RESEARCH ON DESIGN METHOD OF A MULTIPLE DISC WET BRAKE IN LUBRICATED ENVIRONMENT*

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Abstarct: The rmomechanical phenomena occurring between friction pairs greatly change the distributions of lining pressure and friction surface temperature of a multiple disc wet brake. It has become one of the main causes of brake failure. In order to understand these thermomechanical phenomena, several design and material factors that have great influence on thermomechanical phenomena, such as heat transfer coefficient, friction factor, sliding velocity, initial lining pressure and so on, are analyzed. An isothermal design method is proposed for designing a multiple disc wet brake.

Key words: Wet brake Thermomechanical phenomena Heat transfer coefficient Friction factor

0 INTRODUCTION

A multiple disc wet brake mainly consists of friction pairs, an opposing plate and a piston. Each friction pair includes a friction plate and a steel plate. The friction plate is a disc whose metal core is lined on both sides with friction material. The structure of a normal wet brake is shown in Fig.1.

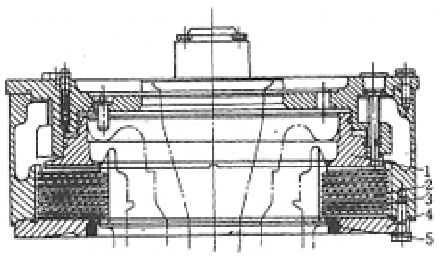

Fig.1 Multiple disc wet brake

Thermomechanical phenomena are caused by non-uniform thermoelastic deformation. Because of non-uniform distributions of heat flux, heat transfer coefficient and friction factor in the radial direction, friction surface temperatures will increase in different rates during an engagement period of a wet brake. In areas of higher temperature, greater thermo-elastic deformation will be seen. Even if there are uniform distributions of initial heat flux, deformations will become non-uniform due to the effects of heat transfer coefficient and friction factor variation. In local regions where deformations are higher, greater pressures must be generated. In turn, this causes higher temperature rising and further increases local pressures due to the thermal expansion in these regions. This process is called thermomechanical phenomena[1]. The phenomena will lead to reduction of the real contact area, increase of surface temperature and wear rate of friction plates. The thermomechanical phenomena are the main cause of the failure of a wet brake [2].

Thermomechanical phenomena effects can be decreased by optimizing the distribution of friction lining pressure. Based on the finite element analysis, an isothermal design method is proposed for designing a multiple disc wet brake.

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1 THEROMECHANICAL PHENOMENA INFLUENCE FACTORS

1.1 Influence of friction factor

The dynamic friction factor between friction plate and steel plate is one of the influence factors on the thermo-mechanical phenomena. When the product of sliding velocity v and lining pressure p is constant, the heat flux q ($q = pvf$) may be different with the change in the friction factor f. So that, a non-uniform thermoelastic deformation will be generated.

Friction materials used in wet brakes are divided into many types according to their compositions, such as sintered bronze-, graphite-, elastomer- and paper-based materials. Material changes can exert outstanding influence on mean dynamic friction factor during an engagement period. Even if the same type of friction material, the mean dynamic friction factor f will be greatly different with the changes in friction surface temperature t, sliding velocity v and lining pressure p[3].

The paper-based friction material was researched in this paper. "Paper" is noted for its high dynamic friction factor and extremely low static/dynamic coefficient ratio. This feature makes the paper-based friction material to be smoother and quieter during a braking period.

In order to obtain the regularity of friction factor varying with temperature t, velocity v and pressure p, an orthogonal experiment on a paper-based friction material is finished in an LBA0049 inertia dynamometer.

In this experiment, the friction factor is defined as the objective value. The parameters, such as temperature t, velocity v and pressure p, are analyzed in a multiple linear regression method. A standard orthogonal table $L_8(2^7)$ is adopted. Parameters z_{2i}, z_{1i}, z_{0i}, Δ_i and x_i are defined as upper limit, lower limit, zero limit, changing range and code variable, respectively. The changing range of all parameters are shown in Table 1. Code variables can only change from −1 to +1.

$$z_{0i} = (z_{2i} + z_{1i})/2 \tag{1}$$

$$\Delta_i = (z_{2i} - z_{1i})/2 \tag{2}$$

$$x_i = (z_i - z_{0i})/\Delta_i \tag{3}$$

Table 1 Code variable scheme

Variable	Temperature (z_1) t/°C	Velocity (z_2) v/(m•s^{-1})	Pressure (z_3) p/MPa
$z_{1i}(-1)$	80	0.3	0.7
$z_{2i}(+1)$	90	0.4	0.9
z_{0i}	85	0.35	0.8
Δ_i	5	0.05	0.1
Code function	$x_1=(z_1-85)/5$	$x_2=(z_2-0.35)/0.05$	$x_3=(z_3-0.8)/0.1$

