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# LOSS MECHANISMS AND EFFICENCY OF PUSHING METAL BELT CVT

**Milan Banić[1], Dušan Stamenković[1], Vojislav Miltenović[1], Jelena Milisavljević[1]**
[1] Faculty of Mechanical Engineering, University of Niš, Niš, Serbia
banicmilan@hotmail.com; dusans@masfak.ni.ac.rs;
vojamiltenovic@yahoo.com; jelena.milisavljevic@masfak.ni.ac.rs

***Abstract:*** *The paper describes the loss mechanisms that occur due to friction within the belt drive as well as belt-slip losses due to friction between metal belt and the pulleys. Paper gives an over review of current models of friction caused losses in metal belt CVT drives as well as the advantages and drawbacks of described models. Paper also discusses impacts of lubricating oils on the transmission efficiency of CVTs as well as effects of metal–metal friction characteristics on the efficiency of a metal V-belt type CVT under various running conditions. The paper summarizes results of investigation of clamping force, lubrication and transmission ratio on the efficiency of metal belt CVT.*

***Keywords:*** *Continuously variable transmission, friction, lubrication, loss modeling.*

## 1. INTRODUCTION

Continuously variable transmission (CVT) has been used for many years in diverse industries. The continuous adjustment of the output speed at constant driving speed is required in many applications. Usage of CVT is especially in the automotive industry as they offer the potential for an improvement in fuel economy relative to discrete ratio transmissions. This arises from the ability to match the engine operating point more beneficially to vehicle requirements as a result of the continuous ratio range. There are many kinds of CVTs [1]: Spherical CVT, Hydrostatic CVT, E-CVT, Toroidal CVT, Power-split CVT, Belt CVT, Chain CVT, Ball-type toroidal CVT, Milner CVT, NUVinci, etc. The review of possible CVT concepts is given on Fig.1. However, belt and push metal belt (or chain) types are the most commonly especially in automotive applications.

Metal push belt CVT consists of cone disks enveloped by a power-transmitting device like belt or chain. Power transmission is done through friction between the disc and chain.

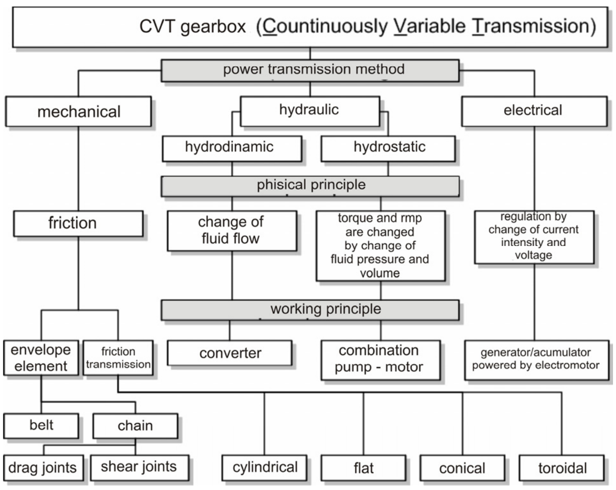


**Figure 1.** *Review of possible CVT concepts [1]*

Varying of transmission ratio is achieved by different position of disc on input and output shaft (Fig.2), i.e. by changing the diameter at which there is direct contact between metal belt and discs. Required pressure force between compressed elements of the chain and working surface is usually achieved by hydraulics via a special hydro aggregates.

