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CALCULATION OF THE STARTING REGIME OF THE POWER TRANSMISSION SYSTEM WITH A HYDRODYNAMIC COUPLING AND A DRIVING MOTOR

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Abstract. *In this paper a grapho-analytical procedure for calculating a starting regime of the power transmission with a hydrodynamic coupling and a driving motor is presented. The presented iterative procedure for solving this problem provides for obtaining the starting power transmission system regime with accuracy determined in advance. The number of iterative steps depends on the defined accuracy.*

Key Words: *Hydrodynamic Coupling, Electromotor, Starting Regime*

1. INTRODUCTION

The layout of the power transmission with a hydrodynamic coupling is shown in Fig. 1. In general, between the driving motor (M) and the hydrodynamic coupling (HDC) a mechanical gear (mm1) can be installed; also, between the hydrodynamic coupling and the driven mechanism a mechanical gear (mm2) can be installed.

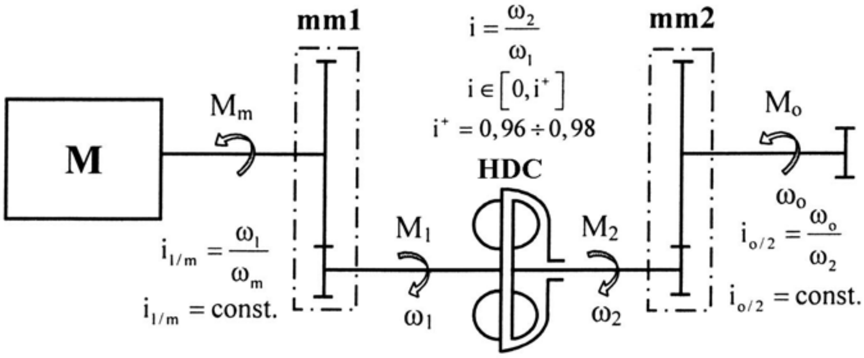

Fig. 1.

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Denotations in Fig. 1 and further in the text:

M_m, ω_m – torque and angular velocity of the driving motor shaft,

M_1, ω_1 – torque and angular velocity of input shaft of the hydrodynamic coupling,

M_2, ω_2 – torque and angular velocity of output shaft of the hydrodynamic coupling,

M_o, ω_o – torque and angular velocity of output shaft of the power transmission,

$i_{1/m} = \omega_1 / \omega_m$ – transmission ratio in mechanical gear **mm1**,

$i = \omega_2 / \omega_1$ – transmission ratio in hydrodynamic coupling,

$i_{0/2} = \omega_0 / \omega_2$ – transmission ratio in mechanical gear **mm2**.

The parameters of the common operation of both the driving motor and the driven unit are determined from the dynamic balance requirement for all components in the transmission chain. Therefore, in addition to the torque characteristics of the driving motor and the load in the output shaft of the power transmission, it is necessary to know coefficients of the torque transformation and angular velocities in all the components of the power transmission. Likewise, for transient operating regimes it has to be known inertia moment of all the rotating masses in the transmission chain.

The driving motor torque, in general, depends on angular velocity (ω_m) and magnitude of the regulation parameter of motor (α_m); $M_m = M_m(\omega_m,\alpha_m)$ – torque characteristic of the motor, which is given by manufacturer, usually as a diagram. For unregulated electromotors, as squirrel cage asynchronous motors, that are often used, and are the subject of this paper, the electromotor's torque characteristic is unambiguous function $M_m(\omega_m)$, given by diagram.

The output shaft torque of power transmission (M_o) can be given as a function of angular velocity of this shaft, $M_o = M_o(\omega_o)$, therewith a shape of this curve (function) during the time (i.e. operating conditions) is changeable.

The dimensionless torque coefficient, transmitted by the hydrodynamic coupling, is defined as:

$$\lambda_M = \frac{M}{\rho R^5 \omega_1^2}\,, \tag{1}$$

where: ρ-density of working liquid, R – maximum internal (circulating) radius of the hydrodynamic coupling, M – torque of the hydrodynamic coupling and ω_1 – angular velocity of the input shaft of the hydrodynamic coupling.

