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Wheel heating model to evaluate brake failures

Tulio Rodrigues, Ramiro Jose Chamorro Coneo, Paola Gonzalez Ramos and Auteliano Antunes dos Santos Junior.

Railway Laboratory (LAFER), State University of Campinas (UNICAMP), Campinas, SP, 13083-970, Brazil

Abstract: Railroad brake systems may experience faults during work, such as wheel lockup or disproportional brake traction forces on the rail vehicle. These faults generate unequal braking in the wheels which, consequently, implies on different heating generated on the wheels-shoe interface. This phenomenon contributes to the development of uneven surfaces on the wheels. In extreme situations, rigging failures can even cause the wheels to overheat and thus lead to defects associated with overheating such as flaws and cavitation. In this work, a wheel heating model considering the effect on the brake failures is developed. The model can calculate the temperature along a route and the stresses generated by thermal loads, allowing the estimative of the wheel life in service. Results and analyzes are presented to demonstrate the accuracy of the proposed model.

Keywords: Railways, Heat Transfer, Finite Element Method, Brake Rigging, Brake Temperature.

1. INTRODUCTION

Historically, several regions of Brazil had their development correlated with the presence of railroads, we can notice this especially in the state of São Paulo, where during much of the 20th century, several cities in the interior had their growth tandem to the railway line that cuts the city. Along with the growth of cities, there was also the modernization of the railway line, but some problems are still a great challenge on the subject.

Failures caused in the brake rigging, induce unequal braking forces that overload the brake system (Picanço et. all., 2018). In extreme cases, or in cases where the correct relief of the braking forces does not occur, severe failures can occur resulting in a risk to the safety of operation. So, the thermal study of the brake induced thermal loads phenomenon in railway composition is a very important subject for better understanding of the problem, modernization and safety improvement of railway compositions.

In the literature there are some studies on the heating of wheels due to braking. Many of these works calculate the temperature in a two-dimensional axisymmetric model by the finite element method (Johnson et. all., 1977), others use a three-dimensional quarter wheel model in order to calculate the internal mechanical efforts (Suchánek et. all., 2018). There are also more sophisticated models that include the residual stresses from the manufacturing process in the analysis and combine the effect of mechanical effort from the wheel/rail contact with thermal load and normal pressure in the shoe due to braking (Vakkalagadda et. all., 2015).

Late in 20th century, Johnson et. all. (1977) started studying the residual stresses in a wheel due to braking using a two-dimensional approach to the problem. Later, Minicucci et. all. (2003) used a validation method to compare the manufacturing residual stresses measured by ultrasound with finite element model simulation. Then, Santos et. all. (2008) developed a numerical model based on finite elements of wheel-rail contact that was able to estimate the fatigue life until the appearance of the crack. Late, Vakkalagadda et. all. (2015), developed a model to calculate the temperature in the wheels during their operation. His approach included 3 steps, a train model that calculate the heat generation and exchange rates between the wheel-shoe and wheel-rail interfaces, then a 2D axisymmetric model estimates the amount of heat transfer between the wheel and shoe considering shoes of different material, geometry and thermal properties and finally a full 3D finite element model was used to estimate the temperature on wheel. Also, the methodology was validated with field data. In 2018, Picanço et. all. developed a numerical evaluation of the effect of braking on the service life of railway wheels regarding surface scaling, using the finite element method an axisymmetric two-dimensional model was used to calculate the temperature distributions on the wheel due to the heat treatment in the manufacturing process and due to braking. In 2020, Kuciej et. all., proposed two numerical models, one 2D and one 3D, in finite elements for estimating wheel temperature during repeated intense braking. The results obtained by the models were compared with experimental data obtained through a full-scale dynamometer test for two different shoe compounds. It was noticed that the 2D model is as efficient as the 3D in terms of temperature estimation during long braking at the considered point of interest. The difference between the models did not exceed 3% and the 2D model considerably surpassed the 3D in terms of computational time. Thus, the power of the two-dimensional axisymmetric tool in estimating the temperature along a stretch is evident. Later in 2021, Tomy et. all. developed a novel semi-empirical model for brake design which calculated the temperature distributions in an isotropic and orthotropic type brake rotor, calibrating his model using a similar approach presented in this work. Also, a literature review is presented about disc braking thermal simulation.

