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DETERMINATION OF FRICTION HEAT GENERATION IN WHEEL-RAIL CONTACT USING FEM

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Abstract. *The modeling of the friction heat generation has become increasingly important in product design process including areas such as electronics, automotive, aerospace, railway (e. g. wheel and rail rolling contact, braking systems, and so on), medical industries, etc. Determination of generated friction heat in the contact of wheel and rail is important for understanding the damage mechanisms on these two bodies such as wear. This paper presents a method to determine the friction generated heat in contact of wheel and rail during normal operation using transient structural-thermal analysis in ANSYS software.*

Key Words: *Wheel-rail, Friction, Heat Generation, FEM, Computer Simulation*

1. INTRODUCTION

Today, computer simulation has allowed engineers and researchers to optimize product design process efficiency and explore new designs, while at the same time reduce costly experimental trials. Generated friction heat in some physical processes like contact of rail and wheel during operation is an influential factor for damage forms and other processes.

There are a great number of studies and research papers dealing with a rail/wheel contact problem.

Ertz and Knothe [1] have concluded that the contact of wheel and rail can be investigated very efficiently with Hertzian contact with polynomial approximation. They have also presented methods for the calculation of contact temperatures using Blok's flash temperature formula. The bulk temperature of the wheel increases with time by continuous friction heat. He has shown that the wheel temperature during normal operating condition cannot be more than double the average temperature of the cold wheel. Lyu et al. [2] have

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analyzed wheel/rail contact including the influence of temperature and humidity on the friction coefficient and wear.

Gallardo-Hernandez et al. [3] have compared temperature in a twin-disc wheel/rail contact in simulation and in experiment using a thermal camera. They have used two methods to calculate temperatures and he has carried out an experiment with 0.5 – 5 % slip. They have also given a diagram of the friction coefficient in the function of slip where the friction coefficient varies up to 0.6.

Wu et al. [4] analyzed wheel/rail contact using thermal-elastic-plastic deformation and residual stress after wheel sliding on a rail. They have simulated a sliding contact process by translating the normal contact pressure and the tangential traction across the rail surface. The results indicate that the friction thermal load of contact between wheel and rail has a significant influence on the residual deformation, plastic strain and residual stress at the rail surface.

Since the friction coefficient plays an important role in study of wheel/rail contact, there is a large number of papers which describe methods for friction measurement. Tomeoka et al. [5] have performed friction control between wheel and rail. Firstly, fundamental tests with two-roller-rigs were carried out to evaluate the friction performances of several types of friction modifiers. Then, for the purpose of realizing the friction control with them, authors developed an on-board system, which sprayed friction modifier from a bogie to the top of rail accurately. Areiza et al. [6] did experimental measurement of coefficient of friction in rails using a hand-pushed tribometer.

Numerical simulations can be used to check the old and to develop new and more efficient designs [7]. Milošević et al. [8] have presented the procedure of modeling thermal effects in braking systems of railway vehicles. Miltenović et al. [9] have presented the basic procedure for determination of friction heat generation in the wheel/rail contact using FEM.

2. Wheel-Rail Geometry

2.1 Rail profile

There are 23 rail profiles specified in the Standard EN 13674-1:2011 [10]. This European Standard specifies Vignole railway rails of 46 kg/m and a greater linear mass for conventional and high speed railway track usage.

The two classes of the rail straightness are specified, differing in requirements for straightness, surface flatness and crown profile. Moreover, the two classes of profile tolerances are specified. Fig. 1 represents rail profile 50E2 which is used in further analyses.

2.2 Wheel profile

UIC CODE 510-2 [11] contains the conditions relating to the design and maintenance of wheels and wheel sets for coaches and wagons used on international services. It covers wheel diameters from 330 to 1000 mm and indicates the permissible axle loads from the standpoint of stresses of the metal used for the wheel and the rail.

UIC CODE 510-2 contains detail coordinates of the wheel rim line. It is valid for a nominal track gauge of 1435 mm and cannot be readily transposed to apply to other track gauges. Fig. 2 represents the wheel profile which is used in further analyses.

