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PREDICTION THERMAL ELASTIC BEHAVIOR OF THE CYLINDRICAL ROLLER BEARING FOR RAILWAY VEHICLES AND CALCULATING BEARING LIFE

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Abstract: *This paper presents a calculation model the thermal elastic behavior of the cylindrical roller bearing for bearings towed vehicle in railways. The present mathematical model allows to define the value of the moment friction due to lubrication, the friction moment due to the radial and axial loads. Heat generated in the bearing calculated based on the value of previously defined moments. Also, it shows the method of calculating the coefficients of conduction and convection of heat in bearing. Using programming system general purpose analyzed thermal elastic behavior of the cylindrical roller bearing for bearings axle towed vehicle in railways. Finite element method determined by temperature values in the characteristic points of the bearing depending on the speed wheel for different time periods. Also, it determined of displacement in characteristic points of bearing due to the heat load in the steady temperature state, maximal stress and bearing life.*
Keywords: *Thermal elastic behaviour, cylindrical roller bearing for railway vehicles, finite element method*

INTRODUCTION

The advent of railway in the mid-nineteenth century certainly represents a significant technological innovation as it is no doubt possible to play a revolutionary role and enormous contribution to the industrial development and economic progress of society. The rapid development of the construction and exploitation railways has made a significant impact on the development of strength science and theory construction, [16], because it appeared a series of new problems (particularly in relationship with the design and construction of railways, bridges, locomotives, vagon and etc..) that should be solved.

Although in comparison with other types of transport railway has a number of advantages in terms of economic viability (lower power energy and especially environmental sustainability), inefficiencies created by the railway regulations have placed restrictions on the industry that have prevented effective competition. Stoppages in innovation railway technology and inadequate answer to significant increase in the quantity goods of small packaging, reducing the goods suitable for carriage by rail (such as ore and coal), have primary explanation because of for example the share railways in freight traffic SAD that is after the Second World War was nearly 70% of intercity ton miles, up in 1975. years dropped to 37% [17].

The European Union treated rail as a transporter of the future and seeks to reaffirm rail transport, with requirement of a competitive, secure and high-quality transport all kinds of goods. Achieving these goals among other things requires the construction of modern wagons customized market challenges, specific technological requirements and systems which to quickly perform the loading / unloading operations [2].

Cylindrical roller bearings for railway are key components of wheel towed vehicles and their cancellation can have disastrous consequences. Bearing temperature is one of the most important parameters bearing whose by monitoring the can be determined state of bearing in exploitation. For this reason were performed tests using different lubricants (grease) and based on the test results it can be concluded that the lubricant has a significant influence on the bearing temperature in exploitation. Roller bearings for railways usually lubricated with grease that are accommodated in a closed housing with bearings so as to ensure proper lubrication. During the rotation of grease comes into contact with rollers and the rings and after the certain period of time leads to the mechanical the degradation of grease. For this reason is very important replace the grease before it has loses her mechanical properties. In paper [11] were performed tests in winter and summer period (in winter temperature was -15 °C and in the summer 20 °C) and led to the values of temperature in the steady state temperature in depending on the type of grease which is used for lubrication. The temperature in winter period was between 15 ÷ 51 °C and in the summer of 33 ÷ 59 °C.

The heat generated due to the rolling wheel on the rails transferred to the entire assembly wheel for railway vehicles. One part of this heat transferred to the bearings. Cole and others [3] examined this influence of experimental and computer modeling. For value generated amount of heat due to the movement of the wheel on rail (Q = 1834 W) in steady temperature state were determined by the *temperature in characteristic points wheel and rings bearing. In this paper is not considered the fact that due to the rotation of bearing in him generates heat which has great influence on his temperature.*

In addition to the heat generated due to the motion of the wheel on rail on heating assembly wheel significantly affected have the heat that is generated due to the braking locomotives and railway freight wagons (at the contact between the brake block and wheel) [4]. The value of the heat generated between the brake block and wheels, as well as the value of the mechanisms heat transfer on wheel calculated mathematically, and using the finite element method were determined by the temperature on wheel. The value of heat generated due to the contact of brake block and wheel was Q = 446 W. Verification of mathematical calculations and finite element methods performed on a laboratory plant on one side brake block of the shaft.