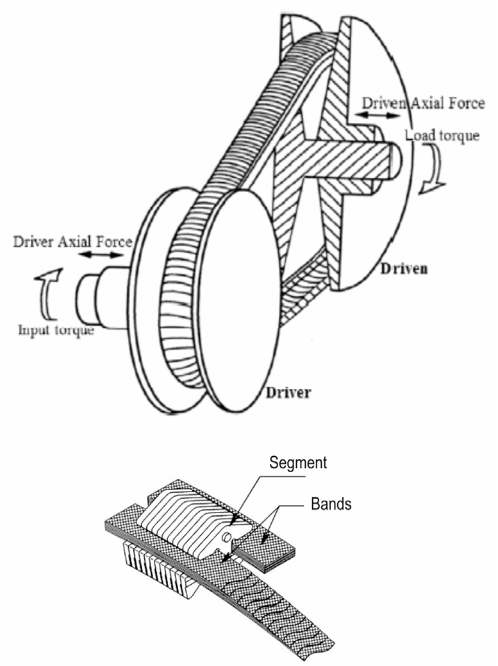


**Figure 2.** *Schematic diagram of a metal-pushing CVT, or a metal V-belt drive*

Metal belt CVTs have been on the market for a number of years now, but so far did not manage to show the improved fuel economy over discrete ratio transmissions with equivalent torque capacity, which was predicted. This deficiency has three main contributing factors. First, engine cannot always be operated at its most fuel efficient condition, even though the transmission ratio could achieve this, due to constraint on the operating point of the engine in the high acceleration level scenarios. The second reason is the lower efficiency of the CVT (0.87 - 0.93) relative to the efficiency of a discrete ratio transmission (0.96 - 0.99) [2]. The inefficiency of CVT has been linked, amongst number of possible inherent losses, to torque losses within the belt mechanism itself and belt slip. Finally, the control logic has not been accurate enough to deliver a desired shifting behaviour. Almost all models for CVT control design assume CVT to be a pure integrator, which is not the best approach for capturing and controlling the dynamics associated with the various components of a CVT system. Moreover, such models would also be inadequate (or rather inaccurate) for the purpose of optimizing fuel economy and acceleration performance of a vehicle [3].

## 2. REVIEW OF METAL BELT CVT FRICTION INDUCED MODELS

Numerous authors investigated the function of the pushing metal belt CVT, while some of them proposed the mathematical models concerning the loss mechanism of a metal belt CVT.

The first significant model, after the pioneer work by Gerbert and Becker was by Micklem et al. [4]. Micklem evolved its model from the first papers concentrating on specific torque loss mechanisms within the belt, over proposal of a viscous shear film between any belt components having a relative motion, to proposal of an elastohydrodynamic lubrication (EHL) regime between the segments and the pulleys to model a belt-slip phenomenon. The final model had a good agreement between measured and calculated slip values between the belt and the pulley.

Kobayashi et al. [5] analysed the slip mechanism by focusing on the distribution of the gaps occurring between the segments, and simulation predicted the slip-limit torque at which the slip ratio increases sharply. The authors analysed the slipping behaviour at the reduction speed ratio and the maximum torque condition, but they did not refer to the transmission efficiency of CVT under a realistic scenario.

Karam and Play [4] adopted a numerical approach to derive global equilibrium equations from the equations of the elements that form the metal belt. They, for the first time proposed that the steel bands actively participate in torque transmission. Noted authors also introduced simplifications regarding assumption that the band segments are always in compression and thus an initial tension exists in the steel bands when at rest, as well as the assumption that the bands are rigid, so their length doesn't change during CVT operation. Simplifications in their work proved to be significant by later work of different authors. Noted authors also developed a concept of dividing the contact path about pulleys into two zones, the adherence zone and the sliding zone. The adherence zone is defined in places where segment and pulley forces are constant, while the sliding zone is defined in a region where compressive forces between segments rise (driving pulley) or fall (driven pulley).

Lee and Kim follow the work of Karam and Play and further divide the pulleys contact paths into three regions: region of constant compressive force and band tension, region of band tension change only and region of variable compressive force and band tension. The noted author's main goal was to further understand the clamping force required for torque transmission in order to improve the efficiency of the transmission. The authors proposed the calculations of clamping forces for specific torque and transmission ratio combinations and calculated the required clamping force in boundary operating regimes. One of the most important conclusion of the noted authors is that in order to have a functional model which is in agreement with experimental data it necessary to implement the variable coefficient of friction between the segments and the pulleys. The coefficient of friction increases linearly with the decrease of transmission ratio.

Kanehara et al. [5] focused on the investigation of effects of friction forces between the steel bands and belt segments. The noted authors indentified six friction coefficients that may be varied to achieve agreement between the model and the experimental data.

