Key Technologies of the Pushbelt CVT
-Status and New Developments-

Francis van der Sluis 1)  Erik van der Noll 2)  Hendrik de Leeuw 3)

1)-3) Bosch Transmission Technology B.V.
Dr. Hub van Doorneweg 120, 5026RA, Tilburg, The Netherlands (E-mail: francis.vandersluis@nl.bosch.com)

Received on July 17, 2012
Presented at the JSAE Annual Congress on May 23, 2012

ABSTRACT: In recent years, the use of Continuously Variable Transmissions (CVT) in automotive applications has seen a strong worldwide growth. With a 10% annual growth rate, the CVT based on the pushbelt principle takes the lead. This growth, that is foreseen to intensify the coming years, is driven by several Key Performance Indicators (KPI). A first KPI is the high level of comfort, which for many consumers defines the benchmark. A second KPI is the excellent fuel economy of CVT equipped drivelines that responds to the present needs in the market. This paper discusses the efforts to support these KPI. While giving an update of the actual status on fuel economy, efficiency, power density and NVH, new developments are highlighted that enable further optimizations of the pushbelt CVT in an ever-demanding market.

KEY WORDS: Power transmission, CVT/Variator, pushbelt, fuel consumption, efficiency, NVH, power density, cost

1. Introduction
In recent years the use of pushbelt CVTs has shown strong growth that is foreseen to rise further with the decision of several OEMs to expand the use of CVT in markets around the world [1,2,3]. As the market leader for pushbelts, BOSCH responds to the increasing demand in volume and variety. In 2011 it produced over 3.5 million pushbelts on a total of 5.9 million CVTs. For 2015, a doubling is expected on a total of 11 million CVTs. In 2012 a new 28/12 and 24/6 pushbelt will be introduced.

The success of the pushbelt CVT is based on several Key Performance Indicators (KPI). This paper discusses how the KPI comfort, fuel consumption, efficiency, NVH, power density and cost contribute to the success, now and in the future.

2. Comfort
As a traditionally subjective KPI, transmission comfort is difficult to characterize. Driver type, region and culture play a role. Data can be found at independent institutes, expert reviews and OEMs with multiple transmission types in their portfolio.

In general, the continuous drive without jerk from torque interruptions or sudden engine speed steps rewards CVT equipped drivelines with the benchmark position (figure 1) [4].

3. Fuel consumption and efficiency
3.1. The CVT in the driveline
Variability enables the driveline to remain in the most efficient point of operation which increases engine efficiency that, with an average value of 20-30%, constitutes the number one driveline loss. Variability is especially beneficial at lower vehicle speeds. CVT based front wheel drivelines thereby form a benchmark on fuel consumption in markets where moderate speed cycles are in use. Figure 2 shows the 2012 data for Japan. Compared with 2009 data [5], it reveals a large scale replacement of AT by CVT.

Also recent dry dual clutch transmission applications (dDCT) show acceptable fuel consumption. As DCTs are mainly linked to DI/charged engines, a direct comparison with CVT applications, typically linked to PFI/naturally aspirated engines, is difficult.

Instead of variability, DCT based drivelines benefit from a somewhat better transmission efficiency as explained in figure 3.
To optimally use variability, the ratio coverage of the CVT variator is steadily increased. The recent introduction of a value of 7 [-] (2) in combination with the new 28/12 belt is enabled by belt and variator measures that will be addressed in section 5.

For a CVT, small measures in the variator directly contribute to an increase in ratio coverage which is contrary to stepped transmission types where the benefit of an inevitable increase of the number of steps must be distributed over the desired increase of ratio coverage and the decrease of the variations in engine speed (figure 3).

Figure 3 shows that the losses in a CVT are the second dominant effect on fuel consumption. They provide a strong potential for improvement as will be explained in the next section.

3.2. CVT efficiency

Figure 4 shows the loss distribution of a CVT. The hydraulic actuation system, supplied by the engine driven pump, and the variator are the main losses and will be discussed next.

3.2.1. Losses in the actuation system

A first focus is the decrease of clamping force in the variator. This measure reduces both the frictional loss in the variator and the loss in the pump system that provides the clamping pressure (figure 4). For a primary input torque $T_{pri}$, primary belt running radius $r_{pri}$ and pulley angle $\lambda$, the required clamping force $F_{cl}$ is:

$$F_{cl} = \frac{T_{pri} \cdot \cos(\lambda)}{2 \cdot r_{pri} \cdot \mu_{est}}$$

Bosch has reported the option to reduce the clamping safety $S_f$ by the feature slip control (16). The increase of the actual coefficient of friction ($\mu$F or $\mu$) between belt and pulley is another option.