Mohan [12] by applying the finite element method performed prediction of thermal and static behavior wheel locomotive and railway freight wagons. The value of the temperature on the flange wheel was 70 °C. Static analysis determined the value of Von Mises's stress depending on the deformations caused by the static load. The maximum displacement was on wheel and amounted to 0.2196 mm, and the maximum stress on wheel 46,34 N/mm². Integrated thermal and static behavior in one model were determined by values of displacements and stresses on wheel. The maximum displacement on the flange wheel amounted to 1.084 mm and maximum stress on wheel amounted to 148.98 N/mm².

Preventive maintenance and remont bearing in the exact prescribed intervals is essential on bearing life. In case when remont bearing don't make in prescribed time intervals may lead to disastrous consequences such as damage axle wheels, damaged parts rail and popping train for rails. Because of this reason it is very important change the inner ring bearing before it reaches his damage due to material fatigue that have disastrous consequences [6]. In this paper analyzes the influencing parameters that lead to damage railway line. Bearing was completely damaged due to the material fatigue that led to the cracking of the inner ring. Slipping of the inner ring on the axle has led to the generation of large of heat generated which led to the change in structure of the material in the bearing and on surface axle.

MECHANISM OF HEAT GENERATION

Introdiction

Bearings assembly wheel railway freight wagons can make different types of bearings (cylindrical roller bearing, spherical roller and taper roller). In this paper discusses the bearing assembly wheel with cylindrical roller bearings (mark of the bearing 324 NJ EC.M1C4 VA301). On Figure 1 shows assembly wheel as well as details bearings mounted on the axle.

This paper considers only the heat generated in the bearing (without heat generated by the movement of wheel on railway line). The heat generated determined on the basis of the friction torque due to lubrication and friction torque due to the load (axial and radial) [1].

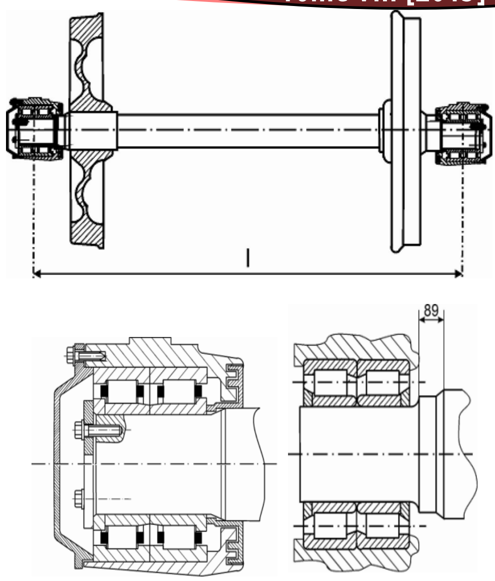


***Figure 1.** Assembly wheel with details bearings*

Determination of the friction torque

$$M = M_0 + M_1 + M_2 \ Nmm \qquad (1)$$

M_0- friction torque due to lubrication,
M_1- friction torque due to the radial load,
M_2- friction torque due to axial load.

$$M_0 = 10^{-7} \cdot f_0(\nu \cdot n)^{\frac{2}{3}} \cdot d_m^3 \ \ Nmm \qquad (2)$$

f_0 – coefficient depended of the bearing type and lubrication type (for cylindrical roller bearing it value is 3),
ν- kinematic viscosity of the lubrication ν=18,
n- bearing speed [rev/min],
d_m - middle bearing diameter (d_m=190 mm).

$$M_1 = f_1 \cdot F_r \cdot d_m \ \ Nmm \qquad (3)$$

$$F_r = \frac{G_1}{4} \qquad (4)$$

$$G_1 = \frac{G}{n_t} \qquad (5)$$

f_1 - coefficient which depends of the bearing type (for cylindrical roller bearing it value is between 0,0003-0,0004),
F_r - radial static load acting on one bearing (F_r= 55181 N),
G_1- radial static load acting on one wheel set (G_1=220725 N),
G- weight of the vehicle,
n_t- number of wheel sets.

$$M_2 = f_2 \cdot F_a \cdot d_m \ \ Nmm \qquad (6)$$

f_2- based n d_m ν i F_a/A,

$$A = k_B \cdot 10^{-3} \cdot d_m^{2,1} \qquad (7)$$

A-surface,
F_a -axial load bearing.

