Push belt CVT developments for high power applications

Arjen Brandsma, Jan van Lith, Emery Hendriks
Van Doorne's Transmissie b.v.

Dr. Hub van Doorneweg 120
Tilburg, 5026 RA, The Netherlands
Phone: +31 13 4640333
Fax: +31 13 4636590
E-mail: arjen.brandsma@nl.bosch.com

The transmittable power of currently available metal V-belt CVTs is limited to approximately 160 kW, depending on the application requirements. This paper presents several ways to increase the power range of metal push belt CVTs.

By the use of alternative CVT layouts, given certain variator characteristics, the transmittable power may be increased. Another option is to improve the belt power capacity by optimising belt design parameters. One of these parameters, the cone angle, also influences a lot of other aspects in CVT design. Therefore the influence of this angle is explained in more detail.

Today's state of the art push belts have a cone angle of 11°. For certain future car applications a different angle is considered.

Keywords / Metal V-belt CVT, Push belt, Power capacity, Cone angle

1. INTRODUCTION

In 1958 the first commercially available belt type CVT was introduced [1]. This rubber V-belt CVT was restricted to 42 kW given a centre distance of 520 mm (double belt application in 1972 with a ratio coverage of 3.9) [2]. For most automotive applications, a significant increase in the power capacity combined with a reduced transmission size was needed for the CVT, to become a serious alternative to other transmission types.

As a consequence, the development of metal belts was initiated, resulting in the introduction of a metal V-belt CVT on the automotive market in 1987 [1], [3].

At this moment, more than 2.5 million metal V-belts have been sold. Currently commercially available CVTs transmit power up to 110 kW (168 mm centre distance, ratio coverage of 5.4). Push belts for higher power applications are being developed, designed and validated at the moment [4], [5].

The transmittable power is determined by CVT design as well as belt design. Both aspects will be discussed in this paper, which is focused on metal push belt types.

2. POWER CAPACITY AND CVT DESIGN

High torque applications require high clamping forces (in order to transmit torque through friction), resulting in a high ring tensile stress. Since the allowable ring stress is determined by fatigue properties, the transmittable torque depends on said properties too. In this section, it is explained how the CVT design affects belt stress and thus the belt power capacity.

2.1 Transmission size and ratio coverage

The design of the variator (i.e. assembly of belt and pulleys) affects both tensile and bending stress. An increase in the belt's minimum running radius, lowers the bending stress in the flexible rings. As a result of this reduced bending stress, higher tensile stress (thus torque) becomes possible, without exceeding the critical fatigue stress. However, this increase in running radius directly affects either the ratio coverage or the centre distance of the variator. This implies that torque capacity (and thus power capacity) is not only determined by fatigue properties, but also by the centre distance and ratio coverage applied (figure 1).

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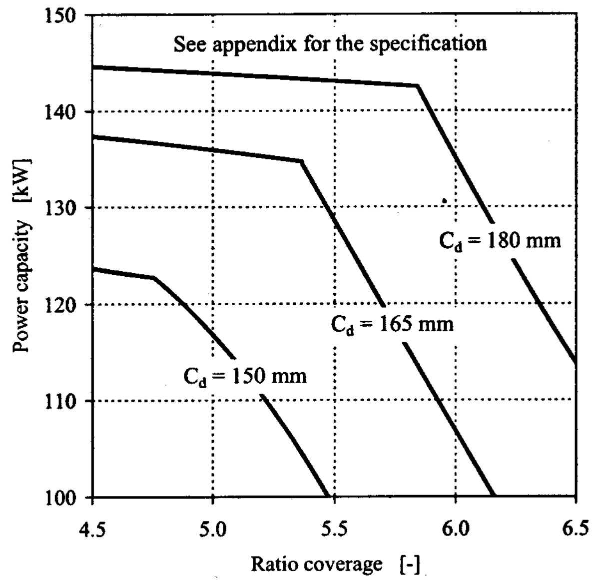

Fig. 1: Variator power capacity as a function of ratio coverage and centre distance for a 30/12 belt type.

