UDC 621.771.252

Analysis of the effect of operating parameters of hydrodynamic bearings on the accuracy of rolling in finishing units of wire mills

Samokhval Volodymyr Mykhailovych

Ph.D. in Technical Science
Dniprovsk State Technical University
volsamokhval@gmail.com

Shtoda Maksym Mykolaiovych

Ph.D. in Technical Science
Dniprovsk State Technical University
maksshtoda@gmail.com

Havrylin Serhii Yuriovych

Ph.D. in Technical Science
Dniprovsk State Technical University
Sgavrilyn@gmail.com

Heimur Kostiantyn Heorhiiovych

Engineer
Dneprovsky integrated Iron and Steel Works

Introduction. The accuracy of rolling is determined by a complex of various factors that can be conditionally divided into technological and constructive ones. The technological factors include the shape of the pass, the deformation degree, the velocity and temperature modes of deformation, and others. The constructive factors are determined by the characteristics of the use equipment. In the most cases, the constructive factors are relatively stable, so the degree of their effect on the accuracy for each of the rolling mills is determined experimentally and taken into account indirectly through the technological factors. For example, when setting up a rolling mill for rolling a certain shape, the gaps in each of the stands are fixed taking into account the "stand spring", i.e. the elastic deformations caused by the rolling force [1, 2].

But in some cases, in case of significant changes in the technological factors, the ambiguous impact of the constructive factors can be observed. For example, almost a tenfold technology-driven increase in the speed between the first and last modules is observed for the finishing units of the wire mills. At the same time, the modules of rolling forces can differ by a factor of 3-5 with the opposite distribution, that is, they decrease from the first modules to the last ones. For the standard stands, such significant changes in the technological factors are not critical and, if necessary, they can be compensated by appropriate design solutions (roll diameters, engine power, spindle sizes in accordance with the appropriate moments, etc.). For most units of the wire mills, this is not possible because all the modules have the same design and common drive [3]. Thereafter, when designing modules, their design parameters are selected according to the conditions of the operability assurance for the entire possible range of changes in the main technological parameters. Under this approach, the possibilities of units regulation to ensure the required rolling accuracy are very limited.

Problem statement. The work objective is to investigate the degree of influence of the structural features of the unit modules and the accuracy of rolling under the conditions of the technology-driven changes in rolling parameters.

From the operating experience of the existing units of the wire mills, the size regulation of the finished wire rod is known to be possible due to the changes in the gaps in the last two modules and due to the changes in the gaps and the size of the semi-finished rolled stocks in the first modules. The gaps adjustment for other modules is considered inefficient due to the rigid kinematic connection from the common drive. The degree of the gap changes effect in the first or last modules is not separately investigated, but is selected until the required size of the wire rod is achieved. In this case, there are certain contradictions. For example, when rolling a wire rod with a diameter of 5.5 mm at the mill stand 400/200 of JSC "DMK", the gap in the last tenth module of the unit 200 should be 1.21 mm according to the calibration scale. The actual gap in this module is from 1.45 to 1.57 mm; it ensures the production of wire rods of specified sizes. Thus, unlike the well-known consideration of the "stand spring" [4], the opposite phenomenon is observed, namely when adjusting the module, a larger gap should be fixed. Such a difference in the configuration of the unit modules can be explained only by the certain design features of the modules, in particular, by various conditions of adjustment and regular operation.

To analyze such structural features of the unit 200, the calculations of the main technological parameters of rolling in the modules when rolling a wire rod with a diameter of 5.5 mm with low-carbon steel were carried out. According to the calculated technological parameters, the operation parameters of the hydrodynamic bearings of the basic module supports for the setting conditions and for the normal mode were determined.

Results. The main technological parameters were calculated for the current calibration of unit 200 of JSC "DMK" (Table 1). The dimensions of the rolls in the passes were determined using the V.K. Smirnov's technique for calculating the extension and other geometric parameters [5]. In this case, the diameters of the rolls (washers) along the collars for all the modules were assumed to be equal to 210 mm. The dimensions of the rolls and the resulting speed rolling rate are calculated in the Table 1.

