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Implicit regulation for automotive variators

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Abstract: A well-known practice uses mechanical variators to realize the so-called continuously variable transmissions (CVTs) for automotive application. A remarkable problem for one of the most used CVTs is given by the hydraulic control of the axial thrust on the half-pulleys that is necessary for torque transmission. It involves a heavy decrease in transmission efficiency.

In the present paper, the possible production of the required axial thrust by a simple spring to eliminate hydraulic losses is analysed. It is shown that the power capacity of such a regulated variator is near to the maximum possible, and the belt torque loss is always considerably lower than the sum of the belt and pump losses of a standard controlled CVT. Moreover, it is not much greater than a slip limit controlled CVT for medium-low vehicle speeds, and slightly lower for higher speeds. Finally, the power capacity by the spring thrust is evaluated when the variator is used in split power transmissions, considering several configurations and steady ideal operation.

Keywords: continuously variable transmissions, mechanical transmission, thrust regulation, automotive application, split power transmissions

NOTATION

c	winding radius parameter
C	torque applied on a pulley
C_{bl}	belt torque loss
C_{pl}	pump torque loss
D	pump displacement
f	belt/pulley coefficient of friction
f'	apparent coefficient of friction $= f/\sin\alpha$
F_0	spring preload
F_z	axial thrust on a pulley
F_{zs}	spring axial thrust on a pulley
h	pulley centre distance
k	radius factor
K	spring stiffness
n	angular speed
n_{1hp}	input speed defined in equation (8)
n_{1sp}	input speed defined in equation (8)
P_{in}	input power
P_v	variator power fraction
q	mass of belt per unit length
R	wrap radius on a pulley
S	actuator area
T_0	belt allowable tension
T_1	tension on the tight belt side
T_2	tension on the slack belt side
V	belt speed
α	pulley wedge half-angle
β	angle defined in Fig. 1
Δn	angular speed loss
Δx	half-pulley axial displacement
$\Delta\theta*$	sliding angle
η_p	pump efficiency
θ_0	idle angle
τ	overall speed ratio
τ_{var}	variator speed ratio

Subscripts

1	primary pulley
2	secondary pulley
in	input
max	maximum
min	minimum
out	output

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1 INTRODUCTION

In automotive power transmissions from the primary engine to the propulsion system, a well-known and common practice uses mechanical variators, where a V-belt connects two expanding pulleys [continuously variable transmission (CVT)]. The fundamental features of belt-type CVTs can be identified with their power capacity (transmissible power–size ratio), coverage ratio

(ratio between the maximum and minimum speed ratio) and efficiency.

The power capacity of a variator is mainly determined by the belt stress. High torque applications require high clamping forces, yielding a high belt tensile stress (torque transmission by friction). Centrifugal forces due to the belt motion also affect the belt stress level [**1**], and thus the power capacity may be maximized by optimizing the primary pulley angular speed. Moreover, given a particular variator, the power can be increased by using alternative CVT layouts.

For example, combining one variator with one or more epicyclic trains into some complex arrangement (split-way or two-path transmission) may amplify the power, though reducing the coverage ratio, or vice versa, in comparison with the simple variator [**2**, **3**]. A multi-mode scheme (that is, a particular combination of two or more split-way schemes commuting to each other by brakes or clutches) can produce a further improvement for both power and coverage ratio [**4**].

Today, wide use is made of Van Doorne's metal pushing V-belt CVT. Although the CVT has effectively an infinite number of gear ratios, allowing the engine to operate near to its maximum efficiency point, the expected reduction in fuel consumption has yet to be realized because existing CVT systems have a lower efficiency than their fixed ratio counterparts. This inefficiency is mainly due to the hydraulic pump required to provide the clamping force on the pulleys and to the losses associated with the belt–pulley coupling itself. The sum of these losses is nearly constant over a very wide range of vehicle speeds, and so the efficiency of a metal V-belt variator never rises above 0.8 at full load and exhibits a sharp decrease at partial load, while the gear efficiency can easily exceed 0.9 [**5**–**8**].

The oil pressure level is controlled by pump choking, but, as the minimum required axial thrust to guarantee the torque transmission undergoes substantial variations with speed ratio and load, the pressure is generally kept at a higher level than required, producing heavy efficiency losses. Such a poor performance is also due to partial utilization of the rated load, as the axial thrust and primary speed are not regulated for reaching the maximum transmissible power. For all these reasons, the convenience of using small variators to be loaded close to their top power appears evident [**2**].

With the aim of providing some increase in variator efficiency, in the present work an analysis is made of the possible production of the required axial thrust by a simple spring (whose force depends on the axial distance between the half-pulleys, i.e. on the speed ratio only) to eliminate hydraulic losses. It is shown that the power capacity of such a regulated variator is near to the maximum possible and the belt torque loss is always considerably lower than the sum of the belt and pump losses of a standard controlled CVT. Furthermore, it is not much greater than a slip limit controlled CVT for medium-low vehicle speeds, and slightly lower for higher speeds.

Moreover, this device is surely much simpler than a hydraulic system, where the oil pressure has to be regulated according to the load, angular speed and speed ratio. Of course, the expected benefits can be achieved mostly if the hydraulic regulation is eliminated for the speed ratio as well, for example using electric actuators, as in some automotive power steering systems, which take up energy only during the transitional states. Finally, the power capacity by spring thrust regulation is evaluated when the variator is used in split power transmissions, considering several schemes and steady ideal operation.

2 VARIATOR MODEL

Several papers have recently been written on the modelling of the metal pushing V-belt transmission. Karam and Play [**9**] have proposed an interesting model that, however, is not easily applicable. Fujii and Kurokawa [**10**] and Gerbert [**11**] derived relations between transmitted torque and pulley thrust, but none of them concentrates on the aspect of the maximum tension level that is reached on the bands, which is important to maximize the variator power capacity [**1**, **2**]. Micklem *et al.* [**12**] state that Coulomb friction cannot predict the belt slip they measured under tests and propose a model based on elastohydrodynamic lubrication concepts, where the entire winding arc is active. By contrast, Fujii and Kurokawa [**10**] and Karam and Play [**9**] presume the existence of an idle arc where the compressive load upon the blocks does not change. The magnitude of this idle arc has an important influence on the axial thrust.

In this paper, a simple model is applied, based on the classic belt theory (Eulero) and on the following assumptions [**13**]:

1. The block and band structure of the belt is treated as a uniform continuum, and so the friction among the bands and between the band set and the blocks is neglected [**14**, **15**]. Besides, it is shown in references [**12**] and [**9**] that the transmitted force between rings and blocks is smaller than the compression force between the blocks, and increasingly so with increasing torque ratio r [**10**] (the torque ratio is defined as the ratio between the torque presently transmitted and the maximum transmittable torque at the slip limit for a given pulley thrust).
2. The V-belt moves along a circular path when engaged with a pulley as the radial stiffness is very large [**12**], and so it is considered to be like a flat belt and an equivalent coefficient of friction is introduced.
3. Finally, the only inertia forces considered are due to belt centripetal acceleration.