2.2 Transmission input reduction

Centrifugal forces acting on a belt, due to belt speed, also affect the ring stress level. Consequently, power capacity is not only restricted by torque, but also by speed (figure 2). In order to reduce speed, an input gear reduction may be applied for certain applications (e.g. rear wheel driven cars). By optimising the gear reduction ratio with respect to ring stresses (through torque and speed), the power capacity may be maximised.

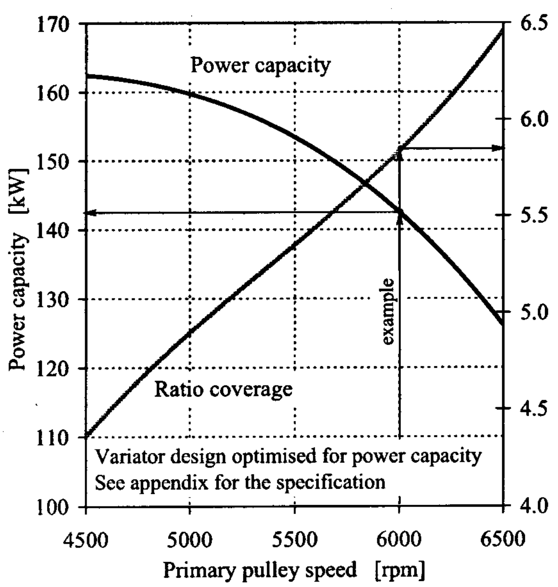

Fig. 2: Variator power capacity as a function of primary pulley speed for a 30/12 belt type.

2.3 Multiple mode CVT

An increase in power capacity may be realised by reducing the variator ratio coverage (section 2.1). By adding a multiple-speed gearbox in series with the variator, a large CVT ratio coverage still remains possible. Such a layout introduces a discrete ratio step as in any geared transmission. With an i^2-CVT the ratio step during the mode change may be avoided [6].

Other layouts such as a variator in parallel with a gear train can also be used to maximise torque capacity and ratio coverage [7].

2.4 Power split CVT

A straightforward option to split the power, is the use of two (or more) belts in parallel [8]. Synchronisation of the belts (in terms of speed and torque) is one of the major problems to overcome in order to avoid additional losses and wear.

Another option is to apply the variator in combination with a planetary gear set. In order to split power, two of the planetary gear inputs are connected with the variator. Depending on the layout, only part of the transmitted power will pass through the variator. This results in an increased power capacity, assuming that the capacity is determined by the variator. However, for power split CVTs, the power capacity is traded off against ratio coverage [9], [10]. Power split CVTs with multiple modes are also known [11], [12].

3. POWER CAPACITY AND BELT DESIGN

An increase in power capacity may also be achieved by upscaling the belt. In this section it is described how the belt may be scaled to higher power capacities, provided the capacity is determined by ring stress.

Another important aspect in optimising belt design, is the specified durability requirement. In automotive applications, it is likely that the time share at full load conditions will decrease with increasing installed power. Nevertheless, this aspect will not be discussed here, since it is beyond the scope of this paper.

3.1 Ring volume

The allowable tensile force level in the belt depends on the cumulated cross section area of all rings (figure 9). Assuming that the ring thickness is limited by bending stresses [13], said area may be increased by increasing the ring width or the number of rings.

3.1.1 Number of rings

Given the explanation above, it will be clear that the transmittable power may be improved by increasing the number of rings adopted in a belt.

The currently available second generation belt has two ring sets containing either 9 or 12 rings, depending on the required power capacity (figure 3). More rings may be used to increase the power capacity. However, the power capacity increases less than proportional to the number of rings installed. Figure 4 shows the nonlinear relation between stress reduction (i.e. power increase) and number of rings in a set. The nonlinearity is caused by several effects:

a) Production variations. Spacings between adjacent rings will introduce additional ring stresses. With an increasing number of rings, the stresses initiated by the spacings will also increase.
b) An increased belt mass, resulting in an increased centrifugal force and consequently an increased ring stress.

The number of rings used in practice is a optimum between several aspects such as power capacity, efficiency and costs.

