

**HYDRO-MECHANICAL CONTINUOUSLY VARIABLE TRANSMISSION****FIELD**

5        This invention relates to a drive system useful as a vehicle propulsion system or stationary equipment drive, combining mechanical and hydraulic power systems.

**BACKGROUND**

10        Hydraulic drive systems are commonly used for large vehicles or stationary equipment. However, as the output speed increases at a given gear setting, the efficiency of the hydraulic drive is correspondingly reduced. This makes  
15 it inefficient to run hydraulic drives at the upper half of the gear setting. This problem may be overcome by having multiple gear settings, but the complexity of the resulting transmission negates the benefits of using a hydraulic drive.

20        An alternative to a hydraulic drive system is a mechanically driven system. However, conventional mechanical drive systems are limited to discrete gear ratios, which do not allow for infinite speed ratios as  
25 found in hydraulic drives. A great deal of power management between the engine and the transmission at all output speeds is necessary for transmission effectiveness. A purely mechanical drive is inadequate to ensure the efficient use of the engine's available power due to the discrete speed  
30 ratios, while a purely hydraulic drive has inherently poor efficiency at higher operational speeds.

With the increasing costs of fuel and more stringent emissions requirements, there is a need for more efficient drive systems for large and small vehicles, as well as  
5 stationary equipment, to replace traditional hydraulic and mechanical drive systems.

It is an object of this invention to provide a more efficient drive system for large and small vehicles and  
10 stationary equipment by combining hydraulic and mechanical power systems.

It is a further object of this invention to provide a transmission system for optimizing use of combined drive  
15 systems.

It is a still further object of this invention to provide a combined drive system with a dual or multiple speed, shift-on-the-fly gearbox for extended speed and  
20 torque ranges.

It is a still further object of this invention to provide an improved steering system for combined drive systems when applied to differential output speed  
25 requirements.

#### **SUMMARY**

The invention comprises a hydro-mechanical continuously  
30 variable transmission (HMCVT) that uses a planetary gear

system to provide a combination of hydraulic and mechanical power for a vehicle or stationary equipment.

The HMCVT has an input planetary gear set with at least a ring gear, a sun gear and a carrier gear. A main power  
5 input is coupled to one of the ring gear, sun gear and carrier gear. A combiner gear is coupled to a main output shaft. An hydraulic branch which has an hydraulic pump and motor, receives an input from a second one of the ring gear, sun gear and carrier gear and provides an hydraulic output.  
10 A mechanical branch receives an input from a third one of the ring gear, sun gear and carrier gear and provides a mechanical output to the main shaft. The combiner gear is operative to combine the hydraulic output with the mechanical output at said main output shaft such that said  
15 transmission decreases a percentage of transmission power arising from said hydraulic branch as said main output shaft increases in speed. An output planetary gear set coupled to said main output shaft and to a power take off shaft. A low speed shifting planetary clutch has an input coupled to the  
20 output planetary gear set. The low speed shifting planetary clutch is configured to create a reduction in a gear ratio of the transmission.  
A high speed shifting planetary clutch is releasably coupled to the main output shaft and, when applied thereto, provides  
25 a predetermined gear ratio between the main output shaft and the power take off shaft. A microprocessor is coupled to the transmission and is operative to sense a shift in

transmission gear ratio and adjust the hydraulic output speed so that, after combining with the mechanical output, a speed of the main output shaft after shifting matches a speed of the power take off shaft before shifting.

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Preferably, the predetermined gear ratio is 1:1.

Advantageously, the microprocessor is coupled to the hydraulic pump and motor, the high and low speed shifting planetary clutches, to an input to the hydraulic branch and to an output of the hydraulic branch.

The HMCVT may also include a 2-speed planetary clutch system to expand the operating parameters of the vehicle or stationary equipment.

The HMCVT may further include a planetary steering system that works with or without the 2-speed planetary clutch system.

