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Simulation of the Joint Operation of an Electric Motor and a Hydraulic Coupling in a Belt Conveyor Drive

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Abstract: The task of ensuring smooth movement of the conveyor belt during the operation of powerful belt conveyors with a long route and several intermediate electric drives is an urgent technical problem. One of the effective ways to solve this problem is to create drives that ensure the joint operation of an electric motor and a hydraulic coupling. In this article, a mathematical model of the joint operation of an electric motor and a hydraulic coupling is proposed. When constructing the mechanical characteristics of their joint work, possible deviations of the electric motor slip from the nominal values are taken into account. To simulate various possible approximating functions that describe the mechanical characteristics of the drive, two of the most accurate mathematical expressions that are used are the Kloss equation and the equation for interpolating hyperbolic functions. With regard to the electric drive of a powerful belt conveyor, the analysis of the appropriate areas of application of these dependencies is carried out, their main advantages and disadvantages are considered.

Keywords: Drive, Belt conveyor, Hydraulic coupling, Mechanical characteristic, Traction force.

1 Introduction

Powerful belt conveyors are a fairly popular type of continuous industrial transport machines used in a wide range of industries. Such machines are most widely used in the mining industry, as well as in the implementation of complex transport and logistics complexes of industrial overloading terminals - ports, open and closed bulk cargo storage warehouses [1, 2].

In the demands of the modern world industry, multi-drive belt conveyors are becoming increasingly common, enabling one to implement conveyor paths of fairly complex route configurations. There are two essential options of implementation of multi-drive belt conveyors: with stationary drives installed

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along the path [3, 4] and with drives moving along the path together with the belt [5, 6]. At this stage of development, the second version of conveyors exists only in the form of several experimental examples, while conveyors with stationary drives installed along the length of the route are widespread, and the engineering approaches used in their creation have proved their effectiveness. This conclusion is confirmed by the analysis of research results and data on the practical use of multi-drive belt conveyors, given in the works from the list of references to this article.

The use of hydraulic couplings in belt conveyor drives is a fairly common design solution that allows for smooth flow of work processes during non-stationary operation of the conveyor. Furthermore, this solution provides a more even distribution of traction force between the conveyor drives while its operation is steady by softening the mechanical characteristics of the drive motors. A hydraulic coupling is a specialized device for the torque transmission that simultaneously performs the function of protecting the drive motor from overloads. In general, the structure of the hydraulic coupling includes a pump wheel (it creates torque), a turbine wheel (it takes the torque and transmits it to the elements of the executive device), and working fluid (it transmits the torque from the pump to the turbine by creating a closed circulating flow) [7].

When constructing mathematical models of traction force distribution within the belt conveyors drive systems and when developing models of drives in a non-stationary mode (running and braking), it is necessary to have an analytical expression that describes the function of the change in the driving torque generated by the motor.

This function depends on the position coordinate of the motor output shaft or the frequency of its rotation. This function is quite often represented as a constant [8 – 12]. In this case, the motion speed of the output shaft is modeled as a variable value, which is a significant assumption and in certain cases significantly distorts the results of studies.

For example, [8] describes a method for approximating the constant value of the electric motor torque when starting a loaded conveyor belt. This method is used at the design stage of powerful belt conveyors in order to determine the static power and other basic parameters of the electric motors. As disadvantages of the method, it is noted that an underestimated torque value is obtained to start the loaded conveyor at zero initial speed of the belt.

For example, the source [8] describes a technique for approximating a constant torque value used by engineers when determining static power and other parameters of electric motors, indicating its negative sides, in particular, the possible insufficiency of the determined torque value for pulling a loaded conveyor from a place at zero initial speed.

In addition, various designs of the hydraulic couplings applied in structures of belt conveyors drives are described in [8]. It should be noted that the natural appearance of the mechanical characteristic of joint work of the electric motor and hydraulic coupling with a possible regulation of the change of working fluid amount, given in [8], is a family of complex curves. Each curve should be approximated using different mathematical models.

