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The Analysis on Hydrodynamic Bearings Performance in High-Speed Modules of Wire Rolling Mills

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Abstract
The present article addresses hydrodynamic bearings performance in high-speed modules of wire rolling mills and is searching for the possibilities to increase the reliability of their work. In order to determine the performance parameters of hydrodynamic bearings, such as the minimum thickness of the lubricant film, the friction coefficient and the temperature of the lubricant heating, we used the calculation method based on the standard method as the parameters depend on the main production and operational factors. This allowed us to reveal that the most important factors are the speed and the temperature of the lubricant input while the load and the lubricant input pressure affect the parameters of the bearings to a small extent. At the lubricant input temperature of more than 85°C, there is a risk of the bearing damage in the last unit modules due to the violation of the laminar flow conditions. According to the results of the analysis, it is recommended to keep the input temperature of lubricants within the range of 60–80 °C. The observance of the proposed recommendations for existing bearing design and actual technological parameters enables the longer operation life of the module, reduction in the downtime related to the need for repairs, and reduction in the repair expenses.
Keywords: PLAIN BEARINGS, HYDRODYNAMIC LUBRICATION REGIME, DYNAMIC VISCOSITY, FRICTION, LUBRICATION INPUT TEMPERATURE, REYNOLDS NUMBER

Introduction

In the production equipment for rolling, both plain bearings and rolling bearings were widely used. Over the past decade, the scope of applications of plain bearings has decreased somewhat. In particular, in the construction of modern working rolling stands for the manufacture of long rolled stock, instead of the earlier common film-lubrication bearings (FLBs), only multi-row cylindrical rolling bearings are used. However, in the cases where high loading capacity is necessary at minimal dimensions, plain bearings remain indispensable.

An example of a successful application of plain bearings is the high-speed modules of rolling mills for wire manufacture. In such units, which usually incorporate from 4 to 10 modules of the same type, closed hydrodynamic plain bearings are used [1,2]. The peculiarity of the performance of such bearings in modules is that there is a significant difference in their performance parameters although the structural parameters are the same.

This could be illustrated by one of the examples as follows: the frequency of rotation of the working roll shafts if the ten-module units increases by 10 times in the course of rolling, while the rolling force on the contrary decreases by about five times. Accordingly, for the bearings of the first modules, extreme operating conditions emerge due to maximum loads and the lowest frequency of rotation, while for the latter ones - the critical parameter is the maximum speed of rotation, which exceeds 10 000 rpm. Moreover, due to the features of the passes, which are usually applied in rolling units, the force of rolling in oval passes is about 1.5 – 2 times higher than that in round ones. Furthermore, such a significant technologically determined difference between the parameters is accompanied by peak loads at stock gripping. These pick loads exceed the parameters of a stable process by 50% on average, and in some cases, the excess reaches 100%. Therefore, for certain modules, the conditions may arise that violate the design parameters of the bearings installed on the working roll shafts, which can lead to their damage and malfunction. In Figure 1, you can see an example of a typical damage of the main support bearing insert of the ninth module block.

Figure 1. External and internal surfaces of the main support bearing insert in module No 9, the insert is out of order due to working roll shaft damage

Accordingly, there arises the need for reconciling the operational parameters, that are determined by the technology of rolling, along with the design and operational parameters of the bearings with the objective to increase the reliability and their service life.

The aim of the research

The aim of this work is to analyze the parameters of hydrodynamic bearings performance at one of the existing high-speed wire rolling mills and to find the ways to increase the reliability of their operation.

Research Materials

The stated analysis is performed for a ten-module block possessing washers with a nominal diameter of 200 mm. The washers with passes are placed in a console on the working roll shafts of the modules. Each working roll shaft is in the housing of the module and is installed on two hydrodynamic bearings. The nominal bearing diameters are: for the main support (closer to the washer) – 140 mm; for the second support – 111 mm. The widths of the bearings are 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 to the pass axis (the plane of the roll shaft force action) is 110 mm.

All the hydrodynamic bearings under analysis are of the same type - each of them is formed by a polished surface of the roll shaft and a steel cylindrical bearing insert fixed with a thermal and press fit in the housing. The working surface of the bearing has a coating of plastic bronze.

Methodology and materials

The analysis of the parameters of the bearings performance has been carried out for the most loaded main support. Taking into account the tolerances, the diameter of the roll shaft bearing of the main support varies from 139.880 to 139.855 mm, the inside diameter of the bearing inserts varies from 140.24 to 140.174 mm. Thus, the gap in the bearing is from 0.294 to 0.385 mm, while an average value is 0.339 mm. The surface roughness of the roll shaft is Ra = 0.2 microns, the surface of the bearing insert is Ra = 1.6 microns. The other design features of the bearing are shown in Figure 2.