The axial load bearing occurs when the train move a curve. For a radius curvature R = 500 m and height superelevation on one side stripes h = 110 mm can be defined by the maximum permitted speed move of the train in a curve, and it is v = 68 km/h. The value of axial load calculated for the speed of move the train in a curve of v = 50 km/h.

Permissible speed of movement of the train in a curve calculated from the following equation:

$$h = \frac{11{,}8 \cdot v^2}{R} \quad (8)$$

for 50 km/h axial load is F_a=16479 N, and coefficient f_2=0,008.

Table 1 shows the values of the friction torque due to lubrication, radial and axial load, and the total value of the friction torque M for the different speed movement of the train.

***Table 1**. The values of the friction torque*

n rev /min	M_0 Nmm	M_1 Nmm	M_2 Nmm	M Nmm
300	629	4193	25080	29902
450	991	4193	25080	30264
700	1107	4193	25080	30380

Determination of the heat generated heat in the bearing

Heat generated in the bearing determined by the equation (9):

$$Q_{uku} = 1{,}05 \cdot 10^{-4} \cdot d_m \cdot n \left[f_0 (\nu \cdot n)^{\frac{2}{3}} \cdot d_m^2 \cdot 10^{-7} + f_1 \cdot F_R + f_2 \cdot F_a \cdot 0{,}1 \right] \quad W \quad (9)$$

In table 2 shows the values of heat generated during movement of the train at speed of 50, 80 and 120 km / h (operational speed bearing). Accepted that the vagon has such a path in exploitation to move 90% straight and 10% in the curve (5 % left and 5 % right curves and the speed of movement in curve v = 50 km/h, this values have been adopted on the basis of recommendations for calculation bearing wheel car).

***Table 2.** Value of the generated heat*

n rev /min	Q_{uku} W
300	231
450	364
700	573

THE HEAT TRANSFER MECHANISMS IN THE BEARING

Heat transfer mechanism at bearing (embedded as shown in Figure 1) are convection due to the rotation, conduction between the inner ring and the axle and the outer ring and housing [1].

Convection due to rotation the bearing

Heat transfer through the bearing is realized only between bearings and surroundings air. Absorbed heat from the grease, in this paper is not discussed. Because of the small difference in temperature radiation could be neglected, coefficient of the heat transfer is calculated according to condition of the flow air through bearing which belong to the turbulent motion. At this transfer, total air flow velocity, caused by the bearing rotation, is calculated from the axial and tangential component.

Surfaces for the axial flow air between the inner and outer track rolling:

$$A_{ax} = \frac{\pi}{4} \cdot (D^2 - d^2) \quad m \quad (10)$$

Axial flow velocity could be calculated as a velocity between two cylinders, from the relation:

$$u_{ax} = \frac{V}{A_{ax}} = \frac{4 \cdot V}{\pi \cdot (D+d)} \quad m^2/s \quad (11)$$

where V-volume air flow obtained from the continuity equation:

$$V = u_{sr} \cdot A_S = \frac{u}{2} \cdot B \cdot s = \frac{1}{2} \cdot d_m \cdot \omega \cdot s \cdot B \quad \frac{m^3}{s} \quad (12)$$

In previous equation is considered medium velocity air through the cross-sectional area $A_S = B \cdot s$.

Tangential velocity component, on the midlle diameter, could be calculated from relation for air flow between movable and immobile cylinder:

$$u_{at} = \frac{\omega \cdot d_m}{2} = \frac{\pi \cdot f \cdot (D+d)}{4} \quad m^2/s \quad (13)$$

where are:

f-frequency bearing Hz,

D- diameter outer ring m,

d- diameter inner ring m.

The resulting air velocity during rotation bearing is obtained from axial and tangential components according to [1].

$$U = \sqrt{u_{ax}^2 + u_{at}^2} \quad m^2/s \quad (14)$$

Coefficient convection is calculate according to [1]:

$$\alpha = (c_0 + c_1 \cdot U^2) \quad \frac{W}{m^2 K} \quad (15)$$

c_0 i c_1 are constants obtained experimentally (c_0=10 a c_1=5).

