СТРУЧНИ РАДОВИ

ПРОФЕССИОНАЛЬНЫЕ СТАТЬИ

PROFESSIONAL PAPERS

CONTRIBUTION TO THE ANALYTICAL CALCULATION OF THE PERFORMANCES OF A MOTOR VEHICLE EQUIPPED WITH A HYDRODYNAMIC TORQUE CONVERTER

Miroslav D. Demić[a], *Jasna* D. Glišović[b]

[a] Academy of Engineering Sciences, Belgrade, Republic of Serbia,
e-mail:demic@kg.ac.rs,
ORCID iD: http://orcid.org/0000-0003-2168-1370

[b] University of Kragujevac, Faculty of Engineering, Department for Motor Vehicles and Motors, Kragujevac, Republic of Serbia,
e-mail:jaca@kg.ac.rs,
ORCID iD: http://orcid.org/0000-0002-8251-1722

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Abstract:

Hydrodynamic torque converters (hereinafter referred to as HDTC) are becoming increasingly widespread in motor vehicles. The internal combustion engine (ICE) and the HDTC system can be viewed as one aggregate the output characteristics of which are uniquely defined by the characteristics of the engine and the corresponding HDTC. Namely, the engine and the hydro-converter can be considered as one system, with a precisely defined output torque and output angular velocity. This creates a possibility that the drive force can be calculated as if it is a mechanical transmission, in which the output torque of the hydro-converter is the input into the planetary gears train (if any). Harmonizing the joint work of the ICE-HDTC is a complex process. An attempt has been made in this paper to make a model of the aforementioned system in order to harmonize more easily the characteristics of the combined operation of the internal combustion engine and the HDTC.

Key words: vehicles, engines, hydrodynamic, torque converter, performances.

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Introduction

The hydrodynamic torque converter (hereinafter referred to as HDTC) is a turbo machine consisting of three or more blade wheels (pump, stator and turbine) and transferring mechanical energy through the working fluid in it (Burciu, 2004), (Kelić, 1985), (Kirchner, 2007), (Laptev, 1973), (Milidrag, 1987), (Demić & Lukić, 2011). For illustration, the scheme of an HDTC on which (1) indicates a pump wheel, (2) is a turbine wheel and (3) is a stator (Laptev, 1973) is shown in Figure 1. The propulsion engine turns the pump wheel so that its blades force the fluid to perform a complex movement. The liquid turns along with the pump wheel and flows from the input to the exit from the blade channels. Due to the energy transfer to the turbine wheel, the speed of the liquid decreases. During its flow through the stator, the fluid changes direction, causing appearance of a reactive torque. In the first approximation, it can be argued that the total energy has not changed (if losses are neglected), but that only redistribution between the kinetic and potential energy of the liquid has occurred. If the redistribution of the kinetic energy is greater, the torque of the turbine is higher than the torque of the pump and vice versa, which depends on the HDTC operation mode. As the mode change is continuous, such is the change in the torque and the angular velocity of the output shaft.

In the application of the HDTC for motor vehicles, in order to achieve harmonization of the characteristics of the combined operation of the internal combustion engine (hereinafter ICE) and the HDTC, an additional mechanical single-speed gearbox is often applied, before the pump wheel. In addition, due to a low transmission ratio, the HDTC is usually combined with a planetary mechanical gearbox installed before the final drive.

The analyses carried out in (Burciu, 2004), (Laptev, 1973), (Demić & Lukić, 2011) showed that the ICE-HDTC systems can be viewed as one aggregate whose output characteristics are uniquely defined by the characteristics of the engine and the corresponding HDTC, which will be discussed below. Namely, the engine and the hydro torque-converter can be considered as one system, with an exactly defined output torque and output angular velocity. In doing so, the driving force of a vehicle can be calculated as if it is a mechanical transmission whose output torque of the HDTC is actually the input to the planetary gearset (if any). It is emphasized that the harmonization of the combined operation of the ICE and the HDTC is a complex task (Milidrag, 1987).

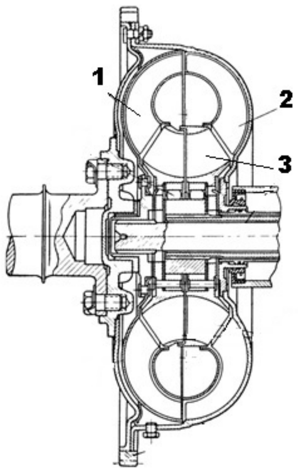

Figure 1 – Scheme of one HDTC
Рис. 1 – Схема одного ГДТКМ
Слика 1 – Шема једног ХДТОМ-а

Therefore, an attempt is made in this paper to simplify the task by using a mathematical model. Furthermore, in the following text, the behaviour of vehicles equipped with HDTCs will be observed in variable modes of motion, which required the definition of a system model that includes non-stationary and stationary operations of the ICE-HDTC system. This problem is very important for traffic in a wider sense because it enables the calculation of performances of motor vehicles.

