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Research Article

Thermo-Mechanical Simulation of Brake Disc Frictional Character by Moment of Inertia

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Abstract: The distribution of temperatures gradient and thermal stress of brake disc has been simulated by FEM code to make brake disc thermal stress more homogenously. In this study, using moment of inertia to simulate the realistic brake process instead of theoretically predefines the train deceleration rate, nonlinear deceleration rate and thermo-mechanical behavior has been revealed. The FEM models build upon LS-DYNA® thermo-mechanical code and contact algorithm. Non-uniform temperature alone disc radial direction was caused by severe friction in short time and the low heat transfer coefficient of its material. Parametric analysis for disc brakes have been carried out by comparison of grouped brake applications conform to UIC code, the main factor cause the high temperature gradient and thermal stress of brake disc is brake force and its initial speed.

Keywords: Brake disc, moment of inertia, thermo-mechanical simulation

INTRODUCTION

Disc brake in brief: For high speed train, the braking technology get fast pace worldwide especially in China, even the main brake method is electricity or eddy current brake but frictional brake is a redundancy backup and reliable way to stop the train in emergency. Most form of frictional bakes on the train is disc brake type mounted on the train axels to dissipate brake energy. Disc brakes are widely used for it simple, powerful and easy maintenance and allow new materials of disc and pad to be brought into use to provide effective friction and minimize the wear.

The premature wear and thermal cracking of brake discs are attributed to high thermal stress, as the speed of train getting faster and during brake process, more heat need to be distribute to brake disc and pad friction pair, which cause higher thermal stress.

In China, the peak speed of express train has reached 380 km/h and has world longest high speed rail. The higher the train speed, the higher kinetic energy the train possesses during transportation, thus need scatter more heat to disc brake system. This study will use a innovated way to simulate the brake disc thermal stress distributions and the main factors cause the high thermal stress.

LITERATURE REVIEW

Comprehensive engineering researches focus on disc thermal and stress filed, thermo-mechanic coupling of disc and pad pair have been performed by Petinrin and Oji (2012), the "Harmony" express have special requirements on the disc brake system and its principal, components and test procedure are discussed by Xiao-Hui *et al*. (2011), also the main character of the brake system and its kinetic energy behavior of express train have been presented by Li *et al*. (2011a). By means of FEM simulate of the carriage brake disc thermal stress, the disc temperature variation and thermal stress distribution were examined, also by compare the effect of the heat conductance, heat capacity and liner expansion factor, suggestions for material and shape of disc were given by Zheng *et al*. (2002). The 380 km/h express, maximum thermal stress can reach 450 MPa and the cast iron disc proved to be suitable for this severe condition simulated by FEM (Li *et al*., 2011b). A new kind of sintered pad proved can be used for the 350 km/h and above brake application test (Li *et al*., 2011c). The arrangement of the pad alone the radial direction of brake disc will affect the heat flux on the disc, the way to optimize the pad arrangement was introduced to reduce the thermal stress by Nong *et al*. (2012).

Even the brake discs have been extensively studied and comprehensive reviews can be founded, the kinetic energy specified by moment of inertia instead of predefine deceleration rate is never performed.

THEORETICAL APPROACHES

The study will use coupled thermo-mechanic code to simulate the frictional disc brakes and to establish the finite element models of the disc and pad pair of the train; the kinetic energy specified by moment of inertia; the LS-DYNA® coupled code is applied and the train brake cycle will be simulated by designate the factors

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such as contact parameter, friction coefficient, initial speed and inertia of the train, then examine the temperature gradient and thermal stress of the disc to find out the main factors affect the disc thermal stress.

The disc is made of 28CrMoV5 gray cast iron by casting process, it has stable friction character and good wear resistance property; the sintered pad has good and stable friction coefficient and the disc pad pair can dissipate the frictional heat effectively. Most current numerical approach on disc brakes preset a brake time or deceleration rate, in practice, the caliper press the pad onto the disc by pneumatic cylinders, the constant deceleration rate hardly be maintained. To reflect the realistic brake cycle, only the initial translational kinetic energy need specified and in turns the FEM code will perform and catch the energy transform process, till the train stops.