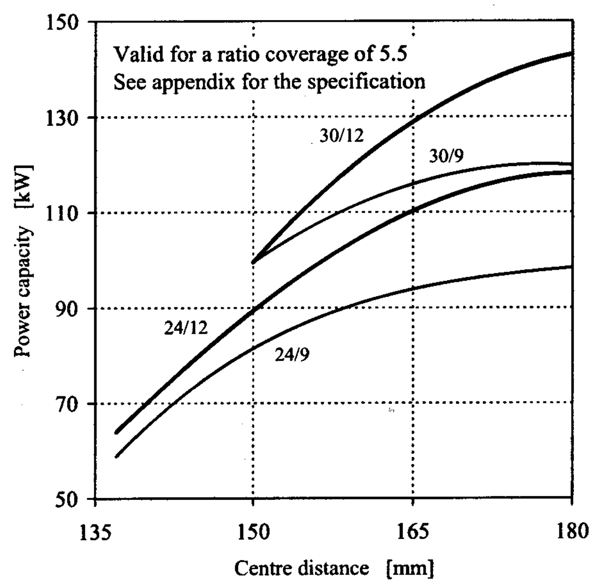

Fig. 3: Variator power capacity as a function of centre distance and belt type.

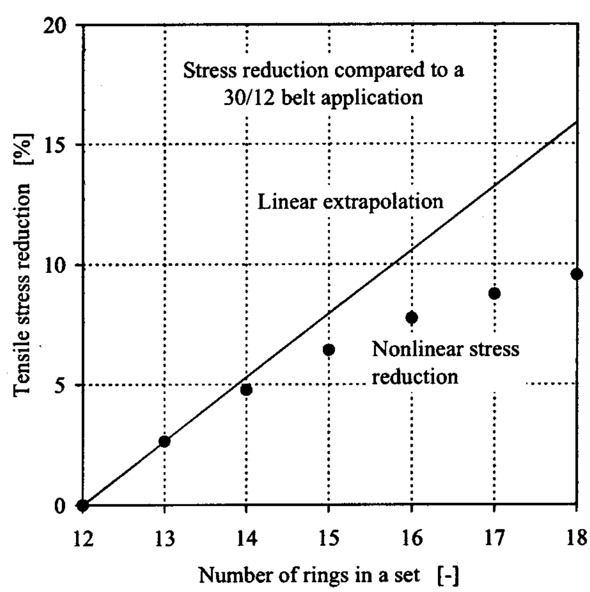

Fig. 4: Ring stress reduction as a function of the number of rings installed.

3.1.2 Ring width

By increasing the ring width higher tensile forces are allowed, which may result in an improved power capacity. Figure 3 shows that a belt with 30 mm wide elements is capable of transmitting more power than a comparable 24 mm belt. Upscaling of the belt by increasing the element width therefore seems to be a logical approach. However, this is only possible to a certain extent.

To ensure sufficient strength, other element dimensions such as flank height need to be scaled together with the element width (figure 9). This results in an increase in element mass (and thus centrifugal forces), which restricts the improvement in power capacity. Depending on the application (low or high speed) the power capacity may even decrease when the width of the elements is increased.

The influence of element width on the ring stress reduction is shown in figure 5.

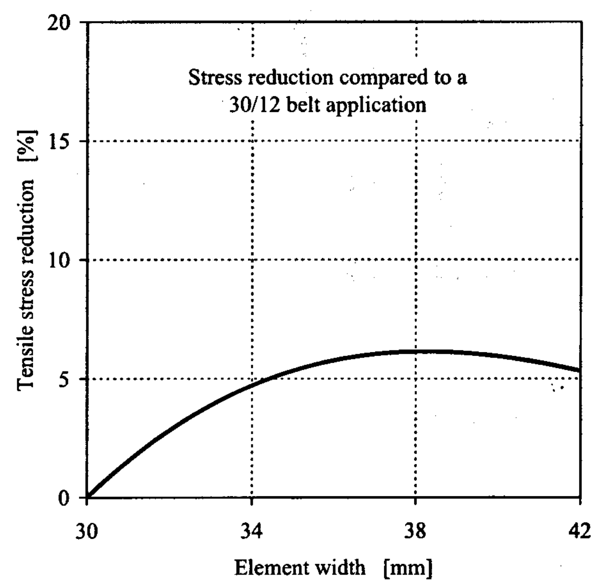

Fig. 5: Stress reduction as a function of element width.