The dimensionless torque coefficient of the hydrodynamic coupling is functional dependence $\lambda_M(i)$, usually given by manufacturer as a diagram or a chart.

The nominal transmission ratio of the hydrodynamic coupling ($i = i^+$) is transmission ratio at maximum efficiency, where $i^+ = 0{,}96 \div 0{,}98$ (0,97).

Efficiency of the hydrodynamic coupling is given as:

$$\eta_{HDS} = i, \text{ for } i \in [0,i^+],$$

and for $i \in [i^+,1]$ efficiency decreases rapidly, so as with $i = 1$ normal power transmission through the coupling is terminated.

In the cases when the flows through the hydrodynamic coupling are auto-modeled in the Reynolds number (for fluid flows where the friction factor does not depend on the Reynolds number), dimensionless torque coefficient $\lambda_M(i)$ does not depend on angular velocity of the coupling's input shaft. According to the data given in the References [2], the fluid flow in the hydraulic coupling can be considered auto-modeled in the Reynolds number if:

$$\omega_1 \geq \frac{0,875 \cdot \nu \cdot 10^5}{R^2}, \tag{2}$$

where ν – kinematic viscosity of the operating fluid [m^2/s].

Using dimensionless $\lambda_M(i)$ characteristic of the hydrodynamic coupling, according to equation (1), a spectrum of parabolas can be defined (drawn) in $M\!-\!\omega_1$ diagram of characteristics

$$M(i) = K\lambda_M(i)\omega_1^2, \quad (K = \rho R^5 = \text{const.}), \quad \text{for } i = \text{const.}, \tag{3}$$

which represent curves of torque transmitted by the hydrodynamic coupling with different values of transmission ratio (for i = const., $i \in [0,1]$).

According to the index in Fig. 1, the torque of the output shafts of mechanical gears **mm1** and **mm2** can be calculated, in steady-state operating regime, by using the following formula:

$$M_1 = \frac{\eta_{mm1}}{i_{1/m}} M_m \quad \text{and} \quad M_o = \frac{\eta_{mm2}}{i_{0/2}} M_2, \quad \text{for } \omega = \text{const.}, \tag{4}$$

where η_{mm1} and η_{mm2} are efficiencies of mechanical gears.

In the steady-state operating system regime, torque and angular velocity of motor, reduced on input shaft of the hydrodynamic coupling, is changing by the law:

$$M_1 = M_{1.m} = \frac{\eta_{mm1}}{i_{1/m}} M_m, \quad \omega_1 = i_{1/m} \cdot \omega_m \quad \text{for } \omega = \text{const.}, \tag{5}$$

Using equations (5), torque characteristic of motor $M_m(\omega_m)$, in steady-state operating regime of the system, can be mapped into the torque characteristic of motor reduced on input shaft of the hydrodynamic coupling $M_{1.m}(\omega_1)$ (sl.2a).

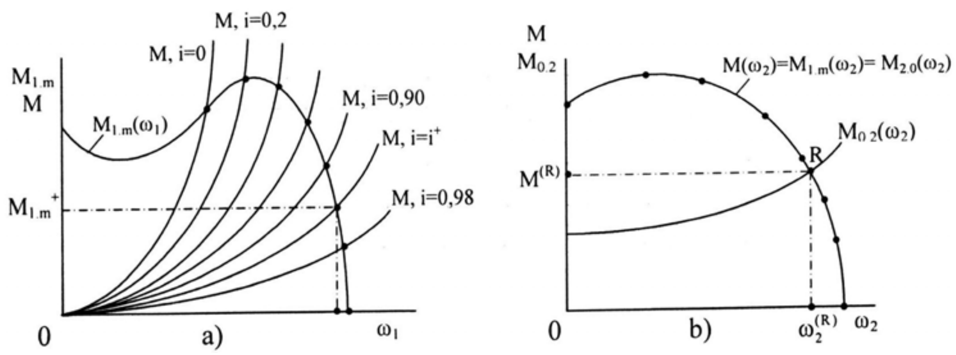

Fig. 2.

In steady-state operating regime of the system, torque transmitted by the hydrodynamic coupling, is equal to the torque of input shaft of the coupling, therefore, the common operating regimes of the motor and the hydrodynamic coupling, for different values of transmission ratio, are defined by intersection points of reduced torque characteristic $M_{1.m}(\omega_1)$ and parabolas defined by the equation (3), as is shown in Fig. 2a.