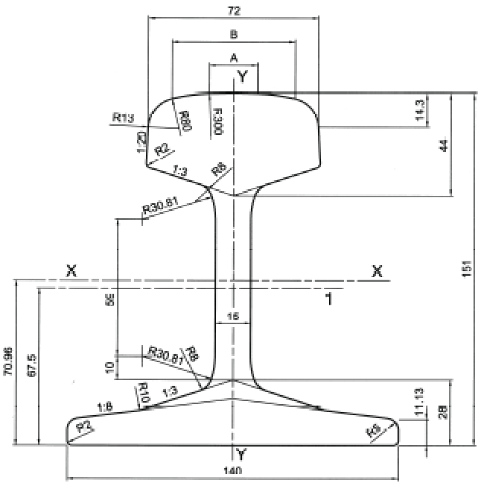


Fig. 1 Rail profile 50E2 [12]

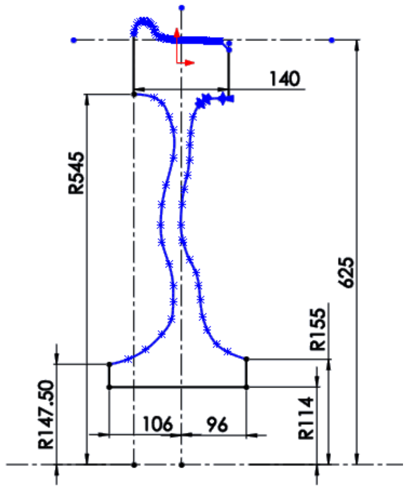


Fig. 2 Wheel profile

3. Analysis Setup

For analysis purposes the simplest case is taken where the wheel speed is constant and when there is only one contact point between wheel and rail. In a case like this slip and friction are also constant. To estimate the generated heat amount is a challenging task that requires a multidisciplinary approach as well as analysis of many influencing factors [12] and the authors of this article offer an approach to simplify the case study.

The basic model of the wheel/rail contact was made in SolidWorks before being exported in ANSYS. For the FEM analysis only the upper part of the rail 10 meters long and the outer ring of the wheel were used.

In the following analyses the ANSYS elements SOLID 226 (3-D 20-Node Coupled-Field Solid, Fig. 3) and SOLID 227 (3-D 10-Node Coupled-Field Solid) were used that support the thermoplastic effect which calculate temperature based on plastic deformation by partial conversion of work into heat.

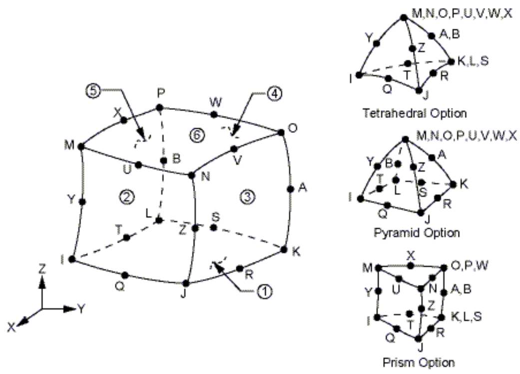


Fig. 3 Element SOLID 226 used in analyses

For meshing of the rail the body SOLID 226 was used. The complete geometry of the wheel body was meshed with SOLID 227 instead of 226 (Fig. 4). Table 1 shows numbers of elements and nodes in the meshed wheel/rail model.

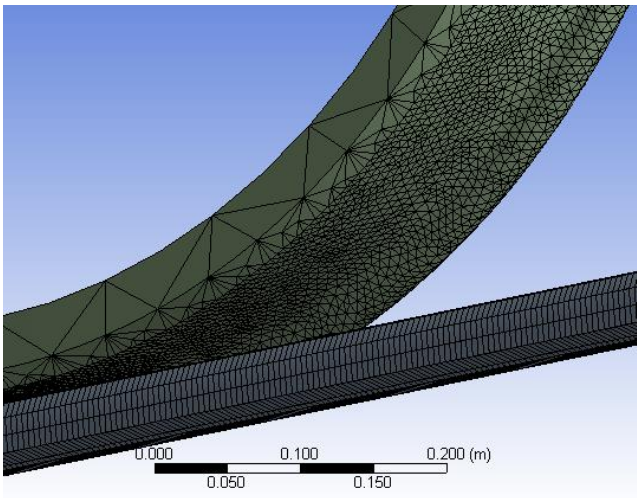


Fig. 4 Mesh of wheel and rail

Table 1 Number of elements and nodes

	Wheel	Rail	Sum
Elements	76502	62400	138902
Nodes	114735	94820	209555

The properties of the material used in the analysis are listed in Table 2.