The temperature of the rolls and the rolling force of the unit modules were determined by the methods of L.V. Andreiuk [6] and V.K. Smirnov [5]. For a more reasonable selection of the rolling power parameters, the calculated data were compared with the experimental data obtained on one of the Japanese wire mills [7]. Based on the results of the comparison of the distribution of the rolling force on the unit modules shown in Fig. 1, the values used for further calculations were obtained by Smirnov's method, as these values the most closely correspond to the experimental data of Noguchi [7].

Washers with passes are cantilevered to rolls, which are situated in the module housing on two hydrodynamic bearings. The nominal diameter of the bearings is the following: for the main support (closer to the washer) - 140 mm; for the second support - 111 mm. The width of the bearings is 105 mm and 54 mm respectively. The distance between the centers of the supports is 309 mm, the distance from the center of the main support of the pass axis (the plane of action of the rolling force) is 110 mm. Through the cantilever placement of the washers with passes, the load on

Table 1. Parameters of passes, rolling and speed rolling rate in unit 200

Module number	Pass dimensions			Rolling dimensions		Section area, mm^2	Lengthening coefficient	Rolling speed m/s
	Width W_p	Depth of incision h_{in}	Gap g	Height	Width			
1	23.65	4.6	1.94	11.14	20.65	185.5	1.209	13.4
2	13.83	6.14	1.51	13.74	13.79	149	1.245	16.6
3	18.89	3.5	1.85	8.85	16.97	120.4	1.238	20.6
4	11.45	4.95	1.13	11.03	11.03	95.6	1.259	25.9
5	16.85	2.70	1.22	6.62	13.9	76.0	1.258	32.6
6	8.97	3.70	1.36	8.66	8.76	59.8	1.271	41.4
7	13.56	2.10	1.06	5.26	10.97	48.2	1.241	51.4
8	7.27	2.92	1.11	6.95	6.95	37.9	1.272	64.4
9	10.24	1.60	0.99	4.19	8.75	30.2	1.255	80.8
10	5.76	2.18	1.21	5.57	5.57	24.2	1.238	100

the main support exceeds the rolling force. For these distances between the supports and the passes axis, this excess is (309 + 110) / 309 = 1.356.

The oil is fed into the bearing through the hole in the liner. The hole diameter is 12 mm. The axis of the hole and the plane of the radial load action form an angle of approximately 120°. The supply of an oil to the working zone of the bearing is provided by a partial circular groove with a depth of 1.5 mm, and two bins with a radius of 25 mm and a depth of 1.5 mm. The width of the bins is 95 mm, the width of the circular groove is 16 mm. One of the bins is placed on the opposite side of the plane, where the minimal gap is located, and the other one is placed approximately at an angle of 90° from this plane.

The design of the hydrodynamic bearings of all the modules of the unit is of the same type, namely each of them consists of a polished surface of the shaft and a cylindrical steel liner fixed by thermal and press fit in the housing. The working surface of the liner has a coating of lead-tin bronze.

The analysis of bearing operating parameters is carried out for the most loaded basic supports of the modules. Taking into account the tolerances, the diameter of the bearing shaft of the basic support varies from 139.880 to 139.855 mm, the inner diameter of the liner varies from 140.24 to 140.174 mm. Thus, the gap in the bearing is from 0.294 to 0.385 mm with an average value of 0.333 mm. The surface roughness of the shaft is Ra = 0.2 μm, the liner surface is Ra = 1.6 μm. Analysis of bearing operating parameters was performed by the calculation method using the standard technique [8]. The calculation method determined by this standard is applied for the round cylindrical bearings of 360, 180, 150, 120 and 90º operating in a stationary mode.

The method is based on the basic hydrodynamics theses, according to which the criterion (number) of Sommerfeld is used to estimate the loading capacity of hydrodynamic bearings. This number is determined by such parameters of the bearing as the pressure from the external load, the viscosity-velocity parameter and the relative gap.