Modelling the combined operation of the ICE-HDTC

In order to better understand the model forming process, the definition of the most important parameters of the HDTC is briefly given, (Burciu, 2004), (Demić & Lukić, 2011).

The hydraulic transmission ratio of the converter:

$$i_h = \frac{n_P}{n_T} = \frac{\omega_P}{\omega_T} \quad . \tag{1}$$

The transmission ratio of the converter:

$$i = \frac{n_T}{n_P} = \frac{1}{i_h} \; , \tag{2}$$

where

ω_P , ω_T, n_P, n_T - angular velocities and revolutions per minute (rpm) of the pump and the turbine wheel, respectively.

Slip:

$$s = \frac{n_P - n_T}{n_P} = 1 - i = 1 - \frac{1}{i_h} . \tag{3}$$

The hydrodynamic torque of the pump and the turbine wheel.

$$M_P = \lambda_P \rho D^5 n_P^2$$
$$M_T = \lambda_T \rho D^5 n_P^2 , \tag{4}$$

where

λ_P, λ_T - torque coefficient of the pump and the turbine wheels, respectively,

ρ - density of the operating fluid, and

D - active diameter of the converter.

The torque Transformation Coefficient:

$$k = \left| \frac{M_T}{M_P} \right| = \frac{\lambda_T}{\lambda_P} . \tag{5}$$

The hydraulic efficiency represents the power ratio of the turbine to the pump:

$$\eta_h = \frac{P_T}{P_P} = \frac{M_T \omega_T}{M_P \omega_P} = \frac{k M_P \omega_T}{M_P \omega_P} = ki = k \frac{1}{i_h} . \tag{6}$$

For illustration purposes, Figure 2 shows the transmission diagram of a vehicle with the HDTC (Burciu, 2004), (Laptev, 1973), (Demić & Lukić, 2011).

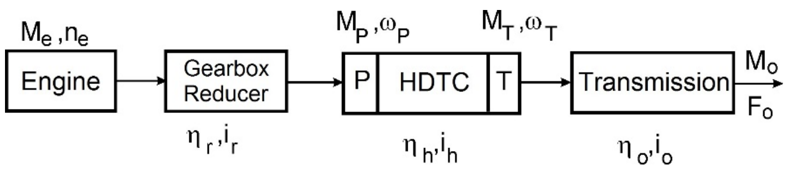

Figure 2 – Scheme of the vehicle transmission with the HDTC
Рис. 2 – Схема трансмиссии автомобиля с ГДТКМ
Слика 2 – Шема трансмисије возила са ХДТОМ

In the case of the straight-line motion, the vehicle overcomes the same resistance forces as in the case of mechanical transmission, so the differential equation of motion of the vehicle can be written (Demić & Lukić, 2011), (Genta, 2003), (Gillespie, 1992), (Janković & Todorović, 2001), (Milliken & Milliken, 1995):

$$R_j = F_o - R_f - R_v - R_u = F_o - \sum R, \tag{7}$$

where
R_j - inertial resistance (resistance of inertial forces),
R_f - rolling resistance,
R_v - aerodynamic resistance, and
R_u - gradient resistance.

Based on (7), it follows:

$$\frac{dv}{dt} = \frac{F_o - \sum R}{m\delta_u}, \tag{8}$$

where
δ_u - rotational mass coefficient which takes into account the mechanical and hydraulic elements, $\delta_u = \delta_m + \delta_h$.

Vehicle speed is given by the expression:

$$v = \frac{2\pi n_e r_d}{i_h i_m i_o}, \tag{9}$$

where
n_e - engine speed-rpm,
i_o - the final drive ratio, and
i_m - the planetary gearset ratio.

The torque on the drive wheels, according to the power and torque transmission scheme shown in Figure 2, is given by the expression:

$$M_o = M_T i_o i_m \eta_o \eta_m, \tag{10}$$

where
η_0, η_m - efficiency of the final drive and the planetary gearset, respectively.

The tractive force on the drive wheels is (Demić & Lukić, 2011):

$$F_o = \frac{M_o}{r_d} = \frac{M_T i_o i_m \eta_o \eta_m}{r_d}. \qquad (11)$$

The torque on the HDTC turbine's output shaft is (Burciu, 2004), (Kelić, 1985), (Kirchner, 2007), (Laptev, 1973), (Milidrag, 1987), (Demić & Lukić, 2011):

$$M_T = kM_P. \qquad (12)$$

Based on (11) and (12):

$$F_o = \frac{kM_P \eta_m \eta_o i_m i_o}{r_d}. \qquad (13)$$

The system shown in Figure 2 can be seen as two subsystems, i.e. two equivalent masses rotating (I, II) (Demić & Lukić, 2011):

- one subsystem includes the masses of the rotational parts, the mass of the shaft and the part of the pump before the converter, all parts being reduced to the pump's shaft, and

- the other subsystem that includes the masses of the rotational and translational moving parts behind the hydrodynamic converter, whereby all parts are reduced to the turbine's output shaft.

