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EXPERIMENTAL DETERMINATION OF THE MECHANICAL STRESSES ON THE WARM ROLLING CYLINDERS

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Abstract:
The rolling mills cylinders are apply to thermo-mechanical stresses that are variable, complex, with extremely marked influences. Therefore, to intensify the rolling processes we need to observe the durability limits. To this purpose it is necessary to know the type of stress, the materials, and a detailed characterizes evaluation, to determine exploitation timing and to compare with previously established values. The paper presents experimental determinations of the mechanical stresses that take place during plastic deformation in rolling cylinders in exploitation. When being used, the laminating rolls are compared to the thermal tensions that cause thermal fatigue. This fatigue is the main cause for laminating rolls break.

Keywords:
experimental, mechanical, stresses, cylinders, fatigue, thermal

1. INTRODUCTION

The researches upon the stresses from the hot rolling mill cylinders represent an important scientific theoretical, experimental and economical issue. The calculus of the mechanical tensions which occur inside the laminating rolls are caused by the deformation process during the lamination, and they are the main elements for performing the sizing of the laminating rolls, as well as the main data for the calculus for the sizing of the rolling mill. We believe that the classical methods of rolls' sizing are not appropriate for most of the stages of the rolling process while producing rolls and laminated products.

This observation was remarked in [5] in which at page 489 say: „Although the solicitation of cylinders is a typical case of the variable solicitation, the usual calculus of dimensioning, is based merely on the static solicitation, without bareback is considered the fatigue".

In the book [4] at page 95, the author show: "The resistance of the hot rolling cylinders", with influence of different types tensional in at large, isn't studied" In the works written on this topic, rolling cylinders are calculated at static strains, which are wrongly considered to be constant in time. It is obvious that for the calculus of cylinders body, the determining of diameter based on the static bending moment does not correspond with the real exploitation strain, quite often thermal shocks lead to breakage of cylinders through shearing of caliber bead in the maximal sections. In the classical calculus of the cylinders, the decisive influence of thermal tension – with major effects in the rolling process – is not taken into consideration. Although rolling cylinder strain is a typical case of variable strain, the usual resistance calculus is based only on static strain, without considering the fatigue resistance in sizing the rolls. The replacement of cylinders takes place practically when the diameter is reduced below the minimum limit corresponding to normal wear, [2], [3].

We calculate mathematically the stresses in order to highlight them and to point out the fact that they do not influence the thermal fatigue of the laminating rolls very much, in order to compare them to those stresses caused by thermal influences.

2. CALCULATING THE MECHANICAL STRESSES

Nowadays specialized literature refers to the classical calculus methods used for the laminating rolls as, [2], [3], [5]:

- the pane of the rolls is calculated for a static bending;
- the journals of the rolls are calculated for twisting and bending;

- the journal blades are calculated for twisting tensions, meanwhile the blade is calculated for twisting and bending.

Forces acting on cylinders in rolling resistance classical calculation are presented in fig.1.

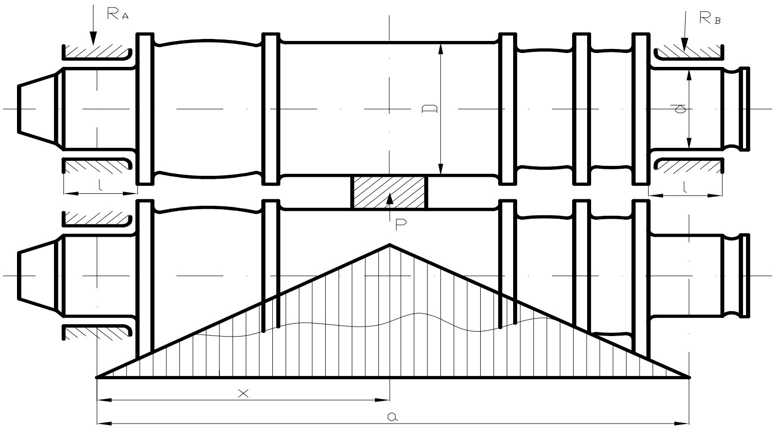

Figure 1. Representation of forces acting on cylinders rolling for the calculation of classical resistance

Nevertheless, we should note that the static stress we have considered for calculating the laminating rolls have not been too accurate as for time constant. The values of these „classic" tensions are determined in order to compare them to other tensions caused by temperature fields, [1].