Akehurst et al. [4], as well as the investigated the loss mechanisms that occur within the belt drive due to relative motion between the bands and segments and between the pulleys and the belt due to pulley deflection effects.

Narita and Priest [5] focused on investigation of losses between all the contact pairs.

The last two models are the most comprehensive and propose improved models that predict both the tension and compression change phenomena in belt elements.

## 3. LOSS MECHANISMS IN PUSH METAL BELT CVT

Figure 3 shows the ideal metal belt geometry. It is clear from the figure that the expression for the length of the belt can be written as:

$$L = R_p \cdot \beta_p + R_s \cdot \beta_s + 2\sqrt{[a^2 - (R_s - R_p)^2} \ . \quad (1)$$

where:

$$\beta_p = 180 - 2\alpha$$
$$\beta_s = 180 + 2\alpha \quad (2)$$
$$\sin\alpha = \frac{R_s - R_p}{a}$$

The push metal belt CVT transmission ratio can be defined in geometric terms as:

$$I = \frac{R_s}{R_p} \ . \quad (3)$$

Transmission ratio greater than 1 corresponds to the reduction ratio, while transmission ratios lower than 1 corresponds to the overdrive ratio.

In loaded operation, the CVT runs at a slightly reduction speed ratio. This difference is known as the slip ratio between the belt and pulley, $S_R$:

$$S_R = \frac{(I_L - I)}{I} \times 100(\text{percent}) \ . \quad (4)$$

where $I_L$ is the speed ratio at a loaded condition.

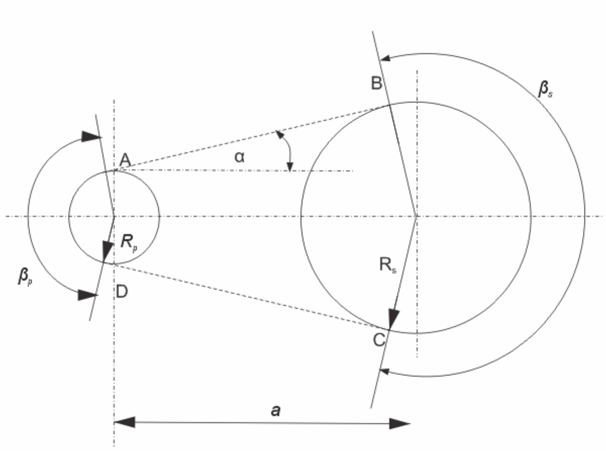


**Figure 3.** *Schematic of ideal metal belt geometry*

As the friction related power losses are one of the main reasons of CVT inefficiency, it is necessary to investigate where the losses are likely to occur. In order to have a power loss a relative motion has to exist between two or more components with resistive force acting between them. Therefore, the following relative motion pairs were indentified in a push metal belt CVT system:

- between the segment side and the pulley
- between the segment and the outermost bands
- between the bands
- between the neighbouring segments

All of these contact pairs with relative motion has to be investigated in order to assess the value of total torque loss.

Friction force between the neighbouring segments as well as the torque losses associated with it can be neglected as assumption can be made that no radial movement is allowed [4].

### 3.1. Friction loss between segment side and the pulley

The friction loss between segment side and the pulley that occurs is based on the relative slipping and contact load between the segments side and the pulley surfaces on both sides of the segment. The noted friction loss $Q_{sp}$ is thus:

$$Q_{sp} = 2 \times \mu_{sp} \times F_{sp} \times V_{sp} \ . \quad (5)$$

where $\mu_{sp}$ is friction coefficient, $F_{sp}$ the normal load and $V_{sp}$ the relative slipping speed. The subscript sp refers to the segment side/pulley contact pair.

The coefficient of friction $\mu_{sp}$ can be calculated from the equation for maximum torque capacity of push metal belt CVT in specific tribological conditions defined by lubricant type ($M_{max}$) [5]:

$$\frac{F_{ax}}{\cos(\lambda)} = \frac{M_{\max}}{2 \times \mu_{sp} \times R_p}. \quad (6)$$

The $F_{ax}$ is the axial clamping force on a pulley and $\lambda$ is half the pulley edge angle. Clamping force on a pulley consists of the static clamp force which is calculated by multiplying the hydraulic pressure by the apply piston area and the centrifugal force generated due to rotation of the piston.

