SZENT ISTVÁN UNIVERSITY

Characteristics of the energy balance of agricultural V-belt drives

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NOMENCLATURE

$F_1;\ F_2$	belt side forces in the tight; loose side	$[N]$
F_H	pre-tensioning force	$[N]$
F_{HN}	pre-tensioning defined for specific setting	$[N]$
$M_1;\ M_2$	torque occurring on drive and driven side	$[Nm]$
M_N	torque defined for specific setting	$[Nm]$
s_E	effective slippage	[%]
b	width of the upper side of the V-belt profile	$[mm]$
d	nominal size of pulley	$[mm]$
f	belt bending frequency	$[s^{-1}]$
i	geometric transmission	$[-]$
s	belt slippage for the whole drive	[%]

Greek Letters:

β_G	geometrical angle of wrap	[°]
β_V	real angle of wrap	[°]
$\omega_1;\ \omega_2$	angular speed of drive and driven pulley	$[rad/s]$
δ	wrap ratio	$[-]$
ξ	flank angle	[°]
ψ	relative angular displacement	[°]

1. INTRODUCTION, OBJECTIVES

In the first chapter the significance of the theme is defined and the objectives of my work are presented.

1.1. Timeliness and significance of the chosen theme

Agricultural machines have undergone significant changes over the last decades. According to the conditions of today's modern agriculture, an important aspect in the development of machines is the enhancement of performance and reliability in addition to economical operation. In order to increase efficiency producers put great emphasis on the development of engines as well as the components of working units and power transmission, which are aimed at improving the overall performance of the machines. In agricultural practice flexible tractive element drive is widely used for the power supply of the units, which encourages my research in the field of belt drives.

Drive belts have undergone mainly material and manufacturing technology development over the last decades, which also entails a kind of constructional development in belt construction. Belt producers have designed different profiles for various demands. Among flexible drives on agricultural machines V-belts have become the most wide-spread. When designing a belt drive engineers choose the belt profile and pulley sizes required for the specific power and revolution, taking into account the place of instalment, gear ratio, drive dynamics, etc. For sizing data from the producers' experience and data from experiments carried out under specific conditions by the producers are available, which do not take into account environmental impacts. The efficiency and lifetime of belt drives designed on the basis of catalogues will not be adequate under extreme conditions characteristic of agricultural machines. In such cases the results of our own many tests conducted in the given circumstances can be safely relied on.

1.2. Objectives

The aim of my research is to determine the factors and connections by studying the power loss of V-belt drives that can help design and presumably optimize the efficiency of V-belts used in agricultural machines. To this end my research work covers two main areas: the study of operational as well as installation and machine setting characteristics.

Research goals in connection with the operational characteristics of V-belt drives:

- Based on the heat build-up of V-belts to determine the drive parameters influencing torque loss and to set up the mathematical model for the temperature increase in V-belts.
- To create the qualitative energy balance of the drive by mapping the components of V-belt drive losses.
- Development of a new experimental method where the relative motion of the V-belt can be studied at the operational revolution of the drive.
- To clarify controversies related to relative motions found in the literature using the experimental method developed by myself.

Research objectives related to the characteristics of V-belt drive installation and machine setting:

In the case of agricultural machinery the geometric adjustment values prescribed by the producers for V-belt drives are often not met. The reason for this can be found in the structure of agricultural machines or in their operation in a specific environment. The study of the effects of real belt running, which is different from the theoretical running, can assist in designing the drives of agricultural machinery. My research goals include:

- The empirical determination of the limit of the geometric setting error (misalignment) of the V-belts used in agricultural machines, where the working operation of the drive is still realized without a decrease in efficiency and lifetime.
- Examining the effect of a geometric setting error on the relative motions of the V-belt.

2. MATERIAL AND METHOD

In this chapter the experimental methods and tools used to accomplish my research goals are presented.

2.1. Examination of temperature rise in V-belt

During my experiments the test parameter was the temperature rise in V-belts, which means the power loss occurring on the peripheral force between the two steady states - between the workshop and the stabilized state of the operating temperature. The temperature of the V-belt was measured with an infra camera type NEC H2640. The thermal camera images were recorded of the active side of the V-belt at a frequency of 0.25 Hz, through which the process of warming could be observed. Temperature data was obtained from the thermal images taken of the active surface of the V-belt by using the evaluation software Image Processor Pro II (Fig. 1).

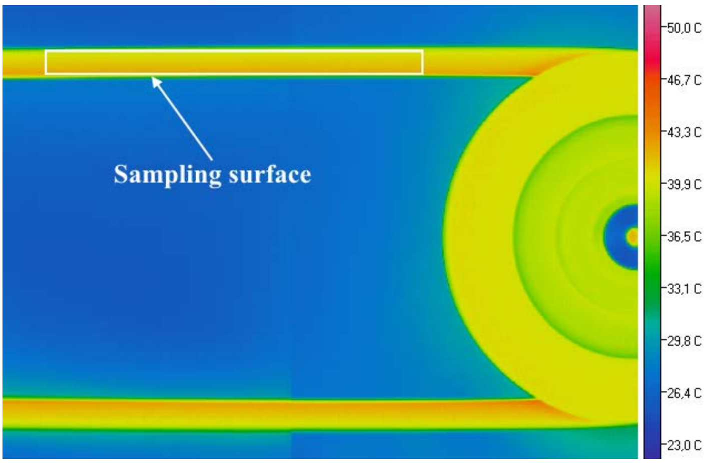


Fig. 1. Infrared camera image with the sampling surface

2.2. Studying the relative motion of the V-belt

The result of motion losses during the power transmission of the V-belt drive is the lower than theoretical angular speed of the driven axis. The value of the slippage relevant to the whole system can be directly determined from the drive's input and output revolution, but no explanation is obtained for the generation of the loss. The quick occurrence of the process makes the observation of relative motions at operating revolutions rather difficult. At 1000 min^{-1} revolution and 180 ° angle of wrap one point of the V-belt stays on the pulley for 30 ms. The so-called high-speed cameras capable of slow-motion observation of fast processes can be used to analyse these short-time processes. High speed means the recording of frames consecutively at a high speed, i.e. the high value of frames per second. During conventional recording the number of recorded images is 25 to 30 per second. During the test method, self-designed and constructed test equipment and an Olympus i-SPEED TR camera (Fig. 2) were used.