There are many researches devoted to determining the resistance to the movement of the load-bearing conveyor belt [9, 13 – 16], in which the resistance values are presented not only as constant parameters, but also as functions of time, belt speed, belt position coordinates, etc. Some of the resistance to motion factors change their values in case of failure of the idlers supporting the belt [17]. This process has a random cause and is accompanied by a number of assumptions in its description [18]. The variability of the resistance nature factors can lead to significant inaccuracies in research under the assumption of the constant motor torque, since the torque developed by the electric motor, especially in steady state, changes in accordance with the resistance moment, affecting the speed of the belt [19].

Modeling of conveyors as electromechanical systems [20] is often accompanied by the opposite effect: constant resistance to motion are opposed to variable electromechanical characteristics of drives.

When electric motors and hydraulic couplings are used together in the structure of belt conveyor drives, it is necessary to synthesize an analytical description of the functions of the mechanical characteristics of their joint work. This makes it possible to more accurately simulate the operation of conveyor drive systems, including intermediate linear drives. Such drives are characterized by unique features of operation, in particular, they have unstable contact of traction and load-bearing belts, which create additional resistance to motion [21, 22]). In some cases, the representation of mechanical characteristics in the form of linear functions distorts the results of modeling the operation of drives. In particular, this is observed when modeling sequential failure of drives of one system or their selective operation in steady-state mode when the conveyor is fully loaded [23]. In order to overcome this drawback inherent in the linear approximation of the mechanical characteristics of the joint operation of electric motors and hydraulic couplings as part of conveyor drives, the idea was expressed in [24] about the expediency of presenting a mechanical characteristic in the form of a hyperbola of a certain type. However, this idea was not implemented in this work.

Within the framework of the design studies when modeling variants of traction force distribution between the conveyor drives, it is necessary to have a range of analytical expressions of mechanical characteristics, each of which will represent any boundary or median value of the permissible range of deviations of the motor slip [25].

2.1 General structure

Taking into account the representation of the mechanical characteristics of the joint operation of the electric motor and hydraulic coupling in the form of a hyperbolic function [23], the family of mechanical characteristics can be represented as a system of equations

$$\begin{aligned}
M_{1t}(n_t) &= \frac{-b_1}{a_1(n_c - n_t) + c_1} + d_1, \\
M_{2t}(n_t) &= \frac{-b_2}{a_2(n_c - n_t) + c_2} + d_2, \\
M_{mt}(n_t) &= \frac{-b_m}{a_m(n_c - n_t) + c_m} + d_m, \\
M_{tp1}(n_t) &= \frac{-b_{p1}}{a_{p1}(n_c - n_t) + c_{p1}} + d_{p1}, \\
&\vdots \\
M_{tpi}(n_t) &= \frac{-b_{pi}}{a_{pi}(n_c - n_t) + c_{pi}} + d_{pi},
\end{aligned} \tag{1}$$

where n_c is the synchronous rounds per minute of the motor; a_1, b_1, c_1, d_1 are function parameters for the first boundary value of the interval of the motor slip; a_2, b_2, c_2, d_2 are function parameters for the second boundary value of the interval of the motor slip; a_m, b_m, c_m, d_m are function parameters for the median value of the interval of the motor slip; a_{p1}, b_{p1}, c_{p1}, d_{p1} are function parameters for the first design case of fluid volume change in the hydraulic coupling; a_{pi}, b_{pi}, c_{pi}, d_{pi} are function parameters for the i-m design case of the change of the hydraulic coupling.

The structural diagram of the mathematical model of the joint operation of the electric motors and hydraulic couplings is represented in Fig. 1.

2.2 Module "Analytical description of the curve of the hydraulic coupling torque coefficient λ"

The initial data for the implementation of this module are:

– Hydraulic coupling torque at the rated slip $M_{gm\ nom}$ [Nm],

– Fluid density of the hydraulic coupling ρ [kg/m^3],

– Active diameter of the hydraulic coupling D_a [m],

– Rated slip of the hydraulic coupling $\varepsilon_{gm\ nom}$,

– Synchronous rounds per minute of the motor shaft n_c [r/min], and
– Multiplicity of the starting moment of the hydraulic coupling k.

