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A MULTIPLE ENGINES, CONTINUOUS VARIABLE HYBRID TRANSMISSION FOR POWERING HEAVY TRACTORS WITH AUTOMOTIVE DERIVED COMMON RAIL DIESEL ENGINES

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ABSTRACT

Even in heavy duty tracked vehicles it is possible to implement Electronic stability control (ESC) system. This is a computerized technology that improves the safety of a vehicle's stability by detecting and reducing loss of traction (skidding). The use of software for the Electronic Stability may render controllable also inherently unstable vehicles. This can be a revolution for heavy tracked vehicles where the unstable dual drive transmission can be used. This very simple transmission system with its extremely high efficiency reduces weight, room and fuel consumption. As for the aircrafts, artificial stability improves handling, giving to the driver the possibility to reach the ultimate dynamic and static limits of the unstable vehicle. The overall performance of the vehicle is then enhanced. The possibility to use multiple engines instead of one, far from being a complication, makes it possible to use off the shelf market solutions for the engines. Automotive derived CRDIDs (Common Rail Direct Injection Diesel) can be used instead of specialized heavy duty direct injection diesel engines. An originally conceived CVT (Continuous Variable Transmission) is introduced for this purpose. This paper demonstrates both the feasibility and the durability of this solution. The TBO (Time between Overhaul) of an automotive CRDID used in a heavy duty vehicle is evaluated.

Keywords: variable hybrid transmission, common rail.

INTRODUCTION

Starting from a good car engine is always a good idea, since they are mass produced in millions of items. The modern diesel engine has the highest thermal efficiency of any internal or external combustion engine due to its very high compression ratio. Up-to-date automotive Common Rail Direct Injection Diesel (CRDID) have a thermal efficiency that exceeds 50%. "The vast majority of Europe’s new cars remain powered by gasoline or diesel motors. Diesel cars account for 55% of all new registrations, gasoline cars for 42%; all other technologies - hybrids, electrics, gas and ethanol-fueled vehicles - combine to make up the remaining 3%."[1]. High power is needed in heavy duty tracked vehicles. For this "market" the limited availability of engines and transmissions greatly affects the designer choice. Development of these engines is limited by the number and new designs are rare. Just to have an idea the first common rail truck was marketed by Renault only in 1999, with a delay of two years from the first cars. This fact is due to the production number of trucks compared with the one of the cars. In 2010, the European Union stock of light-vehicles reached the 239 millions. In the same year, the EU heavy vehicle stock was of only 35 millions [1]. The engine availability for new designs of extremely heavy-duty vehicles is very limited and the availability of transmission-types is even lower. Many of these transmissions may be traced back to WWII. Modern CRDIDs have High Pressure fuel Pumps (HPP) which supplies fuel constantly at high pressure with a common rail to each injector. Each injector has a solenoid/piezoelectric actuator operated by an Electronic

Control Unit (ECU), resulting in an extremely accurate control of injector opening times that can be varied on many control conditions, such as engine speed and loading, altitude, temperature and humidity. This provides engine extremely accurate engine control. It is perfectly possible to control the engine speed with the accuracy of 1/10 rpm with excellent performance and fuel economy. This fact opens new perspectives in the possibility of powering these extremely heavy duty vehicles with more than one engine. The multiengine option was already adopted with success in "sport cars" like the Mercedes A190 Twin (1999). The maintenance level of these modern CRDIDs is extremely reduced with limited scheduled maintenance and build-in, predictive On Board Diagnosis (OBD) systems. OBD controls the emissions and the efficiency of the engine, providing a tool to avoid unnecessary maintenance and improving the engine availability and reliability. This paper introduces new concepts to adapt car engines to these high-powered heavy-duty tracked vehicles. In the first part of this paper, steering of tracked vehicles is introduced, since the steering is an important part of the transmissions. Then, a few available engines from the automotive market are introduced. Automotive CRDID durability is then discussed. Finally a very simple solution for the installation of car CRDIDs on heavy duty tracked vehicles is proposed.

