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AN EXTREMELY COMPACT HIGH RATIO CONTINUOUSLY VARIABLE POWER TRANSMISSION FOR SMALL HYBRID TRACTORS

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ABSTRACT

The planetary gear hybrid powertrain (PGHP) is known as one of the most compact speed reduction system. The PGHP introduced in this paper varies continuously the reduction ratio by using an additional external, speed controlled, power source to the traditional thermal engine. A continuous variable transmission (CVT) can be obtained in this way. An extremely large variation of the reduction ratio can be obtained even by a single stage planetary gearing. An example of a 2 wheel drive (2WD) tractor is introduced herein along with all the calculation necessary for the dimensioning of the gearing system. The efficiency of this system is extremely high since the motor power is added to the engine one.

Keywords: hybrid electric vehicle, planetary gearing, hybrid tractor, CVT.

FOREWORD

An example is proposed to explain the CVT solution. This is the best way to introduce the concept. The data are taken from existing small tractors. However this example suffers from many shortcomings. For example the torque converter/clutch assembly is not included in the model. The mass of the vehicle and the many data needed for the design of a transmission are not included. The focus of this paper is on the transmission concept, not on its design. In general it is a good philosophy to keep the speeds as high as possible down to the final reduction. This is due to the fact that the mass of the transmission goes with the torque, while the power goes with the torque multiplied by the speed. It is a good policy to use the torque curve fully, from the speed where torque begins to be "reasonable" up to the maximum speed. A good design may reduce transmission mass and volume by a factor 4 from an "commercial", apparently "low cost" choice.

Transmission design requires specific knowledge about transmission components, design solutions and it should include a simulation of the dynamics. In general, it is convenient to design the vehicle around the engine and the transmission and not vice-versa. The common practical approach to design the vehicle and then to install the transmission generally outputs very slow, underpowered machines.

This paper introduces the CVT concept and the equations that can be used for its calculation. The numerical example demonstrates the feasibility of this CVT solution.

INTRODUCTION

Among the powertrain systems for hybrid electric vehicles, the planetary gear hybrid powertrain (PGHP) is the most common for its remarkable advantages. PHGP simplify the transmission and improves efficiency. Toyota Prius, Opel Astra, Estima, RX440h, and so forth, use this hybrid configuration. The most common PHGP is also known as a power-split or parallel-series hybrid powertrain. Toyota marketed the Toyota Hybrid System (THS) in 1997 with Prius.

The THS PGHP uses a planetary gear set and an additional generating motor to control the output ring gear speed and the input engine speed. Other configurations of HEV powertrain require transmission devices such as the Continuous Variable Transmissions (CVT), Manual Transmission (MT), and Automatic Transmission (AT). By controlling the speed of the generator, the PGHP can move the engine operation point to a desired high efficiency region. Although the system functions as a CVT, its control system is much more complicated than those of a conventional mechanical CVT. Therefore, the partial load performances of the system should be carefully balanced for the optimal operation of the power sources which can improve the total efficiency. This is a problem for small tractors, where the use is extremely variable from hard mountain to soft sand, from plowing to “fast” road transfers. For this reason the optimization of these hybrid systems may prove to be extremely difficult. On the other hand, modern Common Rail Diesel Engines (CRDIDs) are extremely efficient is a wide range of loads (see Figure-1).

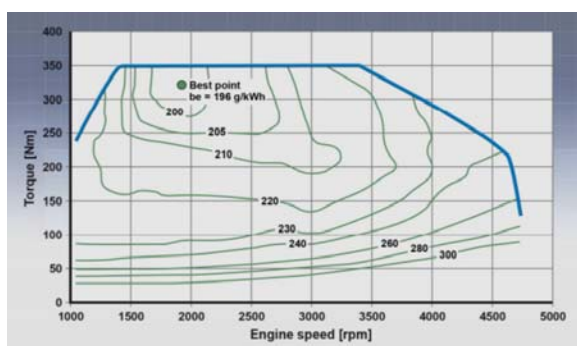

Figure-1.SFC [gr/kWh] of Mercedes -Benz OM 651 DE22 [1].

So the possibility to vary continuously in a wide range the transmission ratio makes it possible to use the CRDID in the area of best efficiency.

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The power split system

The split device consists of a planetary gear box (Figure-2). The planetary gear is a system that consists of several outer gears (planet gears (2)) revolving around a central one, (sun gear (1)). The planets are mounted on a movable carrier (P) which rotates relatively to the sun gear. There is an outer ring gear (annular gear (3)), which meshes with the planet gears. The advantages of planetary gearings are high power density, larger gear reduction in a small volume and multiple kinematic combinations. This planetary gear system is used to achieve our Continuous Variable Transmission (CVT).