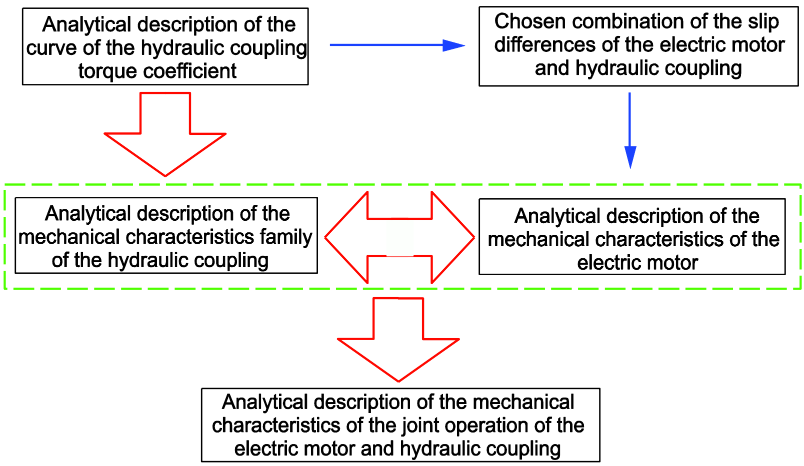

Fig. 1 – *Structural diagram of the mathematical model of the joint operation of the electric motors and hydraulic couplings.*

The module consists of two submodules. The first submodule defines the following parameters:

Nominal rounds per minute of the turbine wheel of the hydraulic coupling consistent with the motor slip

$$n_{gm\,nom} = n_c(1-\varepsilon_{gm\,nom})\,. \tag{2}$$

Nominal value of the hydraulic coupling torque coefficient λ_{nom} consistent with the nominal rounds per minute of the turbine wheel of the hydraulic coupling

$$\lambda_{nom} = \frac{M_{gm\,nom}}{\rho D_a^5 n_{gm\,nom}^2}\,. \tag{3}$$

Starting moment of the hydraulic coupling

$$M_{gm\,s} = kM_{gm\,nom}\,. \tag{4}$$

Maximum value of the hydraulic coupling torque coefficient $\lambda_{\max}$

$$\lambda_{\max} = k\lambda_{nom}\,. \tag{5}$$

The second submodule is reduced to the solution of the system of equations (7), which result in determining the parameters of the curve of the hydraulic coupling torque coefficient (a, b, c, d), also represented as a hyperbola,

$$\lambda(\varepsilon) = \frac{-b}{a\varepsilon + c} + d \,, \tag{6}$$

where ε is hydraulic coupling slide.

The system of equations of the second submodule is as follows:

$$\begin{gathered}
0 = -b/c + d, \\
\lambda_{nom} = \frac{-b}{a\varepsilon_{gm\,nom} + c} + d, \\
\lambda_{\max} = \frac{-b}{a + c} + d, \\
\frac{\lambda_{nom}}{2} = \frac{-b}{(a\,\varepsilon_{gm\,nom})/2 + c} + d.
\end{gathered} \tag{7}$$

2.3 Module “Analytical description of the mechanical characteristic of the squirrel-cage induction motor”

The initial data for the implementation of this module are:

– Nominal power of the electric motor P_{nom} [kW],

– Nominal slip of the electric motor ε_n, and

– Multiplicity of the maximum torque of the electric motor k_{em}.

The following auxiliary parameters are defined, in a preliminary manner, in this module:

Nominal torque of the squirrel-cage induction motor

$$M_{nom} = 9550 \frac{P_{nom}}{n_c(1 - \varepsilon_n)} \,. \tag{8}$$

Breakdown torque developed by the electric motor

$$M_{\max} = k_{em} M_{nom} \,. \tag{9}$$

Breakdown slip of the electric motor

$$\varepsilon_{cr} = k_{em}\varepsilon_n + \varepsilon_n \sqrt{k_{em}^2 - 1} \,. \tag{10}$$

When describing of the mechanical characteristic of the electric motor, the Kloss equation is used [26]

$$M_{em}(n) = \frac{2M_{\max}}{\dfrac{1 - n/n_c}{\varepsilon_{cr}} + \dfrac{\varepsilon_{cr}}{1 - n/n_c}} \,, \tag{11}$$

where n is the rounds per minute of the electric motor.