Kinetic energy and moment of inertia: When train begin to brake, the train lost power and will stop by frictional forces, the imposed kinetic energy theoretically transformed to frictional heat, the work of friction during a brake cycle as below (all units in SI):

$$Q = uF_n S \tag{1}$$

where,
Q = Brake work/energy
u = Friction coefficient
F_n = Brake normal force
S = Brake distance

Now focus on the brake disc, the liner translation must be replaced with rotational motion symbolic, assume rotate about Z axle:

$$E_{Rot} = \tau\theta = I\alpha\theta = I\alpha(\omega_0 t + \tfrac{1}{2}\alpha t^2) \tag{2}$$

where,
E = Kinetic energy
τ = Rotational torque
θ = Angular position
I = Moment of inertia
α = Angular acceleration
ω_0 = Initial angular speed

Energy transform: The kinetic energy can be predefined and according to UIC 541-3, maximum heat dissipation on one friction pair is limited under 18.8 MJ (UIC Code 541-3, 2006), so the initial kinetic energy can be given as following:

$$E_{Rot} = \frac{1}{2} I\omega_0^{\ 2} = \frac{1}{2}\sum_{i=1}^{N} m_i r_i^2 \omega_0^{\ 2} \tag{3}$$

where,
I = Moment of inertia
ω_0 = Initial angular speed

The contact algorithm of LS-DYNA described in its manual of 26.8.4, shows that contact energy for the contact interface is incrementally updated as:

$$E_c^{n+1} = E_c^n + \left[\sum_{i=1}^{nsn} \Delta F_i^s \times \Delta d_i^s + \sum_{i=1}^{nmn} \Delta F_i^m \times \Delta d_i^m\right]^{n+\frac{1}{2}} \tag{4}$$

The contact energy is result of all the slave nodes and master nodes interface force F and incremental distance d, when friction presence the interface energy mainly represents the sliding energy (LS-DYNA, 2006). Frictional heat will be conducted in the 3D spatial of the contact interface, assumes the thermal conductivity k and the specific heat capacity Cp as temperature independent to simplification. Nodal and elements temperature will be updated and exactly represented (Kolleck *et al*., 2008):

$$\frac{\rho Cp}{k}\frac{\partial T}{\partial t} = \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \tag{5}$$

where,
ρ = Density
Cp = Heat capacity
K = Heat conductivity
T = Temperature

FINITE ELEMENT MODELS

The model geometry and properties: The model incorporate of the thermal and mechanical behavior between the brake disc and pad, LS-DYNA® proved to be competence to do the couple analysis. There are several forms of solid elements to choose, hexahedron (or bricks) element is effective and simple. The disc will be solid and rigid to reduce CPU cost and the frictional effect will be captured by an introduced surface mesh. This is a practice way in LS-DYNA® in metal forming and hot stamping approaches (Botticher, 2007). Thermal stress of the disc will then be calculated using the nodal temperature data sequentially. The model and its parameters are illustrated in Fig. 1 and Table 1.

Contact parameters: In Fig. 1, the disc (A) and pad (C) are solid hexahedron elements, the disc surface (B) and steel slice (D) is theoretical introduced as shell elements, which is CPU efficient. Contact of solid C and surface mesh B treated as “surface to surface contact” with friction coefficient as 0.4 (both static and

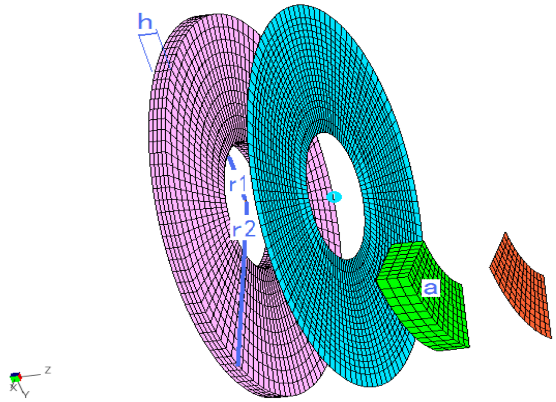


Fig. 1: FEM model arrangement left to right: A-disc B-surface mesh C-pad D-slice

Table 1: Model properties

Items	Units	Disc	Pad
Radius r1	m	0.3200	0.3100
Radius r2	m	0.1000	0.1900
Height h	m	0.0200	0.0400
Area a	m^2	0.2826	0.0396
Node #		5500	520
Element #		4000	324
Type		Bricks	Bricks
E (young's)	G Pa	207	50
Density	Kg/m^3	7830	5250
Poisson R.		0.2800	0.3000
Cp	J/gK	500	400
K	W/mK	50	400

Table 2: Initial parameters

Initial temperature	0°C
Normal force	22.5 kN
Initial angular speed	200 rad/s (320 km/h)
Moment of inertia Izz	600 (3000 kg)
Initial kinetic energy	12 MJ

dynamic friction), by setting FRCENG = 1 in the contact parameter, friction heat will be calculated and the heat will be distributed between the contact interface. Volume mesh A and surface mesh B will be rotated simultaneously without friction, so the friction coefficient is zero and also set FRCENG = 0 to ignore the friction heat (LS-DYNA Keyword User's Manual, 2011). The heat transfer between A and B are enabled by using "surface to surface thermal contact" and adopt a higher heat conductance; thermal contact can be described very accurate taking into account the contact heat transfer coefficient (Lorenz and Haufe, 2008).

