A method for assessing the cross-country capability of a particularly heavy-duty vehicle with all-wheel control based on the compliance of the support surface during curved movement

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Abstract. The article presents a method for assessing the permissible speed of a particularly heavy-duty vehicle based on a road train using a combined hydraulic transformer in the tractor transmission, as one of the indicators of cross-country ability. In this paper, we consider one of the main ways to increase the cross-country ability of heavy-duty vehicles: the creation of heavy-duty vehicles based on a road train with an active trailer link. Increasing the passability of heavy-duty vehicles is primarily aimed at increasing the permissible speed of heavy-duty vehicles and the maximum angles of ascents to be overcome. When justifying the choice of an algorithm for controlling the voltage of a torque Converter inductor based on the speed of particularly heavy-duty vehicles, it is advisable to use the external characteristics of the torque Converter as evaluation indicators: the dependence of the efficiency coefficient, the dependence of the moment created by the pump, and the dependence of the transformation coefficient on the gear ratio of the torque Converter. As criteria for evaluating the feasibility of using a combined torque Converter for especially heavy-duty vehicles, it is advisable to use the maximum increase in efficiency, the moment created by the pump wheel, and the transformation coefficient of torque converters on magnetic fluid in relation to the efficiency, the moment created by the pump wheel, and the transformation coefficient of the torque Converter based on a conventional working fluid in the operational range of gear ratio changes. The external characteristic is calculated after calculating the blade system and selecting the design to determine whether the main parameters of the characteristic correspond to the parameters specified in the task for designing the torque Converter. Such calculations often have to be performed when fine-tuning torque converters with a known design, when the influence of geometric parameters of individual elements of the blade system (input and output angles, wheel radii, the number of blades in the impellers, etc.) on the transforming and loading properties of the torque Converter is investigated.

1. Introduction
The use of hydro-mechanical transmission has raised the transmission design of especially heavy-duty vehicles to a qualitatively new technical level in terms of reducing the load on the transmission and improving the driver's working conditions, but it has not solved the problem of individual power
supply to the wheels depending on their rolling conditions. Various researchers have concluded that
the achieved technical level of mechanical and hydro-mechanical transmission designs of vehicles
with particularly high load capacity (VPHL) exhausted the possibility of further significant increase in
the mobility of multi-axle chassis. Thus, when increasing the power of the power plant VPHL over
600 kW, it is difficult to create mechanical and hydro-mechanical transmissions with high technical
and economic indicators. Therefore, with the growth of power, load capacity, and as a result of the
number of axles, new technical solutions and qualitatively better methods of converting, distributing
and transmitting energy in transmissions are needed VPHL. All this has led to the fact that at present
the country and abroad are searching for new flexible types of transmissions for vehicles of
particularly high load capacity for the automotive industry [1].

2. The study of a method for assessing the cross-country capability of a particularly heavy-duty
vehicle
In this work, we consider one of the above main ways to increase the patency of a VPHL: creation of a
VPHL based on a road train with an active trailer link. Increased cross-country ability of a VPHL it is
aimed, first of all, at increasing the permissible speed of movement of a VPHL and the maximum
angles of ascents to be overcome.

Significant increase in cross-country ability of a VPHL it is possible by improving the technical
and operational characteristics of the hydro-mechanical transmission of the tractor transmission and
the motor-wheels of the trailer device.

To improve the technical and operational characteristics of the hydro-mechanical transmission of a
VPHL it is proposed to use in its composition a combined torque Converter controlled by an external
rotating magnetic field, and as the drives of the wheels of the trailer link - valve-inductor drives [2].

Initial data for solving the problem of estimating the permissible speed of movement of a VPHL
with a hydrodynamic transmission based on a combined torque Converter controlled by a rotating
magnetic field and an active trailer link are divided into the following main groups:

when justifying the choice of an algorithm for controlling the torque Converter inductor strength by
speed of a VPHL it is advisable as the evaluation indicators to use external characteristics of the
Converter:
the dependence of the efficiency,
the torque generated by the pump,
the dependence of the transformation coefficient from the reduction ratio of the torque Converter
[3,4].

As criteria for evaluating the feasibility of using a combined torque Converter for a VPHL, it is
advisable to use the maximum increase in efficiency, the moment created by the pump wheel, and the
transformation coefficient of torque converters on a magnetic fluid in relation to the efficiency, the
moment created by the pump wheel, and the transformation coefficient of a torque Converter based on
a conventional working fluid in the operational range of gear ratio changes [5].

