TELKOMNIKA Indonesian Journal of Electrical Engineering
Vol.12, No.6, June 2014, pp. 4861 ~ 4867
DOI: 10.11591/telkomnika.v12i6.5846

Parameter Analysis of Fluid-Solid Coupling for Wet Brake Disc

Xunjia Zheng*, Tianhong Luo, Ce Jia
College of Mechantronics & Automobile Engineering, Chongqing Jiaotong University,
Chongqing, P.R.China
*Corresponding author, e-mail: xunjia_zheng@163.com

Abstract

Due to the poor conditions of wet brake cooling and the high temperature of the working characteristics, this paper intended to introduce the oil film vadose between the brake linings, considered the variation of volume of grooves on the friction plate, and then proposes the three-dimensional transient dynamic response of fluid-solid coupling mathematical model. By parameter analysis of the disc transient thermal stress field distribution on the friction plate, and according to the real dimension of the brake disc, the distribution of the temperature and the stress of the wet brake disc during the braking are analyzed, the variation of temperature field and stress field in friction plate are obtained. The data curves shown the temperature and the stress are coupled. Finally, the results shows that the mathematical model is effective and feasible.

Keywords: *wet brake, fluid-solid coupling, dynamic response, thermal stress field*

1. Introduction

Because of the characteristics of the strong ability to resist pollution, the large brake torque, wear-resistance and long lifespan and so on, the wet brake disc is widely used in the system of mechanical braking engineering [1-2]. However, during the working process of brake, the wet brake disc can bear extremely severe wear, rising temperature and thermal elastic deformation, resulting in deteriorating the stability of vehicle braking, impacting on the comfort of passengers and reducing its service life [3-5].

The fluid-structure interaction of brake is analyzed by the virtual simulation, which has been widely used. Xing Yutao made a research on the pressure field of the contact surface on the fully enclosed wet multiple disk brake, the transient temperature field of friction plate and the stress field [6]. Payvar P introduces the moving fluid and solid wall flowing boundary layer and thermal boundary layer theory into distribution of the numerical calculation of the square groove fluid velocity field and temperature field [7]. Zagrodzki R carried out the experiment and research on the distribution of temperature field and stress field of the friction pair on the wet multiple disk brake [8].

This paper established the fluid-solid coupling mathematical model of wet-type brake friction plate, through the example to analyze of the wet multiple disk brake of fluid-solid coupling dynamic response, aimed at improving the accuracy of the mathematical model of dynamic response of the braking system, and then delved into wet brake friction mechanism.

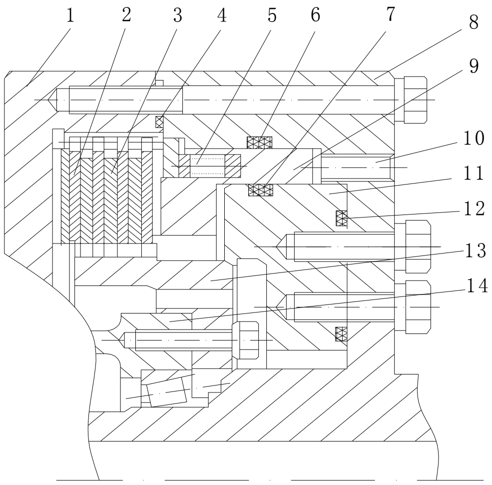
With the continuous development and progress of science and technology, the running speed of machinery and equipment is constantly improved, while the wet brake operation condition is getting worse and worse and the study of the theory of the wet brake friction mechanism is not clear. Therefore, it is imperative to conduct the theoretical research of the the wet brake fluid-structure interaction.

2. Structure and Systematic Principle of Wet Brake Disc

Wet multiple brake disk is mainly composed of brake shell, brake piston, friction plate, oil seal and end plate, etc. Several fixed and rotating braking friction plates are mutually staggered and installed in the brake casing filled with cooling the oil seal; while the fixed friction plates are collected to the brake shell by the spline, and the rotating friction slices rotating with

Received January 23, 2014; Revised March 19, 2014; Accepted April 3, 2014

the hub, are linked with the hub by the internal spline. When the oil liquid from the brake valve flows into the brake, the brake piston presses the braking friction plates installed crisscross, which makes the rotating friction plates moves slowly till stop, thus achieving the goal of the brake. The structure diagram of wet brake is shown in Figure 1.