Since:

$$M(\omega_1) = M(\omega_2), \quad \omega_2 = i \cdot \omega_1, \quad \text{for } \omega = \text{const.}, \tag{6}$$

points of steady-state operating regime of the hydrodynamic coupling from $M(\omega_1) = M_{1.m}(\omega_1)$ characteristic in Fig. 2a can be mapped into $M(\omega_2)$ characteristic (Fig. 2b).

Loading torque M_o and angular velocity of the output shaft of the power transmission, reduced on output shaft of the hydrodynamic coupling, are changing by the law:

$$M_2 = M_{2.0} = \frac{i_{0/2}}{\eta_{mm2}} M_0, \quad \omega_0 = i_{0/2} \cdot \omega_2 . \tag{7}$$

The loading torque of the output shaft of the power transmission can be given by analytic function $M_2(\omega_2)$, and using equation (7) the graph of this function can be mapped into a curve of the loading torque reduced on the output shaft of the hydrodynamic coupling, $M_{2.0}(\omega_2)$, as is shown in Fig. 2b for one form of loading.

In steady-state operating regime of the system, the operating regime of the hydrodynamic coupling is defined by intersection point of function graph of loading torque reduced on the output shaft of the hydrodynamic coupling $M_{2.0}(\omega_2)$ and graph of the torque characteristic of common operating of hydrodynamic coupling and motor $M(\omega_2)$, represented in Fig. 2b with point R. The torque that is transmitted by the hydrodynamic coupling in this operating regime is illustrated as $M^{(R)}$, and angular velocity of the output shaft of the coupling as $\omega_2^{(R)}$. The number of revolutions of the input shaft, in this operating regime ($\omega_1^{(R)}$), is equal to the number of revolutions on characteristic $M_{1.m}(\omega_1)$ (Fig. 2a) that corresponds to the torque $M_{1.m} = M^{(R)}$. The transmission ratio of the coupling is $i^{(R)} = \omega_2^{(R)} / \omega_1^{(R)}$.

Knowing operating parameters of the hydrodynamic coupling ($M^{(R)}$, $\omega_2^{(R)}$, $\omega_1^{(R)}$, $i^{(R)}$), in steady-state operating regime of system, torque and angular velocity of the motor are calculated by using equations (5):

$$M_m^{(R)} = \frac{i_{1/m}}{\eta_{mm1}} M^{(R)}, \omega_m^{(R)} = \frac{\omega_1^{(R)}}{i_{1/m}} .$$

Point R_m ($\omega_m^{(R)}, M_m^{(R)}$) must be on graph $M_m(\omega_m)$ of the motor characteristic, what is also a procedure of accuracy of grapho-analytic method used for solving this problem.

For known $\omega_2^{(R)}$, using the second equation of equations (7), can be also calculated $\omega_0^{(R)} = i_{0/2} \cdot \omega_2^{(R)}$.

2. EQUATIONS OF DYNAMIC BALANCE FOR STARTING OPERATING REGIMES OF THE SYSTEM

The equation of dynamic balance for transmission chain, from the driving motor to the pump impeller output of the hydrodynamic coupling, can be written as:

$$M = M_{1.m} - I^{(r)}_{m.1} \frac{d\omega_1}{dt}, \tag{8}$$

where: M – torque transmitted by the coupling, $M_{1.m}$ – torque of the driving motor (according to equation (5)), reduced on input shaft of the coupling, $I^{(r)}_{m.1}$ - inertia moment of rotating masses in this transmission chain, reduced on the input shaft of the hydrodynamic coupling. For transmission chain shown in Fig. 1, it can be written as:

$$I^{(r)}_{m.1} = \frac{\eta_{mm1}}{i^2_{1/m}} I_m + I_1, \tag{8'}$$

where: I_m – inertia moment of the driving motor shaft and all rotating masses attached (rotor of the electro-motor, operating gear **mm1**), I_1 – inertia moment of the input shaft of the hydrodynamic coupling and all rotating masses attached, including operating fluid in the pump circuit of the hydrodynamic coupling.