Since many years, the advised CoF value for released belts is $\mu_{est} = 0.09$. Continuing developments in the belt-pulley interface and oil have now led to the value $\mu_{est} = 0.10$ for the new 28/12 belt. This means a 10% clamping force reduction. Typically, this force consists of a pressure force, where a pressure $p_1$ works on a cylinder area $A$, and a force $F_a$, containing the centrifugal effect of the oil and, in case applicable, a pulley spring. For identical parameters of equation 1, the new pressure $p_2$ is obtained by:

$$\frac{F_{cl2}}{F_{cl1}} = \frac{p_2 \cdot A + F_a}{p_1 \cdot A + F_a} \Rightarrow p_2 = p_1 \frac{\frac{F_{cl1}}{\mu_2}}{1 - \frac{\mu_1}{\mu_2}} \frac{F_a}{A}$$

The new pressure $p_2$ and the related losses in the hydraulic system decrease more than the ratio between the CoF values. Figure 5 shows the typical pressure reductions for the examples $\mu = 0.10$ and $\mu = 0.16$. At low torque, the benefit is largest. It is therefore important that the minimum pressure of the hydraulic system does not restrict the corresponding low pressures.

A second focus is the oil pump system. The displacement volume of present engine driven pumps is sized to cover the most critical flow condition. These pumps thereby supply a wasted surplus of oil in all other situations.

The use of balanced vane pumps allows one half of the pump to be switched off by short circuiting the outlet and the inlet of the pump half. At higher engine speeds the required power to drive the pump is significantly reduced. Current designs use an on/off solenoid to actively switch the pump from full to half delivery and vice versa (figure 6) (7, 8).
The point of switching is determined from a model based estimation of the actual flow that the pump is supplying to the system. Inaccuracies in the estimation lead to delays in the time of switching and related losses.

A passive switch that uses the lowest pressure in the CVT (for instance lubrication pressure) as the control signal, responds to the actual CVT demand by switching at the correct time. The reduction in actuation losses lies at 10-20%.

The use of a variable oil pump is a further improvement towards a more ideal on-demand oil supply.

Fig. 6 Pump system measures to improve flow on-demand

The growing availability of components for electrification has led to the development of new systems where the power reduction can be stretched much further. Figure 7 shows an example of a variator that is supplied on demand by a small electrically driven pump that is boosted by the main pump. The main pump supplies flow at a lower, eg. lubrication pressure. The engine power that over a typical cycle is required for CVT actuation purposes (figure 4) decreases by approximately 35%.

Fig. 7 Actuation power reduction by electrification

3.2.2. Losses in the variator

Figure 8 shows the loss distribution of the variator. The main losses are the frictional contacts between belt and pulleys and between elements and rings inside the belt (figure 9). This section discusses some short to longer term measures for improvement.

Fig. 8 Variator loss distribution and impact of clamping force.

Fig. 10 Effect of CoF value $\mu=0.10$ on variator efficiency.

The previous result was obtained for a variator without specific measures. Figure 11 shows the results for an experimental variator. The pulley surfaces of this variator were fitted with a new technology to investigate the effect of a significant CoF increase to $\mu=0.16$. Instead of a safety $S_f=1$, a safety $S_f=1.3$ is used for the comparison. At this safety, the variator reaches an efficiency of 96.2% in OD at a very low clamping force. Despite the high robustness of the element flanks, the level of wear that occurred during the test is not acceptable at present. A durable increase of the CoF towards values above $\mu=0.10$ remains a topic under investigation.
This section shortly discusses the main source of excitation in the variator and methods to realize further improvements.

4.1. Polygon effect

The low noise level of the pushbelt primarily results from the small pitch distance between the elements. The small distance increases the passing frequency of the elements at the pulleys. Furthermore, it reduces the so-called polygon effect. This effect can be characterized by the minimum and maximum tangential element speeds that are geometrically defined by the running radius of the belt $r$ and the pitch $p$ (figure 13):

$$\Delta v = \frac{v_{max} - v_{min}}{v_{max}} \cdot 100\% \approx \frac{r - \sqrt{r^2 - (p/2)^2}}{r} \cdot 100\%$$ (3)

The polygon effect causes transversal and longitudinal variations in element speed at the pulley entrance and the pulley exit that subject the belt to local accelerations and decelerations. These variations increase progressively for larger pitch distances.

4.2. Noise optimization

Based on the experiences with early block- and pushbelts, values for $\Delta v$ that lie above 0.07% are considered unacceptable (10). This particular value corresponds with the element thickness of the very first pushbelt that was released for production by Van Doorne’s Transmissie BV in 1986.

