A New Pushbelt to Enhance CVT Technology

- Advances in Efficiency, Power Density and Robustness -

Francis van der Sluis 1)  Dirk Twisk 2)  Michel van de Sanden 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 June 20, 2016
Presented at the JSAE Annual Congress on May 25, 2016

ABSTRACT: Continuously Variable Transmission (CVT) technology is spreading. Fuel consumption is a key driver. Next to driveline synergy optimizations the improvement of CVT components is mandatory to comply with increasing demands. Pushbelt development within Bosch focuses on measures that support both directions. The latest pushbelt release contains measures that improve variator efficiency and ratio coverage at increased torque capacity and wear resistance. This paper explains these measures and their working mechanisms. The newly developed pushbelt combines enhancements in fuel efficiency and power density in a recently released transmission where it supports the world’s largest CVT ratio coverage of 8.7 [1].

KEY WORDS: Power transmission, continuously variable transmission (CVT), mechanism, Pushbelt, Efficiency [A2]

1. Introduction

In 2016, the Continuously Variable Transmission (CVT) has captured a sizeable share of the market. From the initial main market Japan the technology is spreading over the world. Strong growth is seen in the USA and China. In 2015, four out of the five best selling cars in the USA used a CVT [2]. A key driver is the good fuel economy that positions many vehicles with a CVT at the top of the fuel economy lists.

A major part of the benefit is achieved by the synergy between the CVT and the engine. Continuing engine optimizations create new requirements for the CVT. Efficiency, power density and NVH [3] improvements in components and subsystems are mandatory to comply with increasing demands.

As the main producer of pushbelts, Bosch concentrates on the improvement of the pushbelt variator. This heart of the CVT influences many other parts that define CVT characteristics.

This paper discusses measures in efficiency and power density for the latest 24/9 pushbelt release that are based on the latest insights in characteristics of the pushbelt and its interaction with the surrounding components.

2. Market status and requirements

2.1. Efficiency

The efficiency of the driveline ultimately translates into fuel economy and is affected by many factors. The fuel economy of CVT equipped vehicles shows competitive values in the markets and is an important reason for the growth in CVT market shares.

Figure 1 shows homologated fuel economy numbers of vehicles with automated transmissions compared to their manual option in the US market. CVT values usually lie above the manual option. Over the relevant cycle, most vehicles that are equipped with a CVT consume less fuel than the same vehicle with a manual transmission.

The increase of variator efficiency is a requirement for further improvement. The reduction of friction losses that exist in the pushbelt or between the pushbelt and the pulleys or in bearings and in hydraulic seals are areas of attention. Measures that reduce the applied clamping force and hydraulic pressure in the variator improve variator efficiency and reduce actuation power.

Driveline requirements are equally relevant. Ratio coverage but even more variability, the freedom to operate the driveline in any ratio between the LOW and OD ratio, are important.

Especially over the last decade, the requirement to improve fuel economy has initiated an increase in transmission ratio coverage. Values close to 10 are in the market today. The optimal ratio coverage that delivers best fuel economy depends on driveline and cycle characteristics and roughly ranges from 6 to 9.

A large ratio coverage helps to reduce engine speed and to improve drive off performance. New drive cycles like the WLTC with higher vehicle speeds partly drive the increase of ratio coverage. Transmissions with somewhat higher, often speed dependent, internal losses, benefit more from this increase. A reason why Automatic Transmissions (AT) show a stronger increase in ratio coverage by the number of gears. However, for these transmission types a larger ratio coverage does not proportionally improve variability or fuel economy (figure 1).
In between the fixed steps variability remains lacking while the additional components can increase losses. Larger and more costly steps in ratio coverage are therefore needed to reach significant effects.

The pushbelt variator shows its potential here. A similar ratio coverage increase offers a larger and more cost effective increase in variability (figure 2). The ideal measure lies in the reduction of the minimum running radii of the pushbelt. This prevents an increase in space envelope and makes the improvement of CVT ratio coverage a valuable approach for improvement (1, 4, 5).

Fig. 2 Trend between ratio coverage and transmission cost.

2.2. Power Density

The power density of the variator plays a crucial role in a CVT. Power density is defined by the maximum transferred power, ratio coverage, desired durability and space envelope (6).

Fig. 3 Power density trend of the CVT with 24mm pushbelt.