3.2 Cone angle

The cone angle (figure 9) influences the tensile force in the rings as shown in figure 6. It is noted that the magnitude of the belt-sheave normal force (N_1) is determined by the torque transmitted (section 4.1). Given a certain torque level (and thus a certain force N_1), the ring tension (which depends on force N_2) reduces when the cone angle is decreased. Figure 7 shows the relation between cone angle and tensile stress

In other words, for a given ring stress level the power capacity may be increased by reducing the cone angle. The cone angle of currently available push belts is 11°.

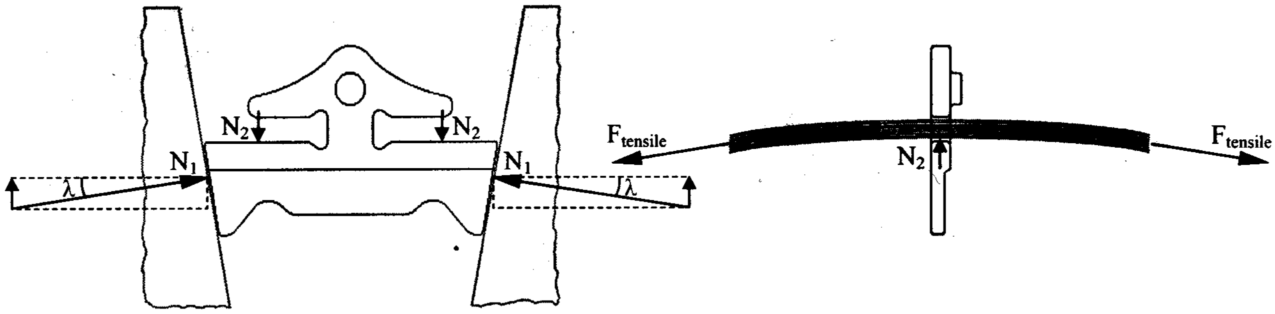

Fig. 6: Normal forces between pulley sheaves, elements and rings. On the left: cross section of a belt clamped between pulley sheaves. On the right: cross section of an element and a ring set.

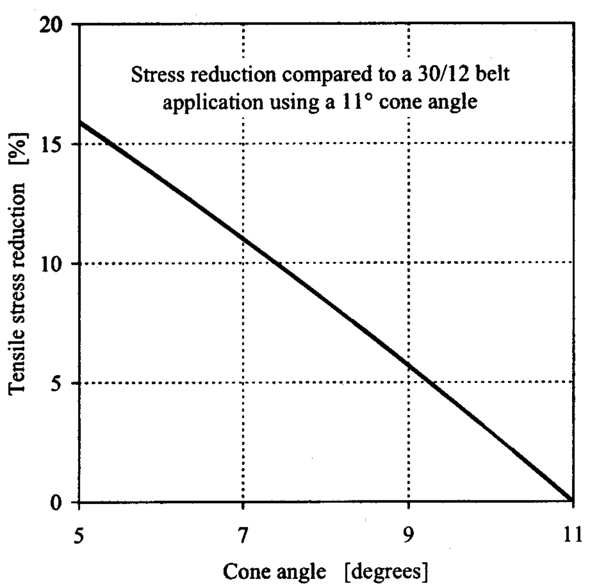

Fig. 7: Stress reduction as a function of cone angle.