In this way, a study considering a simplified wheel heating model to evaluate wheel heating in a course due to brake it's a good approach to the problem. With development, this model could evaluate beyond a simple induced thermal stresses analysis to a full fatigue life analysis considering manufacturing process and track use.

In this work, a train model was developed to. After that, a two-dimensional axisymmetric model was developed in order to estimate the amount of heat transfer between the wheel and shoe considering shoes of different material, geometry and thermal properties. Finally, the finite element method was used to estimate the temperature on the wheel from the results obtained previously. The adopted methodology was validated by comparing the results obtained at the end of the process with data acquired from laboratory tests.

2. METHODOLOGY

In this work, the development of a numerical model of a train wheel in ANSYS was performed, which from heat input data from braking phenomenon could calculated the temperature distribution in the wheel during and after braking. From the thermal modelling of the phenomenon is possible to, subsequently conducted the analysis of which will respond with data relevant to the life and integrity of the part during different conditions of use and also correlate with norms and safety standards adopted by the industry.

As a support base, extensive research was carried out, together with the bibliographic review an article was found (Suchánek et. all., 2018). This article, despite not having validated data, it shares all boundary conditions and input data enabling reproduction work and develop a hybrid approach to the model. The author uses a 3D quarter wheel model that as seen by Kuciej et. all., (2020) could be a good midterm solution in terms of precision versus computational demand.

2.1 MESH STUDY

A quarter wheel model was developed based on a 920mm diameter railway wheel, made of DIN 40Mn4 steel. Two simulation cases were developed considering the use of different materials in the shoe and consequently a different braking power dissipated between the shoe and the wheel. A braking case corresponding to 32 kW was used for the case of a shoe made of cast iron and a power equivalent to 38 kW for the case of a shoe made of composite material, which is most commonly found today in the Brazilian railway scenario.

In Table 1, we can check the thermal properties used in the simulation, in addition, an ambient temperature of 22° C was considered and the data used in the heat input can be checked in Figure 1.

Table 1 - Thermal material properties adapted from (Suchánek et. all., 2018).

Properties	Railway wheel	Air
Density ρ [kg.m^{-3}]	7850	1.170
Heat capacity Cp [J.kg-1.K^{-1}]	486	1100
Thermal coductivity k [W.m^{-1}.K^{-1}]	52	0.026
Emissivity – wheel tread[-]	0.20	-
Emissivity – other surfaces	0.80	
Dynamic viscosity [Pa.s]	-	1.8.10^{-5}

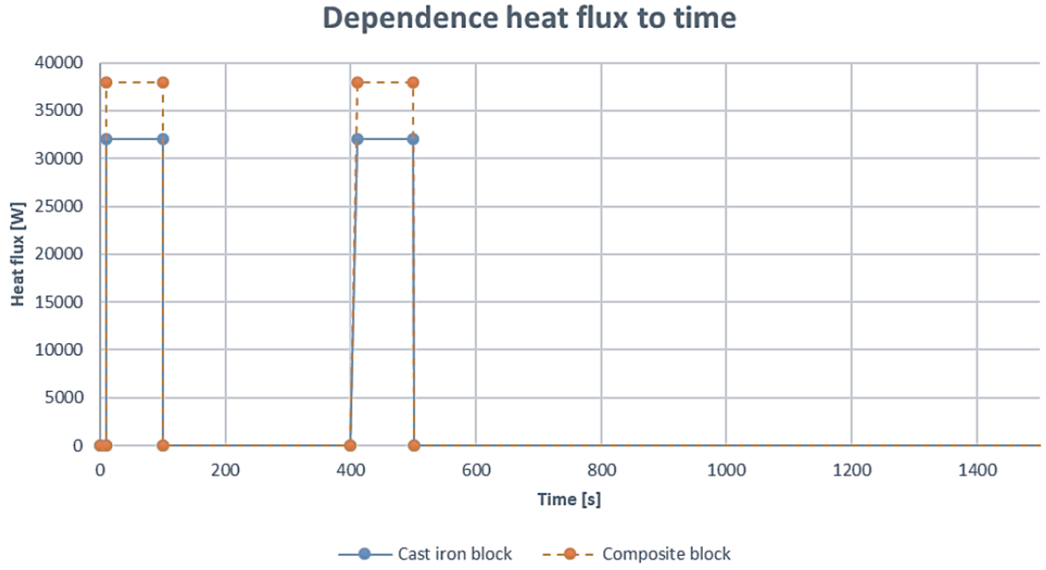

Figure 1 - Braking power dissipated over time (Suchánek et. all., 2018).