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The HMCVT may also include a launch assist device to limit torque applied to the drive pump when the ratio of hydraulic pump displacement to hydraulic motor displacement is small.

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The HMCVT may additionally include a lockup brake coupled to the hydraulic branch input, operative to lock out the hydraulic branch and force all power through the mechanical branch when the transmission output is operating

at a pre-selected percentage of its maximum speed. The lockup brake may be combined with the launch assist device into a single device.

5       The HMCVT may further include an anti-recirculating reverser device operative to allow the transmission output to operate in a reverse direction of motion without developing a recirculating power flow through the mechanical branch.

10       The 2-speed planetary clutch, planetary steering system, launch assist device, lockup brake and anti-recirculating reverser device may be used individually or in any combination of two or more in any given HMCVT.

15       **BRIEF DESCRIPTION OF THE DRAWINGS**

      The invention itself, both as to organization and method of operation, as well as additional objects and  
20   advantages thereof, will become readily apparent from the following detailed description when read in connection with the accompanying drawings:

**Figure 1** shows a planetary gear set with multiple  
25   planetary gears;

**Figure 2** shows a block diagram of an HMCVT in a RSC configuration with all optional components;

30   **Figure 3** shows a block diagram of an HMCVT in a RSC configuration;

**Figure 4** shows a block diagram of an HMCVT in a SCR configuration;

5     **Figure 5** shows a block diagram of an HMCVT in a SRC configuration;

**Figure 6** shows a block diagram of an HMCVT in a RSC configuration with a 2-speed clutch;

10    **Figure 7** shows a block diagram of an HMCVT in a RSC configuration with a planetary steering system;

15    **Figure 8** shows a block diagram of an HMCVT in a RSC configuration with a launch assist device;

**Figure 9** shows a block diagram of an HMCVT in a RSC configuration with a lockup brake;

20    **Figure 10** shows a block diagram of an HMCVT in a RSC configuration with a spur gear reverser;

**Figure 11** shows a block diagram of an HMCVT in a RSC configuration with a bevel gear reverser;

25    **Figure 12** shows a block diagram of an HMCVT in a RSC configuration with a mechanical disconnect reverser;

#### 30     **DETAILED DESCRIPTION**

The hydro-mechanical continuously variable transmission (HMCVT) is designed to split input power between a hydraulic drive branch, using a hydraulic pump and motor, and a

parallel mechanical drive branch, using shafts and/or gears, recombining the power from each branch into a single output.

The HMCVT is based on a planetary gear set **10** as shown in **Figure 1**. A planetary gear set **10** consists of four parts: a carrier gear **12**, a number of planetary gears **14**, a ring gear **16** and a sun gear **18**. The ring gear **16** and the sun gear **18** are connected through the planetary gears **14**. The planetary gears **14** are also connected to the carrier gear **12**. In **Figure 1**, three planetary gears **14a-c** are used, more may be used if necessary.

The planetary gear set **10** is connected to the hydraulic drive pump **22**, the main shaft **26** and the transmission input **40**. A 3 letter code (R=ring gear; S=sun gear; C=carrier gear) has been adopted for purposes of this discussion to describe how the planetary gear set **10** is connected within the transmission: 1<sup>st</sup> letter designates which part of planetary gear set **10** is connected to the hydraulic drive pump **22**; 2<sup>nd</sup> letter designates which part of planetary gear set **10** is connected to the main shaft **26**; the last letter designates which part of planetary gear set **10** is connected to the input **40** from the engine.

A full HMCVT system in a RSC configuration with all optional components connected is shown in **Figure 2**. In the RSC configuration, a hydraulic drive pump **22** is connected to the ring gear **16** of the planetary gear set **10** and the sun

gear 18 is connected to the combiner gear 20 via the main shaft 26. The combiner gear 20 is also connected to a hydraulic drive motor 24. The input 40 from the main engine (not shown) to the HMCVT is received by the carrier gear 12.

