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SEMIAUTOMATIC VARIATOR FOR SPLIT POWER CVT'S

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Abstract

A well known practice uses mechanical variators to realize the so called Continuously Variable Transmissions (CVT) for the automotive application.

A remarkable problem for one of the most used CVT's is given by the hydraulic control of the axial thrust on the half-pulleys which is necessary to the torque transmission. This also involves an heavy decrease in the transmission efficiency.

In the present paper, the production of the required axial thrust by a simple spring is theoretically analyzed for several split way configurations, considering steady state and ideal operation.

Introduction

In the 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 CVT's can be identified with their power to size ratio, coverage ratio (ratio between 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][9] and thus, the power capacity may be maximized by optimizing the primary pulley angular speed. Moreover, given a particular variator, same power can be got increase by use of alternative CVT layouts.

For example, combining one variator with one or more epicyclic trains into some complex arrangement (split way or double way transmission) may amplify the power, though reducing the coverage ratio, or vice versa, in comparison with the simple variator [2][3][4]. A multi-mode scheme (that is a particular combination of two or more split way layouts commuting by brakes or clutches) can produce a further improvement for both power and coverage ratio [5].

Today, a wide use is made of the Van Doorne's metal pushing V-belt CVT [6], whose main drawback is the complexity of the axial thrust regulation on the half-pulleys.

At present, the thrust is generated by the oil pressure provided by a volumetric pump. The pressure level is controlled to guarantee the torque transmission. The minimum required axial thrust undergoes substantial variations with the speed ratio and the torque, and so the pressure is generally kept at a higher level than required.

Due to the reflux type regulation, pressure excess produces heavy efficiency losses, mainly at part load. Moreover, the power losses in the belt-pulley coupling are not small [7]. For these reasons, while the gear efficiency can easily exceed the number 0.9 at full load, a metal V-belt variator does not go beyond 0.8 and exhibits a sharp decrease at partial load [8][9][10].

Such a poor performance is also due to the partial utilization of the rated load, as the axial thrust and primary speed are not regulated for reaching the belt tension limit at any operating condition, but for example, only at the minimum speed ratio. For all these reasons, the convenience of using small variators to be loaded close to their top power appears evident [2].

Aiming at some increase of the engine class and of the transmission efficiency once assigned a variator, 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) is here theoretically analyzed for several split way configurations, considering steady ideal operation.

Of course, such a thrust is redundant at partial load, but does not affect the transmission efficiency since it does not imply any additional work.
Moreover, this device is surely much simpler than a hydraulic system, where the oil pressure has to be regulated in dependence on the load, angular speed and transmission ratio.
The study is carried out by optimizing the primary speed and caring for the proper spring design. The power capacity by this regulation is compared with the other most common strategies.
Of course, the expected benefits would be achieved at the most if the hydraulic regulation could be eliminated for the speed ratio as well, e.g. using electric actuators (like in some automotive power steering) which take up energy only during the transitional states.

Variator features

As far as the governing equations of the variator are concerned, the classic belt theory is applied (Eulero - Grashof).
The V-belt is considered like a flat belt by introducing an equivalent coefficient of friction. The friction among the rings and between the ring set and the blocks is neglected[1] [11][13].
The axial thrusts on the primary and secondary pulley, F_{z1} and F_{z2}, are respectively (see list of symbols at the end):

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

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

whence, the axial thrust appears clearly lower on the driven than on the driving pulley below the adherence limit.
The maximum input torque $C_{m,max}$, in correspondence with the highest admissible belt tension T_0 and the adherence limit (sliding arc $\Delta\theta^*$ equal to the winding arc $\pi - 2|\beta|$ on the smallest pulley) is:

$$C_{m,\max} = T_0 R_1\left[\left(1 - \frac{qV^2}{T_0}\right)\left(1 - e^{-f'(\pi - 2|\beta|)}\right)\right] \qquad \{3\}$$

or else, in dependence on the axial thrust on the primary pulley ($F_{z1,max}$)

