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Design, Analysis &Optimization of Disk Brake

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ABSTRACT

The main goals of this paper which presents the results of a thermal analysis of a braking system of railway vehicles using analytical and numerical modeling of thermal effects during long-term braking for maintaining a constant speed on a down-grade railroad in order to analyze damages of solid wheels braked by blocks, especially on railway .Preliminary works in this area have pointed out the problem of insufficient accuracy of the estimation coefficient. That was specifically expressed in conditions of Many research results have confirmed the dominant influence of thermal loads in regard to mechanical loads and residual stresses induced by high thermal loads in block-braked solid wheel have been registered. Therefore, it is important to determine with high precision the temperature field of the braking system, as well as to emphasize that high thermal loads, in other words, overloads, of wheel very often occur as a result of long-term braking on down-grade railroads or unwanted locking of wheels.

***Keywords*— thermal effects, mechanical loads, residual stresses etc.**

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I . INTRODUCTION

Thermal analysis is involved in almost every kind of physical processes and it can be the limiting factor for many processes. Therefore, its study is of vital importance, and the need for powerful thermal analysis tools is virtually universal. Furthermore, thermal effects often appear together with, or as a result of, other physical phenomena. The modeling of thermal effects has become increasingly important in product de-sign including areas such as electronics, automotive, aerospace, railway (*e. g*. wheel and rail rolling contact, braking systems, and so on), medical industries, *etc*. Computer simulation has allowed engineers and researchers to optimize process efficiency and explore new designs, while at the same time reduce costly experimental trials. The finite element method (FEM) has become the preferred method in performing thermal analysis on many systems and processes in recent years .

With the advances in computer technology and computer aided design (CAD) systems, complex problems can be modeled relatively simply. A FEA thermal analysis is a finite element analysis that looks at how heat affects certain materials and engineering designs. This heat can come in the form of an environmental load such as an ambient temperature of a certain degree affecting a model, or due to friction in a system, effectively converting it into thermal energy. It can also come from processes of conduction through two solids, convection between a liquid and a solid, or radiation such as in space. A thermal analysis is a great way to test a model before the model is built and real world tested in a thermal chamber. It can reduce the time to test a design by weeks, allowing for several redesigns and improvements to be made in the meantime.

Precise prediction of the maximum temperature is needed for the design of many systems, as well as braking systems, especially for both discs and linings. How to handle the high speed spinning of discs is the point of the heat/structure coupled analysis . Transient thermal analysis determine temperatures and other thermal quantities that vary over time. The variation of temperature distribution over time is of interest in many applications such as cooling of electronic packages or quenching analysis for heat treatment. Also of interest are the temperature distribution results in thermal stresses that can cause failure. In such cases, the temperatures from a transient thermal analysis are used as inputs to a structural analysis for thermal stress evaluations. Heat generation controlling is a prerequisite for qualitative weld creation during the friction stir welding process, and it is important to have an adequate mathematical model that is capable of estimating heat generation with satisfying accuracy .

Thermal analysis is the primary stage in the study of braking systems, because the temperature determines the thermo mechanical behavior of the structure. In the braking phase, kinetic energy transforms into thermal energy, resulting in intense heating of the railway wheel. This generates stresses and deformations, whose consequences are manifested by the appearance and the accentuation of cracks on treads of wheel, and eventually fractures of the whole wheel .

II.DESCRIPTION OF THE RAILWAY BRAKING SYSTEM

The disc-brake system is composed of four discs on each wheels axle and sliding bodies that are constituted of two symmetric lining plates with cylindrical pads (18 pad for each side), as illustrated in Figures 1. The brakes are activated by the pneumatic system pressure and slow down rotation of the wheels by the friction caused by pressing brake pads against brake discs.

Fig. 1. A Railway Braking System

Fig. 2 Retrofit of the Brake Shoe with the Brake Disc System.

In Such brake disc system, the wear of wheel tire can be largely reduced comparing to the original brake system which can cause the splitting off the wheel tire from the wheel. With this technology, the railway performance can be improved in term of safety. For these reasons, the objective of this research is to develop and retrofit the basic development of railway brake system with brake disc system using mathematical method. In such method, the brake disc temperature can be investigated based on the conditions in which the brake application and cooling period are considered.