4. CONE ANGLE CONSIDERATIONS

4.1 Variator theory

Clamping force
The minimum clamping force required (which is the axial component of the normal force N_1 of figure 6) is a function of the torque to be transmitted, as is shown in equation (1). At a given torque level, the clamping force increases when reducing the cone angle. However, the increase will be very small (maximal 2%) compared to the currently used 11° angle.

$$F_{ax} = \frac{T_P \cdot \cos(\lambda)}{2 \cdot R_P \cdot \mu} \qquad (1)$$

Self-locking effect
To prevent self-locking of the belt in the radial sense (when clamped between the pulley sheaves), the following condition (2) should be satisfied:

$$\tan(\lambda) > \mu \qquad (2)$$

The self-locking effect is most critical during standstill shifting. Under running conditions, the friction that has to be overcome during radial belt shifting is much lower, since it is only a part of the overall friction (the other part being oriented in the tangential direction for transmitting torque). The overall friction under running conditions is also lower, due to the sliding motion between elements and sheaves (dynamic instead of static friction).

Push versus tensile force
Consider a string of beads as shown in figure 8. When the rope is tensionless, the stack of beads is highly unstable and may buckle easily. To avoid buckling, the tensile force in the rope should exceed the push force between the beads.

In a push belt a similar precondition is valid. The push force between the elements is mainly determined by the torque level. The tensile force in the rings strongly depends both on the clamping force and on the cone angle (section 3.2). Since the clamping force is adjusted to the applied torque (1), the cone angle determines the push/tensile force ratio to a great extent. If the said ratio approaches a value of 1, unstable belt behaviour may be expected.

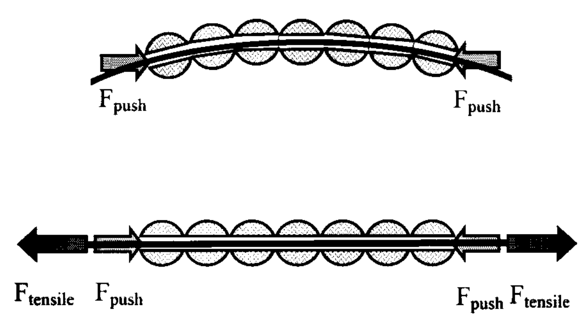

Fig. 8: Push and tensile force in a string of beads.

4.2 Consequences of cone angle changes

Reducing the cone angle has the following consequences:

a) Power capacity increases (see section 3.2). From figure 1 it appears that an improved power capacity, to some extent, may be used for increasing ratio coverage or for reducing the centre distance.

b) Pump flow decreases during shifting of the variator. Assuming that the required pump flow capacity is determined by a specified shift speed, a reduction in the applied pump capacity is possible. Since pump losses depend on pump capacity, a reduction of losses may be expected.

c) Alignment sensitivity decreases. In most applications both pulleys have a fixed and an axially movable sheave. Consequently, the belt will be misaligned in almost every variator ratio. For those applications the misalignment reduces, since it is proportional with $\tan(\lambda)$ [13].

d) Bearing loads reduce. Given a certain torque level, the tensile forces reduce which implies a decrease of bearing loads.

e) Movable sheave displacement reduces. Sheave displacement is proportional to $\tan(\lambda)$ and therefore will decrease with the cone angle. This may have a positive effect on the CVT size.

f) Variator efficiency is influenced by two opposite effects: Power losses due to internal belt friction reduce, since tensile forces decrease. Losses caused by mutual sliding of belt and pulley will increase. The overall influence on the variator efficiency therefore depends on the application.

g) Standstill shifting is more difficult or even impossible depending on the chosen cone angle (see section 4.1).

h) Unstable belt behaviour may occur, depending on the chosen cone angle (section 4.1).

4.3 Discussion

In the past, the optimal cone angle was found at 11° for automotive applications. Smaller angles were not considered feasible, due to the requirement of standstill shifting. This shift requirement was very important for the hydraulically controlled CVTs of that time [14], as will be explained below.

To ensure sufficient clamping force during a drive off situation, the variator needs to be in ratio Low. During an emergency brake, the vehicle wheels may lock completely. Consequently, the variator has to shift down with locked wheels (and pulleys) in order to reach ratio Low. Therefore, standstill shifting was necessary to prevent belt slip, when the variator was not in ratio Low.