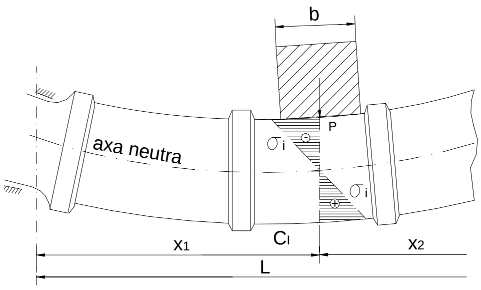

Figure 2. Roll stress caused by rotational bending stress

The bending and twisting stresses inside the pane rolls calibres of the laminating rolls are determined very accurately, avoiding any difficulty, according to current specialized literature, fig.2, [5].

According to fig. 2, bending stresses vary within the roll section, mainly in the compression deformation area, and have negative values. In the opposite side, there are only streching stresses whose values are positive.

Bending stress are determined according to the relation (1).

$$\sigma_i = \frac{M_i}{\frac{I_z}{y}}\alpha_k = \frac{M_i r}{\frac{\pi R^4}{4}} \tag{1}$$

where: M_i – bending moment who stress the roll during the rolling process, considering that the rolling force is does not vary along length, [5].

$$M_i = P\left[\frac{x_1 x_2}{L} - \frac{b}{8}\right] \tag{2}$$

where: x_1, x_2 – the distance between the bearing and the axis of the rolled metal-plate; L – the distance between the axis of the bearings; R – the radius of the roll on the rolling gauge; b – the width of the rolling good corresponding shift of size; α_k – shape factor in case of tension concentration on the rolls already subject to calibration; P – the rolling force obtained after experiments (with an oscillograph); r – the distance from the neutral axis inside the radial section of the roll.

The pane of the rolls is subject to stress and twisting tensions caused by the rolling process. These tensions are determined according to relation (3).

$$\tau = \frac{\alpha_k M_t}{\frac{I_p}{r}} = \frac{M_t}{\frac{R^4}{2r}}\alpha_k \tag{3}$$

where: M_t – twisting moment equal to rolling time, [5]: Δh – a reduced piece of the section, corresponding to the rolling design; r – the radius from the centre of the roll to the surface. The twisting moment is determining with relation (4):

$$M_t = 0{,}5F\sqrt{R\Delta h} \tag{4}$$

Twisting and bending tensions are simultaneous inside the roll section. The numerical calculus of these tensions is made for the fibers inside the roll section that correspond to a specific „r" radius.

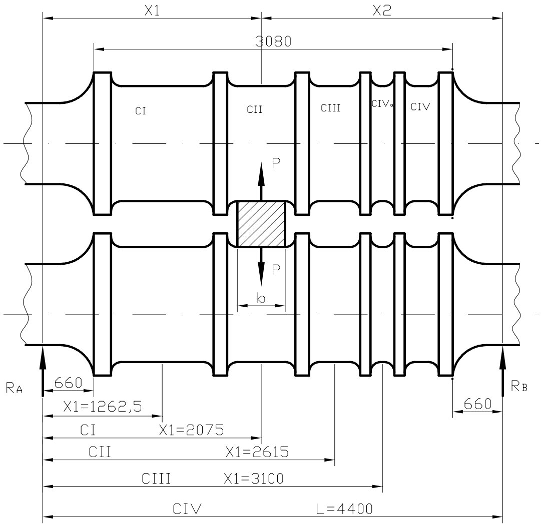

Figure 3. Calculus scheme for the lamination rolls

Contact pressure tensions between the rolling product and the rolls, within the area of deformation, could be determined according to mathematical relations, as well as according to average values of the parameters obtained during the rolling process. In case of a plastic deformation inside the core, we consider that the contact pressure tensions on the surface of the roll gauge are identical to the deformation tensions.