1	annular gear
2	static friction plate
3	dynamic friction plate
4.6.7.12	obturating ring
5	return spring
8	cylinder body
9	piston
10	oil-in
11	terminal pad
13	gear ring
14	wheel hub

Figure 1. Structure Diagram of Wet Brake

Nomenclature

σ_{ij}	total stress of brakes	q_γ	brake fluid velocity
δ_{ij}	Kronecker symbol	h_1, h_2	friction linings and static friction piece sub-volume ratio of the total
p	excess pore pressure of oil		
ν	Poisson's ratio of oil	a_γ	pore fluid thermal expansion coefficient
a_c	coefficient of friction pair thermal expansion	K_γ	bulk modulus
K	bulk modulus of the brake vice	k_l	friction coefficient of permeability
G	Lame constant of friction pair.	g	acceleration due to gravity
b_j	force in spatial coordinate direction	D_T	temperature gradient effecting on seepage
β_i	displacement in spatial coordinate direction	V_{p2}	heat flow rate of fluid
γ_j	relative displacement in spatial coordinate direction	k_{p1}, k_{p2}, k_γ	friction linings, the static friction piece and brake fluid of heat transfer coefficient
ρ_1	density of the friction linings,	C_γ	specific heat capacity of the brake oil
ρ_2	density of the friction plate	T_0	initial temperature
ρ_γ	density of the fluid	T	temperature variation
∇	Hamiltonian operator	$\rho_{\gamma 0}$	initial brake fluid density
V	assuming friction pair volume	l_0	initial porosity of the brake friction
V_{p1}	volume of the friction lining	V_c	volume of the groove
V_{p2}	volume of the static friction piece	ε	represents the overall response
a_{s1}	thermal expansion coefficient of friction linings	l	friction porosity
a_{s2}	static friction piece coefficient	$\bar{\sigma}$	effective stress vector of thermal expansion

3. Dynamic Response of Fluid-Solid Coupling Mathematical Model

Because of existing a large number of grooves in wet brake discs, considering that the friction braking system exists in the two-phase medium as the form of fluid-structure interaction term, assuming that the friction pairs are of isotropy linear elastic material, when braking, two phase flow of the liquid and the solid are in a heat balance, and the brake linings suffer small deformation during braking, and its constitutive relation can be expressed as follow:

$$\begin{cases} \overline{\sigma}_{ij} = 2G\varepsilon_{ij} + \dfrac{vG}{0.5-v}\varepsilon_{kk}\delta_{ij} - Ka_c T\delta_{ij} \\ \overline{\sigma}_{ij} = \sigma_{ij} + p\delta_{ij} \\ \sigma_{ij,j} + b_j = \rho_1\ddot{\beta}_{1i} + \rho_2\ddot{\beta}_{2i} + \rho_r\ddot{\gamma}_i \\ 2\varepsilon_{ij} = \beta_{i,j} + \beta_{j,i} \end{cases} \tag{1}$$

The brake fluid-heat coupling equation of motion:

$$G\nabla^2\beta_i + \frac{G}{1-2v}\beta_{j,ji} - \alpha p_{,i} - Ka_c T_{,i} = \rho_1\ddot{\beta}_{1i} + \rho_2\ddot{\beta}_{2i} + \rho_\gamma\ddot{\gamma}_i \tag{2}$$

Assuming the volume of friction pair is V, including the volume of the friction lining V_{p1}, the static friction piece volume V_{p2} and the volume of the groove V_c, The relationship among the four can be expressed as:

$$\frac{1}{V}\frac{\partial V}{\partial t} = \frac{1}{V}\frac{\partial V_c}{\partial t} + \frac{1}{V}\frac{\partial V_{p1}}{\partial t} + \frac{1}{V}\frac{\partial V_{p2}}{\partial t} = \frac{\partial \varepsilon}{\partial t} \tag{3}$$

So the volume change of friction pair can be written in the follow form:

$$\frac{1}{V}\left(\frac{\partial V_{p1}}{\partial t} + \frac{\partial V_{p2}}{\partial t}\right) = (1-l)\left(h_1 a_{p1} + h_2 a_{p2}\right)\frac{\partial T}{\partial t} - (1-l)\left(\frac{l}{K_{p1}} - \frac{l}{K_{p2}}\right)\frac{\partial P}{\partial t} + \frac{\mathbf{m}}{3}\left(\frac{1}{K_{p1}} + \frac{1}{K_{p2}}\right)\frac{\partial \bar{\sigma}}{\partial t} \tag{4}$$

Where, $\mathbf{m} = [1 \ \ 1 \ \ 1 \ \ 0 \ \ 0 \ \ 0]$, According to the solid model $h_1 = 0.25$、$h_2 = 0.75$。Friction pair's volume will change accordingly when the temperature, oil pressure and effective stress of groove changed.

