машины на динамические и энергетические показатели ее работы проведена путем выполнения серии машинных експериментов.

Результаты. Установлено, что при реостатном пуске асинхронного привода шахтной подъемной машины в моменты переключения сопротивлений ротора в ее элементах возникают пиковые нагрузки, что сопровождается значительными величинами потребляемой мощности. Раскрыты закономерности влияния продолжительности оптимального разгона машины и коэффициента, который определяет важность соответствующих составляющих в структуре оптимизационного критерия, на энергетические и динамические оценочные показатели режима разгона машины.

Научная новизна. Предложен оптимальный режим разгона шахтной подъемной машины по комплексному интегральному критерию с использованием прямого вариационного метода. Полученный в работе оптимальный режим движения описывается непрерывно-дифференцированной функцией, позволяющей уменьшить динамические нагрузки в элементах машины и улучшить энергетические характеристики ее привода.

Практическая значимость. На основе динамического и энергетического анализа оптимизированного режима разгона шахтной подъемной машины установлена рациональная продолжительность режима и коэффициент, определяющий вес составляющих в структуре оптимизационного критерия. Это позволяет обоснованно подходить к реализации найденного режима разгона шахтной подъемной машины, при использовании частотного управления асинхронным электроприводом машины.

Ключевые слова: *шахтный подъем, динамические нагрузки, оптимальное управление движением*

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DYNAMIC MODEL OF INTERACTION OF MECHANISMS ON THE SECTION BETWEEN THE ROLL MILL STAND AND THE COILER IN THE PROCESS OF WIRE WINDING BY GARRETT REEL

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ДИНАМІЧНА МОДЕЛЬ ВЗАЄМОДІЇ МЕХАНІЗМІВ ДІЛЯНКИ КЛІТЬ-МОТАЛКА ПРИ ЗМОТУВАННІ СОРТОВОГО ПРОКАТУ МОТАЛКОЮ ТИПУ ГАРРЕТА

Purpose. The purpose of the present research is to find ways to manage the coiler-mill drive system, which provides an increase in the tension stability of long products when winding on Garrett-type coiling machines at the output mill-coiler section at a level that does not lead to the extension of the profile in the outlet stand caliber.

Methodology. For the purpose of obtaining static and dynamic models of rolling tension at the output mill-coiler section, experimental studies of the winding of long products were carried out. The experimental activities included measuring electrical parameters such as the current of the exhaust cage engine, the current of the pinch roll motor, the speed of the wire feeding machine motor, the current of the coiler motor, the speed of the coiler motor, the position of the coil-laying pipe. The investigations were carried out during the rolling of profiles with a diameter of 16 and 18 mm using a digital recorder "Vizir".

Findings. Taking into account the quantitative and qualitative analysis of the results of the study, a general structural scheme was obtained for the rolling-up model in the mode of stabilizing only the amperage of the winder. Dependencies describing the complex of interrelationships between the mechanisms of the output mill-coiler section and the tension and length of the rolling in this site were obtained. A dynamic model for the interaction of the mechanisms of the mill-coiler section during rolling of long products was developed.

Originality. For the first time, a dynamic model of the interaction of the mechanisms of the mill-coiler section has been created for rolling coils with a coiler of the Garrett type, which takes into account both the direct and indirect influence of the mechanisms on the coiling process, which makes it possible to search for rational methods for stabilizing the tension of rolled products.

Practical value. The above model is a prerequisite for solving the problem of stabilizing the tension of the rolled section of the outlet mill-coiler, which ensures the stabilization of the transverse dimensions of long products when it is coiled by a coiler of the Garrett type.

Keywords: *stabilization of rolling tension, Garrett-type coiler, dynamic model, mill-coiler section*

Introduction. The flat-and-edge rolled steel (rolled section steel) is in great demand in machine building and hardware industry. Further processing of the rolled steel requires its sizing in draw benches.