III.THEORETICAL AND NUMERICAL APPROACH IN DETERMINING THE THERMAL AND STRESS LOADS IN TRAIN DISC BRAKES

The most important part of the vehicle is the braking system. Today the railway vehicles use disc brakes. Braking from 200 km/h to a standstill requires good, reliable brakes, capable of fast response times and durability. Different loads are applied to the disc during braking. Centrifugal, thermo-elastic, friction and brake clamp loads affect the brake disc at the same time. The main problem of braking and stopping a heavy train system is the great input of heat flux into the disc in a very short time. Because of high temperature difference the material is exposed to high tensions. Heat shock, rapid aging and fatigues are the results. To prevent this from happening, analysis need to be made. The problem can be solved only by applying a non stationary and numerical calculation. The analysis is carried out for one model of the disc (Figure 2) and for one large load. [1]

With correct design and correct choice, two major goals have to be assured:

1. safe braking and
2. reliability of brakes during all working regimes

Wrong design and selection of the production procedure and also environmental conditions can lead to catastrophic results. An overview of past researches showed, that brake discs are mostly tested for thermal loads and their effects (thermal cracks, thermal deformation). Besides that, there is great stress on brake pads. The main goal of this model and analysis was to determine the effect of thermal loads on the temperature field and the effect of centrifugal load in a specific brake disc under certain circumstances. To determine the boundary conditions, parts of car brake disc calculation were used. This assumption can be applied because the working physical principal is the same, although the weight distribution ratio is not the same as in cars.

The purpose of this analysis was to define a model for the thermal and centrifugal load. With this model all the necessary parameters (stresses as a consequence of thermal and centrifugal loads), defined by the maker can be calculated.

IV.NUMERICAL APPROACH - CALCULATION OF THE DISC BRAKE MODEL

In the beginning of the analysis the brake disc is simplified. Only one section of the whole disc is used. The disc brake is symmetrical; therefore the model includes 1/12th of the whole disc brake. Only one type of disc was considered.

This was the disc with permitted wearing of 7 mm on both sides. The load, used for this analysis, consists of:

1. input of the heat flux when braking from the top
2. speed to a standstill,
3. a period of 60 s for cooling off,
4. acceleration back to top speed,
5. braking to a standstill.

All the boundary conditions are calculated for braking on a flat track. The material of the disc brake is spherical graphite, defined according to SIST EN 1563:1988 and with the characteristics according to EN-GJS-500-7 (EN-JS 1050) with surface roughness of Ra = 3.2 µm. Disc brakes were machined and prepared on CNC machine tool with cutting conditions which were previously optimized by an intelligent optimization software. Surface roughness and cutting forces acting on the disc during machining were kept constant by continuous adaptation of cutting parameters to current machining conditions.
The disc brake was screwed to the hub. The hub was press fit onto the axle of the vehicle. Only one braking cycle was taken into consideration.

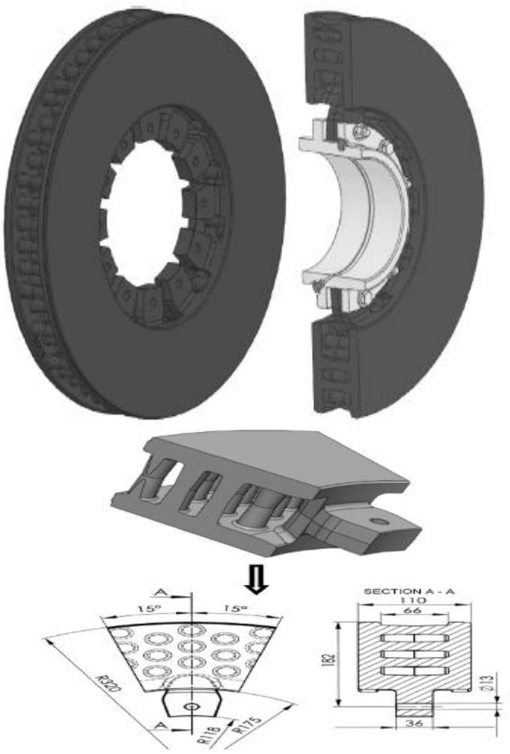

Fig.3. Brake Disc Parts

V. LOAD DETERMINATION

The disc was analyzed for one load case, consisting of the following steps:

1. braking from the top speed of 200 km/h to a
2. standstill, standing still for 60 s
3. accelerating back to top speed,
4. Once again braking to a standstill.