The equation of dynamic balance for transmission chain from the turbine circuit of the hydrodynamic coupling to the shaft of the driven mechanism can be presented in a form:

$$M = M_{2.0} + I^{(r)}_{2.0} \frac{d\omega_2}{dt}, \tag{9}$$

where $M_{2.0}$ – loading torque (according to equation (7)), reduced on output shaft of the hydrodynamic coupling, and $I^{(r)}_{2.0}$ - inertia moment of rotating masses in this transmission chain. For transmission chain shown in Fig. 1 it is:

$$I^{(r)}_{2.0} = I_2 + \frac{i^2_{0/2}}{\eta_{mm2}} I_0, \tag{9'}$$

where: I_2 – inertia moment of output torque of the hydrodynamic coupling and all rotating masses attached, including operating fluid in turbine circuit of the hydrodynamic coupling, I_0 – inertia moment of driven mechanism shaft and all rotating masses attached.

Eliminating M from equations (8) and (9), an equation of dynamic balance for all transmission chain is obtained as:

$$M_{1.m} = M_{2.0} + I^{(r)}_{m.1} \frac{d\omega_1}{dt} + I^{(r)}_{2.0} \frac{d\omega_2}{dt}. \tag{10}$$

In steady-state operating regime $M_{1.m} = M_{2.0}$, so therefore, according to the equation (10), it can be concluded that at the end of the starting regime, acceleration of both shafts of the hydrodynamic coupling (input and output shaft) are gravitating to the zero value**.**

To determine the functional relation between angular velocities of input and output shaft of the hydrodynamic coupling ($\omega_1(\omega_2)$) in starting operating regime of the system, it is necessary to know the universal characteristic of the coupling, that is represented by graph spectrum of torque characteristics $M(\omega_2)$ transmitted by the coupling with different angular velocity of input shaft of the coupling (Fig. 3a). The dash-point line in Fig. 3a represents a graph of function $M_{1.m}(\omega_2)$.

If starting operating regime has a very low value of acceleration ("quasi-static"), according to the intersection points of graphs $M_{2.0}(\omega_2)$ and $M(\omega_2)$, for ω_1=var., in the universal characteristic of the hydrodynamic coupling (Fig. 3a), functional relation $\omega_1(\omega_2)$ could be determined, and its graph is shown in Fig. 3b as a full line. Point R in Fig. 3a represents a steady-state operating regime.

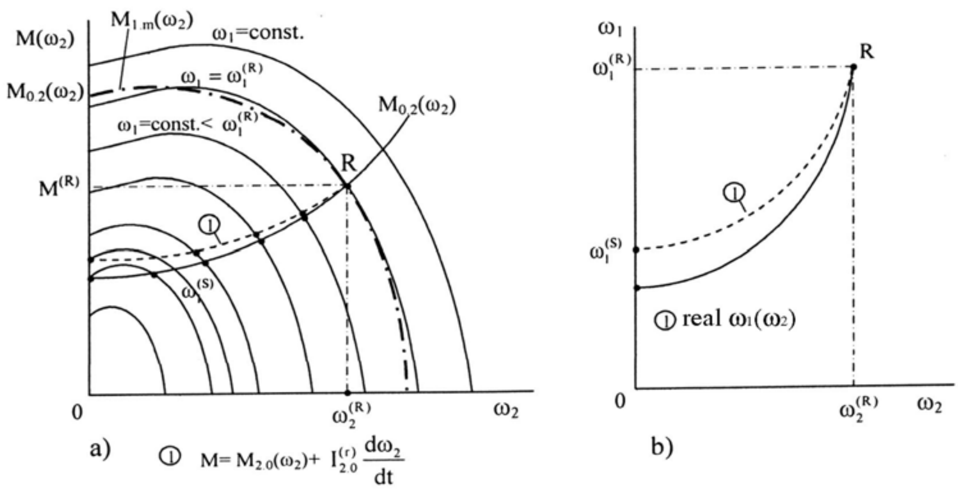

Fig. 3.