Figure 3 shows the trend in power density for CVTs with the 24mm pushbelt. The reference in this figure is the first application that was introduced in the market in 1987. The larger increments in power density are related to the following developments:

- Development of optimized pushbelt loop materials (7, 8)
- Development of new loop manufacturing techniques (6, 7)
- The use of the dual stage concept that enables large CVT ratio coverage with relatively small variators (1)

The increase of power density was thus mainly realized by the optimization of the fatigue critical loops in the belt. Optimization of the other component, the element, brings further benefits. Measures that improve torque capacity like element strength and flank wear resistance thereby are considered.

In this paper the pushbelt nomenclature as shown in figure 4 is followed.

Fig. 4 Pushbelt nomenclature.

3. A New Belt Design

In response to the requirements discussed in the previous chapter Bosch developed a new 24mm pushbelt. Especially the element design shown in figure 5 contains new features like changes in the rocking edge and flank design.

Fig. 5 Reference and new element design with flank detail.

In the following chapters more detail is given about these new features with which the following benefits were achieved:

- An increase in variator efficiency in OD of 1%
- An increase in torque capacity of 4 to 9%
- An increase in flank wear resistance of 46%

4. Improvement of Efficiency

In order to understand the background of efficiency, model and experimental activities are used to create insight in loss mechanisms. This chapter describes how loss modeling supports the quantification of orders of loss. The combination with experimental results under varying product parameters provides confirmation which losses occur in the variator and what measures are suitable to improve efficiency.

4.1. Basic Loss Modeling

Bosch uses advanced modeling techniques to simulate the behavior of the pushbelt variator. Although these techniques provide accurate efficiency estimations (figure 6), the connection between obtained test results and the responsible mechanisms and parameters can still be complicated.

Fig. 6 Simulation of pushbelt variator efficiency in OD.
From advanced models simple loss categories can be derived that depend on input speed \( \omega_i \) and input torque \( T_{p} \). For a fixed ratio and safety a simple loss and efficiency equation can be deduced that contains five components with constants \( c_{m1}, c_{m2}, c_{s1}, c_{s2}, c_{s3} \). Table 1 lists the background of these components:

\[
P_{loss} = \frac{c_{m1}}{T_{p}} \cdot \omega_i + c_{m2} \cdot T_{p}^2 \cdot \omega_i^2 + c_{s1} \cdot \omega_i + c_{s2} \cdot \omega_i^2 + c_{s3} \cdot \omega_i^2
\]

\[
\eta = 1 - \frac{P_{loss}}{T_{p} \cdot \omega_i}
\]

\[
\eta = 1 - c_{m1} \cdot \omega_i \cdot T_p \cdot \frac{c_{s1} \cdot \omega_i}{T_p} + c_{s2} \cdot \omega_i^2 - c_{s3} \cdot \omega_i^2
\]

| No | Power loss | Main background |
|----|------------|----------------|
| 1  | \( c_{m1} \cdot T_{p} \cdot \omega_i \) | Mechanical loss - Coulomb friction |
| 2  | \( c_{m2} \cdot T_{p}^2 \cdot \omega_i^2 \) | Mechanical loss - friction under deformation |
| 3  | \( c_{s1} \cdot \omega_i \) | Spin loss - zero error measuring device |
| 4  | \( c_{s2} \cdot \omega_i^2 \) | Spin loss - viscous shear and dynamic effects |
| 5  | \( c_{s3} \cdot \omega_i^2 \) | Spin loss - acceleration/centrifugal effects |

The loss in the element-loop contact is used as an example. In this contact, schematically shown in figure 7, friction exists between the elements and the inner surface of the inner loops. This friction is dominant at the small wrapped arc.

![Fig. 7 Forces acting in the element-loop contact.](image)

The contact is subject to a normal force by clamp, push and centrifugal forces on the element and by slip that results from the differences in running radii of the element and the inner loops. With the normal force \( F_n \), slip speed \( v_{slip} \) and Coulomb Coefficient of Friction (CoF) \( \mu \), the loss in the contact can be described by:

\[
P_{\text{element-loop}} = \mu \cdot F_n \cdot v_{slip}
\]

\[
F_n = F_p + F_c
\]

