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ALGORITHM DETERMINING THE COMMON OPERATION CHARACTERISTIC OF THE ENGINE-HYDRODYNAMIC TORQUE CONVERTER SYSTEM

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Abstract:
The paper enhances the method used in order to obtain the characteristic of the engine-hydrodynamic torque converter system common operation and the system output characteristic. By undertaking these characteristics can be elaborated an algorithm to analytically determine the traction characteristic that emphasises the vehicles economical and traction performances (slipping, traction power, velocity, hourly and specific fuel consumption according to the traction force). There are presented the specific results for a Diesel engine of 129 kW and a 15″ hydrodynamic torque converter. The paper is useful for the design of vehicles with hydrodynamic torque converter and also for the design of engine-hydrodynamic torque converter systems.

The characteristic of the engine-hydrodynamic torque converter system common operation means the superposing in the same graphic of the following two characteristics: engine's $M_e = f(n)$ and pump's $M_P = \varphi(n_P)$, engine speed n and pump speed n_p being equal (fig.1).

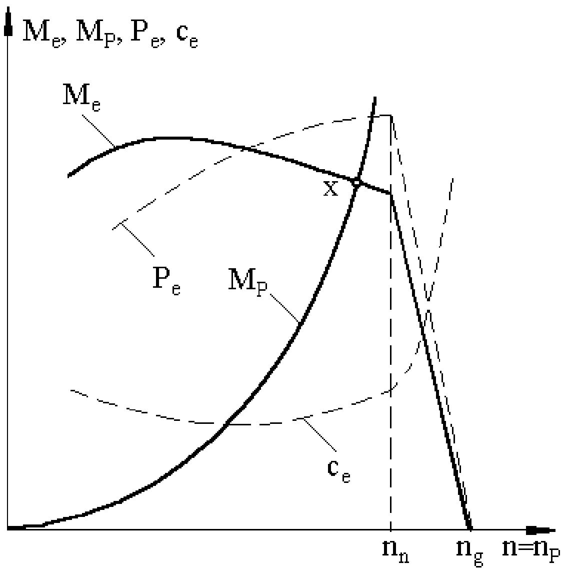

Fig. 1. Characteristic of the engine-hydrodynamic torque converter system common operation – THD

In order to outline this characteristic there is separately analysed:

a) The characteristic of the internal-combustion engine, which is traced within the co-ordinates system M_e, n, where M_e stands for the crankshaft moment of torque (pump's shaft) and n = n_p – stands for the engine/pump speed (there is considered the case of the engine-hydrodynamic torque converter systems without adaptive reductor). Very often, in order to have a complete image of the common operation, there are also traced the curves of both the effective power P_e and the specific fuel consumption $\boldsymbol{c_e}$

As regards the Diesel engines, the moment of torque variation is quite accurately described by the relation:

$$M_e = \begin{cases} M_n\left[\alpha_1 + \alpha_2 \dfrac{n}{n_n} + \alpha_3\left(\dfrac{n}{n_n}\right)^2\right], & \text{if} \quad n < n_n; \\ M_n \dfrac{n_g - n}{n_g - n_n}, & \text{if} \quad n \ge n_n; \end{cases} \tag{1}$$

$$\alpha_1 = \frac{c_1^2 - c_2(2c_1 - 1)}{(c_1 - 1)^2}; \quad \alpha_2 = \frac{2c_1(c_2 - 1)}{(c_1 - 1)^2}; \quad \alpha_3 = \frac{1 - c_2}{(c_1 - 1)^2}; \quad c_1 = \frac{n_m}{n_n}; \quad c_2 = \frac{M_m}{M_n},$$

where:

n_n – stands for the engine nominal speed;
n_m – stands for the engine speed at its maximum torque;
M_n – stands for the engine nominal torque;
M_m – stands for the engine maximum torque.

b) The adimensional characteristic of the hydrodynamic torque converter according to the kinematic transfer ratio, which is defined as ratio of the turbine and pump speed ($i = n_T / n_P$); this characteristic comprises the following curves (fig.2):

- The proportionality coefficient of the pump torque, $\lambda_P = f_1(i)$;
- The conversion coefficient of the hydrodynamic torque converter (ratio of the output torque M_T and input torque M_P), $K = M_T / M_P = f_2(i)$;
- The hydrodynamic torque converter efficiency (ratio of the output power P_T and the input torque P_P), $\eta = P_T / P_P = K \cdot i = f_3(i)$.

