S. A. MASHEKOV, A. E. NURTAZAYEV, E. Z. NUGMAN, A. S. MASHEKOVA, M. L. RAKHMATULIN, A. I. POLESHCHUK

ISSN 0543-5846
METABK 56(3-4) 367-370 (2017)
UDC – UDK 621.771 – 539.417:621.822:621.41=111

INFLUENCE OF THE STANDS CONSTRUCTION ON THE VIBRATION OF THE WORKING AND BACKUP ROLLS OF THE LONGITUDINAL-WEDGE MILL

Received – Primljeno: 2017-03-14
Accepted – Prihvaćeno: 2017-05-10
Original Scientific Paper – Izvorni znanstveni rad

New design of the mill is suggested in the work. MSC.VisualNastran 4D software of the finite element analysis was used for computer modelling of the longitudinal cold rolling process of the strip and it was calculated the stress-strain state and the vibration of the heavy-duty elements of the longitudinal-wedge mill (LWM) with bearings stands and without the bearings stands. As a result of modelling it was established that strips, rolled in the LWM without bearings stands, have longitudinal and transverse flatness, that is the consequence of the working rolls vibration. It was shown that strips, rolled in the LWM with the bearings stands, have not got any wavy surface. It was proved that during rolling in LWM the dangerous vibrations do not fall into the working space of the external loads, therefore construction of the new mill is good enough, in terms of strength at the vibrations.

Key words: longitudinal-wedge mill, rolls, bearings, thin strip, vibration

INTRODUCTION

It has to be noted that currently main direction of the quality improvement of the hot-rolled strips is providing their minimal transverse and cross gage interference, as well as the flatness shape of it [1,2]. For the time being these problems are solved due to creation of the different automatic strips quality control systems of the rolling equipment construction and rolls systems, shift of the working rolls, their intersecting, counter-bending and oil injection system, different options of asymmetric rolling.

Requirements of the customers regarding the quality of the of the strips become higher due to new technologies of automotive steel paint [3]. As a result, a large number of rolls of the 1-st group of surface finish are rejected because of periodic defects as both gage interference, and the surface roughness (change R_a < 4…5 μm). In some mills, there are periodic defects of the roll surface caused as a result of dynamic processes in the stands at a resonant vibration of the mill.

On all mills increase of the vibration to dangerous levels limits productivity, reduces the quality of the finished strips for such indicators as longitudinal gage interference and periodic surface defects among it ribbing, cross shadowbands, as well as increases the consumption of the rolls [4].

According to [4-6] the excitation of vibrations is associated with the presence of the kinematic oscillation sources in the mill, including parametrically excited when changing the stiffness of the elastic connections of the system and friction fluctuations in the deformation zone. Among the latter hypothesis is also considered horizontal oscillations of the rolls carriages in the gap area of the body stand.

In [3,7-9] the different causes of vibration and the mechanism of it amplification, also associated with it periodic surface defects of the strip and the rolls was discussed.

Analysis of these publications about resonant vibration on the high speed cold rolling mills shows, that this problem can be solved only with complex measures, primarily by using stationary vibration control systems of the stands equipment, including visual inspection of periodic defects on the surface of the rolls and the strip. Widely used in practice one of the way dealing with a resonant vibration at various mills is to reduce the rolling speed, which reduces the amplitude of the vibrations, but at the same time reduces the average operating speed of the mill [10].

Another obvious solution is to install damping devices between the carriages of the upper backup rolls and hydraulic gap control (HGC), which are setup on the main frequency of the resonant vibrations in the stand [6].

In our opinion the best method to control the transverse gage interference, flatness and decrease of the resonant vibration is to create the mill with a rational design, optimization of the draft modes and force of hot rolling. With this aim we have developed a new rolling mill construction [11] that allows to assign rational technological parameters of rolling.

S. A. Mashekov, A. E. Nurtazayev, E. Z. Nugman, A. S. Mashekova, M. L. Rakhmatulin, A. I. Poleshchuk – Kazakh National Research Technical University after K. I. Satpayev, Almaty, Kazakhstan

The aim of the work is to calculate the vibration of heavy duty components of the longitudinal-wedge mill (LWM) by using computer simulation and on the basis of the calculations to adjust the construction of new mill stands.

EQUIPMENT, MATERIALS AND METHOD OF THE EXPERIMENT

We have developed 5-stands longitudinal-wedge mill (LWM) with simple construction for rolling thin strips with accurate geometrical dimensions. The strength and stiffness of the working and backup rolls of the new mill were investigated during cold rolling of the strips from D16 alloy with the size of 0,7 × 400 mm.

At this there are two backup rolls in the first three stands and four backup rolls in the last two stands. In addition, the distance between the stands are increased by the amount of forward slip, and the distance between the working rolls is adjusted by worm push mechanisms located above and below mill body.