The normal forces related to clamping (\( F_c \)) and pushing (\( F_p \)) are proportional to the applied clamping force on the belt \( F_{cl} \). The centrifugal force \( F_{cen} \) is a third component. The centrifugal force on the belt is adjusted based on the input torque \( T_{p} \), primary running radius \( r_p \), clamping safety \( S_i \), pulley angle \( \lambda \) and the estimate of the Torque Capacity Factor between belt and pulley \( \text{TCF}_{est} \):

\[
F_{cl} = \frac{T_{p} \cdot \cos(\lambda)}{2 \cdot r_p \cdot \text{TCF}_{est}} \cdot S_i
\]

At a constant speed ratio, safety and Torque Capacity Factor, the clamping force is proportional to input torque. In that case the forces \( F_c \) and \( F_p \) also are proportional to input torque:

\[
F_c \propto T_p
\]

\[
F_p \propto T_p
\]

The centrifugal force is estimated by the element mass \( m_{elem} \), running radius of its center of gravity \( r_{elem} \) and pulley speed \( \omega_{pul} \). At a constant ratio, pulley speeds are a function of input speed \( \omega_i \). The centrifugal force over the wrapped arc can be estimated by:

\[
P_{\text{elem}} = \sum m_{elem} \cdot \omega_{pul}^2 \cdot \omega_i^2
\]

\[
F_{cen} \propto \omega_i^2
\]

At a constant speed ratio, the slip speed \( v_{slip} \) in the element-loop contact in a first approach is proportional to input speed:

\[
v_{slip} \propto \omega_i
\]

For a fixed coefficient of friction and with constants \( c_{m1} \) and \( c_{s1} \), the equations (2) to (6) can be combined to:

\[
P_{\text{element-loop}} = c_{m1} \cdot T_{p} \cdot \omega_i + c_{s1} \cdot \omega_i^2
\]

This relationship contains a 1st order mechanical loss which depends linearly on input torque and input speed and a 3rd order spin loss that only depends on input speed.

By fitting equation (1) on a set of efficiency measurements by least squares optimization, the values of the constants and thereby the size of the individual loss contributions can be determined:

\[
\text{min}_{c_i} \left\| \eta(c_i) - \eta_{\text{measured}} \right\| \rightarrow c_i = \begin{bmatrix} c_{m1} & c_{m2} & c_{s1} & c_{s2} & c_{s3} \end{bmatrix}
\]

A comparison between the contributions of belts with various hardware variations confirms the effects and mechanisms. In the next section an example is given.

4.2. Measures to Improve Efficiency

The new 24/9 belt contains the following efficiency measures:

- Reduction of Rocking Edge – Saddle Distance
- Clamping force reduction by the increase of the Torque Capacity Factor (TCF)

4.2.1. Reduction of Rocking Edge – Saddle Distance

The decrease of slip between elements and loops in a pushbelt variator is a good way to reduce losses. The main slip typically occurs at the small wrapped arc and results from the difference in running radius of the elements and the inner loop. This distance is referred to as the Rocking Edge - Saddle Distance \( \text{RESD}_{var} \) and is defined by the element geometry shown in figure 8.

![Fig. 8 Parameters rocking edge – saddle distance.](image)
Relevant element dimensions are \( \text{RESD}_{\text{el}} \) and the Rocking Edge Radius (RER). In the wrapped arcs of the variator they define \( \text{RESD}_{\text{var}} \) or the radial distance between the element-loop contact and the contact point between elements. The contact point between elements determines element or belt speed.

4.2.2. Test results

Several pushbelt variants were made to check the influence of changes in \( \text{RESD}_{\text{el}} \) and RER. The efficiency of these variants was measured on a variator test rig as shown in Figure 9.

Figure 10 shows measurement and simulation results in OD at 1500rpm for a 24mm variator. All specimens were prepared by processes that are capable for mass production. Dependent on the design and process, significant efficiency improvements can be realized. The new belt with a reduction of \( \text{RESD}_{\text{var}} \) of 0.2mm reaches an efficiency improvement of 0.2%.

![Variator efficiency measurement set-up.](image)

**Figure 9** Variator efficiency measurement set-up.

The obtained efficiency values also include the losses from the belt box and test rig components like flexible couplings, splines, bearings and seals and can therefore not be compared with values from other sources. In model evaluations such losses are included.