A useful relation for calculation of the speed ratio

$\tau_{var} = n_2/n_1 = R_1/R_2$ of the variator in Fig. 1 is given by the constancy of the belt length (see the Notation):

$$L = (\pi + 2\beta)R_1 + (\pi - 2\beta)R_2 + 2h\cos\beta$$

and

$$\beta = \arcsin\frac{R_1 - R_2}{h} \tag{1}$$

The tension distribution is qualitatively as in Fig. 2, where

$$\frac{T_2 - qV^2}{T_1 - qV^2} = e^{-f'\Delta\theta^*}$$
$$\theta_{0_1} = \pi + 2\beta - \Delta\theta^*$$
$$\theta_{0_2} = \pi - 2\beta - \Delta\theta^* \tag{2}$$

The axial thrust on the primary and secondary pulley is respectively

$$F_{z_1} = \frac{T_1 - qV^2}{2f'\tan\alpha}[\theta_{0_1}f' + (1 - e^{-f'\Delta\theta^*})] \tag{3}$$

$$F_{z_2} = \frac{T_1 - qV^2}{2f'\tan\alpha}[e^{-f'\Delta\theta^*}\theta_{0_2}f' + (1 - e^{-f'\Delta\theta^*})] \tag{4}$$

from which it is clear that the axial thrust is on average lower on the driven pulley than on the driving pulley.

The input torque C_1 is

$$C_1 = T_1R_1\left[\left(1 - \frac{qV^2}{T_1}\right)(1 - e^{-f'\Delta\theta^*})\right] \tag{5}$$

or else, depending on the axial thrust on the primary pulley, F_{z_1},

$$C_1 = 2f'R_1\tan\alpha(1 - e^{-f'\Delta\theta^*})\frac{F_{z_1}}{(1 - e^{-f'\Delta\theta^*}) + \theta_{0_1}f'} \tag{6}$$

or, on the secondary pulley, F_{z_2},

$$C_1 = 2f'R_1\tan\alpha(1 - e^{-f'\Delta\theta^*}) \times \frac{F_{z_2}}{(1 - e^{-f'\Delta\theta^*}) + \theta_{0_2}f'e^{-f'\Delta\theta^*}} \tag{7}$$

It is interesting that the transmissible torque is not affected by the primary angular speed ($n_1 \propto V$) if $T_1 < T_0$, but at the highest belt tension, T_0, an increase in the primary speed should be compensated for by a reduction in the axial thrust and thus in the transmissible torque.

3 REGULATION MODES

Figures 3 and 4 show the transmissible torque, power, and primary and secondary axial thrusts for a variator which is always assumed to be working at the highest admissible tension and at the adherence limit (sliding arc $\Delta\theta^*$ equal to the winding arc $\pi - 2|\beta|$ on the smaller pulley). Therefore, in any condition this is the variator with the maximum transmissible power–size ratio. It will therefore be defined as the ideal variator (IV). The power and torque are scaled by their common values at the ends of the speed ratio range for the particular input speed $n_1 = 6047$ r/min, and the axial thrust is scaled by the maximum admissible belt tension T_0, which is here fixed at 5000 N (see other design data in Table 1). The curves are plotted for constant input speeds,