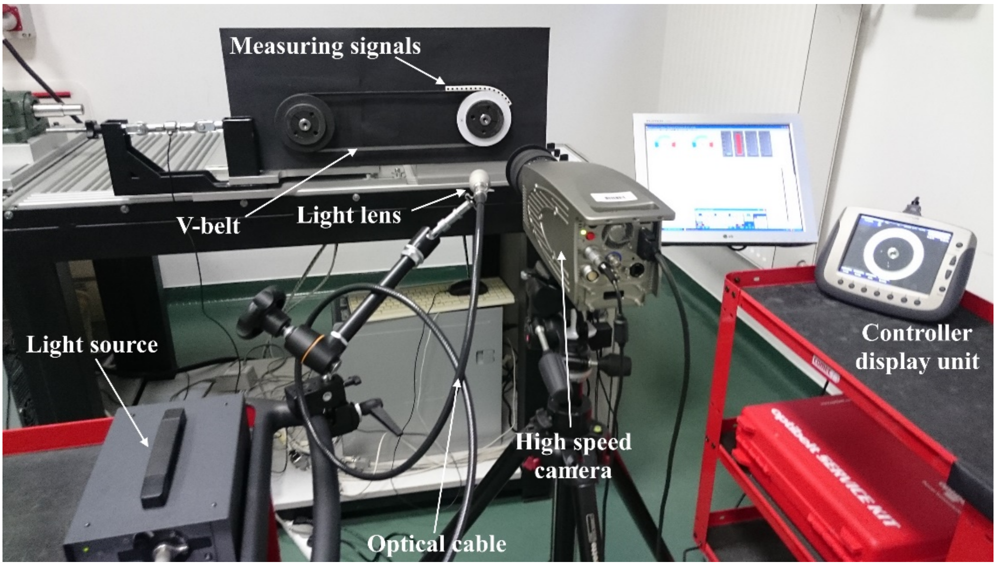


Fig. 2. The experimental arrangement of relative motion tests

In order to observe relative motions the components (pulley, V-belt) were marked with measuring points whose path of motion describes the motion of the machine elements. The correct measuring signal was optimized after several experiments. The motion path described by the measuring points was determined by the i-SPEED Control Pro imaging software belonging to the camera. The motion of the V-belt relative to the pulley was examined along the wrap of the pulley as broken down into two components – radial and circumferential components. The relative motions determined during the experiments describe the motion of the upper side of the V-belt in the pulley groove at the operational revolution of the drive.

3. RESULTS

In this chapter the new scientific results achieved during my research work are presented, which help to understand the operation and optimize the efficiency of V-belt drives used in agricultural practice.

3.1. Drive characteristics affecting power loss

When designing V-belt drives, the power requirements as well as the geometric and kinematic parameters of the driven machine unit are considered as the basis by the engineers, which is adapted to the drive characteristics of the power source by changing the drive. Power transmission can be achieved with several belt drive designs, so it is the designer's task to define the optimal drive design considering different aspects. In the case of theoretical drive arrangements the study of the drive characteristics affecting the power loss of V-belt drives helps with designing the optimal V-belt drive.

3.1.1. The result of the temperature rise test

The factors influencing the torque loss among the drive parameters were determined with the help of the temperature rise of the V-belt, i.e. the difference between the initial and saturation temperature. The regression model was created with the known functions of the drive characteristics to describe the V-belt temperature rise:

$$\Delta T = a_0 + \frac{a_1}{d} + a_2 \cdot f + a_3 \cdot M + a_4 \cdot F_H \, , \tag{1}$$

where a_0 [°C]; a_1 [$°\mathrm{C} \cdot mm$]; a_2[$°\mathrm{C}/Hz$]; a_3 [$°\mathrm{C}/Nm$]; a_4 [$°\mathrm{C}/N$] are the function constants.

Table 1. The variance table of ΔT

	Sum of squares	Degree of freedom	mean square	F/T-value	p
Model	3574.22	37	1158.16	394.75	<0.001
$1/d$	3152.84	4	788.21	29.824	<0.001
f	211.43	1	211.43	8.489	<0.001
M	129.39	2	64.67	6.228	<0.001
F_H	2.43	2	1.22	-0.423	0.675
remainder	99.75	34	3.99		

Table 1 shows that value F of the model is significant, so our model is valid. In addition to the pre-tensioning force, the coefficient of each independent variable is significantly different from 0. With the tested factors, the goodness of the model fit, i.e. what proportion of all the variance the model explains, resulted in value $R^2 = 0.970$. Due to the high correlation the cross effects of the factors are not taken into account. In Table 2 the relationship of the individual parameters was examined in the regression model, where the standardized regression coefficient makes the correct comparison of the factors possible. The Beta value shows how much the independent variable influences the dependent variable.

Table 2. Correlation of parameters in the regression model

Model	coefficient	Standardized regression coefficient Beta	t	p
constant	-17.100		-14.973	<0.001
$1/d$	2317.476	0.888	29.824	<0.001
f	0.472	0.243	8.489	<0.001
M	4.430	0.185	6.228	<0.001

According to Table 2 entering the specific values of coefficients, in case of the model Z/10 belt profile:

$$\Delta T = -17.1 + \frac{2317.476}{d} + 0.472 \cdot f + 4.430 \cdot M. \qquad (2)$$

3.1.2. Energy balance of V-belt drives

During the measurements, the power transmission of the test settings varied between 450 and 1660 W, where the power loss (20-153 W) is also a value which depends on the settings. The resolution of the two tested loss components is estimated by variance analysis, based on the variance of the independent variables (Fig. 3).