The supply of the lubricant to the bearing occurs through the opening (12 mm in diameter) in the bearing insert. The axis of the opening and the plane of radial load action make an angle of approximately 120°. The input of the lubricant to the bearing working zone is provided by a partial circular groove with the depth of 1.5 mm and two cradles made by a radius of 25 mm coming as deep as 1.5 mm. The width of the cradles is 95 mm while the width of the circular groove is 16 mm. One of the cradles is located on the opposite side of the plane where the minimum clearance is, and the other - approximately at an angle of 90° from this plane.

The parameters of the bearings performance for the analysis have been determined by the calculations made according to the standard [3], which calculation method is applicable to round-cylindrical bearings with the arcs of 360°, 180°, 150°, 120° and 90° for a fixed mode of operation.

The method is based on the principle laws of hydrodynamics, the Sommerfeld number is used to assess the load capacity of the hydrodynamic bearings. This number or criterion is determined by the bearing parameters such as: pressure from the external load, viscosity-rate parameter and relative gap:

$$S_o = \frac{F \cdot \psi_{eff}^2}{D \cdot B \cdot \eta_{eff} \cdot \omega_h} = \frac{P_{cp} \cdot \psi_{eff}{}^2}{\eta_{eff} \cdot \omega_h} = \varphi\left(\varepsilon, \frac{B}{D}, \Omega\right)$$

where F is force, acting on the bearing, - radial load (N);

ψ_{eff} – relative clearance of the bearing, as the ratio of the average clearance to the nominal diameter;

D, B – the nominal diameter and width of a bearing (m);

η_{eff} – dynamic viscosity of a lubricant (Pa•s);

ω_h – angular velocity of the shaft (rad/s);

ε – relative eccentricity;

$\frac{B}{D}$ – relative length of the bearing;

Ω – contact angle (radians).

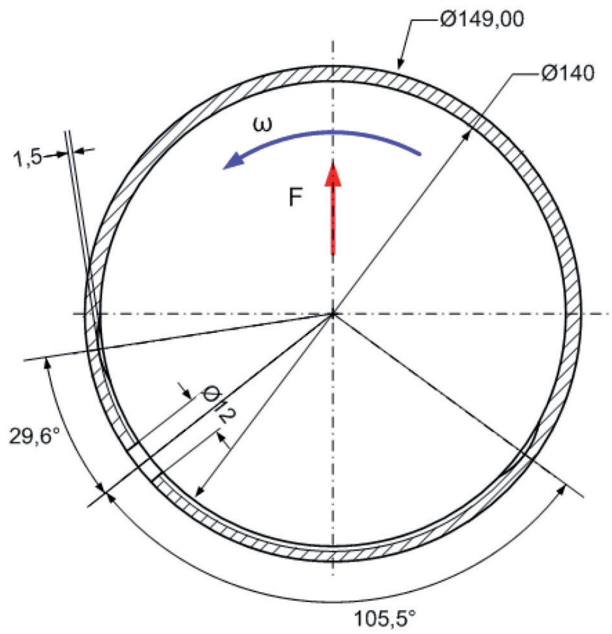


Figure 2. Design parameters of the bearing insert of a hydrodynamic bearing of the main support with the nominal diameter of 140 mm

Moreover, the Sommerfeld number is a function of the geometric parameters of the bearing, such as relative eccentricity, relative length and angle of contact. The functional dependences between the Sommerfeld number, given by the geometric parameters, and some other parameters of the bearing are determined by the standard [4].

The calculation method involves checking the laminar flow of the lubricant according to the Reynolds criterion, checking for the permissible average pressure and calculating the Sommerfeld number. The values of the number S_0 are used to determine other parameters based on the dependencies given in the form of graphs and tables. For example, for the calculated number S_0, we define the relative eccentricity ε as the function of $\varepsilon = \varphi(S_0, B / D, \Omega)$. The obtained value ε allows us to determine the minimum thickness of the lubricant film with the following dependence:

$$h_{\min} = 0,5 \cdot D \cdot \psi_{eff} \cdot (1-\varepsilon) .$$

The friction coefficient in the bearing is determined in a similar way by the specific coefficient of friction. Furthermore, based on the bearing design features, the methodology permits us to determine the frictional force in the bearing, the values of the frictional forces, the intensity of the heat from the frictional forces and the temperature of the bearing or the lubricant at the outlet.