Subsystems I and II are shown in Figures 3 and 4.

The first subsystem

By applying Newton's law to subsystem I (Figure 3), it is obtained:

$$J_1 \frac{d\omega_P}{dt} = M_r - M_P, \qquad (14)$$

where

J_1 - equivalent moment of inertia of the first subsystem, and

M_r - torque of the gearbox reducer given by:

$$M_r = M_e i_r \eta_r, \qquad (15)$$

where

M_e - engine torque,

i_r - transmission ratio of the additional mechanical gearbox, and

η_r - efficiency of the additional mechanical gearbox.

Therefore, differential equation (14) takes the form:

$$J_1 \frac{d\omega_P}{dt} = M_e i_r \eta_r - M_P . \quad (16)$$

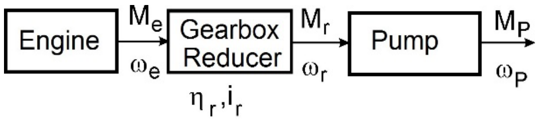

Figure 3 – Subsystem I
Рис. 3 – Подсистема I
Слика 3 – Подсистем I

The inertial moment of subsystem I can be determined as the sum of the hydraulic M_{in}^P and the mechanical M_{in}^M moments of inertia:

$$M_{in}^I = M_{in}^P + M_{in}^M , \quad (17)$$

where

M_{in}^M - moment of inertia of the mechanical part: $M_{in}^M = J_z \dot{\omega}_z$,

M_{in}^P - moment of inertia of the hydraulic part:

$M_{in}^P = M_{in}^M i_r \eta_r = J_z \dot{\omega}_z i_r \eta_r$,

i_r - transmission ratio of the reducer: $i_r = \frac{\omega_z}{\omega_P}$ and

$\dot{\omega}_z$ - angular acceleration of the flywheel: $\dot{\omega}_z = \dot{\omega}_p i_r$,

J_z - moment of inertia of the flywheel.

The moment of inertia of the hydraulic part of subsystem I is:

$$M_{in}^P = J_z \dot{\omega}_P i_r i_r \eta_r = J_z \dot{\omega}_P i_r^2 \eta_r . \quad (18)$$

The summary moment of inertia of subsystem I is:

$$M_{in}^I = \left(J_p + J_z i_r^2 \eta_r \right) \dot{\omega}_P = J_1 \dot{\omega}_P . \quad (19)$$

The equivalent moment of inertia of part I is:

$$J_1 = J_P + J_z i_r^2 \eta_r . \quad (20)$$

Second subsystem

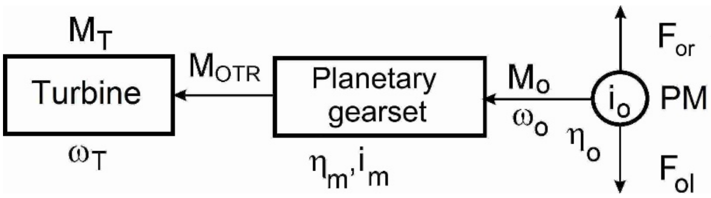

Figure 4 – Subsystem II
Рис. 4 – Подсистема II
Слика 4 – Подсистем II

Based on the distribution scheme of the torque and the tractive force of the vehicle, the resistant moment of the vehicle can be determined, Figure 4:

$$M_{OTR} = \frac{F_o r_d}{i_m i_o \eta_m \eta_o}, \tag{21}$$

where

$F_o=F_{ol}+F_{or}$ - tractive force on the drive wheels, equal to the sum of the tractive forces on the left, F_{ol} and right wheels F_{or},

By applying the Newton's law to subsystem II, shown in Figure 4, it is obtained:

$$J_2 \frac{d\omega_T}{dt} = M_T - M_{OTR}, \tag{22}$$

where

J_2 - equivalent moment of inertia of the second subsystem.

If expression (21) is taken into account:

$$J_2 \frac{d\omega_T}{dt} = M_T - \frac{F_o r_d}{i_m i_o \eta_m \eta_o} . \tag{23}$$

The moment of inertia of the wheels is:

$$M_{intockova} = \sum J_{toc} \dot{\omega}_{toc}, \tag{24}$$

where

J_{toc} and $\dot{\omega}_{toc}$ - moment of inertia and the angular acceleration of the wheels, respectively.