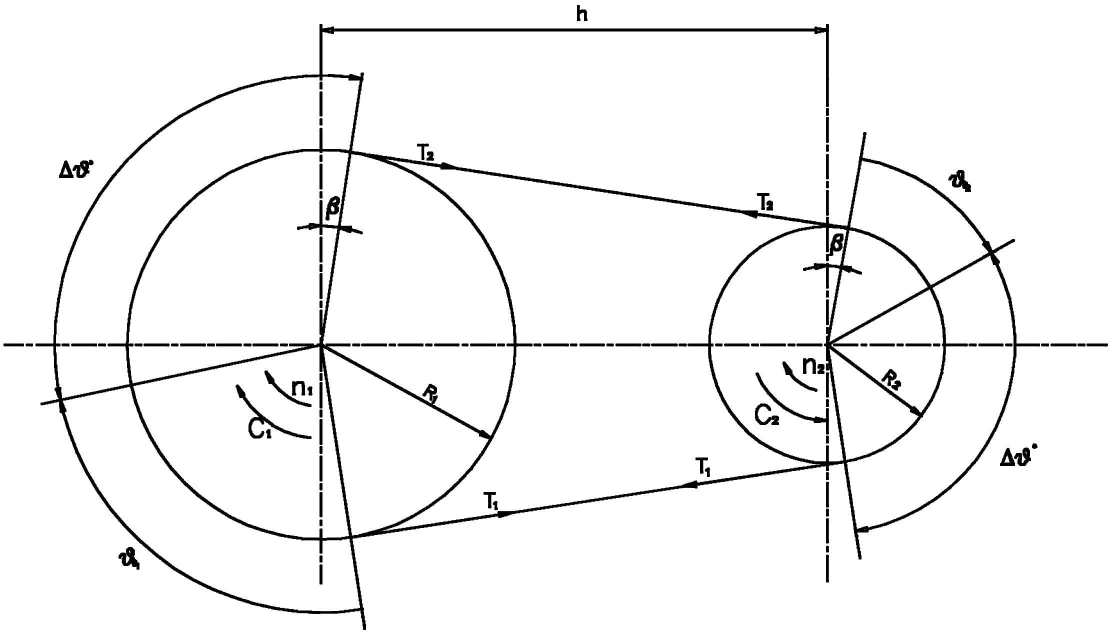

Fig. 1 Variator scheme

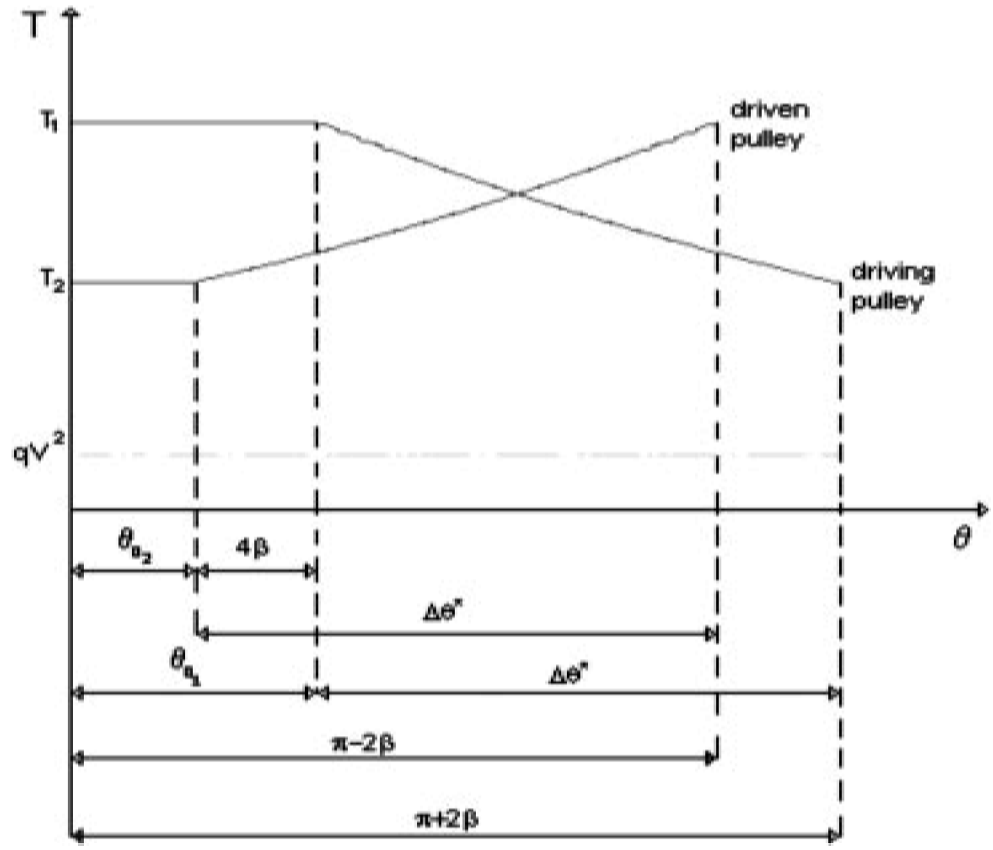

Fig. 2 Tension distribution along the winding arc for $\tau_{var} > 1$

$n_1 = 1$–2–3–4–5–6–7–8000 r/min, plus the two notable values $n_1 = n_{1hp} = 4381$ r/min (highest transmissible power at $\tau_{var_{max}}$) and $n_1 = n_{1sp} = 6047$ r/min (same power at $\tau_{var_{max}}$ and $\tau_{var_{min}}$). For a symmetrical variator (i.e. $\tau_{var_{max}} = 1/\tau_{var_{min}}$),

$$n_{1sp} = \sqrt{\frac{1-\tau_{var_{min}}}{\tau^2_{var_{max}} - \tau_{var_{min}}} \frac{T_1}{qR^2_{min}}}, \qquad n_{1hp} = \sqrt{\frac{T_1}{3qR^2_{max}}} \tag{8}$$

Thus, for the most efficient use of the power capacity, the primary pulley of a simple variator should rotate at $n_1 = n_{1sp}$, and the regulation of the axial thrust on the secondary pulley would be more convenient since it would on average be lower. Actually, the axial thrust is not so finely regulated as for the ideal variator above, and thus the transmissible power is lower than shown in Fig. 3.

In industrial application, for example, for which the definition of technical variator (TV) is introduced here, the manufacturers declare a monotonous increasing characteristic power versus speed ratio at constant input speed, while in Van Doorne's automotive variator (AV) the power is kept constant on increasing τ_{var} at constant n_1 by reducing the secondary axial thrust more than for the IV (Fig. 5).

To compare the power capacity of the spring variator (SV) suggested in this work with the commercial variators and the IV defined above, the technical variator was identified with the ideal variator, though only for $n_1 \leqslant n_{1hp}$, so that the power–speed ratio curves always show a positive slope. As to the AV, the further limitation of a constant torque is imposed, choosing the value of Fig. 3 at $\tau_{var_{min}}$ for $n_1 = n_{1hp} = 4381$ r/min. Thus, Fig. 3 is useful for estimating torque and transmissible power for these variator types as well.

4 POWER CAPACITY

The SV design implies firstly the choice of the spring location either on the primary or on the secondary shaft, and then the calculation of the preload, spring stiffness and angular speed in order to match the SV and the IV performances as closely as possible.

Using a compression spring 'external' to the half-pulleys, in view of the power maximization for a given variator size, the regulation of the axial thrust on the primary pulley is unavoidable. In fact, as the speed ratio increases, the primary pulley closes up, reducing the spring compression (axial thrust), while a spring on the secondary pulley behaves in the opposite way. In both cases the characteristic of axial thrust versus τ_{var} is

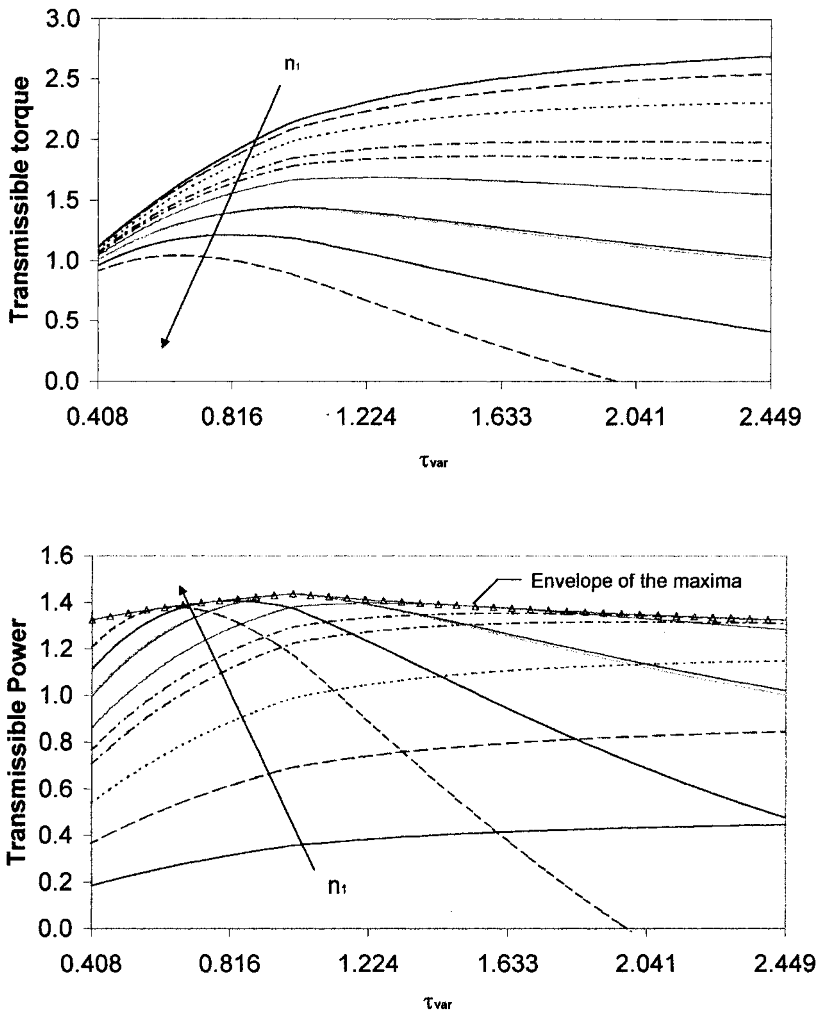

Fig. 3 Transmissible torque and power for the ideal variator

different from the ideal case of Fig. 4, substantially so for the latter case (increasing spring thrust instead of decreasing ideal thrust with increasing τ_{var}), and in a lesser way for the former case, especially at high angular speeds.