Additionally, the force of friction in the analysis was determined for the following conditions: the completely filled bearing gap (loaded and unloaded areas) with the lubricant, the supply of the lubrication through one hole, two cradles and a partial circular groove.

According to the technical characteristics of the bearings, the lubricant is fed at the pressure of 0.35 mPa. That is, the heat emitted from frictional forces scatters due to the heat transfer to the lubricant that is pumped through the bearing. The lubricant consumption in the hydrodynamic bearing is determined by two issues: the lubricant flow through the ends of the bearing due to the hydrodynamic pressure and the lubricant flow rate due to the pressure of the feeder. By comparing the action of frictional forces and the dissipation of heat into the lubricant, we determined the lubricant temperature at the outlet of the bearing.

The method involves an iterative procedure for checking the performance parameters of the bearing, taking into account the change in the viscosity (effective viscosity) after the lubricant heating-up. However, this analysis is limited by only the first step of the calculations.

Results and Discussion

The calculations have been made for the loads obtained from the Y. Noguchi actual data for rolling the rod products with the diameter of 5.5 mm at a speed of 120 m / s in the last module [5]. The rolling force values in the modules were obtained by the experiments and were used to calculate the load on the main supports of the each unit of modules.

Moreover, taking into account the fact that the circular groove and cradles are placed on the arc at 152°, for the calculations we assumed the contact angle of 180°. The relative bearing length was considered to be 0.75.

Moreover, the example illustrated in Figure 3, shows the results of the calculations carried out with the parameters of the main support bearings for all the unit modules at the lubricant input temperature of 60°C (the data at the entrance into the bearing were considered).

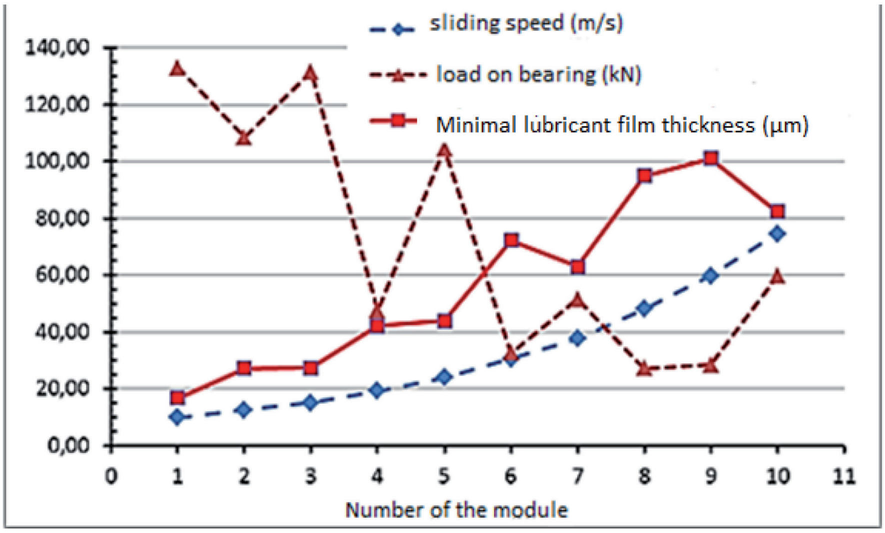
Among all the data obtained, we have analyzed specifically a fairly significant issue of lubricant film minimum thickness, which changes through the modules from 16.7 μm in the first module to 100.9 μm in the ninth one. Thus, for the above mentioned roughness parameters of the working surfaces of the bearing, the mean square roughness is $Ra\Sigma = 1.613$ μm. In order to ensure the hydrodynamic lubrication regime, i.e. the complete separation of the sliding surfaces by the lubricant, the thickness of the film should exceed the mean square roughness as much as 3 times [6]. That is, the critical thickness of the lubricant film for the indicated surface roughness is 4.84 μm. For the further consideration we accepted the minimum allowable thickness of the lubricating film as thick as 5 μm.

Furthermore, from the comparison between the parameters presented in Figure 3a, one can conclude that the effect of the load on the minimum thickness of the lubricant film is manifested at a much lesser extent than the effect of the speed.