If it reduces to a turbine wheel, it follows:

$$M^{T}_{in\ \ tockova} = \frac{\sum J_{toc}\dot{\omega}_{toc}}{i_m i_o \eta_m \eta_o}, \tag{25}$$

where angular acceleration is:

$$\dot{\omega}_{toc} = \frac{\dot{\omega}_T}{i_m i_o}. \tag{26}$$

The final form of equation (25) becomes:

$$M^{T}_{intockova} = \frac{\sum J_{toc}}{i_m^2 i_o^2 \eta_m \eta_o}\dot{\omega}_T. \tag{27}$$

The moment of inertia of the mechanical transmission is:

$$M^{in}_{transm} = J_{transm}\dot{\omega}_{transm}, \tag{28}$$

where the angular acceleration of transmission is:

$$\dot{\omega}_{transm} = \frac{\dot{\omega}_T}{i_m}. \tag{29}$$

The moment of inertia of the transmission reduced to a turbine wheel:

$$M^{T}_{\text{int}ransm} = \frac{J_{transm}}{i_m \eta_m}\dot{\omega}_{transm} = \frac{J_{transm}}{i_m \eta_m}\frac{\dot{\omega}_T}{i_m} = \frac{J_{transm}}{i_m^2 \eta_m}\dot{\omega}_T. \tag{30}$$

The total moment of inertia is:

$$M^{T}_{in\ u} = M^{T}_{in\ \ trans} + M^{T}_{in\ toc}, \tag{31}$$

namely:

$$M^{T}_{in\ u} = \frac{J_{transm}}{i_m^2 \eta_m}\dot{\omega}_T + \frac{\sum J_{toc}}{i_0^2 i_m^2 \eta_m \eta_0}\dot{\omega}_T + J_{turb}\dot{\omega}_T \tag{32}$$

The moment of inertia of subsystem II is, (Figure 3):

$$J_2 = J_{turb} + \frac{J_{transm}}{i_m^2 \eta_m} + \frac{\Sigma J_{toc}}{i_o^2 i_m^2 \eta_o \eta_m}. \tag{33}$$

The total resistance of the inertial forces is equal to the sum of the resistance of the inertial forces due to the accelerated movement of the vehicle and the accelerated movement of the rotating parts:

$$R_j = R_j^{transl} + R_j^{rot} . \quad (34)$$

The inertial force due to the accelerated translation movement of the vehicle is:

$$R_j^{transl} = m\frac{dv}{dt} . \quad (35)$$

The inertial force generated by the accelerated rotational movement of the vehicle parts can be presented as the sum of the inertial forces of the rotational parts of the wheels and the transmission parts:

$$R_j^{rot} = R_j^{transm} + R_j^{toc} , \quad (36)$$

that is,

$$R_j^{transm} = J_2 \frac{\eta_m \eta_o i_m i_o}{r_d} \frac{d\omega_T}{dt} , \quad (37)$$

where

- the inertial force due to the accelerated rotational movement of the transmission elements and

$$R_j^{toc} = \frac{\Sigma J_T}{r_d} \frac{d\omega_{toc}}{dt} , \quad (38)$$

- the inertial force that occurs due to the accelerated rotational movement of the wheels.

The angular velocity at the exit of the turbine wheel is:

$$\omega_T = \omega_{toc} i_o i_m = \frac{v}{r_d} i_o i_m . \quad (39)$$

The angular acceleration of the turbine wheel is:

$$\frac{d\omega_T}{dt} = \frac{i_o i_m}{r_d} \frac{dv}{dt} . \quad (40)$$

The angular acceleration of the wheel is given by the expression:

$$\frac{d\omega_{toc}}{dt} = \frac{1}{r_d} \frac{dv}{dt} . \quad (41)$$

After arranging, the final expression for the resistance of inertial forces is obtained:

$$R_j = \left(\underbrace{m + \frac{\Sigma J_{toc}}{r_d^2} + J_2 \frac{i_m^2 i_o^2 \eta_m \eta_o}{r_d^2}}_{\delta_m} \right) \frac{dv}{dt}, \tag{42}$$

or

$$R_j = m \left(1 + \frac{\Sigma J_{toc}}{m r_d^2} + J_2 \frac{i_m^2 i_o^2 \eta_m \eta_o}{m r_d^2} \right) \frac{dv}{dt}. \tag{43}$$

If (according to 42 and 43) the total resistance of the inertial forces is defined as: $R_j = m \delta_m j$, the mechanical rotational masses coefficient, δ_m, is:

$$\delta_m = 1 + \frac{\Sigma J_{toc}}{m r_d^2} + \frac{J_2}{m} \frac{i_m^2 i_o^2 \eta_m \eta_o}{r_d^2}. \tag{44}$$