Traditional track steering [2, 3, 4]

The most obvious way to steer a tracked vehicle is to have a single engine-transmission driving both tracks, and to slow down one side or the other with brakes. A clutch system is added so that the power is first disconnected from the inside track and then the brake is applied. This is called "clutch and brake" steering. Clutch and brake steering is subtractive and non-regenerative. A tracked vehicle that is driving downhill turns the opposite direction of the driver's intent as the steering clutch is engaged. In fact the track on the side of the turn speeds up due to the loss of engine braking. The situation will be rectified once the brake engages. The clutch and brake systems can achieve wide turn radii only by engaging the clutch instead of the clutch+brake, with imprecise resulting turn. Finally, neutral steer is not possible. A few tracked vehicles use a single engine a separate transmission for each track. To turn the vehicle one track is put in a higher gear ratio than the other. Since this solution is too bulky and complex, there is usually a single gearbox for both tracks, but each final drive has a double epicyclic reduction gearbox. By engaging one of the epicyclic elements with a band brake the final reduction ratio of the drive sprocket is reduced, producing a turn. By disengaging all the brakes, the steering mechanism goes into neutral. This geared steering is almost always used with an auxiliary clutch and brake system. There are negligible efficiency (<1%) and weight penalties (1-2%) for having and additional epicycle into the final drives. This system is not additive. As long as the clutch and brake system is not activated, it is regenerative. It is continuous as it uses band brakes and epicyclic gearing. In its most common form, it only provides a single radius of wide turns in addition to the auxiliary clutch and brake system for very tight turns. Normally the single radius of turn provides a compromise between "high" and low speed driving; being too tight for the former and too narrow for the latter. There is no smooth transition between turning and not-turning.

In Controlled Differential Steering (CDF or Cletrac) the output from the transmission runs through a differential gear. The steering brakes are still present. When one of the brakes is engaged, the differential diverts the power to one track or the other for steering. In Cletrac systems the power is diverted through a series of idler gears. CDF has a small efficiency penalty (1-3%) on the system because of the additional gearing. CDF is additive and regenerative, as one track speeds up while the other slows down. Both tracks are energized at all times during a turn, so the system is also continuous. At top efficiency this system produces a turn of a single radius. However, with reduced efficiency, turns of greater radius can be achieved by slipping the brake. An important problem of CDF comes from the differential. As the tracked vehicles the tank hits a patch of uneven ground, more power will be diverted to the track with less resistance on it. For this reason, a CDF equipped tracked vehicle has the tendency to "self-steer" in off-road conditions. This behavior requires continuous corrections by the driver. A differential locking system may be added to reduce this tendency.

Double (Wilson, Maybach) and triple (Merrith-Brown, Merrith-Maybach) differential systems are an evolution on the controlled differential system. In these systems there are two input shafts leading to the steering unit. One input shaft is from the transmission, while the other comes "directly" and independently off the engine. The power is then provided to the steering mechanism even if the tracked vehicle is in neutral. This gives the tracked vehicle, the ability to turn about on its own axis (neutral steer). Double and triple differential systems also output different turn radii depending on which gear the vehicle is in. The steering brakes are still added to achieve different turn radii.

Double and triple differential systems are still very efficient, but less than CDF or geared steering due to the friction from all the additional gears. These systems have a tendency to self-steer when they hit uneven terrain, however a differential locking system may be added to reduce this tendency.

In Double Differential with Hydrostatic/Electric Steering Drive there are two inputs to the steering mechanism; one leading from the gearbox, and the other leading from a variable-displacement hydrostatic pump or a variable-speed electric motor. This system offers all the advantages of the double/triple differential system, except instead of having one or two discrete turn radii per gear ratio, it provides a smoothly variable range of turn radii based on the displacement ratio of the hydrostatic pump or on the speed of the electric motor. In addition, the pump/motor torque reduces the tendency of the vehicle to self-steer on uneven terrain through a cross-action resistance in the differential. Due to the low efficiency of the hydrostatic motor, the hydrostatic steering drives are substantially less efficient than purely mechanical double or triple differentials. In the case of the electric motor, it depends on the efficiency of the electric motor, which is always a compromise with dimensions of motor and controller. This solution offers the best control and it is widely used in modern tracked vehicles especially in the double differentials version. These highly specialized transmission systems are produced in small numbers and their availability on the market is consequently reduced.

Dual drive inherent instability

The most obvious way to drive the tracks at different speeds is to have an engine for each track. To turn the vehicle, it is sufficient to increase the speed of one engine and decrease the speed of the other. This steering method has not been successful, although some very early tracked vehicles and Ferdinand Porsche's various mechanical abortions did use this method of steering.