During brake cycle, B has a relative rotation speed to C, normal forces on the pad transfer to the surface of disc result the reaction forces. If the sliding speed is high, the resulting thermo-mechanical feedback is unstable, leading to the development of non-uniform contact pressure (Altuzarra *et al*., 2002). So a rigid shell D (function as steel sleeve) has been introduced to keep the pad upper elements displacement the same (by boundary conditions), which will result uniform pressure on the disc and pad contact interface, the pressure and displacement of the pad contact nodes will be less different and avoid local high temperature.

Simulation set-ups:

Baseline model set-ups: The baseline model is chosen according to the normal conditions for a general brake cycle as the initial settings to study the behavior during the brake procedure and fine-turned with the experimental test. The refined parameters described as Table 2.

Moment of inertia is set by ELEMENT_INERTIA and attached to the disc.

Parametric comparison models: To identify the influence of parameters on the affect of thermo-mechanical disc thermal stress, the parameters considered to be normal forces, initial angular speed under the given moment of inertia and are selected conform to the UIC541-3 sintered pad test procedure 3A (assumes 890 mm diameter train wheels). Nine applications have been chosen for comparison which listed in Table 3; all initial temperature is 0°C (273 K)

Table 3: Models setup for comparison

Code	Units	1A	1B	1C	2A	2B	2C	3A	3B	3C
Normal force	kN	10	10	10	15	15	15	22.5	22.5	22.5
Angular speed	rad/s	100	150	200	100	150	200	100	150	200
Liner speed	km/h	160	240	320	160	240	320	160	240	320
Kinetic energy	MJ	4	4	4	9	9	9	16	16	16

Table 4: Brake time and average deceleration of 9 samples, parametric values are in italic

Case no.	Unit	1A	1B	1C	2A	2B	2C	3A	3B	3C
Train speed	km/h	160	160	160	240	240	240	320	320	320
Normal force	kN	10	15	22.5	10	15	22.5	10	15	22.5
Wheel line speed	m/s	44.5	44.5	44.5	66.8	66.8	66.8	89	89	89
Kinetic energy *	MJ	4	4	4	9	9	9	16	16	16
Brake time	s	40	21	15	85	47	30	200	95	55
Deceleration	r/s^2	1.1	2.1	3	0.8	1.4	2.2	0.5	0.9	1.5

*: Moment of inertia is 800 which equiv. to a mass of 4000 Kg

Table 5: Disc nodal temperature and thermal stresses of 9 samples

Case no.	Unit	1A	1B	1C	2A	2B	2C	3A	3B	3C
Peak disc nodal temp.	°C	111	122	130	231	237	260	429	439	484
Peak disc v-m stress	MPa	91	107	100	236	241	255	459	477	530