For a real problem, as is described with equation (9), because of inertia of the transmission chain from the hydrodynamic coupling to the driven mechanism, relation $\omega_1(\omega_2)$ can be determined according to the intersection points of function diagram $M_2(\omega_2)$, on the universal characteristic of the coupling, (for different values of ω_1) and functional diagram $M_{2.0} + I_{2.0}^{(r)} d\omega_2 / dt$, what is shown in Fig. 3a as curve 1. Graph 1 ($\omega_2 = \omega_2^{(R)}$) in point R is becoming graph $M_{2.0}(\omega_2)$, but the law of change $d\omega_2 / dt = f(\omega_2)$ is unknown; therefore, the law of change $\omega_1(\omega_2)$ has to be determined by iterative procedure.

Assume that the fluid flow in the hydrodynamic coupling is auto-modeled in the Reynolds number, using equations (9) and (3), for $\omega_2 = \omega_2^{(s)}$, $\omega_2 = 0$ ($i = 0$), we obtain a formula for calculating angular velocity of input shaft of the hydrodynamic coupling at the moment when its output shaft has been started:

$$\omega_1^{(s)} = \sqrt{\frac{M_{2.0}(\omega_2 = 0) + I_{2.0}\left(\frac{d\omega_2}{dt}\right)_{\omega_1 = \omega_1^{(s)}}}{\rho \lambda_M (i = 0) \cdot R^5}}\ , \tag{11}$$

where $(d\omega_2 / dt)_{\omega_1=\omega_1^{(s)}}$ – acceleration of output shaft of the hydrodynamic coupling in the moment of starting.

Since starting the input shaft of the hydrodynamic coupling cannot take place until the input shaft reaches a suitable number of revolutions ($\omega_1 = \omega_1^{(s)}$), the system's starting regime can be divided into two phases:

1. starting phase until output shaft of the hydrodynamic coupling starts rotating, and,
2. starting phase from the moment of starting output shaft of the hydrodynamic coupling until a steady-state operating regime is established.

Starting regime until the output shaft of the hydrodynamic coupling starts rotating (t_1) can be determined by solving differential equation (8). For $M = M(i = 0) = \rho\lambda_M(i = 0) \cdot R^5 \cdot \omega_1^2$, with changing angular velocity ω_1 from 0 to $\omega_1^{(s)}$:

$$t_1 = I_{m.p}^{(r)} \int_0^{\omega_1^{(s)}} \frac{d\omega_1}{f(\omega_1)}, \text{ where } f(\omega_1) = M_{1.m}(\omega_1) - \rho\lambda_M(i=0) \cdot R^5 \cdot \omega_1^2 \tag{12}$$

Discrete values of function $f(\omega_1)$, that is being used in numerical solving integral (12), are determined as difference between graph ordinate of reduced $M_{1.m}(\omega_1)$ motor characteristic and graph of torque parabola transmitted by coupling with $i = 0$ ($\omega_2 = 0$).

Acceleration of output shaft of the hydrodynamic coupling at the end of the first starting phase, i.e. at the beginning of the second starting phase, is:

$$\left.\frac{d\omega_1}{dt}\right|_{\omega_1=\omega_1^{(s)}} = \frac{1}{I_{m.1}^{(r)}}[M_{1.m}(\omega_1 = \omega_1^{(s)}) - \rho\lambda_M(i=0) \cdot R^5 (\omega_1^{(s)})^2],$$

therefore, according to the equation (10), we obtain an acceleration of output shaft of the hydrodynamic coupling at the beginning of the second starting phase:

$$\begin{aligned}\left.\frac{d\omega_2}{dt}\right|_{\substack{\omega_2=0\\ \omega_1=\omega_1^{(s)}}} &= \frac{1}{I_{2.0}^{(r)}}[M_{1.m}(\omega_1^{(s)}) - M_{2.0}(\omega_2 = 0) - I_{m.1}^{(r)}(\frac{d\omega_1}{dt})_{\omega_1=\omega_1^{(s)}}] = \\ &= \frac{1}{I_{2.0}^{(r)}}[\rho\lambda_M(i=0) \cdot R^5 (\omega_1^{(s)})^2 - M_{2.0}(\omega_2 = 0)].\end{aligned} \tag{13}$$

Knowing an acceleration of output shaft of the hydrodynamic coupling at the beginning of the second starting phase, we can obtain $\omega_1^{(s)}$ in the first approximation, and then, also, the starting regime of the first starting phase. The iterative procedure is completed when the difference between $\omega_1^{(s)}$ and its previous approximation is negligibly low (for instance, less than 1%).

