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THE DYNAMIC SIMULATION FOR A COLD ROLLING MILL MACHINE

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ABSTRACT

For a cold rolling mill machine that represents a mechanical system, the dynamics problems may be solved by using the differential equations of their movement. We must consider all the connections and couplings for the mill machine that have too elastic elements like: springs, shafts and axes deformed through torsion, spring bars, work rolls and back-up rolls.

The vibrations of the mechanical system must be monitored in order to identify those components which cause problems when the damage of the mill occurs during the operation.

In these mechanical systems, mobile rigid permanent couplings are used - which transmit shocks and vibrations - or stretch couplings - which, due to the elastic element, dampen shock and vibration.

Vibrations can appear, too, because of rolling strip that can have undulations.

KEYWORDS: dynamic simulation, mechanical systems, elastic deformations

1. Introduction

This work focuses on aspects of mechanical torque and the aspects resulting from dings (impulsive movements) of a linear character and the vibrations that are produced during the milling impact. The values of the moments of mechanical inertia from electric motor rotors, the couplings, the brake drums and other components can be found in catalogs or can be determined.

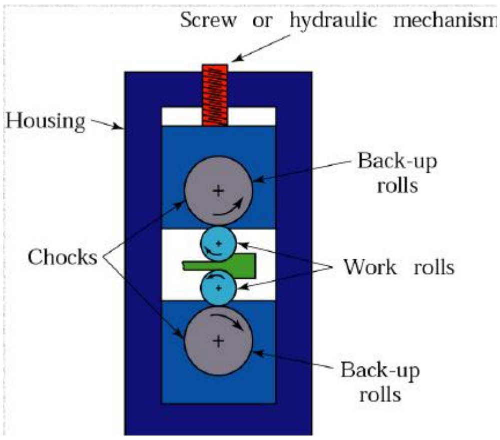

***Fig. 1.** Schedule of a quarto rolling mill machine*

The drive system for a modern drive rolling mill machine has greatest accuracy if the rolls speed and strip tension in correlation with torque, laminated force and rotation angle are taken into account.

The monitoring system incorporates the gap control existing in the rolling mill machine (Fig. 1).

The simulation of the dynamic behavior of some components or parts of rolling mill has contributed to decreasing the hardness of this movement parts, a new and functional design, the decrease of energetical consumption.

Initially, the model accepted for a simulation is composed of resorts which substitute the actual masses in work. The hardware and the software for the function control in work are connected to the simulated model. The model must be simulated in real time function work.

The advantage of this system simulation is that both the hardware and software may be tested, regulated and optimized without any risk in comparison to the the existent system.

To investigate the behavior of rolling mills we must study first of all, the behavior of chosen model. The dynamic simulations show the influence of diverse cause on the stand of rolling mill, during work like: high-frequency torsion vibration in the drive system; eccentricity and state surface of work roll shape; mill speed; reduction; strip tension; roll

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lubrication. The stands models satisfy the most stringent requirements on control dynamics and shock capacity and are largely free of maintenance. The equipment used has the following advantage: no limit in the drive rating; better control performance; the rotor of the engine (machine) has lower moments of inertia; less spare parts; greater flexibility of parts; simple start-up procedure; high reliability and availability; output frequencies from 0 to 600Hz; frequency control; highly accurate speed and torque control; positioning controls; parameter indicator, text display, graphic display.

Now, the modern tandem rolling mills have hydraulic gap controls on all stands. The control loops take into account the position control and work rolls tilt control. The roll forces are measured either indirectly, by pressure transducers attached to the hydraulic cylinders or directly by roll force transducers located in force line of the hydraulic cylinders.

2. Dynamic model of rolling mill machine

Dynamic loads are produced in cold rolling mill machine for strip when change technology parameter, stiffness and forces are periodical. The mathematical models for a mill dynamic model takes into consideration the equation of differential movement during the period of acceleration. For the kinematic schedule of drive mechanism, rigidities shall be regarded as usual at the electric motor shaft or at the input shaft of the work machine mechanism.

These loads appear on systems with variable stiffnesses, in different directions (transmissions) U-Joints, the coupling bars a.s.

Figure 2 shows a schedule of quarto rolling mill. By replacing a real system with an equivalent one, in energetic terms, an accuracy in determining its parameters which greatly affect the accuracy of the dynamic calculation must be kept.