In table 3 shows the values of the coefficient of convection during rotation bearing, depending on the speed.

***Table 3** . Values of the coefficient of convection*

Speed bearing rev /min	Coefficient convection $\frac{W}{m^2 \cdot K}$
250	10,5
450	35,3
700	71,2

Heat convection between ring and housing and ring and axle

The coefficient of thermal conductivity depends on the gap between the outer ring and the housing and the inner ring and axle. Thermal conductivity between the two elements can be determined on the basis of equation [8]:

$$\lambda_{ij} = \frac{ln\left(\frac{r_j}{r_i}\right)}{\frac{ln\left(\frac{r_j}{r_1}\right)}{\lambda_j} + \frac{R_W}{r_1} + \frac{ln\left(\frac{r_1}{r_i}\right)}{\lambda_i}} \quad (16)$$

where are λ_i i λ_j thermal conductivity of ring and housing. Other marks are shown in Figure 2.

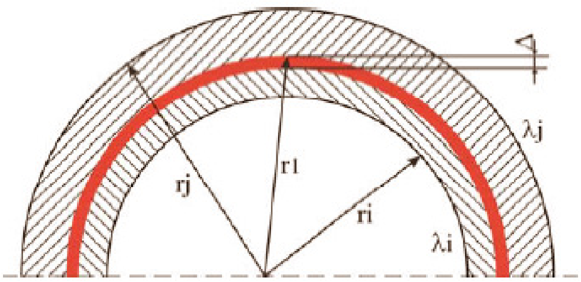


***Figure 2**: Schematic representation the contact between the outer ring and housing [16]*

In the previous equation R_w represents the thermal contact resistance on place of contact of the ring with housing, and according to [8] can be calculated as follows:

$$R_w = \frac{r_1}{\lambda_{ij}} ln\left(\frac{r_1+\Delta}{r_1}\right) \quad (17)$$

where is Δ - clearance between outer ring and housing.

On similar method are determine conduction at contact between the inner ring and axle.

In Table 4 shows the values coefficient thermal conductivity of certain based on the previous equation.

***Table 4** .The values of coefficient thermal conductivity between the ring and the housing and ring and axle*

Place contact	Coefficient conduction λ W/m²K
Inner ring/axle	60,5
Outer ring/ housing	90,9

MODELING THERMAL ELASTIC BEHAVIOR

Figure 3 shows a model of a cylindrical roller bearing modeled using the program system PTC Creo Parametric. This bearing used for the bearing assembly wheel in railways.

Setting the coordinate system, the choice of contact pairs (CONTA 174, 53 contact pairs), defining of heat generated in the bearing, the choice of the type of finite element (SOLID 87, mesh than 8021 elements and 31720 nodes) and defining the elements between which there is conduction and convection heat transfer has been done in the framework of preprocessing. As a result of the previous on Figure 4 shown discretized model considered bearing.

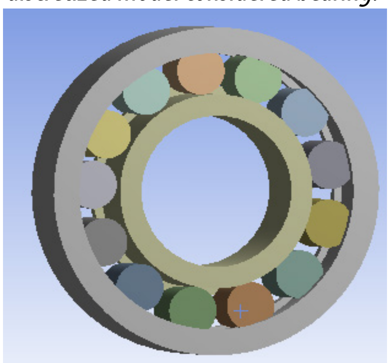


***Figure 3**. Appearance a cylindrical roller bearing for the bearing assembly wheel for railway*

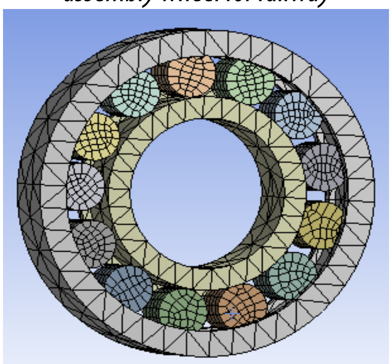


***Figure 4**. Appearance discretized model bearing*

After the calculation and after post processing obtained is graph showing the temperature distribution on elements bearing for speed n = 450 rev/min, respectively with speed moving a train of 80 km/h (Figure 5). The maximum temperature is marked with red color and equals to 54 °C and minimal with blue and equals to 53 °C in the steady temperature state bearing.