The dynamic computer model of the LWM was developed by the finite element analysis software PATRAN NASTRAN [12]. The deflection, vibration and stress-strain state (SSS) of heavy-duty elements of the mill stands were calculated. Also, the LWM with and without bearing stands was investigated.

3D geometrical model of every detail of the mill stand was created in the COMPASS software and then nodes of the working stand was assembled. Six-node and eight-node 3D finite-element mesh was marked down by the Mesh Seed option and the SSS was determined. Elastic connections between the nodes of the stand was modelled by the spring - damper element CBUSH. The force of friction between the rolls was equal to 0,0868, and the initial roll temperature was 20 °C. The computational model of each spherical roller bearing includes three types of components: an outer and an inner ring, and two rows of rollers by 18 in each. Finally, the level of the obtained elastic deformations and stresses in the stands with respect to the required strength criteria was assessed and appropriate changes in the design of stands were entered.

It has to be noted that the rolls were attached to the bearing journal of the bearing unit by three degrees of freedom T_x, T_y, T_z. The material of the rolls is 9X1 steel with the following mechanical properties: modulus of elasticity – 2,1 + 11 Pa; Poisson’s ratio – 0,283; shear modulus – 8,1839 + 10 Pa. 40HS steel is the material of the stands body with the modulus elasticity 214×10^3 MPa, Poisson’s ratio 0,3. The material of the other parts of the mill is steel 45 with the following mechanical properties: modulus of elasticity – 2,034E + 11 Pa; Poisson’s ratio – 0,29; density – 7833,394 kg/m^3. Semi-finished product with the h_0 = 3,5 mm was used as the initial preform. Friction coefficient was equal to 0,25 [13].

Table 1 shows the initial data used for rolling strips in every stand of new mill.

Table 1 **Initial data used for rolling strips**

	1	2	3	4	5
Height of the strip after rolling, mm	2,576	1,708	1,148	0,84	0,7
Draft, %	26,4	33,7	32,8	26,8	16,7
Speed of the strip, m/s	0,5	0,68	1,03	1,526	2,085
Maximal force of rolling, MN	0,095	0,085	0,073	0,054	0,033
Diameter of the backup rolls, mm	220				
Diameter of working rolls, mm	180	150	125	106	94
Length of the deformation zone, mm	8,992	8,07	5,92	3,93	2,3

RESULTS AND DISCUSSION

Studies have shown, that during hot rolling of the strips made of D16 alloy with the width of 400 mm, large by the amount of equivalent stresses do not occur along the cross section of the rolls. Maximum von Mises stresses for the stands 1, 2, 3, 4 and 5 are equal to, respectively: 172,5; 163,4; 151,8; 133,3; 136,6 MPa. Moreover, the maximum stresses occur in the rolls necks of the mill. Maximum received equivalent stress does not exceed the maximum allowable value of the tensile strength for 9Kh1steel (880 MPa).

Finite element model calculations showed that the maximum received values of equivalent stress on the frame of stands (first stand – 112,4 MPa, the second stand – 97,3 MPa, the third stand – 79,83 MPa; the fourth stand – 68,94 MPa; the fifth stand – 56,16 MPa) does not exceed the maximum allowed value 981 MPa of the tensile strength for the material. In this case, the maximum values of the stresses are observed at the housing rocker plate.

As a whole, distribution of the coefficient of resistance for the rolls and the frame struts satisfies the strength condition, when the adopted safety factor is 5 and 10, respectively. The weakest point of the frame is the vertical entablature.

It has to be noted that the calculated values of equivalent von Mises stress does not exceed the upper limit of permissible contact fatigue stress. This fact suggests that even a small deviation from the technological process will not lead to the appearance of defects on the surface of the rolls: cracks, pitting, spalling.

Roll carriages are elastically deformed in the vertical and horizontal planes and they are rotated of slight angle (γ_{max} = 0,00034^o) with respect to the rolling axis. The elastic displacement in the direction of load action for the roll carriages, located from the side of the drive roll, is 1,2 times greater than for the roll carriages, located from the opposite side of the roll.

It is known that during equipment operation the resonant vibrations, which are aroused from the coincidence of itself frequency and external forces frequency, is particularly dangerous. Therefore, it is important to determine the frequency of the external forces. One of the major cause of the resonant frequencies in the construction of rolling mills is the operating frequency of the working rolls drive rotation, which extends the vibration to the mill.

Research has shown that an increase of the rolling speed of new mill without rolling element bearings leads to the relatively high rise of the dynamic loads in the main nodes of the stands and the drive line. Mechanical vibrations of the stands nodes and the drive lines is the reason of the vibration. The study show that in the last stands of the mill compare with the first three stands appear relatively large by the amount forced vibrations due to periodically changing both external forces and rolling speed (Table 2).

