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AN EXTREMELY COMPACT, HIGH TORQUE CONTINUOUSLY VARIABLE POWER TRANSMISSION FOR LARGE HYBRID TERRAIN VEHICLES

Luca Piancastelli[1], Salvatore Migliano[1] and Stefano Cassani[2]
[1]Department of Industrial Engineering, Alma Mater Studiorum University of Bologna, Viale Risorgimento, Bologna, Italy
[2]Multi Projecta, Via Casola Canina, Imola, Italy
E-Mail: luca.piancastelli@unibo.it

ABSTRACT

The planetary gear hybrid power train (PGHP) is known as one of the most compact speed reduction system. The PGHCVT (Planetary Gear Hybrid Continuous Variable Transmission) introduced in this paper varies continuously the reduction ratio by using an additional external, speed controlled, power source to the traditional thermal engine. The movement of the annular gear in the opposite way to the carrier P (Figure-2). In this way, when the annular is still, the minimum speed ratio is achieved. As the annular speeds up the speed ratio increases up to obtaining a still carrier. Theoretically, reverse ratio is possible by further increasing annular speed. However, speed limits on the annular gearing usually prevent the obtainment of a still carrier or a reverse motion. In any case, even a single stage planetary gearing obtains an extremely large transmission ratio variation. An example of 4WD (four Wheel Drive) vehicle is introduced in this paper with a preliminary design of the gearing system. The efficiency of the transmission is extremely high due to the very limited number of sliding contacts.

Keywords: hybrid electric vehicle, planetary gearing, CVT.

INTRODUCTION

In this paper, the torque converter/clutch assembly is not included in the model. The focus of this paper is on the transmission, not on its detailed design. It is a generally accepted design philosophy to keep the speeds as high as possible down to the final reduction on the wheel. In fact, transmission mass and volume depends on the torque, while the power is the torque multiplied by the speed. In thermal engines the vehicle is used from the speed where torque begins to be stable up to the maximum speed. Transmission design requires specific knowledge about transmission components, design solutions and vibrations. This latter depends largely on gear geometry. In general, it is convenient to design the vehicle inside out, starting from payload, engine and the transmission. The common practice to design the vehicle and then to install the power pack generally outputs critically underpowered overheating vehicles. In this field the availability of a CVT makes it possible to downsize the power-pack. In this way the volume and the mass of engine, transmission and fuel tanks can be significantly reduced. This numerical example demonstrates the feasibility of this PGHCVT proposal.

Among the power train systems for hybrid electric vehicles, the PGHP simplifies the transmission improving efficiency. Mayor automotive manufacturers use this transmission configuration. In 1997the Prius HEV (Hybrid Electric Vehicle) introduced the THS (Toyota Hybrid System) power-split (or parallel-series) hybrid power train.

The THS PGHP uses a planetary gear set and an additional generator and motor system to control the output speed in order to keep the input engine speed in the range of maximum efficiency. Therefore, by controlling the speed of the generator, the THS PGHP decouples the engine speed from the wheel speed. In THS PGHP systems the partial load performances are carefully mapped for the maximum efficiency operation of the power sources [1] [2] [3] [4] [5]. This is a problem for terrain vehicles, where the transmission ratio is extremely variable from plowing to "fast" road transfers. For this reason the use the THS PGHP on these hybrid systems is extremely difficult. The better flexibility of the PGHCVT solution makes it possible to solve the "mapping problem".