Figure 10 shows measurement and simulation results in OD at 1500rpm for a 24mm variator. All specimens were prepared by processes that are capable for mass production. Dependent on the design and process, significant efficiency improvements can be realized. The new belt with a reduction of \( \text{RESD}_{\text{var}} \) of 0.2mm reaches an efficiency improvement of 0.2%.

![Variator efficiency increase OD as a function of \( \text{RESD}_{\text{var}} \).](image)

**Figure 10** Variator efficiency increase OD as a function of \( \text{RESD}_{\text{var}} \).

The simulations predict the improvement quite well. The combination of the simple loss model and the test results provides further insight how the changes affect individual loss mechanisms.

Figure 11 gives an example of the loss estimations for the reference belt (upper graphs) and the belt with \( \text{RESD}_{\text{var}}=0.2 \text{mm} \) (lower graphs). The colored areas in the graphs represent the efficiency losses for each order of loss. The markers present the measured efficiency values. The least squares optimization provides an acceptable match between the simple loss model and the measurements.

![Orders of loss for reference and \( \text{RESD}_{\text{var}}=0.2 \text{mm} \) belt.](image)

**Figure 11** Orders of loss for reference and \( \text{RESD}_{\text{var}}=0.2 \text{mm} \) belt.

1st Order mechanical loss associated with Coulomb friction shows a strong reduction for the adapted pushbelt. The driving mechanism is the slip reduction between elements and loops that reduces this torque dependent loss.

2nd Order mechanical loss remains identical, a confirmation that the design change does not affect deformations in the variator.

1st Order spin loss is small in both cases, an indication that the zero-error in the torque measurement also is small.

2nd Order spin loss shows an increase. The small \( \text{RESD}_{\text{var}} \) of 0.2mm reduces the internal damping and losses in the belt but thereby increases dynamic effects. For the new 24\%/9 pushbelt this effect is negligible due to the smaller change of \( \text{RESD}_{\text{var}} \).

3rd Order spin loss reduces especially with speed. The driving mechanism again is the reduction of slip between elements and loops. It now works on the centrifugal component of the normal force in this contact. This mechanism has a larger effect for combinations of high speed and low torque.

4.2.3. Process boundaries

An \( \text{RESD} \) modification requires a review of the production process of the elements. At this moment a fine-blanking process is used that creates a die roll at the saddle edge (figure 8). This die roll prevents small \( \text{RESD}_{\text{el}} \) values. Table 2 lists the options to solve this issue. The new 24\%/9 element falls under option 1. The other options are under investigation for future belt releases.

![Table 2 RESD options and efficiency benefit (24mm belt).](image)

**Table 2** RESD options and efficiency benefit (24mm belt).

| Option | \( \text{RESD}_{\text{var}} \) [mm] | process change | product change | Efficiency benefit [%] |
|--------|---------------------------------|----------------|----------------|-----------------------|
| 1      | 1.2-1.0                         | no             | \( \text{RESD}_{\text{el}} \) / RER | +0.0-0.2               |
| 2      | 1.0-0.6                         | yes            | \( \text{RESD}_{\text{el}} \) / RER | +0.2-0.6               |
| 3      | 0.6-0.0                         | yes            | Redesign       | +0.6-1.2               |

4.2.4. Clamping force reduction

The second measure to improve efficiency is an increase of torque capacity by the optimization of the element flank design. In this way, less clamping force is needed to transmit torque and efficiency is gained by the reduction of friction in the variator.

Copyright © 2017 Society of Automotive Engineers of Japan, Inc. All rights reserved
The Torque Capacity Factor is derived from a slip curve\(^{(10)}\), a test where the maximum transferable primary torque or Break Away Torque (T\(_{BAT}\)) on the variator is defined (figure 12).

In case the clamping force and speed ratio are known, the actual TCF can be determined with:

\[
TCF = \frac{T_{BAT} \cdot \cos(\lambda)}{2 \cdot r_{p,ext} \cdot F_{cl}} \quad \text{or} \quad F_{cl} = \frac{T_{est} \cdot \cos(\lambda)}{2 \cdot r_{p,ext} \cdot TCF_{est}} \cdot S_f
\]

constant speed/torque/ratio/safety \rightarrow F_{cl} \propto \frac{1}{TCF_{est}} \quad (9)

The value TCF\(_{est}\) is used as an estimator for TCF in the CVT. The relationship between TCF and variator efficiency is shown in figure 13. In this case, TCF and TCF\(_{est}\) are chosen equal.

