ISSN 1330-3651 (Print), ISSN 1848-6339 (Online)
DOI: 10.17559/TV-20150809184241

MATHEMATICAL MODELLING OF THERMAL BEHAVIOUR OF CYLINDRICAL ROLLER BEARING FOR TOWED RAILWAY VEHICLES

Vladimir Blanuša, Milan Zeljković, Branko M. Milisavlevich, Branko Savić

Preliminary communication
This paper presents a mathematical model for the prediction of the thermal behaviour of a cylindrical roller bearing for axle assembly of the wheel set of the towed railway vehicles. The mathematical model allows for the heat generated due to lubrication, radial and axial loads at different speeds of the train to be calculated. The method of calculating of the heat conducting and converting coefficients in the bearing is also shown. By using the general purpose software system based on the finite elements method, the thermal behaviour of the above mentioned cylindrical roller bearings was analysed. Temperature values of the bearings are determined by the finite elements method values depending on the speed of the train movement on a straight-line section and the curve (v = 20 km/h, v = 40 km/h, v = 60 km/h, v = 80 km/h and v = 100 km/h), the curve radius (R = 500 m), the ambient temperature of 20 °C and the cant height (h = 110 mm, h = 140 mm, h = 180 mm).

Keywords: *cylindrical roller bearing for towed railway vehicles; finite elements method; mathematical modelling*

Matematičko modeliranje toplinskog ponašanja valjnog ležaja s valjčićima za vučena željeznička vozila

Prethodno priopćemje
U radu je prikazan matematički model za predviđanje toplinskog ponašanja valjnog ležaja s valjčićima za uležištenje vučenih željezničkih vozila. Matematički model omogućava izračunavanje generirane topline uslijed podmazivanja, radijalnog i aksijalnog opterećenja za različite brzine gibanja vlaka. U radu je također prikazan način izračunavanja koeficijenata provođenja i prevođenja topline u ležaju. Uporabom programskog sustava opće namjene na bazi metode konačnih elemenata analizirano je toplinsko ponašanje valjnog ležaja s valjčićima za vučena željeznička vozila. Metodom konačnih elemenata određene su vrijednosti temperature ležaja u ovisnosti o brzini gibanja vlaka (v = 20 km/h, v = 40 km/h, v = 60 km/h, v = 80 km/h i v = 100 km/h), polumjera zakrivljenosti zavoja (R = 500 m), ambijentalne temperature od 20 °C i visine nadvišenja pruge u krivini (h = 110 mm, h = 140 mm, h = 180 mm).

Ključne riječi: *vmatematičko modeliranje; metoda konačnih elemenata; aljni ležaj s valjčićima za vučena željeznička vozila*

1 Introduction

The beginning of rail transport is related to the opening of the railway between Liverpool and Manchester in Great Britain on 15 September 1830. Works on the construction of the railway line between Manchester and Liverpool began as early as the 1820's. The aim was to connect, in this way, the large industrial city of Manchester and its closest deep water port of Liverpool. The length of this railway line was about 56 km.

Further development of the railways in the mid-nineteenth century represents a significant technological innovation that had a very important role and made a huge contribution to industrial development and economic progress of society. Transport of goods by rail has numerous advantages and some of them are: transport of large quantities of goods over long distances, high transport safety, disburdening of roads, lower CO_2 emissions, etc.

The European Union at present treats it as a transporter of the future and it endeavours to reaffirm railway transport on European level, with the requirement of a competitive, secure and quality transport of all kinds of goods. Achieving these goals requires, among other things, the construction of modern train wagons adapted to market challenges, specific technological requirements and systems that allow rapid performance of loading/unloading operations [1].

In addition to the construction of modern towed railway vehicles, the construction of new and reconstruction of old railways lines is essential. Waldemar [2] in his paper discusses the reconstruction of the railway line between Vinkovci-Osijek. The railway line is equipped for the movement speed of the train of 80 km/h and a weight of 22,5 kN per axle, i.e. 80 kN/m, with a total length of 33 339 m with a curve radius of 265 m to 2000 m.

Cylindrical roller bearings for railway vehicles are key components in the wheel assembly of towed vehicles and their failure may lead to large-scale damage. Bearing temperature is one of the most important parameters of bearings, by whose monitoring it is possible to determine the condition of bearings in exploitation. For this reason it is important to know the effect of different lubricants (greases) to the change in bearing temperature over time. Cylindrical roller bearings for railway vehicles are usually lubricated with grease housed in a closed housing together with axle assembly to ensure proper lubrication. As the bearing rotates, the grease comes into contact with the rollers and the rings and after a certain period of time this leads to the mechanical degradation of greases. It is therefore important to replace the grease before it loses its mechanical properties. Based on the tests which were conducted in the winter and summer periods (in the winter period ambient temperature was −15 °C and in the summer period 20 °C), the temperature value in the steady temperature state depending on the type of grease used was obtained. The temperature in winter period ranged from 15 ÷ 51 °C, and in the summer period 33÷59 °C [3].