The regression equation of the friction factor is given as

$$f = b_0 + \sum_{i=1}^{3} b_i x_i + \sum_{j<i} b_{ji} x_j x_i \tag{4}$$

where b_0, b_i, b_{ji} ——Regression coefficients

They can be obtained by least-squares fitting method shown in Table 2.

$$b_0 = \frac{B_0}{N \cdot M} = \frac{\sum_{p=1}^{N} f_p}{N \cdot M} \tag{5}$$

Table 2 Multivariate linear regression analysis of friction factor

No.	(0)	t (1)	v (2)	$t \times v$ (3)	p (4)	$t \times p$ (5)	$v \times p$ (6)	$t \times v \times p$ (7)	$y_{ij} = (f_{ij} - 0.115) \times 10^4$ y_{i1}	y_{i2}	y_{i3}	$\sum_{j=1}^{M} y_{ij}$
1	1	1	1	1	1	1	1	1	–47	–52	–43	–142
2	1	1	1	1	–1	–1	–1	–1	–9	2	–2	–9
3	1	1	–1	–1	1	1	–1	–1	–14	–2	–16	–32
4	1	1	–1	–1	–1	–1	1	1	–14	–10	–15	–39
5	1	–1	1	–1	1	–1	1	–1	–22	–31	–29	–82
6	1	–1	1	–1	–1	1	–1	1	22	22	9	53
7	1	–1	–1	1	1	–1	–1	1	42	32	35	109
8	1	–1	–1	1	–1	1	1	–1	52	43	49	144
y_{k1}		–222	–180	102	–147	23	–119	–19	10	4	–12	2
y_{k2}		224	182	–100	149	–21	121	21				
y^2_{k1}		49 284	32 400	10 404	21 609	529	14 161	361				
y^2_{k2}		50 176	33 124	10 000	22 201	441	14 641	441				
B_i, B_{ji}	2	–446	–362	202	–296	44	–240	–40				
b_i, b_{ji}	0.083	–18.58	–15.08	8.417	–12.33	1.83	–10.0	–1.67				
S_k		8 288	5 460	1 700	3 651	81	2 400	67				
F_k		19.32	12.73	3.96	8.51	—	5.60	—				
a_k		0.01	0.01	0.10	0.01	—	0.05	—				

$w = \sum_{i=1}^{N}\sum_{j=1}^{M} y_{ij}^2$ $Q_k = \frac{2}{NM}\sum_{k=1}^{2} y_{jk}^2$

$p = \frac{1}{NM}\left(\sum\sum y_{ij}\right)^2$ $S_k = Q_k - p$

$f=23$ $f_k=1$ $f_e=18$ $F_{0.01}(1,18)=8.28$ $F_{0.05}(1,18)=4.41$ $F_{0.10}(1,18)=3.01$

$$b_i = \frac{B_i}{N \cdot M} = \frac{\sum_{p=1}^{N} x_{pi} f_p}{N \cdot M} \qquad i = 1, 2, 3 \tag{6}$$

$$b_{ji} = \frac{B_{ji}}{N \cdot M} = \frac{\sum_{p=1}^{N} (x_{pi} x_{pj}) f_p}{N \cdot M} \qquad j > i \tag{7}$$

where N——Number of experiments, $N=8$

M——Repeated number in the same experiment, $M=3$

The marked test of the regression equation coefficients can be gotten by

$$F_k = \frac{S_k / f_k}{S_e / f_e} \sim F_a(f_k, f_e) \tag{8}$$

$$S_e = \sum_{i=1}^{N}\sum_{j=1}^{M} y_{ij}^2 - \frac{1}{N \cdot M}\sum_{i=1}^{N}\left(\sum_{j=1}^{M} y_{ij}\right)^2 + S_k(t \times p) + S_k(t \times v \times p) \tag{9}$$

The regression equation of the friction factor is obtained as

$$f = 0.083\,3 - 18.583x_1 - 15.083x_2 + 8.417x_1x_2 - 12.333x_3 - 10.0x_2x_3 \tag{10}$$

If temperature t, velocity v and pressure p are instead of code variables, the equation of the friction factor can be changed to a new function

$$f = 0.211\,2 - 0.001\,55t - 0.156\,4v + 0.057\,7p + 0.003\,37tv - 0.2vp \tag{11}$$

1.2 Influence of heat transfer coefficient

Friction material surfaces have grooves that allow flow of cooling oil. Heat transfer coefficient distribution will greatly change with oil groove patterns and radial position. Even if the distribution of heat flux is uniform along the radial direction, the thermoelastic deformation will also be non-uniform due to the change of the heat transfer coefficient. So the heat transfer coefficient between cooling oil and steel plate is also one of the influence factors on the thermomechanical phenomena.