Energy efficiency and quality of this process depend mainly on the precision and stability of maintaining the geometry of metal cross section. The bundles of small-section wires (with diameter up to 14 mm) are formed by Stelmor technology which does not change the cross section of the wire. On the other hand, the bigger wire is winded in Garret reels. In these devises the metal is pulled, providing the interaction between the coiling machine and the milling stand. This tension is necessary to bend the metal and align the wire tightly along the coil. At the same time, this tension is applied to the output roll mill. Excessive tension at the mill changes the cross section of the rolled metal at the output of the mill.

Such changes in the tension of metal in Garret reels occur most frequently during the transition to the new layer because of the speed change when the metal first touches the coil surface. The transient process occurs in the system "roll mill stand – rolled metal – coiling machine". Another factor in this system is variation of wire length because of movement of the folding device.

Thus, the first problem with coiling the section wires in Garret reels is the need to stabilize its tension between the roll mill stand and the coiling machine so as not to change the metal cross section in the output mill. This problem is mentioned in [1], but no solution is offered. The problem is complicated by the difficulties of experiments in industrial environment; besides, there are no devices for the measurement of the tension in such conditions. The only solution is to simulate the interaction between the mechanisms involved in metal processing. However, there are still no suitable dynamic models describing the interaction between the output roll mill stand, the metal and the Garret reel. Developing such model would help solving the problem of metal tension stabilization.

Presentation of the main research. The design scheme for the Garret reel model is shown in Fig. 1.

The following assumptions have been made in the model:

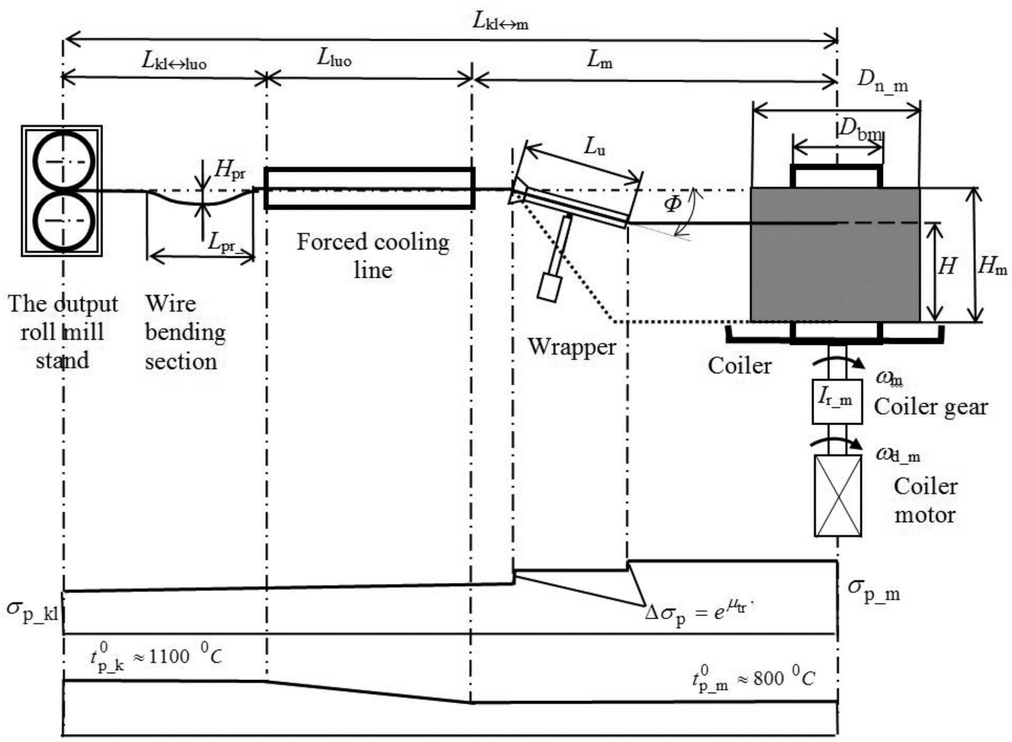

Fig. 1. The design scheme for the Garret reel model:

$L_{kl\leftrightarrow m}$ – the distance between the output mill and the coiling machine; $L_{kl\leftrightarrow luo}$ – the distance between the output mill and the forced cooling line; L_{luo} – the length of the forced cooling line; L_m – the length of the coiling section; D_{n_m} – current coil diameter; l_{pr_0} – the length of bending section; h_{pr} – the height of the bending outlet; l_u – the length of the folder wrapper; φ_u – the tilt angle of the folder wrapper; D_{bm} – the reel drum diameter; H_m – the coil height; h_u – the folder head height; ω_m, ω_{d_m} – the angular speeds of the reel and reel motor; i_{p_m} – the gear ratio of the reel; $t^0_{p\_k}$, $t^0_{p\_m}$ – the temperature of the rolled metal at the stand output and before the coiling machine correspondingly; σ_{p_kl}, σ_{p_m} – the specific tension of the rolled metal behind the stand and before the coiling machine; $\Delta\sigma_p$ – the change of metal tension at its bending point; φ_u – the angle of metal bending; μ_{tr} – the friction factor

- the folder wrapper moves from the flat lower position to the horizontal one, the rolled metal is bent only twice;

- the distance from the input point of the coiling machine to the contact point of the metal and the coiler is constant and does not depend on the coiling layer;

- the additional tension from the master coil wrapper is neglected;

- the metal temperature in the forced cooling section is reduced linearly;

- the calculated elasticity modulus between the stand and the coiling machine $E_{y\_kl\leftrightarrow m}$ equals its mean average value on the section

$$E_{u\_kl\leftrightarrow m}=\frac{L_{kl\leftrightarrow m}}{\dfrac{L_{kl\leftrightarrow luo}}{E_{u\_kl}}+\dfrac{2\cdot L_{luo}}{E_{u\_kl}+E_{u\_m}}+\dfrac{L_m}{E_{u\_m}}},\qquad(1)$$

where $L_{kl\leftrightarrow m}$, $L_{kl\leftrightarrow luo}$, L_{luo}, L_m stand for the length of the wire between the roll stand and the coiling machine and its section between the stand and forced cooling line, the forced cooling line and the coiling machine, correspondingly; E_{u_kl}, E_{u_m} are the elasticity factors of the metal on the stand output and before the coiling machine correspondingly.

The coiling machine is at $L_{kl\leftrightarrow m}$ distance from the final mill stand. At $L_{kl\leftrightarrow luo}$ distance there is a forced cooling line (FCL) whose length is L_{luo}, the distance from the coiling machine to FCL is L_m.

The coiler drum with diameter D_{bm} is driven by electric motor via the gear with ratio i_{r_m}.

As the coiler rotates and the wrapper (tube) moves vertically up to h_u height, the wire is stacked layer by layer and makes a coil of H_m and diameter D_{n_m}.

The angular displacement of the wrapper φ_u is provided by reversal hydraulic drive at given speed.

During the winding, the electric drive operates in current and EMF control modes, i.e. regulates power. The product of specific tension σ_{p_m} and coiling radius under constant speed has also to be constant and correspond to the armature current of the coiling electric drive.

The tension before the stacker reduces because of the double bending of wire by φ_u in the input and output of the wrapper (the tube). Moreover, because of the friction of wire along its way from the mill stand to the coiler, the tension gradually drops to the value, which defines the specific tension after the final stand σ_{p_k}. The additional tension by the wrapper is neglected. The pattern tension diagram is given in Fig. 1.

Between the output mill and FCL there is a sloped groove of l_{p_0} length, the wire bends by h_p outlet in it.

The temperature of the metal at the entrance to the mill is about 1100 °C, after FCL it is approximately 800 °C. The diagram of temperature (assumed that it linearly drops in FCL) is shown in Fig. 1.

The rolling speed V_p for processing wires D_p diameter is given in calibration tables.

The static model of tension on the section between the stand and the coiler describes the wire tension at the key coiling points.

Let the specific tension of the wire before the coiler be σ_{p_m}.

Then, specific tension of the wire before the stacker σ_{p_u} reduces according to Euler's equation with double wrapping angle φ_u

$$\sigma_{p\_u}=\sigma_{p\_m}\cdot e^{-2\cdot\mu_{tr}\cdot\varphi},$$

where μ_{tr} is the friction factor, $\mu_{tr}\approx 0.3$.