2.4 Module "Analytical description of the mechanical characteristics family of the hydraulic coupling"

This module consists of two submodules. In the first module, a number of deviations of the hydraulic coupling is set within the values from 0 to 1. The higher number of the set values favours more accurate approximation of the mechanical characteristic of the joint operation of the electric motor and the hydraulic coupling. The current mathematical model uses the number of slip values relevant to the geometric progression with the common ratio of 2, as follows: 1; 0.5; 0.25; 0.125; 0.0625; 0.03125; 0.015625; 0.0078125; 0.0039; 0.00195.

A coefficient of the hydraulic coupling torque corresponded to each set slip value is determined using (6) with the calculated coefficients of the curve according to the system of equations (7).

The determining factors in this case are the coordinates of the intersection points of the mechanical characteristics of the hydraulic coupling with the operating part of the mechanical characteristics of the electric motor. In case when there is no intersection of the mechanical characteristics of the hydraulic coupling (when slide $\varepsilon_{gm\ nom} = 1$) with the operating part of the mechanical characteristics of the electric motor, the latter may take the maximum load at the stage of its own starting, without developing a maximum torque. This is the limiting circumstance and one of the indicators of appropriateness of selection of hydraulic coupling dimension type.

The family of mechanical characteristics of the hydraulic coupling is a system of equations (12):

$$
\begin{aligned}
M_{gm1}(n) &= \lambda_{\varepsilon(1)}\ \rho D_a^5 n^2,\\
M_{gm(0,5)}(n) &= \lambda_{\varepsilon(0,5)}\ \rho D_a^5 n^2,\\
M_{gm(0,25)}(n) &= \lambda_{\varepsilon(0,25)}\ \rho D_a^5 n^2,\\
M_{gm(0,125)}(n) &= \lambda_{\varepsilon(0,125)}\ \rho D_a^5 n^2,\\
M_{gm(0,0625)}(n) &= \lambda_{\varepsilon(0,0625)}\ \rho D_a^5 n^2,\\
M_{gm(0,03125)}(n) &= \lambda_{\varepsilon(0,03125)}\ \rho D_a^5 n^2,\\
M_{gm(0,015625)}(n) &= \lambda_{\varepsilon(0,015625)}\ \rho D_a^5 n^2,\\
M_{gm(0,0078125)}(n) &= \lambda_{\varepsilon(0,0078125)}\ \rho D_a^5 n^2,\\
M_{gm(0,0039)}(n) &= \lambda_{\varepsilon(0,0039)}\ \rho D_a^5 n^2,\\
M_{gm(0,00195)}(n) &= \lambda_{\varepsilon(0,00195)}\ \rho D_a^5 n^2.
\end{aligned}
\tag{12}
$$

2.5 Module “Analytical description of the joint operation of the squirrel-cage induction motor and hydraulic coupling”

This module also consists of two submodules. In the first submodule expressions from the system of equations (12) are sequentially combined with the (11), resulting in determining the values of rounds per minute of the pump wheel of the hydraulic coupling and their corresponding values developed by the hydraulic coupling torque. Rounds per minute of the turbine wheel are calculated at the known corresponding values of the hydraulic coupling slip.

In the second submodule, based on the obtained values of the hydraulic coupling torques and their corresponding rounds per minute of the turbine wheel, interpolation of the mechanical characteristics of the of the joint operation of the electric motor and hydraulic coupling is performed on the basis of the system of equations (1). The unknown coefficients of the mathematical function approximating the mechanical characteristic are calculated similarly to the coefficients of the mathematical function approximating the hydraulic coupling torque coefficients (system of equations (7)), since both of these functions are expressed by hyperbolic equations, which include four unknown coefficients. To determine the unknown coefficients, it is necessary to set the coordinates of the four points through which the graph of the mechanical characteristic passes.

2.6 Example of building the mathematical model of the joint operation of the squirrel-cage induction motor and hydraulic coupling

For an example, a mathematical model of the joint operation of the electric motor and hydraulic coupling with the parameters:
$P_{nom} = 250$ kW, $\varepsilon_n = 0.017$, $k_{em} = 1.9$, $M_{gm\,nom} = 1658$ Nm, $\rho = 890$ kg/m^3, $D_a = 0.57$ m, $\varepsilon_{gm\,nom} = 0.05$, $n_c = 1500$ r/min, $k = 1.8$ is considered.

The nominal rounds per minute of the turbine wheel of the hydraulic coupling corresponding to the nominal slip, according to (2), is $n_{gm\,nom} = 1425$ r/min.