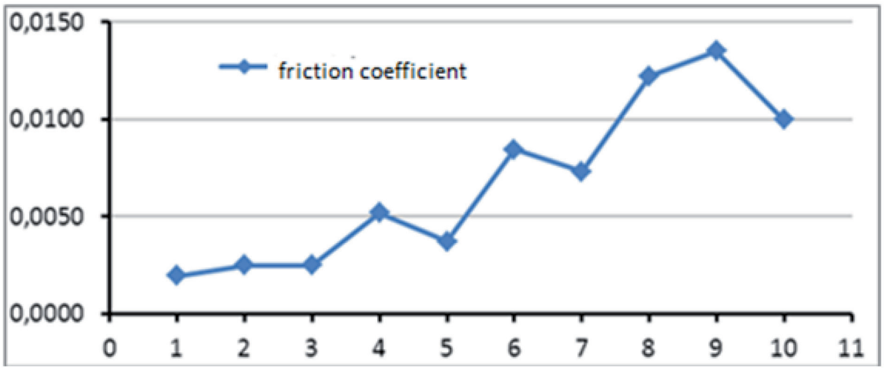
The other parameters of the bearing vary approximately proportionally to the sliding speeds throughout all the modules. These parameters are friction coefficient, frictional power and the temperature of the lubricant at the outlet (figure 3b; 3c; 3d respectively). From this we conclude that the speed is the dominant factor here.

In general, it can be admitted that all the calculated parameters of the bearings installed in the unit modules are in optimal ranges of possible value changes if a feeding temperature is 60°C.

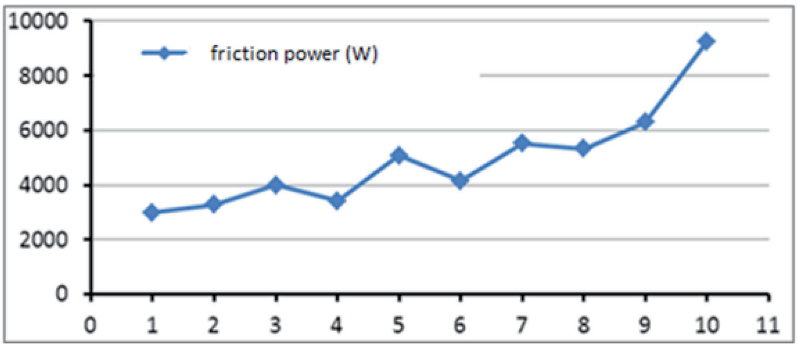
In order to carry out the analysis how the lubricant input temperature affects the bearings performance,



3a)



3b)



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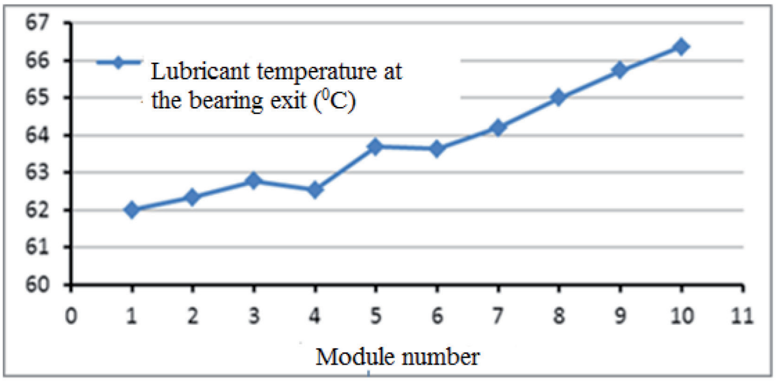


3d)

Figure 3. The calculated performance parameters of the bearing at lubricant input temperature of 60 °C

the similar calculations with other input temperatures of the range 40 °C - 100 °C have been performed and the results are shown in Figure 4. It is evident from the data that the increase in the input temperature of the lubricant, accompanied by a certain decrease in viscosity, leads to the decreases in the values of the lubricant film thickness, the friction coefficient and the lubricant heating-up. That is, the increase in the input temperature in general positively influences the parameters of the bearings, but only to certain values.

Thus, at the lubricant input temperature of 80 °C, the thickness of the lubricant film in the bearings of module No.1 is 4.24 μm (Figure 4a), which is less the critical value (5 μm). It is obvious that, at the speeds typical for the first module, the viscosity of the lubricant at the input temperature of 80 °C or more is not sufficient to form a film, which would meet the conditions of a hydrodynamic lubrication regime with complete separation of the surfaces. Therefore, the lubrication temperature of 80 °C can be considered as the temperature at which the bearings can operate steadily in transient modes between the hydrodynamic and combined lubrication modes. However, the further increase in the input lubricant temperature for the first modules is undesirable.