$$C_{m,\max} = \begin{cases} 2F_{z_1,\max} f' R_1 \tan\alpha & \text{for } t_{var} \le 1 \\ \dfrac{2F_{z_1,\max} f' R_1 \tan\alpha\left(1 - e^{-f'(\pi - 2|\beta|)}\right)}{\left(1 - e^{-f'(\pi - 2|\beta|)}\right) + 4f'|\beta|} & \text{for } t_{var} \ge 1 \end{cases} \qquad \{4\}$$

or, on the secondary pulley ($F_{z2,max}$):

$$C_{m,\max} = \begin{cases} \dfrac{2F_{z_2,\max} f' R_1 \tan\alpha\left(1 - e^{-f'(\pi - 2|\beta|)}\right)}{\left(1 - e^{-f'(\pi - 2|\beta|)}\right) + 4f'|\beta| e^{-f'(\pi - 2|\beta|)}} & \text{for } t_{var} \le 1 \\ 2F_{z_2,\max} f' R_1 \tan\alpha & \text{for } t_{var} \ge 1 \end{cases} \qquad \{5\}$$

It is interesting that the torque is not affected by the primary angular speed ($n_a \propto V$) if $T_1 < T_0$. For $T_1 = T_0$ on the contrary, a speed increase must be compensated by a reduction of the axial thrust and thus of the transmissible torque.

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[1] For rubber V-belts, a higher accuracy can be achieved considering the "wedging effect" (Gerbert and other similar theories [14][16][17]). Nevertheless, these theories are not consistent with the experimental results for metal V-belts, save choosing a different coefficient of friction for each pulley [12]. An empirical equation for the pulley thrust balance is presented in [12], but this formula does not consider the maximum tension level that is being reached.

At last, a useful relationship for the calculation of the variator transmission ratio $t_{\mathrm{var}} = n_b / n_a = R_1 / R_2$, is given by the constancy of the belt length L:

$$L = (p + 2b)R_1 + (p - 2b)R_2 + 2h\cos b \qquad \{6\}$$

Regulation modes

By applying the above formulas and the data of table II, which are typical of the Van Doorne variator for Fiat, Ford and Nissan cars, the transmissible torque, power and the primary and secondary axial thrusts can be derived as shown in figures 1 and 2, for the ultimate power capacity of the variator, that is assuming always at the highest admissible tension working. Let's define this as the ideal variator I.V., regulated for the ultimate strength and the adherence limit. The power and torque ordinates are scaled by their common values at the ends of the speed ratio range for the particular input speed n_a=6047 rpm; the axial thrust is scaled by the highest belt tension T_0, which is here fixed at 5000 N.

The curves are plotted for constant input speeds n_a=1-2-3-4-5-6-7-8000 rpm, plus the two notable values n_a=4381 rpm (highest transmissible power at $t_{\mathrm{var}_{Max}}$) and n_a=6047 rpm (same power at $t_{\mathrm{var}_{\max}}$ and $t_{\mathrm{var}_{\min}}$).