Preventive maintenance of the wheel assembly and overhaul of bearings in exactly prescribed time intervals is essential to the exploitation life of the bearing. If the overhaul of bearings is not performed in defined time intervals, consequences such as damage to the axle, damage of the part of the railway and train derailment may arise. For this reason it is very important to change the inner bearing ring before its damage due to the fatigue of the material occurs [4]. The authors analysed the important parameters that had led to damage of the part of the railway line and train derailment. Bearing was damaged due to the fatigue of the material, which led to cracking of the inner ring. Due to the slippage of the inner ring on the axle, there had been generating of large amounts of heat which led to changes in the structure of the material in the bearing and on the surface of the axle.

The heat generated due to the rolling of the wheel on the rail is transferred to the entire wheel assembly of towing and towed vehicles. Cole et al. [5] examined this influence by experimental and computational modelling. The generated heat is given to the flange of the wheel and temperature measurement in specific points of the wheel assembly was made by using thermocouples placed in characteristic points. The paper has not discussed the generation of heat due to rotation of the bearing, which has great influence on its temperature.

Realization of movement of railway vehicles creates a problem that is associated with the process of wear of the elements of the wheel – rail. Wear between the wheel and the rail is a complex phenomenon that depends on many factors. One of the main areas in which there is a process of wear and tear is in the lateral contact between the wheel and rail flange (wheel–rail flange contact) [6]. Due to wear, heat is generated in two different places between the wheel and the rail head and between a rail flange of the wheel when the train moves. In this paper, the dimensions of the surfaces that are in contact when the train is moving are determined and the forms used to calculate these surfaces are also shown. When contact between the wheel and rail head occurred, the maximal temperature was 350÷400 °C for the speed of the train from 210 km/h for the rectilinear movement, and when the train was moving on a curve at a speed of 60 km/h and curve radius of 600 m, the temperature ranged from 200÷250 °C. In case of wheel–rail flange contact (side contact) maximum temperature was generated at the maximum speed of the train movement and it was 600÷700 °C, and when the train was moving at a speed of 60 km/h and curve radius of 600 m, the temperature ranged from 600÷700 ° C.

Milošević and others [7] in their work determine the temperature value of the wheel during the braking of the train, due to the contact of the brake discs with the wheel. By using finite element method they obtain the temperature value of the wheel at high and low pressure of the discs with the wheel. The analysis of breaking was performed for the train movement speed of 20, 40 and 60 km/h and for braking time of 300 seconds. The maximum temperature value is obtained at the maximum speed of the train movement of 60 km/h and high pressure of the discs to the wheel and this temperature is 772,5 °C. The minimal temperature is for the speed of the train movement of 20 km/h and a low pressure is 172,3 °C.

This paper presents a mathematical model for the prediction of the thermal behaviour of a cylindrical roller bearing for the towed railway vehicles. The advantage of this mathematical model in relation to the mathematical model presented in the paper [8] is that the heat generated due to lubrication, under radial and axial loads is being determined particular. We should also consider the fact that the generated heat for three values of cant height on one side of the railway was calculated h = 110, h = 140 and h = 180 mm.

2 Material and methods

2.1 The mechanisms of generating heat

This paper deals with the heat generated only in the bearing (excluding the heat generated by the wheel moving across rail) which is calculated on the basis of the friction torque due to lubrication and friction torque under load (axial and radial) [9].

2.1.1 Friction torque determination

$$M = M_0 + M_1 + M_2,\ \mathrm{N\cdot mm} \tag{1}$$

M_0 - the friction torque due to lubrication, N·mm
M_1 - the friction torque due to radial load, N·mm
M_2 - the friction torque due to axial load, N·mm

$$M_0 = 10^{-7} \cdot f_0 (\upsilon \cdot n)^{\frac{2}{3}} \cdot d_{\mathrm{m}}^3, \tag{2}$$

f_0 - coefficient dependent on the type of bearing and on the type of lubricant (for roller bearing it is 3),
υ - kinematic viscosity of the lubricant υ = 18,
n - number of bearing speed,
d_{m} - mean diameter of the axle box d_{m} = 190 mm.

$$M_1 = f_1 \cdot F_{\mathrm{r}} \cdot d_{\mathrm{m}}, \tag{3}$$

$$F_{\mathrm{r}} = \frac{G_1}{4}, \tag{4}$$

$$G_1 = \frac{G}{n_{\mathrm{v}}}, \tag{5}$$

f_1 - coefficient is dependent on the type of bearing and for roller bearing it is 0,0003÷0,0004,
F_{r} - radial load of the bearing (F_{r} = 55 181 N),
G_1- radial load per one one axle (G_1 = 220 725 N),
G - weight of the vehicle,
n_{v} - number of wheel sets.

$$M_2 = f_2 \cdot F_{\mathrm{a}} \cdot d_{\mathrm{m}}, \tag{6}$$

f_2 - based on n, d_{m}, υ and F_{a}/A,

$$A = k_{\mathrm{B}} \cdot 10^{-3} \cdot d_{\mathrm{m}}^{2,1}, \tag{7}$$

A - area,
F_{a} - axial load of the bearing.

