this case the graph has rather stepped character. In the A case the displacement increases extremely after exceeding the specified load.

![Figure 7. The belts transverse stiffness](image)

Figure 8 includes the results of measurement of the Shore hardness according to the D scale for three surfaces of the each belt. The shown values are averages from 10 points. As it is seen the hardness of each surface is similar between considered belts.

![Figure 8. Shore hardness](image)
4. The results and discussion

The bench tests have been conducted on the complete light two-wheeled vehicle drive system so the experiments refer to real exploitation conditions. The research has been performed for three fixed gear ratios of the CVT and with the full throttle position in each case. The powertrain has been loaded in the whole range of engine revs in quasi-static conditions. The measurement of the force transmitted on the engine brake lever and the brake shaft angular speed allows to calculate the power behind CVT. The released characteristics have been compared with the engine external characteristic. In this way the overall CVT efficiency has been estimated. Simultaneously, the belt slip on both pulleys have been examined. The measurement of the angular speed and winding radii for each pulley has enabled to estimate the pulley circumferential velocity for radii related to belt back side. The belt back side circumferential velocity has been measured by optical head. The relative difference of these velocities determinates the slip value. This approach to the issue enables to become independent from the necessity of the belt neutral axis determination. Every graph presented below is the average from three measurements carried out in the same conditions.

Figures 9-11 show the comparison of CVT overall efficiency and belt slip on both pulleys between examined belts for gear ratio 0,9. As it can be seen the efficiency does not exceed 70% for belts A and B. For the C belt maximum efficiency is lower about 5%. This difference corresponds with the belt slip which maximal value on both pulleys concerns just the belt C. The slip on both pulleys is congruous however it is slightly higher for the driven pulley. The reason lies in fact that for considered gear ratio the belt pitch radius is less on the driven pulley. The belt has there worse work conditions among other by its flexural rigidity.

Figures 12-14 refer to gear ratio 1,8. The maximal efficiency is higher than previously and reaches almost 80% for the B belt. The value for A belt is insensible smaller. The C belt maximal efficiency (72%) is again clearly less than others. The belt slip value is noticeably greater on the driving pulley which winding radii for this gear ratio is much lower than the driven pulley. As before the slip of the C belt is about 2% greater than for others strips.

Figures 15-17 relate to the greatest examined gear ratio 2,8. In this case the maximal efficiency is connected with the B belt and its value exceeds 80%. The efficiency for A and C belt is similar and even 5% less than for the B case. The belt slip on driving pulley is much higher than previously and for C belt it reaches 18%. This value is at least 5% greater than for others belts. However, on the driven pulley this quantity is very similar for all the strips and close to 0. 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 worth to note that changes of the overall efficiency against the engine revs correspond with the engine torque characteristic. In each case this relationship can be approximated by a second-order polynomial which maximum is in the range 4000-5000rpm. On the other hand the slip is almost constant in the whole range of engine revs.
Gear ratio: 0.9

Figure 9. CVT overall efficiency for gear ratio 0.9

Figure 10. Belt slip on driving pulley for gear ratio 0.9

Figure 11. Belt slip on driven pulley for gear ratio 0.9
Gear ratio: 1.8

Figure 12. CVT overall efficiency for gear ratio 1.8

Figure 13. Belt slip on driving pulley for gear ratio 1.8

Figure 14. Belt slip on driven pulley for gear ratio 1.8
Gear ratio: 2.8

Figure 15. CVT overall efficiency for gear ratio 2.8

Figure 16. Belt slip on driving pulley for gear ratio 2.8

Figure 17. Belt slip on driven pulley for gear ratio 2.8
The experiment results clearly show a substantial influence of the belt properties on the transmission efficiency. As it can be seen the B belt is characterized by the lowest power losses for each gear ratio. Simultaneously, the smallest speed losses associated even if with belt extension are related with the belt A. The reason of this phenomenon results from the fact that the A belt has the greatest cord stiffness (figure 5) and also relatively high ‘cogs side’ stiffness (figure 6). Furthermore, the coefficient of friction for the A belt is the highest. The features accompanied with the B belt entails therefore the significant reduction of the torque losses. It must be noted here that this belt is characterized by the lowest cord and ‘cogs side’ stiffness and outstandingly linear transverse stiffness. As it results from the above, strip features limiting specified type of losses contributes to increasing other losses at the same time. The greatest slip is connected with the C belt for each case. It is also accompanied with the lowest efficiency except the highest gear ratio. The following behavior could be explained by its flexural rigidity which is distinctly the largest for the C strip. Additionally this belt features the smallest coefficient of friction for both pulleys.

5. Conclusions
1. The available rubber V-belts are varied in term of their construction details, used materials and stiffness parameters.
2. The carried out experiments indicate significant influence of the belt properties on the overall efficiency as well as the belt slip.
3. The high belt tensile stiffness and coefficient of friction entail slip reduction. On the other hand the largest overall efficiency refers to belt with the greatest susceptibility.
4. The gear ratio has substantial contribution for the belt slip and CVT overall efficiency. The belt slip amount is extremely determined by winding radii and it could be significantly different for driving an driven pulley. Such a trend has been observed for each considered belt.
5. The slip value indicates that this is significant part of the total power losses.
6. The measured maximal CVT overall efficiency slightly exceeds 80% and it corresponds with the research presented e.g. in works [2] and [3]. However it is still low value in reference to standard mechanical transmissions.

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