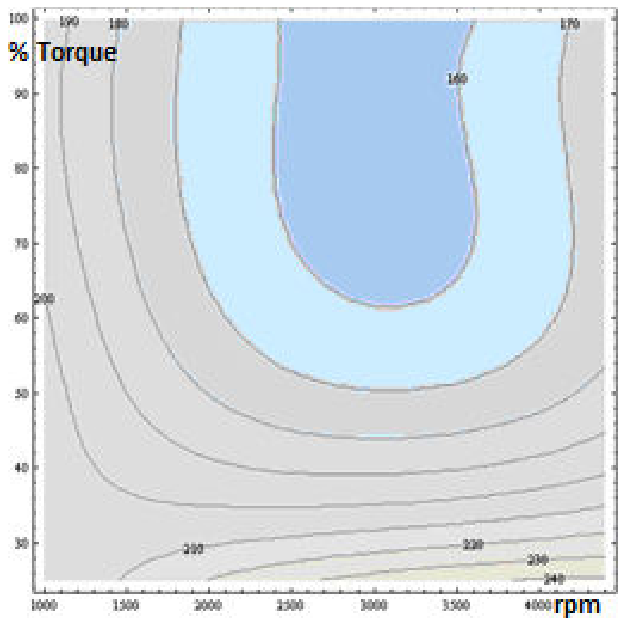

Figure-1. SFC (Specific fuel consumption) of the CRDID of Table-2.

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Therefore, the wide area of best efficiency of the CRDIDs (Common Rail Direct Injection Diesel) gives the possibility to reduce the installed power and to reduce fuel consumption.

THE POWER SPLIT SYSTEM

The PGHCVT consists of a planetary gear box (Figure-2). The planetary consists of several planet gears (2) revolving around the sun gear (1). The planet gears are mounted on a movable carrier (P) which rotates relatively to the sun gear. The annular gear (3) rotation is used to achieve the CVT. The largest CVT advantage is the optimization of the operating point of the engine and the consequent optimization of fuel consumption and torque.

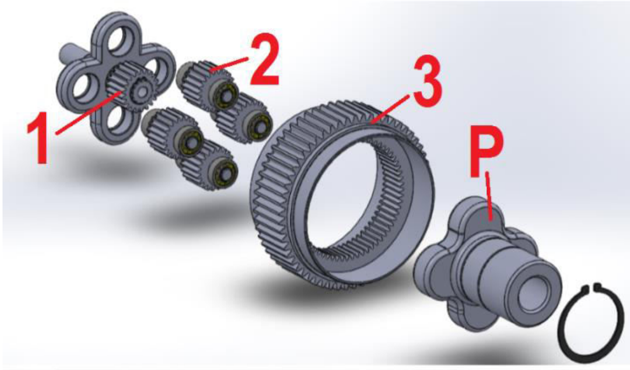

Figure-2. Planetary gearing.

In the PGHCVT system the reduction ratio is achieved by powering the shaft (1) and connecting the carrier (P) to the wheels through the final reduction gears. The annular gear 3 moves in a direction opposite to the carrier P. As the speed of the annular gear 3 increases the transmission ratio is changed up to maximum design value. Vehicle data are summarized in Table-1.

Table-1. Vehicle data.

Symbol	Description	Value	Unit
Df	Front wheel diameter	1.070	m
Dr	Rear wheel diameter	1.742	m
V_{max}	Maximum speed	11.11	m/s
V_{min}	Minimum speed	0.08	m/s
$rpm_{max}, n_{1,max}$	Maximum power engine speed	6000	rpm
rpm_{torque}	Maximum torque engine speed	4100	rpm
P	Maximum engine power	115.6	kW
T	Maximum engine Torque	201	Nm
i_{final}	Differential transmission ratio	4	-

Due to the difference between the rear and the front wheel diameters we need to a gearing that makes rear shaft slower to make tires rotating with the same vehicle speed (equation 1).

$$\omega_a r_a = \omega_p r_p \rightarrow \frac{\omega_p}{\omega_a} = \frac{r_a}{r_p} = \frac{0.535m}{0.871m} = 0.614 \qquad (1)$$

Now we have to determinate the engine that may be able to move the whole tractor and give enough power to overcome friction and rolling resistance. The worst condition for our vehicle is plowing.

Several experimental tests relating to plowing measured an actual average speed v_{me} = 1.30 m / s. Furrows 1.16m wide and 0. 49m deep led us to estimate a power of $P_r \approx 100$kW.

The principal data of a commercial CRDID of 155CV≈115.6kWare summarized in Table2.

Table-2. Engine data.