Axial load of the bearing depends on the movement speed of the train on a curve, the curve radius and cant height of one side of the rail.

2.1.2 Determination of the amount of heat generated developed in the bearing

$$Q = Q_0 + Q_1 + Q_2,\ \mathrm{W} \tag{8}$$

$$Q_\mathrm{u} = M_\mathrm{t} \cdot \omega, \tag{9}$$

$$\omega = \frac{\pi \cdot n}{30}, \tag{10}$$

Q_0 - heat generated due to lubrication, W
Q_1 - heat generated due to the radial load, W
Q_2 - heat generated due to axial load, W

$$Q_0 = 1{,}05 \times 10^{-4} \cdot n \cdot f_0 (\upsilon \cdot n)^{\frac{2}{3}} \cdot d_\mathrm{m}^3 \cdot 10^{-7}, \tag{11}$$

$$Q_1 = 1{,}05 \times 10^{-4} \cdot d_\mathrm{m} \cdot n \cdot f_1 \cdot F_\mathrm{r}, \tag{12}$$

$$Q_1 = 1{,}05 \times 10^{-4} \cdot d_\mathrm{m} \cdot n \cdot f_2 \cdot F_\mathrm{a} \cdot 0{,}1. \tag{13}$$

The generated heat is in a function of the movement speed of the train in rectilinear motion and movement speed on a curve, the curve radius, cant height of the one side of the railway. Tab. 1 shows the values of heat generated at cant height of h = 110, h = 140, h = 180 mm and a curve radius R = 500 m.

Table 1 Values of the generated heat

Velocity of the train movement (km/h)	Rectilinear	20	40	60			80			100		
	On the curve	20	40	40	50	60	40	50	60	40	50	60
Cant height h = 110 mm												
Generated heat, W		138	209	270	242	194	329	298	256	393	363	320
Cant height h = 140 mm												
Generated heat, W		178	277	341	292	236	402	354	298	470	421	370
Cant height h = 180 mm												
Generated heat, W		217	357	420	355	300	484	418	365	551	486	432

3 Heat transfer mechanisms in the bearing

Mechanisms of heat transfer in bearing are convection due to the rotation, conduction between the inner ring and the axle and the outer ring and housing.

3.1 Convection due to rotation of the bearing

Heat transfer through the bearing is realized only between the bearing and the surrounding air. The absorbed heat of the grease is not treated in this paper. Since due to small differences in temperature radiation can be neglected, the heat transfer coefficient is calculated from the conditions of the air flow through the bearing in the turbulent flow.

In this transfer the total rate of air flow due to the rotation of the bearing is calculated from the axial and tangential components.

Area for axial air flow between inner and outer track of rolling:

$$A_\mathrm{ax} = \frac{\pi}{4} \cdot \left(D_1^2 - d_1^2 \right),\ \mathrm{m} \tag{14}$$

D_1 and d_1 see the references [5] or in a catalog of bearings.

The axial velocity can be determined as a flow velocity between two cylinders from the relation:

$$u_\mathrm{ax} = \frac{V}{A_\mathrm{ax}} = \frac{4 \cdot V}{\pi \cdot (D_1 + d_1)},\ \frac{\mathrm{m}^2}{\mathrm{s}} \tag{15}$$

whereby V is air flow volume obtained from the continuity equation:

$$V = u_\mathrm{sr} \cdot A_\mathrm{s} = \frac{u}{2} \cdot B \cdot s = \frac{1}{2} \cdot d_\mathrm{m} \cdot \omega \cdot s \cdot B,\ \frac{\mathrm{m}^2}{\mathrm{s}} \tag{16}$$

Medium air speed through the cross-section was considered in the previous relation $A_\mathrm{s} = B \cdot s$.

The tangential component of the flow velocity at the mean diameter is obtained from the relation for the flow of air between the movable and immovable cylinder:

$$u_\mathrm{at} = \frac{\omega \cdot d_\mathrm{m}}{2} = \frac{\pi \cdot f \cdot (D + d)}{4}, \tag{17}$$

whereby:
f - is the bearing frequency, Hz
D - is diameter of the outer ring, m
d - is diameter of the inner ring, m

The resulting air velocity in reversing bearings is obtained from the axial and tangential components.

$$U = \sqrt{u_\mathrm{ax}^2 + u_\mathrm{at}^2},\ \frac{\mathrm{m}^2}{\mathrm{s}} \tag{18}$$

Convection coefficient is calculated according to:

$$\alpha = c_0 + c_1 \cdot U^2,\ \frac{\mathrm{W}}{\mathrm{m}^2\mathrm{K}} \tag{19}$$

c_0 and c_1 are constants obtained experimentally (c_0 = 10 a c_1 = 5) [9].