As no slip occurs when the drive torque is at no load, the relative slipping velocity $V_{sp}$ may be calculated from [6]:

$$V_{sp} = \frac{V_s \times S_R}{100}. \quad (7)$$

Relative slipping between the belt and the pulley occurs on the pulley with the smaller active arc with the belt because there are larger gaps between the neighbouring segments on a smaller arc pulley, due to the smaller radius of curvature, and the gaps facilitate slipping [7]. Thus, slipping occurs on the primary pulley in the case of a reduction speed ratio and on the secondary pulley in the case of an overdrive speed ratio. Slipping occurs mainly on a smaller active arc pulley which was experimentally confirmed in [5].

### 3.2. Friction loss between and the outermost bands

The bands in contact with the segment slide relative to the segment shoulder both in the arc part of the belt and in the straight part of the belt. The friction loss $Q_{br}$ based on this phenomena is [4]:

$$\begin{aligned} Q_{sb} &= Q_{sb-a} + Q_{sb-st} \Rightarrow \\ Q_{sb} &= \mu_{sb} \cdot N_{sb-a} \cdot V_{sb-a} + \mu_{sb} \cdot N_{sb-st} \cdot V_{sb-st} \end{aligned} \quad (8)$$

where $N$ is normal load and μ is the friction coefficient. Subscripts *sb-a* and *sb-st* refer to the segment shoulder/outermost band interfaces in the arc part of the belt and in the straight part of the belt.

The normal load $N_{sb-a}$ between the segment shoulder/outermost band in the active arc part is obtained [5]:

$$N_{sb-a} = -\frac{T_1}{\mu_{sb}} \cdot \left[e^{-\mu_{sb}\beta_s} - 1\right] + \frac{m_b V_b^2}{\mu_{sb}} \cdot \left[e^{-\mu_{sb}\beta_s} - 1\right] \quad (9)$$

where $m_b$ is the mass per unit of length of a band.

The normal load $N_{br-st}$ between the segment shoulder/outermost band in the straight part of the belt can be determined as:

$$N_{sb-st} = (T_1 + T_2) \cdot \tan\alpha \quad (10)$$

where $T_1$ and $T_2$ are the band tensions in upper side of the belt and lower side of the belt, respectfully. In order to simplify the estimation of the friction induced losses, band tensions $T_1$ and $T_2$ are considered as constant independent of the drive torque.

The difference in band tensions can be determined from the transmitted torque by:

$$\frac{M}{R_p} = (T_2 - T_1) + C_1. \quad (11)$$

while the constant $C_1$ can be obtained from:

$$C_1 = \frac{\left[(M/R_p) + (\mu_{sb}\mu_{sp}/\mu')(F_{ax}/\cos\lambda) + m_s V_s^2 \mu_{sb}\beta_s\right]}{(1 - \mu_{sb}\beta_s)} \quad (12)$$

The actual values of band tension can be determined in last point of contact arc where $T_2 = 0$.

The relative slipping tangential velocity $V_{sb-a}$ of the segment shoulder to the outermost band in the arc part of the segment can be calculated as:

$$V_{sb-a} = V_s \cdot d_R \cdot \left| \frac{1}{R_s} - \frac{1}{R_p} \right|. \quad (13)$$

where $d_R$ is the distance from the contact circle to the segment shoulder.