3-21% of the input power of the tested V-belts is loss. Most of the power loss, 75-92% is torque loss and the remaining part is motion loss. The torque loss originated mainly from the bending of the V-belt (internal friction) determined by the radius of the bending of the belt and its frequency. The frictional loss of the contact surfaces of the force-locking drive is manifested in a complex manner. On the one hand, from the repeated deformation of the surface of the V-belt, which is realized as torque loss and made up of the relative displacement of the belt element. Motion loss is influenced by the frictional conditions of the contacting surfaces determined by the size or change of the transferred peripheral force (the course of the deformation along the curve length) and the pre-tensioning of the V-belt.

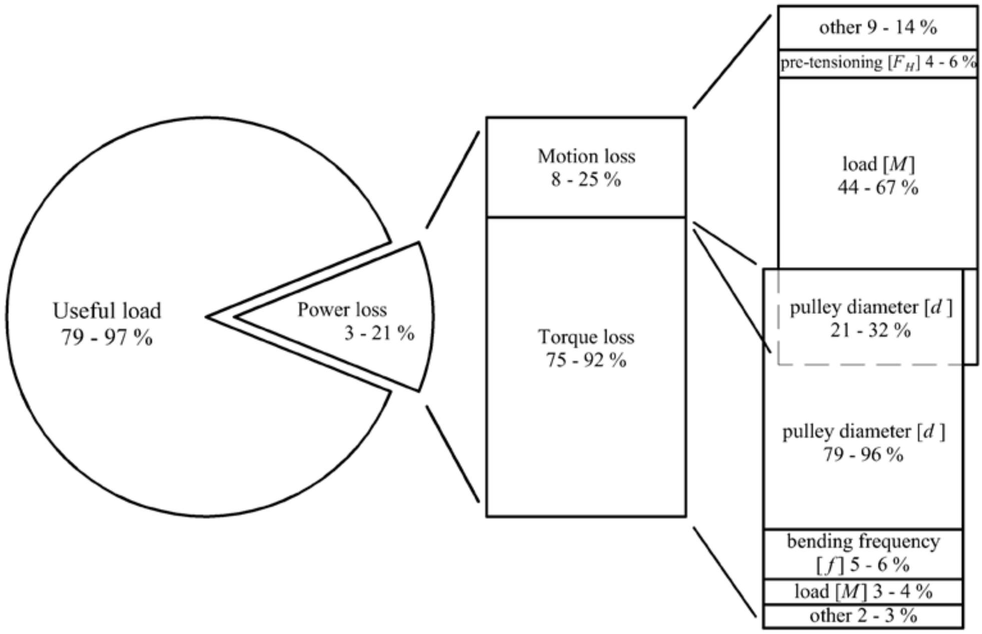


Fig. 3. Qualitative energy balance of V-belt drives
(Z/10 profile; $d = 60 - 180\ mm$*;* $i = 1$*;* $f = 10 - 20\ s^{-1}$*;*
$M_1 = 3 - 18.3\ Nm$*;* $F_H = 50 - 300\ N$*;* $a = 345 \pm 10\ mm$*)*

3.2. Belt relative motions

The motion loss due to the force-locking engagement of the V-belt and pulley reduces the theoretical speed of the driven axis, which is influenced by a number of factors. This phenomenon is explained by the micro-level examination of relative motions of the V-belt. During the evaluation of the measurement data, the engagement of the V-belt into the pulley groove and its circumferential slip were determined, and these motions were divided into sections along the wrap.

3.2.1. The radial motion of the V-belt in the groove

The radial relative motion of the V-belt is shown in Fig. 4 in the no-load drive and during power transmission on the drive and driven pulleys. The middle section of the radial motion of the V-belt is the same as the statements provided by the literature, but it cannot be clearly divided into further sections on either pulley. There is no significant radial relative motion without loading the drive, but on the drive and driven pulleys going towards the tight section the effective bearing radius of the V-belt decreases. Along the wrap with the increase in the belt force, the elastic V-belt cross-section suffers increasing deformation and with this the belt section penetrates the groove deeper.

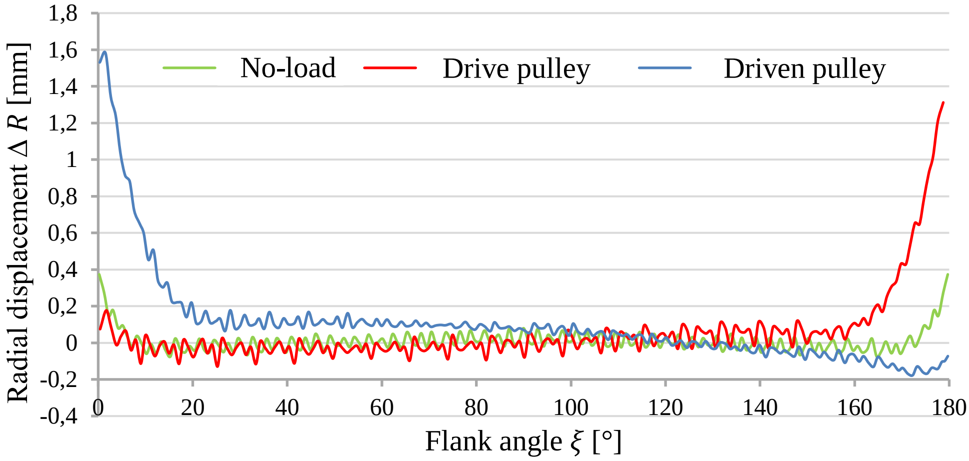


Fig. 4. Radial relative motions as a function of the flank angle
(Profile Z/10; $d = 118\ mm$; $i = 1$; $L_w = 1142\ mm$; $f = 10\ s^{-1}$; $F_H = 190\ N$; $a = 386\ mm$)