5 A detailed view of the RSC configuration is shown in **Figure 3.**

A full HMCVT system in a SCR configuration is shown in **Figure 4.** In the SCR configuration, a hydraulic drive pump 10 22 is connected to the sun gear 18 of the planetary gear set 10 and the carrier gear 12 is connected to the combiner gear 20 via the main shaft 26. The combiner gear 20 is also connected to a hydraulic drive motor 24. The input 40 to the HMCVT is received by the ring gear 16.

15 A full HMCVT system in a SRC configuration is shown in **Figure 5.** In the SRC configuration, a hydraulic drive pump 22 is connected to the sun gear 18 of the planetary gear set 10 and the ring gear 16 is connected to the combiner gear 20 20 via the main shaft 26. The combiner gear 20 is also connected to a hydraulic drive motor 24. The input 40 to the HMCVT is received by the carrier gear 12.

In theory, the carrier gear 12, ring gear 16 and sun 25 gear 18 may be connected to the input 40, drive pump 22 and main shaft 26 in any combination. However, the above three configurations have tested as the most practical for application as transmissions for large vehicles.

Mathematically, it can be shown that in the HMCVT, the power is split such that the power from the hydraulic system (including the drive pump **22** and the drive motor **24**) combines with the mechanical system (including the main shaft **26**) to equal 100% of the total power less efficiency losses. It can further be shown that the percentage of mechanical power increases as the output speed increases, with a corresponding decrease in hydraulic power. The result is a more efficient use of the input energy **40** than in a strictly hydraulic or strictly mechanical transmission.

It can also be shown that the torque ratio between the ring gear **16** and the sun gear **18** is only dependent on the gear ratio between the ring gear **16** and the sun gear **18**. This means that the final gear ratio of the HMCVT can be set by the choice of ring gear **16** and sun gear **18**.

To prove: define the following terms: h-hydraulic, m-mechanical, i-input, specific speed ( $O_x$ ) is ratio of  $x$  ( $x=h,m,i$ ) gear speed to input (i) gear speed.

Define a constant  $R$  as the speed of the m-gear when the h-gear is not turning:  $R=O_m \mid O_h=0$ . Then define  $O_m=RS$ , where  $S$  reflects the actual speed of the output (as a value from 0 to 1).  $R$  and  $S$  are used to make the equations independent of the actual configuration of the planetary gear set **10**.

Since  $O_m$  is linear in  $S$ ,  $O_h$  must also be linear with  $S$ , as a function of  $(1-S)$ , since  $O_h=0$  when  $S=1$ . At  $S=1/R$ ,  $O_m=1$ . This means that at  $S=1/R$  the i-gear and m-gear are turning at the same speed. Considering the planetary gear model in **Figure 1**, this means that the ring gear **16** and sun gear **18** are turning at the same speed. For this to occur, the planetary gears **14** must not be turning, meaning that the carrier gear **12** is also turning at the same speed as the ring gear **16** and sun gear **18**.

More generally, when any two of the gears of the planetary gear set **10** are moving at the same speed, so is the third gear. Using this result, we then get  $O_h=(R/R-1)(1-S)$ .

The power split then becomes  $P_h=1-S$  and  $P_m=S$ . This also means that two forms of power recirculation can occur: "overdrive" when  $S>1$  and "reverse" when  $S<0$ .

In the physical HMCVT, the combiner gear **20** and planetary gear set **10** are responsible for controlling the distribution of power between the hydraulic and mechanical systems. As the output speed changes, the power split between the hydraulic and mechanical systems is changed as described above. When the output is motionless (speed=0), main shaft **26** is also motionless (0 rpm). As the output moves, the drive pump **22** must pump fluid and, initially, all the power is derived from the drive pump **22** and drive motor

24. As the output speed increases, the main shaft 26 and the connected drive motor 24 (through the combiner gear 20) must turn faster. As a result, the drive pump gear (in the RSC configuration, ring gear 16) turns slower, due to the effect of the planetary gear system 10 and the need to maintain a constant torque ratio.