At the same time, the Sommerfeld number is a function of the bearing geometric parameters, such as relative eccentricity, relative length and angle

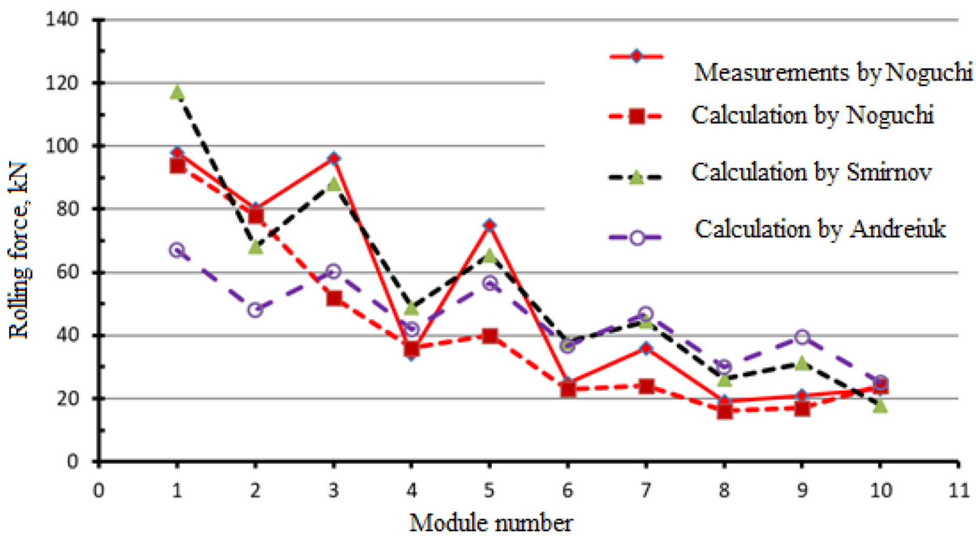

Figure 1. Calculated and experimental values of the rolling force in the modules of the finishing unit

of contact. Functional dependencies between the Sommerfeld number, the given geometric parameters, and some other bearing parameters are determined by the standard [9].

The calculation procedure involves checking the laminarity of the oil flow according to the Reynolds criterion, checking of permissible mean pressure, and calculating the Sommerfeld number. The value of the number is used to determine the other parameters by dependencies represented in the form of graphs and tables. For example, for the calculated value S_o, let us determine the relative eccentricity ε as a function $\varepsilon = \varphi(S_o, B / D, \Omega)$. The obtained value ε makes it possible to determine the minimum thickness of the oil film, the friction coefficient, the frictional force, the frictional power, the intensity of the heat flow from the frictional forces, and the temperature of the bearing or lubricant at the output.

Considering that the circular groove with bins are placed on the arc of 152º, an angle of contact 180º was assumed for calculations. Calculations were carried out taking into account the filling of the entire gap (loaded and unloaded sections) of the bearing with oil, the supply of oil through one hole, two bins and a partial circular groove. In the calculations, the oil was assumed to be supplied at a pressure of 0.35 MPa.

Calculations of operating parameters of bearings of basic support at the lubricant supply temperature of 60 °C and at the motor speed of the unit of 1350 rpm, according to the technique [10], are given in Table 2.

From the data obtained, we should note a quite significant minimal film thickness of the oil, which varies from 9.85 µm in the first module to 110.82 µm in the last tenth one. For the above-mentioned roughness parameters of the bearing surfaces, the oil film thickness is critical; from the viewpoint of providing the hydrodynamic lubrication rate, the film thickness of the oil is 4.84 µm.

The considerable thickness of the oil film in the operating area of the bearings, especially for the latest modules of the unit, affects significantly the gap in the passes. As shown in the Figure 2, in case of the absence of rotation, the shafts fall on the surface of the liners under the influence of their own weight. In this case, the gap on the washers collars will be minimal. If the separating force F acts on the shafts in case of the absence of rotation, the lower shaft will not change its position, and the upper one will move upwards by the value of the diametrical gap 2δ. For the considered bearings of the basic support, the average value of the bearing gap is 0.34 mm (Figure 2, b).

