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Combined hydraulic power vehicle transmission modes

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Abstract. Achieving high efficiency is a key objective of optimal vehicle drive train design. Hydraulic power transmissions with hydraulic torque converters are commonly used in wheeled and tracked self-propelled vehicles. Hydraulic torque converters are self-adjusting, and have a relatively low efficiency. The study investigates design options for self-propelled vehicle combined hydraulic power transmission. In order to improve their average efficiency, the engine power is transferred in two combined modes: single- and double-flow. We propose hydraulic power transmission layouts where the hydraulic torque converter operates with regular or summing differentials. Screw and free wheel clutches are used as drive train controls. They enable automated switchover of the torque transmissions modes. The proposed drive trains with switchable hydraulic torque converters are designed for both road and off-road self-propelled vehicles. They can be combined with robotic fixed-ratio gearboxes or as a part of a vehicle drive train consisting of a conventional clutch, and a manual gearbox.

1. Introduction

Hydraulic power transmissions (HPT) are commonly used in wheeled and tracked self-propelled vehicles. HPTs consist of a hydraulic power transmission and a fixed-ratio gearbox [1]. The reasons for extensive use of hydraulic power transmissions with hydraulic torque converters in vehicle/tractor drive trains are as follows:

– automatic ratio adjustment controlled by the drive train load as the engine torque is continuously transferred
– damping drive train torsional vibration that extends service life and ensures smoother operation of the engine and drive train components
– high capacity with relatively compact size and low weight compared to mechanical transmissions
– better drive train controllability and general vehicle handling, specifically in demanding operating conditions [2].

Typically, vehicle/tractor HPTs use a comprehensive hydraulic torque converter with interlocks. An HPT gearbox is a planetary or regular gear with gear shifters: gear clutches, synchromeshes, friction-type clutches, and brakes. An HPT also contains a hydraulic system that supplies fluid to the drive train components, lubricates interfaces, and automatically control the hydraulic torque converter and the gearbox [3].

Hydraulic torque converters are self-adjusting, but their feature low efficiency as compared to mechanical gears. Hydraulic torque converters have the lowest efficiency in a low ratio range. For this reason it is desirable to develop an HPT with a hydraulic torque converter having a high efficiency throughout the entire ratio range.

2. Problem Statement

In an HPT a hydraulic torque converter is installed either in series, or in parallel with the power flow. For the in-series option, the self-propelled vehicle engine power as a single flow is transmitted consequently through the hydraulic torque converter and the gearbox to the output shaft. In this case

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power losses in the drive train are progressively increased reducing the gearbox/drive train efficiency [4].

For the in-parallel option, the engine power flow is split into two flows. One of them passes through the hydraulic torque converter, and the other one passes through the mechanical gear. In double-flow HPTs, power losses are lower, and efficiency is higher as compared to single-flow drive trains. The disadvantages of double-flow HPTs are reduced converter capacity, and poorer torsional vibration damping.

There are two categories of split-flow HPTs: external and internal flow split designs. In drive trains with internal power flow split the engine power is split in a special (multi-turbine, or multi-stage) hydraulic torque converter. Multi-turbine, or multi-stage hydraulic torque converters have a high conversion ratio, but they are complicated and their application in vehicle drive trains is limited [5].

HPTs with external power flow split are commonly used in self-propelled vehicle drive trains. In such HPTs the engine power flow is split outside of the drive train. A portion of the power is carried over a hydraulic power transmission. and another portion goes through a mechanical gearbox; then the power flows are merged and supplied to the P3E output shaft. In this arrangement a portion of the power goes through a low-efficiency hydraulic torque converter, and another portion is delivered to the output shaft either directly, or through a highly efficient mechanical gearbox. Both power flows are merged at the output shaft. It results in higher overall drive train efficiency.

In double-flow HPTs the engine power is commonly split into two flows or is merges into a single flow with planetary geared differentials. Double-flow HPTs fall into two categories by the mutual arrangement of the differential and the hydraulic torque converter. The first category includes HPTs where the differential is located at the transmission input. In this case, it is a regular differential ("input differential" layout) The second category includes HPTs where the differential is located at the transmission output. In this case, it is a summing differential ("output differential" layout.)