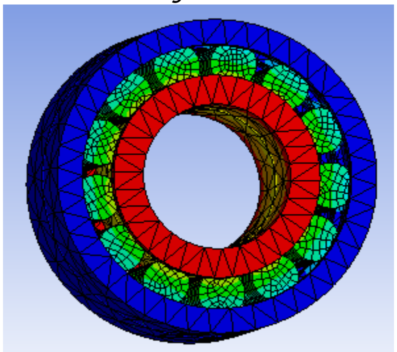


***Figure 5**. Graphical representation temperature on elements bearing*

Based on results of modeling in Figure 5 show temperature changes on rings bearing in depending from time for three different speeds movement of the train (v=50, v=80 i v=120 km/h). On diagram maximum temperature represents temperature of the inner ring and the minimum temperature is the temperature of the outer ring. Observation diagrams shows in Figure 6 can see that the steady temperature state bearing with highest speed motion (n = 700 rev/min, v = 120 km/h]) reached in shortest period of time.

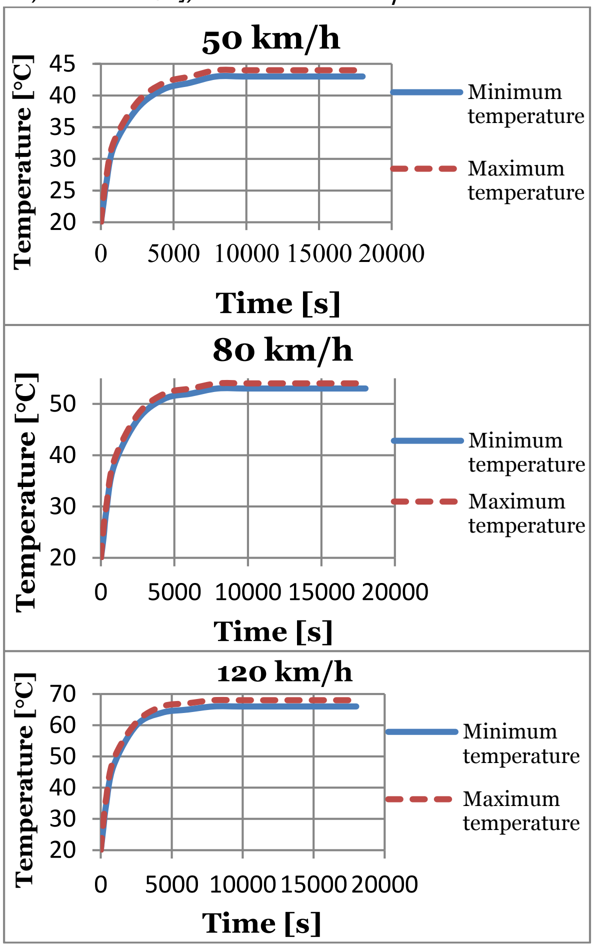


***Figure 6**. Change of the temperature on rings bearing depending on the time and speed of movement*

Based on the previously defined thermal model (temperature fields) can be determined by displacement in nodes bearing. The thermal load is defined on basis of the results of thermal analysis, and it represents the temperature of in nodes of the bearing. On elastic model is necessary to define limits displacement and it is shown in Figure 7.