![Figure 12 Example slip curve](image1)

![Figure 13 Relationship between TCF and efficiency (simulation).](image2)

This relationship can also be derived similar to equation (1). Considering that torque is proportional to clamping force and inversely proportional to TCF and with constants \(c_1\), \(c_2\) and \(c_3\):

\[
\eta = 1 - c_1 - \frac{c_2}{TCF} - c_3 \cdot TCF
\]

At larger TCF values the positive effect on efficiency reduces. In case the minimum actuation pressure restricts the reduction of clamping force, a maximum benefit is reached sooner.

When an improved TCF is not applied to reduce clamping force, the higher CoF increases element-pulley loss. Best variator efficiency is reached when TCF\(_{est}\) equals the actual TCF.

An increase in torque capacity can be achieved by pushbelt, pulley or oil measures. Bosch cooperates with multiple parties to optimize the TCF in the variator and takes the responsibility for measures in the belt. The flank design is such a measure.

The detailed geometry of the flank profile is a relevant aspect that, besides efficiency, also influences the power density of the pushbelt. Details are therefore presented in the next chapter.

5. Improvement of Power Density

The power density of a variator is defined by the mechanical power that can be transferred through a given variator for a given ratio coverage\(^{(6)}\). The new pushbelt can handle increased input power and ratio coverage. The measures are discussed next.

5.1. Ratio Coverage

Customer requirements on ratio coverage stretch the limits of the pushbelt. Small minimum running radii provide the largest benefit but require additional measures to keep robustness at the same level. For the new belt, the smallest running radius in OD was reduced by 4mm. The following main design verifications were done (figure 14):

1. Verification of the element geometry to prevent contact between element legs at the smaller running radii.
2. Verification of loop strength. Smaller running radii lead to smaller bending radii and higher stress. Therefore the optimized Phytime material\(^{(7)}\) was chosen for the loops.
3. Verification of dynamic performance due to larger belt misalignment in the ultimate ratios.

![Figure 14 Reduction of minimum running radius in OD ratio.](image3)

5.2. Element Flank Design

The flank is a partner in the most heavily loaded contact in the variator. Its design must offer the best compromise between torque capacity and wear resistance of the tribological partners.

Figure 15 shows a 3D-scan together with the base profile of the reference and new flank design. The total length of the element flank was unchanged. The new design has the following features to improve TCF and wear resistance:

- An increase in the number of profile tops from 32 to 45
- Wider profile tops
- A profile depth reduction of 34%
5.2.1. Torque Capacity Factor (TCF)

The TCF is a parameter on variator level. It is influenced by the coefficient of friction (CoF) between individual elements and the pulley. In the variator this CoF depends on local variations in load and slip speed over the wrapped arcs. These result from variator design choices like stiffness and play in belt and pulley components and from the way that the variator is geometrically actuated. The varying conditions on the individual element make that TCF values are below individual CoF values.

The Stribeck curves in figure 16 provide an indication for the effects that operating conditions and design choices in variator and pushbelt can have on the CoF and thereby on the TCF. The curves show the CoF (µ) as a function of the slip speed condition in the contact. Other main parameters that affect this relationship are the dynamic viscosity of the oil, the average pressure in the contact as defined by the load and contact area and the roughness parameters of the contact partners.

Relevant design parameters of the variator, belt and element flank are set along these relationships to improve the TCF.

At the wrapped arc, pulley and element deformation introduce radial and tangential slip that cause a deviation from the ideal circular trajectory of the element. This is known as spiral running.

The axial stiffness plays a role in both mechanisms. The axial stiffness of the pushbelt by the small pitch and consequently large number of elements on the wrapped arcs is relatively high. The optimization of pulley shaft and pulley sheave stiffness therefore offers the most benefit. This insight is pursued in many new CVT designs \(^{[1,11]}\).

The dynamic viscosity of new types of oil applied in recent CVT introductions shows a decrease \(^{[6,12]}\). This is driven by the desire to reduce drag loss in the transmission by oil shear in tight gaps in for instance pumps and open clutches.

In the loaded element-pulley contact, a low viscosity extends the boundary lubrication range and improves the drainage of oil from the contact between element and pulley at element entry. Both mechanisms increase the CoF \(^{[13,14,15]}\). The oil drainage from the contact area during element entry is also improved by the larger number of drainage channels in the new element flank.