Relative slipping tangential velocity $V_{sb-st}$ of the segment shoulder to outermost band in the straight part of the belt can be calculated as:

$$\begin{aligned} V_{sb-st} &= V_s \cdot \frac{d_R}{R_p}, \quad (I < 1), \\ V_{sb-st} &= V_s \cdot \frac{d_R}{R_p}, \quad (I > 1) \end{aligned}. \quad (13)$$

### 3.3. Friction loss between the bands

Band set in of a push metal belt is usually composed of 12 sheets of thin steel. The sum of the friction loss $Q_{bb}$ arising from relative slip between the bands in contact is given by [5]:

$$\begin{aligned} Q_{bb} &= Q_{bb-a} + Q_{bb-st} \Rightarrow \\ Q_{bb} &= \mu_{bb} \cdot \sum_{n=2}^{12} \left[ N_{bb-a}(n) \cdot V_{bb-a}(n) \right] + \\ &+ \mu_{bb} \cdot \sum_{n=2}^{12} \left[ N_{bb-st}(n) \cdot V_{bb-st}(n) \right] \quad (2 \le n \le 12) \end{aligned} \quad .(14)$$

where $N$ is normal load and $\mu$ is the friction coefficient. Subscripts *bb-a* and *bb-st* refer to the band/band contact in the arc part of the belt (defined by warping angle) and the straight part of the belt.

The normal load $N_{bb-a}(n)$ acting on the $n^{\text{th}}$ band in the arc part can be expressed by:

$$N_{bb-a}(n)=\sum_{i=n}^{12}\left\{\begin{array}{l}-\dfrac{T_{1(i)}}{\mu_{sb}}\left[e^{-\mu_{sb}\beta_s}-1\right]+\\+\dfrac{m_b(i)V_b^2}{\mu_{sb}}\left[e^{-\mu_{sb}\beta_s}-1\right]\end{array}\right\}(2\le n\le 12)\ .(15)$$

Constant $T_1(n)$ is given by

$$T_1(n)=\frac{13-n}{12}T_1\quad . \tag{16}$$

The normal load $N_{bb2st}(n)$ between the bands in the straight part of the belt can be written as:

$$N_{bb-st}(n)=\sum_{i=n}^{12}\left[T_1(i)+T_2(i)\right]\cdot\tan\alpha\ \left(2\le n\le 12\right).(17)$$

Relative slipping tangential velocity $V_{bb-a}(n)$ between the bands in the arc part of the belt can be expressed as:

$$V_{bb-a}(n)=V_s t\left|\frac{1}{R_s}-\frac{1}{R_p}+\frac{d_R t}{R_s R_p}+\frac{(n-2)t^2}{R_s R_p}\right|(2\le n\le 12)\ (18)$$

Relative slipping tangential velocity $V_{bb-st}(n)$ between the bands in the straight part of the belt can be expressed as:

$$\begin{array}{l}V_{bb-st}(n)=V_s\cdot\dfrac{t}{R_p},\quad (I<1),\\ V_{bb-st}(n)=V_s\cdot\dfrac{t}{R_p},\quad (I>1)\end{array}\quad . \tag{19}$$

In both expressions $t$ denominates band thickness.

### 3.4. Friction coefficients

Determination of values of $\mu_{sb}$ and $\mu_{bb}$ during the operation of push metal belt CVT is very difficult. Those values can be determined experimentally on a ring to disk tribometer [5] in which frictional conditions (contact pressure and slipping velocity) are similar to frictional condition of an actual CVT. Values of $\mu_{sb}$ and $\mu_{bb}$ are determined according to the values of slipping velocities $V_{bb}$ and $V_{sb}$. If experimental values of friction coefficient are not available they can be selected between 0.08 and 0.14 [5].

Value of friction coefficient primarily depends on clamping force value (which defines the contact pressure), oil type, transmission ratio, slipping velocity and oil temperature. It is observed that lubricants which enable higher overall efficiency tend to lower values of friction coefficient [5]. Moreover, it is also observed that in low contact pressure scenario there is little influence of contact pressure value on friction coefficient. So, to improve the efficiency of push metal belt CVT it is necessary to establish the lower value of the friction coefficient under a lower contact pressure condition.

Although most of the authors use classic Coulomb friction law to describe band segment and band to band contact such models lack the flexibility to adjust the speed relationships of the bands on the driver and driven pulleys to meet the requirement that each band effectively shares part of the load torque. Much better results are obtained via the creep-rate-dependent law [8].