The present pushbelts, with their element thickness of 1.5 and 1.8 [mm], show a very small polygon effect in the ultimate running radii (figure 13). A further reduction of the effect and its consequences is possible by applying a variable element thickness. This spreads the frequency of the elements that enter the pulley (polygon frequency) over a wider range (11).

4.3. Slenderness ratio

The low noise level of the pushbelt variator makes it difficult to find and characterize the exact sources of noise. Besides further measures for optimization, the research therefore also focuses on new technologies that help to visualize the noise behavior of the variator. One of them is a technology called Planar Nearfield Acoustic Holography (PNAH) (12).
With this technology, sound pressure levels originating from the source are visualized by an acoustic camera. This camera consists of multiple microphones in a planar grid that is positioned very close to the source plane. The technology was used during the development of the new 28/12 pushbelt.

To first identify the work point settings for the visualization, speed sweeps were performed on the variator test rig. For this purpose, the variator was prepared with acceleration sensors that were mounted in the secondary pulley\(^{13}\). Figure 14 shows the spectrogram for this location that resulted from a speed sweep at a variator speed ratio of 0.7 [-].

**Fig. 14  Spectrogram secondary pulley accelerations.**

At the secondary pulley speed of 1832 [rpm], an acceleration peak is found that by its frequency can be connected to the polygon effect. The element passing frequency apparently crosses a natural frequency of 5000 [Hz] at this point.

Figure 15 shows the acoustic hologram (sound pressure) at these settings. At the fixed sheave of the secondary pulley, sound radiation hotspots can be recognized that identify this specific location as a main source of noise.

**Fig. 15  Acoustic hologram 28/12 belt at ratio 0.7 (5000Hz).**

By combining the theoretical understanding of the working principle with the experimental techniques of acceleration measurements, PNAH, and modal analysis, the source of the noise and its excitation mechanism were identified. From this knowledge, specific measures can be derived to further improve the noise performance of the pushbelt variator at the source and to prevent expensive noise countermeasures outside the variator.

### 5. Variator Power Density

Variatior power density \((PD)\) is defined as the maximum power \(P_{\text{max}}\) that, for a certain ratio coverage \(RC\) and over a certain belt life (factor \(f_{\text{dur}}\), can be transferred through a certain variator space envelope \(V_{\text{var}}\). The accompanying formula is:

\[
PD = \frac{P_{\text{max}} \cdot RC \cdot f_{\text{dur}}}{V_{\text{var}}}
\]

The durability factor typically depends on the fatigue limit of the ring material and the related number of cycles before failure. The space envelope is determined from the characteristic dimensions of the variator like the center distance between the pulleys, the pulley diameters, the width of the pushbelt elements and the maximum stroke of the pulley cylinders.

The latest pushbelt introductions show that the power density value of the variator has been significantly increased by the following parameters:

- reduced element width for 28/12 belt (30 → 28: 4%)
- increased ratio coverage for 28/12 belt (6 → 7: 17%)
- increased belt-pulley CoF for 28/12 belt (0.09 → 0.1: 8%)
- reduced number of rings for the 30/10 belt and the 24/6 belt (up to 25%)
- reduced clamping safety \(S_f\) (1.3 → 1: up to 16%)

Some of the product changes that were required to achieve these improvements will be addressed in the next section.

#### 5.1. Ring design

Belt life is dominated by the fatigue life of the maraging steel rings. In the variator, the rings are subjected to tensile, bending and contact stresses. In order to increase power density, the fatigue properties of the Durimphy ring material (18Ni 9Co 5Mo 0.5Ti balance Fe) needed to be improved.

From failure mode evaluations it became clear that small nonmetallic TiN inclusions, originating from the titanium content in the base material, were responsible for fatigue crack initiation. This initiation was located at the boundary between the core and the nitrided zone as shown in figure 16\(^{14}\).

**Fig. 16  Ring fracture surface analysis and failure mode.**
To prevent the formation of TiN inclusions, titanium was removed from the material. The cobalt content was increased to compensate for the precipitation hardening function of titanium.

During typical accelerated overload fatigue tests, the new Phytime material (18Ni 16.5Co 5Mo balance Fe) increases belt durability by a factor 5 as shown in figure 17.

The main failure mode of the rings has changed. Crack initiation now occurs at tiny imperfections on the ring surface that in some cases can be traced back to certain process steps in the production process of the rings. The prevention of such imperfections coming into existence is a next target. A first focus was the development of a new thermal deburring process to avoid the forming of imperfections during the current stone-tumbling process. A second focus is the development of a new hardening and nitriding process that treats complete ring sets in a single set-up to avoid the handling and treatment of individual rings.