Due to the rotation of the shafts in the operating area of the bearings, a film of oil will be formed, figuratively speaking, "the shafts will run over". It results in a reduction in the distance between the shaft axes by the value equal to double thickness of the oil film. However, the total movement of the axes from the separating force and from the formed oil films is (2δ – 2hmin).

For example, in case of the module adjusting, which reduces to the specified gaps according to the approved calibration scale, the gap is determined by rolling of wire at “creep” speed through the washer collars (electrode of 3 mm in diameter made of mild steel). In this case, the gaps in the bearings of the upper shafts are selected, but the thickness of the film is minimal because of the insignificant rotation frequency of the shafts.

"Creep" speed corresponds to 10 rpm rotation frequency of block motors. Calculation of minimal oil film thickness was made for this speed according to the method of table 2 and considering 20 kN

Table 2. Parameters of 1350 rpm motor bearing operation

Module #	Frequency of shaft rotation rpm n_j	Angular velocity c^{-1} ω_j	Velocity of sliding m/sec U_j	Number of Reynolds Re	Radial load kN F	Average Pressure N/mm² p_{cp}	Number of Sommerfeld So	Eccentricity ratio ε	Minim. thickness of film um h_{min}	Coeff. of friction f
1	1214.2	127.08	8.90	39.97	159.1	10.83	14.73	0.94	9.85	0.0016
2	1532.0	160.35	11.22	50.44	92.2	6.27	6.77	0.83	28.60	0.0026
3	1858.6	194.54	13.62	61.19	119.4	8.13	7.22	0.84	27.02	0.0025
4	2370.7	248.13	17.37	78.05	66.3	4.51	3.14	0.72	47.09	0.0040
5	2937.0	307.41	21.52	96.69	88.8	6.04	3.40	0.73	45.20	0.0038
6	3760.4	393.59	27.55	123.80	51.3	3.49	1.53	0.69	52.10	0.0061
7	4614.6	482.99	33.81	151.92	60.2	4.10	1.47	0.68	53.67	0.0063
8	5900.0	617.53	43.23	194.24	35.6	2.42	0.68	0.52	81.32	0.0098
9	7346.6	768.95	53.83	241.86	42.4	2.88	0.65	0.51	82.91	0.0101
10	9155.8	958.31	67.08	301.42	24.2	1.65	0.30	0.35	110.82	0.0159
	Critical values:			838.68		10				

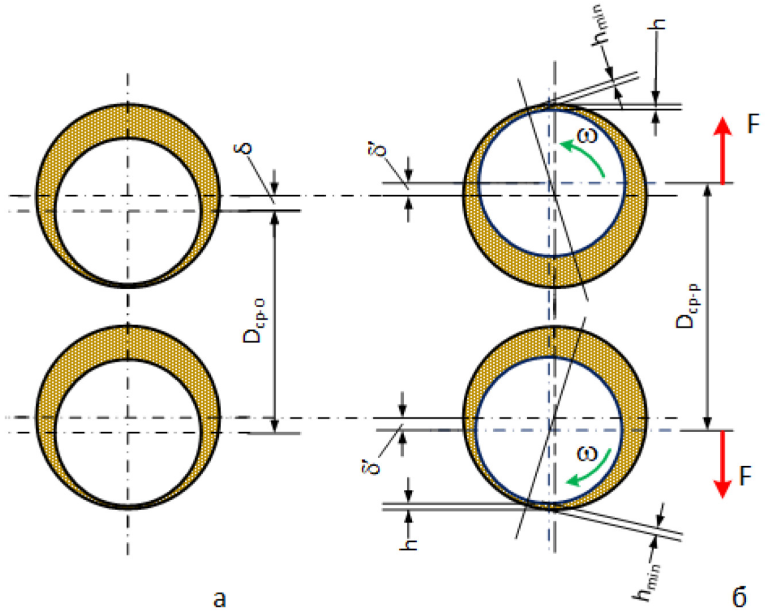

Fig. 2. A diagram of hydrodynamical bearings of unit blocks

separating force. Equal oil film thicknesses for all modules were obtained; the thickness was 2.24 um. Such insignificant oil film thickness can't drastically change the distance between shaft axes. Therefore, if the upper roller bearing clearance is completely eliminated (0.34 mm), the upper shaft axis "influences" on oil film thickness in the same was as the lower shaft axis does. Thus, the distance between axes makes