In a few dual drive electric hybrid vehicles (Porsche Tiger, IS-6...); the energy was recovered from the slowing track by running the electric motors in reverse to act as generators and then giving that power to the motor of the speeding up that. This made the hybrid dual drive steering regenerative (up to 65%). However larger motors were required to handle the additional power.

If the transmissions have reverse gearing, dual drive provides infinitely variable turn radii and allows neutral steer. It is very mechanically efficient, since there is no power flow from the transmission to the steering mechanism. Unfortunately, dual drive tracked vehicles are inherently unstable. This means that they are poor at manually driving in straight lines. In fact it is difficult to balance manually the speed of both engines. Also the tracked vehicle will tend to veer if it hits irregular terrain, being the two drive sprockets not cross-linked.

Enhanced stability for unstable tracked vehicles and dual drive

Stability system is successfully installed in cars since 1997, when Mercedes Benz developed the ESP (Electronic Stability Program). The car system uses 4 sensors on wheels to measure rotational speeds, a 3D sensor to measure accelerations, velocities and angles, the throttle position and the steer angle. An ECU (Electronic Control Unit) measures the car slip and, in case of necessity, cuts the throttle and the direction by applying brake on each wheel in a proper way. The objective is to reduce the error on desired direction inputted by the driver via the steering wheel. The control system is a digital PID (Proportional Integrative Derivative) control. The ESP is superimposed on an ABS (Antilock Braking System) that avoid wheel blockage. This system works very well in many conditions and turns off when adherence is too low or control is beyond recovery leaving to the driver a normal car. For every car model a proper tuning is to be made. Normally a car simulator calculates the optimum values to be inputted into the ECU, then standard experimental tests are performed on special test circuits. These were designed for the specific application of the ESP/ABS. These systems are difficult to implement on a tracked vehicle in uneven terrain. This due to the extremely variable conditions that, in most cases, turn the ESP off, leaving the driver to steer manually the vehicle. Brake efficiency on tracked vehicle is also reduced by vehicle inertia and the track-soil friction. For the electronic stability program a dual drive is highly recommended since it can control independently the velocity of the two tracks. Single track velocity can be easily controlled by the CRDID ECU (or FADEC). The diesel braking capability reduces the necessity to use an external brake enhancing the endurance of the brake and reducing its thermal sizing. The unstable behavior of the dual-drive tracked-vehicles greatly improves the response of the system. However, the tracked vehicle system is highly not linear. In this case an Improved Electronic Stability Program (IESP) based on a fuzzy logic algorithm may be the best choice [5]. The IESP may use the same hardware (sensors/actuators) of a standard ESP (Mercedes-Benz™); only the control system is completely different. It is sufficient a software upgrade to convert a car from ESP to IESP. The IESP reads the driver steering angle and the dynamic condition of the vehicle and selectively acts on throttle and track-engine speed in order to put the vehicle on the required direction even during a sudden and complete loss of adherence. The fuzzy logic advantage is the capability of self-tuning. Once the inertia data of the vehicle are introduced into the software, the fuzzy control system does not need any further tuning. On the contrary the standard ESP, which is based on a traditional PID control system, needs to be adapted to every car model and the tuning differs from sedan, cabriolet and 2 volumes of the same car. The traditional ESP tuning process is long and expensive and experimental tests are required. Traditional ESP cannot recover direction from a spin and cannot control the car direction after a tyre burst. The only known limit of the IESP proposed in [5] is small oscillations in very limit condition. This oscillation affects not only the yaw axis, which is normal, but also the pitch and the roll giving the impression to the driver of an unstable and unsafe handling. However, this “unsafe felt” l behaviour takes place in a condition very close to the physical limit of the vehicle dynamic. In this case these oscillations may be a good warning to the driver to behave more properly. “Physic cannot be fooled” as Richard Feynman said about the famous Shuttle accident. As for the aircrafts, artificial stability improves handling, giving to the driver the possibility to reach the ultimate dynamic and static limits of the unstable vehicle. The overall performance of the vehicle is then enhanced. The extremely simplified solution of the dual drive transmission with its extremely high efficiency reduces also the fuel consumption.

Available CRDID from the automotive market

A few CRDIDs from the automotive market are known to the authors: their performances are summarized in Table-1.

Table-1. CRDIDs from the automotive field.