Scheme of the algorithm for estimating the permissible speed of movement of a VPHL with a
hydrodynamic transmission based on a rotating magnetic field controlled combined torque Converter
includes the following main stages:
1) preparation and input of initial data;
2) modeling of system dynamics «internal combustion engines - hydrodynamic transmission based
on a combined TC – WT VPHL» with speed control of a VPHL.
3) the feasibility of using a hydrodynamic transmission based on a combined torque Converter
controlled by a rotating magnetic field is evaluated by comparing the combined transmission options
with each other and with existing (known) transmissions of traditional design;
4) formation of requirements (General, special, special) for the developed structures by
hydrodynamic transmission based on a combined torque Converter controlled by a rotating magnetic
field of a VPHL.
Initial data for modeling the dynamics of the system "internal combustion engine-hydrodynamic transmission based on a combined torque Converter controlled by a rotating magnetic field-wheel travel of a VPHL" under the action of operational disturbances and speed control of a VPHL due to the automatic control system.

| Initial data for mathematical modeling of an internal combustion engine |
| Source data for mathematical combined TC |
| Source data for mathematical modeling of WT |
| Исходные данные для математического моделирования прямолинейного движения of a VPHL |
| Source data for mathematical modeling of information sensors: speed sensors, tachogenerator, linear speed sensor |
| Source data for mathematical modeling of amplifier-converters |
| Initial data for mathematical modeling of feedback between internal combustion engines and load |

**Figure 1.** Structure of initial data for modeling the dynamics of the system "internal combustion engine-hydrodynamic transmission based on a combined torque Converter controlled by a rotating magnetic field-wheel travel of a VPHL" under the action of operational disturbances and speed control of a VPHL due to an automatic control system.

The external characteristic is calculated after calculating the blade system and selecting the design to determine whether the main characteristic parameters (such as M, N, n, i, K0, P, d75, etc.) correspond to the parameters specified in the task for designing the torque Converter.

Such calculations often have to be performed when fine-tuning torque converters with a known design, when the influence of the geometric parameters of individual elements of the blade system (input and output angles, wheel radii, number of blades in the impellers, etc.) on the transformative-loading properties of the torque Converter is investigated.

Equations of moments. According to the theorem on the moment of the amount of movement, the second change in this moment is equal to the moment of external forces acting on this wheel. Second change in the moment of the amount of fluid movement in the impeller:

$$\frac{dL}{dt} = \rho Q (c_{u2} r_2 - c_{u1} r_1)$$

(1)

The moment of external forces on the pump MH is determined by the force action of the channel walls and blades driven by the motor on the fluid flow:

$$M_H = Q \rho (c_{uH2} r_{H2} - c_{uH1} r_{H1})$$

(2)
The moment of the amount of movement on the pump increases as the flow moves from the inlet to the outlet, and the torque is a positive value. The moment of the amount of fluid movement on the turbine is reduced, and the torque is a negative value:

\[ M_T = Q\rho (c_{uT2}r_{T2} - c_{uT1}r_{T1}) \]  

Torque generated at the reactor:

\[ M_P = Q\rho (c_{up2}r_{P2} - c_{up1}r_{P1}) \]  

This moment is transmitted to the body with which the reactor is rigidly connected, and can be positive or negative [6].

It follows from equations (3) and (4) that if the external moment is zero, then in the General case \( r_2c_{u2} = r_1c_{u1} = r_c = \text{const} \) that is, in the absence of energy transfer and in the absence of friction, the motion of the liquid is described by the law: \( r_c = \text{const} \). According to this law, the fluid moves in the inter-wheel gaps of torque converters and hydraulic couplings.

Determination of the flow rate \( Q \) in the working cavity of the torque Converter. The transfer of energy from the drive shaft to the slave in hydrodynamic transmission is accompanied by energy losses to overcome various resistances. Losses in hydrodynamic transmission can be divided into three types - hydraulic, mechanical and volumetric, which are taken into account by the corresponding efficiency: hydraulic \( PG \); volumetric \( po \) and mechanical \( pmex \).

Hydraulic losses are counted as specific energy or pressure losses. The viscosity of the working fluid causes a change in the \( H_{H} \) and \( H_{T} \) heads. So, pump head

\[ H_H = H_{TH} - h_{mot,H} \]

where \( h_{mot,H} \) - hydraulic losses in the pump.
Figure 3. The scheme of movement of fluid in a hydrodynamic transmission:
a - in the torque Converter; b - in the fluid coupling

Turbine head

$$H_T = H_{CT} - h_{pot,T}$$  \ ((6)\)

where $h_{pot.T}$ - hydraulic losses in the turbine.