3.2.2. Ranges of V-belt relative motions

The literature breaks the relative motion of the V-belt into four ranges. Along the wrap it describes the ranges of belt running-up and running-down, as well as ranges of adhesion and flexible slippage. The latter two ranges were created on the basis of the flexible slippage theory. Measurements conducted at a speed of several orders of magnitude less than the actual operation showed the existence of the four ranges on the driven pulley, but not on the drive pulley. In the case of experiments performed at low belt speed, neither the influence of inertial forces, nor the rheological properties of the V-belt play a role. In addition, even the relative speed of the contacting surfaces does not affect the friction connection.

The wrap curve length was divided into three sections with the help of the radial components. The borders of the sections were defined by the horizontal coordinate of the intersection of the lines fitted to the curve. In order to correct the actual angle of wrap, a proportion was created called wrap ratio δ. The wrap ratio was determined by the ratio of real and geometrical wrap:

$$\delta = \frac{\beta_V}{\beta_G}, \tag{3}$$

where:

β_V real angle of wrap defined from the relative motions [°].
β_G the angle of wrap defined from the geometry [°],

Table 3 shows the real angle of wrap of the V-belt drive and the wrap ratio in no-load drive as well as on the drive and driven pulleys. The real wrap is 91% of the angular domain calculated from the geometry in the case of unstrained drive. On the drive pulley this value drops to 87%, the wrap of the driven pulley equals the no-load state.

Table 3. The real angle of wrap of the V-belt and the wrap ratio

No-load drive		Drive pulley		Driven pulley	
real angle of wrap [°]	δ	real angle of wrap [°]	δ	real angle of wrap [°]	δ
163	0.91	157	0.87	163	0.91

3.2.3. The circumferential motion of the V-belt in the pulley groove

Effective slippage (s_E) gives the difference in the angular speed of the pulley and the V-belt with respect to the angular speed of the pulley when rotating, i.e. the momentary belt slippages in circumferential direction:

$$s_E = \frac{\omega_t - \omega_{sz}}{\omega_t} \cdot 100\ [\%]. \tag{4}$$

The slippage of the drive considering the whole system is made up of belt slippages emerging on the drive and driven pulley. Fig. 5 shows that angular slippages differ on the pulleys involved in the drive. The slippages measured on the two pulleys were illustrated on a diagram with the tight belt section on the left and the loose section on the right. On the drive side the angular speed of the pulley is greater than the angular speed of the belt$(\omega_t > \omega_{sz})$, so the slippage takes positive values. In the ranges of the belt running-up and running-down the increased relative motion results in significant effective belt slippage values on both the drive and the driven pulley.

On both pulleys the frictional relationship between the active surface of the V-belt and the groove wall is formed in the range of the real angle of wrap β_V as the drive elements are in contact in this curve section. However, the effect of the belt running-up can be observed at the beginning of the real wrap (on the *A*-*B*, and *E*-*F* curve lengths). Similarly, the relative motions emerging during the belt running-down are already present at the end of the real wrap (in the *C*-*D* and *G*-*H* sections). The upper side of the V-belt slows down or accelerates due to the deformation coming from its bending, thereby affecting the effective angular slippage.

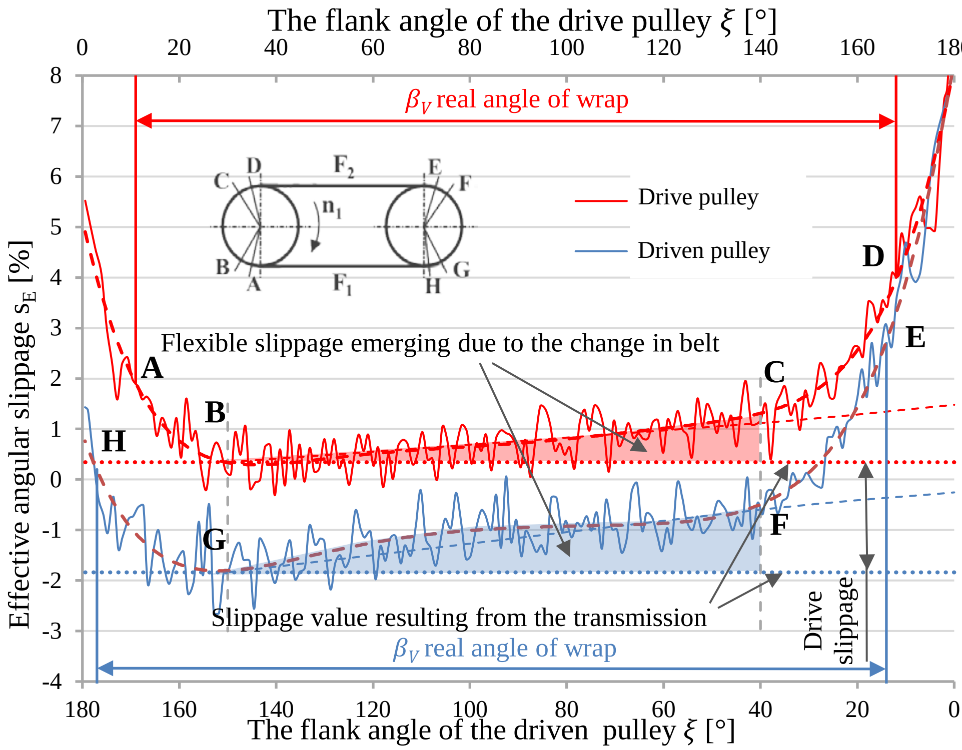


Fig. 5. Effective angular slippage as a function of the flank angle
(Profile Z/10; $d = 118\ mm$; $i = 1$; $L_w = 1142\ mm$; $f = 10\ s^{-1}$;
$M_1 = 10.7\ Nm$; $F_H = 190\ N$; $a = 386\ mm$; $s = 2.32\ \%$)