In addition, under the conditions of the bearings installed in module No 10 the following is registered: at 90 °C of lubricant input temperature, the calculated Reynolds number exceeds the critical value, which for the existing ratio of nominal diameter to clearance is 838.7 [7]. This critical value exceeding indicates that the condition of the laminar flow of the lubricant is not fulfilled. In its turn, the violation of the uniformity of the lubricant flow in the bearing working zone (i.e. the transition of the flow into the turbulent one) leads to a sharp decrease in bearing loading capacity. Accordingly, in such operation mode, a continuous lubricant film is not formed and the working surfaces of the bearing are not separated. Thus, the transition is possible not only to the lubrication regime of marginal conditions, but even to dry friction (with the destruction of the film lubricant). In this case, we observe the contact of the working surfaces, their heating and intensive wear - such a mode of operation is unacceptable for hydrodynamic bearings and results in performance failure.

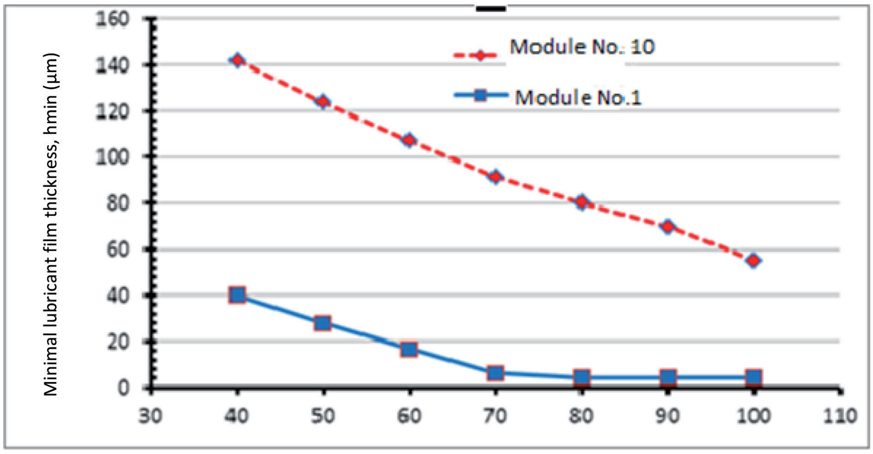
Furthermore, the example of the bearing destruction shown in Figure 1 is directly connected with the lubricant laminar flow violation, when the lubricating film does not separate the surfaces. The destruction of the bronze coating of the bearing insert started with this very issue, followed by the local swelling or welding of the bearing insert steel base and the shaft, the failure of the thermal and press fit of the bearing insert and its further scroll with the subsequent heating up, grip and destruction. In this case, the nature of the damage to the bearing insert corresponds to the peculiarities of the load distribution for the overhung shaft, the calculations for which were made with a media solid-state modeling [8, 9, and 10]. According to the calculations, the maximum pressure acts on the segments located closer to the plane of external force action, as shown in Figure 5. Therefore, according to Fig. 4d, the lubricant input temperature should not exceed 85 °C.

On the other hand of the problem, based on the dependencies shown in the graphs (Figures 4b and 4c), we observe that at the lubricant input temperature of less than 60 °C, the values of the friction coefficient and the lubricant heating-up temperature are relatively high. This is especially true for module No10. The high values of the friction coefficient lead to unproductive power losses. In addition, increased heating of the lubricant causes a decrease in the viscosity of the lubricant in the working area of the bearings and the gradual heating of the entire mass of the lubricant being pumped through the bearings. This means that the bearing would actually work at a slightly higher temperature (and the respective effective viscosity) than the temperature accepted as suitable for the lubricant feed. Hence, the optimum input temperature for supplying the lubricant should be within the range of 60 - 80 °C.

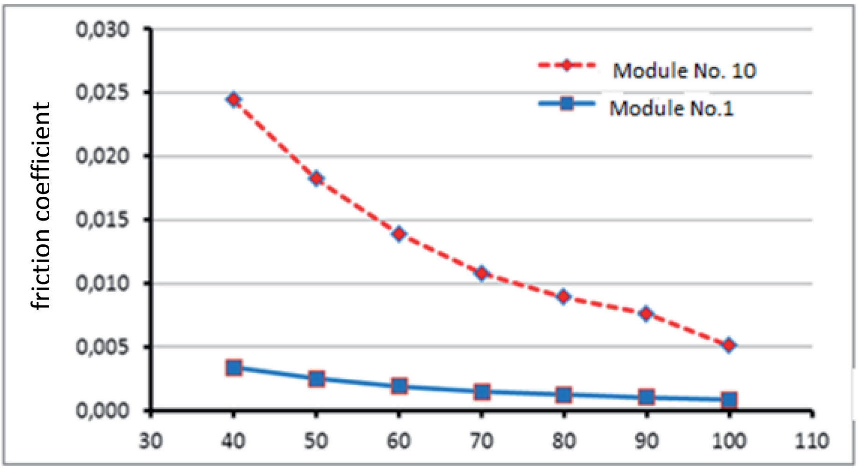
Taking into account the fact that the temperature regime in the bearings under consideration is maintained by the heat transfer to the lubricant, pumped through the bearing, we have also performed the analysis to reveal how the lubricant input pressure influences the parameters of the bearings. According to the technical characteristics of the unit, the nominal feed pressure is 0.35 mPa (3.5 kgf/cm^2). All the above calculations were made for this very pressure. That is why the existing calculations have been supplemented by the calculations with pressure decreases down to 0.3 mPa, 0.25 mPa and 0.2 mPa.