From the principle of operation of hydrodynamic gears, it follows that

$$H_T = H_T + h_{pot,R} + h'_{pot}$$  \ ((7)\)

Formula (7) expresses the balance of specific energy. From the energy balance equation in the torque Converter (7), the flow rate $Q$ can be obtained for various operating modes. In hydraulic machines, the flow deviation is taken into account at the output of the impellers by the coefficient:

$$\mu = \frac{c_{u2}}{c_{u2,\infty}}$$  \ ((8)\)

where $c_{u2}$, $c_{u2,\infty}$ - projections of the absolute velocity on the circumferential one, respectively, for a finite and infinite number of blades (see Fig. 4, a, b, v).

Figure 4. velocity Triangles taking into account the finite number of blades:
a - in the reactor; b - in the pump; v - in the turbine
Figure 5. Pressure characteristic of the torque Converter: a-head balance, $H_{tH\omega}$ и $H_{t\Gamma \omega}$ - theoretical pressure of the pump and turbine; shaded area-hydraulic losses in the torque Converter; b-hydraulic efficiency

where $C_{u2 \text{mer}}$ и $C_{u2 \text{circ}}$ the meridional and circumferential components of the speed at the exit of the previous wheel.

Flow rate value for different $i$:

$$Q = \frac{2\lambda + ic}{2a} \sqrt{4(b+ic)^2 - 4a(d+mt^2 + 2j)}$$  (13)

The calculation of the external characteristic is preceded by the calculation of the internal parameters of the torque Converter $Q$ and $H$ for various $i$. the flow rate is determined by the equation. To determine the coefficients of the equation related to the geometric parameters of the blade system of the torque Converter, you must first determine the following calculated values: $F_{H2}$, $F_{T2}$, $F_{P2}$, - the cross-sectional area of the flow, normal to the direction of the speed cm, at the exit of the corresponding impellers.

The areas are determined taking into account the constraint of the flow by the blades according to the formula:

$$F = 2\pi br\lambda.$$  (14)

In addition, determine the coefficient of resistance of the wheel

$$k = \frac{\lambda}{4R_{cp}} \cdot \frac{\lambda}{F_{\rho \omega \text{cp}}},$$  (15)

where $\lambda$ - the experimental resistance coefficient is accepted within 0.06-0.08;

$R_{cp}$ - the average hydraulic radius of the channel;

$F_{\omega \text{cp}}$ - average area of wheel channels, normal to the direction of relative flow rate.

The $K$ coefficient for each wheel is calculated for several sections (see figure 6). figure 6 shows $\Delta l_m$ - the distance along the midline in the Meridian section between the selected points; $\Delta l_m = \frac{\Delta l}{\sin \beta}$ - the actual distance along the average current line between the selected points; $\beta$ - the current angle of inclination of the blade at the point (taken from the conformal diagram); $l = \Sigma \Delta l$ - length of the middle line of the blade profile; $t = \frac{2\pi r}{z}$ - current blade pitch at the given radius $r$; $\Delta l_m = \frac{\Delta l}{\sin \beta}$ - the width of the channel between the blades; $\delta$-the thickness of the blade at this point $f = ab$ - the area of the channel between the blades, normal to the direction of velocity $\omega$ (b is the width of the channel at a given radius $r$ in the meridional plane); $f_{\omega \text{cp}} = \frac{\sqrt{n}}{\sqrt{1/f_{\omega \text{cp}}^2 + 1/f_{\omega \text{cp}}^2 + \cdots}}$ - average area of the channel ($n$ is the number of selected calculation points); $F_{\omega \text{cp}} = f_{\omega \text{cp}} \cdot z$ - the average area of the wheel channels, normal to the direction of relative speed; $4R_{r \text{cp}} = \frac{2a b}{a + b}$ - the four-fold hydraulic radius of the channel at this point; $4R_{r \text{cp}} = \frac{4R_{r1} + \cdots + 4R_{rn}}{n}$ - the average value of the quadruple hydraulic radius.
It is advisable to calculate the value of K in tabular form. After calculating the geometric parameters, we determine the coefficients a, b, c, etc., and the flow rate Q according to the equation for different values of i. The known flow rates are used to determine the pressure values \( H_{\text{eH}} = f(i) \) and \( H_{\text{ef}} = f(i) \) and calculate the hydraulic moments on the impellers:

\[
M_T = \frac{\rho g Q H_{\text{eH}}}{\omega_H}
\]

Determine the hydraulic efficiency:

\[
\eta_T = \frac{H_{\text{ef}}}{H_{\text{eH}}} = f(i)
\]

Figure 6. Calculation of the coefficient of resistance \( k \) (points 1-7 correspond to the considered blade sections)

Torque converters that provide a constant mode of engine operation when the resistance to movement changes are usually called opaque. The load characteristic of an opaque torque Converter is represented by a single square parabola (Fig. 7, a).