At the beginning of the test cycle, the brake disc has the same temperature as the surrounding environment, which is 20 °C. Because the train never stops on an inclined track, it has to stop on a horizontal track. Braking to standstill on a horizontal track is the most challenging load case. Because of the speed reduction, the influence of the wind is smaller and therefore a smaller heat transfer coefficient (10 W/ (m2 K)) was considered.In the analysis the disc was checked for temperature and stress arrangement. During the whole braking cycle, the prescribed temperature of 350 °C was not exceeded. This temperature was not reached even with subsequent braking to a standstill. The air humidity and the influence of the heat radiation were not considered in the calculation. [1]

Table 1. Material Properties

Mass of the vehicle, kg	70 000
Maximal load per axle, kg	17500
Number of axles per vehicle	4
Number of discs per axle	2
Start speed $v0$, m/s	56
Deceleration a, m/s^2	1.4
Braking time ts, s	34
Effective radius of the disc brake $r$disc, m	0.247
Radius of the wheel $r$wheel, m	0.460
Friction coefficient disc μ	0.4

In consideration of weight distribution and the fact that the bogie consists of two axles, one brake disc from the bogie carries 12,5 % of the whole braking force (Figure 4).

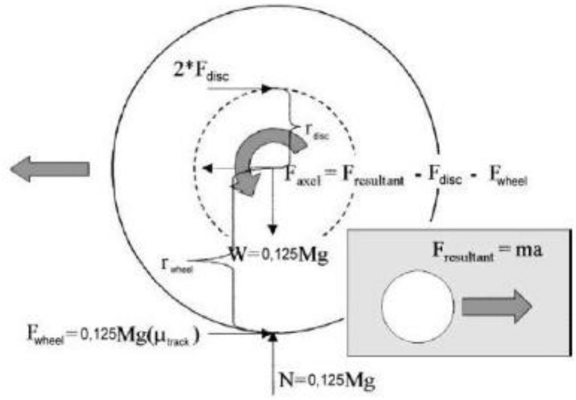

Fig. 4.Forces on Disc

DETERMINATION OF LOADS, FIXING, MESH AND MATERIAL PROPERTIES.

LOADS**:** the symmetrical boundary condition, which is located on the edge of the selected section, was modelled with slide supports in radial direction. The heat flux did not flow thru those supports. But id did flow thru both sides of the disc.

FIXING**:** fixing was made in the points, where the disc was mounted (screwed) onto the hub

MESH**:** the creation of a mesh volume was conducted automatically by the software package Abaqus CAE 6.7.1. The mesh consists of 86713 tetrahedral elements (element code C3D4AT – allows linear thermo – deformational analysis). The average size of the elements is 6mm and the number of nodes is 18135.

MATERIAL**:** for the analysis of the disc, certain physical properties of materials, given in Table 2, were required.

Table 2.Material Properties

Heat conductivity λ, W/mK	35.2
Density ρ, kg/m3	7100
Specific heat cp, J/kgK	515
Module of elasticity E, MPa	169000
Poisson number ν	0.275