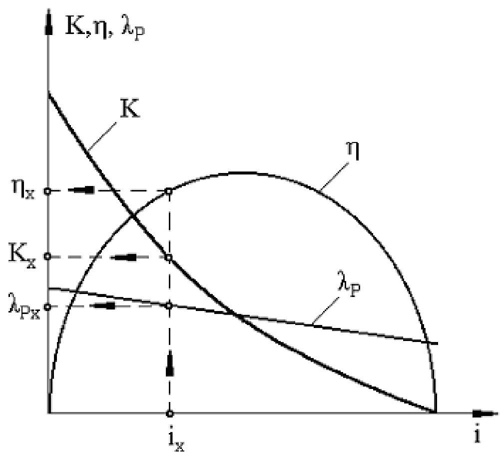

Fig. 2. Adimensional characteristic of the complex engine-hydrodynamic torque converter system.

In order to emphasise the load characteristics of the engine-hydrodynamic torque converter system, there is outlined its input load representing the dependency between the pump torque and the speed ($M_P = f(n_P)$) for i = constant. This characteristic is traced for a range of values of the conversion ratio, for example: i = 0; 0.1; 0.2;…i_{max}.

There is used the relation

$$M_P = \gamma \lambda_P n_P{}^2 D^5 \text{ [Nm]}, \tag{2}$$

where:

γ – stands for the specific weight of the motive fluid, in în N/m^3;
λ_P – stands for the proportionality coefficient of the pump moment, in m^{-1}. min^2;

n_P – stands for the pump speed, in rot/min

D – stands for the active diameter of the engine-hydrodynamic torque converter system

The input characteristic (or the load characteristic) of the engine-hydrodynamic torque converter system may be outlined following the succession:

- There is adopted a value for i, for example i_x (see fig. 2);
- The graphic λ_P = f(i) reveals the corresponding value of the λ_P coefficient (in the example considered λ_{Px});
- There is calculated the constant A corresponding to the ratio adopted $i = i_x$: $A = \gamma\lambda_P D^5$ and $A_x = \gamma\lambda_{Px} D^5$;
- There is traced the parabola $M_P = A n_P{}^2$, and $M_{Px} = A_x n_P{}^2$;
- In an analogue way there is traced the parabola for other values of the conversion ratio i (fig. 4).

Within the present paper there was studied and analysed the engine-hydrodynamic torque converter system as part of a Diesel engine of 129 kW and a hydrodynamic torque converter with a 15″ active diameter.

The main parameters of this engine in the speed characteristic are: M_n = 560 Nm – nominal coupling; M_m = 825 Nm – maximum coupling; n_n = 2200 rot/min – nominal speed; n_m = 1400 rot/min – speed corresponding to the maximum coupling; n_g = 2400 rot/min – idle running maximum speed.

By making use of the polynomial regression, there were obtained the following functions for the curves that form the adimensional characteristic of the engine-hydrodynamic torque converter system:

$$\begin{aligned} &K(i) = -1{,}32i^3 + 2{,}128i^2 - 3.554i + 3{,}131; \\ &\eta(i) = K(i)\cdot i; \qquad (3) \\ &\lambda_P(i) = \left(-4{,}485i^3 + 3{,}437i^2 + 0{,}306i + 1{,}836\right)10^{-6}. \end{aligned}$$

These curves were traced in figure 3.

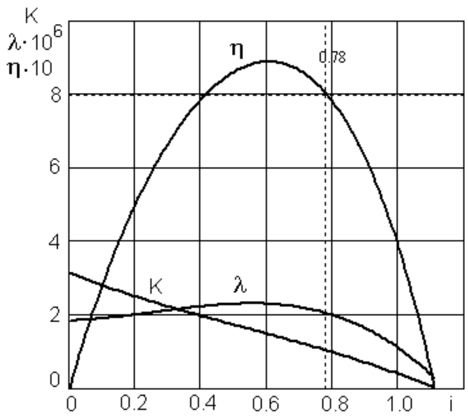

Fig. 3. Adimensional characteristic of the engine-hydrodynamic torque converter system submitted to analysis.

All the characteristics considered in the paper were calculated for 221 values of the ratio i_j (j = 0, 1, 2, …220). The conversion ratio i_j is related to the following values:

$$A_j = \lambda_j \gamma D^5; \quad M_{P\,j} = A_j n^2.$$

As it is known, the characteristic of the engine-hydrodynamic torque converter system common operation is traced for a diminished engine torque: $M'_e = aM_e$ (where a = 0.9…1). As expressed in the paper $M'_e = 0.93M_e$. There is considered solely the power passing through the engine-hydrodynamic torque converter system.

The common operation points (of intersection between the M'_e and M_{Pj} curves) are determined by using, for example, the function root of MathCAD. The figure 4 presents the characteristic of the engine-hydrodynamic torque converter system common operation for the 10 common functioning points.