The specific tension of the wire after the stand reduces by the value of friction in the forced cooling line and passing grooves – σ_{tr}

$$\sigma_{tr}=\mu_{tr}\cdot\rho_s\cdot g\cdot(L_{luo}+L_{kl\leftrightarrow luo}-l_{pr\_0}),$$

where ρ_s is specific weight of the steel, $\rho_s = 7850$ kg/m^3; g is free fall acceleration $g = 9.8$ m/s^2.

Thus, the tension after the stand σ_{p_kl} will be defined as

$$\sigma_{p\_kl}=\sigma_{p\_m}\cdot e^{-2\cdot\mu_{tr}\cdot\varphi_u}-\sigma_{tr}.\qquad(2)$$

The wrapping angle φ_u depends on the stacker position and the height of its head h_u by

$$\varphi_u=\arcsin\frac{H_m-h_u}{l_u},$$

where H_m is total vertical displacement of the stacker head; l_u is the length of the tube.

For small angles we can assume

$$\varphi_u\cong\frac{H_m-h_u}{l_u}.$$

Thus, the correlation between specific tensions before the coiler and after the stand is described by exponential dependence from the stacker position.

It should be noted that the wire feeding machine between the final stand and the forced cooling line produces additional transportation tension T_{tr_t} and then equation (2) can be rewritten as

$$\sigma_{p\_kl}=\sigma_{p\_m}\cdot e^{-2\cdot\mu_{tr}\cdot\varphi_u}-\sigma_{tr}+\Delta\sigma_{tr\_t},\qquad(3)$$

where $\Delta\sigma_{tr\_t}$ is additional tension before the stand, produced by the effort of the wire feeding machine.

The last two components in (3) are constant values. Since we have discarded the wire feeding machine from the model, the correlation of specific tensions before the coiler and after the stand is described as

$$\sigma_{p\_kl}=\sigma_{p\_m}\cdot e^{-2\cdot\mu_{tr}\cdot\varphi_u}-\sigma_s,\qquad(4)$$

where σ_s is constant value (σ_s = const), satisfying the condition $\sigma_{tr}\le\sigma_s\le 0$.

The initial conditions for simulation are easy to obtain by giving the initial value of wire tension before the coiler and the initial position of the stacker.

It would be natural to take the reference tension σ_{p_mz} as initial value of the specific tension before the wire, under the power regulation it equates

$$\sigma_{p\_mz}=\frac{\eta_m\cdot U_{eds}}{V_p\cdot S_p}\cdot\left(I_{m\_z}-I_{m\_hh}\right),$$

where η_m is the coiler efficiency, $\eta_m \approx 0.95$; U_{eds} is the coiler motor EMF; V_p is rolling speed (calibrated); S_p is the area of the cross-section with diameter D_p, $S_p = \frac{\pi \cdot D_p^2}{4}$; I_{m_hh} is the no load current of the winder drive; I_{m_z} is the given current.

The initial (given) tension of the wire behind the output mill stand can be calculated by the equation (4) using an arbitrary constant value of σ_s from the allowed range and initial position of the stacker. The latter should be taken as its lowest position (the position in which it first captures the wire), i. e. for $h_u = 0$ $\left(\varphi_u = \frac{H_m}{l_u}\right)$.

Then, the specific tension in the wire at the initial time instant $\sigma_{p\_kl(0)}$ can be calculated as

$$\sigma_{p\_kl(0)} = \sigma_{p\_m} \cdot \mathring{a}^{-2\cdot\mu_{tr}\cdot\frac{H_m}{l_u}} - \sigma_s.$$

It is proper to take the initial tension of σ_{p_mz} of, e. g. 5 N/mm^2, and then calculate $\sigma_{p\_kl\_0}$ and given value of current I_{m_z} during coiling. This will provide correct matching of the control plant model and the model of electric drive at initial time instant.

We shall assume that the initial tension σ_s in the model equals zero, considering that the transporting effort of the wire feeding machine is equal to the friction force in the forced cooling section.

Let us examine the dynamic model of the wire tension in the section between the final stand and the coiler.