Table 2 Material properties used in this study

Parameter	Unit	Value
Density	[kg/m^3]	7850
Coefficient of Thermal Expansion	[C^{-1}]	$1.2\cdot10^{-5}$
Young's Modulus	[Pa]	$2\cdot10^{11}$
Poisson's Ratio	[-]	0.3
Tensile Yield Strength	[Pa]	$2.5\cdot10^{8}$
Tensile Ultimate Strength	[Pa]	$4.6\cdot10^{8}$
Isotopic Thermal Conductivity	[W/mC]	60.5
Specific Heat	[J/kgC]	434

The analysis of the friction heat generation in the wheel/rail contact was defined as a direct coupled transient structural-thermal analysis. The rate of frictional dissipation of contact elements in ANSYS is evaluated using the frictional heating factor and is given by:

$$q = FGHT \cdot \tau \cdot v \tag{1}$$

where τ is the equivalent frictional stress, v is sliding rate and *FGTH* is fraction of frictional dissipated energy converted into heat (the default value of 1 was used).[13]

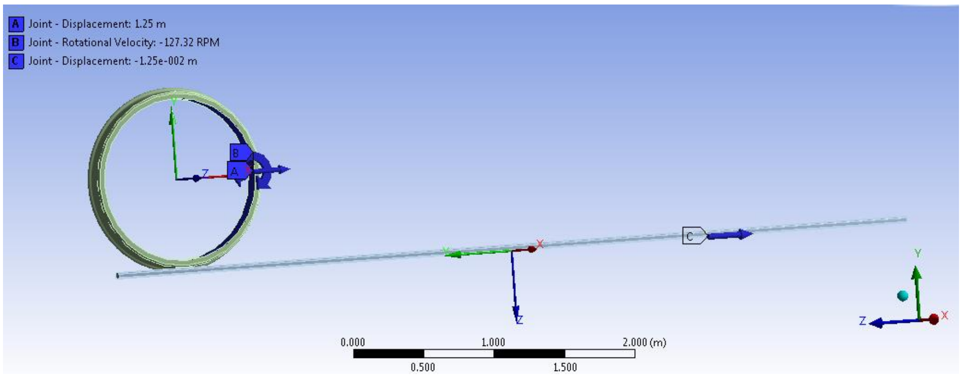


Fig. 5 Analysis setup of wheel and rail for case with sliding 1% and speed 30 km/h

Due to limited computer resources, analyses were limited on 0.6 sec which is enough for wheel to make more than one whole rotation. During normal operation there is sliding between the wheel and the rail. In order to take into account the sliding effect, the rail was translated during an analysis in the same time with the wheel in the same direction 1%, 2% or 3%, which represent some of slip ratios [14]. This moving of the rail represents the sliding that is expected during the normal train operation. For analyses purposes, the wheel was analyzed with the speeds of 30 km/h and 60 km/h. However, because of the slip ratio, the exact speeds of the wheel during the simulation were 29.7 km/h and 59.4 km/h in the case when the sliding was 1%, 29.4 km/h and 58.8 km/h for 2% sliding and 29.1 km/h and 58.2 km/h for 3% sliding. At the same time, the pressure load on the wheel was 5 t and 10 t representing the weight of the wagon. The friction coefficient between the wheel and the rail was 0.1 or 0.3.

5. RESULTS

Tables 3 and 4 present the results for generated temperatures of the analyzed friction heat generation of the wheel and the rail in the contact area. These results represent relative increase of the temperature, since, for the simulation circumstances, the atmosphere temperature was 22 °C and weight of 5 t. In Table 3 temperatures for speed 30 and 60 km/h and the friction coefficient 0.1 are given, while Table 4 gives temperatures for 30 and 60 km/h and the friction coefficient 0.3.

Table 5 gives temperatures on wheel and rail for the speed 30 km/h, the friction coefficient 0.3 and with different weights (5 and 20 t). Temperatures in the tables are average values of the highest temperatures of wheel or rail area.