Symbol	Description	Value	Unit
$rpm_{max}, n_{1,max}$	Maximum power engine speed	6000	rpm
rpm_{torque}	Maximum torque engine speed	4100	rpm
P	Maximum engine power	115.6	kW
T	Maximum engine Torque	201	Nm

Since the shaft p is connected to the traction wheels and the engine moves shaft 1, the transmission ratio is calculated by equation (2):

$$\boldsymbol{\tau_{1,p}} = \frac{\boldsymbol{\omega_p}}{\boldsymbol{\omega_1}} \qquad (2)$$

In order to achieve V_{min} and V_{max}, the front wheels have to spin with these angular speeds (3) (3):

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$$\omega_{amax} = \frac{v_{max}}{r_a} = \frac{11.111\frac{m}{s}}{0.535\,m} = 20.768\,\frac{rad}{s} \quad (3)$$

$$\omega_{amin} = \frac{v_{min}}{r_a} = \frac{0.008\frac{m}{s}}{0.535\,m} = 0.0156\,\frac{rad}{s} \quad (4)$$

But considering the differential transmission ratio, the slow shaft after the CVT has these speeds (5) (6).

$$\omega_{lmax} = 4\omega_{amax} = 83.072\,rad/s \quad (5)$$

$$\omega_{lmin} = 4\omega_{amin} = 0.0624 rad/s \quad (6)$$

Being that the chosen engine has a stable torque erogation curve between 2000rpm and 5000rpm we can take this as the upper limit and calculate the maximum gear ratio (7).

$$\tau_{max} = \frac{\omega_{lmax}}{\omega_{1max}} = \frac{83.072\,rad/s}{523.599\,rad/s} = 0.159 \quad (7)$$

From the Willis' equations (8):

$$\begin{aligned} \omega_1^0 &= \omega_1 - \omega_p \\ \omega_2^0 &= \omega_2 - \omega_p \\ \omega_3^0 &= \omega_3 - \omega_p \\ \omega_p^0 &= \omega_p - \omega_p = 0 \end{aligned} \quad (8)$$

The ordinary gearing relative to this planetary one has 1 and 3 as input and output shaft (9) (10).

$$\boldsymbol{\tau_{1,3}} = \frac{\omega_3^0}{\omega_1^0} = \frac{\omega_3^0}{\omega_2^0}(-\mathbf{1})\frac{\omega_2^0}{\omega_1^0} = -\frac{z_1}{z_3} \quad (9)$$

$$\boldsymbol{\tau_{1,p}} = \frac{\omega_3 - \omega_p}{\omega_1 - \omega_p} \rightarrow \boldsymbol{\omega_p} = \frac{\tau_{1,3}}{\tau_{1,3}-1}\boldsymbol{\omega_1} - \frac{1}{\tau_{,13}-1}\boldsymbol{\omega_3} \quad (10)$$

At last we obtain equation (11).

$$\boldsymbol{\tau_{1,p}} = \frac{z_1}{z_1+z_3} + \frac{z_3}{z_1+z_3}\frac{\omega_3}{\omega_1} \quad (11)$$

Assuming to use spur gears with involute profile we must respect the noninterference condition that determines the minimum amount of teeth that the pinion should have according to the gear ratio of the mechanism (equation 12).

$$\boldsymbol{z_{min}} \geq \frac{2\tau}{-1+\sqrt{1+\tau(2+\tau)\sin^2\alpha}} \quad (12)$$

It changes from internal gear to external gear but the results can be summarized as below.

Table-3. Significant values of z_{min}.

τ	1	½	¼	0	- ¼	- ½
z_{min}	13	15	16	17	20	23

For example, we can choose the number of teeth of the satellite and the ratio between the number of teeth of the sun and the planetary, as z_2=21 e z_1/z_3=1/3.

On the planetary gearing of figure 2, equation (13) applies:

$$r_3 = r_1 + 2r_2 \rightarrow z_3 = z_1 + 2z_2 \quad (13)$$

Equation (13) shows that the numbers of teeth are uniquely determined, with z_1=21 and z_3=63.