As boundary conditions, the same ones followed by the author were adopted, the CAD drawing of the wheel was obtained through reverse engineering considering the main dimensions and a section of the part obtained in the original paper, when simulated the result seen in Figure 2 was observed. The model showed good correlation with the author despite some data that was unavailable and imprecision in the reverse engineering CAD obtaining method.

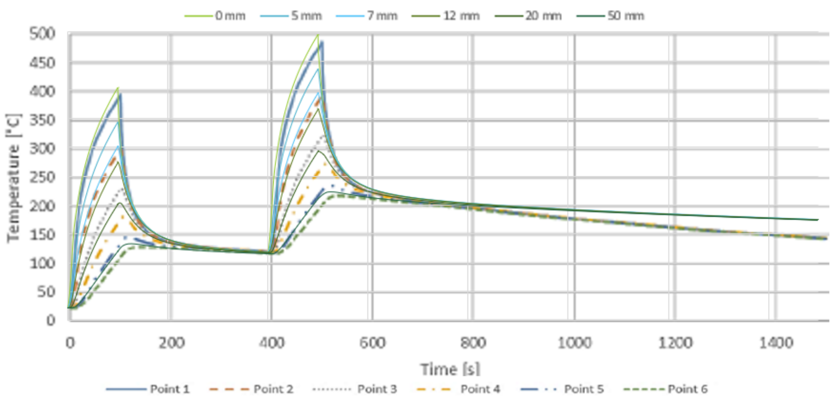

Figure 2 - Temperature data from points inside the wheel comparing the reproduced model with results found by (SUCHÁNEK et al., 2018)

Then, a mesh independence method was started. For this study, the same model proposed previously for the reproduction of the work by (Suchánek et. all., 2018) was considered and the mesh refinement region was discretized into 3 main portions, as seen in Figure 3.

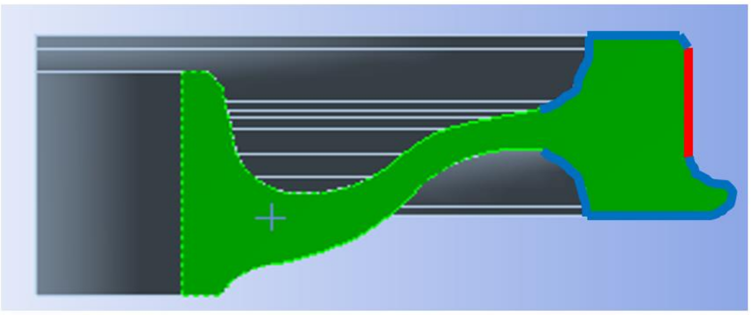

Figure 3 - Regions of interest used in the mesh independence study.

• Contact region mesh: Hatched in red in Figure 3, this region comprises the points at which the heat input due to braking is directly in contact and, consequently, is the running surface of the wheel. It is a region of interest for optimization, since it comprises a fundamental part of the problem.

• Region of interest mesh: Hatched in blue in Figure 3, it is the region where the highest thermal gradients will be located in the part and comprises most of the zone thermally affected by the braking phenomenon. Furthermore, in this region, the portions of the most difficult geometry discretization are composed (sharp curves and details such as the flange); and the refinement of this region is of great importance, in order to obtain a good discretization of the study geometry.

• Fill/base mesh: Hatched in green in Figure 3, it composes the inner region of the part and the other outer portions of the part. Despite composing most of the elements due to its large size compared to the other regions of study, it theoretically has a lower degree of importance in the thermal analysis because it is located in the most distant regions and with lower thermal amplitude.

With these portions defined, 3 grades of refinement were settled for each region, A, B and C (Table 2) and a combinatory analysis was run which the results of the bests setups are shown in Table 3-4.

Table 2 - Parameters used in the independence mesh analysis.