On the drive side, after section *A-B*, the frictional condition needed to transfer the peripheral force is created, which determines the value of the effective angular slippage. In section *B-C* due to the decreasing belt tension along the wrap, the belt element moves out of the groove. As a result, the angular speed of the V-belt decreases during its circular motion, and the effective angular slippage increases. On the drive pulley belt slippage is thus made up of the relative displacement developing during the transfer of the peripheral force and of the so-called flexible slippage developing from the deformation of the belt bending and due to the decrease in belt tension.

The V-belts entering and exiting the driven pulley also result in significant effective angular slippage. On the pulley the upper side of the V-belt falls behind at the beginning of the running-up (E) and then changes direction at the place $s_E = 0$. Similarly, this process takes place before the running-down. The V-belt's angular speed is greater than the angular speed ($\omega_t < \omega_{sz}$) of the driven pulley, the force flow is from the V-belt towards the direction of the pulley. Thus, slippage takes negative values and also increases the drive's motion loss. The nature of the angular slippage curve is similar to the change in the slippage measured on the drive pulley, but the direction of the V-belt running-up is the opposite. At the end of the actual wrap on the tight belt section (G), where the belt force almost equals the force emerging in the tight belt section, belt slippage is determined by the friction conditions. Prior to this (in section *G-F*), as a result of the increasing tension in the belt, the flexible slippage already mentioned at the drive pulley takes place.

Belt slippage does not occur on the pulleys involved in the drive to the same extent. In the experiments 85% of the motion loss is generated on the driven pulley $i = 1$ during transmission. The slippage characteristic of the transfer of the peripheral force can be defined by the extreme value of the angular slippage curve at the real angle of wrap near the tight belt section of the two pulleys, where flexible slippage does not yet have an effect on the drive pulley and no longer has an effect on the driven pulley. The difference between the two defined slippage values specifies the slippage of the drive for the whole system.

3.3. Geometric machine setup errors

Measurements were taken to map the geometric setup errors of V-belt drives operating on agricultural equipment. The results show that the pulleys are not always located in the center plane of the drive. The maximum tolerances regarding V-belt drive alignment are given by the producers as a function of the pulley diameter while the nature of the occurring fault and the belt profile is not taken into account. The maximum permissible deviation may occur due to the parallel misalignment of the pulleys (Fig. 6a) or the angular misalignment of the axes (Fig. 6b). In both cases the straight belt sections undergo extra bending (ϑ)and the sidewalls experience larger friction where running onto and off the pulley. In the case of parallel misalignment occurrence the friction increases on both sides, in the case of angular misalignment it is strained more on one side only. Here a strained and an unstrained side was explained.

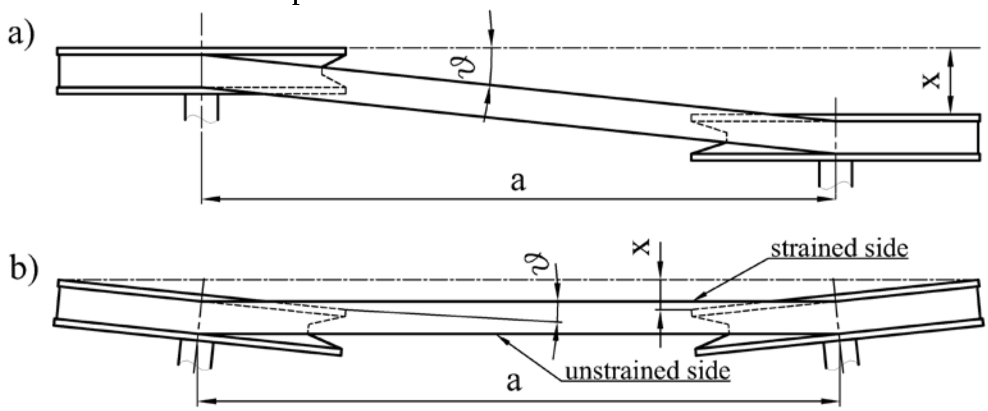


Fig. 6. Understanding the setup errors of pulleys
a) parallel misalignment; b) angular misalignment

3.3.1. Definition of permissible geometric machine setup error

Manufacturers prescribe such a small size interval for the geometry setting of V-belts that cannot be maintained in agricultural equipment. The permissible axial deviation of the pulleys was determined by examining the loss components relative to the setup of the pulleys in one plane. The ϑ inclination angle of the straight belt section originating from setup errors is defined by the a axis spacing and the axial deviation of x pulleys. The effect of the geometric setup error is different according to the belt profile because the width of the V-belt section and with this the lateral bending rigidity change. During my experiments, the parallel error values were not adjusted to the diameter of the pulley, like by the belt manufacturers, but to the size of the belt cross section.

The parallel misalignment error affects the slippage of the whole system favorably up to the value of the profile width. Initially, even a slight deviation significantly reduces the revolution loss of the driven axis. Approaching or exceeding the V-belt profile width, belt slippage is increased due to the error. The V-belt's temperature rise is slightly increased initially as a function of the set error value, but in excess of the profile width the torque loss suddenly increases. Based on these tendencies, when determining the setup errors not the pulley diameter but the width of the belt profile prevails. The permissible value of the geometric setup error of the pulleys was defined for one third of the profile width ($x = b/3$) as the efficiency of the belt drive does not change with the experimental setups up to such a deviation. From the point of view of service life this setup error value is also appropriate, and the belt temperature increase does not exceed 10%. At the suggested error limits, the power transmission of the V-belt drive can be maintained operationally, up to 35 axis spacing and belt profile width ratio.