The thrust produced by a spring extension, Δx_1, is

$$F_{zs_1} = F_0 - K\Delta x_1, \qquad \text{where } \Delta x_1 = 2(R_1 - R_{min}) \tan \alpha \tag{9}$$

The spring parameters F_0 and K can be determined by optimizing the SV with reference to the IV, imposing the same axial thrust at the τ_{var} range end-points, for the primary speed $n_1 = n_{1sp}$ which yields the highest average transmissible power. The comparison of axial force, transmissible torque and power with the IV is shown in Fig. 6.

Of course, the present variator SV:

(a) cannot work at input speeds $n_1 > n_{1sp}$, otherwise the belt breaks;
(b) for any input speed $n_1 \leqslant n_{1sp}$, has the same transmissible power at $\tau_{var_{min}}$ and at $\tau_{var_{max}}$;
(c) for a given τ_{var}, shows a linearly increasing transmissible power with input speed (the lower the primary speed compared with n_{1sp}, the more SV is penalized in comparison with IV).

When optimized at the most convenient angular speed according to the coverage ratio, the spring variator permits the same size as the ideal variator, while the technical variator and the automotive variator require larger

Table 1 Design data

Description	Symbol	Value	Unit of measure
Minimum winding radius	R_{min}	28.7	mm
Maximum winding radius	R_{max}	70.2	mm
Centre distance	h	155	mm
Belt mass per unit length	q	1.6	kg/m
Groove half-angle	α	11	deg
Friction coefficient	f	0.1	
Allowable belt tension	T_0	5000	N
Belt length	L	632	mm
Pump displacement	D	9.72	cm^3/rev
Pump efficiency	η_p	0.7	
Secondary actuator area	S_2	10	cm^2

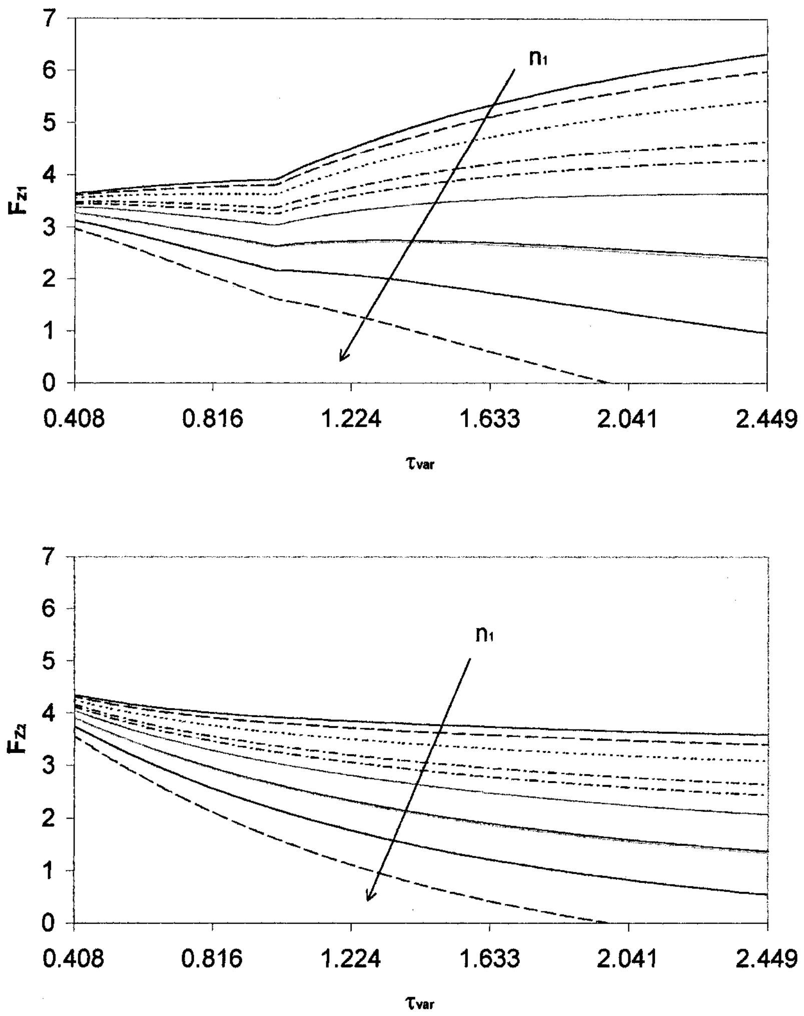

Fig. 4 Axial thrust on the primary pulley, F_{z_1}, and secondary pulley, F_{z_2}, for the ideal variator

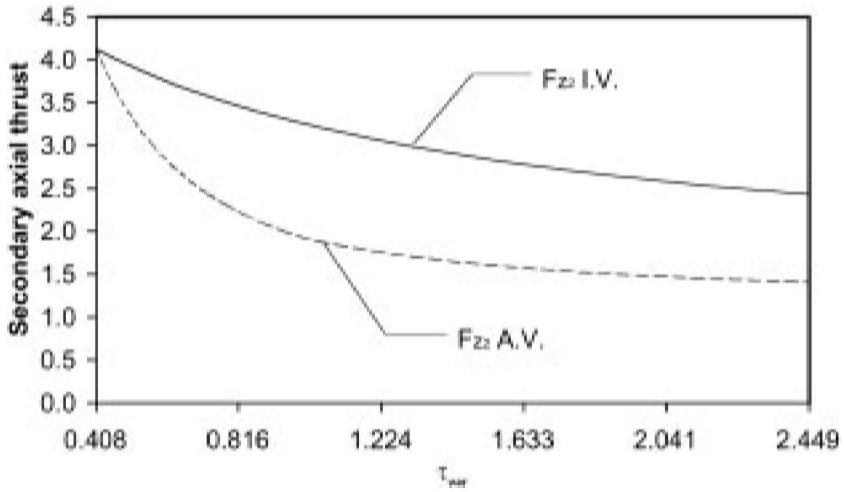

Fig. 5 Secondary axial thrust at $n_1 = 4381$ for the automotive variator and the ideal variator

sizes, as shown in Table 2. In the case of an automotive engine, however, where the maximum torque must be transmitted at an angular speed that is about two-thirds of the maximum power speed, the ideal variator gains in comparison with the spring variator because it can transmit a higher torque at a lower speed, while the spring variator cannot exceed the transmissible torque at the highest primary angular speed.