In double-flow HPTs as the max torque ratio is reduced, its efficiency, capacity, transparency increase, and the hydraulic torque converter effective diameter decreases. Also, double-flow HPTs with differentials can be equipped with standard, commercially available hydraulic torque converters.

One way to improve hydraulic power transmission/hydraulic torque converter efficiency in operation is increasing the average transmission efficiency by combining its single- and double-flow modes. It can be done by introducing controllable components into the original hydraulic power transmission design. The extra components would enable two engine power flow modes. In the low ratio range (0...0.6) of the hydraulic torque converter a single-flow transmission/torque conversion is used; in the high hydraulic torque converter ratio range (0.6...0.85) a double-flow arrangement is used.

Figure 1 shows nondimensional properties of single- and double-flow hydraulic power transmissions vs. the hydraulic torque converter **i** ratio: 1) Transmission efficiency curves **η** for single-flow (curve 1) and double-flow (curve 2) modes; 2) Torque ratio **K** curves for single-flow (curve 3) and double-flow (curve 4) modes.

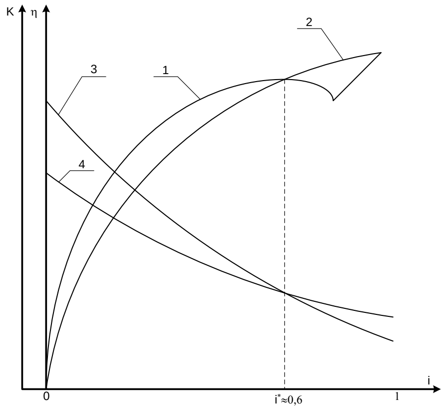

Figure 1. Efficiency (**η**) and torque ratio (**K**) variations in HPTs with single/double-flow modes

The single-flow mode is used for vehicle breakaway and acceleration. The transmission is switched over from the single- to the double-flow mode at the curve intersection (**i*** torque ratio). Besides, the single-flow mode can be utilized for extended vehicle riding in challenging or off-road conditions. While driving in favorable road conditions, the double-flow mode is more efficient, and the transmission operates without switching over to the single-flow mode.

3. Theory

Below we shall consider the original double-flow HPT kinematics used as input data for subsequent study. Dynamic and motion analysis of simple double-flow hydraulic power transmissions with regular and summing differentials are considered in the book [6].

Figure 2 shows a kinematic diagram of a simple double-flow hydraulic power transmission with a regular differential. The HPT consists of a casing (not shown), the input shaft Pos. 1, and the hydraulic torque converter Pos. 2 comprising of the impeller Pos. 3, the runner Pos. 4, and the reactor wheel Pos. 5. The reactor wheel is connected to the HPT casing with the free wheel clutch Pos. 6. The HPT output shaft Pos. 7 is connected to the runner. The input shaft is connected to the carrier Pos. 9 of the differential Pos. 8. The satellite gears Pos. 11 are attached to the carrier. The differential's central (sun) gear Pos. 10 is also connected to the runner and the HPT output shaft. The differential's crown gear Pos. 12 is connected to the hydraulic torque converter's impeller.

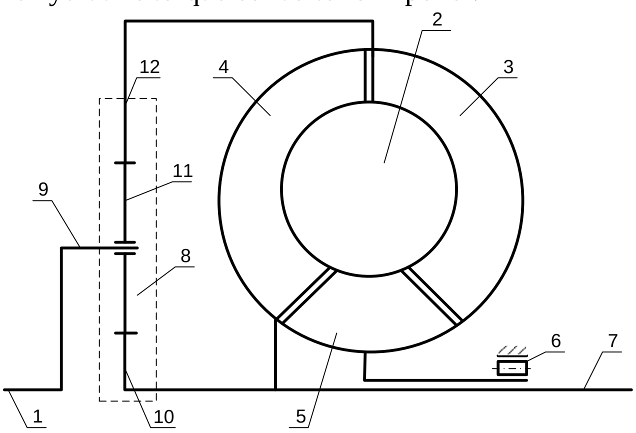

Figure 2. Kinematic diagram of a simple double-flow hydraulic power transmission with a regular differential