Engine	**Automotive power** (HP)	**Ultimate power** (HP)	**Crankcase**
Audi V12tdi	500@4,000rpm	900@5,200rpm	CGI
Audi V8tdi	327@4,000rpm	600@5,200rpm	CGI
Fiat 2000jtd	190@4,500rpm	250@5,200rpm	Cast Iron
Peugeot 1600 HDI	115@3,800rpm	200@5,000rpm	All. alloy
Fiat 1300jtd	95@4,400rpm	200@6,000rpm	Cast Iron

A few comments are needed on the ultimate power concept. Automotive engines have the advantage of torque at low rpm. It is the torque that moves the wheels through the transmission. In the case of tracked-vehicles the used range is from 50% to 100%. Intake and exhaust manifold are usually replaced and all the parts not strictly necessary are removed. The engine "naked" of Table-2 is the engine without accessories and turbocharger. In a few cases also the crankshaft and the camshafts are replaced. The head is also tuned up, with larger valves, polished ducts and re-worked swirl-ducts. Surface treatments may easily improve wear resistance. Just as a first approximation the output power of a tracked-vehicle conversion can be evaluated with (1).

$$P_{\max} = \frac{T_{\max Auto} e \pi n_{\max}}{30} \quad (1)$$

Where $T_{maxAuto}$ is the maximum torque of the automotive engine, e is a factor that takes into account that the conversion from Euro 6 to the much less stringent requirements of heavy duty vehicles makes it possible to eliminate several pressure drops in the air-exhaust system of the engine. With the use of SCR (Selective Catalytic Reduction) e can be around 1.15 (15% increase in torque) in many engines. Table-2 summarizes the values calculated with (1) and the ultimate values (racing field).

Table-2. Summary of results of equation (1).

Engine	**Automotive power** (HP)	**Calculated with** (1) (HP)	**Ultimate power** (HP)
Audi V12tdi	500@4,000rpm	692	900@5,200rpm
AudiV8tdi	327@4,000rpm	420	600@5,200rpm
Fiat 2000jtd	190@4,500rpm	290	250@5,200rpm
Peugeot 1600 HDI	115@3,800rpm	138	200@5,000rpm
Fiat 1300jtd	95@4,400rpm	137	200@6,000rpm

So, even with this extremely simplified method, the results are acceptable, with the exception of the FIAT 2000jtd. As it can be seen the available commercial CRIDIDs are in the range of 168HP for the dual "SmartCDI" up to the 1, 384HP of the dual "AudiV12tdi".

CRDID durability

The maintenance schedule is influenced by the “weariness” parameter. Historically TBO has been expressed in “hours”. In some cases it meant the total number of revolutions of the crankshaft measured by a device installed on the crankshaft. Another parameter is the lubricant consumption rate. When this rate overcomes the limit given by the manufacturer, the engine should be overhauled. The availability of the FADEC with OBD (On Board Diagnosis) makes it possible to improve the TBO criteria with a more sophisticated algorithm. The result is an on-line indication of the residual engine life for proper TBO scheduling.

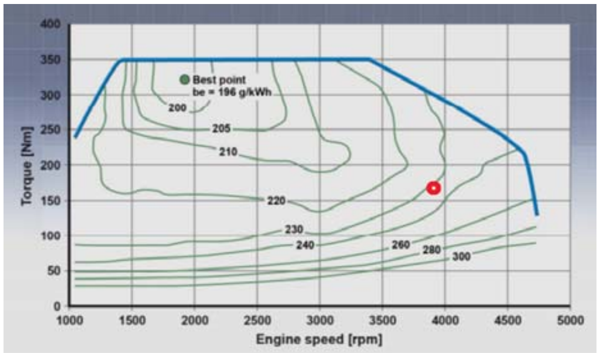

Figure-1. SFC [gr/kWh] of OM 651 DE22 [6]; 196 g/kW=42.6% efficiency.

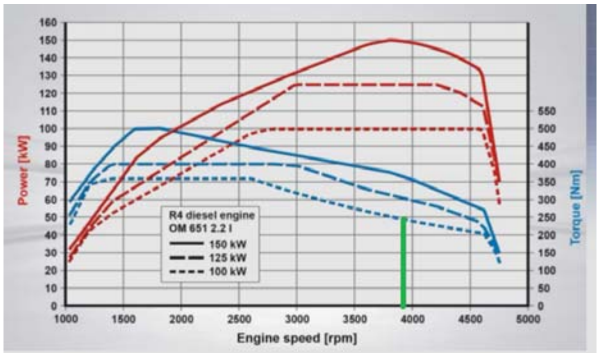

Figure-2. Power and torque curves for main OM 651 variants [6].