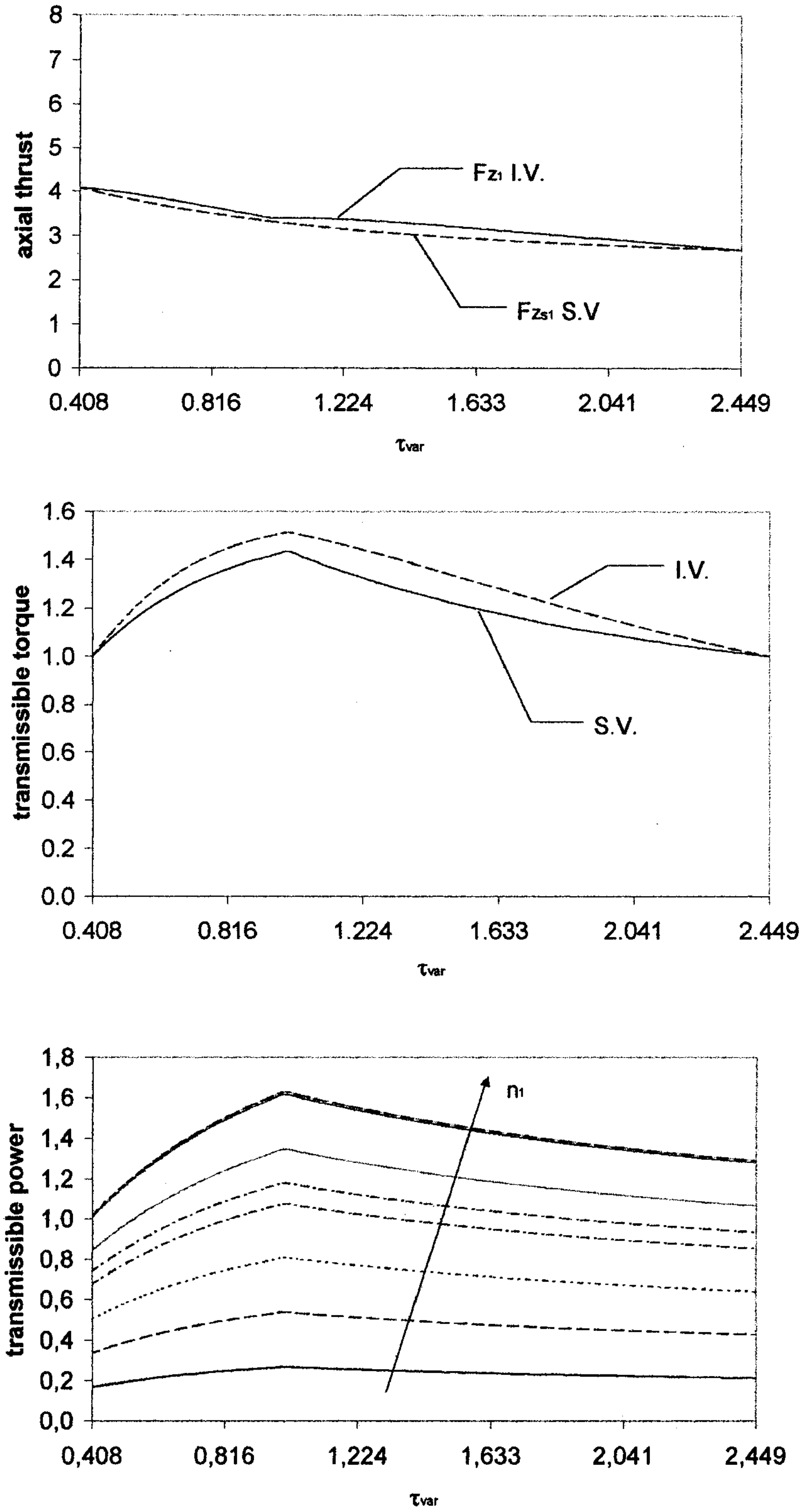

Fig. 6 Axial thrust, transmissible torque and power for the spring variator when optimized at $n_1 = n_{1sp} =$ 6047 r/min

5 EFFICIENCY

Thus, the SV reduces the complexity of the axial thrust regulation by making it automatic and passive (no power consumption) and allows a very good power capacity. However, the spring thrust is excessive at partial load, bringing about a consequent rise in the belt power losses. Therefore, the belt losses of the spring variator were compared with the sum of the belt and pump losses for the cases of standard and slip limit control (the sliding

Table 2 Summary of the variator class

Speed ratio τ range	IV			SV			TV			AV		
	Two-ET	Mono-ET	Simple	Two-ET	Mono-ET	Simple	Two-ET	Mono-ET	Simple	Two-ET	Mono-ET	Simple
1/2–1	0.286	0.453	0.728	0.382	0.509	0.728	0.365	0.453	0.756	0.784	0.784	1.307
1/4–1	0.660	0.701	0.872	0.713	0.764	0.872	0.774	0.784	0.963	1.176	1.176	1.307
1/6–1	1.000	1.000	1.000	1.000	1.000	1.000	1.259	1.307	1.307	1.307	1.307	1.307
0–1/2–1	0.402	0.907	—	0.600*	1.019	—	0.640	0.907	—	1.568	1.568	—
0–1/4–1	1.049	1.071	—	1.049	1.098	—	1.164	1.164	—	2.091	2.091	—
0–1/6–1	1.521	1.532	—	1.634	1.661	—	1.869	1.869	—	2.875	2.875	—
−1/2–1	0.804	1.814	—	1.432	2.037	—	1.280	1.814	—	3.137	3.137	—
−1/4–1	1.581	1.952	—	1.702	2.059	—	1.976	2.015	—	3.660	3.660	—
−1/6–1	2.248	2.251	—	2.270	2.294	—	2.507	2.507	—	4.444	4.444	—

* Obtained realizing the regulation of the axial thrust on the variator secondary pulley.

arc is fixed at 80 per cent of the winding arc on the smaller pulley).