Considering $\omega_1 = \omega_1(\omega_2)$, we can also write: $\frac{d\omega}{dt} = \frac{d\omega}{d\omega}\frac{d\omega}{dt}$,

therefore, differential equation (10), which is used for calculating a starting regime of the second starting phase (t_2), can be written in this form:

$$\left.\begin{aligned} & dt = F(\omega_2)\cdot d\omega_2 , \\ \text{where:} \quad & F(\omega_2) = \frac{I_{2.0}^{(r)} + I_{m.p}^{(r)} \dfrac{d\omega_1}{d\omega_2}}{M_{1.m}(\omega_1(\omega_2)) - M_{2.0}(\omega_2)} \end{aligned}\right\} \qquad (14)$$

In the moment of obtaining a steady-state operating regime ($d\omega_1/dt = 0$, $d\omega_2/dt = 0$), when $\omega_1 = \omega_1^{(R)}$ and $\omega_2 = \omega_2^{(R)}$, function $F(\omega_2)$ achieved infinite value, therefore, in numerical integration of differential equation (14), this equation integrates with negligible error from ω_2=0 to $\omega_2 = 0,99\omega_2^{(R)}$:

$$t_2 = \int_0^{0,99\omega_2^{(R)}} F(\omega_2)\cdot d\omega_2 \ . \qquad (14')$$

To obtain subintegral function F (ω_2), it is necessary to determine a graph of function $\omega_1(\omega_2)$, and after that, a graph of functions $d\omega_1/d\omega_2 = f_1(\omega_2)$ and $\varphi(\omega_2) = M_{1.m}(\omega_1(\omega_2)) - M_{2.0}(\omega_2)$.

Graph of function $\omega_1(\omega_2)$ is interpolated according to the intersection points of torque graph $M(\omega_1,\omega_2)$ in the universal characteristic of hydrodynamic coupling, and graph of load torque characteristics of output shaft of the hydrodynamic coupling.

$$M(\omega_1, \omega_2) = M_{2.0}(\omega_2) + I_{2.0}^{(r)} \frac{d\omega_2}{dt} , \qquad (15)$$

Acceleration of output shaft of the hydrodynamic coupling at the end of the first starting phase – at the beginning of the second starting phase, is known ($(d\omega_2/dt)_{\omega_1=\omega_1^{(s)}}$); therefore in the first iterative step for solving the second starting phase, it can be assumed that a linear changing of $d\omega_2/dt$, and equation (15) becomes:

$$M(\omega_1, \omega_2) = M_{2.0}(\omega_2) + I_{2.0}^{(r)} \left(\frac{d\omega_2}{dt}\right)_{\omega_1=\omega_1^{(s)}} \frac{\omega_2^{(R)} - \omega_2}{\omega_2^{(R)}} , \qquad (15')$$

Drawing, in the universal characteristic of the coupling, a graph of function described by the right side of equation (15') (curve 1 in Fig. 3a), according to the intersection points of this graphic and graphs of torque characteristics $M(\omega_1, \omega_2)$, the graphic of function $\omega_1(\omega_2)$ can also be interpolated (curve 1 in Fig. 3b). According to graph $\omega_1(\omega_2)$, it is easy to interpolate also graph $f_1(\omega_2)=d\omega_1/d\omega_2$ ($f_1(\omega_2)=tg\alpha(\omega_2)$ – angle (ω_2) of the inclination of a straight line which contacts graph (ω_2), for variable ω_2 measured toward abscissa).

Graph of function ω_1

$$\varphi(\omega_2) = M_{1.m}(\omega_2) - M_{2.0}(\omega_2) , \qquad (M_{1.m}(\omega_2) = M_{1.m}(\omega_1(\omega_2))) ,$$

which is in the denominator of subintegral function $F(\omega_2)$, can be easily interpolated according to the serial, previously defined, differences between ordinates of graphs $M_{1.m}(\omega_2)$ and $M_{2.0}(\omega_2)$, shown in Fig. 3a.