Heavy duty CRDID durability (TBO) based on Load Factor (LF) and Power Factor Just for explanatory reasons, since fuel consumption is not significatively affected by the automotive engine size, a 100 kW CRDID was chosen. Just as an example, from Figures 1 and 2 it is possible to obtain Table-1.

Table-3. Typical fuel consumption at average power settings.

Fuel consumption	
Max (100%-100 kW)	20.91 lxh
Max continuous (92%-92 kW)	18.47 lxh
Off road (73% 73 kW)	14.05 lxh
Road (60% 60 kW)	11.55 lxh

The values calculated in Table-3 are just for explanatory reasons; the true data of the engine should be supplied by the Manufacturer.

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Table-4. Typical 1 hour long "heavy duty cycle".

Power setting (%)	**Time** (min)	**Duration** (%)	**Fuel burned** (l)	**Fuel burned** (kg)
100	0.6	1	0.2	0.18
92	6	10	1.85	1.55
73	41.4	69	10	8.37
60	12	20	2.3	1.92
Total	60	100	14.35	12.02

Table-2 summarizes a typical 1 hour long "heavy duty cycle". From Table-4, it is possible to calculate an approximated fuel consumption of 14.35 l (12.02 kg). In a very simplified durability model, an engine has a lifetime that can be measured in weight of fuel burned. You can run that mass of fuel through the engine in a short time period if you are extracting large amounts of power, or you can take much more time to burn the same amount if you only extract small amounts of power. The Load Factor (LF) represents the relationship between fuel burned and the number of hours you are taking to burn it. At max continuous power, the fuel burned would have been 20.91 l. Hence the LF can be calculated (2).

$$LF = \frac{Fuel_{Burnt}}{Fuel_{\max ratedpower}} = \frac{14.35}{20.91} = 0.68 \qquad (2)$$

The typical small car load factor is 0.44. It is then possible to calculate the load factor ratio LF_{ratio} (3).

$$LF_{ratio} = \frac{0.68}{0.44} = 1.54 \qquad (3)$$

A small car used for typical automotive light duty will go for 250,000 km without rebuild when properly maintained. At the average speed (city car) of 28 km/h, this means a TBO=8, 930 h. On a pure LF basis the aircraft engine will last 3, 800 h (4).

$$TBO_{heavy\_vehicle} = \frac{TBO_{automotive}}{LF_{ratio}^2} = \frac{8930}{(1.54)^2} \approx 3800\,[h] \qquad (4)$$

The basic concept is that the LF reduces the life with a quadratic law. However in a heavy duty conversion, it is possible to increase the power output of about 15% (1). The TBO of the automotive conversion will be then 2, 470h (5).

$$TBO_{heavy\_vehicle} = \frac{TBO_{automotive}}{LF_{ratio}^2 P_{ratio}^3} = \frac{8930}{(1.54)^2 (1.15)^3} \approx 2470\,[h] \qquad (5)$$

The durability (TBO) of the automotive conversion is then satisfactory.

The CVT for the dual drive powerpack

In the case of CRDIDs, the practical limitation is 6,000 rpm, due to the common rail injection system inherent limits. Commercial high power Direct Injection Diesels run at about 2,000 rpm. Their torque value should be divided by a factor ranging from 2.5 up to 3.0 for a direct comparison with automotive derived CRDIDs. For this reason an automotive CRDID of 600Nm may be compared with a 1, 800Nm classical heavy duty diesel engine. The adoption of a Continuous Variable Transmission (CVT) further increases the advantage, providing the right transmission ratio for every condition.

In the proposed CVT, the main engine is connected to a planetary gearbox. Its annular gear has the velocity controlled by a generator in order to obtain an epicyclical continuous variable drive for the rear sprockets. The generator outputs a continuously variable brake torque, from stall up to the maximum vehicle speed. The stall condition will obtain the maximum reduction ratio with the annular external gear "locked". The electric power obtained in this way is recirculated to the front sprockets to obtain an hybrid, interconnected, four sprockets, dual drive transmission system. In this way it is possible to reduce and control track tensioning, while offering the possibility to move the vehicle even in case of single engine failure (Figure-3).