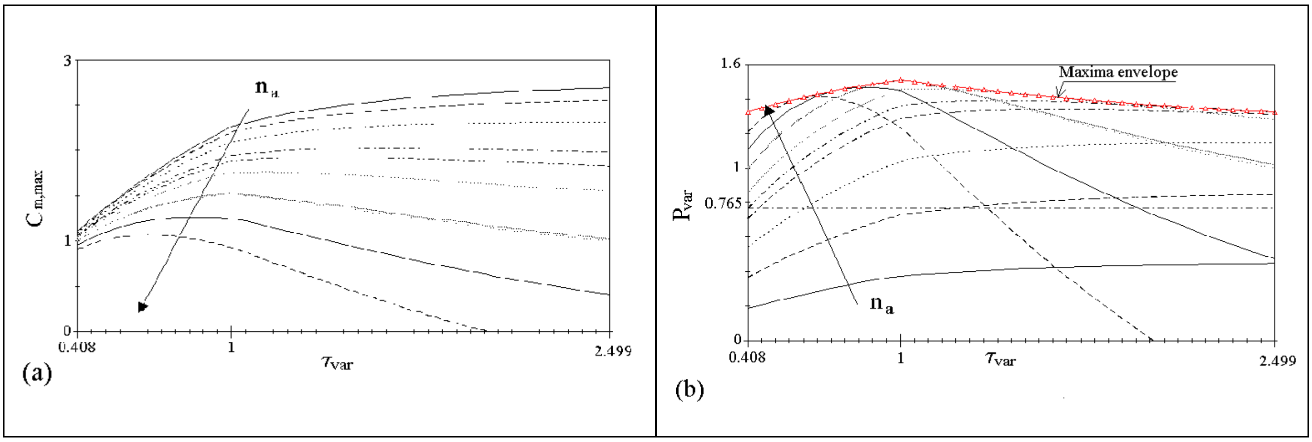

Figure 1 Transmissible torque (a) and power (b) versus speed ratio I.V..

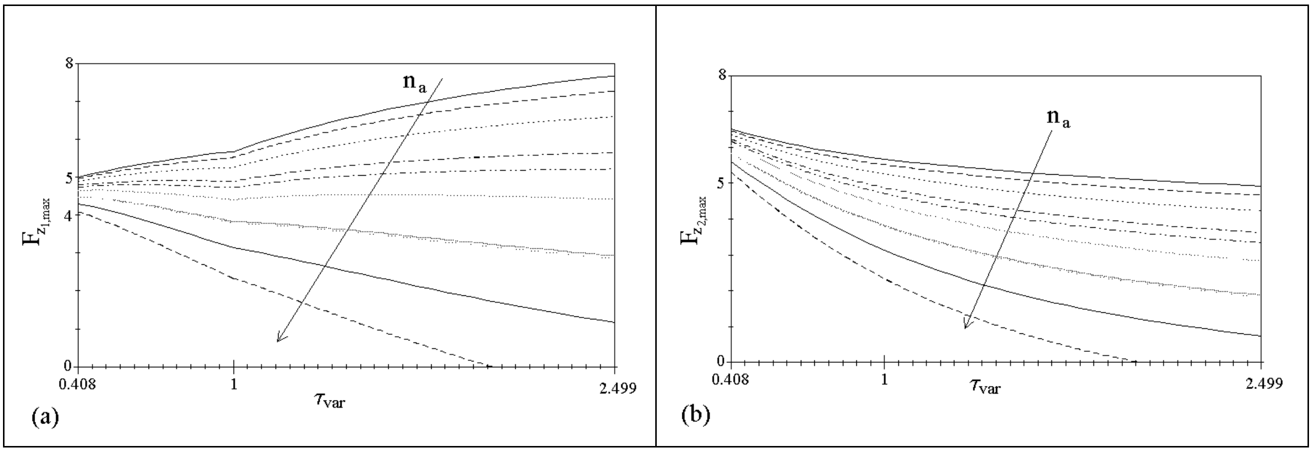

Figure 2 Axial thrust on the primary (a) and secondary pulley (b) versus speed ratio I.V..

For the most efficient use of the power capacity, the primary pulley should rotate at 6047 rpm; moreover, if angular speeds below 6047 rpm should also be required, the axial thrust regulation on the secondary pulley would be more convenient since it would be lower on the average [11].

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 figure 1b. Of course, the axial thrust is always lower than case I.V. which is loaded at the adherence limit and highest belt tension.