Tab. 2 shows the values of the coefficients of convection in reversing bearings depending on the bearing speed.

Table 2 The values of the coefficients of convection in reversing bearings depending on the bearing speed

Bearing speed		The coefficient of heat transfer, W/(m^{2}K)
rev/min	km/h	
115	20	11,6
230	40	16,6
288	50	20,3
346	60	24,9
461	80	36,5
577	100	51,6

3.2 Heat conduction between the ring and the housing, i.e. the ring and the axis

Heat conduction depends on the clearance between the outer ring and the housing and inner ring and the axis. Thermal conductivity between the two elements can be determined on the basis of relations [10]:

$$\lambda_{ij} = \frac{\ln\left(\frac{r_j}{r_i}\right)}{\frac{\ln\left(\frac{r_j}{r_1}\right)}{\lambda_j} + \frac{R_\mathrm{w}}{r_1} + \frac{\ln\left(\frac{r_1}{r_i}\right)}{\lambda_i}}, \tag{20}$$

whereby λ_i and λ_j are thermal conductivity of the ring or housing.

Other markings are shown in Fig. 1.

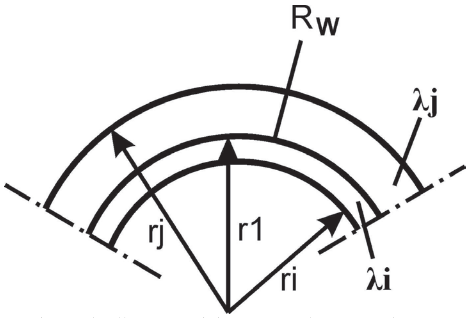

Figure 1 Schematic diagram of the contact between the outer ring and the housing [10]

In the previous relation R_w represents thermal contact resistance at the contact point between the ring and the housing and it can be calculated from the relation:

$$R_\mathrm{w} = \frac{r_1}{\lambda_{ij}} \cdot \ln\left(\frac{r_1 + \Delta}{r_1}\right), \tag{21}$$

where Δ - clearance between the observed elements.

It is assumed that the clearance is the same along the entire circumference between the ring bearing and the housing.

In a similar manner, by replacing the radius mark and the clearance value between the inner ring and the axle, conductivity at this contact point is determined.

Tab. 3 shows values of the heat conduction coefficient determined based on previous relation.

Table 3 Values of the conduction coefficient between the outer ring and housing, and the inner ring and the axle

Contact point	Conduction coefficient λ (W/m^{2}K)
Outer ring /housing	90,9
Inner ring/axle	60,48

4 Modelling of the thermal behaviour of the cylindrical roller bearing

Modelling of the thermal behaviour of the cylindrical roller bearing for rail vehicles has been done by applying the program system based on the finite elements method.

View of the wheel assembly with the cylindrical roller bearing is given in Fig. 2.

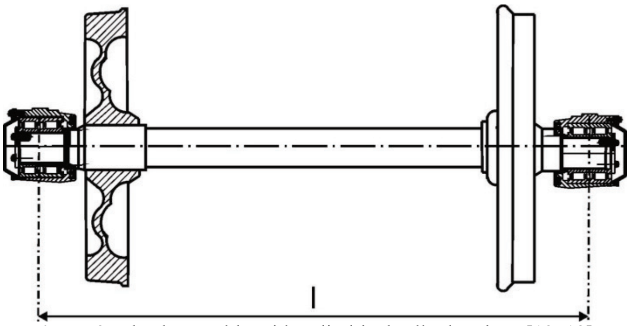

Figure 2 Wheel assembly with cylindrical roller bearings [12, 13]

Model of the wheel assembly for towed vehicles and model of the cylindrical roller bearing (NJ 324 EC.M1C4 VA301, DIN EN 12080) which is used for the axle box of the wheel assembly modelled by applying the program system PTC Creo Parametric is shown in Fig. 3 [11]. In the modelling of the assembly in question, it was taken into account that between the shaft and the inner bearing ring there should be the fit press in the range of 37 to 74 μm in accordance with 3250.VPI04-3. For the permissible maximum temperature of bearing in exploitation (150 °C) the fit press change is 3 μm, due to very small differences of the linear expansion coefficient regarding the materials of the shaft and bearing ring.

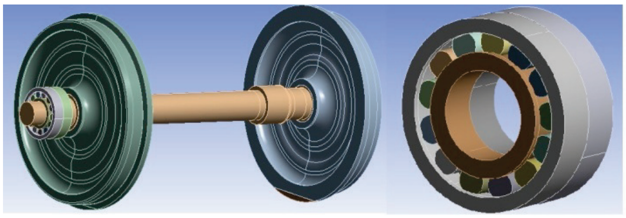

Figure 3 Model of the wheel assembly and the cylindrical roller bearing

Adjusting of the coordinate system, choice of contact pairs (CONTA 174, 26 contact pairs), defining of the generated amount of heat in the bearing, selection of the finite element type (SOLID 87, mesh of 8021 elements and 31 790 nodes) and defining elements between which heat conduction and convection exists is done as part of pre-processing.