The research has revealed that the input pressure does not affect either the minimum thickness of the lubricant film or the friction coefficient, i.e. the hydrodynamics in the bearings does not change with the decrease in the input pressure. However, the decrease in the amount of the lubricants pumped through the bearings due to a pressure drop results in a slight increase in the temperature of lubricants at the outlet of the bearing.

Moreover, the values of this heating are 60 °C, 70 °C and 80 °C for the lubricant being fed in the first



3a)



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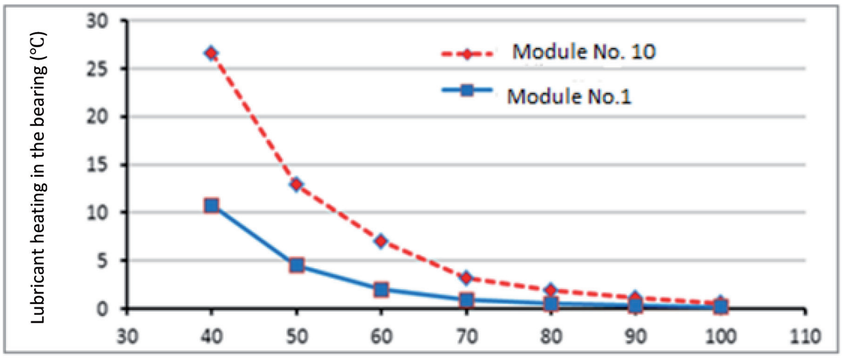

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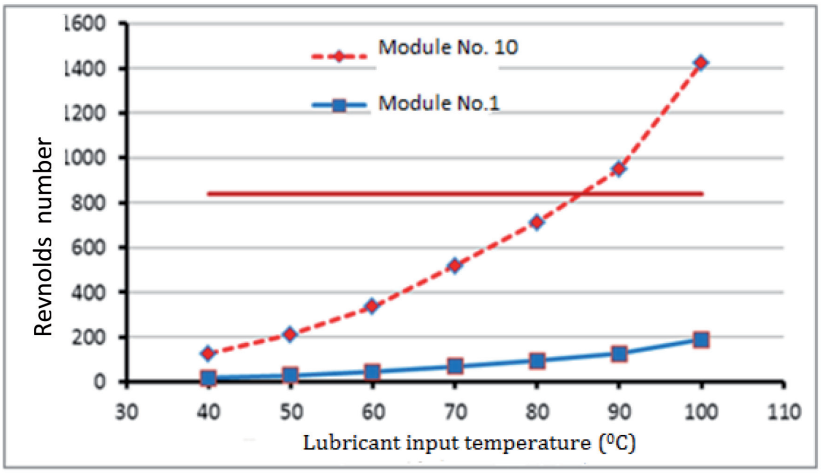

3d)

Figure 4. The dependence of the bearing performance parameters on the lubricant input temperature