The HPT efficiency is higher than that of the original hydraulic torque converter. The transmission also has zero circulating power but features a significantly lower torque ratio. The capacity is increased in the lower engine angular velocity range. The impeller angular velocity exceeds that of the engine. It is controlled by Z_{12}/Z_{10}, where Z_{10} is the teeth number of the central gear, Z_{12} is the teeth number of the planetary ring gear. This may negatively affect the converter performance. A double-flow transmission is also less efficient at torsional vibration damping.

It is possible to arrange a simple double-flow HPT with a regular differential where the central gear is connected to the impeller, while the ring gear is connected to the runner. Such a drive train's properties are similar to that of the drive train shown in Figure 2, but due to even higher impeller angular velocity there are more chances for fluid cavitation in the hydraulic torque converter.

Figure 3 shows a kinematic diagram of a simple double-flow hydraulic power transmission with a summing planetary differential.

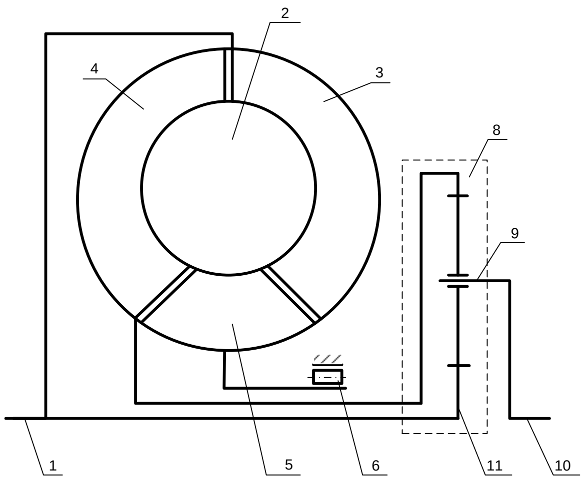

Figure 3. Kinematic diagram of a simple double-flow hydraulic power transmission with a summing differential

This HPT also consists of a casing (not shown), the input shaft Pos. 1, the three-wheel hydraulic torque converter assembly Pos. 2 comprising of the impeller Pos. 3, the runner Pos. 4, and the reactor wheel Pos. 5. The reactor wheel is connected to the HPT casing with the free wheel clutch Pos. 6. The HPT output shaft Pos. 7 is connected to the runner. The output shaft Pos. 10 is connected to the carrier Pos. 9 of the regular differential Pos. 8. The satellite gears Pos. 11 are attached to the carrier. The differential's central (sun) gear Pos. 11 is also connected to the runner and the HPT output shaft. The differential's crown gear Pos. 12 is connected to the hydraulic torque converter’s runner.

The HPT efficiency is higher than that of the original hydraulic torque converter. The transmission also has zero circulating power but features a significantly lower torque ratio. The transmission’s capacity and transparency are higher. In a certain torque ratio range the hydraulic torque converter operates as a hydraulic brake, i.e. the runner rotates backwards. A double-flow transmission is also less efficient at torsional vibration damping. As compared to double-flow HPTs with summing differentials the impeller angular velocity in this arrangement is always equal to the engine angular velocity.

It is possible to arrange a simple double-flow HPT with a summing differential where the central gear is connected to the input shaft and the impeller, while the ring gear is connected to the runner. The key properties of such a transmission are identical to that of the transmission shown in Figure 3.

4. Results of engineering

We will consider HPT/hydraulic torque converter arrangements that combine single- and double-flow modes.

Figure 4 shows the kinematics of the first two-range HPT option with a regular differential that has a three-wheel hydraulic torque converter assembly and two controllable friction couplings.