We shall define the tension effort during the coiling according to the Hooke's law

$$\sigma_{p\_m} = E_{u\_kl\leftrightarrow m} \cdot \frac{\Delta l_{kl\leftrightarrow m}}{L_{\bar{i}\_kl\leftrightarrow m} - \Delta l_{kl\leftrightarrow m}},$$

where $E_{u\_kl \leftrightarrow m}$ is the weighted mean value (Young's module) in the section between the final stand and the coiler (calculated according to (1)); $\Delta l_{kl \leftrightarrow m}$ is total elastic strain of the wire before the coil; $L_{p\_kl \leftrightarrow m}$ is the length of the wire between the output stand and the coiler given the current value of specific tension of the wire.

The length of the wire between the output stand and the coiler consists of three next components

$$L_{p\_kl \leftrightarrow m} = L_{kl \leftrightarrow m} + \Delta l_u + \Delta l_{pr}, \tag{5}$$

where Δl_u is increment of the wire length when the wrapper bends from horizontal position; Δl_{pr} is increment of the wire length because of its bending.

The stretch of the wire when the wrapper bends from its horizontal position is

$$\Delta l_u = l_u - \sqrt{l_u^2 - \left(H_m - h_u\right)^2}, \tag{6}$$

and, according to V. Alshits, in order to elongate the wire by bending

$$\Delta l_{pr} = \frac{\pi^2 \cdot h_{pr}^2}{4 \cdot l_{pr\_0}},$$

where l_{pr_0} is the length of the bending section; h_{pr} is wire bending outlet height.

The height of the outlet bending in turn is related to the tension of the rolled metal with the following functional relationship

$$h_{pr} = \frac{h_{pr\_0}}{1 + \frac{\sigma_{p\_kl}}{\sigma_{kr}}},$$

where σ_{p_kl} stands for specific tensions of the rolled metal after output mill stand; h_{pr_0} is the height of the outlet bending with zero tension; σ_{kr} is critical specific tension.

For the last two parameters, the expressions are

$$h_{pr\_0} = \frac{5}{384} \cdot \frac{\rho_s \cdot g \cdot l_{pr\_0}^4}{E_{u\_kl} \cdot r_{i\_p}^2};$$

$$\sigma_{kr} = \frac{\pi^2 \cdot E_{u\_kl} \cdot r_{i\_p}^2}{l_{pr\_0}^2},$$

where r_{i_p} is the radius of inertia of the rolling metal cross section in the vertical plane. For the circular section, $r_{i\_p} = 0.25 \cdot D_p$ (see in [2]).

On the other hand, the length of non-stressed rolling metal between the coiler and mill stand $L_{pn\_kl \leftrightarrow m}$, in the current instant of time τ is a function of rolling metal linear velocity after output mill stand and linear velocity of the coil surface difference

$$L_{pn\_kl\leftrightarrow m}(\tau) = \int_0^{\tau} (V_{p\_kl} - V_{p\_m}) \cdot dt + L_{p\_kl\leftrightarrow m(0)},$$

where V_{p_kl} is the linear velocity of the rolling metal after output mill stand; V_{p_m} is the linear velocity of the rolling metal at the point of contact with the surface of coil, $L_{p\_kl \leftrightarrow m(0)}$ is length $L_{pn\_kl \leftrightarrow m}$ at the initial instance time ($\tau = 0$).

Then the elastic elongation of the rolling metal at the section of output mill stand-coiler $\Delta l_{kl \leftrightarrow m}$, is defined as

$$\Delta l_{kl \leftrightarrow m} = L_{p\_kl \leftrightarrow m} - L_{pn\_kl \leftrightarrow}.$$

The linear velocity of the rolling metal after mill stand can be calculated with the expression

$$V_{p\_kl} = \frac{\left(1 + s_{op}\right) \cdot D_{kl} \cdot \omega_{dv\_kl}}{2 \cdot i_{r\_kl}},$$

where s_{op} is leading ratio; D_{kl} is the diameter of mill stand rolls; ω_{dv_kl} is the angular velocity of the mill stand motor; i_{r_kl} is the gear ratio of the mill stand.

