A Concept of Hydraulic Power Transmission for Heavy Self-Propelled Mining Vehicles

N Trushin

1 Tula State University, Polytechnic Department, Tula, 300012, Russian Federation

E-mail: trushin@tsu.tula.ru

Abstract. The study considers a concept of hydraulic power transmission with a multi-stage hydraulic torque converter. The design objective is expanding the operational envelope of a self-propelled vehicle, increasing the power train efficiency in the entire engine torque transformation range. Two hydraulic torque converter (HTC) designs are proposed. The HTC uses uncoupled turbines (centrifugal, axial, centripetal) in a series arrangement. Each turbine operates in its optimal range of the HTC ratios. Each turbine is connected to the output shaft with individually controlled couplings. To optimize the hydraulic torque converter performance, a dedicated stator for each turbine is proposed. It is activated/deactivated concurrently with the respective turbine. The proposed hydraulic power transmission is designed for heavy self-propelled vehicles operated in mining and construction. The design can also be used in railway engine power trains to replace the complicated multistage hydraulic transmissions consisting of two or three conventional hydraulic torque converters. The hydraulic power transmission kinematics and principle of operation are protected by the Russian patent No. 2716378. The study is supported by Tula State University as research project No. 2019-22.

1. Introduction

Hydraulic power transmissions (HMT) with torque converters single-stage are extensively used in various self-propelled vehicles. The most common hydraulic torque converters have three wheels: an impeller connected to the input (drive) shaft, a turbine connected to the output (driven) shaft, and one immovable stator. There are also single-stage four-wheel hydraulic torque converters with two stators. In combined hydraulic torque converters the stator is attached to the casing with a freewheel clutch to enable the fluid coupling mode. Now power train designers extensively use the hydraulic torque converter lock-up clutch, even on heavy vehicles (quarry dump trucks, blade graders, bulldozers, etc.) which was not the case. The reason is fuel saving and total standardization of the self-propelled vehicle power trains [2].

The advantages of HPTs with hydraulic torque converters are fully utilized in heavy machinery that operates under variable loads in construction and ore mining industries [3, 4]. It is the driver for the continuous car/tractor hydraulic torque converter production growth for the last 70-75 years. On June 10, 2020 Precision Report released Global Torque Converter Market Insights and Forecast to 2026 (https://www.precisionreports.co/global-torque-converter-market-15780667). The global Torque Converter market size is projected to reach US$ 5949.3 million by 2026, from US$ 5745.7 million in 2020, at a CAGR (Compound Annual Growth Rate) of 3.3% during 2021-2026. Torque Converter market competitive landscape provides details and data information by manufacturers. The report
offers a comprehensive analysis and accurate statistics on production capacity, price, the revenue of Torque Converter by the player for the period 2015-2020. It also offers detailed analysis supported by reliable statistics on production, revenue (global and regional level) by players for the period 2015-2020. Details included are company description, major business, company total revenue, and the production capacity, price, revenue generated in Torque Converter business, the date to enter into the Torque Converter market, Torque Converter product introduction, recent developments, etc. The major vendors covered: Schaeffler, ZF, Aisin, Valeo, Kapec, Transtar, Exedy, Yutaka Giken, Borgwarner, Sonnax Industries, Hitachi Nico Transmission Co.

It means that hydraulic torque converters are still a key component of self-propelled vehicle power trains. Correspondingly, there is a demand for a comprehensive enhancement of HPT/hydraulic torque converter designs [5].

2. Problem statement

The primary issue with single-stage hydraulic torque converters is a relatively narrow engine torque adjustment range, and low efficiency in the low ratio range ($i<0.6$). The max torque ratio ($K$-factor) for single-stage hydraulic torque converters does not exceed $K=3.5-4$, while for most commercially available hydraulic torque converters the max torque ratio is $K=1.8-2.5$ [1, 2]. To eliminate this shortcoming, a single-stage hydraulic torque converter is combined with a mechanical $n$-speed gearbox. The number of speeds in self-propelled vehicle HPTs can be as high as 8-10 and there is a trend of adding more speeds up to 12-16 and more [6, 7]. The more speeds the gearbox has, the more complicated its mechanical component and the control system are, and the more engine power is used for auxiliary functions reducing the overall drive train efficiency.

Multi-stage torque converters have higher ratios. In such converters, the ratio can be as high as $K=4.5-6$ or even higher. Besides, in the low ratio range, multi-stage torque converters have higher efficiency compared to single-stage ones. With the high ratio of multi-stage torque converters, the number of speeds in mechanical gearboxes can be reduced. It would simplify the power train kinematics and control system.

Multistage torque converters are divided into two-stage and three-stage ones, with two or three turbines, are simultaneously connected to the output shaft, respectively. A separate stator is coupled to each turbine in multistage torque converters so that the torques on each turbine have the same direction. A two-stage torque converter usually has five wheels: one impeller, two turbines and two stators. Three-stage torque converters have six to seven wheels: one impeller, three turbines and two or three stators.

