Two power split path hydrodynamic drive train of the commercial tractor

M V Vyaznikov and A M Vyaznikov*
IES UB RAS, Department of Transportation Vehicles Mechanics, 620049, Ekaterinburg, Russia

* priss1912@yandex.ru

Abstract. The article presents hydromechanical drive trains with a differential link at the input and output. As a differential link, three-link differential mechanisms with mixed gears mesh are used. The possible schemes of two-path hydrodynamic drive trains are investigated. It is shown that the operation mode of a two-path hydromechanical drive train is possible when the turbine wheel of the torque converter rotates in the direction opposite to the impeller rotation. This factor is taken into account when choosing the characteristics of a two-path hydromechanical drive train. It has been established that for the most common two-path hydromechanical drive train scheme, the planetary gear set characteristic reduction increases its maximum efficiency, decreases the torque converter active diameter and increases the relative speed ratio of the three-link differential spider pinion. It is shown that a two-path hydromechanical drive train in comparison with a one-path drive train allows for using more efficiently the engine transforming property to change the torque when changing the output load on the engine, which is very important for commercial tractors operating under variable load conditions.

1. Introduction

The operating mode peculiarities of commercial tractors, modularized with various operating equipment, are characterized by high dynamic loads acting on the transmission structure elements [1, 2]. Therefore, the design of such tractors actively comprises hydromechanical drive trains (HMDT) with hydrodynamic transformers (TC) or complex hydrodynamic drive trains, providing automatic change of torque as the output load on the tractor changes.

The technique of designing and calculating one-path (full-power) HMDT is currently quite well developed [1-7].

In one-path HMDT, the efficiency, the kinematic and power gear ratios are equal to the product of the efficiency and the gear ratios of the corresponding mechanisms. These drives have a wide range of regulation, but relatively low efficiency [3-7].

Two-path HMDTs have a higher efficiency, in them, power is transmitted by two paths through mechanical and hydraulic links. Such a transmission usually consists of a TC and a differential mechanism, implemented in the form of a three-link differential mechanism (TDM) with mixed or external gearing. In this case, only part of the power is transmitted through the TC, while the
remaining power is transmitted through a mechanical drive (gear), which has a much higher efficiency than the TC.

2. Main part
Currently, there are no clear techniques for selecting the main characteristics of two-path HMDT, therefore, the study of selecting such characteristics for commercial tractors is topical.

The main characteristics of two-path HMDT are the kinematic and power ratios and efficiency.

Depending on the location of the differential link in relation to the TC, there are two-path HMDTs with a differential link at the input or at the output [3, 4, 6, 7]. As an example, we consider the most common scheme of a two-path HMDT with a differential link at the output (Figure 1) [3, 4, 7].

In this drive system, power is transmitted from the driving to the driven shaft through two paths. The first power path is transmitted in a purely mechanical way through the sun gear to the pinions and then to the carrier connected to the driven gear shaft. Here, there are only mechanical power losses, the drive efficiency in this power path is high.

The second power path is transmitted through the TC to the epicyclic gear and then through the pinions to the carrier. There are power losses both in the TC with low-efficiency and in the mechanical drive part with high-efficiency. Here, the power losses are higher.

Thus, the TDM carrier sums up the power of two paths.

Figure 1. Scheme of a two-path HMDT with a differential link at the output: a is sun gear; c is epicyclic gear; в is carrier; Bo is pinion; H is impeller; T is turbine wheel; P is reactor; $M_1$, $M_2$ are torques on the HMDT driving and driven shafts; $n_1$, $n_2$ are rotational speeds of the HMDT driving and driven shafts
This is due to the fact that, through the TC, only part of the power is transmitted from the HMDT main shaft. The other part of the power is transmitted through mechanical gear links with high efficiency.

The selection of the main characteristics of the two-path HMDTs will be considered by means of the HMDT scheme with a differential link at the output shown in Figure 1.
Figure 2 shows all possible schemes for two-path HMDTs with a differential link, made in the form of a TDM with mixed gearing [3, 4, 7]. In a number of schemes shown in Figure 2, in the power circuit, a circulating power $N_c$ appears. As a result, the efficiency of such a drive is $\eta_{HMDT} < \eta_{TC}$, and the circulating power $N_c$ additionally loads the HMDT mechanical or hydraulic parts.

In Schemes 1, 2, 7, 8, the circulating power $N_c$ is absent [3, 4, 7]. These schemes increase the drive efficiency, reduce the power ratio (transformer ratio) and allow for reducing the size of TC compared to full-power drive train. The most rational is Scheme 1, which is widely applied (see Figure 1).

In Schemes 3, 4, 9, 10, the circulating power overloads the TC, which leads to an increase in its size, a decrease in the drive efficiency and an increase in its power ratio [3, 4, 7].

In Schemes 5, 6, 11, 12, the circulating power overloads the mechanical drive parts, which leads to a decrease in drive efficiency and power ratio as compared to the TC. Therefore, these schemes are irrelevant [3, 4, 7].