Therefore, we must point out that in case of warm rolling, contact pressure tensions could not be determined according to Hertz Relation, who corresponds only to elastic stress, [2].

If we use the Exelud [2], [3], [4], [5] formula for contact pressure, the tension for the contact pressure is determined according to the relation (5); we use the results of the experiments for specific rolling parameters.

$$\sigma_{pc} = \frac{F_{max}}{B_m \sqrt{R\Delta h}} \tag{5}$$

where: F_{max} – the highest rolling force; B_{max} – the average width of the rolled goods – the deformation core; $\sqrt{R\Delta h}$ - the length of the contact arc; R – the radius of the rolling gauge; Δh – reducing the height of the rolled good during several use.

Fig. 3 describes the mathematical calculus method we have used for determining the twisting, bending, and contact pressure stress within the rolls of the rolling equipment. The rolling gauges are situated on their surface, according to their size, determined by the axis of the bearings. The rolling forces stress the gauges of the 9.2 tonns ingot, according to the rolling scheme for each process during the rolling process. In order to determine the values of the forces, we have used the oscillograph to measure the parameters of the industrial rolling process – we have processed 10 ingots.

The experimental rolling mill is endowed with a plant for the determination of the lamination forces and of the variations of temperature fields in cylinders, which uses the electronic calculus technique, fig.4. The forces of lamination is measured in temporally experimentations of a help installation finded in the endowment rolling mill, in the aim verification of the stress from cylinders in order to subjected to excessive forces, which can produce ruptures or the damage of the rolling mill. In fig.5 presents the montage of tension-meter 1, which takes over half of rolling forces, transmits in bearing holder 3. The tension-meter is located under the axial bearing, lied on the head of the pressure screw 4, in a rigid metallic box, with the steel tie, at the superior cylinder's 5 equilibrate bend, fig.5, [4],[6].

Table 1. The results of the oscillograph analysis for industrial rolling

No. of stages	Gauge	Rolled good section [mm x mm]	Average rolling time [s]		$t_{mi}+t_{ri}$ [s]	Average rolling force F_m[kN]
			Time t_{mi}	Return time t_{ri}		
0	-	760/830 x 730/800	-	-	-	-
1	I	720 x 730/800	0,92	2,569	3,489	8527
2	I	640 x 735/800	0,889	4,505	5,394	11021
R	-	735/805 x 640	-	-	-	-
3	I	700 x655	0,928	2,099	3,027	8555
4	I	610 x 680	1,174	3,030	4,204	10905
5	I	530 x700	1,351	2,469	3,820	11010
6	I	450 x 720	1,161	6,110	7,271	10260
R	-	720 x450	-	-	-	-
7	II	600 x480	1,965	2,585	4,550	9582
8	II	500 x505	1,614	4,812	6,426	9999
R	-	505 x 500	-	-	-	-
9	I	390 x 525	1,496	2,546	4,042	65543
10	I	330 x 545	1,526	5,540	7,066	7621
R	-	545 x 330	-	-	-	
11	IV	430 x355	2,297	2,257	4,554	8410
12	IV	350 x 375	4,349	6,114	10,463	8682
R	-	375 x 350	-	-	-	
13	III	280 x 380	2,297	11,183	13,945	6927
The total value of the average rolling cycle of a 9.2 tonns ingot 78,251 s = 1,304 minutes						