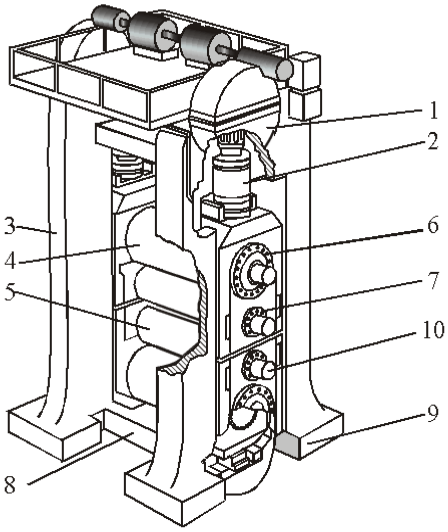

Fig. 2. *Quarto rolling mill machine* [2]*: 1-mechanism of positioning system; 2-positioning system; 3-mill frame; 4-back-up roll; 5-work roll; 6-back-up roll bearing; 7-work roll bearing; 8-lower traverse of rolling mill frame; 9-the sole of rolling mill frame; 10-work roll shaft*

The signals from the position transducers on each side of the stand are subtracted to produce the value for the roll tilt controller, which alters the tilt of centerline stand.

For the calculation of the dynamic rolling stands we can imagine a system with springs and masses in rotation as shown in Figure 3. The considered oscillating system is formed from a straight shaft (negligible mass) and the elastic constant of the torsion represents C = K. There can be treated oscillating systems with rotating masse from 2, 3 to n.

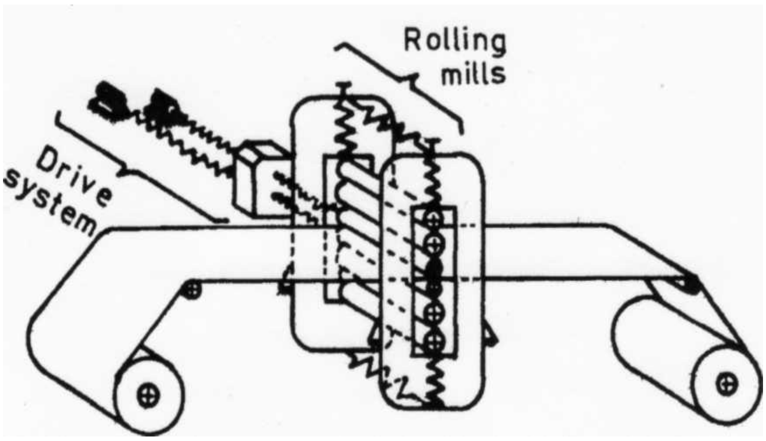

Fig. 3. *Stand for dynamical simulation* [3]

The roll force control is measured for each roll work. The total tension reference value is calculated in the control system in accordance with the rolling program and the strip dimensions and is sent to the tension controller.

The laminated torque can be estimated by [3]:

$$T = 0.5 * F * L \tag{1}$$

where T is torque (Nm); F-roll force (N); L-contact length (m).

The Power required by the rolling mill electrical engine is calculated as follows

$$P = 2\pi * N * F * L \tag{2}$$

where P is Power (in J/s =Watt or in-lb/min); N- rolls rotational speed (RPM); F- roll force (N).

The tension is measured by tension-meters mounted under the bearing blocks of the deflector rolls. The control system depends on the speed with respect to the gain and time response. The gain of the tension controller is matched out adaptively to the properties of the controlled system.

The equipment used has the following advantages [4]:

- total digital DC speed control in production line, closed loop control of tension and automatic control of speed;
- hydraulic automatic gauge control of five stands including pre-control AGC, monitoring AGC and mass flow AGC;
- positive/negative bending roll of working rolls, positive bending rolling of intermediate rolls;
- equipping with the collection system, display, storage and data output, including fault diagnosis and alarm system;
- five stands with the function of automatically decreasing speed and tension during passing welding seam;
- quick roll change;
- oil-air lubrication for all rolls;
- ferromagnetic filter and plate filter for rolls cooling system.

3. Schedule of reductions for cold rolling strip

Reduction schedule for obtaining cold rolled strip steel with dimensions of 0.3 × 1490 mm, the mill consists of five stands arranged in tandem.