A considerable advantage of the HMCVT lies in the unique ability of the configured systems as shown in **Figures 2-5** to enable a driven output on both ends of the transmission via a common shaft 26. This is particularly useful in vehicles or stationary equipment that require duplicated output shafts to two drives such as tracks and/or differentials.

Furthermore, one or both of the outputs can be engaged or disengaged eliminating the need for a transfer case when configured for multiple output drives.

The HMCVT speed can be controlled in any conventional manner, however an electronic control system is preferred to best optimize the power splitting in connection with the output speed. Furthermore, the electronic control system can also include control means for the two-speed transmission system, planetary steering system, launch assist device, lockup brake and anti-recirculatory reverser discussed below.

### Two-speed transmission

An additional modification to optimize the use of the HMCVT is a two-speed planetary clutch system as shown in

5 **Figure 6.** The output **26** is connected to the input of an additional 2-speed shifting planetary **30**. Power enters the shifting planetary through the shifting planetary sun gear **31** and exits through the shifting planetary carrier gear **35**. In low speed operation, the shifting planetary ring gear **33**  
 10 is held fixed by a low-speed clutch or brake **32**, creating a reduction in the gear ratio. Shifting to high gear for high-speed operation is accomplished by releasing the brake **32** and applying a high-speed clutch **34** to effectively give a 1:1 gear ratio. Microprocessor **41** is coupled to the pump **22**  
 15 and motor **24**, to the input and output gears **43** and **45**, respectively, of the hydraulic branch, to the low speed clutch **32** and the high speed clutch **34** and to the power take off shaft **47**. The microprocessor **41** controls the operation of the shifting as well as other functions.

20 The 2-speed planetary clutch system provides an extended range of available speeds and torques to the operator. The result is an increased operating envelope for the vehicle or stationary equipment.

### 25 Planetary Steering System

Another useful modification for the HMCVT is a planetary steering system as shown in **Figure 7**. As shown,

power is sent from the main input **40** to two steering planetaries **52**, one for each output drive on either side of the transmission. The steering system also includes a closed-loop hydraulic pump **54** that is driven in relation to the current engine RPM. The output of the pump **54** is connected to a hydraulic motor **56** that drives a cross shaft assembly referred to as a zero shaft **58**.

The zero shaft **58** is connected to the sun gear **60** of the left and right steering planetaries **52** and the left and right sun gears **60** are driven by the motor **56** in opposite directions. Therefore, when the zero shaft **58** turns, the speed of the inside drive of the vehicle decreases and the speed of the outside drive increases.

The result of the planetary steering system is a high-precision steering system that provides quick reaction times while maintaining good driving characteristics during straight-ahead motion.

#### Launch Assist Device (LAD)

One characteristic of the HMCVT is that at low output speeds, the pump **22** is set to a very low displacement and the motor **24** is set to a high displacement. In theory, this could create a very large torque multiplication through the hydraulic branch of the HMCVT. However, in that scenario, the hydraulic pressures generated would exceed those that can be withstood by the system. Therefore, the hydraulic

ratio must be reduced to limit pressure to acceptable levels. Unfortunately, this corrective measure also reduces the output torque at very low speeds.

5        To solve this problem, an energy absorber, called a Launch Assist Device (LAD) **70** is attached to the pump **22** as shown in **Figure 8**. The LAD **70** provides an initial resistance to the gear element of the planetary **10** driving the hydraulic branch. This resistance limits the torque  
10    available to the pump **22** and allows the mechanical branch to reach its full torque output at very low speeds. As a result, the torque is available at the output of the HMCVT.

      The LAD **70** is only required at very low speeds and  
15    should be gradually phased out as the speed increases. As shown, the LAD **70** is a modulated brake assembly. However, other devices, such as a fluid coupling or a torque converter, could be used.