In the industrial application for example, for which the definition of technical variator T.V. is here introduced, the manufacturers declare a monotonous increasing characteristic power/speed ratio at constant input speed [18], while in the automotive variator (Van Doorne), which is hereafter called A.V., the power is kept constant on increasing t_{var} at constant n_a, by reducing the secondary axial thrust more than for case I.V.. In this manner, the efficiency improves because the hydraulic regulation is “active”, that is by the pump choking.
In order to compare the spring variator S.V. of the present analysis, with the commercial variators and the ideal variator I.V. defined above, the technical variator T.V. was identified with the ideal variator, though only for n_a 4381 rpm, so that the power-transmission ratio curves show always positive slope. As to the variator A.V., the further limitation of a constant torque is imposed, choosing the value of figure 1b at $t_{\mathrm{var}_{\min}}$ for the same speed (n_a 4381 rpm). Thus, figure 1b is useful for estimating torque and power for these variators types as well. In particular, the maximum dimensionless value 0.765 can be got for the variator A.V..
The belt tension T_1 of the variator A.V. is obviously lower than T_0 in almost the whole working range (curves n_a>4381 rpm are quite hypothetical).
The variator S.V. reduces the complexity of the axial thrust regulation by making it automatic and passive, with no power waste neither influence on efficiency. The axial thrust is automatically produced by a suitably preloaded spring between the movable half-pulley and an axially fixed wall.
Therefore, the S.V. 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 to let the S.V. performances approach the ideal variator I.V. as close as possible.
Using a compression spring “external” to the half-pulleys, in view of the power maximization for a given variator size, the axial thrust regulation on the primary pulley is unavoidable. In fact, on increasing the transmission ratio, the primary pulley closes up reducing the spring compression (axial thrust), while the spring behaves in the opposite way on the secondary pulley. In both cases, the diagram axial thrust versus t_{var} is different from the ideal case of figure 2, but this in a substantial way in the latter case (increase of the spring thrust instead of the ideal decrease, on increasing t_{var}), and in a lighter way in the former case, especially at a high angular speed.
The thrust produced by the spring extension $D_{x1,2}$ is:

$$F_{z_{s1,2}} = F_0 - KDx_{1,2} \qquad \{7\}$$

where the half-pulley axial displacements are given by:

$$Dx_1 = 2(R_1 - R_{\min})\tan a \ ; \ \ Dx_2 = 2(R_2 - R_{\min})\tan a \qquad \{8\}$$

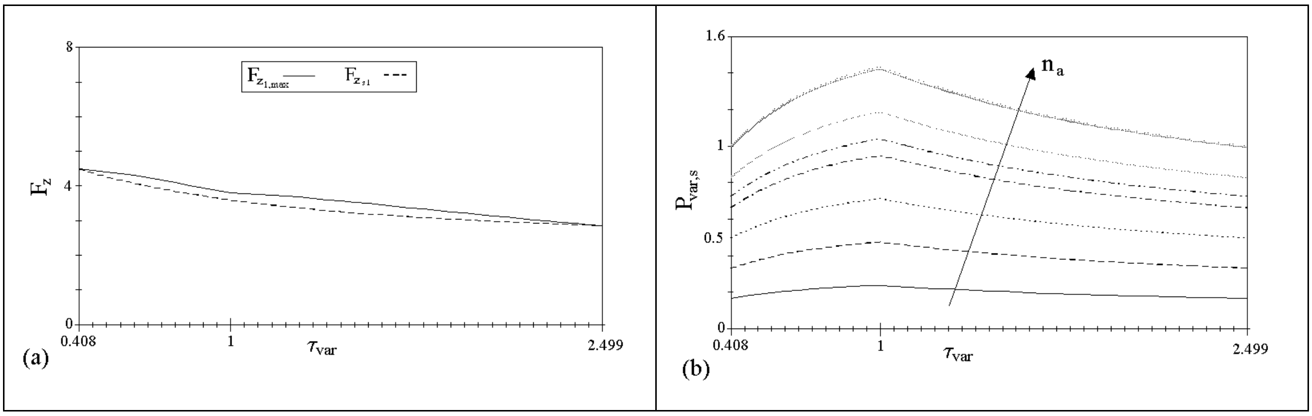

Figure 3 Axial thrust comparison between I.V. and S.V. (a) and transmissible power by S.V. (b). The S.V. is optimized for n_a=6047 rpm

Single way transmission

The spring parameters F_0 and K can be determined by optimizing the variator S.V. with reference to the I.V., imposing the same axial thrust at the t_{var} range endpoints, for the primary speed 6047 rpm, which yields the highest average power.