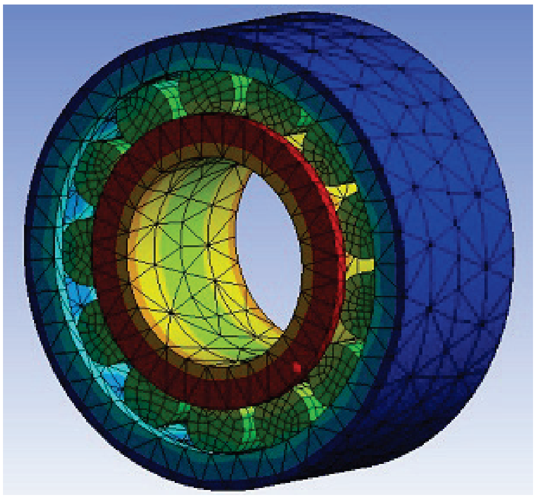

Figure 4 Graph of the layout of temperatures per bearing elements

After calculating and final post-processing, the graph of the layout of temperatures per bearing elements for the number of bearing speed n = 450 rpm, i.e. train movement speed of 80 km/h, height of the cant h = 110 mm and train movement speed round the curved rail of 40 km/h (Fig. 4).

Based on the results obtained with mathematical modelling, maximal bearing temperature of 50 °C has been determined, as well as the time of reaching the stationary temperature condition of 100 min.

5 Results and discussions

Results of mathematical modelling shown in the following figures have been systematized in such a way that they show the effect of the cant height of one side of the rail on the value of the bearing temperature at different speeds of train movement.

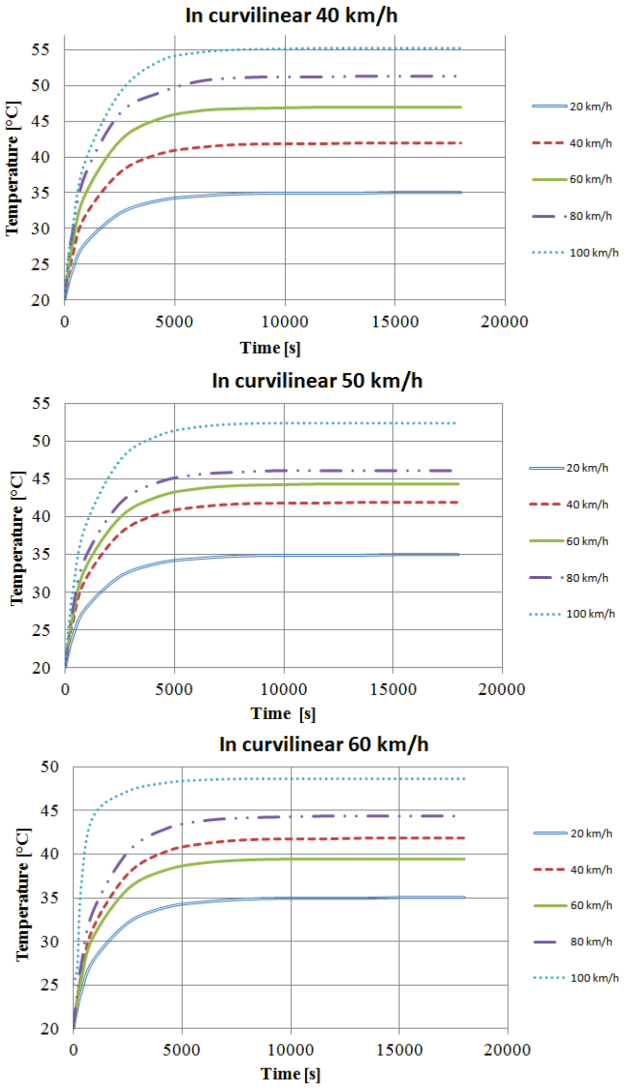

Figure 5 Change in bearing temperature at cant height h = 110 mm

Analysis of the thermal behaviour has been done for the curve radius of R = 500 m, cant height h = 110 m, h = 140 and h = 180 mm, ambient temperature of 20 °C as well as for speeds:

- 20 km/h rectilinear and 20 km/h curvilinear,
- 40 km/h rectilinear and 40 km/h curvilinear,
- 60 km/h rectilinear and 40 km/h curvilinear,
- 60 km/h rectilinear and 50 km/h curvilinear,
- 60 km/h rectilinear and 60 km/h curvilinear,
- 80 km/h rectilinear and 40 km/h curvilinear,
- 80 km/h rectilinear and 50 km/h curvilinear,
- 80 km/h rectilinear and 60 km/h curvilinear,
- 100 km/h rectilinear and 40 km/h curvilinear,
- 100 km/h rectilinear and 50 km/h curvilinear,
- 100 km/h rectilinear and 60 km/h curvilinear.