## 20    Lockup Brake

      At the upper limit of the HMCVT operating range, the displacement ratio between the motor **24** and the pump **22**  
decreases to the point where the amount of torque available  
25    to the pump **22** is insufficient to keep it turning. With the speed of the hydraulic pump **22** at zero, all the power is transferred exclusively through the mechanical branch. Unfortunately, most currently available pump and motor

designs include some degree of internal leakage, preventing the HMCVT from reaching a pure 100% mechanical state.

This problem can be solved by using a lockup brake **80** as shown in **Figure 9** to prevent the pump **22** from turning at the end of the operating range. The lockup brake **80** is attached to one of the gears **10** driving the pump **22**. When the upper end of the operating range is reached, the lockup brake **80** is activated, forcing the HMCVT into a pure 100% mechanical mode. The lockup brake **80** can then be deactivated when the HMCVT operating range falls out of the upper regions. The activation/deactivation point for the lockup brake **80** will be determined by the operating conditions and parameters for the vehicle or stationary equipment using the HMCVT.

#### Anti-Recirculatory Reverser

The simplest way to reverse the direction of the final output of the transmission is to reverse the drive motor. When this happens, the power in the mechanical branch is negative and the power in the hydraulic branch is greater than the input power. What is effectively happening is that the drive motor must reverse the direction of the mechanical output of the split speed, feeding power upstream through the mechanical branch. To balance out the power equations, the hydraulic branch must transfer an amount of power equal to the input plus the recirculated power from the mechanical branch.

In order to accommodate the increased power levels, both the hydraulic and the mechanical branches must have increased component strength and/or size, which is not always practical or desirable. Therefore, an additional reverser subsystem that avoids the need to reinforce the hydraulic is desirable.

One potential reverser subsystem **90** as shown in Figures **10** and **11** involves the installation of a reversible gear at the input. The input **40** is connected to a sliding clutch/synchronizer assembly **90**. Depending on which gear is engaged with the clutch/synchronizer **90**, the HMCVT components, including the drive pump **22** and motor **24**, will rotate in either the forward or reverse direction. Either spur gears (**Figure 10**) or bevel gears (**Figure 11**) can be used in the clutch/synchronizer **90**, however, bevel gears allow for a 90-degree change between the engine power output and the main axes of the transmission (the final power output). In either case, if steering planetaries **52** (see **Figure 7**) are present, they are connected to the power output through a preset gear ratio and a constant direction so that the steering will function regardless of the forward/reverse direction of the clutch/synchronizer **90**.

An alternative subsystem for the reverser shown in **Figure 12** disconnects the mechanical branch during reverse hydraulic operation and locks the mechanical output to the

transmission housing (not shown). In this case, the input shaft always rotates in the same direction. The mechanical output may be selectively coupled to either the main shaft 26 or the transmission housing by means of a clutch/synchronizer assembly 112. When the mechanical output is connected to the main shaft 26, forward output speeds result. When the mechanical output is connected to the housing, the mechanical branch is locked out of the transmission. Reverse output speeds can then be achieved by reversing the direction of the drive motor. Power flows exclusively through the hydraulic branch and cannot be recirculated back through the mechanical branch.

Accordingly, while this invention has been described with reference to illustrative embodiments, this description is not intended to be construed in a limiting sense. Various modifications of the illustrative embodiments, as well as other embodiments of the invention, will be apparent to persons skilled in the art upon reference to this description. It is therefore contemplated that the appended claims will cover any such modifications or embodiments as fall within the scope of the invention.