The power losses of the belt consist of the speed loss and torque loss, defined as follows:

$$n_2 = n_1 \frac{R_1}{R_2} - \Delta n_2, \qquad C_1 = C_2 \frac{R_1}{R_2} + C_{bl} \tag{10}$$

The speed loss Δn_2 is mainly due to the slip between the belt and the pulley surfaces. It is smaller than the torque loss [**8**, **16**] and moreover is proportional to the torque transmitted and inversely proportional to the axial thrust [**14**], so it can be neglected in this analysis.

The belt torque loss, C_{bl}, is due to radial friction forces which constrain the belt to enter the pulley at a greater radius and to exit at a smaller radius than the equilibrium radius. This implies an additional torque to force the belt into the pulleys and then to remove it. An empirical model has been proposed by Micklem *et al.* [**15**] who, for both primary and secondary pulleys, suggest that

$$\begin{aligned} &R_{in} = R/k, \qquad R_{out} = Rk \\ &R\sqrt{1-k^2} = \text{constant} = 5.5\ \text{mm} \end{aligned} \tag{11}$$

They use elastohydrodynamic lubrication theory to model the variator behaviour [**8**], in contrast to the present analysis, but, bearing in mind that their torque loss mechanism is formally analogous to the imperfect flexibility of the conventional belts, this type of torque loss is assumed, but using the same experimental constant of reference [**15**], that is

$$\begin{aligned} &R_{in} = R + c, \qquad R_{out} = R - c, \qquad c = R(1-k) \\ &k = \sqrt{1 - \left(\frac{5.5\ \text{mm}}{R}\right)^2} \end{aligned} \tag{12}$$

$$\begin{aligned} C_{bl} &= (T_1 + T_2)c_1 + (T_1 + T_2)c_2 \frac{R_1}{R_2} \\ &= (T_1 + T_2)R_1[(1-k_1) + (1-k_2)] \end{aligned} \tag{13}$$

Figure 7 compares the torque loss measured by Micklem *et al.* [**15**] and the values given by (13) for an input torque of 32 N m. The agreement is good and the slopes of the calculated and experimental curves are very close. Therefore, the energy involved in the process appears to be the same whichever model is applied, even though the true causes of the torque loss are different in the metal pushing and in the rubber V-belt.

It can be shown that the belt torque loss has only a very small dependence on the transmitted torque and increases with increasing axial thrusts. The input speed has little effect on belt torque loss when τ_{var} is small but a greater effect at higher τ_{var}. For given input speed and axial thrust, the loss is maximum at $\tau_{var_{max}}$, minimum at about $\tau_{var} = 1$ and intermediate at $\tau_{var_{min}}$.

Finally, the torque absorbed by the hydraulic pump for standard and slip limit controlled variators is expressed as

$$C_{pl} = \frac{F_{z_2}}{A_2} \frac{D}{2\pi\eta_p} \tag{14}$$

In fact, the pump outlet pressure is always equal to the pressure in the secondary actuator and the pump rotates at the primary angular speed.

By applying the above formulae and the work diagrams of a standard controlled variator driven by an 1100 cm^3 petrol engine [**8**], the torque losses have been derived in Fig. 8 over a vehicle speed range from 15 to 140 km/h.

Supposing for the slip limit variator that the control system sets the same speed ratio as in the standard case at the various vehicle speeds, Fig. 9 shows the torque losses in this case. Optimizing the SV as explained above (Section 5) but with reference to the slip limit variator, the spring parameters $F_0 = 20.5$ kN and $K = 4.6$ kN/cm were obtained, and Fig. 10 shows the corresponding belt torque loss.

Finally, Figs 11 and 12 compare the total losses and the secondary pressures for the three cases considered. In the case of the SV, the secondary pressure is representative of the axial thrust that must be applied on the

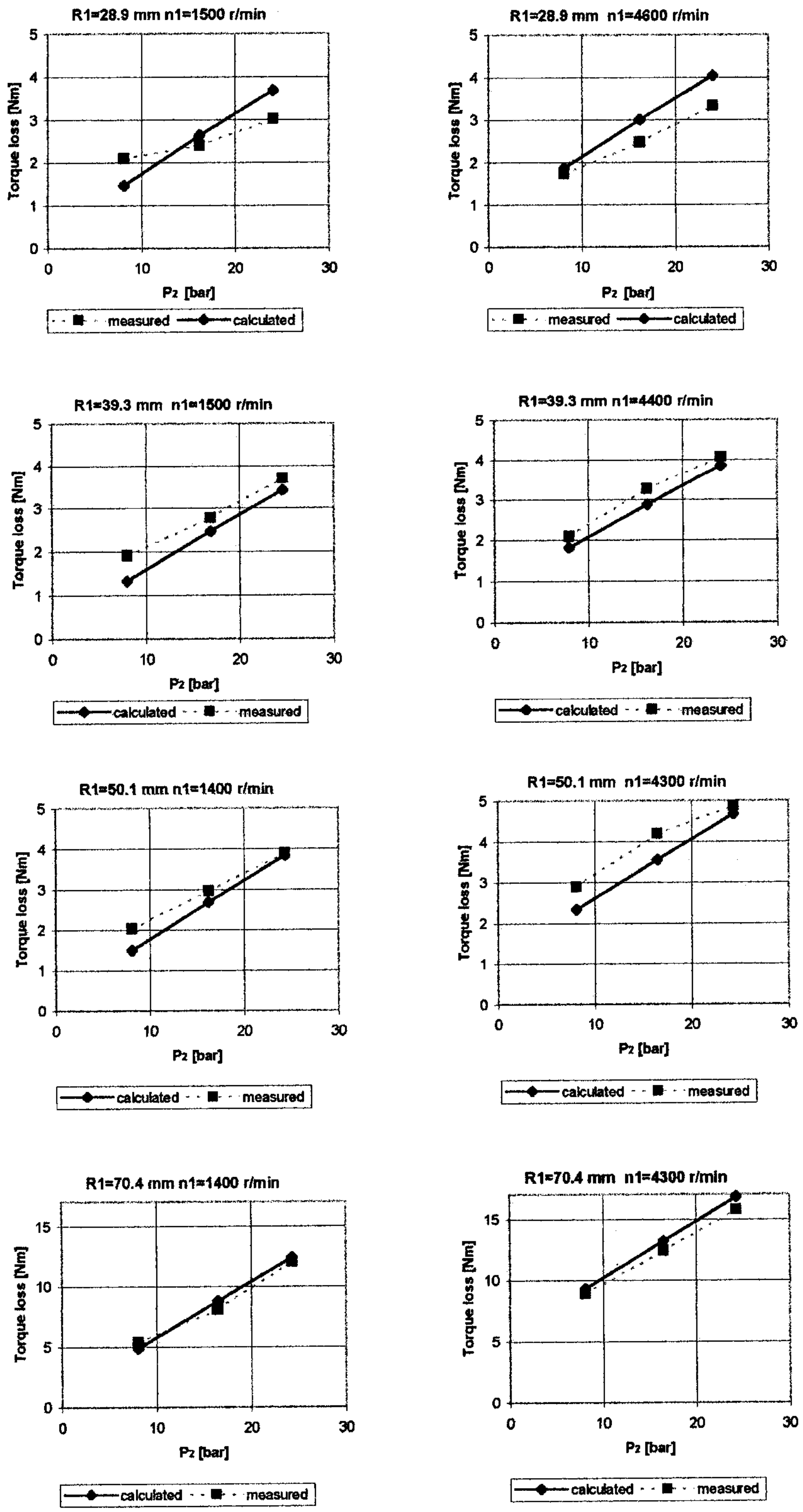

Fig. 7 Comparison between measured [**15**] and calculated torque loss for an input torque of 32 N m

secondary pulley to balance the spring thrust on the primary pulley.