The leading ratio in turn depends on the tension of the rolling metal before and after the mill stand, as it is known from the studies of V. N. Vydrin, A. P. Chekmarev. In our case the rolling before the mill stand is carried out with automatic stabilization of rolling metal bending, i.e. the rolling metal has a small stable tension before the output mill stand. Then the following expression can be written

$$s_{op} = s_{op\_0} + k_{s\_\sigma} \cdot \sigma_{p\_kl},$$

where k_{s_σ} is the influence ratio of the rolling metal preliminary tension on the leading ratio in the mill stand caliber. For the final mill stands it can be assumed that $k_{s\_\sigma} \approx 4 \cdot 10^{-4}(\text{N/mm}^2)^{-1}$; s_{op_0} is leading ratio in the final mill stand without preliminary tension of the rolling metal.

The quantity s_{op_0} can be evaluated with the calibration data: the rated value of the rolls diameter by calibration D_{kl_k}, the angular rotation speed of the motor by calibration $\omega_{dv\_kl\_k}$, and calculated linear velocity of the rolling metal after the output mill stage by calibration – $V_{p\_kl\_k}$

$$s_{op\_0} = \frac{i_{r\_kl}}{D_{kl\_k}} \cdot \frac{V_{p\_kl\_k}}{\omega_{dv\_kl\_k}} - 1.$$

The initial conditions of dynamic tension model parameters can be calculated at the same initial instant. It can be assumed that at the initial instant the angular rotation of mill stand motor $\omega_{dv\_kl(0)}$, corresponds to the calibrated angular rotation speed – $\omega_{dv\_kl\_k}$

$$\omega_{dv\_kl(0)} = \omega_{dv\_kl\_k}.$$

Then the expression becomes

$$V_{p\_kl(0)} = \frac{\left(1 + s_{op\_0} + k_{s\_\sigma} \cdot \sigma_{p\_kl(0)}\right) \cdot D_{kl} \cdot \omega_{dv\_kl(0)}}{2 \cdot i_{r\_kl}},$$

assuming that $V_{p\_m(0)} = V_{p\_kl(0)}$ and considering that winding of rolling metal starts from the first layer ($n_{sl} = 1$), for the angular speed of coiler motor at the initial instance of time $\omega_{dv\_m\_0}$ the following expression is valid

$$\omega_{dv\_m(0)} = \frac{2 \cdot V_{p\_kl(0)} \cdot i_{r\_m}}{D_{bm} + D_p},$$

where D_{bm} is the diameter of the coiler drum; i_{r_m} is gear ration of the coiler.

The length of the rolling metal between the coiler and mill stand at the initial instance of time $L_{p\_kl \leftrightarrow m\_0}$, can be calculated by equation (5) with known values of Δl_u and Δl_{pr} for initial instance of time $\Delta l_{u(0)}$ and $\Delta l_{pr(0)}$.

Because at the initial instance of time the stacker is in the down position, then from equation (6) it follows

$$\Delta l_{u(0)} = l_u - \sqrt{l_u^2 - H_m^2},$$

and for $\Delta l_{pr(0)}$

$$\Delta l_{pr(0)} = \frac{\pi^2 \cdot h_{pr(0)}^2}{4 \cdot l_{pr\_0}},$$

where $h_{pr(0)}$ is the value of bending at the initial instance of time

$$h_{pr(0)} = \frac{h_{pr\_0}}{1 + \dfrac{\sigma_{p\_kl(0)}}{\sigma_{kr}}}.$$

In this way the dynamic model of mechanisms interaction at the section of the mill stand-coiler for Garret type coiler can be represented with structural diagram, shown in Fig. 2.