The nominal value of the hydraulic coupling torque coefficient corresponding to the nominal rounds per minute of the turbine wheel of the hydraulic coupling, according to (3), is $\lambda_{nom} = 1.525 \cdot 10^{-5}$.

The starting moment of the hydraulic coupling, taking into account (4), is $M_{gm\,p} = 2984.4$ Nm.

The maximum value of the hydraulic coupling torque coefficient, according to (5), is $\lambda_{\max} = 2.745 \cdot 10^{-5}$.

The system of equations (7) of the second submodule for the mentioned initial data and additional calculated parameters takes on form (13)

$$
\begin{aligned}
0 &= \frac{-b}{c} + d, \\
1.525 \cdot 10^{-5} &= \frac{-b}{0.05a + c} + d, \\
2.745 \cdot 10^{-5} &= \frac{-b}{a + c} + d, \\
\frac{1.525 \cdot 10^{-5}}{2} &= \frac{-b}{\frac{0.05}{2}a + c} + d.
\end{aligned} \tag{13}
$$

Solving the system of equations (13) results in an expression for the function of the hydraulic coupling torque curve:

$$
\lambda(\varepsilon) = \frac{-9.45 \cdot 10^{-9}}{6.04 \cdot 10^{-3}\varepsilon + 3.21 \cdot 10^{-4}} + 2.91 \cdot 10^{-5}. \tag{14}
$$

The diagram of the function is shown in Fig. 2.

The maximum torque developed by the electric motor, according to (9), is $M_{\max} = 3076.5\ \text{Nm}$. The critical slip of the electrical motor, according to (10), is $\varepsilon_{cr} = 0.05976$. Thereby, the mechanical characteristic of the electric motor, defining by (11), takes on form (15):

$$
M_{em}(n) = \frac{2 \cdot 3076.5}{\frac{(1 - n/1500)}{0.05976} + \frac{0.05976}{(1 - n/1500)}}. \tag{15}
$$

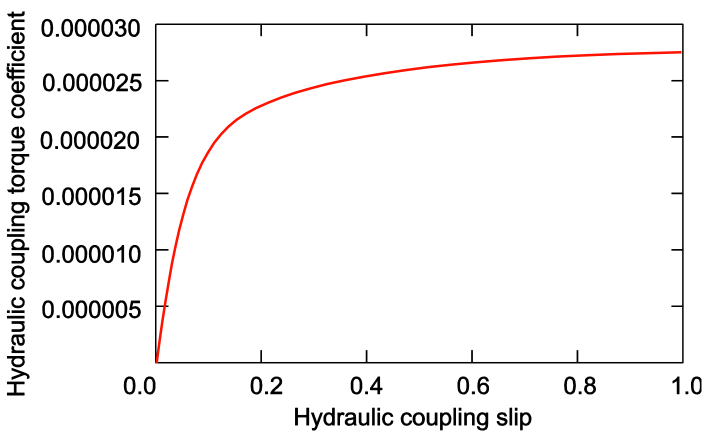

Fig. 2 – *Diagram of the function of the hydraulic coupling torque curve.*

Substituting sequentially the values of the hydraulic coupling slip (1; 0.5; 0.25; 0.125; 0.0625; 0.03125; 0.015625; 0.0078125; 0.0039; 0.00195) into (14), a set of corresponding values of the hydraulic coupling torque coefficient $\lambda(\varepsilon_i)$ is defined.

Taking into account the given set, the family of the mechanical characteristics of the hydraulic coupling takes on form of the system of equations (16)

$$\begin{aligned}
M_{gm(1)}(n) &= 2.76\cdot 10^{-5}\ \rho D_a^5 n^2,\\
M_{gm(0,5)}(n) &= 2.63\cdot 10^{-5}\ \rho D_a^5 n^2,\\
M_{gm(0,25)}(n) &= 2.39\cdot 10^{-5}\ \rho D_a^5 n^2,\\
M_{gm(0,125)}(n) &= 2.03\cdot 10^{-5}\ \rho D_a^5 n^2,\\
M_{gm(0,0625)}(n) &= 1.56\cdot 10^{-5}\ \rho D_a^5 n^2,\\
M_{gm(0,03125)}(n) &= 1.06\cdot 10^{-5}\ \rho D_a^5 n^2,\\
M_{gm(0,015625)}(n) &= 6.36\cdot 10^{-6}\ \rho D_a^5 n^2,\\
M_{gm(0,0078125)}(n) &= 3.44\cdot 10^{-6}\ \rho D_a^5 n^2,\\
M_{gm(0,0039)}(n) &= 1.68\cdot 10^{-6}\ \rho D_a^5 n^2,\\
M_{gm(0,00195)}(n) &= 7.12\cdot 10^{-7}\ \rho D_a^5 n^2.
\end{aligned} \tag{16}$$