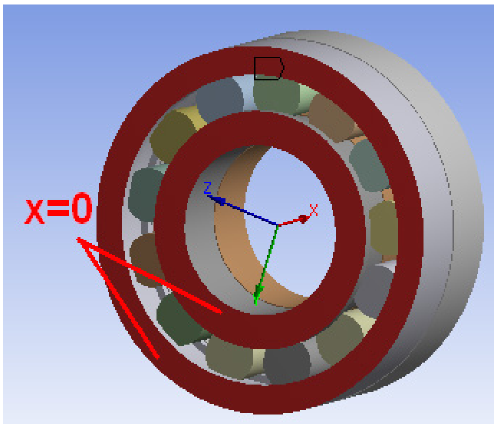
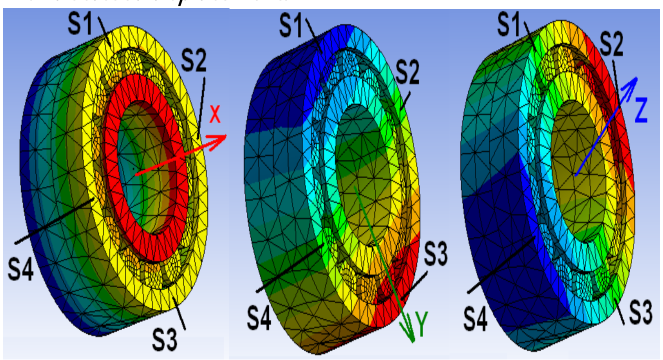


***Figure 7.** Constraints displacement*

In Figures 8 a. (displacement in the direction of x-axis), 8 b. (displacement in the direction of y-axis) and 8 c. (displacement in the direction of z-axis) shows the computer models of cylindrical rolling bearing for railway after heat load for speed n = 700 rev/min in the direction x, y and z axes. In figures shown characteristic points in which discussed displacement.



a. b. c.

***Figure 8.** The results of computer modeling of elastic behavior of the bearing after effects of heat load*

The obtained results can be seen that maximum displacement in the axial direction (direction x-axis) was 51 [μm], and in the radial direction displacement was 55 [μm]. Displacement values in characteristic points S1, S2, S3 and S4 are shown in Table 5.

***Table 5.** Values displacement in characteristic points*

Axis	Values displacement characteristic points μm			
	S1	S2	S3	S4
x	51	51	51	51
y	-55	0	55	0
z	0	55	0	-55

Stress (Von Misses's) appearing in bearing have a maximum value at the place of contact rollers and outer ring bearing and the amount 58 N/mm². In Figure 9 shows the distribution of stress on the bearing.

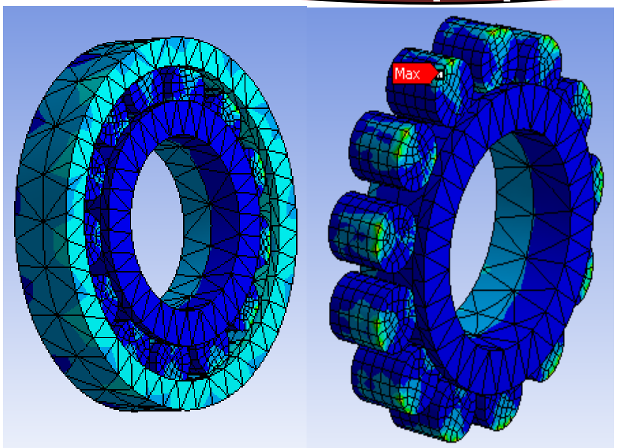


***Figure 9.** Distribution of stress on the bearing*

Bearing life for towed vehicles on railways determined following equation:

$$L_{10} = \left(\frac{C_r}{P_r}\right)^{\frac{10}{3}} \cdot \pi \cdot D_k \cdot 10^{-3} \qquad (18)$$

where are:

C_r-basic radial dynamic load rating (C_r=465 KN)

P_r-radial equivalent dynamic load acting on one bearing

$$P_r = F_r \cdot f_d \qquad (19)$$

f_d-factors of additional forces (f_d=1,2÷1,4)

D_k-diameter of the vehicle wheel (D_k=0,92 m)

On basis equation (18) and previously shown calculations obtained bearing life, which is 1 million kilometers.

CONCLUSION

In this paper were analyzed thermal elastic behavior cylindrical roller bearings for railway. Mathematical modeling of cylindrical roller bearing defined by the finite element method.

Based on the results of thermal behavior can be seen that the maximum temperature bearing was 68 °C which is considerably less than the maximum permissible temperature of 120 °C.

Maximum stress in bearing is 58 N/mm² which is much lower than the permissible stress in bearing.

The calculated bearing life corresponds to the required recommendations EN C (2006) 3345.

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