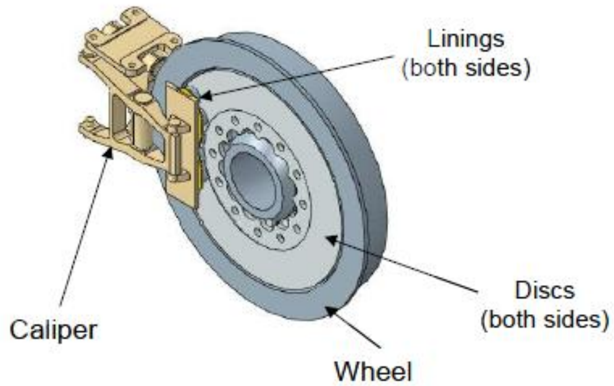

Fig.5.Components of Disc Brakes

VI. THERMAL ANALYSIS METHOD USING ABAQUS/EXPLICIT

From the above-mentioned studies, the short time increments caused by high rotation speed of discs made analysis impossible in practical computational time. Abaqus/Explicit, which requires very short computational time for each increment, might be able to solve this problem. Ordinary fully coupled heat structure analysis was tried but the stable time increment for the explicit integration was unacceptably small. So we assumed that the contact conditions were not changing during the braking event because of the wear on the linings and that all parts were made rigid bodies for the deformation degrees of freedom. Because all elements had only temperature degrees of freedom, the stable time increment increased by two digits and the problem could be analyzed in practical computation time.

HEAT TRANSFER ON THE SURFACES

Conductive heat transfer between the parts was considered using thermal contact properties. Abaqus/Explicit can handle heat generated due to frictional sliding but in this problem we apply heat flux directly instead of using this functionality because the penetrations were excessive between the rigid surfaces of the discs and the linings when that functionality was used.

MEASUREMENT METHOD

Nabtesco has brake testing facilities that can reproduce up to 480km/h train speeds (See Figure. 7.) The emergency brake patterns from various initial speeds were tested using these facilities. Temperatures of discs and linings were measured by thermocouples and surface temperature contours were taken by thermal video. Measurements were repeated twice for each initial speed condition.

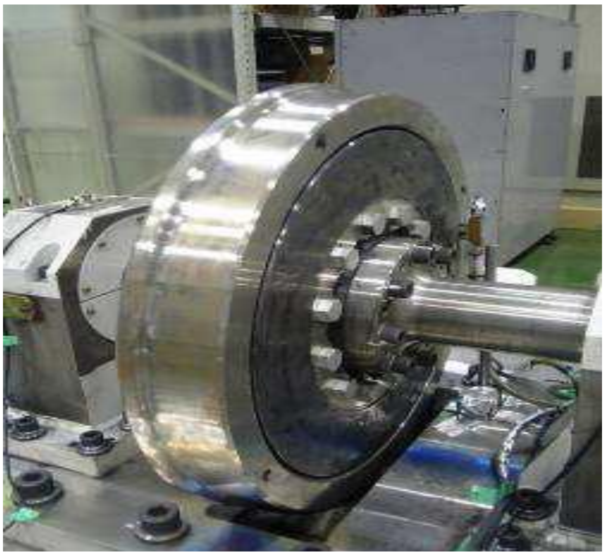

Fig.6.Exprimentation Setup

VII. RESULTS AND DISCUSSION

TEMPERATURE ON SURFACES

Temperature contours of the measurements and the analysis results are shown in Figure 9. In both figures, the temperature distributions were concentric and increased with time. The temperature distributions of analysis were in good agreement with the measurements besides the low temperature regions, where the measured values of thermal video were influenced by the reflection from high temperature regions.

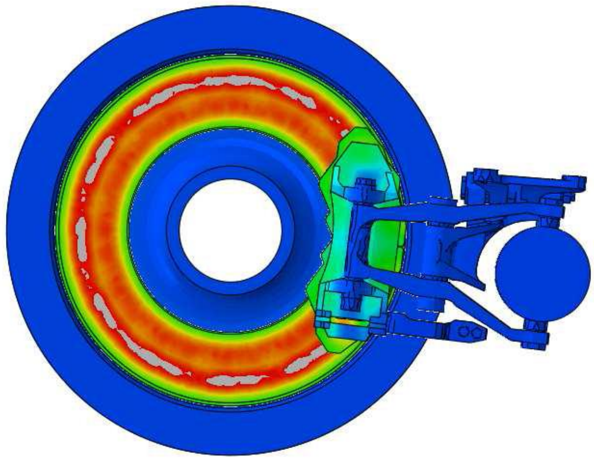

Fig. 7. Temperature contours of experiment and analysis

COMPARISON OF TEMPERATURE BETWEEN ANALYSIS AND MEASUREMENTS

The results of the analyses and the measurements are shown in Table 1. As the largest error was 10%, the analysis results were in good agreement with the measurements. Time series plots of the analyses and measurement results are shown in Figures:

• Discs (Figures 7,8) While Temperatures of Disk-1 and Disk-2 are almost identical in the analyses, the measurements temperatures of Disk-1, which is on inside, are higher by 50K than the temperatures of Disk-2, which is on outside. This difference seemed to come from the measurements environment such as airflow around the wheel.

• Linings (Figures 8,9)

The plots of analyses results are smooth but there are some fluctuations in the measurements of
the linings. These fluctuations are possibly caused by an imbalance in braking force, wears on
the linings and thermal deformation of the discs.

• Overall results

Analysis results of overall time series change and maximum temperature agreed with the
measurements results, besides the analysis could not simulate the fluctuations mentioned above.

Simulation of thermal stresses has been performed with the sequential approach. The results how that during hard braking, high compressive stresses are generated on the disc surface in circumferential direction which cause plastic yielding. But when the disc cools down, the compressive stresses transform to tensile stresses. Such results for a single braking operation have been presented where the plasticity model is taken to be the von Misses yield criterion with nonlinear isotropic hardening, and both the hardening and the yield limit are temperature dependent. [6]

For repeated braking it is important to use the kinematic hardening model as the isotropic hardening model cannot represent the Bauschinger effect. It has been shown in that in grey cast iron, for a cyclic loading resulting in plastic deformation in both tension and compression, the kinematic hardening model gives a somewhat better agreement with experimental data than isotropic hardening. Results of an analysis for repeated braking are presented, where the plasticity model is taken to be the von Mises yield criterion with linear kinematic hardening and both the hardening and the yield limit are temperature dependent. Figure shows the temperature distribution on the disc surface after a brake application during this analysis. A ring of high temperatures, called hot band, can be distinguished in the middle of the disc surface. Figure 15 shows a ring in the middle of disc surface, at the end of brake application, with relatively higher compressive circumferential stresses which roughly corresponds to the ring of high temperatures. The disc is cooled after this braking operation, completing one brake cycle. It is assumed that braking conditions are same for all the brake cycles so they generate similar temperature history. Hence the temperature history generated during one brake cycle is merged three times in a sequence It can be seen that residual tensile stresses in circumferential direction are predicted with both hardening models but with the kinematic hardening model these stresses are lower in magnitude as compared to the isotropic hardening model. [6]

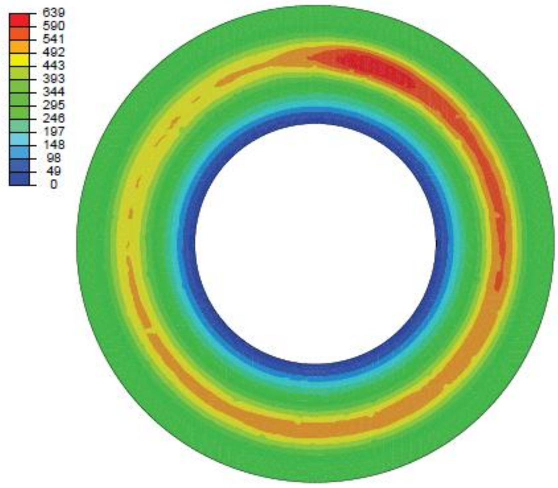

Fig .8.After brake application a ring of high temp.develop on disc surface.