Verification of the criterion of correct selection of the hydraulic coupling on its rated characteristics (verification of the absence of intersection of the hydraulic coupling mechanical characteristics, when sliding $\varepsilon_{gm\,nom} = 1$, with the operating part of the mechanical characteristics of the electric motor) is to build and analyze the graphs of the mechanical characteristics of the electric motor and the family of mechanical characteristics of the hydraulic coupling (Fig. 3). The operating part of the mechanical characteristics of the electric motor is shown in Fig. 3 corresponds to the range of the rounds per minute from 1410 to 1500 rpm.

When modeling, application of the Kloss equation (11) allows one to model with a high degree of accuracy exactly the operating part of the mechanical characteristics of the electric motor that is not a linear function. In this case the operating part corresponding to the rounds per minute range from 0 to 1410 r/min is imaginary and does not represent the real view of the mechanical characteristic of the electric motor, which, as a rule, corresponds to significantly higher values of the starting torque. Thus, when analyzing the diagram in Fig. 3, the intersection points of the curves of the hydraulic coupling and the idle part of the electric motor characteristic can be neglected.

In case of interpolation of the mechanical characteristic of the joint operation of the electric motor and hydraulic coupling on the basis of the system (1), the following dependence of the torque, transmitted by the hydraulic coupling, on the rounds per minute of the turbine wheel of the hydraulic coupling is formed

$$M_{1t}(n_t) = \frac{-1.47 \cdot 10^6}{4.98(n_c - n_t) + 428.16} + 3241.51. \tag{17}$$

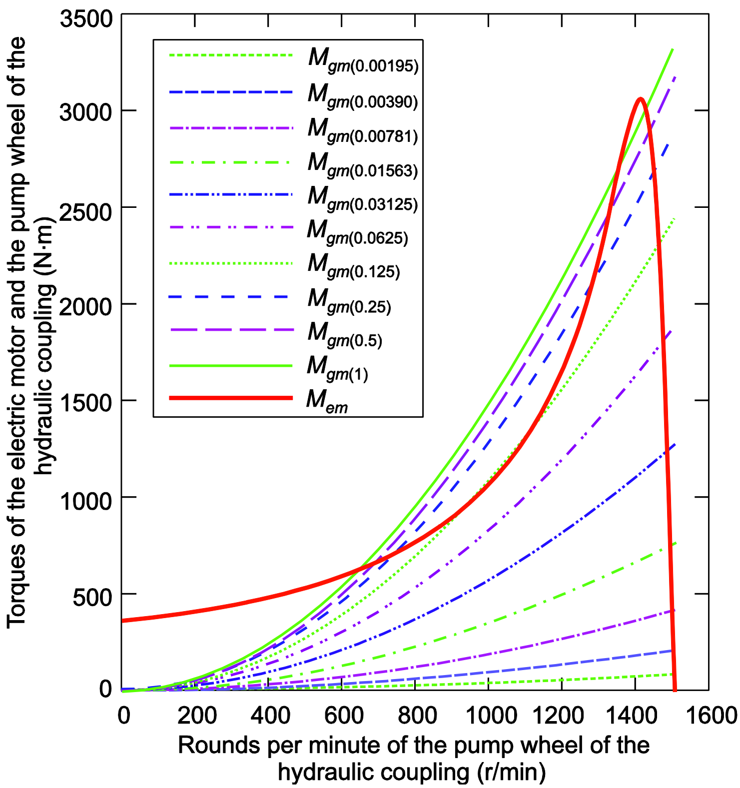

Fig. 3 – *Functions of the mechanical characteristics of the pump wheel f the hydraulic coupling.*