The durability improvements can be exchanged with the other power density parameters in equation 4. The increased fatigue limit of the ring material enables the discussed improvement of power capacity, space envelope and ratio coverage.

The reduction of ring load is another way to improve the durability of the pushbelt. The clamping force is the main contributor to the tensile stresses in the rings. The increase of the coefficient of friction between belt and pulley for the new 28/12 belt to a value $\mu = 0.10$ enables a reduced clamping force that decreases tensile stresses in the rings by 4%. The reduced stress level results in a power density increase of 8%.

5.2. Element design

Also the parameters of the element can deliver a contribution to the improvement of power density. In this section, the center of gravity, the flank angle and the clearance angle are addressed.

5.2.1. Element center of gravity

The position of the center of gravity (CoG) of the element affects its dynamic behavior. In the variator, the elements pass through several stages of kinetic energy (figure 18). The relevant parameters are the element mass $m_d$ and inertia $J_d$, the running radius of the CoG $r_{CoG}$ and rocking edge $r$ and the rotational speeds (figure 19).

The energy of the element while at the wrap angles of the pulleys $E_{k,p}$ can be calculated with:

$$E_{k,p} = \frac{1}{2} \left( m_d \cdot r_{CoG,p}^2 \right) \omega_p^2$$

At the straight parts, the kinetic energy is:

$$E_{k,s} = \frac{1}{2} \left( m_d \cdot r_{s,p/s}^2 \right) \omega_s^2$$

The kinetic energy of the elements is lowest at the straight parts. It is highest at the smallest running radius. The main reasons are:

- the CoG that lies above the rocking edge ($r_{CoG,p} > r_{p,s}$)
- the higher rotational speed of the pulley with the smallest running radius.
- the rotational energy of the element at the pulleys.

At the entrance of the straight loose part, the surplus of kinetic energy serves as a booster. Here, at the exit of the secondary pulley, the longitudinal play or endplay between the rings and the elements in the belt needs to be overcome by an element speed that is locally higher than the average belt speed.

For belts suitable for higher rotational speeds, like the new 28/12 belt, this energy needs to be larger. The CoG is therefore positioned higher in the element by increasing size and mass of the element head. The inertia $J_d$ and mass $m_d$ of the element (and belt) were maintained at approximately the same level.
Figure 19 shows the difference in kinetic energy (E_{k.s}-E_{k.l}) that occurs between an element at the secondary wrap angle and an element at the straight part. Both the results of the 30/12 belt and the new 28/12 belt are given. The higher CoG position of the 28/12 belt increases the high speed capability of the variator. This feature thereby improves the power density of the variator.

5.2.2. Element flank angle

The difference in angle between the element flank and the pulley flank determines whether the contact between the two is concentrated at the upper area or lower area of the element flank. A contact at the lower area of the element flank is undesirable for the following reasons (figure 20):

- the bending torque and stress over the element width can become too large which may lead to an early element and belt failure.
- the bending torque and stress over the thickness of the element is too large which also may lead to an early element and belt failure.
- the bending torque over the thickness of the element leads to an undesirable increase of the pitch behavior of the element.

These effects are prevented when the flank angle of the element is made slightly larger than the flank angle of the pulley. This feature is shown in figure 20 (\(\lambda_r \neq 0\)). The clamping load on the element flank shifts to the upper area of the flank.

The smaller bending radius must also be allowed by the element geometry. The previous element design put a constraint on the minimum running radius by its clearance angle (figure 20). For this purpose, the clearance angle of the new element is increased to enable the belt to physically run at the smaller running radius.

5.3. Pushbelt environment

Besides measures on a pushbelt level, also measures in the surrounding components can have a large impact on the power density of the variator. The BOSCH Group therefore co-operates with CVT manufacturers to identify the main parameters and where possible, take the appropriate measures for improvement.

Within the responsibility of the CVT manufacturer and its suppliers, the following topics have an impact on power density:

- the applied clamping safety \(S_c\) and control method\(^{(5, 9)}\)
- the lay-out of the (hydraulic) actuation circuit\(^{(5, 9)}\)
- oil parameters\(^{(15)}\)
- pulley surface and geometry parameters\(^{(15, 16, 17)}\)
- variator stiffness\(^{(18)}\)
- variator bearings\(^{(19)}\)

Obviously, most of these topics also influence other KPI.