A low viscosity can however also increase CoF values in other contacts in the variator and thereby increase the losses there.

Oil viscosity reduces with temperature. By reducing the oil storage capacity of the drainage channels in the flank profile, local temperatures increase. This was one of the considerations for the decrease of the profile depth of the new flank profile.

The contact area between the element and the pulley can also be influenced by the flank design. To increase the CoF, a larger contact area is chosen. Over the mixed lubrication regime the contact load is partly carried by asperities and partly by the fluid film. A larger contact area increases the relative contribution of the asperities which increases the initial CoF.

The initial roughness characteristics of the reference and new belt and their contact partner were kept identical. The discussed measures led to an initial TCF increase of about 5%.

Over life, wear reduces roughness and increases the contact area. The high initial CoF and TCF values thereby decrease. To keep an improved TCF over life, the growth of contact area compensates the reduction in roughness.

A reduction of slip speed improves the CoF. Various slip mechanisms occur in the variator (figure 17). Slip already occurs outside the theoretical wrapped arc. Here, the translating elements pass compression and decompression zones where they are in an early or late contact with the rotating pulley and are axially compressed or decompressed.
5.2.2. Wear Resistance of the Flank

Material and oil properties were unchanged. An increase in wear resistance therefore depends on the modified parameters as discussed in the previous section. Next to the improvement of TCF, the wear resistance of the new element flank needed to be increased for reasons of:

- Increased torque and clamping force levels.
- Increased slip speeds by increased variator speed levels.
- Increased slip speeds and clamping forces by the use in dual-stage variators\(^{(1)}\). During the stage shift the variator shifts fast under high actuation forces.

Previously, the increased contact area of the new element flank was discussed. To remain effective, this increased needs to be durable. The change in contact area over wear can be shown by a Bearing Area Curve (BAC) that is often used to characterize surface textures. The desired shape of this curve was another optimization criterion for the flank profile design.

Figure 18 shows the BAC of the base profile of the reference and new element. Over the depth of the profile, the new element flank maintains to have a larger contact area.

The final flank profile that results from the manufacturing process, with process steps like tumbling, differs from the base profile. Figure 18 also shows the BAC that results from a 3D element scan as shown in figure 15. The final bearing area is somewhat larger.

![Bearing Area Curve for reference and new flank.](image)

Fig. 18 Bearing Area Curve for reference and new flank.

To validate the profile, wear validation tests were performed on tribotester level. In this tester single element flanks are tested under a contact force of 600N against a rotating pulley at varying slip speeds. Figure 19 shows a typical result. Under identical operating conditions the new element demonstrated a reduction of 46% in wear depth while the contact area still remained larger.

![Reference and new profile after wear (tribotester level).](image)

Fig. 19 Reference and new profile after wear (tribotester level).

The results show that the larger contact area and thereby the increase in TCF is durable over belt life. To confirm these results, TCF measurements on variator level were performed. The belts were subjected to a series of load cycles using reference pulleys and reference oil. In between cycles the TCF was measured in the OD and LOW ratio by means of a slip curve (figure 12).

Figure 20 shows the TCF for the reference and new belt. The improvement in the LOW ratio stabilizes around 9%. In the OD ratio it stabilizes around 4%.

![TCF over life for reference and new belt.](image)

Fig. 20 TCF over life for reference and new belt.

The increased TCF value enables a reduced clamping force level. The variator efficiency benefit associated with this lower clamping force level in OD is 0.2%. The reduced clamping force also lowers the power requirement of the actuation system. In hydraulic systems the maximum pressure level is usually defined by the variator. This level can therefore be reduced by 4 to 9%.