## 4. EFFICENCY INFLUENCE PARAMETERS

As already noted, values of clamping force and transmission ratio, as well as the oil type and its temperature has an effect to friction coefficient, belt slip and thus system efficiency. Noted influence parameters cannot be considered independently as numerous experimental investigations show strong correlations between them. For instance applying higher torque capacity oil to CVT units could contribute to reduce the maximum required pulley clamping force, and thus lower losses in hydraulic pump which is one of the main contributors to overall losses.

The clamping force has to be precisely controlled in order to achieve maximal efficiency. For a given clamping force there is a maximum limit due to friction at which no slip occurs. If this frictional force is overcome then the belt begins to slip over the pulley surface. Further increasing the torque continues to increase the rate of slip. In low load scenario it is desirable to decrease the clamping force and thus improve efficiency.

Push metal belt efficiency is at maximum at transmission ratio equal to 1. For optimal clamping force the decrease of efficiency is roughly the same in reduction and overdrive operating regimes. With increase of over clamping the decrease of efficiency is grater in the reduction regime than in overdrive. The difference gets larger as the clamping force rise.

The oil type affects not only the efficiency, but torque capacity as well. Experimental investigations showed that lower viscosity index oils have a greater torque capacity, but lower efficiency. From the viewpoint of the improvement of transmission efficiency, it is clarified that the priority for the performance of push metal belt CVT should be focused on the higher transmittable torque capacity.

Metal belt CVT operate in boundary lubrication regime. Friction surfaces are practically in direct contact, thus the load is transferred by the surfaces roughness and waviness elements. This causes deformation of contact surfaces which has also effect on efficiency [4].

## 5. CONCLUSIONS

Paper indentifies the reasons of friction related losses in push metal melt CVT. Friction loss is caused by slipping between the belt and pulley, the segment and band, and between the bands, while the slipping between the segments can be neglected.

It also gives review of models for modelling the push metal belt CVT as a tribological system.

A model of assessment of friction related losses in push metal belt CVT is also presented as well as the main influence parameters on efficiency.

## REFERENCES


[1] A. Miltenović, M. Velimirović, M. Banić, M. Milovančević: Modern trends in development and application of CVT, Konstruisanje mašina - Journal of Mechanical Engineering Design, Vol. 11, No1, pp.14-22, 2009.

[2] V. Miltenović, M. Velimirović, M. Banić, A. Miltenović: Increase of Energy Efficiency of Windturbines by Application of CVT, in: *Proceedings of International conference on Gears - Gears 2010*, 04-06.10.2010, Munich, pp. 1095-1106.

[3] N. Srivastava, I. Haque: A review on belt and chain continuously variable transmissions (CVT): Dynamics and control, Mechanism and Machine Theory, Vol. 44, No 1, pp 19-41, 2009.

[4] S. Akehurst, N. D. Vaughan, D. A. Parker, D. Simner: Modelling of loss mechanisms in a pushing metal V-belt continuously variable transmission. Part 1: torque losses due to band friction, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, Vol. 218, No 11, pp 1295-1306, 2004.

[5] K. Narita, M. Priest: Metal–metal friction characteristics and the transmission efficiency of a metal V-belt-type continuously variable transmission, Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, Vol. 221, No 1, pp 11-26, 2007.

[6] C. Morgan, R. Fewkes: Development of a belt CVT fluid test Procedure using the VT20/25E belt box for the DEX-CVT specification, SAE paper 2002-01-2819, 2002.

[7] T. A. Stolarski: Tribology in machine design, Batterworth-Heinemann, Oxford, 1990.

[8] S. Akehurst: An Investigation Into the Loss Mechanisms Associated with a Pushing Metal V-Belt Continuously Variable Transmission, PhD thesis, Department of Mechanical Engineering, University of Bath, Bath, 2001.

[9] K. Narita, J. Deshimaru, M. Kato: The influence of lubricating oil on the performance of a metal V-belt-type continuously variable transmission, in: *Proceedings of JSME International Conference on Motion and power transmissions*, 2001, Fukuoka, pp.730-735.

[10] D. Stamenković, M. Đurđanović: The possibility of static friction coefficient calculation, in: *Proceedings of 5$^{th}$ International Conference on Tribology*, 2005, pp. 465-469.