For low carbon steel with percentage and reduced thickness ε_{tot} = 86%.is adopted

So, it will be used as a rough hot rolled strip and pickled (dimension 2 × 1490 mm) form of roller weighing 30 t.

3.1. Determining maximum reductions - admissible

By friction grease provided by technology, the friction coefficient varies from the first to the last pass on the rolling speed (v = 2.5 ...30 m / s) between f = 0.08 ...0.043 (anointing with oil). Based on these values, it results:

$$\Delta h_{max} = D\left(1-\frac{1}{\sqrt{1+f^2}}\right) = 585\left(1-\frac{1}{\sqrt{1+(0{,}08...0{,}043)^2}}\right) \tag{3}$$

$$\Delta h_{max} = 1.87...0.54 \text{ mm}$$

Depending on the driving engine power reduction is [4]

$$\Delta h_{max} = (32...33{,}8)\,\frac{K \cdot N_{nom}}{\psi \cdot p_m \cdot b \cdot v} \tag{4}$$

where k - is a driving motor overload factor, ranging from 1.5 ... 2 (K = 1.75 was adopted); Ψ = coefficient of rolling force positioning on arc length of contact. For cylinders with the leaf surface "mirror" Ψ = 0.48, p_m represents mean pressure lamination, which is approximatively as follows:

$$p_m = 1.15 \cdot \sigma_c \tag{5}$$

Knowing that the initial state annealed, strip has a flow limit equal to σ_0 = 22 daN / mm^2, there follows:

- for the first pass (admitting ε_1 reduction = 27%):

$$\sigma_{c1} = \sigma_0 + 17_{1}^{0.6} = 22 + 1.7 \cdot 27^{0.6} = 34.3 \text{ daN/mm}^2 \tag{6}$$

$$p_{m1} = 1.15 \cdot 34.3 = 39.4 \text{ daN / mm}^2$$

- for the final pass (in which ε_{tot} = 85%):

$$\sigma_{c5} = \sigma_{c0} + 3.4 \cdot \varepsilon_{tot}^{0.6} = 22 + 3.4 \cdot 85^{0.6} = 70.9 \text{ daN/mm}^2 \tag{7}$$

$$p_{m5} = 1.15 \cdot 70.9 = 81.5 \text{ daN / mm}^2$$

The amount of reduction to the first passage was adopted according to the final thickness. Depending on the deflection of the work rolls in the last pass,

$$\Delta h_{max} = (F_{max} / p_m.b)^2 \cdot 1/R \quad (8)$$

where F_{max} is the rolling force, which can ensure an arrow (bending) for rolling cylinders with Δd_{max}, respectively strip thickness tolerance that is:

$$\Delta d_{max} = \pm\ 0.05 \text{ mm} \quad (9)$$

$$F_{max} = \frac{37.6 \cdot \Delta d_{max} \cdot E \cdot D_s^4}{L^2\left(10 \cdot L \cdot +24 \cdot l\,\dfrac{15D_s^2}{L}\right)} \quad (10)$$

E = modulus of elasticity of the back-up rolls. E = 2.2 · 10^4 daN/mm^2; D_s = diameter of back-up rolls D_s = 1525 mm; L = roll length; l = spindles roll length; L = 1700 mm; l = 1000 mm.

Depending on the adopted values, we obtain

$$F_{max} = 12.6 \cdot 10^5 \text{ daN} \quad (11)$$

and it results

$$\Delta h_{max} = 0.14 \text{ mm} \quad (12)$$

Given the maximum value reductions - allowable, calculated according to three limiting criteria it results that, regarding the reduction scheme to be fixed, higher reductions than 1.87 mm at first pass and 0.14 mm at the last pass will not be allowed.

Considering the maximum reduction values - allowable calculated according to three limiting criteria, results that the established reductions scheme, will not allow larger reductions of 1.87 mm at first pass to 0.14 mm at the last pass.

3.2. Establishing the crossing reductions

For the first pass, we adopted, depending on the final strip thickness (h_5 = 0.3 mm), a ε_1 equal reduction = 27%. For other passages, the following factors shall be taken to reduce the degree of amplification from a switch to another: C = 1.555; C' = 0.90.