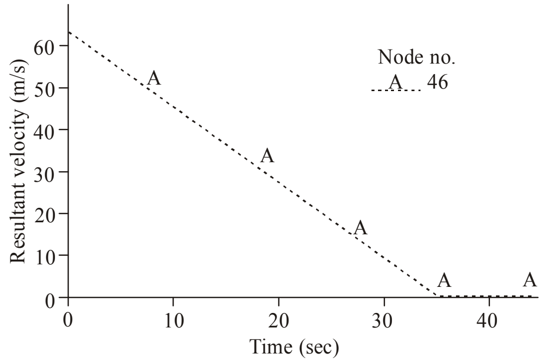


Fig. 2: Edge nodal line speed of disc vs. time

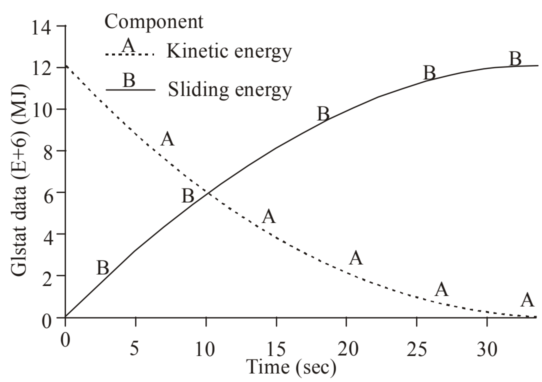


Fig. 3: Energy convert and balance during brake

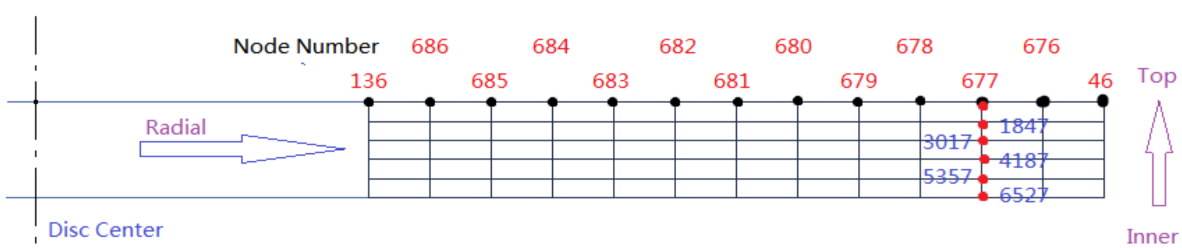
and moment of inertia Izz is 800 which have an equivalent mass of 4000 kg, the simulation time is till the train stops.

RESULTS AND EXPLANATIONS

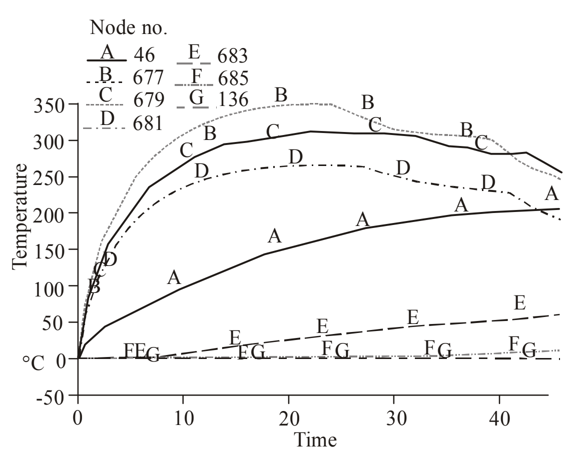
Baseline brake model: This sample model will dissipate 12 MJ kinetic energy by the brake and pad, time to stop the train was 35 sec, hot band appeared (Panier *et al*., 2001) and the resultant velocity of brake procedure is show in Fig. 2 with a nearly constant deceleration. During a brake cycle, kinetic energy transformed to heat to heat up the disc and pad, the material property and volume different cause the pad temperature higher than the disc temperature. In LS-DYNA kinetic energy is converted into sliding energy (mainly frictional energy) as Fig. 3 and detailed in its theory below.

For contact interactions, when a penetrating node is first detected, by checking the sign of the projected normal distance to the closest master segment, a penalty force is first calculated based on the stiffness and the absolute penetration value.

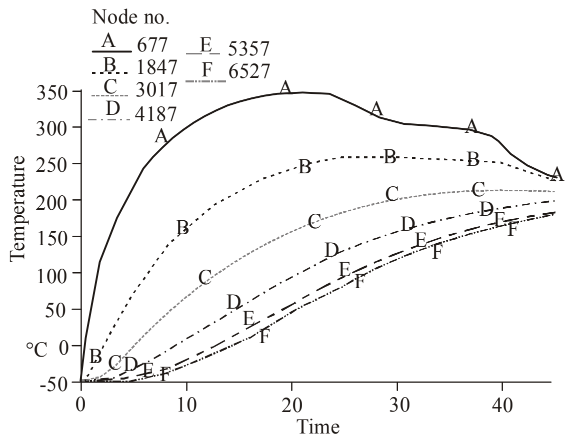
This force is then resolved in a local coordinate system embedded at the master element nodes to determine the normal and shear components. The sliding resistance is then computed using the friction parameters of the master segment and the normal force component.



(a) Disc node locations



(b) Disc radial nodes temperature vs. time



(c) Disc nodes surface to inner temperature vs. time

Fig. 4: Disc temperature gradient distributions

Consider the heat dissipation, the disc nodes contact with the pad will be directly heated and the untouched nodes were heated by heat conduct, along the radial direction, surface nodes of the disc temperature varies and showed in Fig. 4, the temperature gradient along the disc radial direction also increase against the time as Fig. 4b. And the inner nodes which are heat up and reduce temperature gradient as Fig. 4c.