In situations where interpolation is not possible, the following empirical expression (with a high degree of precision of the final result) can be used instead of (17):

$$M_t(n_t) = \frac{-b\rho D_a^5 n_{n1}^2}{a\left(1 - \dfrac{n_t}{n_{n0.00195}}\right) + c} + d\rho D_a^5 n_{n1}^2, \tag{18}$$

where $n_{n0.00195}$ is the rounds per minute of the pump wheel of the hydraulic coupling when its slip is 0.195 % at the intersection point of the curve of the hydraulic coupling and the operating part of the mechanical characteristic of the electric motor; n_{n1} is the same parameter when the hydraulic coupling slide is 100 %.

The graphs of the mechanical characteristics of the joint operation of the electric motor and hydraulic coupling, (17) and (18), are presented in the Fig. 4.

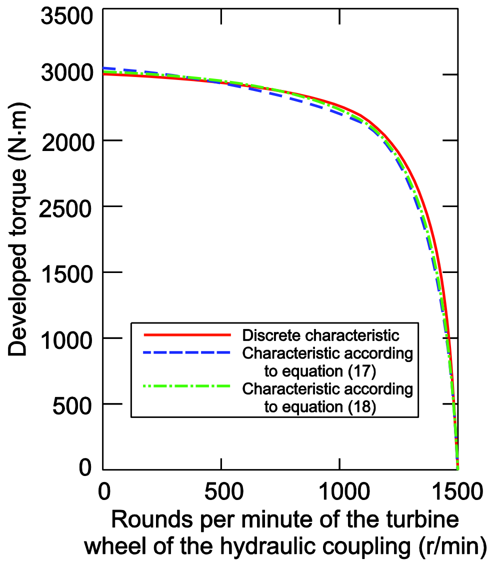

Fig. 4 – *Mechanical characteristic of the joint operation of the electric motor and hydraulic coupling.*

3 Discussions

The feasibility of using (17) or (18) is determined by the calculation goals.

For instance, considering the graphs shown in Fig. 4, it would be more appropriate to use (18) when modeling the irregularity of the traction forces of belt conveyor drives developed in the steady state. The given relation exhibits a high gradient of slope of the operating part (the range of rounds per minute is from 500 to 1000 r/min), and this, in turn, leads to more considerable spread in values of the traction forces of drives (the system is intentionally placed into the deteriorated conditions in comparison with the possible real scenarios of the conveyor operation) [23].

Application of (17) is appropriate when modeling the distribution of traction forces between the conveyor drives in case of successive breakdown of the latter

[23]. In this situation, in the range of rounds per minute from 500 to 1000 r/min the (17) exhibits a high gradient of slope of the mechanical characteristic in comparison with the equation (18), which again put the system into more unfavorable design conditions.

The difference of the torque values of the turbine wheel, determined using (17) and (18), when its rounds per minute is 1350 r/min (the largest apparent discrepancy of the mechanical characteristics in Fig. 4), is 70.0 Nm or 3.5 %. The deviation value of the mechanical characteristic given in the example confirms the described argumentation of the planning selection in different design situations between (17) and (18). It should be noted that (17) uses the higher initial torque during the running of the turbine wheel of the hydraulic coupling in comparison with the discrete (experimental) characteristic and torque value determined using (18). The discrepancy value is 42.4 Nm or 1.4 %.

While taking into account the nominal slip of electric motors in the range of $\pm$20 % (two boundary values and one median value which corresponds to the nominal operation mode), the family of the mechanical characteristics of the joint operation of the electric motor and hydraulic coupling (1) using the (17) takes on form

$$\begin{aligned}
M_{t1}(n_t) &= \frac{-9.28\cdot 10^4}{0.306(n_c - n_t) + 26.984} + 3210.13,\\
M_{t2}(n_t) &= \frac{-1.06\cdot 10^5}{0.351(n_c - n_t) + 30.955} + 3286.34,\\
M_{tm}(n_t) &= \frac{-1.47\cdot 10^6}{4.98(n_c - n_t) + 428.16} + 3241.51.
\end{aligned} \tag{19}$$