Figure 7. Transparency of torque converters:

a-load characteristic of an opaque torque Converter; b-load characteristic of a transparent torque Converter; C-external characteristic of a torque Converter: 1-direct transparency; 2-reverse the transparency

In General, when the speed of the drive shaft and the torque on it change, the transparency coefficient is determined from the ratio:

\[
\Pi = \frac{\lambda_{M1_{\text{max}}}}{\lambda_{M1_{\text{min}}}}
\]
Where $\lambda_{M1}$ max and $\lambda_{M1}$ min are taken respectively $M_{1\text{max}}$ and $M_{1\text{min}}$.

Determination of disk friction moments and overall efficiency.

The total efficiency of the torque Converter is calculated by taking $n_0=0.95\div0.97$ and determining the moments of disk friction and mechanical losses.

Moment of disk friction of adjacent surfaces $M_{D} = \frac{N_d}{\omega_A}$ We denote $M_{1\text{mex}}$ and $M_{2\text{mex}}$ -respectively, the moments of mechanical losses in the seals and bearings of the leading and driven parts of the torque Converter.

For traction modes of operation of the torque Converter forward when $i < 1$.

\[
\begin{align*}
M_1 &= M_{HT} + M_{A\text{HT}} + M_{A\text{HP}} + M_{1\text{mex}}; \\
M_2 &= M_{HT} + M_{A\text{HT}} - M_{A\text{TP}} - M_{2\text{mex}}. \\
\end{align*}
\] (20)

If $i > 1$, the $M_{HT}$ is preceded by a minus sign.

For reversible operation modes:

\[
\begin{align*}
M_1 &= M_{HT} + M_{A\text{HT}} - M_{A\text{HP}} - M_{1\text{mex}}; \\
M_2 &= M_{HT} + M_{A\text{HT}} + M_{A\text{TP}} + M_{2\text{mex}}. \\
\end{align*}
\] (21)

If the supports of the torque converters are made in the form of rolling bearings, and the seals are labyrinth, then $M_{1\text{mex}} = M_{2\text{mex}} \cong 0$. If there are cast-iron sealing rings or cuffs on the shafts, the moments $M_{1\text{mex}}$ and $M_{2\text{mex}}$ should be determined experimentally for each type of torque Converter. With this in mind, we determine the overall efficiency and build the external characteristic of the torque Converter $M_1, M_2, \eta = f(i)$ for $n_1 = \text{const}$.

\[
\eta = \frac{M_2}{M_1}, \frac{n_2}{n_1}
\] (22)

Figure 8. Diagram of the disk friction moments in the torque Converter

If the calculated parameters do not match the specified ones, we Refine the geometry of the blade systems theoretically or experimentally.

3. Conclusion

Thus, the proposed method makes it possible to assess the cross-country passability of wheeled vehicles of especially heavy carrying capacity with all-wheel control, taking into account curvilinear movement.

References

[1] Reimpell J and Sponagel P 1995 Fahrwerktechnik: Reifen und Räder (Vogel Fachbuch, Würzburg)
[2] Sova A N, Mazlumyan G S, Egorov O V, Eruslankin S A, Shadrin S S, Sova V A The results of modeling and evaluating the dynamics of a road train with an active trailer link based on valve-inductor electric machines *IOP Conference Series: Materials Science and Engineering*

[3] Sova A N, Mazlumyan G S, Egorov O V, Eruslankin S A, Shadrin S S, Sova V A The results of modeling and evaluating the relative slip coefficients during the interaction of the pneumatic tire with the supporting surface to determine the forces acting in the contact patch *IOP Conference Series: Materials Science and Engineering*

[4] Sova A N, Mazlumyan G S, Egorov O V, Eruslankin S A, Shadrin S S The results of modeling and evaluating the dynamics of the active semitrailer *IOP Conference Series: Materials Science and Engineering*

[5] *Dynamics of the "Road - tire - car - driver" system*. Under. ed. A A Khachaturova (Moscow: Mechanical Engineering, 1976)

[6] Mazlumyan G S, Trifonova G O, Trifonova O I, Presnyakov LA, Bulanov S V 2017 *Blade machines and hydrodynamic transmissions (textbook)* (Moscow: branch of FSUE “TSENKI” - KBTHM) ISBN 978-5-7962-0225-8