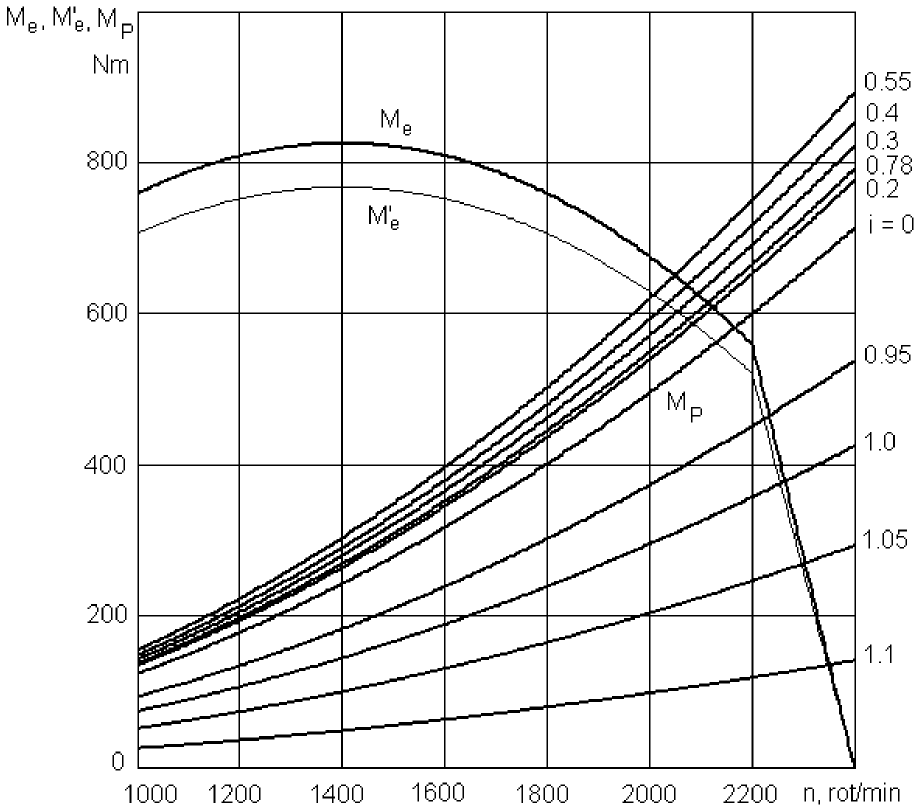

Fig. 4. The characteristic of the engine-hydrodynamic torque converter system common operation.

A common operation point i_j has the co-ordinates (n_j, M_e'), and (n_{Pj}, M_{Pj}). At the output, at the turbine shaft, this point is given the parameters:

– speed: $n_{Tj} = n_j \cdot i_j$ [rot/min];

– torque: $M_{Tj} = M_{Pj} \cdot K_j$ [Nm];

– power: $P_{Tj} = \dfrac{n_{Tj} \cdot M_{Tj}}{9550}$ [kW];

– system efficiency: $\eta_s = \dfrac{P_T}{P_n}$, if $P_e = P_n = \text{const}$.

In figure 5 there is presented the output characteristic of the engine-hydrodynamic torque converter system.

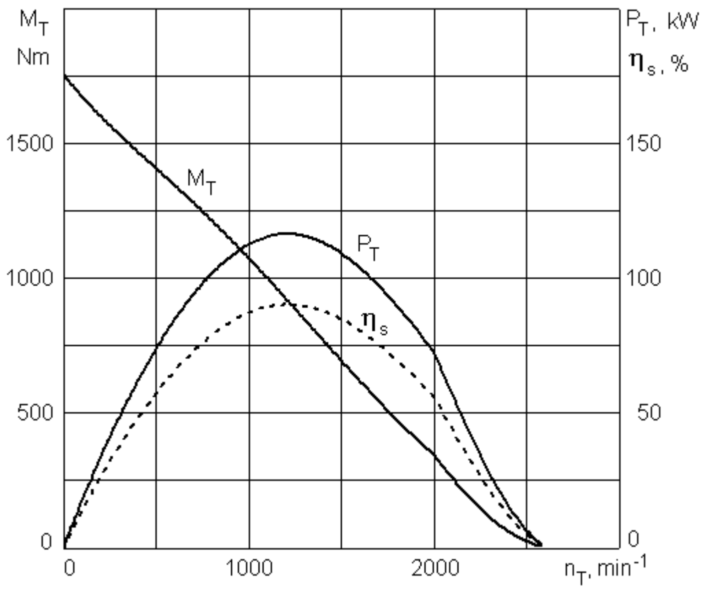

Fig. 5. Output characteristic of the engine-hydrodynamic torque converter system.

It is commonly known that vehicles using these systems have an ideal traction characteristic provided the condition $M_T \cdot n_T$ = constant is met, that is the variation of the output torque M_T is a equiangular hyperbola taking into account the shaft speed n_T. Although the characteristic of the system submitted to analysis is not the ideal one, it is recommended for any traction vehicle due to the high degree of adaptability. Nevertheless, these systems are efficient in a relatively reduced rotational speed interval.

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