Figure 4. The assambly of experimental rolling mill

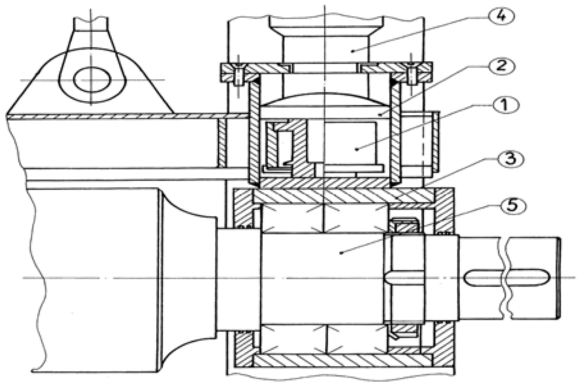

Figure 5.The tensiometer's montage with resistive traductors

Table 2. Calculating the contact bending stresses on the roll surface

No. of stages	Gauge	Rolled good section [mm x mm]	Gauge radiu R [mm]	Coef α_k	Bending stresses [daN/mm²]				
					σ_0 $\Delta r = 0$	σ_1 $\Delta r = 1{,}5$	σ_2 $\Delta r = 3$	σ_3 $\Delta r = 6$	σ_4 $\Delta r = 9$
0	-	760/830 x 730/800	-	-	-	-	-	-	-
1	I	720 x 730/800	620	1,39	5,09	5,08	5,07	5,04	5,02
2	I	640 x 735/800	620	1,39	6,91	6,57	6,55	6,52	6,49
R_s	-	735/805 x 640	-	-	-	-	-	-	-
3	I	700 x 655	620	1,39	5,21	5,19	5,18	5,16	5,13
4	I	610 x 680	620	1,39	6,63	6,61	6,59	6,56	6,53
5	I	530 x 700	620	1,39	6,66	6,64	6,63	6,60	6,56
6	I	450 x 720	620	1,39	6,18	6,17	6,15	6,12	6,09
R_s	-	720 x 450	-	-	-	-	-	-	-
7	II	600 x 480	592,5	1,39	8,55	8,53	8,51	8,47	8,42
8	II	500 x 505	592,5	1,39	8,55	8,53	9,51	8,47	8,42
R_s	-	505 x 500	-	-	-	-	-	-	-
9	I	390 x 525	620	1,39	4,26	4,05	4,04	4,02	4,00
10	I	330 x 545	620	1,39	4,49	4,48	4,47	4,45	4,42
R_s	-	545 x 330	-	-	-	-	-	-	-
11	IV	430 x 355	605	1,40	5,91	5,89	5,88	5,85	5,82
12	IV	350 x 375	605	1,40	6,02	6,07	6,05	6,02	5,99
R_s	-	375 x 350	-	-	-	-	-	-	-
13	III	280 x 380	600	1,40	5,80	5,78	5,77	5,74	5,71

Tabelul 3. Calculation of winding tension for all crossings of a rolling cycle of Lingo 9.2 tonnes at depths r_i

No. of stages	Gauge	Rolled good section [mm x mm]	Gauge radiu R [mm]	Coef α_k	Tensions of twisting [daN/mm²]				
					τ_0 $\Delta r = 0$	τ_1 $\Delta r = 1{,}5$	τ_2 $\Delta r = 3$	τ_3 $\Delta r = 6$	τ_4 $\Delta r = 9$
0	-	760/830 x 730/800	-	-	-	-	-	-	-
1	I	720 x 730/800	620	1,52	1,80	1,80	1,80	1,80	1,78
2	I	640 x 735/800	620	1,52	1,99	1,98	1,98	1,97	1,96
R_s	-	735/805 x 640	-	-	-	-	-	-	-
3	I	700 x 655	620	1,52	1,77	1,76	1,76	1,75	1,74
4	I	610 x 680	620	1,52	2,09	2,08	2,08	2,07	2,06
5	I	530 x 700	620	1,52	1,99	1,98	1,98	1,97	1,96
6	I	450 x 720	620	1,52	1,85	1,84	1,85	1,83	1,82
R_s	-	720 x 450	-	-	-	-	-	-	-
7	II	600 x 480	592,5	1,54	2,40	2,39	2,40	2,38	2,37
8	II	500 x 505	592,5	1,54	2,29	2,28	2,28	2,27	2,25
R_s	-	505 x 500	-	-	-	-	-	-	-
9	I	390 x 525	620	1,52	1,38	1,38	1,38	1,37	1,36
10	I	330 x 545	620	1,52	1,19	1,18	1,18	1,18	1,76
R_s	-	545 x 330	-	-	-	-	-	-	-
11	IV	430 x 355	605	1,51	1,92	1,91	1,91	1,90	1,89
12	IV	350 x 375	605	1,51	1,64	1,64	1,64	1,63	1,62
R_s	-	375 x 350	-	-	-	-	-	-	-
13	III	280 x 380	600	1,55	1,47	1,46	1,46	1,45	1,44