The variation of groove volume be expressed as:

$$\frac{1}{V}\frac{\partial V_c}{\partial t} = -\nabla q_\gamma + na_\gamma\frac{\partial T}{\partial t} - \frac{n}{K}\frac{\partial p}{\partial t} \tag{5}$$

Simultaneous Equation (3), (4) and (5), the flow continuity equation may be written in the following form:

$$\begin{aligned} \nabla q_\gamma = -\frac{\partial c}{\partial t} + \left[la_\gamma + (1-l)(h_1 a_{p1} + h_2 a_{p2})\right]\frac{\partial T}{\partial t} - \\ \left[\frac{l}{K_\gamma} - (1-l)\left(\frac{l}{K_{p1}} - \frac{l}{K_{p2}}\right)\right]\frac{\partial p}{\partial t} + \frac{\mathbf{m}}{3}\left(\frac{1}{K_{p1}} + \frac{1}{K_{p2}}\right)\frac{\partial \bar{\sigma}}{\partial t} \end{aligned} \tag{6}$$

The heat flux is expressed as a friction linear function of forces which act on the brake; the equation of fluid motion can be expressed as:

$$q_\gamma = -\frac{k_l}{\rho_\gamma g}\left(\nabla p + \frac{\rho_\gamma}{l}\ddot{w} + \rho_\gamma\ddot{\mu}\right) - D_T\nabla T \tag{7}$$

Brake fluid have a significant impact to the brake thermal conductivity, the heat flow rate can be defined as:

$$q_F = \rho_\gamma q_\gamma C_\gamma T - \left[h_1(1-l)k_{p1} + h_2(1-l)k_{p2} + lk_\gamma\right]k\nabla T - (T + T_0)D_T\nabla p \tag{8}$$

Assumed the solid-liquid fluid thermal equilibrium between friction and brake, the heat balance equation can be expressed as:

$$-\nabla q_F = \frac{\partial\left[\left(V_{p1}\rho_{p1}C_{p1} + V_{p1}\rho_{p1}C_{p1} + V_c\rho_\gamma C_\gamma\right)T\right]}{V\partial t} - (T_0+T)a_\gamma K_\gamma \nabla q_\gamma - (T_0+T)Ka_c\frac{\partial \varepsilon}{\partial t} \quad (9)$$

The friction pair and brake fluid are in mass conservation:

$$\begin{cases} \dfrac{\partial\left(V_{p1}\rho_{p1}\right)}{V\partial t} = 0 \\ \dfrac{\partial\left(V_{p2}\rho_{p2}\right)}{V\partial t} = 0 \\ \dfrac{\partial(V_c\rho_\gamma)}{V\partial t} = -\rho_\gamma \nabla q_\gamma \end{cases} \quad (10)$$

Simultaneous Equation (8), (10), the heat balance equation can be expressed as:

$$\begin{aligned} &\left[(1-l)(h_1\rho_{p1}C_{p1} + h_2\rho_{p2}C_{p1}) + l\rho_\gamma C_\gamma\right]\frac{\partial T}{\partial t} + (T_0+T)\left(Ka_c\frac{\partial\varepsilon}{\partial t} + a_\gamma K_\gamma \nabla q_\gamma\right) = \\ &-C_\gamma T q_\gamma \nabla\rho_\gamma - \rho_\gamma C_\gamma q_\gamma T + k\nabla^2 T + \nabla k \nabla T + \nabla(D_\gamma \nabla p) \end{aligned} \quad (11)$$

Brake fluid equation of state is obtained as follows:

$$\rho_\gamma = \rho_{\gamma 0}\exp(-a_\gamma T + p / K_\gamma) \quad (12)$$

$$\nabla\rho_\gamma \approx \rho_{\gamma 0}(-a_\gamma T + p / K_\gamma) \quad (13)$$

According to Equation (11), (12), (13), another form can be obtained:

$$\begin{aligned} &\left[(1-l)(h_1\rho_{p1}C_{p1} + h_2\rho_{p2}C_{p1}) + l\rho_\gamma C_\gamma\right]\frac{\partial T}{\partial t} + (T_0+T)\left(Ka_c\frac{\partial\varepsilon}{\partial t} + a_\gamma K_\gamma \nabla q_\gamma\right) = k\nabla^2 T \\ &-C_\gamma T q_\gamma \rho_{\gamma 0}(-a_\gamma T + p/K_\gamma) - \rho_\gamma C_\gamma q_\gamma T + (k_\gamma - h_1 k_{p1} - h_2 k_{p2})\nabla l \nabla T + \nabla(D_\gamma \nabla p) \end{aligned} \quad (14)$$