The comparison of axial force and power with the ideal variator I.V. is shown in figure 3.

Of course, the present variator S.V.:

- cannot work at input speeds n_a>6047 rpm, otherwise the belt breaks,
- transmits the same power at the t_{var} range endpoints,
- transmits a linearly increasing power with the primary angular speed n_a. The lower is the primary speed the more penalized is S.V. in comparison with I.V.

The variator S.V., when optimized at the most convenient angular speed in dependence on the coverage ratio of the transmission, permits same sizes (class) as the ideal variator I.V. while the technical variator T.V. and the automotive variator A.V. require larger dimensions, as shown in tab. II.

Double-way transmission

In order to reduce the variator dimension (class) necessary for an assigned nominal power it is convenient to split the transmission into two ways in parallel, the first one with a continuously variable ratio (variator V) and the other whit a fixed ratio (gear G): the two ways converge into an epicyclic train E.T. on one side, and are linked together (mono E.T. scheme) or converge into a second epicyclic train E.T. (two E.T. scheme) on the other side, as in figure 4 [2].

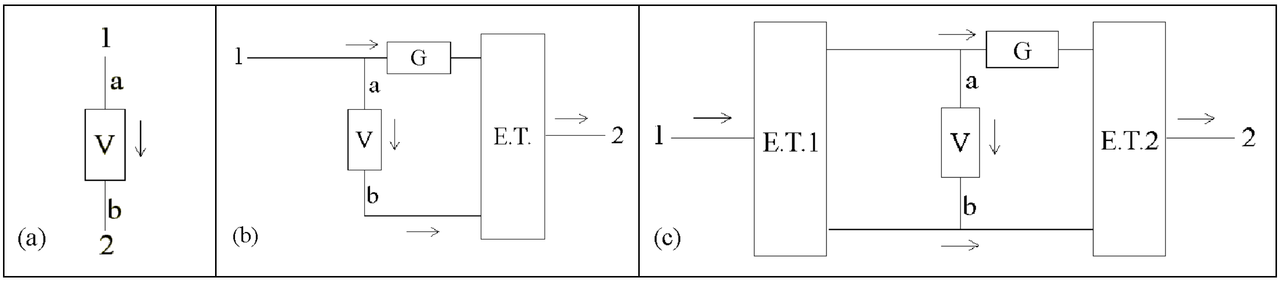

Figure 4 Simple variator (a); mono E.T. scheme (b); two E.T. scheme (c).

As the variator is generally the main source of power loss in the transmission, this solution should provide a higher efficiency of the whole transmission with a narrower coverage ratio or vice-versa, a wider coverage ratio with a lower global efficiency. In fact, the variator power fraction will be lower or higher than the nominal one respectively.

By this split way transmission type, with assigned nominal power P_1 and engine speed n_1, depending on the schemes, the variator power fraction P_{v_a} and its input speed n_a are not constant but functions of the global transmission ratio $t=n_2/n_1$. I.e. (see figure 4):

for n_1=cost. and P_1=cost. $\qquad P_{v_a} = P_{v_a}(t) \quad n_a = n_a(t) \qquad t = t(t_{\text{var}}) \qquad t_{var}=n_b/n_a$ {9}

Therefore, comparing with the single way, the diagrams of the power versus t_{var} for n_a=constant are here more important (in the limit, the variator could transmit the envelope of the power maxima in figure 1b), together with the possible penalization of the S.V. regulation (or T.V. and A.V.) compared with the ideal regulation I.V..

In the choice of the optimal scheme, a comparison becomes necessary of the power demand from the scheme $Pv_{demand} = P_{v_a}(t)$ with the available power from variator $Pv_{available}$ in order to determine the class. $Pv_{available}$ is obtainable from eqs. {4}{5}, while Pv_{demand} from equations of the type {9} [2][3].