The reason of rather high resonant vibration in the rolls of the proposed mill is that it has not rolling element bearings and not horizontal arrangement of the spindles of the LWM drive. At this, the rolls nodes had not sufficiently high rigidity in the horizontal plane. This resulted in the displacement of the work rolls in the horizontal plane. Thus, even small gaps between the bearings, roll carriages and housing windows, caused by the tolerance clearances and wear, led to a horizontal displacement of the vertical axial plane of the work rolls with respect to the backup rolls. In other words, working rolls was in an unstable position and their axes were skewed. This has led to negative consequences as an unpredictable fluctuation of the rolls gaps.

It should be noted that although the construction of the LWM without stands bearings does not allow to occur harmful resonant vibrations, we have attempted to change the construction of the mill in the way its own frequency was brought out of its operating range and then evaluate the resonant vibrations, thereby get high stock before the appearance of the abovementioned variations. Therefore, in the construction of new mill we used rolling element bearings and spindles of special design.

Studies have shown, that during rolling, in the last two stands of new mill with the rolling element bearings, relatively large in the magnitude forced vibrations occur. However, their value is significantly lower, than during rolling on the mill without rolling element bearings (Table 2).

Thus, rotation of the rolls through the rolling element bearings allowed to position the spindles strictly horizontally. Whereas use of the spindles of special design allowed to eliminate horizontal displacement of the working rolls. At the same time LWM with the bearings stands allowed to transmit torque to the working rolls of the mill stands with high stock to resonant vibrations (Table 2). It is expected that all this will contribute to the production of strips with precise geometric dimensions and without surface defects.

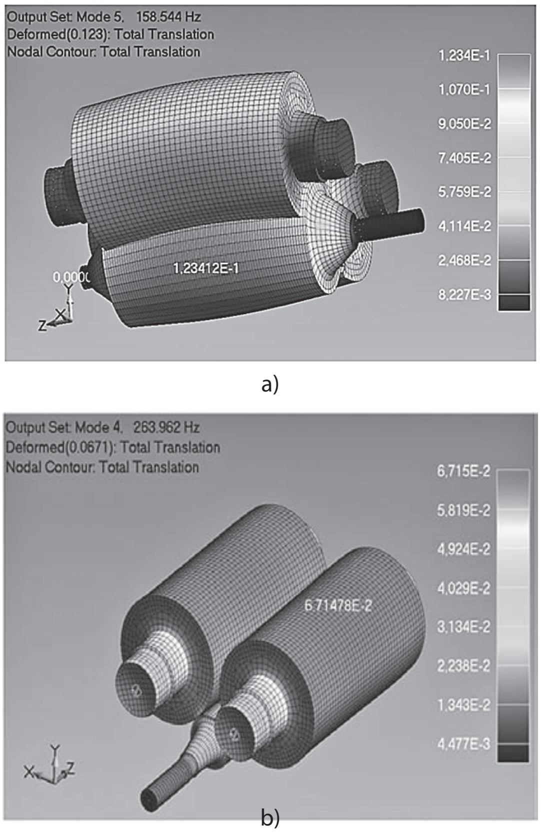
The following results were gathered in assessing the natural frequencies of the frame assembly with a roller assembly: vertical vibrations of tension-compression of the housing post with the rolling nodes and the bar corresponds to the frequency range of 28 - 136 Hz, whereas forced oscillation of the roll unit equals to 6 - 60 Hz.

The visualized results show the way of the rolls vibrations distribution to the bending of the working and backup rolls (Figure 1). Studies have shown that the pattern of distribution of total displacement in three directions consistent with the deformed shape of the rolls. The maximum deflection in the backup rolls changed in the range from 0,113 to 0,0964 mm and in the working rolls changed from 0,133 to 0,0378 mm.

The modeling found that the strips, rolled on the LWM without rolling elements bearings, have a wavy surface with the oscillation period of about 15 – 20 mm. It should be noted that such a wavy surface is not detected in the strips rolled on the LWM with the bearings stands. According to modern ideas about the process of the longitudinal rolling appearance of the waves on the surface of the strips is explained by the relatively large vibrations of the rolls.