Considering all that has been defined so far, it is not difficult to do the interpolation of subintegral function $F(\omega_2)$and, subsequently, to solve numerical integral (14'). In this way, the first approximation of starting regime of the second starting phase (t_2) is obtained.

By solving differential equation (14), for different discrete values of ω_2, from interval $\omega_2 \in (0, 0,99\omega_2^{(R)})$,

$$t_2 = \int_0^{\omega_2} F(\omega_2)\cdot d\omega_2 \ , \quad \text{for} \quad \omega_2 \in (0, 0,99\omega_2^{(R)}) \ , \tag{16}$$

therefore, knowing functional relation $\omega_1=\omega_1(\omega_2)$, graphs $\omega_2(t)$ and $\omega_1(t)$ can be interpolated, and accordingly, also:

$$\psi_1(t) = \frac{d\omega_1(t)}{dt}, \quad \psi_2(t) = \frac{d\omega_2(t)}{dt} \ , \tag{17}$$

and

$$\varphi(t) = M_{1.m}(\omega_2(t)) - M_{2.0}(\omega_2(t)) \ , \tag{18}$$

According to equation (10), written in the form:

$$M_{1.m} - M_{2.0} = I_{m.p}^{(r)}\frac{d\omega_1}{dt} + I_{2.0}^{(r)}\frac{d\omega_2}{dt} \ ,$$

It is obtained that at every moment (t) of the second starting phase the next equation must be satisfied:

$$\left.\begin{aligned} \varphi(t) &= \psi(t) \\ \psi(t) &= I_{m.p}^{(r)}\cdot\psi_1(t) + I_{2.0}^{(r)}\cdot\psi_2(t) \end{aligned}\right\} , \tag{19}$$

To verify validity of functional relation $\omega_1(\omega_2)$, i.e. validity of assumed functional graph $M_{2.0}(\omega_2) + I_{2.0}^{(r)} d\omega_2 / dt$ (curve 1 in Fig. 3a), functional difference $\varphi(t) - \psi(t)$ is determined, and based on that difference also $\varphi(\omega_2) - \psi(\omega_2)$. Value of functional difference between $\varphi(\omega_2)$ and $\psi(\omega_2)$ shows us a method of correction for functional graph $M_{2.0}(\omega_2) + I_{2.0}^{(r)} d\omega_2 / dt$ (curve 1 in Fig. 3a). When is $\varphi(\omega_2) > \psi(\omega_2)$, the curve 1 has to be further from curve $M_{2.0}(\omega_2)$, and when is $\varphi(\omega_2) < \psi(\omega_2)$, the curve 1 has to be closer to curve $M_{2.0}(\omega_2)$.

The correction of functional graph $M_{2.0}(\omega_2) + I_{2.0}^{(r)} d\omega_2 / dt$ is followed by the calculation of the second approximation. For angular velocities of hydrodynamic coupling that satisfy inequation (2), the universal characteristic of the coupling $M(\omega_1,\omega_2)$ may be defined also using analytic functions. This enables creating a program for solving this problem by personal computer.

CONCLUSION

In this paper an iterative procedure for calculation a starting regime of power transmission with a hydrodynamic coupling and a driving motor, with *a priori* accepted accuracy, is presented. The number of iterative steps depends on required result accuracy.

If functional dependence $\omega_1(\omega_2)$ is determined according to the “quasi-static” model, the calculation could be considerably simplified, and it will lose its iterative nature. On the other hand, in addition to reduced accuracy of the calculation, it is more important that in this case miscalculation could not be estimated.

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PRORAČUN VREMENA ZALETA PRENOSNIKA SNAGE SA HIDRODINAMIČKOM SPOJNICOM I POGONSKIM ELEKTROMOTOROM

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U radu je izložen grafo-analitički postupak proračuna vremena zaleta prenosnika snage sa hidrodinamičkom spojnicom i pogonskim elektromotorom. Izložen iterativni postupak rešavanja zadatka omogućava da se vreme zaleta sistema može odrediti sa unapred usvojenom tačnošću. Broj iterativnih koraka proračuna zavisi, naravno, od usvojene tačnosti proračuna..

Ključne reči: *hidrodinamička spojnica, elektromotor, vreme zaleta.*