The coefficient of thermal expansion allows the calculation of thermal strain and stress for any of the mechanical material models. In Fig. 5 the thermal stress is calculated based on the nodal temperatures in the D3PLOT file, it's so called Stress Initialization process in LS-DYNA. The disc centrifugal stress also calculated by implicit code, compare to the peak thermal stress up to 420 MPa in Fig. 5, the centrifugal stress is always under 30 MPa showed in Fig. 6 and can be neglected.

Parametric comparison models: The deceleration rate listed in Table 4 for the sample cases 1A-1B-1C, 2A-2B-2C and 3A-3B-3C, the deceleration is increased if the normal forced magnified prearranges the same initial angular speed and normal forces. The normal forces dramatically reduce the brake time and also cause more friction heat to dissipate. Take the maximum disc nodal temperature in Table 5 for study, normal force increase disc temperature and higher initial kinetic energy will cause higher temperature, which means more energy need to be transformed.

Temperature gradient causes the thermal stress and strain in the disc; the final temperatures from the previous simulation are used as initial conditions in the sequent transient thermal analysis. The simplicity of the thermal only approach considered to be accurate enough for a stress initialization analysis, the disc thermal stress is proportional to the disc temperature

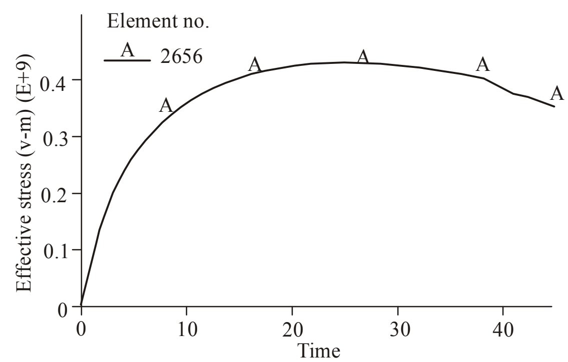


Fig. 5: Peak disc thermal stress during brake (GPa) element # 2656-containts node # 677

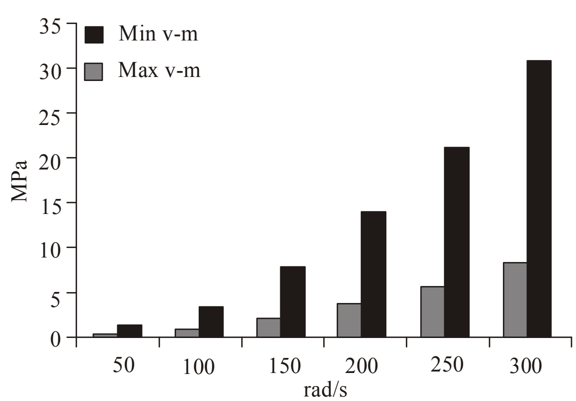


Fig. 6: Maximum and minimum disc centrifugal stress at given initial angular speed

gradient; Showed in Fig. 7 the sample 3C maximum stress can reach 530 MPa.

The higher train speed will transform more kinetic energy to heat and cause higher thermal stresses, the series 3 samples results extreme thermal stresses which will cause fatigue quickly and series 2 samples also

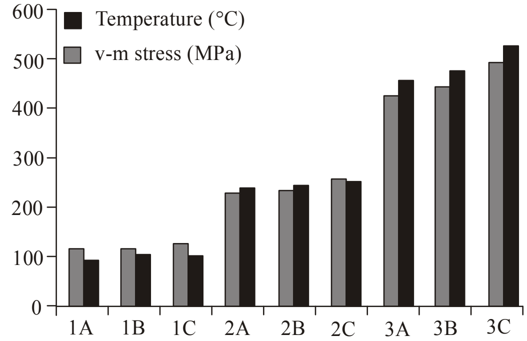


Fig. 7: Disc nodal temperature and stresses

have high stresses compare to series 1 samples at low train speed, so life time of disc is mainly affected by the train speed.

CONCLUSION

The numerical approaches by moment of inertia instead of predefine decelerate to indentify the factors influence in disc brakes are performed by a considerable amount of work in this study. Energy transform and heat dissipation process have been performed by thermo-mechanical coupled analytics code. The results reveal the non-uniform temperatures gradient alone disc radial direction and thermal stress caused due to temperature gradient of the disc. By comparison influences of the parameters of normal force and initial speed under given moment of inertia, initial speed is the dominate factor and the normal force dramatically increase the thermal stress and count to be second factor. Methods to reduce temperature gradient will be the future study.

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