Table 3 Relative temperatures increase [K] on wheel and rail for speed 30 and 60 km/h and friction coefficient 0.1

	Wheel		Rail	
	30 km/h	60 km/h	30 km/h	60 km/h
Sliding				
1%	6.4	8.1	4.4	5.7
2%	13.1	16.1	8.8	10.4
3%	19.7	21.5	13.3	16.2

Table 4 Relative temperatures increase [K] on wheel and rail for speed 30 and 60 km/h and friction coefficient 0.3

	Wheel		Rail	
	30 km/h	60 km/h	30 km/h	60 km/h
Sliding				
1%	7.5	8.9	4.9	6.3
2%	15.2	16.6	9.6	11.6
3%	20.8	22.9	15.7	17.8

Table 5 Relative temperatures increase [K] on wheel and rail for speed 30 km/h, friction coefficient 0.3 and different weights

	Wheel		Rail	
	5 t	10 t	5 t	10 t
Sliding				
1%	7.5	10.8	4.9	7.11
2%	15.2	20.2	9.6	13.3
3%	20.8	26.3	15.7	19.6

Figs. 6 to 12 show comparisons of temperatures of the wheel/ rail contact areas for varying: speed 30 and 60 km/h, friction coefficient 0.1 and 0.3, sliding 1, 2 and 3 % and weight 5 and 10 t.

Fig. 6 gives comparisons of the temperature increase for speeds of 30 and 60 km/h and the friction coefficient 0.1. Compared with temperatures on the rail, temperatures on the wheel are approximately 66 – 68 % higher for the speed of 30 km/h, also, for the speed of 60 km/h temperatures on the wheel are approximately 70 – 75 % higher.

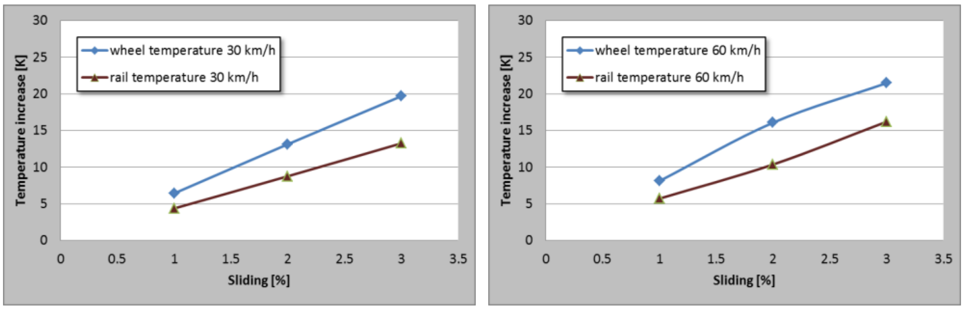


Fig. 6 Comparisons of the temperature increase for speeds of 30 km/h and 60 km/h and friction coefficient 0.1

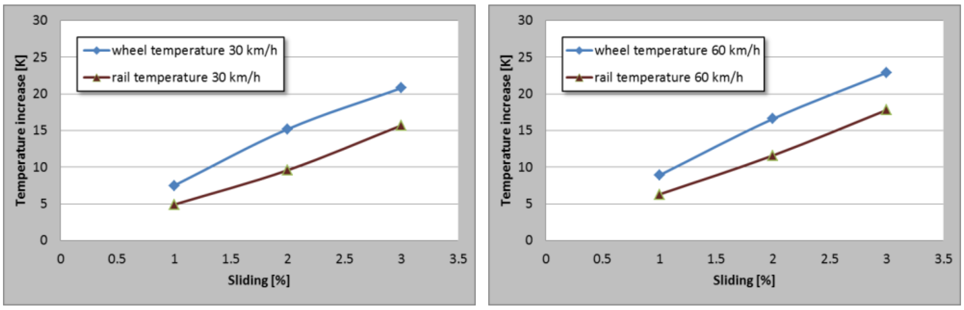


Fig. 7 Comparisons of the temperature increase for speeds of 30 km/h and 60 km/h and friction coefficient 0.3

Fig. 7 represents comparisons of the temperature increase for speeds of 30 and 60 km/h and the friction coefficient 0.3. Compared with temperatures on the rail, temperatures on the wheel are approximately 63 – 74 % higher for the speed of 30 km/h, also, for the speed of 60 km/h temperatures on wheel are approximately 69 – 77 % higher.