Region	Case A	Case B	Case C
Contact region mesh	2,5 mm	5 mm	10 mm
Region of interest mesh	5 mm	10 mm	15 mm
Fill/Base mesh	10 mm	15 mm	20 mm

Table 3 - Results of the mesh independence study for the thermal case.

Mesh	0 mm		10 mm		Time (Hours)
	Med.	Max.	Med.	Max.	
A - A - A	0	0	0	0	01:42:00
B - B - B	0,940926	8,574	2,12895041	4,57	00:09:13
C - C - C	0,56364	3,05	6,50043802	14,06	00:02:27
A - A - C	0,034339	0,27	0,96266116	2,44	00:10:08
A - C - A	0,007033	0,04	2,32121488	5,18	00:23:15
C - A - A	0,980702	7,39	4,87749587	10,61	00:22:56
C - C - A	0,452512	2,44	9,38847934	20,42	00:09:35
C - A - C	0,817793	7,977	11,6977562	24,64	00:07:24
A - C - C	0,018289	0,13	1,3097686	2,56	00:05:25

Table 4 - Results of the mesh independence study for the thermomechanical case.

Mesh	0 mm		25 mm		Time (Hours)
	Med.	Max.	Med.	Max.	
A - A - A	0	0	0	0	42:55:30
B - B - B	1,14	9,37	2,32	5,79	5:25:32
C - C - C	1,32	13,27	7,52	22,86	2:03:08
A - A - C	0,56	4,25	1,75	6,63	5:55:59
A - C - A	0,17	3,24	1,56	7,82	13:39:20
C - A - A	1,21	14,58	5,89	15,65	13:26:06
C - C - A	0,54	6,58	12,68	52,35	5:44:37
C - A - C	0,93	8,457	14,87	29,52	4:26:20
A - C - C	1,08	7,32	3,21	4,14	3:09:51
A - B - B	0,21	5,24	1,45	5,27	10:34:48

The most refined case (A – A – A) were taken as a reference, then point by point over time the solutions were overlapped and at each point the result difference were calculated. So a arithmetic mean was taken for each case and also the maximum discrepancy value was also taken for analysis.

Thus, it can be noted in the thermal case that, an A – A – C or A – C – C mesh can faithfully represent the problem, as well as in the case of an A – A – A mesh, however, with a lower computational cost. This situation is due to the fact that, the fill/base region of the part has a good part of the elements, however, it is not so relevant for this case study.

In the thermomechanical case, we can observe the performance of some main models in Table 4. It is possible to notice that, unlike the thermal mesh independence case, the refinement near the base influences the results, since the support in this case study is positioned in this region and now the model A – B – B proved to be the model that satisfactorily describes both the thermal part as well as the residual stress from the thermal study in question, reducing 4x the analysis solution time. With these results in hand, the necessary mesh density for each of the analysis were settled.

3. MATERIALS AND METHODS

In the experimental procedures, the railway dynamometric bench present in the structure of LABVIA – Vehicle Via Interaction Laboratory – Unicamp, presented in Figure 4, was used. The bench consists of a complete module that simulates the brake system of a wagon, to apply braking loads on the shoe, and an electric motor that has inertia disks attached to its axle along with the wheel.

Figure 4 - LABVIA railway inertial dynamometer

Two tests rounds composed of ramp tests with varied duration were carried out in order to capture peculiarities related to the test and obtain, together with the repeatability of the test, a greater range of data. The Ramp test procedure consist of the following steps:

1. Brake shoe performance on a ramp will be determined by measuring the friction retarding force produced by a constant intensity application of the brake shoe to the wheel at a constant speed of 30 km/h.
2. Light braking condition implies a normal brake shoe force of 3200 N, with a permissible variation of 4%.
3. The heavy braking condition implies a normal brake shoe force of 4900 N, with a permissible variation of 4 %.
4. The temperature during all tests must not exceed 250 °C and, in addition, a maximum temperature of 80 °C is required for the start of the test.
5. The tests will be repeated from 3 to 5 times for each test composition, so that it is possible to evaluate the repeatability and the influence of the initial temperature condition in the part on the results obtained.
6. In the first round of tests, 4 K-type acicular thermocouple sensors were placed on the lower support.
7. In the second round of tests, one sensor is positioned on the lower support and 3 sensors are positioned on the upper support, as in the first battery, all are K-type acicular thermocouples.
8. Along with the continuous measurement thermocouples, point measurements were collected with a portable infrared thermocouple between tests in order to gauge and calibrate the models.
9. A fan and exhaust fan were used throughout the test, to circulate air through the wheel and brake shoe, to simulate the movement of the wagon during the test, but the wheel and the shoe were enclosed in a acrylic safety box.