It can be seen that, especially in the low vehicle speed zone, both the standard controlled CVT and SV keep the clamping force decidedly too high as regards the real input torque. Nevertheless, the SV torque loss is always considerably lower than the sum of the belt and pump losses of the standard controlled CVT. Moreover, it is not much greater than the slip limit controlled CVT for medium-low speeds, and slightly lower for higher speeds.

With increase in the vehicle speed the efficiency rises exponentially from 0.37 to 0.89 for standard CVT and

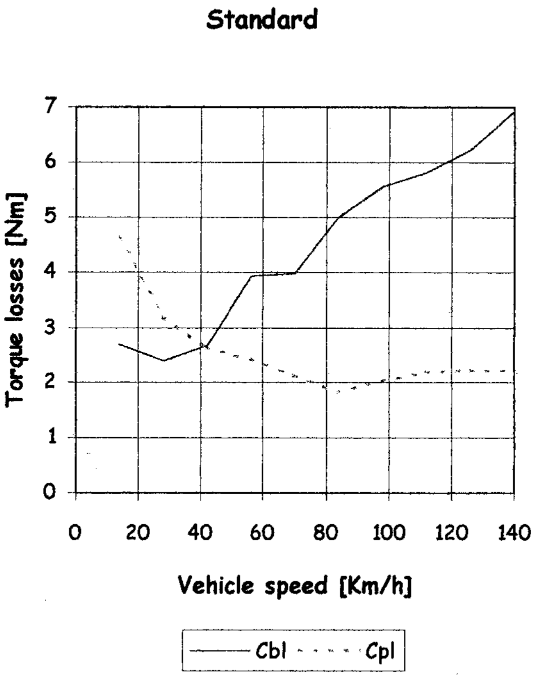

Fig. 8 Torque losses in the standard variator

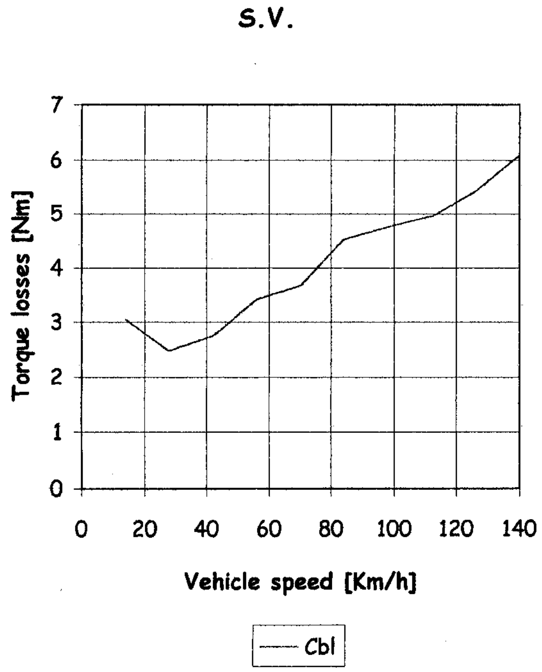

Fig. 10 Torque losses in the spring variator

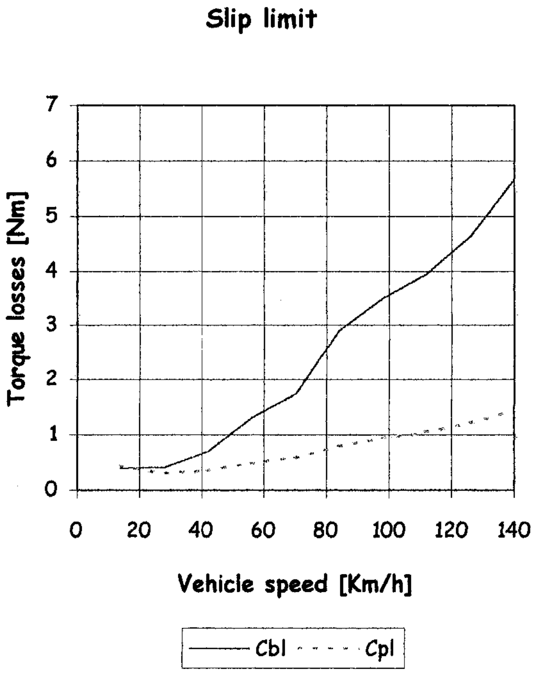

Fig. 9 Torque losses in the slip limit variator

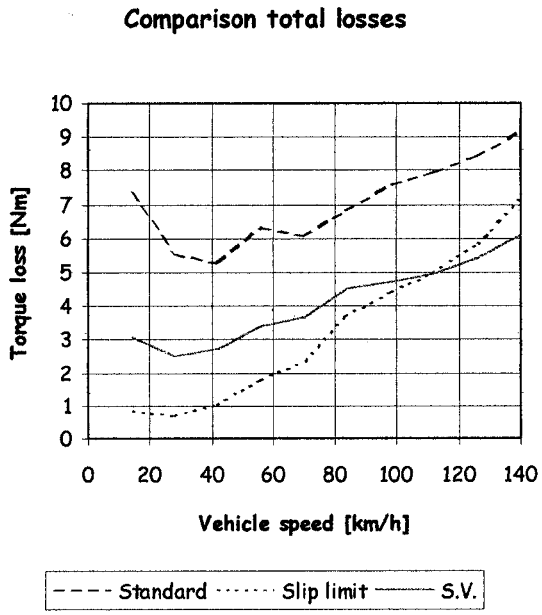

Fig. 11 Comparison of the overall torque losses of the standard, slip limit and spring variators

from 0.74 to 0.93 for SV, while it goes from 0.93 to 0.92 for the slip limit variator. However, when all the other losses are included (slip, bearings, gears, clutch) the efficiency settles at lower values [**5**–**8**].