The changes of groove porosity can get according to Equation (3), (4):

$$\frac{\partial l}{\partial t} = \left(1 - \frac{K}{h_1K_{p1} + h_2K_{p2}}\right)\frac{\partial\varepsilon}{\partial t} + \left[\frac{Ka_c}{h_1K_{p1} + h_2K_{p2}} - (1-l)(h_1a_{p1} + h_2a_{p2})\right]\frac{\partial T}{\partial t} + \frac{1-l}{h_1K_{p1} + h_2K_{p2}}\frac{\partial p}{\partial t} \quad (15)$$

The Equation (15) can be simplified as:

$$l = l_0 + \left(1 - \frac{K}{h_1K_{p1} + h_2K_{p2}}\right)\varepsilon + \left[\frac{Ka_c}{h_1K_{p1} + h_2K_{p2}} - (1-l_0)(h_1a_{p1} + h_2a_{p2})\right]T + \frac{1-l_0}{h_1K_{p1} + h_2K_{p2}}p \quad (16)$$

The porosity gradient of friction pair can be expressed as:

$$\nabla l = \left(1 - \frac{K}{h_1K_{p1} + h_2K_{p2}}\right)\nabla\varepsilon + \left[\frac{Ka_c}{h_1K_{p1} + h_2K_{p2}} - (1-l_0)(h_1a_{p1} + h_2a_{p2})\right]\nabla T + \frac{1-l_0}{h_1K_{p1} + h_2K_{p2}}\nabla p \quad (17)$$

Simultaneous the Equation (14) and (17):

$$\left[(1-l)(h_1\rho_{p1}C_{p1}+h_2\rho_{p2}C_{p1})+l\rho_\gamma C_\gamma\right]\frac{\partial T}{\partial t}+(T_0+T)\left(Ka_c\frac{\partial e}{\partial t}+a_\gamma K_\gamma\nabla q_\gamma\right)=k\nabla^2 T$$
$$-C_\gamma Tq_\gamma\rho_{\gamma 0}(-a_\gamma T+p/K_\gamma)-\rho_\gamma C_\gamma q_\gamma T+\nabla\left[(T+T_0)D_T\nabla p\right]+(k_\gamma-h_1k_{p1}-h_2k_{p2})\nabla T\cdot \quad (18)$$
$$\left\{\left(1-\frac{K}{h_1K_{p1}+h_2K_{p2}}\right)\nabla\varepsilon+\left[\frac{Ka_c}{h_1K_{p1}+h_2K_{p2}}-(1-l_0)(h_1a_{p1}+h_2a_{p2})\right]\nabla T+\frac{1-l_0}{h_1K_{p1}+h_2K_{p2}}\nabla p\right\}$$

Overall, the Equation (18) is the wet brake friction pairs fluid-solid coupling equation.

4. The Numerical Analysis of Fluid-Structure Interaction

In order to test the accuracy of the establishment of wet heat flow fluid-solid coupling model on the brake friction pair, this section will make a comparative analysis of the data of the wet brake emergency braking measured by the numerical calculation and experiment of heat flow and solid coupling model.

Setting a certain wet brake actual structure as an example, the related parameters of calculation and analysis are shown in Table 1 to 3. The total braking time is 3.5s and the original braking speed is 50km/h

Table 1. Wet Brake Characteristic Parameters of each Component Materials

Parameter	$\rho/(kg\cdot m^{-3})$	E/GPa	ν	$\lambda/(W\cdot m^{-1}\cdot K^{-1})$	$c/J\cdot(kg\cdot K)^{-1}$	a/K^{-1}
Dynamic friction plate	7900	210	0.3	42	480	1.27e-5
Friction linings	1125	0.3	0.25	0.241	385.2	1.3e-5
Static friction tablets	7900	210	0.3	42	480	1.27e-5
Brake oil	857	—	—	0.126	9.85	3e-4

Note: the brake viscosity $\mu = 20\times10^{-6}(m^2\cdot s^{-1})$

Table 2. Geometrical Dimensions mm

Parts	Inside diameter	Major diameter	Thickness
Dynamic friction plate	32	155	3
Friction linings	115.3	155	1.56
Static friction tablets	111.4	156	2.5