## WE CLAIM:

1. A hydro-mechanical continuously variable transmission for a vehicle or stationary equipment, comprising:

- 5 a) a input planetary gear set having at least a ring gear, a sun gear and a carrier gear;
- b) a main power input coupled to one of said ring gear, said sun gear and said carrier gear;
- c) a combiner gear coupled to a main output shaft;
- 10 d) a hydraulic branch having a hydraulic pump and motor, said hydraulic branch receiving an input from a second one of said ring gear, said sun gear and said carrier gear and providing an hydraulic output;
- 15 e) a mechanical branch receiving an input from a third one of said ring gear, said sun gear and said carrier gear and providing a mechanical output to said main shaft and wherein said combiner gear is operative to combine said hydraulic output with said
- 20 mechanical output at said main output shaft such that said transmission decreases a percentage of transmission power arising from said hydraulic branch as said main output shaft increases in speed;
- f) an output planetary gear set coupled to said main
- 25 output shaft and to a power take off shaft;
- g) a low speed shifting planetary clutch having an input coupled to said output planetary gear set,

said low speed shifting planetary clutch being configured to create a reduction in a gear ratio of said transmission;

h) a high speed shifting planetary clutch releasably coupled to said main output shaft and, when applied thereto, provides a predetermined gear ratio between the main output shaft and the power take off shaft; and

(i) a microprocessor coupled to said transmission and operative to sense a shift in transmission gear ratio and adjust hydraulic output speed so that, after combining with said mechanical output, a speed of said main output shaft after shifting matches a speed of said power take off shaft before shifting.

2. A transmission according to claim 1, wherein said predetermined gear ratio is 1:1.

3. A transmission according to claim 1, wherein said microprocessor is coupled to said hydraulic pump and motor, to said high and low speed shifting planetary clutches, to an input to said hydraulic branch and to an output of said hydraulic branch.

4. A transmission according to claim 1, wherein said main output shaft is coupled to said sun gear of said output planetary gear set, a ring gear of said output planetary

gear set is coupled to said low speed shifting planetary clutch, and a carrier of said output planetary gear set is coupled to said power take off shaft.

5 5. A transmission according to claim 1, including a launch assist device to limit torque applied to said hydraulic branch at low output speed, comprising an energy absorption device connected to said hydraulic branch input to provide resistance to a gear driving said hydraulic branch.

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6. A transmission according to claim 5, wherein said energy absorption device is controlled by said microprocessor, which causes said energy absorption device to limit torque applied to said hydraulic branch at very low  
15 speeds.

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7. A transmission according to claim 1, including a lockup brake coupled to said hydraulic branch input, operative to lock out said hydraulic branch and force all power through  
20 said mechanical branch when said main output shaft is operating at a pre-selected percentage of its maximum speed.

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8. A transmission according to claim 1, including an anti-recirculating reverser device operative to allow said main  
25 output shaft to operate in a reverse direction of motion without developing a recirculating power flow upstream through said mechanical branch.

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9. A transmission according to claim 1, including a planetary steering system comprising:

- a) a hydraulic pump;
- b) a hydraulic motor;
- c) a zero shaft assembly with one input coupled to said hydraulic motor and two outputs geared to rotate at the same speed but in opposite directions; and
- d) two planetary gear sets with identical gear ratios aligned coaxially with said main shaft, an input for each planetary gear set coupled to one of said zero shaft outputs, a second input for each planetary gear set coupled to said main shaft and an output for each of said planetary gear sets directed to opposite sides of said vehicle or stationary equipment, said hydraulic pump and hydraulic motor operative to control the amount of power available for said zero shaft.

10. A transmission according to claim 8, wherein wherein said anti-recirculatory reverse comprises a clutch/synchronizer assembly coupled to said mechanical power output, said clutch assembly operative to selectively engage said main shaft for forward output operation, a housing of said transmission for reverse output operation,

and nothing for neutral output operation, such that in reverse output operation said mechanical output is prevented from rotating and power flows solely through said hydraulic branch.

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11. The transmission according to claim 1, wherein said anti-recirculatory device comprises a clutch/synchronizer coupled to said main power input, said clutch/synchronizer having two gears located coaxially on either side of said  
10 clutch/synchronizer, each of said gears driving a gear train of equal and opposite gear multiplication, said gear trains coupled to one of said planetary gears and forward, neutral and reverse rotation of said main output shaft determined by said clutch/synchronizer engaging one of said two gears, the  
15 other of said two gears and neither of said two gears, respectively.

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