6. A New Application

The new belt was released for mass production and introduced into the market. The first customer application was a new dual stage CVT\(^{(1)}\). Table 3 shows the main characteristics.

| Requirement                      | Reference | New  |
|----------------------------------|-----------|------|
| CVT torque capacity [Nm]         | 150       | 150  |
| Belt design (width/ nr of loops) | 24/9      | 24/9 |
| Loop material\(^{(2)}\)          | Durimphy* | Phytime* |
| Variator LOW / OD ratio\(^{(1)}\) | 2.2 / 0.55 | 2.2 / 0.458 |
| Variator / CVT ratio coverage\(^{(1)}\) | 4 / 7.3 | 4.8 / 8.7 |
| Minimum running radius OD [mm]   | Ref. -4   |      |
| Variator center distance [mm]\(^{(1)}\) | 147       | 147  |
| Max torque/speed [Nm]/[rpm]      | 150/6500  | 175/7200 |
| TCF\(_{ref}\) release [-]        | 0.09      | >0.10 |
| Variator efficiency benefit in OD| Ref. +1%  |      |
| Fuel economy benefit (NEDC)\(^{(1)}\) | Ref. +3%  |      |

* Durimphy and Phytime are trade names of ArcelorMittal.
The new belt enabled a ratio coverage increase from 4 to 4.8. This was achieved by a reduction of the minimum secondary running radius. The ratio coverage of the transmission increased from 7.3 to 8.7.

The improvement of ratio coverage was realized within the same center distance and radial space envelope of the variator. The pulley angle remained 11 deg. The axial space envelope slightly increased to accommodate the larger pulley stroke from the wider ratio range.

For this application, variator efficiency in OD could be improved by 1%. The measures in the variator and CVT unit improved fuel economy on the NEDC cycle by 3\%(1).

7. Conclusions

- A new pushbelt with improved efficiency and power density characteristics was designed and released.
- The main efficiency measures are the reduction of slip between elements and loops by geometrical measures in the element and the reduction in the required clamping force by a new element flank profile. The efficiency of the variator in the OD ratio is improved with 1%.
- The reduction in required clamping force enables a 4% to 9% reduction of CVT actuation power.
- Main power density measures are the design for higher torque and ratio coverage, a new loop material and a new flank profile. The enlarged contact area between belt and pulley leads to an increase in wear resistance of 46%.
- The new belt was released in a first customer application. Here it enables a ratio coverage increase of 19% to supports the world’s largest CVT ratio coverage of 8.7. The new belt helps to realize a 3% improvement in fuel economy on the NEDC cycle\(^{(1)}\).

References

(1) Naotoshi Okamura et al., : Development of new generation CVT with auxiliary gearbox, SAE 2016-01-1109 (2016)
(2) www.statista.com/statistics/276419/best-selling-cars-in-the-united-states/, (access date March 31 2016)
(3) Erik van der Noll et al., : Optimizing the acoustics of the pushbelt CVT, Automobiltechnische Zeitschrift (ATZ) Vol.117, No. 02/2015 p.36-39 (2015).
(4) Koen Laurijssen, VT5 CVT, Next generation high efficiency CVT, webinar, https://vimeo.com/174481750, (access date October 7 2016)
(5) Brian Schneidewind et al., : Development of New continuously variable transmission for 2.0-Liter Class Vehicles, proceedings CTI Symposium, Novi-Michigan, USA (2015).
(6) Francis van der Sluis et al., : Key technologies of the pushbelt CVT, International Journal of Automotive Engineering, Vol.4, No.1, p.1-8, 20134096 (2013).
(7) Bert Pennings : Material developments to optimize pushbelt CVT power density, proceedings CTI Symposium Novi-Michigan, USA (2016).
(8) Katsuhiko Ohishi et al., : Development of High Fatigue Strength Maraging Steel for CVT Belt, SAE 2015-01-1102 (2015).
(9) Ahmet Kahraman : Modeling of power losses of automotive power transmission systems, proceedings CTI Automotive Transmission Symposium Berlin, Vol.2, p.433-446 (2013).
(10) Mark van Drogen et al., : Determination of variator robustness under macroslip conditions for a pushbelt CVT, SAE 2004-01-0480 (2004).
(11) Fumikazu Maruyama et al., Development of new CVT for compact car, SAE 2015-01-1091 (2015)
(12) Koichiro Inukai et al., Development of high-efficiency new CVT for midsize vehicle, SAE 2013-01-0365 (2013)
(13) Daniel Wilkerson et al., Real World 3-Element Screening Test for CVT Lubricant Applications, proceedings CTI Symposium Novi-Michigan, USA (2015)
(14) Takahiro Fukumizu et al., Development of Continuously Variable Transmission Fluid for Fuel Economy, SAE 2013-01-2584 (2013)
(15) Sang Yeob Cha et al., Development of Next-Generation Continuously Variable Transmission Fluid Technology, SAE 2012-01-1670 (2012)