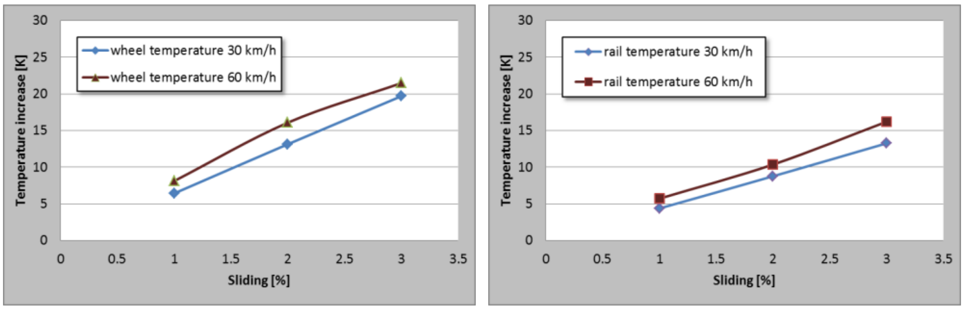


Fig. 8 Comparisons of the temperature increase on the wheel and rail for speeds of 30 km/h and 60 km/h and friction coefficient 0.1

Fig. 8 shows comparisons of the temperature increase on the wheel and rail for speeds of 30 and 60 km/h and the friction coefficient 0.1. Compared for the cases of the speed of 30 km/h, temperatures on the wheel are approximately 16 % higher for the speed of 60 km/h, also, for the rail, temperatures are approximately 19 % higher for the speed of 60 km/h.

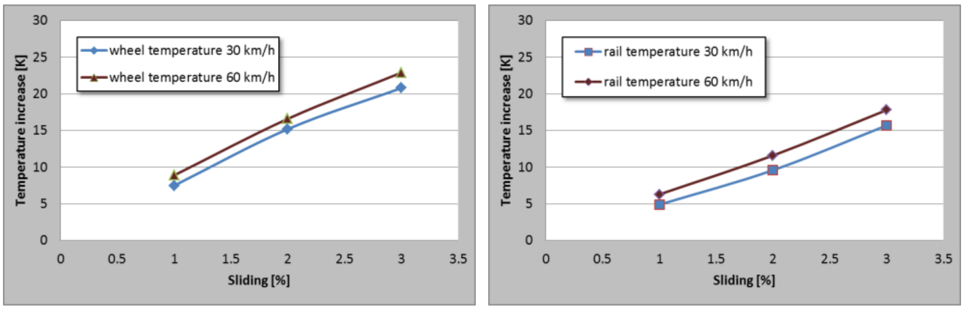
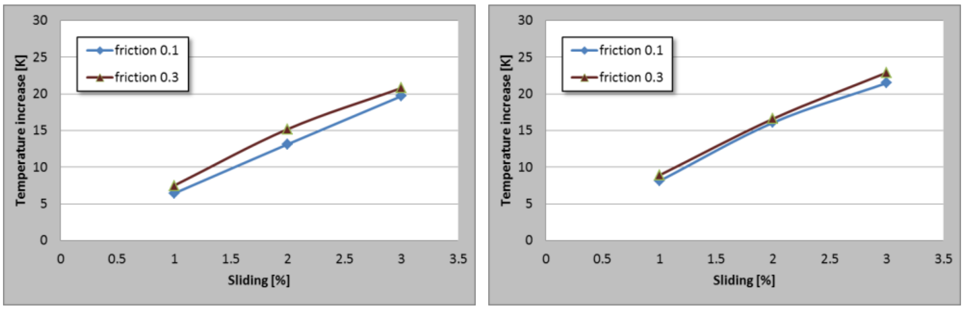


Fig. 9 Comparisons of the temperature increase on the wheel and rail for speeds of 30 km/h and 60 km/h and friction coefficient 0.3

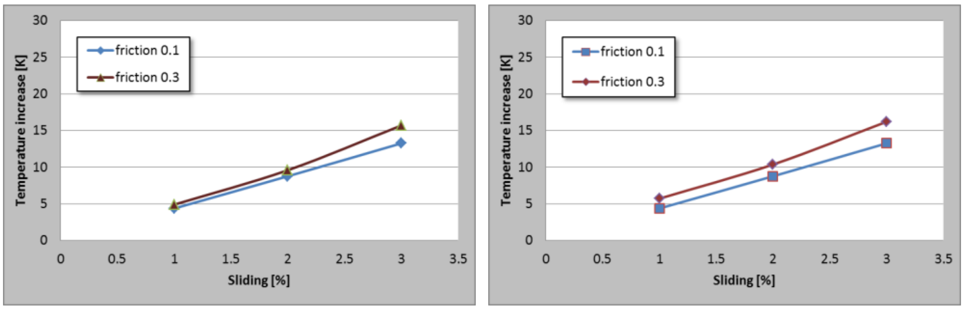
Fig. 9 gives comparisons of the temperature increase on the wheel and rail for speeds of 30 and 60 km/h and the friction coefficient 0.3. Compared for the cases of the speed of 30 km/h, temperatures on the wheel are approximately 12 % higher for the speed of 60 km/h, also, for the rail, temperatures are approximately 17 % higher for the speed of 60 km/h.