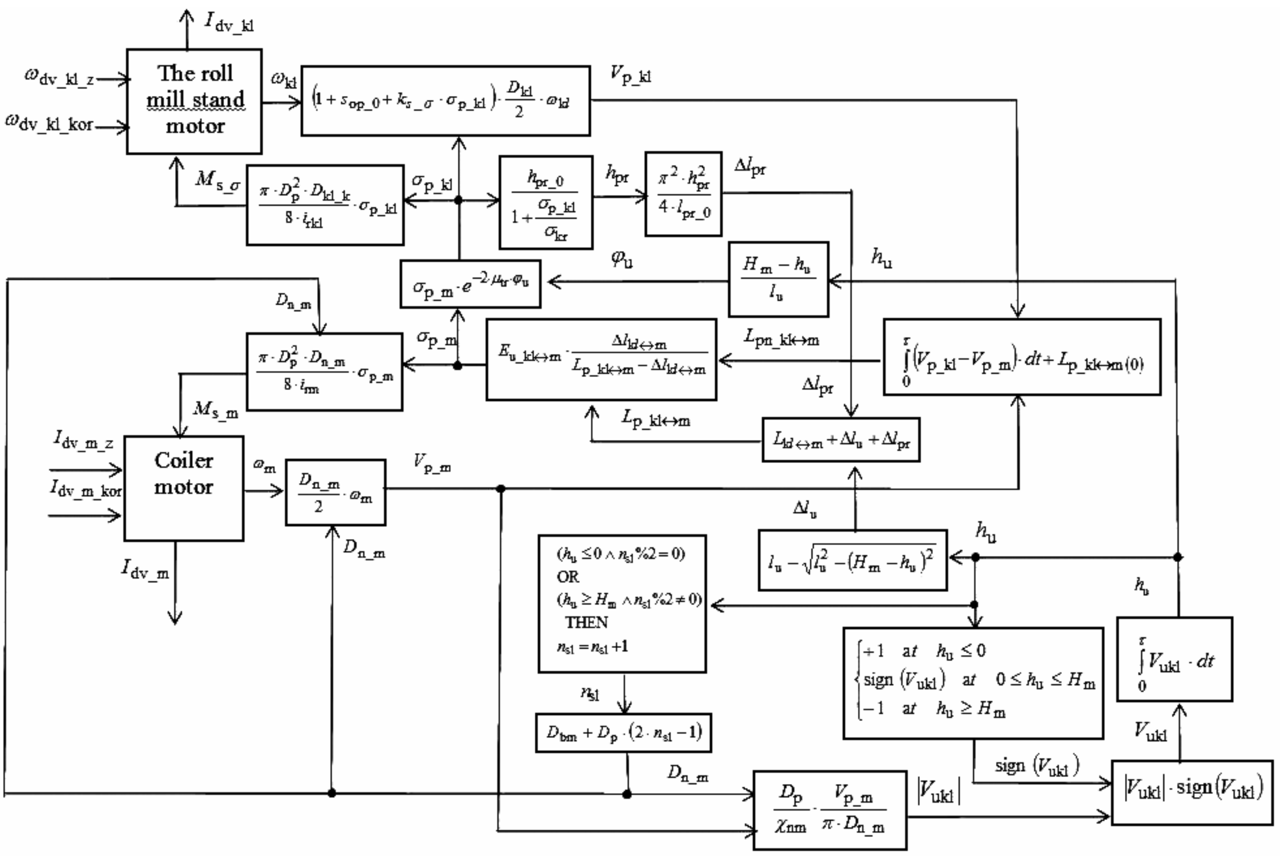

Fig. 2. General structural diagram of model which represents winding of rolling metal with stabilization of coiler current only

Conclusions.

1. The solution for tension stabilization at the section of mill stand-coiler requires developing a dynamic model of mechanisms interaction for Garret type coiler.
2. The dependences which describe a set of relationships, both external and internal, between mechanisms at the section of output mill stand-coiler and tension or length of rolling metal on this section are obtained.
3. The integrated dynamic model of mechanisms interaction at the section of the mill stand-coiler for Garret type coiler have both linear and non-linear blocks, which does not allow obtaining rational parameters and method for tension stabilization at the mill stand-coiler section.
4. For searching of tension stabilization rational methods, Heuristic method for control structures and their combinations definition as well as analysis of influence on basic mechanisms at the output mill stand-coiler section is used, i. e. by controlling the angular velocity of output mill stage or coiler current.

References.

1. Beshta, O.S., Kuvaiev, V.M., Ivanov, D.O., Politov, I.V. and Dinnik, Yu.O., 2015. The problems of controlling the winding of long products at small section-wire mill PAT "ArcelorMittal Kriviy Rig". *Theory and practice of metallurgy*, 3–6, pp.78–81.