The angular velocity of the pump wheel is:

$$\omega_P = i_h \omega_T = i_h i_m i_o \frac{v}{r_d}. \tag{45}$$

The angular acceleration of the pump wheel is:

$$\frac{d\omega_P}{dt} = \frac{di_h}{dt} i_m i_o \frac{v}{r_d} + i_h i_m i_o \frac{1}{r_d} \frac{dv}{dt}. \tag{46}$$

If a shift is introduced: $\frac{di_h}{dt} = \frac{di_h}{dv} \frac{dv}{dt}$, the angular acceleration of the pump wheel is:

$$\frac{d\omega_P}{dt} = \frac{i_m i_o}{r_d} \left(v \frac{di_h}{dv} + i_h \right) \frac{dv}{dt}. \tag{47}$$

The torque that is transferred from the engine to the gearbox reducer according to Figure 1 is:

$$M_r = M_e i_r \eta_r, \tag{48}$$

further:

$$J_1 \frac{i_m i_o}{r_d} \left(v \frac{di_h}{dv} + i_h \right) \frac{dv}{dt} = M_r - M_P. \tag{49}$$

Bearing in mind the traction balance of the vehicle, and after the corresponding transformations:

$$m\left[\delta_m + \frac{bC}{m}J_2 + kJ_1\left(v\frac{di_h}{dv} + i_h\right)\right]\frac{dv}{dt} = bkM_r - \Sigma R, \tag{50}$$

where

$$C = \frac{i_0 i_m}{r_d} \quad b = \frac{i_m i_0}{C}. \tag{51}$$

Based on expression (51), the equation for the rotational masses coefficient of the hydraulic part of the transmission is:

$$\delta_h = \frac{i_o i_m}{m}\left[J_2 + kJ_1\left(v\frac{di_h}{dv} + i_h\right)\right]. \tag{52}$$

The analysis of expression (52) shows that the rotational masses coefficient, in the case of the HDTC, depends on the inertial parameters of the mechanical part of the transmission, and also on the transformation of the fluid energy in the HDTC.

Now the traction balance equation can finally be written (Demić & Lukić, 2011):

$$m\left(\delta_m + \delta_h\right)\frac{dv}{dt} = \frac{\eta_m \eta_o}{i_m i_o}M_r - \Sigma R. \tag{53}$$

or

$$m\left(\delta_m + \delta_h\right)\frac{dv}{dt} = \frac{M_e k i_m i_o i_r \eta_m \eta_o \eta_r}{r_d} - \Sigma R. \tag{54}$$

A system of differential equations that describes the motion of a hydrodynamic transmission system is:

$$\begin{aligned} &J_1\frac{d\omega_P}{dt} = M_r - M_P, \\ &J_2\frac{d\omega_T}{dt} = M_T - M_{opt}^T, \\ &m\left(\delta_m + \delta_h\right)\frac{dv}{dt} = \frac{\eta_m \eta_o r_d k}{i_m i_o}M_r - \Sigma R. \end{aligned} \tag{55}$$

Dynamic simulation

In order to research the reliability of the applied mathematical model of the combined operation of the ICE-HDTC, it was considered appropriate to apply it to the example of a Florida diesel engine prototype (Wong, 2001). In this case, the engine and the HDTC were used and their experimental characteristics are shown in Figures 5-7 (indicated by B).

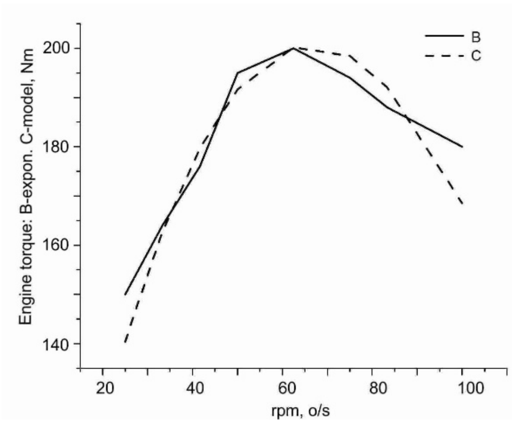

Figure 5 – Torque of the applied engine
Рис. 5 – Крутящий момент испытуемого двигателя
Слика 5 – Обртни момент посматраног мотора

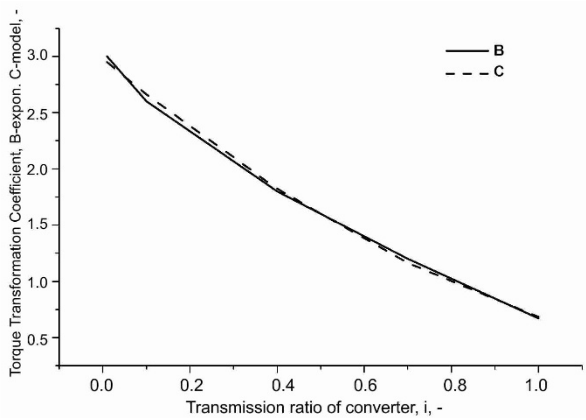

Figure 6 – Torque Transformation Coefficient
Рис. 6 – Коэффициент трансформации крутящего момента
Слика 6 – Коефицијент трансформације обртног момента