Moreover, considering also the losses due to slip, the difference between the slip limit variator and the SV should decrease, because the slip losses are small when the variator is working with high clamping forces but are likely to become much greater for low clamping forces as required by the slip limit variator. Furthermore, in this case, the control system must give a very rapid pressure increase when the driver calls for maximum acceleration, and it should be verified whether the higher slip values can affect the life of the transmission [**8**].

As far as the SV is concerned, the expected benefits can be achieved mostly if the hydraulic regulation is eliminated for the speed ratio as well, for example using electric actuators that take up energy only during the transitional states (a device of this kind is produced

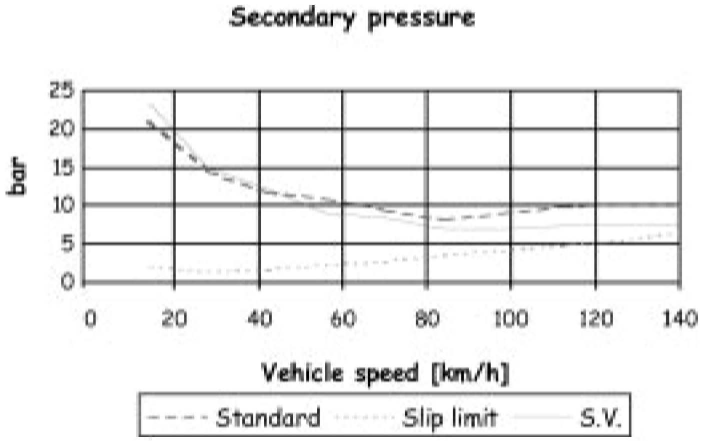

Fig. 12 Comparison of the secondary pressure of the standard, slip limit and spring variators

for competition scooter variators), but they must be sufficiently powerful to balance the high spring thrust.

6 SPLIT-WAY TRANSMISSIONS

As the variator is generally the main source of power loss in the transmission, it can be convenient to split the transmission into two parallel paths, the first one with a continuously variable ratio (variator V) and the other with a fixed ratio (gear G): the two paths converge into an epicyclic train (ET) on one side, and are linked together (mono-ET scheme) or converge into a second epicyclic train (two-ET scheme) on the other side, as in Fig. 13.

In fact, a two-path transmission may amplify the overall transmissible power, though reducing the coverage ratio (or vice versa) in comparison with the simple variator, transmitting by the variator only a fraction of the total power. This solution should provide a higher overall efficiency with a narrower coverage ratio (or vice versa) [**2**]. However, a multimode scheme (i.e. a particular combination of two or more split-way layouts commuting by brakes or clutches) can produce a further improvement for both power (and thus efficiency) and coverage ratio [**4**].

By a split-way transmission with assigned nominal power P_{in} and engine speed n_{in}, depending on the schemes, both the variator power fraction P_v and its input speed n_1 are not constant but functions of the overall speed ratio $\tau = n_{out}/n_{in}$; i.e. (see Fig. 13) [**2**], for $n_{in} =$ constant and $P_{in} =$ constant,

$$P_v = P_v(\tau), \qquad n_1 = n_1(\tau), \qquad \tau = \tau(\tau_{var}) \qquad \tau_{var} = n_2/n_1 \tag{15}$$

Therefore, for a given variator, the diagrams of the transmissible power versus τ_{var} for $n_1 =$ constant are more important than for the single path (in the limit, the variator could transmit the envelope of the power maxima in Fig. 3). The power capacity by spring thrust regulation has been compared with the IV, TV and AV for several schemes of split-way transmission in steady ideal operation.

In the choice of the optimal scheme, it becomes necessary to compare the variator power fraction $P_v = P_v(\tau)$ (i.e. the part of the total power that the scheme transmits by the variator) with the variator transmissible power, $n_1 C_1$, in order to determine the class of the variator (required size for an assigned nominal power P_{in}). The transmissible power is obtainable from equations (6) and (7), while the variator power fraction $P_v = P_v(\tau)$ can be obtained from equations such as (15).

The design of the SV consists in this case in calculating the spring preload and stiffness, the primary angular speed n_{in} and the aperture A_a (where A_a is a characteristic parameter of two ET schemes and represents the ratio between the variator input speed n_1 at maximum and at the minimum overall speed ratio τ [**2**]) in order to

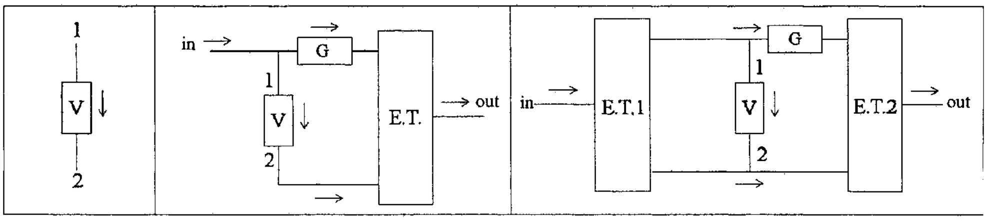

Fig. 13 Simple variator, mono-ET scheme and two-ET scheme

minimize the variator class; i.e. in finding [**17**]

$$n_{\text{in}}, F_0, K, A_\text{a}$$

$$\text{so that}\left[\max\frac{P_\text{v}}{n_1 C_1}\quad\text{for } \tau_{\min}\leqslant\tau\leqslant\tau_{\max}\right]\text{is minimum}\tag{16}$$

Operating in this way and restricting the ratio F_0/K to limit the spring axial size, the results of Table 2 can be obtained. The following conclusions can be drawn from Table 2:

1. The 'ideal' regulation IV allows clearly the smallest variator class for any condition.
2. The regulation AV at constant torque, typical in the automotive application, gives the highest variator classes, and is therefore the worst and least suitable one for a two-path transmission.
3. The typically industrial regulation TV is better than the AV but penalizes the performance of the simple variator with a wide coverage ratio.
4. The regulation SV is the best one for the simple variator and quite efficient for the two-path transmission in terms of variator class. It is certainly to be preferred to the three others because of the simple regulation of the axial thrust, which is implicit, with only a moderate impact on efficiency.

7 CONCLUSIONS

The optimized spring variator has the following advantages:

1. It has nearly the greatest transmissible power–size ratio either for single- or for two-path transmission.
2. It applies an axial thrust similar to the standard controlled CVT.
3. It has an efficiency considerably greater than that of the standard controlled CVT, especially for low vehicle speeds, and close to that of the slip limit variator.
4. It is much simpler than a hydraulic system, but its expected benefits would be best achieved if the hydraulic regulation could be eliminated for the speed ratio as well, for example using an electric actuator.

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