Depending on the values of these degrees of reduction per pass, the absolute reduction in thickness of the strip passes will be:

$$\Delta h_2 = \varepsilon_2 \cdot h_1 = 0.46 \text{ m};\ h_2 = h_1 - \Delta h_2 = 1.00 \text{ mm} \quad (14)$$

$$\Delta h_3 = \varepsilon_3 \cdot h_2 = 0.36 \text{ mm};\ h_3 = h_2 - \Delta h_3 = 0.64 \text{ mm} \quad (15)$$

$$\Delta h_4 = \varepsilon_4 \cdot h_3 = 0.21 \text{ mm};\ h_4 = h_3 - \Delta h_4 = 0.43 \text{ mm} \quad (16)$$

$$\Delta h_5 = \varepsilon_5 \cdot h_4 = 0.13 \text{ mm};\ h_5 = h_4 - \Delta h_5 = 0.30 \text{ mm} \quad (17)$$

Setting the stress values of each pass band will be according to: $\sigma_n = (0.2...0.5)\sigma_{c.med}$.

Calculation of average yield strength of the material after each pass is:

$$\sigma_{cn} = \sigma_{c0} + a \cdot \varepsilon_{tot_n}^{0.6} = 22 + 3.4 \cdot \varepsilon_{tot_n}^{0.6} \quad (18)$$

Calculation of tensile forces inside of strip is done by the next equation:

$$T_i = \sigma_i \cdot S_i = \sigma_i \cdot h_i \cdot b_i \quad (19)$$

3.3. Establishing the minimum speed rolling

Speed of the last mill stand is adopted with a value v_5 = 30m / s.

Minimum speed to other stands will be: v1 = 6.16 m / s, v2 = 9.0 m / s; v3 = 14.06 m / s; v4 = 20.93 m / s.

3.4. Calculation of the rolling forces

For the case of cold rolling it is recommended rolling force calculation using the Stone formula [5]:

$$F = p_m \cdot B \cdot l_c = (K_0 - \sigma_m).(e^m - 1)\ B.l_c / m \quad (20)$$

where: $K_0 = 1.15 \cdot \sigma_{c.med}$

The calculation of rolling force is performed in two stages: determining a first approximation rolling force without taking into account the elastic deformation of the cylinder. In this case:

$$I_c = \sqrt{R \cdot \Delta h} \quad (21)$$

The value obtained for the rolling force is calculated by the arc length of contact, taking into account the elastic deformation of rolls.

For rolling of carbon steel, the coefficient of friction when using emulsions of mineral oils is f_0 = 0.06 ...0.11.

It was found that the elastic deformation of the rolling mill work rolls, in contact with the arc length of the first passage increases by about 9.5%, which entails an increase in the rolling force by about 13.0%. Also, as the material of the strip is hardening, the relative increase in length of the contact arc is the greatest.

For other passages similarly resulting calculation were obtained, and the results were summarized in Table 1.

Table 1. *Schedule of reduced for rolling a strip of 0.3 × 1500 mm carbon steel as a continuous train of five rolling stands*

Number of passes	**h mm**	**Δh, mm**	**Reduction on passing ε, %**	**total reduction ε_{tot}, %**	**σ_c daN/mm²**	**σ_0 daN/mm²**	**v m/s**
0	2.0	-	-	-	22	6	-
1	1.46	0.54	27	27	46.56	10	6.16
2	1.00	0.46	21.2	50	57.55	21	9.0
3	0.64	0.36	36.0	68	64.75	30	14.06
4	0.43	0.21	32.4	78,5	68.60	33	20.93
5	0.30	0.13	29.1	85	70.88	17	30.0

f	**I_c^*, mm**	**I_c^{**}, Mm**	**P_m^*, daN/mm²**	**P_m^{**}, daN/mm²**	**Mill force*, tf**	**Mill force**, tf**	**Tensile strength tf**
-	-	-	-	-	-	-	15
0,061	12.57	13.76	40.16	41.08	757	848	21.9
0,057	11.60	13.28	58.74	61.30	1022	1221	31.5
0,051	10,26	12,04	62,72	66,70	965	1205	28,8
0.047	7.84	9.68	66.77	73.15	785	1062	21.3
0.043	6.17	8.30	81.15	93.65	751	1166	7.65

* -- for undeformed rolls
** -- for deformed rolls

A lot of tracking tests on the work and backup rolls were contours of the roughing mill and have been finished. Generally, work roll wear contours are wearing and the partial peak exists while the backup roll wear is generally non-uniform along the entire length of the roll.