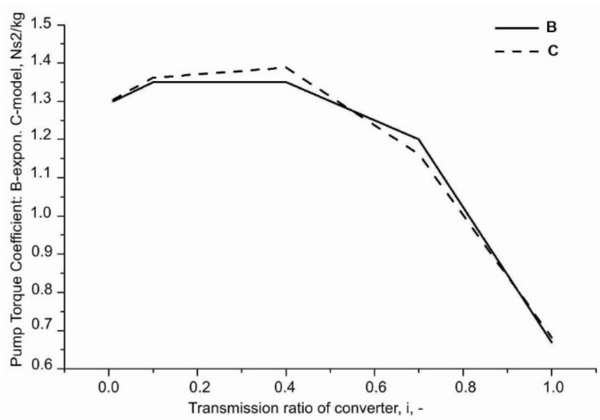

Figure 7 – Pump Torque Coefficient
Рис. 7 – Коэффициент крутящего момента насоса
Слика 7 – Коефицијент момента пумпног кола

For the purpose of easier dynamic simulation, the experimental data in Figures 5-7 were approximated by polynomials. Bearing in mind that the observed engine has regulated characteristics, its torque could not be more precisely modelled with the second-degree polynomial (Simić, 1988), so the minimum of the sum of squared errors was satisfied by using the fifth-degree polynomial. The HDTC data were approximated by the third-degree polynomial. The calculation of the unknown coefficients of polynomials was carried out using the program in Pascal, based on optimization methods.

The coefficients are shown in Tables 1-3.

Table 1 – Engine torque
Таблица 1 – Крутящий момент двигателя
Табела 1 – Обртни момент мотора

a_0	$4.337*10^{1}$
a_1	$5.064*10^{0}$
a_2	$4.351*10^{-2}$
a_3	$7.163*10^{-14}$
a_4	$6.173*10^{-7}$

Table 2 – Torque Transformation Coefficient
Таблица 2 – Коэффициент трансформации крутящего момента
Табела 2 – Коефицијент трансформације обртног момента

a_0	$1.167*10^{0}$
a_1	$1.161*10^{-1}$
a_2	$9.779*10^{-4}$

Table 3 – Coefficient of the pump torque
Таблица 3 – Коэффициент крутящего момента насоса
Табела 3 – Коефицијент момента пумпног кола

a_0	$9.720*10^{-1}$
a_1	$5.073*10^{-14}$
a_2	$3.313*10^{-5}$

The vehicle and the HDTC used in the dynamic simulation are given in Table 4 (Zastava automobili, 2008).

Table 4 – Basic data on the used vehicle and the HDTC
Таблица 4 – Основные данные об б/у автомобиле и ГДТКМ
Табела 4 – Основни подаци о коришћеном возилу и ХДТОМ-у

m	1200, kg
r_d	0.273, m
i_m	1; 1,5, -
I_0	8.5, -
i_r	0.35, -
φ	0.7, -
η_0	0.98, -
η_r	0.96, -
η_m	0.98, -
f	0.02, -
$K*A$	0.525, Ns^2/s^2
l_0	2.5, m
J_p	0.92,kgm^2
J_t	0.90, kgm^2
J_{transm}	1, kgm^2
J_{toc}	5,kgm^2
J_z	0.1, kgm^2
D	0.11, m

It is noted that the harmonization of the engine and the HDTC characteristics is achieved by changing the active diameter D of the pump wheel or the gear ratio, i_r, of the gearbox reducer, which is installed between the engine and the HDTC (Laptev, 1973), (Demić & Lukić, 2011):

$$i_r = \sqrt[3]{\frac{D^5 \lambda_P \rho \omega_e^2}{M_e \eta_r}} \tag{56}$$

where
D - active diameter of the HDTC,
M_P - torque on the pump wheel,
λ_P - torque coefficient of the pump, and
ω_P - angular velocity of the pump wheel.

In practice, the HDTC transmission ratio is chosen from the requirements for ensuring the best traction and speed characteristics of the vehicle and fuel consumption. If i_r is calculated from the conditions of the vehicle starting from a standstill, then λ_P has the value for i_h = 0. In addition, ω=(0.75÷0.85)*ω_p for motor vehicles with diesel engines, where ω_p is the angular velocity at the maximum engine power.

In this case study, the calculated value for i_r is 0.35, which means that instead of the reducer, it is necessary to use a speed multiplier gearbox.