$$(2\delta - 2h_{min}) = (0{,}34 - 2 \cdot 0{,}00424) = 0{,}3315 \text{ mm.}$$

Nevertheless, in case of continuous operation of a block shaft, both rotation frequency and oil film thickness will be much greater. For instance, the 10th module clearance established due to "creeping" speed will decrease and make $(2\delta - 2h_{min}) = (0{,}34 - 2 \cdot 0{,}11082) = 0{,}118$ mm.

Therefore, the clearance difference of an established and continuous process will make 0.3315 - 0.118 = = 0.213 mm. Such value of the clearance change can significantly influence on ready section dimensions and even bring them out of limits.

Calculated clearance differences of modules at stable operation and various temperatures of oil supply are given in fig. 3.

It is obvious that the less temperature of oil supply, the greater oil film thickness; it leads to clearance difference growth. At higher temperatures, both oil film thickness and clearance difference decrease.

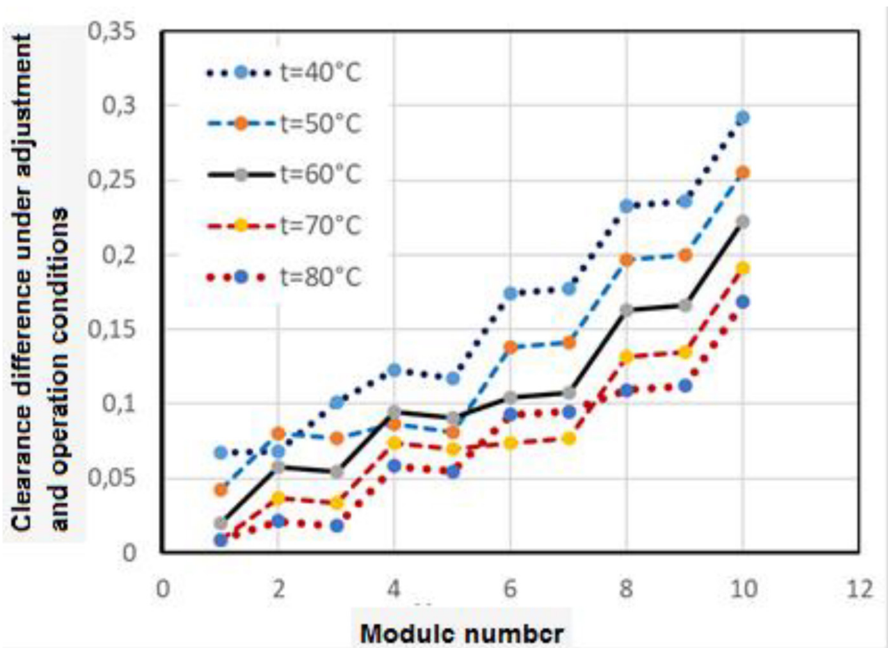

Fig. 3. Clearance difference at stable operation of unit module main support bearings

However, velocity or shaft rotation frequency is the dominating technological factor, which defines clearance difference. Clearance difference almost proportionally grows together with increasing velocity of unit modules. It is clear that the highest difference in clearances is seen in the last unit blocks; therefore, the discovered designed feature of bearings shall be considered in this case.

In particular, clearance difference in 8-10 modules exceeds 0.1 mm at any temperature of oil supply. At oil supply temperature of 60ºC and stable operation in modules 8-9-th, it is recommended to set clearance by 0.16 mm greater than the clearance indicated in the calibration table, whereas for 10-th module clearance shall be increased by 0.21 mm. For instance, if 1.21 mm clearance shall be provided in 10-th module, it is necessary to set 1.42 mm clearance, when adjusting.