Table 2 **The values of own and forced oscillations during rolling strips on the longitudinal-wedge mill**

Stands No.	The natural frequencies of the mill stands, Hz						The frequency of the forced oscillation, Hz	High stock before the resonant frequency:					
	1-st	2-nd	3-rd	4-th	5-th	6-th		1-st	2-d	3-rd	4-th	5-th	6-th
Longitudinal-wedge mill with the rolling element bearings													
1	55,12	70,02	143,7	214,7	266	403,1	5,71	9,65	12,3	25,2	37,6	46,6	70,5
2	58,27	71,45	136,3	248,8	260,3	323,3	9,07	6,43	7,88	15,1	27,4	28,7	35,6
3	61,39	72,67	132,7	252,7	258,3	266,9	16,48	3,73	4,41	8,06	15,4	15,7	16,2
4	84,58	162,8	240,3	261,3	290,9	302,6	30,52	2,77	5,34	7,88	8,56	9,53	9,92
5	86,91	167,7	242,4	263,9	282,9	303,5	55,60	1,56	3,02	4,36	4,75	5,09	5,46
Longitudinal-wedge mill without the rolling element bearings													
1	34,21	40,17	93,63	164,1	189,3	213,5	6,56	5,22	6,12	14,2	25,0	28,8	32,5
2	43,81	57,19	98,56	189,7	212,4	233,2	12,17	3,6	4,7	8,1	15,6	17,5	19,1
3	53,52	60,44	118,7	201,5	227,3	251,6	19,72	2,71	3,07	6,02	10,2	11,5	12,8
4	64,61	168,4	183,8	221,4	251,3	272,1	38,61	1,67	4,36	4,76	5,74	6,51	7,05
5	71,82	158,5	213,3	237,8	263,7	283,8	63,24	1,14	2,51	3,38	3,76	4,17	4,49



a – fifth stand without rolling elements bearings;
b – fifth stand with rolling elements bearings

Figure 1 The shape of the rolls fluctuations due to vibration load during rolling strips on the LWM with (*a*) and without bearing stands (*b*)

CONCLUSION

It is shown that the use PATRAN NASTRAN software of finite element analysis is an effective means to study the influence of different rolls vibration on forming the strips thickness during cold rolling.

It was established that strips, rolled on the LWM with rolling elements bearings, have not got a wavy surface.

It is proved that when rolling in the LWM critical vibrations do not fall within the operating range of existing external loads, so the construction of the new mill is good enough in terms of strength during vibrations.

REFERENCE

[1] S.S. Voronin, D.Yu. Usatyi, V.R. Gasiyarov, A.A. Radionov. The use of the axial shift system of rolls to correct the roll gap on hot plate thick-rolling mills. Russian Internet Journal of Industrial Engineering, 3 (2015) 1, 45-49. http://indust-engineering.ru/issues/2015/2015-1.pdf

[2] A.N. Kulik, A.V. Danko, G.V. Kolesnikov, A.S. Kartavenko. A method for reducing the transverse thickness variation by asymmetric rolling. Materials processing by the pressure, 20 (2009) 1, 225-228.

[3] E.A. Garber, V.P. Naumchenko. Investigation of the causes of oscillation in the stands of continuous-rolling mill. Rolling production. 1 (2003), 10 – 12.

[4] P.V. Krot. Investigation of the “ribbing” defect and high-frequency oscillations of cold-rolling strip mills. Proceedings, IV conference proceedings of rollers, Magnitogorsk, 2001, “Chermetinformation” OJSC, Moscow, 2002, pp. 225-227.

[5] A.N. Marjuta, P.V. Krot. High frequency rolling mills chatter – mathematical identification and simulation. Proseedings, 1st International Symposium on “Multi-Body Dynamics Monitoring and Simulation Techniques”, University of Bradford, UK, 1997, pp. 407-419.

[6] P.V. Krot. Parametric oscillations in rolling mills. Proceedings, Collection of scientific works of NMU, Vol. 3, National Mining University, Dnipro, 2002. Pp. 15-21.

[7] G. Helekal, R. Luftensteiner, C. Riegler, H. Aigner. Automatic vibration analysis and trending for complex drives. Steel Technology, (2002) 3, 43 – 49.

[8] Yu. V. Rybakov, G.N. Subbotin. Determination of vibration sources causing resonance phenomenon in cold rolling mills. Rolling production, 1 (2003) 10, 13-16.

[9] B.R. Hardwick. A technique for the detection and measurement of chatter marks on roll surfaces. Steel Technology, (2003) 2, 64–70.

[10] B.R. Hardwick. Application of vibration monitoring to cold mill process. Iron and Steel Engineer, (1999) 2, 39-45.

[11] S.A. Mashekov, B.N. Absadykov, E.Z. Nugman, A.S. Mashekova, E.P. Orlova, M.E. Isametova, N.K. Smagulova, A.I. Polechuk, M.L. Rakhmatulin, M.V. Gladkov. Continuous mill for rolling strips made of steel and alloys. Republic of Kazakhstan, Patent 31750.

[12] E.K. Rybnikov, S.V. Volodin, R.Yu. Sobolev. Engineering calculations of mechanical structures in MSC.Patran-Nastran. Parts 1, 2. MSC.Software Corporation, 2003, pp. 130 and 174.

[13] G.S. Nikitin. Theory of continuous longitudinal rolling. Moscow. MNTU after N.E. Bauman publisher, 2009, pp. 399.

Note: The responsible for English language is the lecturer of University Almaty, Kazakhstan.