6. Cost

6.1. Transmission cost and transmission price

Compared to other transmission types, the manufacturing cost of a pushbelt CVT is relatively low. This is the result of the specific design of the CVT that contains a smaller number of parts. Besides this, the more expensive component groups are relatively few in number. This results in a CVT cost that is approximately 75% of the cost of a DCT (figure 21).

The lower cost is reflected in the add-on price that consumers pay to upgrade from a manual to an automatic transmission.
Figure 22 shows the result of an investigation of these add-on transmission prices for the market in Europe/Germany. The shown add-on prices include a value added tax (VAT) of 19%.

Although the Automated Manual and 4-speed AT provide a lower add-on price, the gradual comfort and fuel economy driven worldwide replacement of these transmission types by either 6AT, DCT or CVT, leads to the conclusion that the actual comparison should be made between the latter three transmission types.

From these main competitors, the pushbelt CVT shows the lowest add-on transmission price. The difference even can be a little bit larger since the base transmission for the pushbelt CVT in most cases is a 5-speed manual. For the other main competitors, the base transmission typically is a 6-speed manual.

Fig. 22 Consumer add-on transmission price Europe/Germany.

For the future, the required demand for variability is expected to enhance the benefit of the CVT. The cost of relatively small changes to variator dimensions is expected to be significantly smaller than the cost that will accompany the required variability improvement of stepped transmission types. The cost of measures to improve CVT efficiency will only slightly reduce this benefit.

6.2. Pushbelt cost

The cost of the pushbelt forms a minor share in the total cost of the CVT transmission. Measures are considered to reduce the cost of this component.

The cost of the pushbelt is partly determined by the number of applied rings. The exact number of rings that is required to fulfill the application therefore is important.

On the other hand, the improvement of ring durability by the development of new ring material and production processes has reduced the number of rings that is required. This has enabled the development of downsized variants of already existing belts that have a smaller number of rings. Examples are the 30/10 belt (originally 30/12) and the new 24/6 belt (originally 24/9).

7. Conclusions

The following conclusions are drawn:

- In 2012 a new 28/12 pushbelt will be introduced that enables a benchmark CVT ratio coverage of 7 for engine displacements above 2ltr. A new 24/6 belt will be introduced for smaller engine displacements.
- The increasing market share of the pushbelt CVT results from the high score on the Key Performance Indicators comfort, fuel consumption, efficiency, NVH, power density and cost.
- The BOSCH group dedicates itself to a continuing effort to improve the KPI of the pushbelt in order to keep delivering benchmark products at the highest possible quality that contribute to the success of the Continuously Variable Transmission.

References

(1) Seabaugh, Honda says its cars will be industries most fuel efficient in three years. (www.Motortrend.com 2011)
(2) Press release, Nissan’s advanced technologies – PURE DRIVE (2011)
(3) Greimel, Two new Hyundai transmissions boost MPG (Autoweek 2010)
(4) Scholz, Automatisierte Schaltgetriebe, (AutoMotorSport magazine (Germany) 2011)
(5) van der Noll et al., Innovative self-optimizing clamping force strategy for the push belt CVT (SAE 2009-01-1537)
(6) Waku, Newly developed CVT for compact vehicles with start-stop system (CTI 2011)
(7) van der Sluis et al., The two-stage pushbelt CVT (CTI 2002)
(8) Ohashi et al., Development of high-efficiency CVT for luxury compact vehicle (SAE 2005-01-1019)
(9), van der Sluis et al., Efficiency optimization of the pushbelt CVT (SAE 2007-01-1457)
(10) Roovers, Noise reduction in pushbelt CVT’s (VDI Berichte Nr. 977, 1992)
(11) Tsukuda et al., Toyota’s new belt CVT for 1.3 ltr FWD cars (SAE 2006-01-1305)
(12) Scholte, Improved source localization techniques in planar nearfield acoustic holography (2004)
(13) Hodate et al., Development of a Method for Analyzing CVT Casing Radiated Noise Induced by Belt Excitation Forces (SAE 2005-01-1460)
(14) Pennings et al., New CVT push-belt design featuring a new maraging steel to cover all FWD powertrains (SIA 2005)
(15) Yamazaki et al., Research on improvement of transmission efficiency by improving the friction coefficient between element and pulley of the belt CVT (JSAE 20075644)
(16) Ichijo, Research of V-surface angle for metal pushing V-belt CVT (JSAE 20115360)
(17) Kruessmann, Driving CVT into a new area (VDI Getriebe in Fahrzeugen 2011)
(18) Lee et al., Study on power loss in pushing V-belt CVT (FISITA F2006P161)
(19) Takemura et al., Development of Long Life Pulley-Supporting Bearing for Belt-CVT (SAE 2005-01-0873)