Figure-3 shows a possible solution a heavy duty dual drive tracked vehicle. The rear sprockets S1 and S3 are powered by engine 1 and engine 2. The CVT of S1 is controlled by the generator G1 that energizes the sprocket s4 on the opposite side through motor M1. In this way it is possible to power the right track in case of failure of engine 2. In normal operation, the tension of the right track is controlled by M1. The battery and the drives of M1, M2, G1 and G2 control the electric power recirculation. In case of full failure of the electric power system, the "safety clutch" on the generators G1 and G2 are engaged and the slowest transmission ratio is available. The steering is steel possible by engaging selectively the brakes installed on sprocket S1 (left track) and S3 (right track). Also a full electric mode with motor M1 and M2 is possible if the battery has sufficient energy.

In this system the reduction ratio is achieved by powering the shaft (1) and connecting the planetary pinions carrier (P) to the rear wheels (through reduction gears). In order to achieve the maximum reduction ratio the annular gear (3) is kept to the slowest speed by the "braking torque" of the generator G. As the speed of the

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generator shaft increases the transmission ratio is reduced up to direct (τ=1) when ω3= ω1= ωP→ ω2=0. The heavy vehicle data are summarized in Table-5.

Table-5. Heavy vehicle data.

Symbol	Description	Value	Unit
D	Sprocket diameter	0.67	m
V_{max}	Maximum speed	18.56	m/s
V_{min}	Minimum speed	0.36	m/s
rpm_{max}, $n_{1, max}$	Maximum power engine speed	5,200	rpm
P	Maximum engine power	441	kW
T	Maximum engine Torque	1,150	Nm

It is then possible to calculate the rear wheel average speed at V_{max} and V_{min} (6) (7).

$$n_{\min} = \frac{60 \times V_{\min}}{\pi D} = 10.26[rpm] \tag{6}$$

$$n_{\max} = \frac{60 \times V_{\max}}{\pi D} = 529.6[rpm] \tag{7}$$

In order to achieve V_{min} and V_{max} at rpm_{max}, the following gear ratios should be obtained i_{max}=506.72 and i_{min}=5.67. It is possible to assume that the final reductions on the wheel totalize i_{final}=i_{min}=9.82. The CVT should achieve then i_{cvt}=51.56.

A two stage planetary gearing is then required. However, since the generator does not work at stall condition (ω_3=0), it is better to increase i_{CVT}=56. It is divided into two stages: i_1= i_2=7.484.

$$\left\{\begin{aligned} & i_1(\omega_3 = 0) = \frac{z_1 + z_3}{z_1} \\ & \tau_{12} = \frac{z_1}{z_2} \\ & \tau_{23} = \frac{z_2}{z_3} \end{aligned}\right\} \Rightarrow i_1(\omega_3 = 0) = 1 + \frac{1}{\tau_{12}\tau_{23}} \tag{8}$$

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

Equations (8) and (9) make it possible to evaluate z_1, $z_2$and z_3. The final choice is z_1=17, z_2=47 and z_3=111. The true gear ratio i_{1true}=7.53 and i_{CVT}=56.69.

The "emergency condition", with the brake on the generator inserted, will then guarantee a maximum speed of 2.6 km/h (10).

$$V_{emergency} = \frac{rpm_{\max}}{i_{CVT}\, i_{final}\, 3.6} = 2.6[km/h] \tag{10}$$

The complete planetary gearing may have an efficiency of around 98% (η_{tot}=0.98). It is then possible to write the system of equations (11).

$$\left\{\begin{aligned} & P_{diss} = C_1\omega_1\left(\eta_{tot} - 1\right) \\ & P_1 = C_1\omega_1 \\ & P_3 = -C_3\omega_3 \\ & P_p = C_p\omega_p \\ & P_1 + P_3 + P_p + P_{dis} = 0 \\ & C_1 + C_3 + C_p = 0 \end{aligned}\right\} \tag{11}$$

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 three are the power, torque speed relations of the three shafts. Shaft p goes to the wheels through the differential and the final reduction. Shaft 3 is the annular gearing one and it is connected to the generator G through an ordinary gearing. Shaft 1 comes from the engine. The unknown are the output torque C_p, the annular gearing torque C_3, the power output of the generator P_3, the output power to rear wheel P_p and the dissipated energy P_{diss}. At the minimum speed V_{min} with the engine at the maximum rpm, it is possible to calculate ω_p=10.56 [rad/s] (12) and ω_3=0.96 [rad/s] (13).

$$\omega_p = n_{\min} i_{final} \frac{\pi}{30} = 10.56[rad/s] \tag{12}$$

$$\omega_3 = \omega_p - \frac{rpm_{\max}}{i_{CVT}} \frac{\pi}{30} = 0.96\,[rad/s] \tag{13}$$

In this condition the results of the system of equations (11) are the following: C_p=-37.6 [kNm], C_3=36.8 [kNm], P_3=35 [kW], P_p=397 [kW] and P_{diss}=8.8 [kW]. The front sprocket (S4) can then be continuously powered with P_3=35 kW (see Figure-3) that is approximately 9% of the power available at the rear sprocket (S1) (see Figure-3).