The design of the S.V. consists in this case in calculating the spring preload and stiffness, the pri-

[2] Further details about double way transmission can be found in [2][3].

mary angular speed n_1 and opening A_a [3] in order to minimize the variator class, that is in finding

$$n_1, F_0, K, A_a \text{ so that } \left[\max \frac{P_{v_{demand}}}{P_{v_{available}}} \quad \text{for } t_m \le t \le t_M \right] \text{ is minimum.} \qquad \{10\}$$

Operating in this manner and restricting the ratio F_0/K to limit the spring axial size, the results of table I can be obtained.

Conclusions

Summing up, the following conclusions descend from table I:

- the “ideal” regulation I.V. allows clearly the smaller variator class in any condition;
- the regulation A.V. at constant torque, typical in the automotive application, gives the highest variator classes, and is then the worst and least suitable one for a double way transmission;
- the typically industrial regulation T.V. is better than A.V., but penalizes the performance of the simple variator with a wide coverage ratio;
- the regulation S.V. is the best one for the simple variator and quite efficient for the double way transmission in terms of variator class. It is certainly to be preferred to the other three because of the simple regulation of the axial thrust, which is implicit, with only a moderate impact on the efficiency.

Gear'r Change	I.V.			S.V.			T.V.			A.V.		
	Two E.T.	Mono E.T.	Simple	Two E.T.	Mono E.T.	Simple	Two E.T.	Mono E.T.	Simple	Two E.T.	Mono E.T.	Simple
I÷II	0,286	0,453	0,728	0,382	0,509	0,728	0,365	0,453	0,756	0,784	0,784	1,307
I÷IV	0,660	0,701	0,872	0,713	0,764	0,872	0,774	0,784	0,963	1,176	1,176	1,307
I÷VI	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÷I	0,402	0,907	-	0,600(*)	1,019	-	0,640	0,907	-	1,568	1,568	-
0÷III	1,049	1,071	-	1,049	1,098	-	1,164	1,164	-	2,091	2,091	-
0÷VI	1,521	1,532	-	1,634	1,661	-	1,869	1,869	-	2,875	2,875	-
RM÷I	0,804	1,814	-	1,432	2,037	-	1,280	1,814	-	3,137	3,137	-
RM÷III	1,581	1,952	-	1,702	2,059	-	1,976	2,015	-	3,660	3,660	-
RM÷VI	2,248	2,251	-	2,270	2,294	-	2,507	2,507	-	4,444	4,444	-

Table I Variator class summary

(*) Axial thrust regulation on the secondary pulley.

[3] The opening A_a is a characteristic parameter of two E.T. schemes [2][3].

Variator design data

Description	*Symbol*	*Value*
Minimum winding radius	R_{min}	2.87 cm
Maximum winding radius	R_{max}	7.02 cm
Pulley center distance	h	15.5 cm
Belt mass per unit length	q	1.64 kg/m
Grove half-angle	a	11°
Coefficient of friction	f	0.05
Admissible tension of belt	T_0	5000 N
Belt length	L	63.2 cm
Minimum transmission ratio	$t_{\mathrm{var}_{\min}}$	0.408
Maximum transmission ratio	$t_{\mathrm{var}_{\max}}$	2.449
Variator coverage ratio	A_{var}	6

Table II

List of the symbols

$A_a = n_{a_M} / n_{a_m}$

f belt/pulley coefficient of friction

$f' = f/\sin a$ apparent coefficient of friction

F_0 spring preload

h pulleys center distance

K spring stiffness

n_a angular speed of the variator primary shaft

n_{a_m} angular speed of the variator primary shaft at minimum global transmission ratio t_m (double way schemes)

n_{a_M} angular speed of the variator primary shaft at maximum global transmission ratio t_M (double way schemes)