The numerical calculus of stress caused by bending stress according with relation (1) is presented in table 2. The numerical calculus of stress caused by bending stress according with relation (3) is presented in table 3.

The process of stresses determination, when tensions are caused by the contact pressure, according to relation (5), corresponds to the result on the contact surface between the rolled good and the rolls. The numerical calculus has been performed according to new characteristic parameters, used for industrial rolling of 9.2 tons ingots – as in table 4.

Determination of stress caused by contact pressure by the relationship (5) corresponds to the effect of surface contact between laminate and cylinders. Calculations made after the parameters resulting from the characteristics of industrial rolling ingots of 9.2 tons are presented in table 5.

Table 4. Calculation of pressure stress on the contact surface of rolling cylinders in the area of hole deformation

No. of stages	Gauge	Rolled good section [mm x mm]	Gauge radius R [mm]	Reduction Δh[mm]	Average width B_m[mm]	Tension σ_{pc}	
						$\sqrt{R\Delta h}$	σ_{pc} [daN/mm^2]
0	-	760/830 x 730/800	-	-	-	-	-
1	I	720 x 730/800	620	40/110	765	261,1	4,26
2	I	640 x 735/800	620	80	767	222,7	6,46
R	-	735/805 x 640	-	-	-	-	-
3	I	700 x655	620	35/105	640	255,1	5,23
4	I	610 x 680	620	90	655	236.2	7,04
5	I	530 x700	620	80	682	222,7	7,24
6	I	450 x 720	620	80	707,5	222,7	6,51
R	-	720 x450	-	-	-	-	-
7	II	600 x480	592,5	120	467,5	266,6	7,68
8	II	500 x505	592,5	100	497,5	243,4	8,25
R	-	505 x 500	-	-	-	-	-
9	I	390 x 525	620	115	514	267,0	4,76
10	I	330 x 545	620	60	536,5	192,8	7,36
R	-	545 x 330	-	-	-	-	-
11	IV	430 x355	605	115	342,5	263,7	9,30
12	IV	350 x 375	605	80	362,5	220,0	10,88
R	-	375 x 350	-	-	-	-	-
13	III	280 x 380	600	95	365	238,7	7,94

3. RESULTS

According to the analysis of the twisting, bending, and contact pressure tensions we have already highlighted in Tables no. 2, 3, 4:

- the bending tensions (table 2) are the highest in case of the outside layer and go deep to 15-20 mm underneath the surface of the gauge; the highest values are σ_I = 8,55 daN/mm^2, and they grow smaller as they get closer to the core of the gauge. By the time they reach its axis, they are null.
- the twisting tensions (table 3) produced during the rolling (rolling time) have higher values. The highest value is τ_I = 2,29 daN/mm^2 at the surface of the gauge. Thus, these tensions could be ignored. But still, we have to point out the general rule for determining the tensions inside the roll, where they get more and more small; and by the time they reach the axis they are null.
- other tensions caused by stress and contact pressure (table 4) influence only the surface of the gauges, in the area of the deformation core, and their highest value is σ_{pc}= 10,88 daN/mm^2.

Generally, we could point out that mechanical tensions we have used for classical resistance calculus for the rolling process have insignificant values. These calculations are not valid in case of real industrial processes. If we consider the thermal tensions, we would be able to come up with a complete study about the genuine industrial process situations, because thermal influences are the main cause for thermal fatigue in case of lamination rolls. Those influences are also valid in case of favorable operation conditions when we use the lamination rolls.

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