In the first tests round, the initial focus was on familiarization with the test equipment, along with repeatability, thus ensuring reliability in the data acquired. 8 continuously heavy test runs were carried out, some in sequence and thus also capturing the phenomenon of cooling during the process in order to calibrate the model, the thermocouples were all located at the bottom of the wheel, just after the shoe, considering the direction of rotation of the wheel. All on the same support spaced 25mm along the contact surface between the wheel and the shoe named Temperature_0 to 3 being Temperature_0 the sensor located near the flange and 3 the sensor on the opposite side.

After a discussion of the results obtained experimentally and an initial comparison with the results obtained by the model, a second round of tests was elaborated, now varying in intensity and period of force application according to test schedule seen in Table 5.

Table 5 - Tests round 2 schedule.

Test	Test Type	Duration
1	Heavy	30 min.
2	Heavy	15 min. (Braking) + 5 min. (Coasting) + 10 min. (Braking)
3	Light	30 min.
4	Heavy	10 min. (Braking) + 10 min. (Coasting) + 10 min. (Braking)
5	Heavy	15 min. (Braking) + 5 min. (Coasting) + 10 min. (Braking)
6	Heavy	30 min.

In order to investigate the influence of the positioning of the sensors on the wheel, temperature acquisition at a point outside the contact region between the shoe and the wheel and acquisition of new data such as the ambient temperature at the location. With this, the upper support was introduced on the bench and 1 control sensor was kept in the lower support (Temperature_0) in order to compare the results obtained in the new round of tests. In the intermittent tests, periods of brake application were alternated with periods of pure rolling in order to better investigate the cooling conditions during the test. The other thermocouples were located in the upper support, being Temperature_1, a sensor displaced 40mm from the centre of the contact region near the flange, Temperature_2, the sensor positioned in the centre of the contact region and Temperature_3, the sensor positioned in the sweetening radius between the side of the wheel and the contact surface. Outside the heat input region via the shoe, in tests 5 and 6, this sensor was disconnected from the support and positioned at a considerable distance from the wheel in order to capture the ambient temperature in the test.