The concept of CVT was conceived in 1490 by Leonardo Da Vinci. The idea was then developed in the last two centuries, due to a growing need of comfort and economic efficiency. CVT makes it possible vary continuously the rotational speed of the input shaft independently from the rotation speed of the output one. The largest resulting advantage is the optimization of the operating point of the engine and the consequent optimization of fuel consumption and torque.

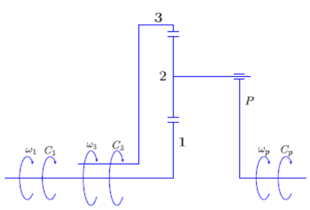

Figure-2. Planetary gearing.

In our case the CVT transmission is obtained by a planetary gearing. In this system the reduction ratio is achieved by powering the shaft (1) and connecting the carrier (P) to the rear wheels through reduction gears. As the speed of the motor shaft increases the transmission ratio is reduced up to maximum value.

Table-1. Tractor data

Symbol	Description	Value	Unit
D	Traction wheel diameter	1.114	m
V_{max}	Maximum speed	11.11	m/s
V_{min}	Minimum speed	0.36	m/s
rpm_{max},$n_{1,max}$	Maximum power engine speed	2600	rpm
rpm_{torque}	Maximum torque engine speed	1500	rpm
P	Maximum engine power	55.2	kW
T	Maximum engine Torque	270	Nm
i_{final}	Differential transmission ratio	4	-

Since the shaft p is connected to the traction wheels and the engine moves shaft 1, the transmission ratio will be (1):

$$\tau_{1p} = \frac{\omega_p}{\omega_1} \tag{1}$$

From the Willis' equation:

$$\begin{aligned} \omega_1^0 &= \omega_1 - \omega_p \\ \omega_2^0 &= \omega_2 - \omega_p \\ \omega_3^0 &= \omega_3 - \omega_p \\ \omega_p^0 &= 0 \end{aligned} \tag{2}$$

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

$$\tau_{1p} = \frac{\omega_3^0}{\omega_1^0} = \frac{\omega_3^0}{\omega_2^0}(-1)\frac{\omega_2^0}{\omega_1^0} = -\frac{z_1}{z_3} \tag{3}$$

$$\tau_{1p}^0 = \frac{\omega_3 - \omega_p}{\omega_1 - \omega_p} \Rightarrow \omega_p = \frac{\tau_{13}^0}{\tau_{13}^0 - 1}\omega_1 - \frac{1}{\tau_{13}^0 - 1}\omega_3 \tag{4}$$

This value (4) can be inserted in equation (1).

$$\tau_{1p}^0 = \frac{z_1}{z_1 + z_3} + \frac{z_3}{z_1 + z_3}\frac{\omega_3}{\omega_1} \tag{5}$$

Equation (5) is a line that intersects the x-axis on y=-z1/z3 and the y-axis on x=z1/(z1+z3) (Figure 3).

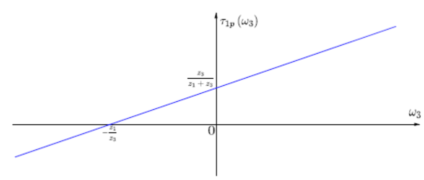

Figure-3. Line of equation (5).

The tractor data are summarized in Table-1.
It is then possible to calculate the traction wheel average rotational speed at V_{max} and V_{min} (6) (7).

$$n_{min} = \frac{60 \times V_{min}}{\pi D} = 6.19[rpm] \tag{6}$$

$$n_{max} = \frac{60 \times V_{max}}{\pi D} = 190.49[rpm] \tag{7}$$

In order to achieve V_{min} and V_{max} at rpm_{max}, the following gear ratios should be obtained $i_{min,total}$=420.17 and $i_{max,total}$=13.64. Since the differential gearing has a

reduction ratio i_{final}=4, the CVT should achieve i_{minCVT}= i_{minCVT}= i_{min}=105.04 and i_{maxCVT}= i_{max}=3.41.
The planetary gearing will be calculated with i_{min}=3.41 that will be obtained with the annular gear locked (ω_3=0).
The first step is to calculate τ_{12}=1.4184 and τ_{34}=0.2925 (8),(9).

$$\begin{cases} i_{\min}(\omega_3 = 0) = \dfrac{z_1 + z_3}{z_1} \\ \tau_{12} = \dfrac{z_1}{z_2} \\ \tau_{23} = \dfrac{z_2}{z_3} \end{cases} \Rightarrow i_{\min}(\omega_3 = 0) = 1 + \frac{1}{\tau_{12}\tau_{23}} \tag{8}$$

$$\begin{Bmatrix} m_{n1} = m_{n2} = m_{n3} \\ r_1 + 2 \times r_2 = r_3 \end{Bmatrix} \Rightarrow z_1 + 2 \times z_2 = z_3 \Rightarrow \tau_{12} = \frac{1}{\tau_{23}} - 2 \tag{9}$$