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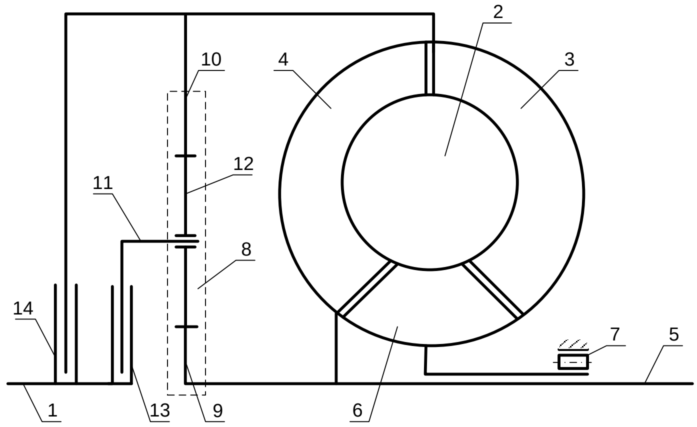

Figure 4. The kinematics of a two-range hydraulic power transmission with a regular differential and two friction clutches

The transmission includes the input shaft Pos. 1. and the hydraulic torque converter Pos. 2 consisting of the pump Pos. 3, and the turbine Pos. 4 connected to the output shaft Pos. 5. The reactor Pos. 6 is attached to the free wheel clutch Pos. 7. The regular differential Pos. 8 consists of the central (sun) gear Pos. 9 connected to the hydraulic torque converter's runner and the transmission output shaft. The crown gear Pos. 10 is connected to the hydraulic torque converter's impeller. The differential carrier Pos. 11 carries the satellite gears Pos. 12. The controllable friction clutch Pos. 13 connects and disconnects the carrier to the input shaft. The second controllable friction clutch Pos. 14 connects the input shaft to the crown gear and the hydraulic torque converter's runner. The friction clutches are controlled by the HPT hydraulic system.

In the first range this HPT operates in the single-flow mode. The friction clutch Pos. 13 is disengaged, and the friction clutch Pos. 14 is engaged. The torque is delivered from the input shaft to the clutch Pos. 14 and to the impeller Pos. 13, and then to the runner Pos. 4, and the output shaft Pos. 5. The carrier Pos. 11 in this mode rotates freely. In the single-flow mode an HPT has an enhanced torque ratio that depends of the hydraulic torque converter specifications.

In the second range the HPT operates as a double-flow transmission with enhanced efficiency. In this mode, the friction clutch Pos. 13 is disengaged, the friction clutch Pos. 14 is engaged and connects the input shaft to the crown gear Pos. 10 and the impeller Pos. 3. The differential splits the engine torque into two parallel flows which are merged at the output shaft.

To ensure the engine power flow continuity as the self-propelled vehicle moves and the HPT ranges are shifted, engagement of one clutch and disengagement of the other clutch shall overlap. Regardless of the HPT mode, both friction clutches shall be sized for the max engine torque.

For the HPT under consideration, a third mode is possible as both Pos. 13 and Pos. 14 friction clutches are engaged. In this case the entire transmission is blocked: both the differential and the hydraulic torque converter are blocked. The ratio in this case is 1. The hydraulic torque converter fluid can be drained, and the engine torque would be transferred from the input to the output shaft with max efficiency. This HPT mode can be used to drive a self-propelled vehicle in favorable road conditions.

Figure 5 shows the second option of a two-range HPT with a regular differential. The friction clutch Pos. 14 is replaced with the friction clutch Pos. 15 [7]. In a single-mode version of this HPT configuration the friction clutch Pos. 13 is disengaged, the differential carrier is disconnected from the input shaft and rotates freely. The torque from the input shaft is transferred to the impeller and to the engaged free wheel clutch Pos. 15. In the double-flow mode the friction clutch Pos. 13 id engaged, the carrier Pos. 14 is connected to the input shaft. Since the impeller and the crown gear Pos. 10 connected to it rotate faster than the input shaft, the free wheel clutch is disengaged.