2. Mezhetskii, G.D. and Zagrebin, G.G. eds., 2016. *Structural resistance*. Moscow: Izdatelsko-torgovaia korporatsia "Dashkov i K".

Мета. Пошук способів управління комплексом електроприводів „моталка – кліть“, що забезпечують підвищення стабільності натягу сортового прокату при змотуванні на моталках типу Гаррета на ділянці „випускна кліть – моталка“ на рівні, що не призводить до витягування профілю в калібрі випускної кліті.

Методика. Задля отримання статичної й динамічної моделей натягу прокату на ділянці „випускна кліть – моталка“ були проведені експериментальні дослідження процесу намотування сортового прокату. В експериментальних заходах проводилось вимірювання таких електричних параметрів, як струм двигуна випускної кліті, струм двигуна трайбапарата, частота обертання двигуна трайбапарата, струм двигуна моталки, частота обертання двигуна моталки, положення труби укладальника витків. Дослідження проводили при прокатці профілів діаметром 16 і 18 мм за допомогою цифрового реєстратора „Візир“.

Результати. З урахуванням кількісного та якісного аналізу результатів дослідження була отримана загальна структурна схема моделі змотування прокату в режимі стабілізації тільки струму моталки. Отримані залежності, що описують комплекс взаємозв'язків між механізмами ділянки «випускна кліть – моталка» та натягом і довжиною прокату на цій ділянці, а також розроблена динамічна модель взаємодії механізмів ділянки «кліть–моталка» при змотуванні сортового прокату.

Наукова новизна. Уперше створена динамічна модель взаємодії механізмів ділянки „кліть – моталка“ при змотуванні сортового прокату моталкою типу Гаррета, що враховує як прямий, так і непрямий вплив механізмів на процес змотування, який дозволяє здійснити пошук раціональних методів стабілізації натягу прокату.

Практична значимість. Наведена модель є передумовою вирішення проблеми стабілізації натягу прокату на ділянці „випускна кліть – моталка“, що забезпечує стабілізацію поперечних розмірів сортового прокату при його змотуванні моталкою типу Гаррета.

Ключові слова: *стабілізація натягу прокату, моталка типу Гаррета, динамічна модель, ділянка „кліть – моталка“*

Цель. Поиск способов управления комплексом электроприводов „моталка – клеть“, обеспечивающих повышение стабильности натяжения сортового проката при сматывании на моталках типа Гаррета на участке „выпускная клеть – моталка“ на уровне, который не приводит к вытягиванию профиля в калибре выпускной клети.

Методика. Для получения статической и динамической моделей натяжения проката на участке „выпускная клеть – моталка“ были проведены экспериментальные исследования процесса намотки сортового проката. В экспериментальных мероприятиях проводилось измерение таких электрических параметров, как ток двигателя выпускной клети, ток двигателя трайбапарата, частота вращения двигателя трайбапарата, ток двигателя моталки, частота вращения двигателя моталки, положение трубы укладчика витков. Исследования проводились при прокатке профилей диаметром 16 и 18 мм с помощью цифрового регистратора „Визир“.

Результаты. С учетом количественного и качественного анализа результатов исследования была получена общая структурная схема модели сматывания проката в режиме стабилизации только тока моталки. Получены зависимости, описывающие комплекс взаимосвязей между механизмами участка „выпускная клеть – моталка“ и натяжением и длиной проката на этом участке, а также разработана динамическая модель взаимодействия механизмов участка „клеть – моталка“ при сматывании сортового проката.

Научная новизна. Впервые создана динамическая модель взаимодействия механизмов участка „клеть – моталка“ при сматывании сортового проката моталкой типа Гаррета, которая учитывает как прямое, так и косвенное влияние механизмов на процесс сматывания, что позволяет осуществить поиск рациональных методов стабилизации натяжения проката.

Практическая значимость. Приведенная модель является предпосылкой решения проблемы стабилизации натяжения проката на участке „выпускная клеть – моталка“, что обеспечивает стабилизацию поперечных размеров сортового проката при его сматывании моталкой типа Гаррета.

Ключевые слова: *стабилизация натяжения проката, моталка типа Гаррета, динамическая модель, участок „клеть – моталка“*

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