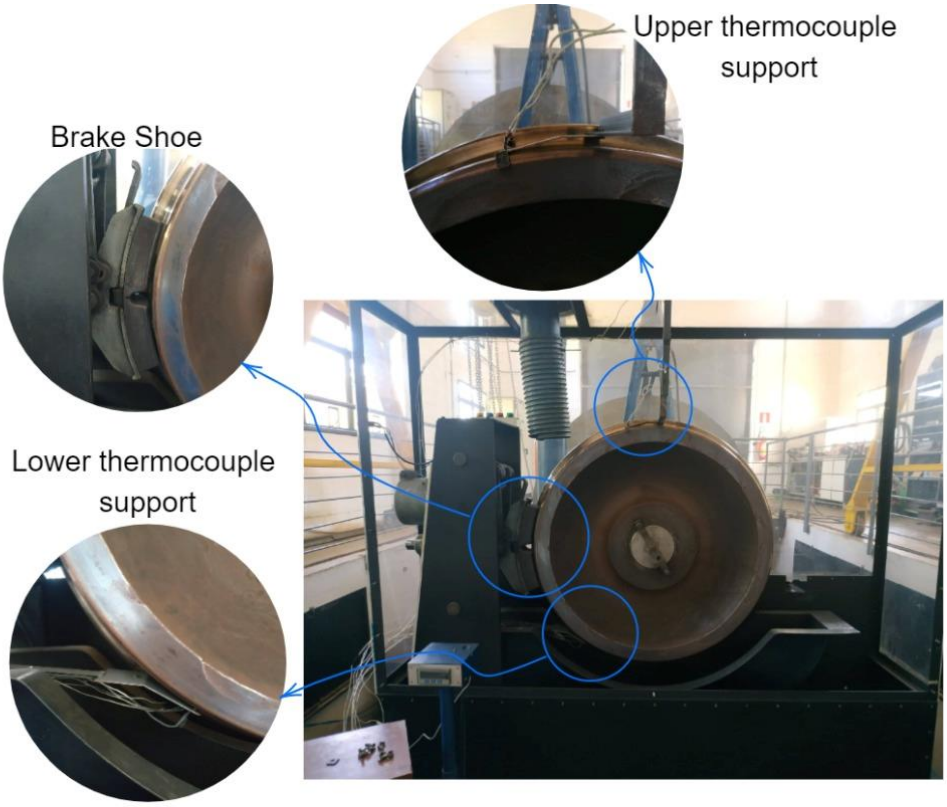

Figure 5 - Positioning of thermocouple supports on the dynamometer bench.

4. ANALISYS AND RESULTS

A comparison between the simulated models and the experimental data of the temperature distribution on the wheel surface is described in this chapter. The first round of tests, the data were compiled and the average of the values of a double heavy braking test was extracted, which was compared with results obtained in simulation. The models had their heat input value from 60% to 90% total heat and convective coefficient variated from 10 to 100 W/m^{2*}°C, in order to find an input configuration that best described the phenomenon of heating due to braking in the current test case. Several models were proposed, but it was observed in the results, characteristics inherent to the altered parameters when comparing the curves of the acquired models with the experimental data.

It was noted that at every moment of heat input and brake withdrawal, that is, at the beginning and end of each test, there was a discrepancy with the model having a more aggressive variation in temperature at these points, compared to the real data. At this point, the influence of the conductivity coefficient of the material used in the simulation was investigated and a smoothing in the discrepancy was noticed. Nevertheless the simulation data always had a different characteristic from the experimental temperature curves observed at this point. The data of the material used until then defined for the steel class used by the wheel according to the AAR standard and already used in the model until then as well as in the model of Picanço et. all. (2019).

In addition, it was noted that for the model to meet the observed maximum and minimum values. A very high value of convective coefficient would have to be used, as we can see in model 4 presented in Figure 6. This model has a convective coefficient 10x greater than that observed by Júnior et. all. (1996) in their determination of the parameter for a wheel in service. This shows a value that is impossible to obtain with the proposed experimental configuration. In view of this, a new round of tests was proposed to investigate the influence of the sensor position on the observed temperature and also its positioning in the wheel profile for the same reason.