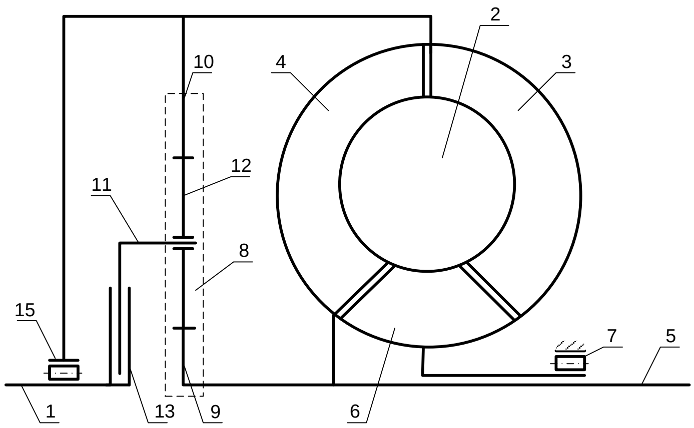

Figure 5. The kinematics of a two-range hydraulic power transmission with a regular differential and one friction clutch

A similar approach to combining a single- and double-flow HPT modes can be applied to a summing differential transmission. Figure 6 shows the kinematics of the first two-range HPT option with a summing differential equipped with a hydraulic torque converter assembly and two controllable friction clutches [3].

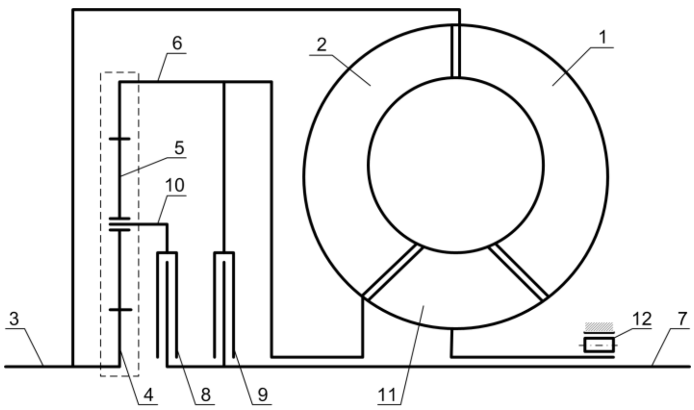

Figure 6. The kinematics of a two-range hydraulic power transmission with a summing differential and two friction clutches

The hydraulic torque converter consists of the impeller Pos. 1 and the runner Pos. 2. The latter is connected to the input shaft Pos. 3. The sun gear Pos. 4 (the differential is highlighted with a dash line) is permanently connected to both the input shaft and the impeller. The runner is connected to the crown gear Pos. 6. The reactor wheel Pos. 11 is attached to the free wheel clutch Pos. 12. The carrier Pos. 10 with the satellite gears Pos. 5 can be engaged/disengaged to the output shaft Pos. 7 using the controllable friction clutch Pos. 8. The second he controllable friction clutch Pos. 9 engages the runner to the output shaft. The Pos. 8 and 9 friction clutches are controlled by the HPT hydraulic system.

In the first range this HPT operates in the single-flow mode. The friction clutch Pos. 8 is disengaged, and the friction clutch Pos. 9 is engaged. The torque is delivered from the input shaft to the the impeller Pos. 13, and then to the runner Pos. 2, and the output shaft Pos. 7. The carrier Pos. 10 rotates freely. In the single-flow mode an HPT has an enhanced torque ratio that depends of the hydraulic torque converter specifications.

In the second range the HPT operates as a double-flow transmission with enhanced efficiency. In this case, the friction clutch Pos. 9 is disengaged, the friction clutch Pos. 8 is engaged and connects the input shaft Pos. 3 to the carrier Pos. 10; the runner Pos. 2 drives the crown gear Pos. 6. The engine torque is split into two parallel flows to be merged at the carrier.