Table 3. Calculation Parameters

Variable	G	K_γ	K_{p1}	K_{p2}	T_0	l	D_T
Parameter	1	4.1	0.2	175	293	0.253	1.7e-11
Unit	Mpa	Gpa	Gpa	Gpa	K	—	$m^2/s\cdot$℃

During braking, since heat generated is much higher than the speed of transfer to the internal, this makes the friction surface temperature is higher than the internal, and there is a large axial temperature gradient. With the passage of time, the heat impact on the temperature field is decreased with the input intensity decreases. Because the surface of the cooling conditions were better, the disc surface temperature decreased rapidly. Finally, the temperature of axial points gradually unified.

As it can be seen from the curves in Figure 2, dynamic friction piece outer has particularly tremendous changes in temperature fluctuations, due to the large linear velocity at external of friction disk, and the turbulence of brake oil has a relatively large impact on the temperature; The other hand, due to the inside friction disk small linear velocity, brake oil exists only within the narrow groove, brake temperature fluctuations is smaller. Finally, and the brake temperature is stable at about 134 ℃ when the brake time to reach 3.26s, and the brake disc temperature stabilized at between 122 ℃ and 134℃.

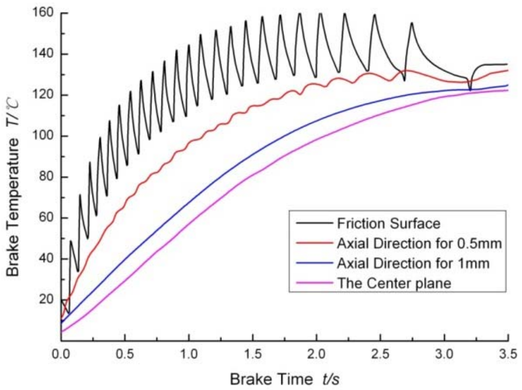


Figure 2. Different Point Temperature at r =114.5mm

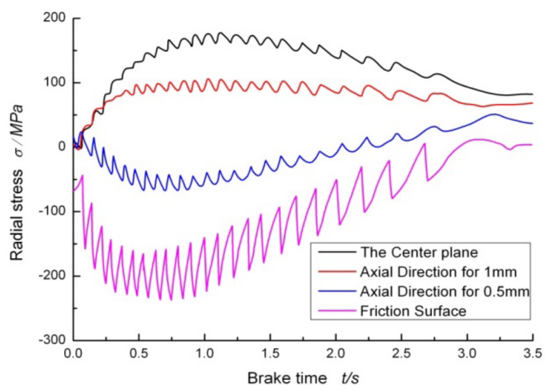


Figure 3. Different Point Radial Stress at r =114.5mm

From the Figure 4, another discovery is gained. In the contact area of the disc, due to the combination effect of the braking pressure, friction force and thermal stress, at the beginning of braking, with the temperature rising, the parts of the equivalent stress value and the maximum stress has reached or exceed the material yield limit stress. Although it does not reach the strength of the material limit, the partial plastic deformation will occur, further more, with braking increasing and the temperature decreasing, the corresponding circulation of stress adds and the fatigue crack damage accumulation of plastic deformation will lead to fatigue crack. However, if the braking strength is intensified in a single emergency brake, and the temperature gradient is big, which may result that the hermal stress of the friction surface may exceed the strength of the steel plate material limit, at this time, the outer space of the material will suffer great damage and then the initial crack happens. Compared with Figure 2 and 3, we can clearly see that the circumferential stress is much larger than the radial stress. With the increasing number of brake, the alternating thermal stress will lead to the common brake disc crack in actual situation, the radial thermal fatigue crack. In the case of the unreasonable of the brake, the stress of the brake cannot apply the force evenly on the on friction lining, and the contact area of the contact pressure will be more unreasonable, seriously, which is easy to cause the thermoelastic instability phenomenon.