Conclusions. Design features of hydrodynamic slider bearings of unit block main supports in case of certain relation of process parameters, for instance, velocity, influence on oil film thickness, which can rise up not only to hundredths, but to tenths of millimetre. Such oil film thickness can influence on roller clearance and, as a result, on section precision.

According to made calculations of technological parameters and bearing operation parameters, it has been defined that for a quick rolling process in a 200 unit, and if the other three modules are adjusted, it is necessary to consider the decrease of roller clearance due to significant oil film thickness in main support bearings.

If the 200 PAT DMK unit is adjusted to compensate the significant oil film thickness, it is recommended to set clearance in 8,9-th modules by 0.16 mm more and in 10-th module by 0.21 mm more than in calibration tables.

References

1. In particular, clearance difference in 8-10 modules exceeds 0.1 mm at any temperature of oil supply. At oil supply temperature of 60°Ivanchenko F.K., Calculation of machinery and mechanisms of rolling workshops: [Textbook for university students] / F.K. Ivanchenko, V.M. Grebenik, V.I. Shiryaev. – Kiyev: Higher school. – 1995. – 455 pages.
2. Vydrin V.N, New aspects of rolling precise sections/ V.N. Vydrin, V.G. Dukmasov, V.S. Nagornov – Chelyabinsk: Uzhny-Ural publishing house, 1970. – 107 pages.
3. Kugushin A.A., High-speed wire rolling/ A.A. Kugushin, Y.A Popov -Moscow: Metallurgy, 1982. – 144 pages.
4. Rolling production process: reference book in 2 volumes / [M.A. Benyakovsky, K.N. Bogoyavlensky, A.I. Vitkin at al.]. – Moscow: Metallurgy. – 1991 –. - Book 1. – 440 pages.
5. Smirnov V.K., Calibration of milling rollers: [Textbook for universities]/ V.K. Smirnov, V.A. Shilov, Y. V. Ignatovich. – Moscow : Metallurgy, 1987. – 368 pages.
6. Andreyuk L.V. - 8 books. : Rolling theory. Materials of All-Union scientific and technical conference "Theoretical problems of rolling production", (Dnepropetrovsk, 25 – 28 September, 1975). – Moscow: Metallurgy, 1975. – pages 383 -385.
7. Noguchi Yukio, Harutoshi Ogai, Kunihiko Okamura, Takayoshi Hizume, Koji Tanabe, Koji Yoshimura (1999) Characteristics of Continuous Wire Rod Roll Shaft and Precision Roll Shaft System. Nippon Steel Technical Report. No 80, July. pp. 79–83. Available at: http://www.nssmc.com/en/tech/report/nsc/pdf/8016.pdf.
8. Hydrodynamic radial slider bearings that operate under steady state conditions. Circular cylindrical bearings. Part 1. Calculation method: DSTU GOST ISO 7902-1:2003 (GOST ISO 7902-1-2001, IDT). – [Valid since 01.07.2004]. – Kyiv: Derzhspozhivstandard of Ukraine, 2008. – 28 pages. –(National Standard of Ukraine).
9. Hydrodynamic radial slider bearings that operate under steady state conditions. Circular cylindrical bearings. Part 2. Functions used for calculation: DSTU GOST ISO 7902-2:2003 (GOST ISO 7902-2-2001, IDT). – [Valid since 01.07.2004]. – Kyiv: Derzhspozhivstandard of Ukraine, 2008. – 28 pages. –(National Standard of Ukraine).
10. Yershov S. The Analysis on Hydrodynamic Bearings Performance in High-Speed Modules of Wire Rolling Mills / SerhiiYershov, VolodymyrSamokhval, SerhiiHavrylin, YuriiAntonov, MaksymKutvitski //Metallurgical and Mining Industry. – 2017. - # 7. – pp. 36-45.