3.3.2. V-belt relative motions due to setup error

The geometric setup error (misalignment) of the pulleys greatly affects the slippage of the V-belt drive of the whole system. With relative motion experiments further tests were made to explain the slippage changes caused by parallel misalignment. On the drive pulley independently of the extent of the error the V-belt occupies its position firmly on a smaller diameter and retains its radial position until the exit begins. The loaded side of the V-belt is deformed prior to wedging due to the F_{gh_1} axial force developing because of the setup error (Fig. 7). At the end of the running-up, the belt force F_1 emerging in the tight belt section and force component F_{gh1} originating from the setup error result in the belt cross-section reaching its smallest dimension and going deeper into the groove. Due to the highly tight V-belt section the load-bearing capacity of the force-locking contact increases and the flexible slippage significantly decreases. However, increasing the setup error, the normal force on the unloaded belt side decreases to such an extent that the V-belt can no longer deliver the same torque with the same slippage.

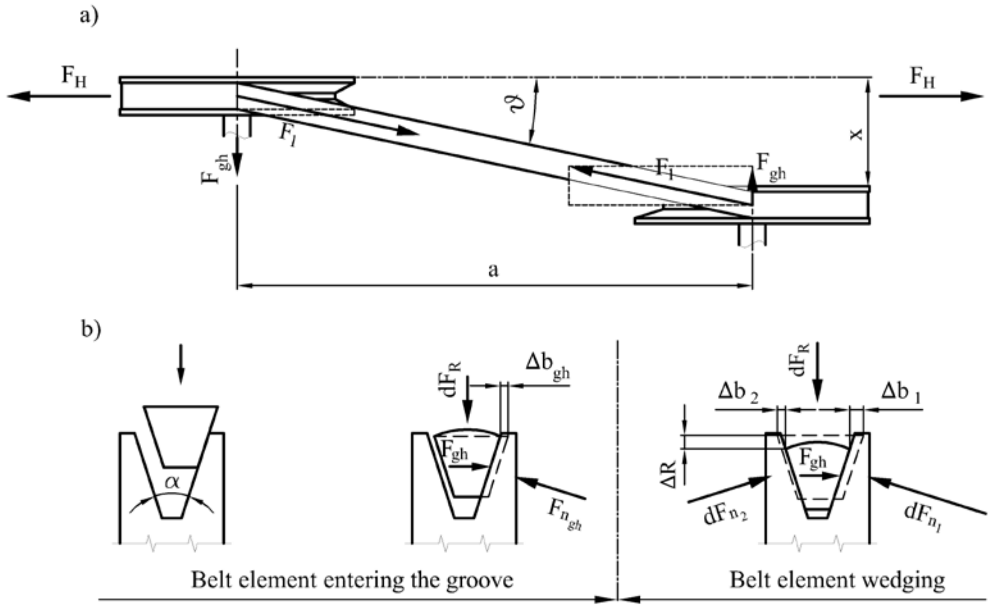


Fig. 7. Forces affecting the V-belt during the running-up phase, in case of setup error
a) a top view of the belt drive and b) the position of the belt cross section at a given moment in the wedge groove

On the driven pulley, the V-belt runs down from the loose belt section, thus the axial force component is smaller. Because of the smaller forces the belt section – opposite the drive pulley – gradually take its place in the groove of the pulley. Depending on the size of the setup error on the driven side, the circumferential relative motion may decrease or increase. Similarly to the drive pulley, the asymmetrical load on the active belt sides affects the force-locking contact of the V-belt and the pulley.

4. NEW SCIENTIFIC RESULTS

1. Mathematical model of temperature rise in V-belt

The mathematical model of the temperature rise in V-belt was determined using the known functional relationships of the drive parameters. Temperature rise in V-belt as a function of the drive characteristics:

$$\Delta T = -17.1 + \frac{2317.476}{d} + 0.472 \cdot f + 4.430 \cdot M,$$

where: d -the nominal diameter of the V-belt pulley $[mm]$,
f -the belt bending frequency $[s^{-1}]$,
M -the torque affecting the drive $[Nm]$.

It was determined on the basis of the variance analysis of variables that the size of the diameter of the pulley influences the heating-up of the belt to the greatest extent. The frequency of bending and the effect of the load is nearly the same, but at the same time has a smaller effect than the pulley diameter. The pre-tensioning force has no significant effect on the belt warming. In the case of normal profile V-belts and casting pulleys the relationship Z/10 can be applied in the following range with a margin of error of 5%. The model's validity limits:

$$d_{min} \leq d \leq 3 \cdot d_{min} \,,$$

$$10\ s^{-1} \leq f \leq 20\ s^{-1},$$

$$0 \leq M \leq M_N \,,$$

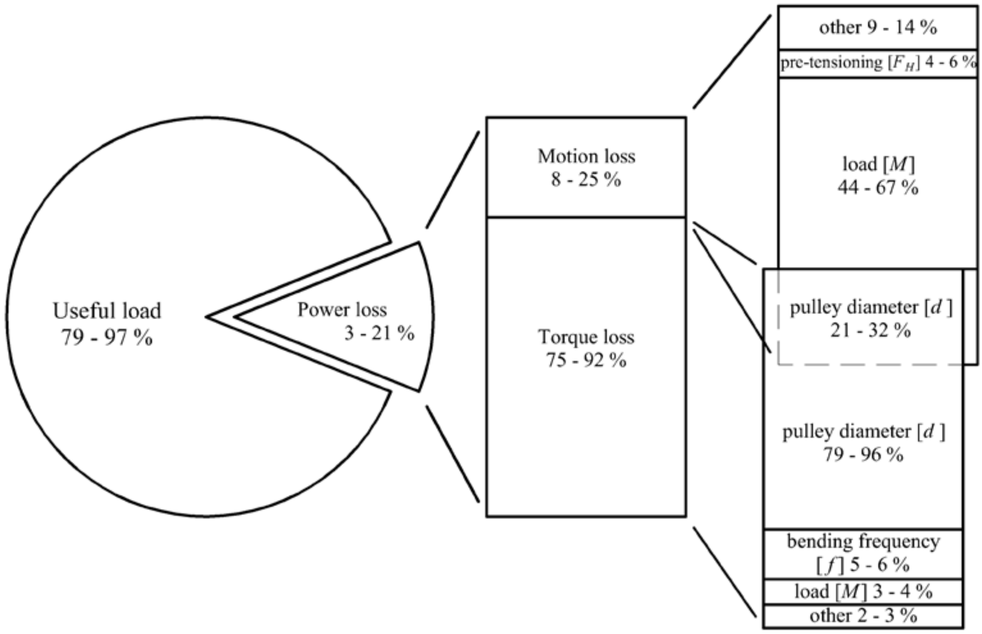
$$0.5 \cdot F_{HN} \leq F_H \leq 1.5 \cdot F_{HN} \,,$$

$$i = 1 \,,$$

where: d_{min} - the nominal diameter of the smallest pulley specified for the belt profile $[mm]$,
M_N - the load torque calculated for the given setup $[Nm]$,
F_{HN} pre-tensioning force specified for the given setup $[N]$.

2. The qualitative energy balance of the V-belt drive

Using the results of the drive parameter tests the qualitative energy balance of the V-belt drive was developed. The power loss of the V-belt drive was divided into two components, torque and motion loss. The torque loss derives essentially from the bending of the belt (internal friction), which is determined by the radius of the bending (d the diameter of the pulley) and its frequency (f the frequency of bending). With force-locking drives, the transmission of the peripheral force contributes to the loss in a complex way. On the one hand, it is made up of the repeated deformation of the surface layers of the V-belt (internal friction), which is realized as torque loss, on the other hand of the relative displacement of the belt element. Motion loss is influenced by the friction of the surfaces, which is determined by the amount of the transmitted force (Mload torque), and the deformation along the V-belt curve-length (d pulley diameter) and prestressing (F_H). The test limit of the energy balance is described in my thesis titled "Mathematical model of temperature rise in V-belts".



The energy balance of V-belt drives

3. The test method for the relative motions of the V-belt

There is no such method in the literature that could be used to examine the motion of the V-belt in the pulley groove under conditions similar to operation. An experimental method was developed and a drive test dynamometer was created to determine the relative motion of the V-belt. Under laboratory conditions, the actual load of the drive ($M = 10.7\ Nm$) was produced at operational revolution ($n_1 = 924\ min^{-1}$). Images were taken of the V-belt drive at high speed ($2000\ images/s$). In order to observe the relative motions, the components of the drive were marked with signs indicating the motion of the pulley and the upper side of the V-belt. After image processing the motion path described by the measuring points was developed with 0.28% relative error, from which the radial and circumferential relative motion of the upper side of the V-belt and the current belt slippage along the wrap were determined. The commercially available drive elements used during the experiment do not require conversion, so the real belt motion can be measured.

4. Relative motions of V-belts

Through laboratory tests it was proved that the relative motion of the V-belt along the wrap, under operating conditions, can be divided into three ranges (belt running-up, real wrap, belt running-down) by defining the breakpoints of the radial path. As a quotient of the real and theoretical angle of wrap, a new parameter was defined, the wrap ratio, which can be used to clarify the rope friction model when describing the operation of the belt drive:

$$\delta = \frac{\beta_V}{\beta_G},$$

where: β_G - the angle of wrap determined from the geometry [°],
β_V - the real angle of wrap determined from the relative motions [°].

By interpreting the effective slippage (s_E) it was determined that belt slippages differ on the pulleys involved in the drive. Belt slippage is made up of the relative displacement developing during the transmission of the peripheral force, of the deformation of the V-belt bending and of the flexible slippage emerging due to the change in belt tension. The slippage emerging at the transmission of the peripheral force can be defined by the extreme value of the angular slippage curve at the real angle of wrap near the tight belt section of the two pulleys. The difference between the two extreme values gives the slippage value for the whole drive.

5. Geometric setup error of pulleys

The real belt running is different from the theoretical running in all cases, especially in the case of large, self-propelled, plate-structured agricultural machines. It was demonstrated through experiments that the loss emerging due to the geometric misalignment is caused by the increased friction of the active belt sides. The permissible value of the parallel misalignments of the pulleys was determined for one third of the profile width $(x = b/3)$, where the loss emerging on the motion and peripheral force were both taken into account. Below the margin of error determined by me the loss of motion (belt slipping) can be significantly reduced. By further increasing the error, it reaches and exceeds the slippage value of the drive layout adjusted in the same plane. The small asymmetrical load of the belt sides increases the groove effect by the increased wedging of the belt, but in case of a greater deviation than the margin of error the belt slippage increases due to the additional load.

5. CONCLUSIONS AND PROPOSALS

During my research by testing the loss factors of V-belts I obtained such results which help to better understand the operation of the drive and help with the design and optimization processes of V-belt drives used on agricultural machines.

Through experiments I created the mathematical model of the temperature rise in V-belts and that of the slippage for the whole system as a function of the drive parameters. Based on the variance of the individual variables, I determined the drive characteristics influencing loss and the weight of each factor. Using the results of the drive parameter tests I set up the qualitative energy balance of the V-belt drive, where the power loss was divided into two components, torque and motion loss. The tests can be extended to other belt types, and additional research task will be to establish simulation models to describe the operation of the drive.