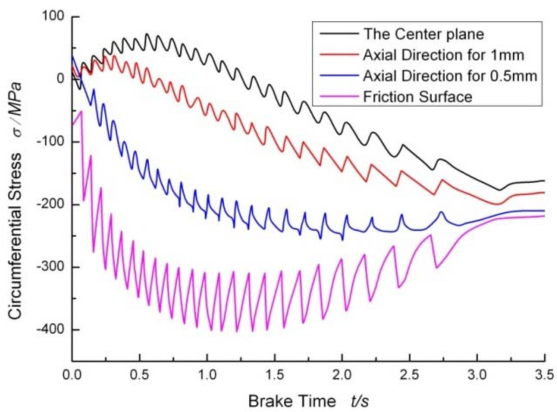


Figure 4. Different Point Circumferential Stress at r =114.5mm

In the Figure 2~4, we can find that the brake disc temperature field and stress field are coupled. At the beginning of the brake, the disc temperature rises fast, and at the same time the stress also changes fast. When the temperature of the disc trends to uniformity, the stress of the plate on the same axial section also has the tendency to be more consistent. While more closer to the surface of the friction, more prominently the temperature curve behaves, and these fluctuations are due to the exchange effect between the heat stroke of moving heat source and the heat convection. With the time extension and the decreasing strength of the inputting heat flow, the tendency of the frequency of the fluctuation is showed from fast to slow. This kind of heat stroke and uneven temperature distribution of the corresponding thermal stress led to fluctuations, which will cause thermal shock and thermal fatigue.

5. Conclusion

(1) This paper sets up the wet brake friction pair of fluid-solid coupling mathematical model, taking the actual size, materials, oil groove form and the influence of fluid-structure coupling effect into consideration, which can better response the dynamic property of the temperature field, stress field and seepage field in the braking process of the wet brake.

(2) In the process of braking, the friction pair exists strong thermal shock, and the brake disc temperature field, stress and seepage field have the coupling relationship, varying from the time constantly.

(3) The theoretical calculation data curves is in accord with the classic trend data, thus it shows that the establishment of the wet brake fluid solid coupling mathematical model of this paper is accurate in certain degree.

(4) This research considers the oil film between the brake linings state of motion as seepage, and gives an example of fluid-solid coupling for wet brake disc. Through analysis and validation, this may conclude that the proposed method makes good use of the advantages of the wet brake disc for heavy truck, and gets an efficiency and accuracy mathematical model.

Acknowledgements

This work was financially supported by a Grant from the National Natural Science Foundation of China (51375519), and Chongqing Jiaotong University postgraduate education Innovation Fund project.

References

[1] GUAN Di-hua, SU Xin-dong. An Overview on Brake Vibrations and Noise. *Engineering Mechanics.* 2004; 21(4): 150-155.
[2] PENG Chunbao, ZHOU Changfei, WANG Chongjian. Study on high temperature problems of wet multi-disc brake in non-braking state based on ANSYS. *Mining & Processing Equipment*. 2013; 41(5): 16-20.
[3] Li Jie, Wang Min, Wang Zhi-Yong, Zhou Guang-Ming. Transient thermo-mechanical coupling simulation of wet brake friction disk on tracked vehicle. *Journal of Beijing Institute of Technology (English Edition).* 2011; 20(1): 71-76.
[4] Zivanovic, Zlatomir. Thermal load of multidisc wet friction assemblies at braking regime. *Journal of Mechanical Engineering*. 2012; 58(1): 29-36.
[5] Song Yunpu. Parameter design for the energy-regeneration system of series hydraulic-hybrid bus. *Telkomnika.* 2012; 10(6): 1293-1300.
[6] XING Yu-tao, ZHAN Kai, LIU Da-Wei, YAN Tian-yi. Finite Element Analysis Of Close Wet Multi-Disc Brake Temperature Field. *Mining & Metallurgy.* 2007; 16(1): 57-60.
[7] Payvar P, Lee YN, Miokowyca WJ. Simulation of heat transfer to flow in radial grooves of friction pairs. *Int J Heat Mass Transfer.* 1994. 37(2): 313-319.
[8] Zagrodzki R. *Analysis of temperatures and stresses in wet friction disks involving thermally induced changes of contact pressure*. SAE Technical Paper 9892025. 1998.
[9] Sun Dongye, Hu Fengbin, Qin Datong, Liu Zhenjun. An Experimental Analysis on Thermal Stress Distributions in a Multiple Disc Wet Brake. *China Mechanical Engineering*. 2010; 21(16): 2006-2010.
[10] LI Wen-jun. Feature and Application of Wet Multi Disc Brake. *Coal Mine Machinery*. 2013; 34(8): 190-191.
[11] Yunpu Song. Study on the energy-Regeneration-based velocity control of the hydraulic- Hybrid vehicle. *Telkomnika.* 2012; 10(7): 1700-1707.
[12] Jiang Yong. Optimal design method with time-varying uncertainty for wet multi-disc brakes. *Transactions of the Chinese Society of Agricultural Engineering.* 2012; 28(9): 52-56.