Due to non-linearity, differential equations (55) were solved numerically using the Kutta-Merson method in the program realized in Pascal. The initial integration step was 0.001, s, and the integration was carried out at 500,000 points, which ensured the reliability of the results in the area 0.002-500, Hz (Bendat & Piersol, 2010). Bearing in mind that the aim of the simulation was to calculate the performance of a vehicle with an HDTC, this frequency range is acceptable.

It was considered appropriate to analyse the performance of the motor vehicle, and therefore a dynamic factor is calculated (Simić, 1988) according to the formula:

$$D_V = \frac{F_0 - R_v}{mg} = f\cos(u) + \sin(u) + \frac{\delta_m}{g} j. \tag{57}$$

where the labels from the preceding text, that is, the usual labels from (Simić, 1988), were used, so here they will not be specifically explained.

It was concluded by the analysis that only the results of a dynamic factor simulation in the case of 100% fuel injection in the engine and a transmission ratio of the planetary gearset 1, i.e. 75% of the fuel injection in the engine and the transmission ratio of the planetary gearset 1.5, should be shown here in Figures 8 and 9 (and not the performance of the HDTC, which is treated as a black box).

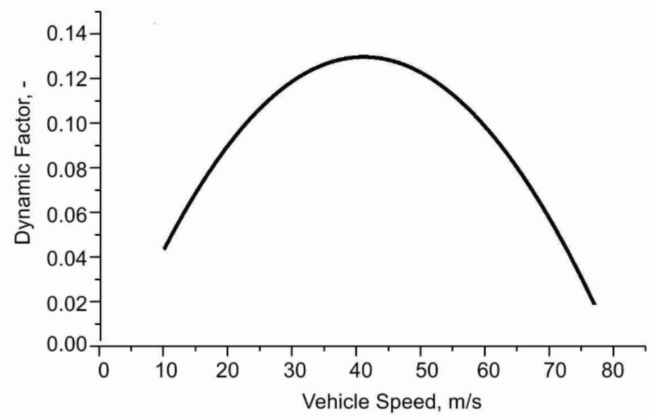

Figure 8 – Dynamic factor for 100% fuel injection in the engine and transmission ratio of the planetary gearbox equal 1
Рис. 8 – Динамический коэффициент для 100% впрыска топлива в двигатель и передаточного числа в планетарной коробке передач равен 1
Слика 8 – Динамички фактор за 100% довода горива у мотор и преносни однос у планетарном преноснику једнак 1

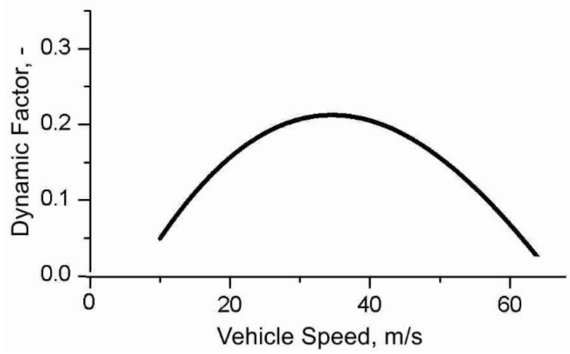

Figure 9 – Dynamic factor for 75% fuel injection into the engine and transmission ratio of the planetary gearbox equal 1.5
Рис. 9 – Динамический коэффициент для 75% впрыска топлива в двигатель и передаточного числа в планетарной коробке передач равен 1.5
Слика 9 – Динамички фактор за 75% довода горива у мотор и преносни однос у планетарном преноснику једнак 1,5

Results and analysis

Using all the calculated values of the dynamic factor, which are partially shown in Figures 8 and 9, and the procedures described in detail in the references (Demić & Lukić, 2011), (Janković & Todorović, 2001), (Simić, 1988), the maximum values of the dynamic factor and the speed at which it occurs (D_{max} and V_{Dmax}), the maximum vehicle speed (V_{max}), the maximum vehicle acceleration (j_{max}) and the maximum slope the vehicle can climb (u_{max})are calculated. The values were calculated using a program developed in Pascal, and the data are shown in Table 5.

Table 5 – Maximum values
Таблица 5 – Максимальные значения
Табела 5 – Максималне вредности

Fuel supply,%	i_m, -	D_{vmax}/V_{Dvmax}, -/m/s	v_{max}, m/s	j_{max}, m/s^2	u_{max}, %
100	1	0.129/30.97	62.47	0.413	10.9
50	1	0.088/26.54	53.34	0.257	6.8
100	1.5	0.292/25.73	54.71	0.775	27.2
75	1.5	0.212/24.59	51.76	0.547	19.2
50	1.5	0.133/22.61	46.98	0.322	11.3

Based on the data in Table 5, it can be determined that a better performance is achieved in the case of the maximum fuel supply for both transmission ratios in the planetary gearbox. With the reduction in the amount of fuel brought into the engine, the performance of the vehicle also drops. It should be noted that with the transmission ratio in the planetary gearbox equal 1, it was impossible to move the vehicle (the calculated speed of the vehicle had values below 0).