Assuming that the annular gear is still, equation (14) must apply for correct meshing.

$$\boldsymbol{h} = \frac{z_1+z_3}{N_s} = \frac{21+63}{N_s} = \frac{84}{N_s} = \frac{4x3x7}{N_s} \quad (14)$$

For a correct operation therefore we take Ns=4 and put the satellites symmetrically and oriented at 90 ° each other.

Once we collected all these data, we can calculate the needed angular speed of the planetary to fulfill the maximum transmission ratio by using the inverse equation (15).

$$\omega_3 = \omega_{1max}\left[\left(1+\frac{z_1}{z_3}\right)\tau_{max} - \frac{z_1}{z_3}\right] = \\ = -63.53\,rad/s \quad (15)$$

In order to obtain the minimum ratio the solar has to rotate at 191 rpm (16).

$$\omega_{1min} = \frac{\left[\omega_{lmin} - \frac{z_3}{z_1+z_3}\omega_3\right]}{\frac{z_1}{z_1+z_3}} = \\ = 190.84\,rad/s \quad (16)$$

The complete planetary gearing may have an efficiency of around 96% (η_{tot}=0.96). It is then possible to write the system of equations (17).

$$\begin{cases} P_{diss} = C_1\omega_1(\eta_{tot}-1) \\ P_1 = C_1\omega_1 \\ P_3 = C_3\omega_3 \\ P_p = C_p\omega_p \\ \omega_p = \tau\omega_1 \\ P_1+P_3+P_p+P_{diss} = 0 \\ C_1+C_3+C_p = 0 \end{cases} ; \quad (17)$$

Where the first equation evaluates the dissipated energy, the last is the equilibrium of the system around the rotation axis of the planetary gearing and the second-to-last is the energy conservation equation. The remaining equations are the power, torque speed relations of the four shafts [6]. Shaft p goes to the wheels through the differential gearings. It is then possible to evaluate $C_1(\omega_1, C_p)$ (18) and $C_3(C_1)$ (19).

$$C_1 = -\frac{C_p}{\frac{z_1+z_3}{\omega_1 z_1+\omega_3 z_3}\left[\omega_1\eta_{tot}+\frac{\omega_1 z_1(1-\eta_{tot})+z_3(\omega_3-\eta_{tot}\omega_1)}{z_1(\omega_1+\omega_3)}\omega_3\right]} \quad (18)$$

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$$C_3 = C_1 \frac{\omega_1 z_1 (1-\eta_{tot}) + z_3(\omega_3 - \eta_{tot}\omega_1)}{z_1(\omega_1+\omega_3)} \tag{19}$$

We can also calculate the angular speed after the CVT (20) and its resistant torque (21).

$$\omega_r = 4\frac{v_{me}}{r_a} = 4\frac{1.30\frac{m}{s}}{0.535m} = 9.72\, rad/s \tag{20}$$

$$|C_r| = \frac{P_r}{\omega_r} = \frac{97000W}{9.72\frac{rad}{s}} = 9979.8\text{ Nm} = C_p \tag{21}$$

In order to have that value of ω_r using the (16) we can calculate$\omega_1 = 229.47\, rad/s$, and now we have all the data to solve the torque equations (18), (19).

$$\begin{cases} C_1 = 178.723\, Nm \\ C_3 = -907.169\, Nm \end{cases}; \tag{22}$$

The power to be supplied to the annular gearing to run at $\omega_3$with torque C_3 is $P_3 \approx 58$kW.

To provide the power necessary it is convenient to use two commercial electric motors with rated power of 30 kW at rated speed of 1465rpm≈ $153.414\frac{rad}{s}$.

We define the last reduction ratio between the electric motor and the outer annulus:

$$\tau_e = \frac{\omega_3}{\omega_{ne}} = \frac{63.53 rad/s}{153.414 rad/s} = 0.414 \tag{23}$$

It was chosen to split the needed power in two twin engine using an ordinary gearing from the electric motors to the annular gear (Figure-3). The electric motors not only increase the transmission ratio but also contribute to the output torque.