n_b angular speed of the variator secondary shaft

n_1 angular speed of the whole transmission primary shaft

n_2 angular speed of the whole transmission secondary shaft

q belt linear density

R_1 primary pulley radius

R_2 secondary pulley radius

T_0 belt admissible tension

T_1 tension on the tight belt side

T_2 tension on slack belt side

V belt linear speed

a half-angle of V-belt cross section

b $\arcsin(R_1 - R_2)/h$

Dq^* sliding arc

q angular coordinate along winding arc

q_{0_1} angular width idle on primary pulley

q_{0_2} angular width idle on secondary pulley

t global transmission ratio

t_m minimum global transmission ratio

t_M maximum global transmission ratio

t_{var} variator transmission ratio

References

[1] A. Brandsma - J. Van Lith - E. Hendriks, "Push belt CVT developments for high power applications", Int. Congress on Continuously Variable Power Transmission CVT 99, Eindhoven, The Netherlands, September 16-17,1999.

[2] M. Andolina - A. Beccari, "Continuous Variation Transmission in Automotive Application: Extension of the Working Range of Automotive Transmission to Start and Reverse Motion, with Minimization of the Variator Dimension", Int. Congress on Continuously Variable Power Transmission CVT 96, Yokohama, Japan, 1996.

[3] M. Andolina - A. Beccari - F. Sorge ,"Trasmissione della potenza meccanica in doppia via con variazione continua del rapporto di trasmissione", Internal Report, Ins. of Machines, University of Palermo, Palermo, Italy, April 1996

[4] H.Vahebzadeh et al., “A Split-Torque, Geared-Neutral Infinitely Variable Transmission Mechanics”, SAE n° 905089, (1990).

[5] A. Beccari, "Schemi bimodali ottimizzati per la trasmissione di potenza con due vie in parallelo e rapporto di trasmissione variabile con continuità", Internal Report, Ins. of Machines, University of Palermo, Palermo, Italy, December 1997

[6] Trasmissione a variazione continua per la FIAT Uno Selecta", ATA maggio 1987

[7] T. Ide, "Effect of Power Losses of Metal V-Belt CVT Components on the Fuel Economy", Int. Congress on Continuously Variable Power Transmission CVT 99, Eindhoven, The Netherlands, September 16-17,1999.

[8] A. Beccari - F. Sorge, "Experimental Results for a Variable Speed Differential Drive", II British - Italian Workshop on Heat Engines, Bath, March 1991

[9] B. Van Vuuren , “Efficiency Improvement in CVT Systems", SITEV 1990 Technical Forum.

[10] A. Beccari - F. Sorge, "Improved Performance of a C.V. Transmission for Passengers Cars", 3° Int. Conference "Innovation and Reliability in Automotive Designand Testing" Firenze, Italy, April 8-10, 1992.

[11] A.Beccari - M. Cammalleri, "Variatore automobilistico a regolazione implicita", XIV Congresso nazionale AIMETA, Como, Italia, 6-9 Ottobre 1999

[12] T. Fujii - T. Kurokawa, "A Study of a Metal Pushing V-Belt Type CVT - Part 1: Relation between transmitted torque and pulley thrust”, SAE Transaction Vol. 102 Journal of Passenger Cars- SEC6 930666, (1993)

[13] M. Cammalleri, "Trasmissione meccanica in doppia via con variatore a regolazione semplificata", Thesis, Ins. of Machines, University of Palermo, Palermo, Italy, 1997.

[14] B. G. Gerbert, "Force and Slip Behaviour in V-belt Drives", Acta Polytechnia Scandinavica, Mech. Eng. Series n°67, Helsinki (1972)

[15] S. Kanehara - T. Fujii - O. Fujimura, "Characterization of a CVT Using a Metal V-Belt at Transitional States", Int. Congress on Continuously Variable Power Transmission CVT 99, Eindhoven, The Netherlands, September 16-17,1999.

[16] F. Sorge, "A Qualitative-Quantitative Approach to V-belt Mechanics", ASME J. of Mechanical Design v.118 (1996)

[17] F. Sorge, "A Simple Model for the Axial Thrust in V-belt Drives", ASME J. of Mech. Des. v.118 (1996).

[18] Texrope-Marzorati, "Technical Performance of Belt Variator", Italy.