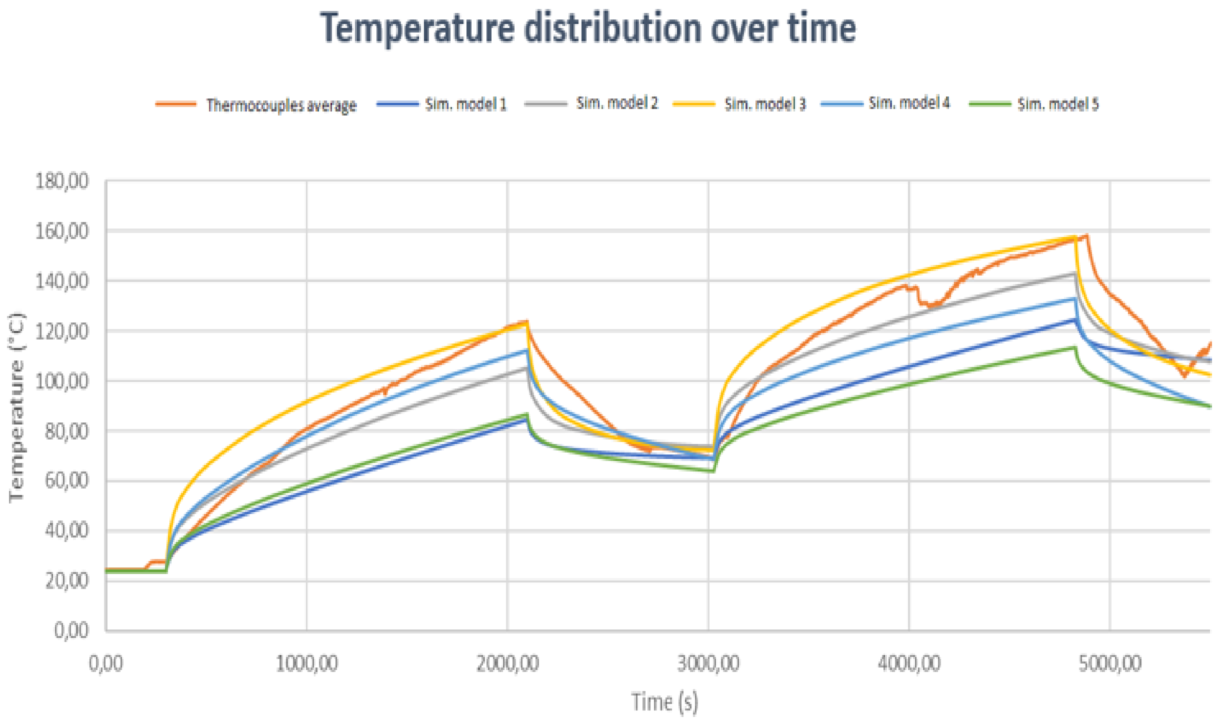

Figure 6 - Comparison between simulation models and thermocouples average in tests round 1.

With the second round of tests completed, in its first test, a greater influence of friction was observed in the first test of the second round. A high and rapid increase in temperature was observed 100% higher than the normal values observed in the first round of tests. Several hypotheses explained this problem. On explanation could be some troubles in the alignment of the sensor in relation to the wheel in all the tests of the second round, especially in the upper support. Always a thermocouple at some point or during the entire test had its contact pressure slightly altered due to vibration and oscillation of the test bench and consequently this sensor overheated due to friction and the measurement was lost.

The sudden effect of friction on the temperature rise can be observed at the beginning of tests, where small movements adjusting the supports caused the pressure of one of the sensors momentarily against the wheel and peaks were observed in the temperature value acquired so far. Such peaks are also present in some other moments, where during the cooling the supports were adjusted in an attempt to reduce the friction effect and acquire the measurement in all the sensors. However, sometime after the application of the braking load, the problem returned to persist and it was found impractical to obtain temperature data from all sensors simultaneously in the experiment.

Although the difference caused by friction in these cases of excessive pressure is remarkable, it cannot be neglected. As a whole in the evaluation of the observed results, because as we can see at the beginning of the tests, Figure 7 shows there is an increase in temperature in Temperature_1 Test 2 sensor and this increase due to the start of wheel rolling before the brake load is applied. At this point, we notice an increase of 10 °C to 15 °C depending on the sensor, relative to the difference of 30% to 40% of the static measurement.