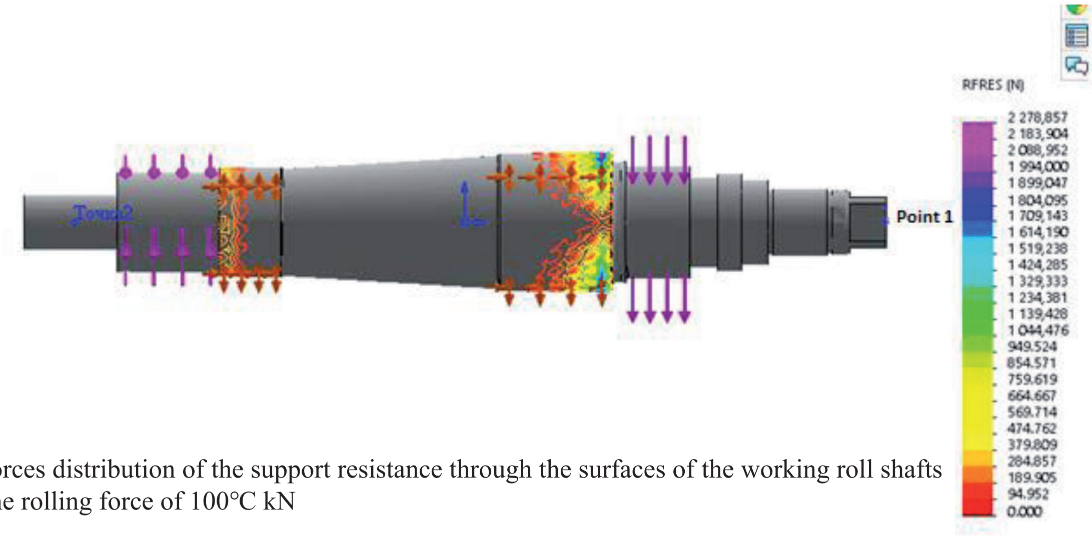

Figure 5. The forces distribution of the support resistance through the surfaces of the working roll shafts in a module at the rolling force of 100°C kN

and the tenth modules (refer to Figure 6). As can be seen from the data obtained, the pressure reduction, even if it is down to 0.2 mPa, does not lead to critical heating the lubricant. That means that the pressure of the lubricant input does not significantly affect the parameters of the module bearings.

Eventually, in order to check the previously made conclusion about the comparatively insignificant influence of the load on the bearings parameters, we have performed an analysis of the increased loads effect for modules No 1 and No 10. The loads on the main support bearings for these modules were 132.887 kN and 29.832 kN (rolling forces of 98 kN and 22 kN, respectively). All the above calculations for modules No 1 and No 10 have been made for these loads. The earlier results have been supplemented with the calculations for the loads increased by 50% and by 100%, at other parameters being constant (inlet lubricant temperature of 60 °C, pressure of 0.35 mPa, standard speed mode). These results are shown in Figure 7.

The obtained data have proven the conclusion about the relatively insignificant influence of loads on the performance parameters of the bearings: as the load was increasing, the thickness of the minimum film in the bearings was decreasing, which led to a decrease in the coefficient of friction. The other parameters were not changing

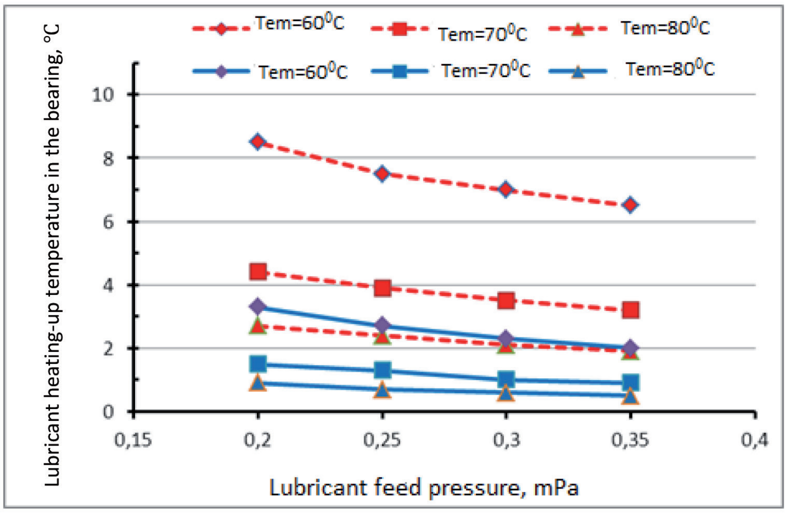

Figure 6. Lubricant heating-up temperature dependences on lubricant input pressure. A variety of the input temperatures in the bearings of module No 1 (solid lines) and module No 10 (dash lines)

significantly. For example, the lubricant temperature at the output of the bearings decreased by tenths of a degree when increasing loads.

However, under double load condition, the minimum thickness of the lubricant film decreases from 107 µm to 82.3 µm in module No.10 while in module No 1 the range of the changes is from 16.7 µm to 4.2 µm. Thus, for module No 1, in which the technologically determined loads are approximately four times higher than those in module No 10, the minimum thickness of the lubricant film decreases to critical values with a double load increase. Thus, the hydrodynamic lubrication mode can undergo transition into the mixed lubrication. In addition, with a double increase in the load on the bearings in module No 1, the average pressure exceeds the permissible one for bronze inserts of the bearing : 18.08 N / mm^2> 15.0 N / mm^2.