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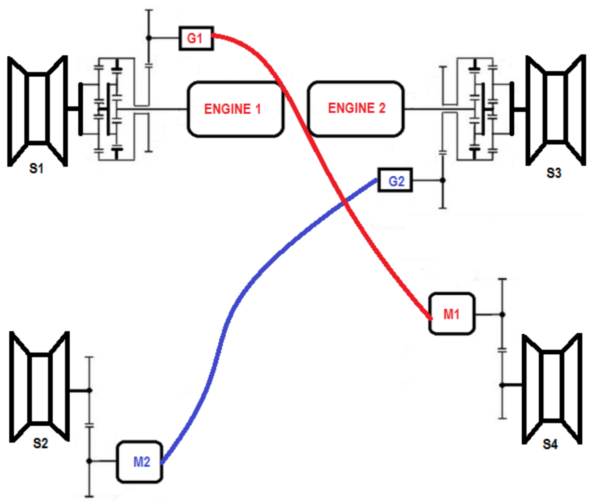

Figure-3. Proposed CVT hybrid system (S1-S2 left track, S3-S4 right track).

From Figure-3 the system can be explained. A large reduction ratio from the double stage epicyclical gearbox is obtained by keeping the annular gears at very low speed through the opposing torque of the generator G. G produces electrical energy that can be recirculated to the front wheel sprocket through the electric motor M or/and can charge the batteries. As speed is increased the opposing torque of the generator G is progressively reduced. In this way, the rotational speed of the annular gears is augmented. The reduction ratio of the double epicyclical gearing is reduced accordingly. Therefore, the amount of energy available from G is progressively reduced. This process ends at the top speed where the annular gear runs at the same speed of the input (engine) and no energy is available from G. An emergency “all mechanical mode” is possible by installing a brake on the generator. In this case the maximum reduction ratio is obtained. A “silent”, full electric, mode can be obtained by powering the electric motor with the battery energy. Since, during the larger part of the life of the engine, the vehicle runs at speeds below the maximum one, the original generator installed on the engine may be replaced by the one installed on this transmission.

Disclaimer

An example is proposed to explain the CVT solution. This is the best way to introduce the concept. The data are taken from existing heavy duty vehicles. However the example suffers from many shortcomings. For example the torque converter is not included in the model. The mass of the vehicle and the many data needed for the design of a transmission and steering device 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 a "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 fact is particularly common in heavy duty tracked vehicles, with masses that often run out of designer control.

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.

CONCLUSIONS

This paper introduces a hybrid CVT steering device concept for a tracked vehicle. This concept is combined with dual drive steering and electronic directional stability software. In this way smaller, more efficient automotive derived engines can be used. This system uses a planetary gearbox. The high torque, double stage, high density gearbox has the annular gear coupled to an electric generator, that controls annular-gear-speed. In this way the reduction ratio can be varied continuously from slow to full speed. When the annular gear is moving at low rotational speed the reduction ratio is high. When the annular gear rotates as the same speed of the input shaft a direct drive (reduction ratio=1) to the final reduction is obtained. The electric energy from the generator can be used to energize the other sprocket. In this way the track tension can be reduced. An example was calculated for a heavy vehicle and one of the possible solutions is depicted in Figure-3. In this way high efficiency is also obtained.

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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)	-	
$i_{x,y}$	Transmission ratio from gear "x" (input) to gear "y" (output)	-	
i_{final}	Differential transmission ratio	-	4
ω_p	Carrier speed	rpm	
ω_1	Sun gear speed	rpm	
ω_3	Annular gear speed	rpm	
$\omega^0{}_x$	Rotational speed of the gear "x" of the equivalent ordinary gearing	rpm	
$\tau^0{}_{xy}$	Transmission ratio from gear "x" (input) to gear "y" (output) or the relative ordinary gearing	-	

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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		