First, the parabolic flow needs to be defined. If there exists a predominant direction of a three dimensional flow, the diffusions of momentum, heat, mass, etc, can be neglected in this direction. If there is no reverse flow, the flow is called a parabolic flow [4].

Before introducing the mathematical equations on the groove heat transfer problem, a description of certain simplifying assumptions will be given in the mathematical analysis of laminar flow and heat transfer in a single groove (Fig.2).

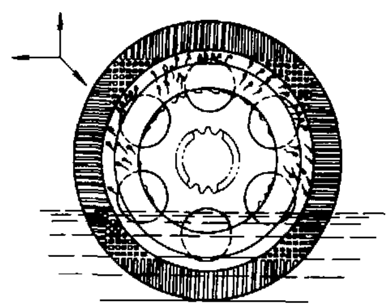


Fig.2 Scheme of groove pattern

(1) In the main flow direction, the diffusions of the heat and mass are neglected, and the convection is the main influence on the heat transfer between the cooling oil flow and plates.

(2) Because a great difference between the thermal conductivity of the steel plate and paper-based friction plate, most of the heat generated in a braking period is absorbed by the steel plate. The thermal conductivity between friction plate and cooling oil can be thought to equal zero.

The above assumptions reduce the amount of computations by orders of magnitude. The three dimensional flow will be reduced to a series of two dimensional flows. The equations in the Cartesian co-ordinates x, y, z (Fig.2) to be solved are as follows.

Continuity

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{12}$$

Navier-Stokes equations

$$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = \rho X - \frac{\partial p_f}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right)$$

$$\rho\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = \rho Y - \frac{\partial p_f}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right)$$

at $x=0,\ 0 \leqslant y \leqslant d;\ u=0,\ v=0$

at $x=b,\ 0\leqslant y\leqslant d;\ u=0,\ v=0$

at $y=0,\ 0\leqslant x\leqslant b;\ u=0,\ v=0$

at $y=d,\ 0\leqslant x\leqslant b;\ u=0,\ v=0$ (13)

where u, v, w ——Velocity components in the x, y and z directions

b, d, l ——Width, depth and length of groove

ρ ——Oil density

μ ——Dynamic viscosity

p_f ——Pressure of oil flow

X, Y, Z ——Mass force components in the x, y and z directions.

The results of u and v can be obtained by a numerical solution using the finite difference method. The fully developed distribution of the velocity w can be obtained from the following equation

$$\frac{\partial w}{\partial z}\approx 0 \qquad \frac{\partial^2 w}{\partial z^2}\approx 0 \tag{14}$$

$$\rho\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}\right)=\rho X-\frac{\partial p_f}{\partial y}+\mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}\right)$$

at $x=0$ or $x=b,\ 0\leqslant y\leqslant d;\ w=0$

at $y=0$ or $y=d,\ 0\leqslant x\leqslant b;\ w=0$ (15)

When the first assumption is satisfied, $\partial^2 t/\partial z^2$ equals zero. The temperature distribution can be obtained from the following equations

$$\rho\left(u\frac{\partial t}{\partial x}+v\frac{\partial t}{\partial y}+w\frac{\partial t}{\partial z}\right)=\frac{\lambda_f}{c_p}\left(\frac{\partial^2 t}{\partial x^2}+\frac{\partial^2 t}{\partial y^2}\right)$$

at $y=d,\ 0\leqslant x\leqslant b;\ t=t_m$

at $y=0,\ 0\leqslant x\leqslant b;\ \dfrac{\partial t}{\partial n}=0$

at $x=0,\ 0\leqslant y\leqslant d;\ \dfrac{\partial t}{\partial n}=0$

at $x=b,\ 0\leqslant y\leqslant d;\ \dfrac{\partial t}{\partial n}=0$

at $z=0,\ 0\leqslant x\leqslant b,\ 0\leqslant y\leqslant d;\ t=t_0$ (16)

where t_m ——Temperature of the steel plate

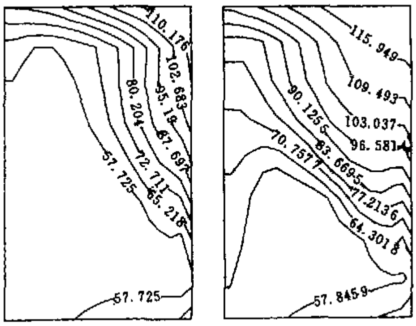
t_0 ——Initial temperature of cooling oil

$\partial t/\partial n$ ——Normal temperature gradient

c_p ——Specific heat of oil

λ_f ——Thermal conductivity of oil

On the basis of the velocity fields of the cooling oil obtained by Eqs.(12)~(15), the temperature distributions on each cross sections can be gotten by the Eq.(16) by the finite difference method. According to the parameters shown in Table 3, when z equals 35.5 mm and 71 mm, The results are shown in Fig.3. The average temperature gradient at the normal direction between the steel plate and cooling oil is given by