The family of the mechanical characteristics of the joint operation of the electric motor and hydraulic coupling (1) using (18) takes on form

$$\begin{aligned}
M_{t1}(n_t) &= \frac{-b\rho D_a^5 1418.2^2}{a\left(1 - n_t/1498.51\right) + c} + d\rho D_a^5 1418.2^2,\\
M_{t2}(n_t) &= \frac{-b\rho D_a^5 1435.59^2}{a\left(1 - n_t/1499\right) + c} + d\rho D_a^5 1435.59^2,\\
M_{tm}(n_t) &= \frac{-b\rho D_a^5 1427.07^2}{a\left(1 - n_t/1498.75\right) + c} + d\rho D_a^5 1427.07^2.
\end{aligned} \tag{20}$$

In the system of equations (20) the coefficients *a*, *b*, *c*, *d* are defined in accordance with (6) and (14).

The graphical interpretation of the boundary mechanical characteristics of the joint operation of the electric motors and hydraulic couplings corresponding to the deviations of the nominal motor slip values in the range of +20% are presented in Fig. 5.

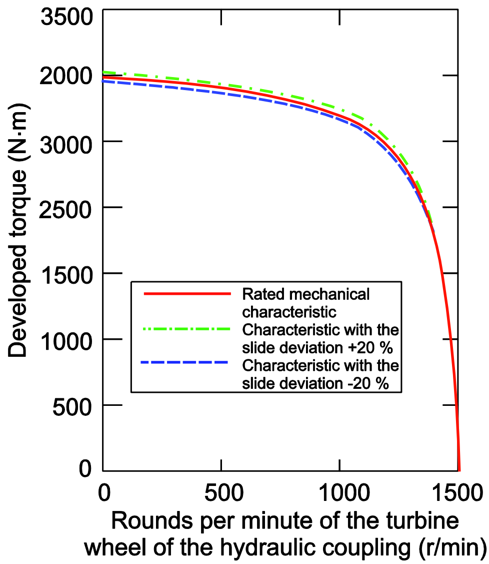

Fig. 5 – *Mechanical characteristics of the joint operation of the electric motor and hydraulic coupling taking into account the slip deviations.*

The graphs in Fig. 5 are built using the system of equations (20) which represent running characteristics in the relation to the discrete (experimental) mechanical characteristic most accurately. For the convenience of the visual analysis, the graphs are built in the same system as the rated mechanical characteristic.

When analyzing the graphs presented in Fig. 5, the difference of the torques on the turbine wheel of the hydraulic coupling at the moment of the beginning of its running, taking into account the boundary dependencies, is 73.4 Nm or 2.47%. In the process of further running, the given deviation, when approaching to the rated values of the rounds per minute and developed torque, is smoothed.

Therefore, the impact of the deviations of the nominal slip values of the electric motors in the drive structures of the belt conveyor drives on distribution of the traction forces is seriously reduced during their joint operation with the hydraulic couplings. This fact may be taken into account by engineers when analyzing distribution of traction forces between the drives of belt conveyors, selecting driven equipment and designing control systems of heavily-loaded belt conveyors.

4 Conclusion

The article describes a developed mathematical model designed to simulate the joint operation of electric motors and hydraulic couplings as part of belt conveyor drives, which takes into account possible individual differences of electric motors slipping from nominal values. Two alternative approaches to the specified modeling problem are considered based on the use of various possible approximating functions to describe the mechanical characteristics of an electric motor, the features of the application of the two most accurate mathematical equations are shown, the areas of their application, advantages and disadvantages are described. This allows the designer to establish two alternative analytical equations for determining the mechanical characteristics of the joint operation of the electric motor and the hydraulic coupling, which reflect the change in torque developed by the turbine wheel of the hydraulic coupling, depending on its rounds per minute, as well as to perform their comparative analysis.

Since with different probable slipping differences characteristic of the same electric motors, the same hydraulic couplings must be filled with working fluid individually for each specific electric motor, the method proposed in this article allows us to reasonably determine the necessary degree of compensation for differences. This ensures guaranteed nominal mechanical characteristics of the drives and a stable mode of tension of the conveyor belt during its operation.

Further research directions involve conducting experimental studies of the possibilities of hydraulic couplings to compensate for slipping differences of electric motors. The obtained results of these studies will improve the efficiency of regulating the belt tension mode based on discrete control of hydraulic couplings.

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