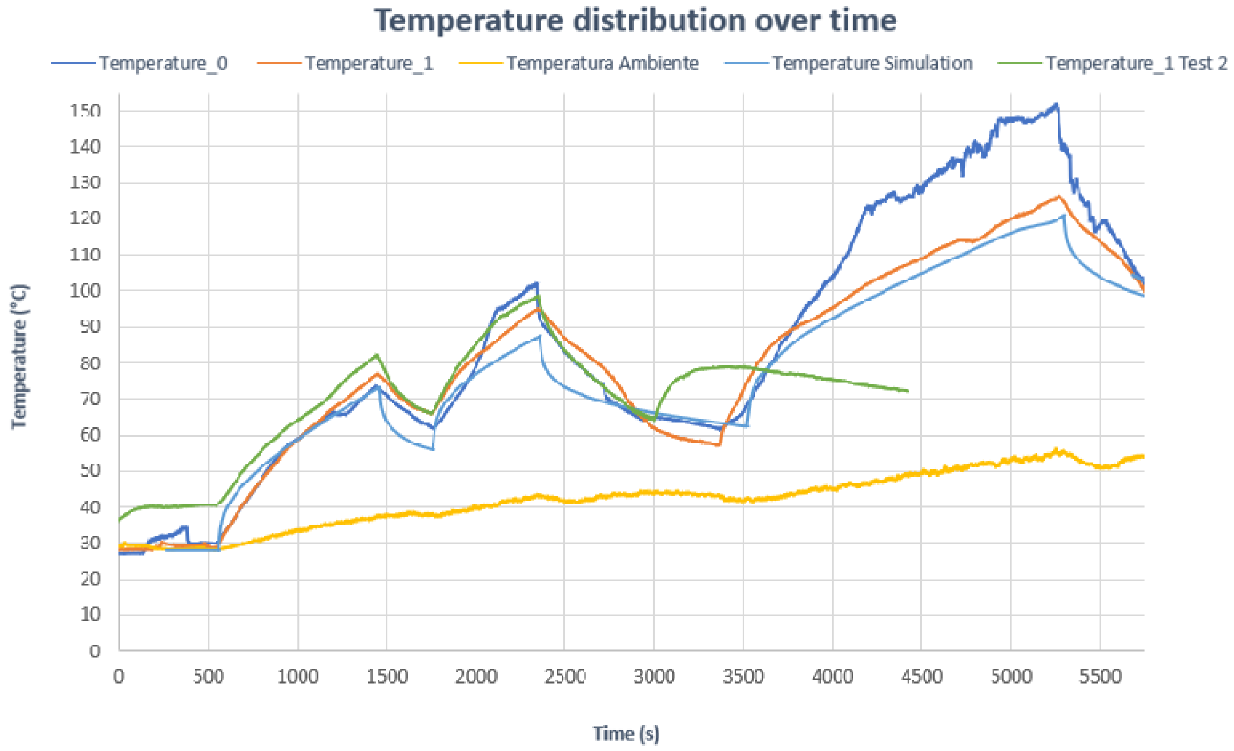

Figure 7 - Comparison between simulation and experimental data in test 5 + 6 in round 2.

Were observed after the second round, slight thermal variation at the beginning and end of braking phase when compared with the simulation data regardless of the positioning and support used. This phenomenon can then be explained by the thermal inertia occurring in thermocouples in sliding contact measurements, as observed by NOSKO et al., 2021.

Despite this, we can compare the results obtained with the proposed thermal model considering the effects caused by friction and thermal inertia in thermocouples. In Figure 7, we can see a better match in the cooling data over time using the rate of power converted into heat by the braking phenomenon as 70% and the connective coefficient of h = 30 W/m^2°C.

In addition, in this test, the ambient temperature was measured to adjust the reference temperature in the convective heat exchange over time, but this parameter did not show great influence on the observed result.

Even in this final comparison there are discrepancies due to friction and thermal inertia, however, considering the data presented, it can be said that the model similarly describes the heating phenomenon due to braking observed in the test.

5. CONCLUSION

The braking effect comprises several external factors and variables. A finite element model was developed to determine the temperature on the wheel caused by the braking phenomenon.

In addition, it was noticed that due to failures in the helm, this temperature can reach high values, in order to significantly reduce the mechanical resistance, and consequently the useful life of the part, and induce permanent defects or cause a catastrophic failure.

It is observed that the developed tool has robustness due to the mesh independence study carried out and it was also possible to find a mesh model that describes the model's response well at a relatively low computational cost when compared to the braking and cooling time simulated in the model.

Analysing the results, after the initial rise of temperature there are observed discrepancies lower than 10% in some tests considering all the variation due to griding thermocouples. This indicates that for long periods of brake application the model is capable of represent the thermal load on the wheels that is the case of long stretches of decent in mountains. Also, the model is more adequate to cargo composition as their brake pattern matches where the model has it’s best fit.

We can notice that with the preliminary results obtained, we already see that the model is capable of capturing the phenomenon of heating caused by braking and that with improvement. The proposed model will be able to estimate with high precision the temperatures and investigate from the voltage results generated by the thermomechanical model possible sources of failure due to overheating during braking on national railways.

Thus, the development of this work will assist in the analysis and diagnosis of failures in order to capture their source according to the traces observed after the failure and also in the validation of alarms and procedures currently adopted in the industry.

6. FUTURE WORK

This work is part of the Unicamp railway research group and is part of a master's work that will be developed by the author, thus having its continuity in the validation of the tool including a infrared temperature sensor in a third round of tests and future analyses, such as the inclusion of the cooling effect caused by the rails and also a fatigue life analysis and an analysis of the safety and reliability parameters adopted by the industry for the solution of real problems, such as a stuck brake.

7. ACKNOWLEDGMENTS

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