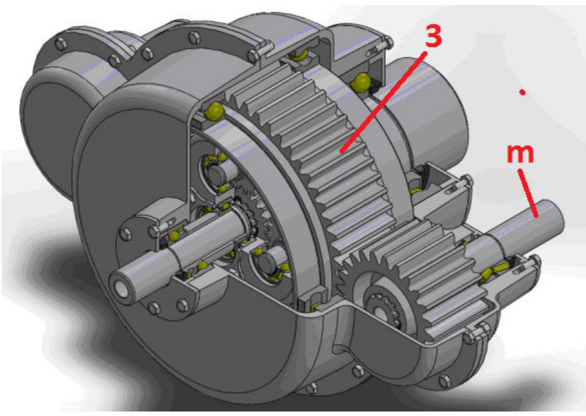

Figure-3. Electric motor ordinary reduction gearing.

A possible configuration of the solution described in the previous paragraphs is depicted in Figure-5.

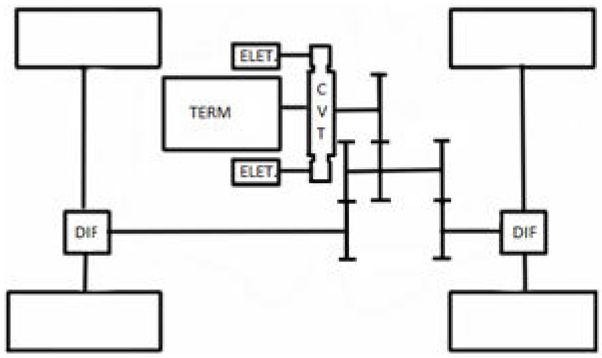

Figure-4. An example of the hybrid power assembly.

The internal combustion engine moves both the wheels and the generator. The solution is extremely simple and the number of components in comparison with a traditional transmission is reduced [7] [8] [9]. The possibility to have infinite transmission ratio makes it possible to reduce the engine size with benefits on manufacturing and operating costs. The electric motor contributes to the output torque in any condition. It is also possible to have reverse speeds, but it may be convenient to include a reverser after the engine. This choice depends on vehicle requirements and on engine performance. A clutch with a dynamic decoupling device or a torque converter should generally be included. In this example the engine runs the wheels and the generator. The electric energy from the generator is recirculated to the motor, through a suitable power electronic drive. A battery is generally included in the system. The transmission is extremely compact as it can be seen in Figure-4. Several other solutions are possible on the same concept.

CONCLUSIONS

An extremely compact hybrid planetary transmission PGHCVT is introduced in this paper. This solution makes it possible to vary continuously the speed ratio in a wide range. In this way it is possible to obtain also a reverse gear ratio. The two input shafts can be powered by any power source; the torque output will depend on both. Very high outputs can be obtained at low speed with large benefits on vehicle performance. Electronic differentials are easily obtainable with this system.

SYMBOLS

Symbol	Description	Unit	Value
Df	Front wheel diameter	m	1.070
Dr	Rear wheel diameter	m	1.742
V_{max}	Max vehicle velocity	m/s	11.11
V_{min}	Min vehicle velocity	m/s	0.08
rpm_{max}	Maximum power engine speed	rpm	6000
rpm_{torque}	Maximum torque engine speed	rpm	4100
P	Maximum engine power	kW	115.6
T	Maximum engine torque	Nm	201

τ_{xy}	Transmission ratio from gear "x" (input) to gear "y" (output)	-	
i_{final}	Differential transmission ratio	-	4
ω_p	Carrier speed	rad/s	
ω_1	Sun gear speed	rad/s	
ω_3	Annular gear speed	rad/s	
ω^0_x	Rotational speed of the gear "x" of the equivalent ordinary gearing	rad/s	
τ^0_{xy}	Transmission ratio from gear "x" (input) to gear "y" (output) of the relative ordinary gearing	-	
z_x	Number of teeth of gear "x"	-	
P_{diss}	Dissipated power	W	
P_x	Power on gear "x"	W	
C_x	Torque on gear "x"	W	
η_{tot}	Efficiency of the gearing	-	0.96

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