By analysing the effect of the transmission ratio in a planetary gearbox, it was found that all performance parameters are better in the case of a reduction in it, except for the maximum speed which in this case is lower. This is understandable when taking into account the facts in (Demić & Lukić, 2011), (Janković & Todorović, 2001), (Simić, 1988), which relate to the theory of the movement of motor vehicles with classical-mechanical gears.

It should be noted that some data whose values had to be approximately determined were used in this paper, such as: moments of inertia, masses of individual parts of the aggregate, etc., which inevitably leads to errors in the simulation results. Therefore, it is necessary to pay more attention in the following period to the experimental determination of the inertial parameters of the vehicle and its aggregates as a whole, or of the constituent elements. The used model vehicle (engine)-passenger makes it easier to harmonize their combine operation.

Conclusion

The developed ICE-HDTC model makes it easy to calculate and analyse the performance of motor vehicles. In the following period, attention should be paid to the experimental determination of the necessary parameters in order to improve the reliability of the simulation results. The used model makes it easier to harmonize the combined ICE - HDTC operation in variable vehicle regimes.

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ПРИЛОЖЕНИЕ К АНАЛИТИЧЕСКОМУ РАСЧЕТУ ЭФФЕКТИВНОСТИ АВТОМОБИЛЯ С ГИДРАВЛИЧЕСКИМ ТРАНСФОРМАТОРОМ КРУТЯЩЕГО МОМЕНТА

Мирослав Д. Демич[а], *Ясна* Д. Глишович[б]
[а] Академия инженерных наук, г. Белград, Республика Сербия
[б] Университет в г. Крагуевац, Инженерный факультет, Департамент автотранспортных средств и двигателей, г. Крагуевац, Республика Сербия

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Резюме:

Гидродинамические трансформаторы крутящего момента (далее именуемые ГДТКМ) становятся все более распространенными в автомобильной промышленности. Система ГДТКМ и двигатель внутреннего сгорания (ДВС) рассматриваются как агрегат, выходные характеристики которого однозначно определяются характеристиками двигателя и соответствующим ГДТКМ. А именно, двигатель и гидротрансформатор можно рассматривать как единую систему с точно определенным выходным крутящим моментом и угловой скоростью на выходе. Таким образом создается возможность расчета движущей силы так, как это делается при расчете механической трансмиссии, входной сигнал в планетарную коробку передач которой (если есть) является выходным крутящим моментом гидротрансформатора. Согласование совместной работы ДВС – ГДТКМ является весьма сложным процессом. В данной статье была сделана попытка представить моделирование вышеописанной системы, с целью упрощения процесса согласования характеристик совместной работы двигателя внутреннего сгорания и ГДТКМ.

Ключевые слова: транспортные средства, двигатели, гидродинамический, гидротрансформатор, производительность.

ПРИЛОГ АНАЛИТИЧКОМ ИЗРАЧУНАВАЊУ ПЕРФОРМАНСИ МОТОРНОГ ВОЗИЛА СА УГРАЂЕНИМ ХИДРАУЛИЧКИМ ТРАНСФОРМАТОРОМ ОБРТНОГ МОМЕНТА

Мирослав Д. Демић[а], *Јасна* Д. Глишовић[б]
[а] Академија инжењерских наука, Београд, Република Србија
[б] Универзитет у Крагујевцу, Факултет инжењерских наука, Катедра за моторна возила и моторе, Крагујевац, Република Србија

ОБЛАСТ: машинство, моторна возила и мотори
ВРСТА ЧЛАНКА: стручни рад
ЈЕЗИК ЧЛАНКА: енглески

Сажетак:

Хидродинамички трансформатори обртног момента (ХДТОМ) налазе све ширу примену код моторних возила. Систем мотор са унутрашњим сагоревањем (МСУС)–ХДТОМ може се посматрати као један агрегат, чије су излазне карактеристике једнозначно дефинисане карактеристикама мотора и одговарајућег ХДТОМ-а. Наиме, мотор и хидротрансформатор могу се посматрати као један систем, са тачно дефинисаним излазним моментом и излазном угаоном брзином. То ствара могућност да се погонска сила може израчунати као да се ради о механичкој трансмисији, чији је улаз у планетарни преносник (ако постоји) излазни момент хидротрансформатора. Усаглашавање заједничког рада МСУС–ХДТОМ представља сложен процес. У овом раду настојано је да се поменути систем моделира, како би се што једноставније могле усагласити карактеристике заједничког рада мотора са унутрашњим сагоревањем и ХДТОМ-а.

Кључне речи: возила, мотори, хидродинамички, трансформатор обртног момента, перформансе.

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