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AN HYBRID, EXTREMELY SIMPLE, STEERING SOLUTION FOR TRACTORS WITH CONTINUOUSLY VARIABLE TRANSMISSION RATIO

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ABSTRACT

The implementation of an electrically controlled dual drive inherently stable heavy duty tracked vehicle is introduced in this paper. The proposed solution makes it possible to obtain a CVT (Continuously Variable Transmission) by using two single stage planetary gearings for each sprocket. An additional motor is added to the annular gear. By controlling the motor speed, it is possible to vary the transmission ratio continuously and to obtain also continuous reverse speed. The torque of the electric motor and the engine are added. It is then possible to obtain extremely high torque at very low speed. This track steering mechanism is simple, continuous, efficient and additive; this means that, when transitioning to a turn, the mean track velocity remains the same. It is also additive, since keeping both tracks moving helps prevent the tracked vehicle from getting stuck. Driving in a straight line is easy. Neutral steering can be achieved. Finally, the energy from the slowed track can be transferred to the sped up one in a regenerative way. An example is proposed in this paper to clarify the concept.

Keywords: Continuously Variable Transmission (CVT), steering, tractors.

INTRODUCTION

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 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 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.

Tracked vehicles steering solutions

Steering a tracked vehicle is very different from steering a wheeled vehicle. In a typical four wheeled car, the front two wheels can be pivoted to point the vehicle. The power is delivered to the traction wheels via differential gear (s). This type of transmission allows the radial velocity difference between the wheels to be accounted for, and the wheeled vehicle can theoretically turn without any slippage or skidding. With a few exceptions, like the universal carrier and tetrarch, tracked vehicles have to rely on skid steering. If one track is moving faster than the other, the forward and rearward sections of the tracks slip laterally over the ground and the vehicle turns. The ideal track steering mechanism is simple, continuous, efficient and additive; this means that, when transitioning to a turn, the mean track velocity remains the same. Tests have shown that additive steering works better when the vehicle is in deep mud or sand, since keeping both tracks moving helps prevent the tracked vehicle from getting stuck. In addition the ideal tracked vehicle steering mechanism should make driving in a straight line easy. Neutral steering should also be allowed, that is, that the tracked vehicle can turn within its own length. Additionally, the energy from the slowed track should be transferred to the sped up one in a regenerative way.

Traditional track steering

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

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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. The first vehicle with a double differential with hydrostatic steering drive was the Char B (1935).

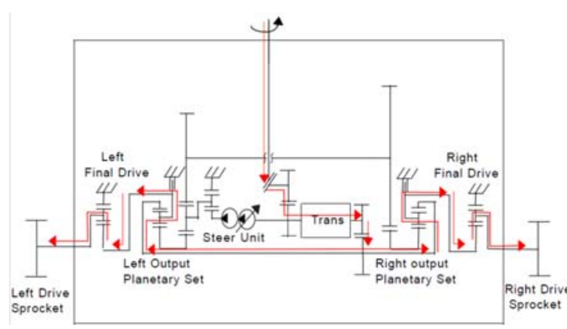

Figure-1. T Double differential with Hydrostatic Steering Drive [29].

The power split system

The split device consists of a planetary gearbox (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

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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.

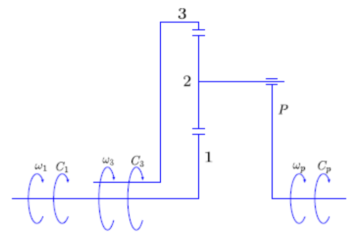

Figure-2. Planetary gearing.

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. 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	3,000	rpm
rpm_{torque}=rpm_{min}	Maximum torque engine speed	900	rpm
rpm_{min}	Min engine speed	900	rpm
P	Maximum engine power	1,103	kW
T	Maximum engine Torque	3,754	Nm
i_{final}	Final reduction gear ratio	1.68	-

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

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

From the Willis' equation:

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

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

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

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

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

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

The heavy vehicle 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).

$$n_{min} = \frac{60 \times V_{min}}{\pi D} = 3.22[rpm] \qquad (6)$$

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$$n_{\max} = \frac{60 \times V_{\max}}{\pi D} = 166.32[rpm] \quad (7)$$