For the HPT under consideration, a third mode is possible as both Pos. 8 and Pos. 9 friction clutches are engaged. In this case the entire transmission is blocked: both the differential and the hydraulic torque converter are blocked. The ratio in this case is 1. The hydraulic torque converter fluid can be drained, and the engine torque would be transferred from the input to the output shaft with max efficiency. This HPT mode can be used to drive a self-propelled vehicle in favorable road conditions.
Figure 7 shows the second option of a two-range HPT with a summing differential. One of the friction clutches is replaced with a friction clutch [8].

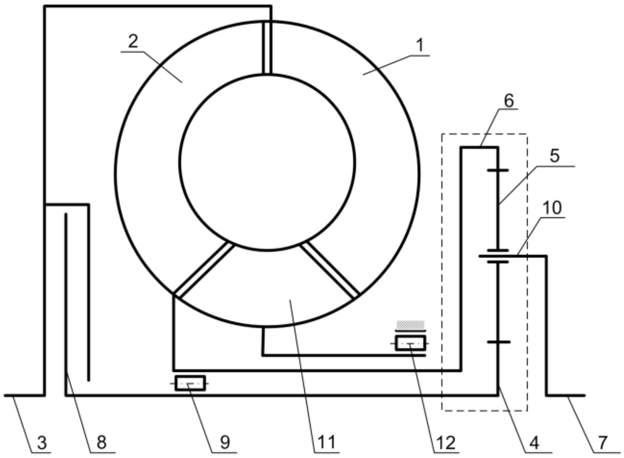

Figure 7. The kinematics of a two-range hydraulic power transmission with a summing differential and one friction clutch

The hydraulic torque converter consists of the impeller Pos. 1 and the runner Pos. 2. The latter is connected to the input shaft Pos. 3. The sun gear Pos. 4 (the differential is highlighted with a dash line) is connected to the input shaft with the clutch Pos. 8. The runner is connected to the crown gear Pos. 6. The reactor wheel Pos. 11 is attached to the free wheel clutch Pos. 12. The carrier Pos. 10 with the satellite gears Pos. 5 are permanently connected to the output shaft Pos. 7. The free wheel clutch Pos. 9 is installed between the differential's sun and crown gears.
In the single-flow range for this HPT configuration the friction clutch Pos. 8 is disengaged. The input shaft Pos. 3 drives the impeller Pos. 1. The runner Pos. 2 drives the crown gear Pos. 6 and the sun gear Pos. 4 through the free wheel clutch Pos. 9. The carrier drives the output shaft. Since the angular velocities of the crown and sun gears are equal, the free wheel clutch Pos. 9 blocks the differential (the differential ratio is 1.)
In the double-flow HPT range the friction clutch Pos. 8 is engaged. It connects the sun gear Pos. 4 to the input shaft Pos. 3. In this case the differential sums the power flows coming from the input shaft and the runner. Since the sun gear angular velocity exceeds that of the crown gear (due to the impeller and the runner slippage) the free wheel clutch Pos. 9 is disengaged (uncoupled.) In this way the free wheel clutch partially automates the HPT range switchover with just one controllable free wheel clutch; it somewhat simplifies the control system. The free wheel clutch and the friction clutch shall be sized for the max engine torque.

5. Conclusion

Hydraulic power transmissions with switchable hydraulic torque converters are designed for both road and off-road self-propelled vehicles. Extra controls enable an extended range for the original hydraulic torque converter ratio. Free wheel clutches automate single/double-flow switchover. The proposed HPTs can be combined with robotic fixed-ratio gearboxes or as a part of a vehicle drive train consisting of a conventional clutch, and a manual gearbox.
The proposed HPT kinematics combines the advantages of a single-flow hydraulic power transmission (higher torque ratios and better efficiency in a lower ratio range) and of a double-flow hydraulic power transmission (better efficiency in a higher ratio range). As a result, the combined HPTs can improve

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the average self-propelled vehicle drive train efficiency as compared to a drive train with a single-flow HPT while maintaining high torque ratio in a low ratio range.

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