Fig. 5 shows a change in the bearing temperature over time at train speeds of 20, 40, 60, 80 and 100km/h for rectilinear movement, speeds of 40, 50 and 60 km/h for curvilinear movement, and cant height h = 110 mm.

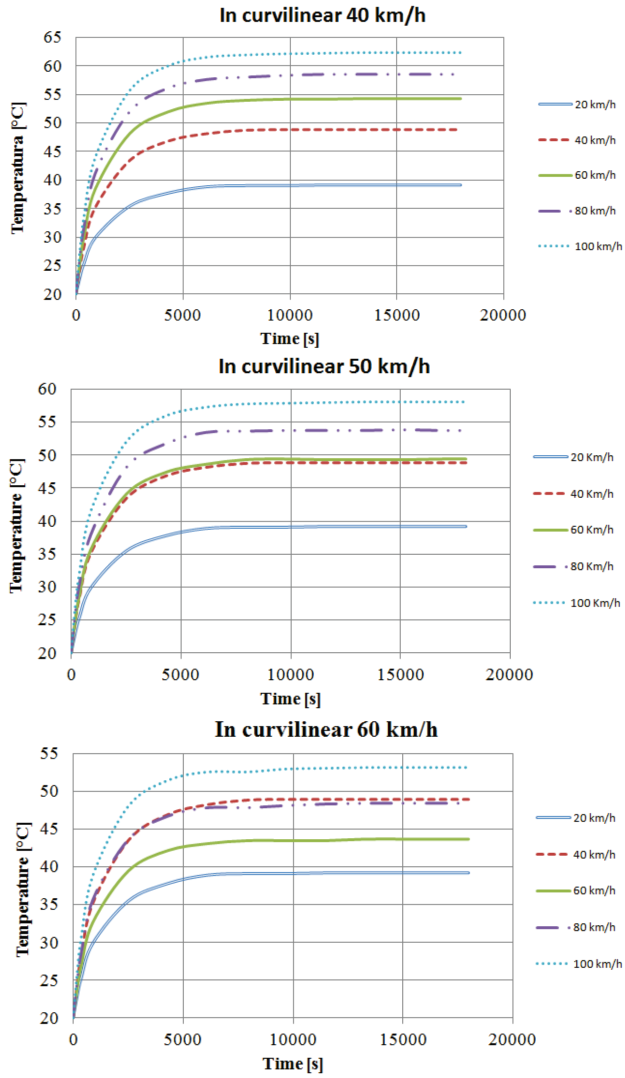

Figure 6 Change in bearing temperature at cant height h = 140 mm

Based on the results shown in Fig. 5, it is seen that the maximal bearing temperature occurs at train speed of 100 km/h for rectilinear movement and speed of 40 km/h for curvilinear movement. Maximal bearing temperature for train speed of 100 km/h is 55 °C, and for train speed of 80 km/h the temperature is 51°C.

Also, the diagram prompts the conclusion that by increasing speed when rounding a curve, the bearing temperature is reduced, provided other limitations are observed.

Fig. 6 shows the values of bearing temperatures for different train speeds at cant height on one side of the rail h = 140 mm.

Based on the results shown in Figure 6, it can be seen that the maximal temperature to which a bearing is heated is 63 °C for maximal train speed at rectilinear movement of 100 km/h and curvilinear movement speed of 40 km/h.

Based on Fig. 6 and diagram which shows the train speed while rounding a curve of 60 km/h and Tab. 1, a case in point, showing that greater generated heat may give lower bearing temperature, is observed.

This is the result of a higher convection coefficient which appears at a higher number of bearing speed (speeds). At speeds (rectilinear and on a curve) of 40 km/h, generated heat amounts to 277 W, while at rectilinear movement speed of 80 km/h and on a curve of 60 km/h the generated heat is 298 W. Generated heat at speeds of 40 km/h and convection coefficient of 16,6 W/(m^{2}K) results in a bearing temperature of 49 °C, while at rectilinear movement speed of 80 km/h and 60 km/h on a curve and convection coefficient of 36,5 W/(m^{2}K), bearing temperature of 48 °C is obtained.

Fig. 7 shows the values of bearing temperatures for different train speeds with cant height on one side of the rail h = 180 mm.