In order to achieve V_{min} and V_{max} at rpm_{max}, the following gear ratios should be obtained i_{min}=929.69 and i_{max}=18.32. Since the differential gearing has a reduction ratio i_{final}=1.68, the CVT should achieve i_{minCVT}= i_{minCVT}= i_{min}=175.76 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 i_{12}=0.7 and i_{23}=3.42 (8) (9).

$$\left\{\begin{array}{l} i_{\min}(\omega_3 = 0) = \dfrac{z_1 + z_3}{z_1} \\ i_{12} = \dfrac{z_2}{z_1} \\ i_{23} = \dfrac{z_3}{z_2} \end{array}\right\} \Rightarrow i_{\min}(\omega_3 = 0) = 1 + i_{12} i_{23} \quad (8)$$

$$\left\{\begin{array}{l} m_{n1} = m_{n2} = m_{n3} \\ r_1 + 2 \times r_2 = r_3 \end{array}\right\} \Rightarrow z_1 + 2 \times z_2 = z_3 \Rightarrow i_{12} = \frac{1}{i_{23} - 2} \quad (9)$$

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

$$z_{\min} = 2k \frac{1 + \sqrt{1 - (\frac{2}{i} - \left(\frac{1}{i}\right)^2) \sin^2(\theta)}}{(2 - \frac{1}{i}) \sin^2(\theta)} \quad (10)$$

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 is then possible to calculate i with $\omega 3 \neq 0$ (11). This can be obtained from equation (5) (Figure-3)

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

With i_{min}=175.76 and the engine at rpm_{max}, it is possible to calculate ω_3= $\omega_{3,lim}$=-127.8 [rad/s] = -1221 rpm. It is also useful to calculate the angular velocity of the planetary gearing at the maximum torque (rpm_{Tmax}=900 rpm), with i_{min}. The result is ω_3= $\omega_{3limTmax}$=-38 [rad/s] = -366 rpm.

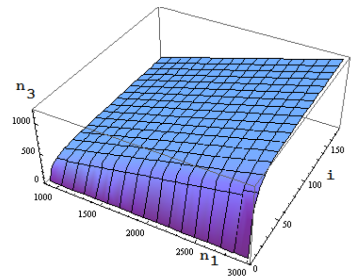

Figure-3. n_3 (annular gear speed) with gear ratio i and engine rpm (n_1) (11).

As it can be seen n_3 has the maximum value at {i_{min},rpm_{max}}→ n_{3max}=1,201 rpm. The minimum value is at {i_{max},rpm_{min}}→ n_{3max}=0 rpm. The complete planetary gearing may have an efficiency of around 97% (η_{tot}=0.97). It is then possible to write the system of equations (12).

$$\left\{\begin{array}{l} P_{diss} = C_1 \omega_1 (\eta_{tot} - 1) \\ P_1 = C_1 \omega_1 \\ P_3 = C_3 \omega_3 \\ P_p = C_p \omega_p \\ \omega_p = \dfrac{1}{i} \omega_p \\ P_1 + P_3 + P_p + P_{diss} = 0 \\ C_1 + C_3 + C_p = 0 \end{array}\right\} \quad (12)$$

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), $C_p(n_1,n_3)$(14) and the total output torque C_{total} (15).

$$C_3 = -C_1 \omega_1 \frac{1 - \eta_{tot} i}{\omega_1 - \omega_3 i} \quad (13)$$

$$C_p = C_1 \times i \times \frac{\eta_{tot} \omega_1 - \omega_3}{\omega_1 - \omega_3 \times i} \quad (14)$$

$$C_{total} = |C_3| + |C_p| = \left| -C_1 \omega_1 \frac{1 - \eta_{tot} i}{\omega_1 - \omega_3 i} \right| + \left| C_1 \times i \times \frac{\eta_{tot} \omega_1 - \omega_3}{\omega_1 - \omega_3 \times i} \right| \quad (15)$$

It is then possible to calculate C_3 and C_p at maximum power and at maximum torque. The heavy duty internal engine has a torque that follows the linear equation (16)

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$$C_{ice} = C_1 + \frac{rpm - rpm_{torque}}{rpm_{max} - rpm_{torque}} \left(C_1 - C_{max} \right) \tag{16}$$

From equation (13), it is then possible to choose the electric motor (or the internal combustion engine) that acts on the annular gearing 3