Nowadays more sophisticated control strategies are possible with electronic controllers. Therefore, it is possible to prevent belt slip, even if the variator is not in ratio Low during a drive off situation. Moreover, new techniques and CVT layouts, that prevent standstill shifting are currently applied:

a) Drive off clutch between variator and vehicle wheels. This ensures that the belt is always running when the engine runs, even when the wheels are locked [15].

b) DNR set between variator and vehicle wheels. Enables disconnection of the belt from the (locked) wheels, when required [16].

c) Anti-lock Braking System (ABS). Prevents wheel lock. This means that the variator may shift to ratio Low, simultaneously with the decrease in vehicle speed. Therefore, ratio Low can then be reached even during emergency stops.

Given the above developments, it appears that the variator requirements are changing: Standstill shifting becomes less important, while the interest in high power applications is growing. As a consequence, for certain applications the 11° cone angle may no longer be the best option.

5 FUTURE DEVELOPMENTS

Van Doorne's Transmissie is currently developing a new push belt type to accommodate the changing variator requirements. This belt type, which is in the development phase, is characterised by the following parameters:

a) Element width = 33 mm
b) Number of rings per set = 14
c) Cone angle = 7°

This so-called 33/14 belt extends the current second generation belt programme. A power capacity increase of 50 %, compared to the 30/12 belt type is foreseen with this belt. Standstill shifting will be more difficult, given current CVT lay outs.

Several OEMs have shown serious interest in the 33/14 belt. Start of production is scheduled at 2002 or 2003.

6 CONCLUSIONS

The power capacity of push belt CVTs is mainly determined by ring stress. The ring stress level in the belt is not only influenced by load specification and belt design, but also by CVT design.

Push belts are scalable with respect to power capacity. Three parameters suitable for scaling are discussed: number of rings, element width and cone angle.

Drive line developments improved the feasibility of small cone angles. Therefore, smaller angles may be used in future applications. To increase the powerrange of push belt CVTs a new 33/14 belt type with a 7° cone angle is under development.

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NOMENCLATURE

C_d	Variator centre distance	[m]
F_{ax}	Axial force between belt and sheave	[N]
F_{push}	Push force between elements	[N]
$F_{tensile}$	Tensile force in a ring set	[N]
N_1	Normal force between sheave and element	[N]
N_2	Normal force between element and inner ring	[N]
R_P	Running radius of the belt between the primary pulley sheaves	[m]
T_P	Torque on primary pulley	[N•m]
λ	Cone angle	[-]
μ	Coefficient of friction	[-]

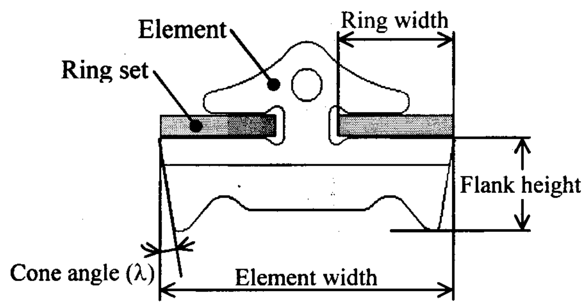

Fig. 9: Belt cross section.

APPENDIX

Figures 1, 2 and 3 show the allowable belt specification as advised by the belt supplier. The figures are indicative only. All charts are valid for the following settings:

- Assumed engine characteristics:
  Engine speed at maximum torque level =
  0.67 • engine speed at maximum power level
  Maximum engine torque =
  1.11 • torque at maximum power level
  Primary pulley speed and torque are identical with engine speed and torque.
- Primary pulley speed at maximum power level = 6000 rpm (not for figure 2)
- Safety factor =
  Max. possible torque / Max. nominal torque = 1.3
- Ratio coverage = 5.5 (not for figure 1 and 2)
- Top ratio = ratio where the maximum vehicle speed is achieved = 0.6
- Centre distance = 180 mm (not for figure 1 and 3)
- Belt type = 30/12 (not for figure 3)
  The mentioned belt code should be interpreted as follows: "element width [mm] / number of rings per set"