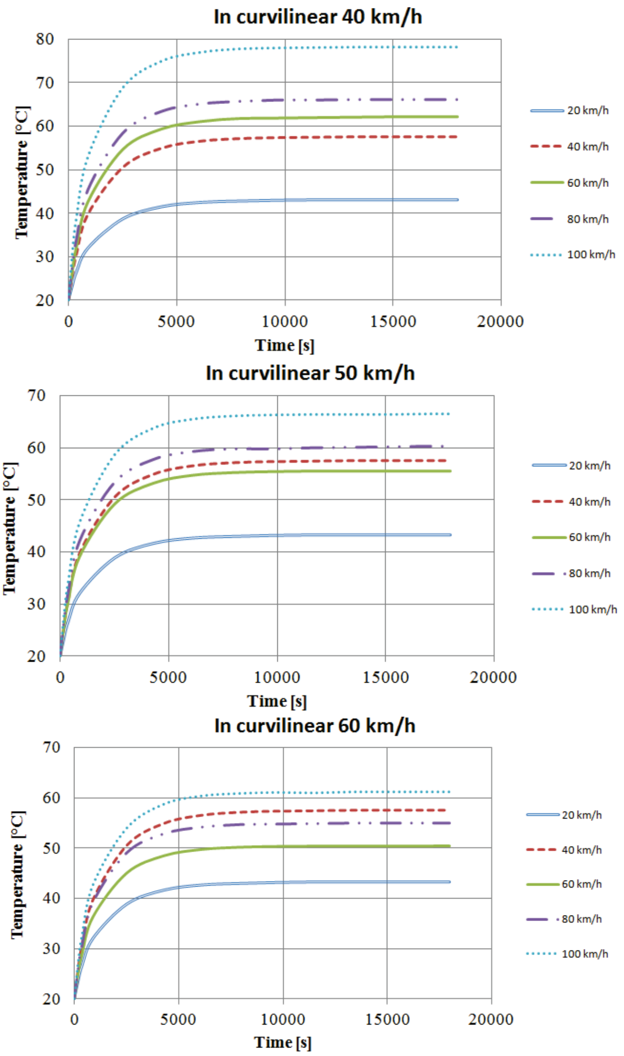

Figure 7 Change in bearing temperature at cant height h = 180 mm

By observing the results from Fig. 7 it is seen that the maximal bearing temperature at train rectilinear movement speed of 100 km/h and speed on a curve of 40 km/h amounts to 79 °C. Increase of speed while rounding a curve, generated heat is lowered and inherently bearing temperature for higher speeds of train movement (60, 80, 100 km/h).

Fig. 7 also shows that bearing temperature at train speeds of 40km/h is higher than at train speeds of 20 km/h which is the result of higher generated heat at higher train speed.

By analysing the results shown in Figs. 5, 6 and 7, it can be concluded that the highest bearing temperature at cant height h = 180 mm, train speed of 100 km/h at rectilinear movement and speed on a curve of 40 km/h amounts to 79 °C. At these speeds and cant heights of h = 110 mm and h = 140 mm bearing temperature has significantly smaller values and amounts to 55 °C, for h = 110 mm and 63 °C for h = 140 mm.

Fig. 8 shows the effects of speed on a curve on bearing temperature for train rectilinear movement speeds of 60, 80 and 100 km/h and cant heights h = 110, h = 140 and h = 180mm.

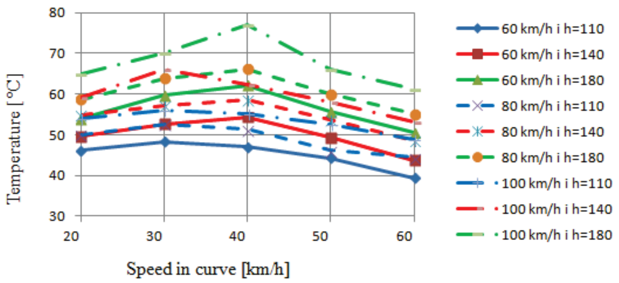

Figure 8 Effect of speed movement on a curve on bearings temperature for different train rectilinear movement speeds and different cant heights

Based on the result chart, it can be ascertained that maximal bearing temperature at speed of 40 km/h on a curve and for cant height h = 180 mm. Also, it can be stated that by increasing cant height, the bearing temperature is also increased.

Effects of the cant height on the value of bearing temperature for different train speeds at rectilinear movement and movement round a curve are shown in Fig. 9.

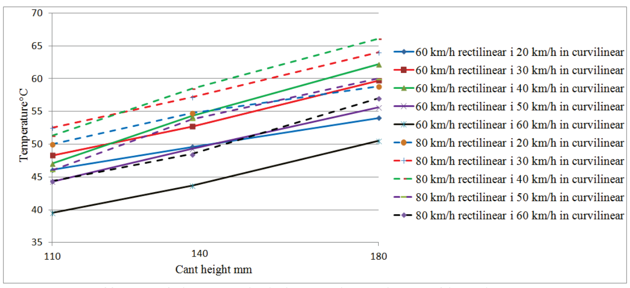

Figure 9 Effects of the cant height on the values of bearing temperatures for different train speeds at rectilinear and movement round a curve

By increasing the cant height, bearing temperature is increased at all train speeds. Maximal train rectilinear movement speed of 80 km/h is shown because that is the maximal speed of freight trains in the region.