It is known that the hydraulic torque converter turbine type significantly affects its performance. For instance, a centrifugal turbine performs better in the low ratio range ($i=0-0.4$), an axial turbine--in the mid ratio range ($i=0.4-0.6$), and a centripetal turbine--in the high ratio range and in the fluid coupling mode ($i=0.6$) [8]. For these reasons, multi-stage hydraulic torque converters with various types of turbines are used on vehicles. Lysholm-Smith [9], Twin Disc [10], Brockhouse-Salerni [11, 12], Packard [13], SRM [14], Volvo [15] have been used on vehicles. Such torque converters are used in power trains of heavy vehicles, buses, tractors, tanks, railway engines [3, 4]. In Russia, two-stage hydraulic torque converters are used in standard HPTs of the TGM series diesel shunters equipped with 750 and 1,200 hp diesel engines. (http://www.kalugaputmash.inni.info/produt/gidoperedachi).

The disadvantages of multi-stage hydraulic torque converters are a relatively complicated design, reduced efficiency in the high ratio range ($i>0.8$). Mostly they do not support the fluid coupling mode. Combined versions of two-stage torque converters are also known. They can switch to the fluid coupling mode as the ratio exceeds 0.8 (for example, the above-mentioned Brockhouse-Salerni torque converter.) However, in the fluid coupling mode combined multistage torque converters have lower efficiency compared to single-stage combined torque converters due to a relatively large number of rotating wheels. To improve the power train efficiency both single-stage and multi-stage hydraulic torque converters are equipped with a lockup clutch.
A separate stator is coupled to each turbine in multistage torque converters so that the torques in each turbine would have the same direction. The vane and blade airfoil geometry for multi-stage hydraulic torque converters is a result of a tradeoff since all the turbines are rigid attached to the output shaft and rotate at the same angular velocity. It is also a significant drawback of multi-stage hydraulic torque converters.

3. Design results
The design objective is eliminating the disadvantages of HPTs with single-stage hydraulic torque converters, expanding the operational envelope of a self-propelled vehicle, increasing the power train efficiency in the entire engine torque transformation range. The proposed HPT is patented, Russian patent No. 2716378 [16].

Fig. 1 shows a kinematic diagram of the proposed HPT: a configuration with a three-stage lock-up hydraulic torque converter with three centrifugal, axial, and centripetal turbines, three controlled stators, and three clutches that connect the turbines to the output shaft. The HPT components are hydraulically controlled.

![Hydraulic power transmission, option 1. Kinematic diagram.](image)

In the first HPT option, a three-stage hydraulic torque converter comprises of a centrifugal pump, the T1 first stage centrifugal turbine, the T2 second stage axial turbine, and the T3 third stage centripetal turbine. The T1, T2, T3 turbines are connected to the output shaft with the C1, C3, C3 clutches, respectively.

The hydraulic torque converter has three stators: S1, S2, S3. Each stator S1, S2, S3 is connected/disconnected to the fixed casing with the B1, B2, B3 brakes. As one of the B1, B2, B3 brakes are engaged, one of the S1, S2, S3 stators is activated. Each of the three stators is designed for optimal joint operation with one of the T1, T2, T3 turbines.

The lock-up clutch (LUC) locks the hydraulic torque converter by directly coupling the output shaft to with the input shaft. A LUC is optional.

The HPT operates as follows.

The pump supplies the fluid pressure flow. The fluid enters first the turbine T1, and then the turbines T2 and T3. In the low ratio range (i=0-0.3) the T1 centrifugal turbine is in operation; in the
mid ratio range \((i=0.3-0.6)\) the T2 axial turbine is in operation; in the high ratio range \((i=0.6-0.98)\) the T3 centripetal turbine is in operation. The C1, C2, C3 automatically controlled clutches alternatively connect and disconnect the T1, T2, T3 turbines with the output shaft as the self-propelled vehicle gets going and accelerates. Generally, as one clutch is engaged, the other two clutches are disengaged. This logic is implemented by the control system.

As one of the C1, C2, C3 clutches is engaged, one of the S1, S2, or S3 stators (the one optimized for joint operation with the specific turbine) is also activated. For instance, the S1 stator operates together with the T1 turbine, the S2 stator operates together with the T2 turbine, and the S3 stator operates together with the T3 turbine. The stators are activated and deactivated by engaging and disengaging the B1, B2, or B3 brakes. For instance, as the C1 clutch is engaged, the B1 brake is also engaged automatically, as the C2 clutch is engaged, the B2 brake is engaged, and as the C3 clutch is engaged, the B3 brake is engaged. As one brake is engaged, the other two are disengaged. In this way, when the vehicle gets going and accelerates and the hydraulic torque converter ratio increases, the T1, T2, T3 turbines are progressively activated and deactivated while the respective S1, S2, S3 stators are also activated and deactivated. After accelerating and reaching the cruise speed, the hydraulic torque converter is locked with the LUC.