The output characteristic of a two-path HMDT is the dependence of the torques $M_1$ on the driving and $M_2$ on the driven transmission shafts and of its efficiency $\eta_{HMDT}$ on the reduction ratio $\kappa$ at a constant rotational speed $n_1$ of the driving shaft. It can be built according to the TC output characteristic for a given magnitude of the planetary set characteristic $\kappa$.

Let us consider the technique of building the output characteristic of a two-path HMDT (see Figure 1) with the TC, which output characteristic is shown in Figure 4.

It should be noted that in this two-path HMDT scheme, it is possible to have an operation mode when the turbine wheel of the TC rotates in the direction opposite to the impeller rotation (turbine wheel counter-rotation). This factor must be taken into account when building the output characteristics of a two-path HMDT.

The results obtained are used to build an output characteristic of a two-path HMDT (Figure 5).

The output characteristics of one-path and two-path HMDTs (see Figure 1) with different characteristics $\kappa$ of the planetary set and an output characteristic of the TC, shown in Figure 4, are presented in Figure 6.

From the analysis of the schemes given, it follows that two-path HMDTs have higher magnitudes of maximum efficiency. At the same time, with a decrease in the characteristic of the planetary set $\kappa$, the efficiency magnitude increases.
The change in the characteristic of the planetary set $\kappa$ also influences other parameters of the HMDT output characteristic. Thus, an increase in $\kappa$ leads to a decrease in the HMDT transparency and an increase in its transformer coefficient. The HMDT load characteristic is the dependence of the HMDT driving shaft torque $M_1$ on the rotational speed $n_1$ of its driving shaft. Since the HMDT driving shaft is connected to the engine shaft, then when the torque $M_1$ changes on the driving shaft, the engine load will change. This characteristic is sometimes called the input characteristic of the HMDT. To build the load characteristic of the HMDT the expression (4) is used.

To test the possibility of joint work, the output speed characteristic of the engine is combined with the load characteristic of the HMDT.

When coordinating the loading characteristic of the HMDT with the full load characteristic of the engine, it is assumed that the HMDT maximum efficiency should correspond to the engine running in a nominal mode.
Figure 7. The computational scheme of the "Engine – Two-path HMDT": $M_d, n_d$ is torque and rotational speed of the engine shaft; $u_p$ is the power ratio of the matching gear reducer

The coordination of the load characteristic of the HMDT with the full load characteristic of the engine can be performed in two ways: by changing the active diameter $D$ of the Torque Converter and by selecting a corresponding parameter of the planetary set $\kappa$ [8-11].

When coordinating the load characteristic of a two-path HMDT with the full load characteristic of the engine, it is desirable to provide an intersection of the load characteristic left-hand parabola with the engine torque curve at point A (see Figure 8) corresponding to the maximum torque $M_{d1}$.

This will allow the most complete use of the transforming property of the engine to change the magnitude of the torque moment when the output load on the engine changes. This is especially important for engines of constant power, in which the adjustability factor in magnitude of torque can reach values of 1.4-1.65 [1-3].

The extreme left parabola in Figure 8 corresponds to the mode of operation, when $\lambda_1$ has the maximum value (point 1 in Figure 5), the extreme right parabola is built for the minimum value of $\lambda_1$ (point 4 in Figure 5).

Calculations have showed that in a two-path HMDT, the left parabola of the load characteristic intersects the torque curve of a diesel engine at point B, located closer to point A, which corresponds to the value $M_{d1}$ of its maximum torque, compared to one-path HMDT. Consequently, the considered scheme of a two-path HMDT as compared with a one-path allows the most efficient use of the engines transforming property to change the torque value when the output load on the engine changes.

Experimental studies were carried out at the testing bench (Figure 9), which included an electric dynamometer with a measuring shaft as a drive, and an eddy-current brake as a loading device. As a working fluid in the HMDT, torque converter fluid MGT TU 38.401.220-80 was used, the supply of
which to the HMDT was carried out by an automatic pumping unit. When removing the output characteristic of the two-path HMDT and TC, the fluid temperature at the input to the HMDT was maintained in the range of 75-90 °C.

Figure 9. Kinematic bench scheme: a is sun gear; c is epicyclic gear; c is carrier; H is impeller; T is turbine wheel; P is reactor; $M_1$ and $M_2$ are torque, respectively, on the HMDT driving and driven shafts; $n_1$ and $n_2$ are rotational speed, respectively, of the HMDT driving and driven shafts

3. Conclusions
The described technique allows for determining all the necessary parameters of a two-path HMDT at the design stage. It was established that with a decrease in the characteristic of the planetary set $k$, the maximum efficiency of the two-path scheme of the HMDT increases, the active diameter $D$ of the torque converter decreases, and rotational speed of the TDM planetary pinions increases. An increase in the characteristic of a planetary set $k$ leads to a decrease in the HMDT transparency and to an increase in its transformer coefficient. The two-path HMDT allows the most complete use of the engine-transforming property to change the magnitude of torque when the output load on the engine changes, which is very important for commercial tractors.

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