Then it is possible to calculate z_1=11.26 and z_2=19.72, by assuming k=1 (10).

$$z_{\min} = 2k \frac{1 + \sqrt{1 - (2\tau - \tau^2)\sin^2(\theta)}}{(2 - \tau)\sin^2(\theta)} \tag{10}$$

By assuming that, for manufacturing purpose, it is better to have z_{min}>17, the final choice is calculate z_1=29, z_2=21 and z_3=71.
It then possible to calculate τ with $\omega 3 \neq 0$ (11). This can be obtained from equation (5).

$$\omega_3 = \omega_1 \left[\left(1 + \frac{z_1}{z_3} \right) \tau - \frac{z_1}{z_3} \right] \tag{11}$$

With i_{min}=105.04 and the engine at $rpm_{max,\ it}$ is possible to calculate ω_3= ω_{3lim}=-107.6 [rad/s] = -1028 rpm. It is also useful to calculate the angular velocity of the planetary gearing at the maximum torque (rpm_{Tmax}=1500 rpm), with i_{min}. The result is ω_3= $\omega_{3limTmax}$=-62.05 [rad/s] = -593 rpm.
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 (12).

$$\begin{cases} P_{diss} = C_1\omega_1(\eta_{tot} - 1) \\ P_1 = C_1\omega_1 \\ P_3 = C_3\omega_3 \\ P_p = C_p\omega_p \\ \omega_p = \tau\,\omega_p \\ P_1 + P_3 + P_p + P_{diss} = 0 \\ C_1 + C_3 + C_p = 0 \end{cases} \tag{12}$$

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. Shaft p goes to the wheels through the differential gearings. It is then possible to evaluate $C_3(n_1,n_3)$ (13) and $C_p(n_1,n_3)$(14).

$$C_3 = -C_1\omega_1 \frac{1 - \dfrac{\eta_{tot}}{\tau}}{\omega_1 - \dfrac{\omega_3}{\tau}} \tag{13}$$

$$C_p = \frac{1}{\tau} C_1 \frac{\eta_{tot}\omega_1 - \omega_3}{\omega_1 - \dfrac{1}{\tau}\omega_3} \tag{14}$$

It then possible to calculate C_3 and C_p at maximum power and at maximum torque (see Table-2).

Table-2. Values calculated for i_{max} and i_{min} at T_{max} and P_{max}.

Engine Power (kW)	**Engine speed** (rpm)	**n_3** (rpm)	**C_1** (Nm)	**C_3** Motor Torque (Nm)	**C_P** (Nm)	**Motor Power** (kW)
55.2 (P_{max})	2,600	0	203	468	-671	-
55.2 (P_{max})	2,600	-1,028	203	476	-679	51
42.4 (T_{max})	1,500	0	270	624	-894	-
42.4 (T_{max})	1,500	-593	270	634	-904	42

From the data of Table-2, it is possible to choose the electric motor that acts on the annular gearing 3 (Figure-4).

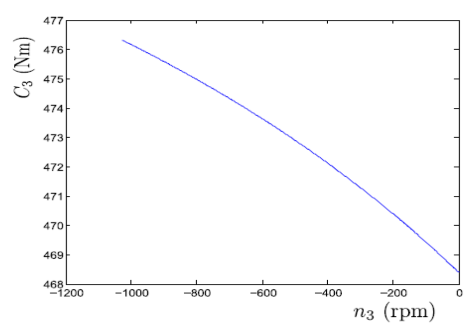

Figure-4. Torque-rpm curve required to the electric motor.

Electric motor choice

It was chosen to use an ordinary gearing from the electric motor to the annular gear (Figure-5).

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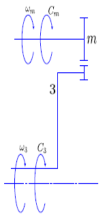

Figure-5. Electric motor ordinary reduction gearing.

From Table-2 the nominal data to choose the electric motor are $|C_3|$=476 [Nm] and n3=1028 rpm. It will also be necessary to have $|C_3|$=593 [Nm] and n3=634 rpm. The motor should have a stall torque $|C_3|$=624 [Nm]. It is possible to have an electric motor with the parameters shown in Table-3.

Table-3. Electric motor data.