To enable the fluid coupling mode the C3 clutch is engaged, the C1 and C2 clutches are disengaged, and the B1, B2, B3 brakes are also disengaged. In the fluid coupling mode, only the pump and the T2 turbine operate. All the C1, C2, C3 clutches can be concurrently engaged. As a result, all the wheels are combined into a single wheel. Another option is a concurrent engagement of the C2 and C3 clutches that join the turbines T2 and T3.

Fig. 2 shows a kinematic diagram of the second hydraulic power transmission configuration with a simpler two-stage hydraulic torque converter that has two axial and centripetal turbines and two controllable stators.

![Figure 2. Hydraulic power transmission, option 2. Kinematic diagram.](image)

The two-stage hydraulic torque converter consists of a centrifugal pump, the T2 first stage axial turbine, the T3 second stage centripetal turbine, and two controllable S2 and S3 stators. The clutches
C2 and C3 connect the turbines T2 and T2 to the output shaft. The brakes B2 and B3 activate and deactivate the stators S2 and S3, respectively. The hydraulic power transmission components T1, S1, C1, and B1 (refer to Fig. 1) are not presented in Fig. 2 since the original HPT kinematics was simplified.

The second HPT option operates basically in the way as the first one. In the 0-0.6 ratio range the T2 axial turbine operates; as the ratio exceeds 0.6 and in the fluid coupling mode, the T3 centripetal turbine operates. The HPT control system concurrently engages, for instance, the C2 clutch or the T2 brake, or the C3 clutch and the T2 brake. The S2 and S3 stators can be connected to the casing with freewheel clutches to automatically disengage the stators as the hydraulic torque converter switches to the fluid coupling mode. In the fluid coupling mode, the pump and the T3 turbine operate, the C3 clutch is engaged, the B2 and B3 brakes are disengaged. The hydraulic torque converter can also by locked with the LUC.

4. Conclusion
The proposed HPT offers a wider operational envelope compared to the existing designs. The independent operation of the turbines at the first and second stages and the corresponding three stators makes it possible to optimally profile their blades and thereby provide higher efficiency in the entire ratio range. The HPT control system can implement flexible control logic and change the hydraulic power transmission properties online depending on the riding conditions and on the power train load. As compared to multi-flow hydraulic power transmissions with two or three hydraulic torque converters, the proposed hydraulic power transmission for all other conditions being equal has smaller dimensions and weight. The proposed HPT can be used on hoisting, construction, earth-moving, and railroad machinery.

5. References
[1] Heisler H 2002 Advanced Vehicle Technology Second Edition Butterworth-Heinemann p 654
[2] Nanney M J 2007 Light and Heavy Vehicle Technology Fourth Edition Oxford: Elsevier p 671
[3] Duffy O C, Heard S A, Wright G 2019 Fundamentals of Mobile Heavy Equipment Jones & Bartlett Learning p 1406
[4] Huzij R, Spano A, Bennet S 2014 Modern Diesel Technology: Heavy Equipment Systems 2nd Edition NY Delmar, Cengage Learning p 583
[5] Hossay P 2020 Automotive Innovation: The Science and Engineering behind Cutting-Edge Automotive Technology Boca Raton, FL: CRC Press/Taylor & Francis Group p 308
[6] Naunheimer H, Bertsche B, Ryborz J, Novak W 2011 Automotive Transmissions Second Edition Berlin Heidelberg: Springer-Verlag p 740
[7] Fischer R, Kçuçuk F, Jürgens G, Najork R, Pollak B 2015 The Automotive Transmission Book (Switzerland: Springer International Publishing) p 355
[8] Mazalov N D, Trusov S M 1971 Hydromechanical Automotive Gear Boxes (Moscow: Mashinostroenie) p 296
[9] Lysholm A J R 1933 Hydraulic variable speed power transmission (Patent US1900118)
[10] Shorts W F 1955 Vehicle power transmission (Patent US2727601)
[11] Salerni P M 1942 Hydraulic power transmission apparatus (Patent US2293767)
[12] Gatiss A L 1950 Improvements in or relating to hydraulic transmission apparatus (Patent GB640727)
[13] Misch H L, Lucia C J 1953 Transmission (Patent US2630893)
[14] Ahlen K G 1954 Hydrodynamic torque converter (Patent US2690053)
[15] Kronogard S-O 1964 Hydrodynamic torque converter (Patent US3154924)
[16] Trushin N N 2020 Hydromechanical transmission of vehicle (Patent RU2716378)
Acknowledgments

We thank the personnel of the Research Library, Patent and licensing department Tula State University, and of the Tula Region Research Library for their assistance with reference and patent search, and innovative design development. The study is supported by Tula State University as research project No. 2019-22.