I developed an experimental method to determine the motion of the V-belt in the pulley groove. From the coordinates of the measuring points placed on the V-belt and the pulley I determined the motion of the upper side of the V-belt relative to the pulley. The experimental method allows for the kinematic examination of other types of belt types and other drives at operating speeds.

By analysing the relative motions I proved that the relative motion of the V-belt along the wrap, under operating conditions, can be divided into three ranges (belt running-up, real wrap, belt running-down). I defined a wrap ratio (δ), which I interpreted as the quotient of real and theoretical wrap. The rope friction model used for the sizing of the V-belt drive can be specified with this ratio. By introducing the effective angular slippage (s_E) I divided the elementary slippages occurring on the pulleys, which in a complex way are made up of the relative displacement developing during the transmission of the peripheral force, of the deformation of the V-belt bending and of the flexible slippage emerging due to the change in belt tension.

Among the special conditions occurring on agricultural machines I dealt with the problem of geometric setup errors. I determined the limit of the parallel misalignment of the pulleys where the V-belt drive can be used without any efficiency reduction relative to the theoretical layout. Via relative motion tests I found that as a result of the misalignment the asymmetric load on the belt sidewalls can increase the wedge effect through the increased wedging of the belt element. However, due to a misalignment exceeding the width of the belt profile, the load capacity of the force-locking link is decreased because of the increased additional loads.

6. SUMMARY

The flexible tractive element drive is a widespread power transmission solution for agricultural equipment. It is essential to be familiar with the behavior of the above-mentioned drive for the design and optimal operation of machine structures, especially in the agricultural environment.

By studying the power loss of V-belt drives the purpose of my research work was to gain more knowledge about the operation of the drive and to study the impact of specific agricultural conditions on power transmission. In order to realize the purpose of the research I overviewed the literature on V-belt drives where I found that the experiments conducted so far are not suitable for testing the drive in real conditions. Thus I encountered contradictions even in the case of basic research. In addition, there is not any literature available in relation to drives operating in the agricultural environment.

As the first step I worked out an experimental method to study the torque and motion loss of the V-belt drive, where I used commercially available V-belts and pulleys and created operation-like conditions in order to be able to apply my results better in practice. Torque loss was analysed by the temperature rise in the V-belt. At the macro level motion loss was determined through the angular speed of drive axes and at the micro level through the V-belt relative motions.

Next, it was important to clarify the composition of the power loss in the case of the theoretical drive setting. I created the mathematical model of the temperature rise of belts as a function of the drive parameters, and I determined the impact of each parameter by weighting the individual variables. As a result of the series of experiments I set up the mathematical model of the drive slippage for the whole system, and I also determined the impact of the drive parameters on the studied phenomenon. Using the results I created the energy balance of the V-belt drive and analysed the impact of each component. By developing a new experimental method I clarified the knowledge regarding the relative motions of the V-belt.

After the analysis of the environmental conditions characteristic of agricultural machinery I examined the effect of the geometrical adjustment errors of the pulleys on the operation of the V-belt drive. By empirically determining the error limit of the machine adjustment I established an error value, where performance transmission of the drive is still realized without decrease in efficiency. Via relative motion tests I found that as a result of the adjustment error the asymmetric load on the belt sidewalls can increase the wedge effect through the increased wedging of the belt element.

7. PRIORITY PUBLICATIONS RELATED TO THE THEME OF THE DISSERTATION

Proofread articles in an international language

1. **Gárdonyi, P.**, Kátai, L., Szabó, I. (2015): Examination of drive misalignment and v-belt temperature conditions, International journal of science, technics and innovations for the industry, Vol. 12. pp. 56-59, ISSN 1313-0226
2. **Gárdonyi, P.**, Kátai, L., Szabó, I. (2015): Relationship between the drive installation and v-belt temperature conditions, Mechanical Engineering Letters, Vol. 13, pp. 81-87, HU ISSN 2060-3789
3. Kátai, L., Szendrő, P., **Gárdonyi, P.** (2016): The Power Transmission Stability and Efficiency of V-belts, Progress in Agricultural Engineering Sciences, Vol. 12, pp. 25-49.
4. **Gárdonyi, P.**, Nagy, D., Gergely, Z., Bércesi, G. (2017), Developing test equipment suitable for testing torque transfer systems used in agriculture, Poljoprivredna tehnika, Vol. 42 (4), pp. 27-36, ISSN 0554-5587

Proofread articles in Hungarian

1. **Gárdonyi, P.**, Kátai, L., Szabó, I. (2014): A hajtás beállítási hiba és az ékszíjak melegedési viszonyainak kapcsolata, GÉP, LXV. évf. 6-7. szám, 26-29. o., ISSN 0016-8572
2. **Gárdonyi, P.**, Kátai, L., Szabó, I. (2015): Az ékszíjtárcsa átmérők és az ékszíjak melegedési viszonyainak kapcsolata, Műszaki Tudományos Közlemények, 2015 (3), 151-154. o., ISSN 2393-1280
3. **Gárdonyi, P.**, Szendrő, P., Kátai, L., Szabó, I. (2015): Mezőgazdasági gépeken alkalmazott ékszíjak élettartamának és hatásfokának kísérleti vizsgálata, Mezőgazdasági Technika, LVI. évfolyam, 2015. október, 2-5 o. HU ISSN 0026-1890
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5. **Gárdonyi, P.** (2016): Ékszíjhajtás veszteségét befolyásoló hajtásjellemzők vizsgálatának módszere, GÉP, LXVII. évf. 5-6. szám, 82-85. o., ISSN 0016-8572
6. **Gárdonyi, P.**, Kátai, L., Szabó, I., Balassa, Zs. (2017): Ékszíj belső súrlódási veszteségének vizsgálata üzemhasonló körülmények között, GÉP, LXVIII. évf. 3. szám, 66-69. o., ISSN 0016-8572