Symbol	Description	Value	Unit
P_m	Nominal power	55	kW
C_m	Nominal torque	178	Nm
$n_{max,motor}$	Nominal speed	2,956	rpm
$C_{max,m}$	Max Torque	534	Nm
$C_{s,m}$	Stall Torque	427.2	Nm

It is then possible to calculate the ordinary gearing ratio (15).

$$i_{3,m} = \frac{n_{\max,motor}}{n_{3\max}} = \frac{2956}{2.87} = 2.87 \qquad (15)$$

The motor is verified (16) (17) (18).

$$C_{s,m} i_{3,m} \geq \max(C_{3sP\max}, C_{3sT\max}) \Rightarrow 247.2 \times 2.87 \geq \max(671{,}904) \Rightarrow 1226 > 904 \qquad (16)$$

$$C_{\max,m} i_{3,m} \geq C_{3,T\max} \Rightarrow 534 \times 2.87 \geq 634 \Rightarrow 1532 > 634 \qquad (17)$$

$$C_m i_{3,m} \geq C_{3,P\max} \Rightarrow 178 \times 2.87 \geq 476 \Rightarrow 510 > 476 \qquad (18)$$

The torque available at wheels is summarized in Table-4, the available torque is the sum of the (electric) motor and the one of the (internal combustion) engine. It is possible to control the transmission ratio by varying the speed of the motor but this will affect the maximum output torque available to the wheels.

Table-4. Torque available at wheel.

Torque at wheels (Nm)	**Engine torque** (Nm)	**Engine speed** (rpm)	**Motor torque** (Nm)	**Motor speed** (rpm)	**Reduction ratio**
91,731	203	2,600	165	1,028	420.17
9,098	203	2,600	163	0	13.64
121,885	270	1,500	217	593	420.17
122,020	270	1,500	220	0	13.64

The values of Table-4 were calculated assuming a unitary efficiency of the gearings. As it can be seen the electric motor adds it torque to wheel with the ratio calculated with equation (19).

$$i_{m,wheel} = i_{3,m} i_{3,1} = i_{3m} \frac{z_3}{z_1} = 2.87 \frac{71}{21} = 9.7 \qquad (19)$$

The electric motor not only increases the transmission ratio but also contributes to the output torque.

A few application examples

A possible configuration of the solution described in the previous paragraphs is depicted in Figure-6.

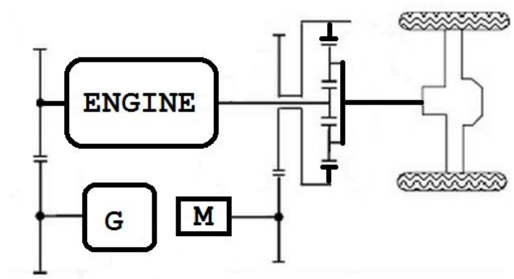

Figure-6. An example of the hybrid power assembly.

The internal combustion engine moves both the wheels and the generator G. Since the speed of the wheels is controlled by the electric motor M, it is possible to select the optimum speed for the engine and for the wheels. The solution is extremely simple and the number of components in comparison with a traditional transmission is reduced. 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 re-circulated 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-7. Several other solutions are possible on the same concept.

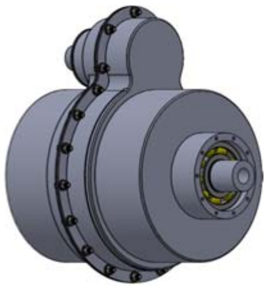

Figure-7. Transmission assembly.

A 4 wheel drive vehicle with an electronic central differential gearing is shown in Figure-8.

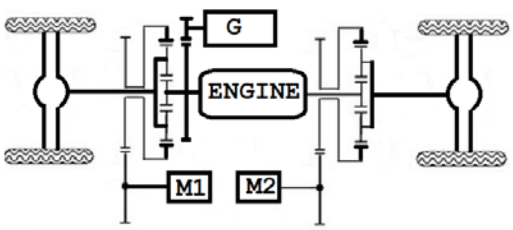

Figure-8. 4WD example.

In this case the central differential gearing is obtained by different speed ratio of the front and rear wheels. These speed ratios are obtained with different rotation speed of the motor M1 and M2. In the example of Figure-8 a single generator G is used. Several other configurations are possible.

CONCLUSIONS

An extremely compact hybrid planetary transmission is introduced in this paper. This solution makes it possible to vary continuously the speed ratio in a wide range by controlling the speed of an electric motor that acts on the annular gear. 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.

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Symbols

Symbol	Description	Unit	Value
D	Rear Wheel Diameter	m	1.114
V_{max}	Max vehicle velocity	m/s	11.11
V_{min}	Min vehicle velocity	m/s	0.36
rpm_{max}	Maximum power engine speed	rpm	2,600
rpm_{torque}	Maximum torque engine speed	rpm	1,500
P	Maximum engine power	kW	55.2
T	Maximum engine torque	Nm	270
τ_{xy}	Transmission ratio from gear "x" (input) to gear "y" (output)	-	
$i_{x,y}$	gear ratio from gear "x" (input) to gear "y" (output)	-	$I_{xy}=1/\tau_{xy}$
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	-	0.97
m	Electric motor		