At maximum power (1,103kW@3,000rpm) we will have at $i=i_{max} \rightarrow$ C_3=8,233 Nm, n_3=0 and C_3=8,336 Nm, n_3=-1201.31 rpm at $i=i_{min}$ (14). At maximum torque (3, 754Nm@900rpm) we will have at $i=i_{max} \rightarrow$ C_3=8,685 Nm, n_3=0 and C_3=8,794 Nm, n_3=-360 rpm at $i=i_{min}$ (14). The second engine or motor should then have a stall torque C3stall=8, 685 Nm a max torque C3max=8, 794 Nm at 360 rpm and a max power of P_{mmax}=8, 366*1, 201.31*π/30,000=1,048 kW. These value are easily obtainable with an electric motor, but not with an internal combustion engine. In this case the speed of the annular gear should never arrive at zero.

The torque available at the final reductions will be the sum of the torque of the two engines or of the electric motor+engine (15) (Figure-4).

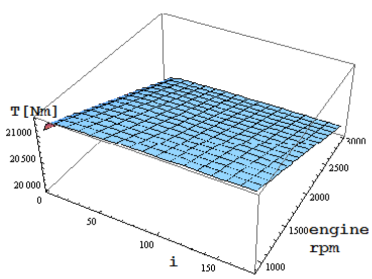

Figure-4. Torque available at the final reduction.

As it can be seen the torque is extremely effective, since its maximum is available at low rpm, were it is more needed.

The new steering mechanism

The new steering mechanism is depicted in Figure-5. In this solution we have 2 internal combustion engines: engines 1 power directly the sprockets, while engine 2 generates the electric power for the two motors M1 and M2. By varying the speeds of M1 and M2 it is possible to vary continuously the speed ratio of each sprocket. It is also possible to reverse the speed always by varying the relative speed of engine 1 and motors 1 and 2.

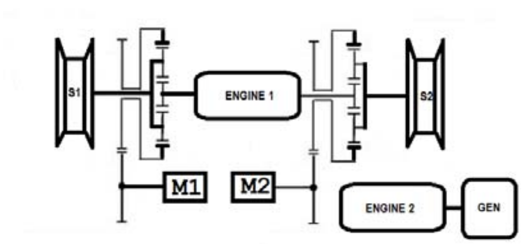

Figure-5. a possible solution for the proposed CVT.

In this case the steering system is obtained by different speed ratio of the left (S1) and right (S2) sprockets. These speed ratios are obtained with different rotation speeds of the motor M1 and M2. In the example of Figure-6 a single generator G is used. Several other configurations are possible. Note the extreme simplicity of the transmission of Figure-5, when compared with the one of Figure-1. The final reductions and the torque converter(s) are not included in Figure-5. This later is not considered in the calculations. The reverse system should be dimensioned from a dynamic model of the vehicle. Additional reverser (s) can be added for more flexibility and performance.

This track steering mechanism is simple, continuous, efficient and additive; this means that, when transitioning to a turn, the mean track velocity remains the same. It is also additive, since keeping both tracks moving helps prevent the tracked vehicle from getting stuck. Driving in a straight line is easy. Neutral steering can be achieved. Finally, the energy from the slowed track can be transferred to the sped up one in a regenerative way.

Several other solutions based on the same concept are possible. With accurate design it is also possible to use an internal combustion engine in place of the electric motor.

CONCLUSIONS

The implementation of an electrically controlled dual drive inherently stable heavy duty tracked vehicle is introduced in this paper. This vehicle has to use a computerized control system to achieve the directional stability. This solution makes it possible to obtain a CVT by using two single stage planetary gearings for each sprocket. An additional motor is added to the annular gear. By controlling the motor speed it is possible to vary the transmission ratio continuously and to obtain also continuous reverse speed. The torque of the electric motor and the engine are added. It is then possible to obtain extremely high torque at very low speed. This track steering mechanism is simple, continuous, efficient and additive; this means that, when transitioning to a turn, the mean track velocity remains the same. It is also additive, since keeping both tracks moving helps prevent the tracked vehicle from getting stuck. Driving in a straight line is easy. Neutral steering can be achieved. Finally, the energy from the slowed track can be transferred to the sped up one in a regenerative way. An example is proposed in this paper to clarify the concept.

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