Similarly, Fig. 10 represents comparisons of the temperature increase on the wheel for the same speed and friction coefficients 0.1 and 0.3. Compared with the temperatures for the case with the friction coefficient 0.1, temperatures on the wheel for the speed of 30 km/h are approximately 10 % higher for the friction coefficient 0.3, also, for the speed of 60 km/h temperatures on the wheel are some 6 % higher for the friction coefficient 0.3.



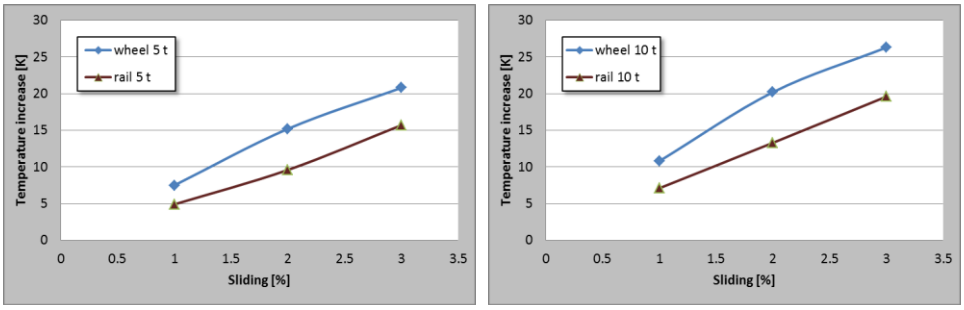
a) 30 km/h b) 60 km/h

Fig. 10 Comparisons of the temperature increase on the wheel for the same speed and friction coefficients 0.1 and 0.3



a) 30 km/h b) 60 km/h

Fig. 11 Comparisons of the temperature increase on the rail for the same speed and friction coefficients 0.1 and 0.3



a) 5 t b) 10t

Fig. 12 Comparisons of the temperature increase on rail and wheel for speed 30km/h, friction coefficient 0.1 and different weights

Fig. 11 gives comparisons of the temperature increase on the rail for the same speed and friction coefficients 0.1 and 0.3. Compared with the temperatures for the case with the friction coefficient 0.1, temperatures on the rail for the speed of 30 km/h are some 14 % higher for the friction coefficient 0.3, also, for the speed of 60 km/h temperatures on the wheel are approximately 10 % higher for the friction coefficient 0.3.

Finally, Fig. 12 depicts comparisons of the temperature increase on the rail and the wheel for the speed of 30 km/h, the friction coefficient 0.1 and for different weights. Compared with the temperatures for the case with the weight of 5 t, temperatures on rail are approximately 31 % higher for the weight of 10 t and for the wheel the temperatures are some 32 % higher.

6. Conclusion

The paper presents an approach to determine the friction generated heat in the wheel/rail contact by using of the FEM transient structural-thermal analysis in ANSYS software.

The performed research shows that temperatures on the wheel are greater than temperatures on the rail (66-75%) in all the cases of slip ratios, speeds and friction coefficients. Moreover, the increase of the sliding leads to increase of friction generated temperature on the wheel/rail contact surface.

The increase of the speed (30 to 60 km/h) leads to 12 – 16 % increase of the temperature on the wheel and 17 – 19 % on the rail. The increase of the friction coefficient (0.1 to 0.3) leads to 6 – 10 % increase of the temperature on the wheel and 10 – 14 % on the rail contact surface. The increase of the weight (5 t to 10 t) leads to 31 – 32 % higher temperatures on the rail and wheel.

Further research should compare temperatures obtained by means of simulation with real temperatures in the exploitation. Also, it would be very interesting to calculate friction generated heat in the wheel rail contact in the cases of acceleration and braking of trains, as well as in the cases when there are two contact points between the wheel and rail.

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