4. Vibration analysis

The dynamic behavior of rolling mill is related to the vibrations that occur during operation. These vibrations can occur because of wear in bearings and spacing due to component wear rolling mill, cylinder roughness or even destruction of the strip surface which is laminated. The vibrations in the rolling tandem occur when acceleration or deceleration different underlain by different reasons.

This analysis shows that the alternating shear stress of a certain depth from the roll surface caused by roll contact pressure is very important.

The type of conventional wear of rolls makes contact stress between the edge of backup and work rolls increase significantly, contact stress between rolls in the middle of the roll had little change in the edge of the roll occurs sometimes [5].

The ideal solution is to obtain a uniform distribution of the pressure by improving the surfaces and the contour of the backup rolls and of the work rolls (Cao, J. G. 1999).

This wear is in correlation with the amplitude of the vibration measured in the stand 3, 4 and 5 of rolling mill machine.

The axial distribution of contact stress between the work roll and the backup roll could be influenced by the strip width and wear in different service period. It could be known by comparative analysis: the peak of the contact stress between rolls gradually increases with the increase of the strip width.

When the strip width increases, the peak of contact stress between rolls is increased with 106.3%. The rolling force per strip width is increased from 16 kN/mm to 27 kN/mm.

The distribution trends of contact stress between rolls are the same for different width strips, and the non-uniformity coefficient of contact stress distribution between the rolls are 1.35, 1.39 and 1.43, respectively [5]. It could be seen by the analysis of contact pressure between rolls that the peak position of contact stress between rolls is essentially the same, which is about 205mm from the edge side of the backup roll barrel.

The position control is in action when the gap is open, during the beginning of the operation and a subsystem of the gauge control.

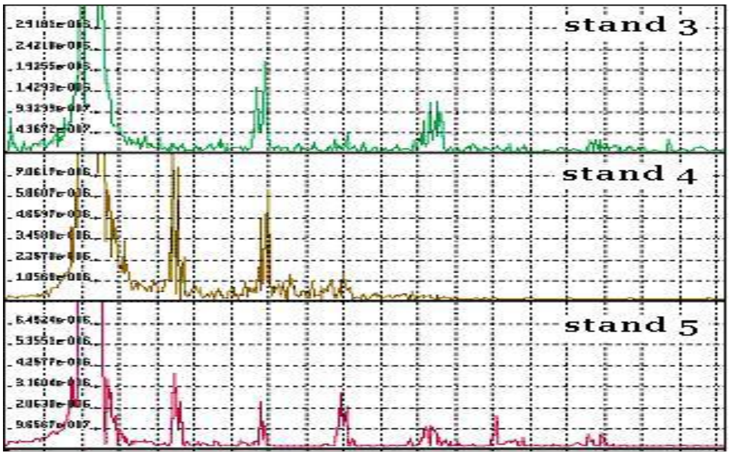

Fig. 3. *Vibrations spectrums in the stands no. 3, 4 and 5* [6]

For the rolls tilt control the signals from the position transducers on each side of the stand are subtracted to produce the actual value for the tilt controller, which alters the tilt about the centerline of the stand. Atypical response time is approx. 25 ms. The control loop is effective during rolling and is only switched during calibration after the calibration force has been reached. A typical response time is approx. 30 ms.

The gain of the tension controller is matched to the properties of the controlled system.

5. Conclusion

Nowadays, the modern rolling mills have hydraulic gap controls on all stand loops, positioning control, tilt control, speed, forces and tensions control. The roll forces are measured either indirectly by pressure transducers attached to the hydraulic cylinders or, directly by roll force transducers located in the line of force of the hydraulic cylinders. To predict the real roll force, it is necessary to introduce all variables into a corrective neural network. A corrective neural network can eliminate the errors in the roll force prediction.

This network can reduce the prediction errors at 21%. The additional variables, which are not introduced in the mathematical model, were necessary for the substitutive model only. The chemical composition of coil and temperature variable must be considered for the network.

The use of proposed system milling (stand) improves the accuracy of work parameters (in particular rolling force).

Some parameters not considered in the mathematical model can be easily introduced in a system neural network.

The system proposed can predict the work parameters and also correct wrong work milling as an option for the final adequate configuration (thickness and flatness and mechanical characteristic) for rolling sheet.

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