6 Conclusion

This paper analyses the thermal behaviour of a cylindrical roller bearing for the axle box of the wheel assembly of the towed railway vehicles. Based on the results of thermal behaviour, it may be concluded that maximal temperature which will be developed by the bearing at 79 °C which is lower than the allowed maximal bearing temperature (which relatively little affects the change of the shaft and bearing fit press). It should be taken into consideration that this temperature is reached during 5 hours of speed without stopping the train and train rectilinear movement speed of 100 km/h and curvilinear movement speed of 40 km/h. In practice, the train usually stops at stations and pauses.

Analysis of the thermal behaviour has been done for different rectilinear and curvilinear train movement, curve radius of R = 500 m (curve radius may vary from 250÷2000 m), cant height h = 110, h = 140 and h = 180 mm and ambient temperature of 20 °C. The aim of future research of the said issue is detailed consideration of the effects of the specified parameters and identification of their effects on the values of generated heat.

7 References

[1] Brkljač, N. Proračunski modeli nosećih konstrukcija sa primenom na rešenja železničkih vagona za prevoz tereta. Doktorska disertacija, Fakultet tehničkih nauka, Novi Sad, 2013.
[2] Waldemar, A. Reconstruction of the railway line Vinkovci – Osijek. // Technical Gazette. 17, 4(2010), pp. 553-557.
[3] Lundberg, J.; Parida, A.; Soderholm, P. Running temperature and mechanical stability of grease as maintenance parameters of railway bearings. // International Journal of Automation and Computing. 7, 2(2010), pp. 160-166. DOI: 10.1007/S11633-010-0160-1
[4] Gerdun, V.; Sedmak, T.; Šinkovec, V.; Kovše, I.; Cene, B. Failures of bearings and axles in railway freight wagons. // Engineering Failure Analysis. 14, 5(2007), pp. 884-894. DOI: 10.1016/j.engfailanal.2006.11.044
[5] Cole, D. K.; Tarawneh, M. C.; Fuentes, A. A.; Wilson, M. B.; Navaro, L. Thermal models of railroad wheels and bearings. // International Journal of Heat and mass transfer. 53, 9-10(2010), pp. 1636-1645. DOI: 10.10/6/j.ijheatmasstransfer.2010.01.031
[6] Spirygin, M.; Lee, Soo K.; Yoo, Hee. H.; Kashura, O.; Popov, S. Numerical calculation of temperature in the wheel-rail flange contact and implications for lubricant choice. // Wear, 268(2010), pp. 287-293. DOI: 10.1016/j.wear.2009.08.014
[7] Milošević, M.; Stamenković, D.; Milojević, A.; Tomić, M. Modeling thermal effects in braking systems of railway vehicles. // Thermal science. 16, 2(2012), pp. 515-526. DOI: 10.2298/TSCI120503188M
[8] Blanuša, V.; Zeljković, M.; Živković, A. Prediction of thermal elastic behavior of the cylindrical roller bearing for railway vehicles and calculating bearing life. // Acta tehnica corviniensis-Bulletin of Engineering. 1(2015), pp. 21-26.
[9] Bossmanns, B.; Jay, F. A thermal model for high speed motorized spindles. // International Journal of Machine Tools and Manufacture. 39(1999), pp. 1345-1366. DOI: 10.1016/S0890-6955(99)00005-X
[10] Jedrzejewski, J. Effect of the thermal contact resistance on thermal behavior of the spindle radial bearings. // International Journal of Machine Tools and Manufacture. 28, 4(1998), pp 409-416. DOI: 10.1016/0890-6955(88)90054-5
[11] FKL. // Tehnička dokumentacija za ležaj NJ 324 EC.M C4 VA301. Br. 9560, Temerin, 2009.
[12] Horvath, T. A vasúti kocsik forgóvázai. Műszaki Könyvkiadó. Budapest, 1987.
[13] Sostarics, G.; Balog, V. Vasúti járművek. Tankönyvkiadó, Budapest, 1991.

Authors' addresses

Vladimir Blanuša, M.Sc.
The School of Higher Technical Professional Education
Školska 1, 21000 Novi Sad, Serbia
blanusa@vtsns.edu.rs

Milan Zeljković, Ph.D.
University of Novi Sad
Faculty of technical sciences
Trg Dositeja Obradovića 6, 21000 Novi Sad, Serbia
milanz@uns.ac.rs

Branko M. Milisavlevich, Ph.D.
The School of Higher Technical Professional Education
Školska 1, 21000 Novi Sad, Serbia
branmyl@gmail.com

Branko Savić, Ph.D.
The School of Higher Technical Professional Education
Školska 1, 21000 Novi Sad, Serbia
savic@vtsns.edu.rs