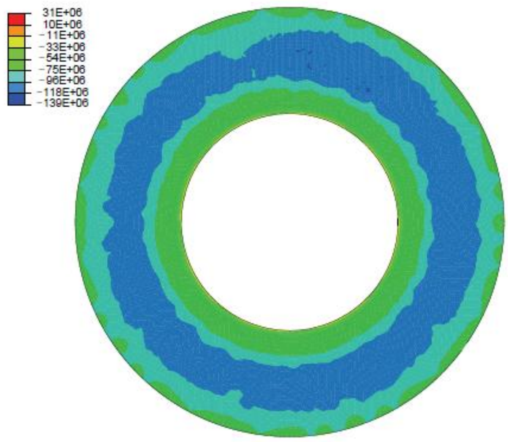

Fig.9. Circumferential stresses at the end of brake application during first cycle with the linear kinematic hardening model

After the first cycle an approximately stable stress-strain loop is obtained for the linear kinematic hardening model. So if the fatigue life data for the disc material is known, its fatigue life can be assessed. [6]

Furthermore results also show the appearance of tensile stresses in radial direction during braking and cooling of the disc. But the residual radial stresses are compressive as compared to the residual circumferential stresses which are tensile. This is indeed in agreement with the observation that radial microcracks on disc surfaces are more marked than circumferential ones, even when macroscopic cracks do not appear. Figure 17 shows a ring in the middle of the disc surface, at the end of three brake cycles, where effective plastic strain is relatively higher. So the material in this area is most susceptible to fatigue cracks. The simulation results presented in the first two papers predict one hot band in the middle of the disc. It has been explained by showing the contact pressure plots at different time steps. Convex bending of the pad due to thermal deformations is the major cause of contact pressure concentration and hence appearance of hot bands. In the first two papers wear of the pad is not considered as it does not show much influence on the temperature distribution during a single braking operation for a pad without wear history and hence on the stresses. The results show that when wear is considered, different distributions of temperature on the disc surface are obtained for each new brake cycle. After a few braking cycles two hot bands appear on the disc surface instead of only one, which is in agreement with experimental observation. This sequential approach has proved tremendously cheap in terms of computational time when compared to a fully coupled Lagrangian approach. Significantly lower computational resources required to simulate a disc brake by using the sequential approach gives the freedom to perform multi-objective optimization studies.The mass of the back plate, the brake energy and the maximum temperature generated on the disc surface during hard braking are optimized. The design variables are the applied load of braking, Young's modulus of friction material and the thickness of back plate. The results indicate that a brake pad with lowest possible stiffness will result in an optimized solution with regards to all three objectives. The results also reveal a linear relation of applied braking load and brake energy. Another interesting result is the trend of a decrease in maximum temperature with an increase in back plate thickness.

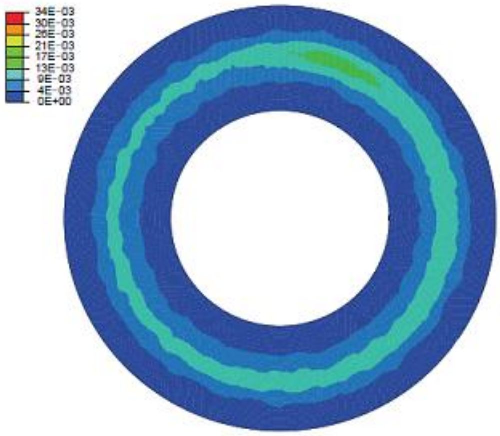

Fig.10 Effective plastic strain at the end of third brake cycle kinematic hardening model.

VIII. CONCLUSION

The disc heats up to high temperatures. The highest temperature is 175 °C. Considering the material, the design of the brake disc and the outcome of the results, we can conclude, that the model of the disc is adequate. Also the set target of the analysis meets the makers standard. The analysis should also be run with different boundary conditions to see, how a higher starting temperature of the brake disc affects the end results. In addition, other boundary conditions should be taken into account as shear stresses, residual stresses and the effect of service life with cyclic load, which the disc material must daily support without failing or breaking down. In this analysis, those boundary conditions were not included.

Fully coupled heat structure analysis of Abaqus/Explicit was tried to analyze the temperature of high speed rotating brake discs but the time increments turned out to be too small to analyze them in practical computational time. The rigid body constraint for the deformation degrees of freedom was introduced to solve this problem. The whole procedures of emergency braking were analyzed in practical computational time while maintaining the error of max temperature within 10%.

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