Conclusions

Using the standard calculation method, the analysis has been carried out on the performance parameters of the hydrodynamic plain bearings in high-speed modules of the wire rolling mills. The following parameters were analyzed as ones having the response functions: Reynolds number, minimum film thickness of the lubricant in the bearing working zone, friction coefficient, lubricant temperature at the outlet of the bearing. The other parameters were analyzed as the factors of the influence functions: speed of sliding in the working zone of the bearing, load, lubricant input temperature and lubricant input pressure.

The calculations have revealed that the most significant factors are the speed and the input temperature of the lubricant. As the speed increases, the values of all the parameters under consideration increase. Different is the affect of the lubricant input temperature: when the temperature increases, the value of the lubricant viscosity decreases, resulting in the decreases in the thickness of the lubricant film, in the friction coefficient and in the temperature of the lubricant at the outlet of the bearing.

The research has found out that for certain values of the speed and the input temperature, the bearings can lose their working capacity. Thus, for module No 10, which operates at the maximum production speed, if the lubricant input temperature reaches above 85 °C, the condition of laminar flow (Reynolds number exceeds the critical value) is violated, and turbulence occurs, which leads to the failure of the bearings.

Based on the results of the present article calculations, we claim that the lubricant input temperature within the range of 60 °C to 80 °C is optimal for the hydrodynamic plain bearing in terms of energy consumption and reliability level.

The load and the pressure of the lubricant supply affect the working conditions of the bearings to a small extent. According to the received data, the two times decrease in the lubricant input pressure and/or double-load increase do not shift the working conditions of the bearings beyond the acceptable values.

The observance of the proposed recommendations for available bearing designs and actual production parameters is able to increase the life of the module units, to reduce the downtime associated with the need for repairs, and to reduce the costs of repairs.

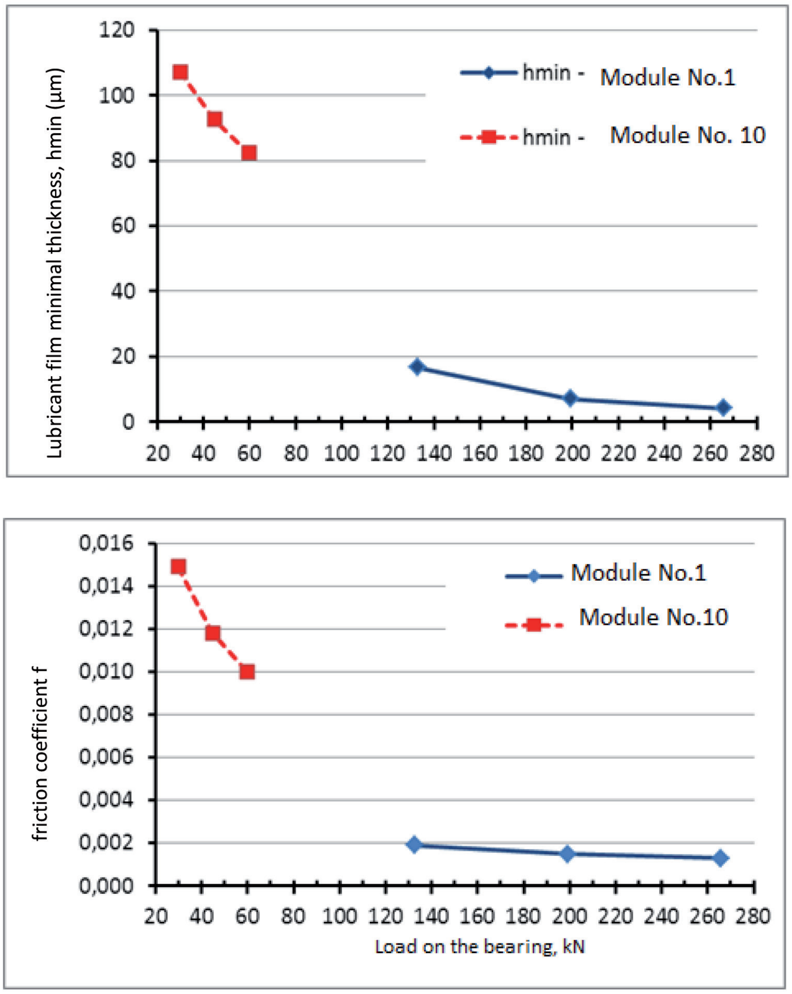

Figure 7. Dependences of minimum film thickness and friction coefficient on applied forces for modules No 1 and No 10

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