(a) (b)

Fig.3 Temperature field at z=35.5 mm and z=71 mm

$$\left.\frac{\partial \bar{t}}{\partial n}\right|_{n=0}=-\frac{\displaystyle\int_0^b \left.\frac{\partial t}{\partial y}\right|_{y=0}\mathrm{d}x}{b} \tag{17}$$

Table 3 Structural and physical parameters

Inside radius of friction plate r_i / mm	160.5
Outside radius of friction plate r_o / mm	231.5
Width of oil groove b / mm	3.18
Depth of oil groove d / mm	0.64
Length of oil groove l / mm	71
Mass flow rate of oil in a single groove m_i /(kg·s^{-1})	3.24×10^{-3}
Initial temperature of cooling oil t_0 /°C	50
Specific heat of oil c_p /(J·kg^{-1}·K^{-1})	2177
Thermal conductivity of oil λ_f /(W·m^{-1}·K^{-1})	0.126

Finally, the average of heat transfer coefficient $\bar{\xi}$ along the radial direction is calculated by

$$\bar{\xi}=-\frac{\lambda_f}{t_m-t_\infty}\frac{\displaystyle\int_0^b \left.\frac{\partial t}{\partial y}\right|_{y=d}\mathrm{d}x}{b} \tag{18}$$

t_∞ is defined as the qualitative temperature and calculated by the following equation

$$t_\infty=\frac{\int_0^d\int_0^b wt\mathrm{d}x\mathrm{d}y}{\int_0^d\int_0^b w\mathrm{d}x\mathrm{d}y} \tag{19}$$

Fig.4 illustrates its heat transfer coefficient distribution of a multiple parallel groove in the radial direction. It shows that the cooling effect near the inner radius of the friction plate is obviously better than one near the outer radius. In designing a wet brake, greater heat flux should be generated near the inner radius to keep the uniform thermoelastic deformation along the radial direction of the plates.

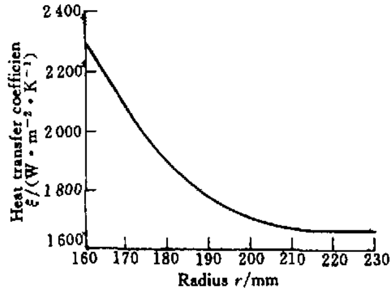


Fig.4 Geometric and material parameters of a brake

2 FEA MODEL OF THERMOMECHANICAL PHENOMENA

The scheme shown in Fig.1 can be described in Fig.5. It mainly includes friction plates, steel plates, an opposing plate and a piston. In order to establish the relevant FEA model, the following key points need to be considered.

(1) The most important structural character of a multiple disc wet brake is gaps between friction plates and steel plates. In order to calculate the pressure distributions between the friction pairs, the gap elements are used. The FEA model with gap elements becomes a non-linear model.

(2) The friction factor is not constant. It will change with the different using conditions, such as plate surface temperature t, sliding velocity v and lining pressure p.

(3) Heat transfer coefficient between cooling oil and steel plate is not constant in the radial direction.

In the FEA model, each component is assumed to be a linear elastic object. The system external forces include the hydraulic pressure p' acting on the piston and the supporting force acting on the opposing plate by floating oil ring in the axial direction. The hydraulic pressure p' is defined to a dimensionless dimension as follows

$$p' = \frac{pA}{F} \tag{20}$$

where F——Resultant force applied to the piston

A——Gross area of a single friction surface

p——Lining pressure

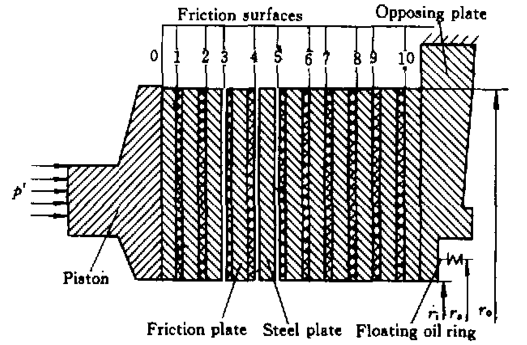


Fig.5 Scheme of a real multiple disc wet brake

When the geometric and material parameters of a multiple disc wet brake are given in Table 4.

Table 4 Geometric and material parameters of a brake

Thickness of steel plate g_s / mm	2.4
Thickness of friction plate g+2g_f / mm	2.74+2×1.18
Oil ring acting radius r_s / mm	174
Elasticity modulus of steel E_s / GPa	200
Elasticity modulus of friction material E_f / GPa	2.1
Poisson ratio of steel μ_s	0.3
Poisson ratio of friction material μ_f	0.2
Oil pressure p /MPa	2.5

In order to verify the FEA model, an experiment on initial lining pressure distribution is finished. The results by finite element analysis and experiment are shown in Fig.6.

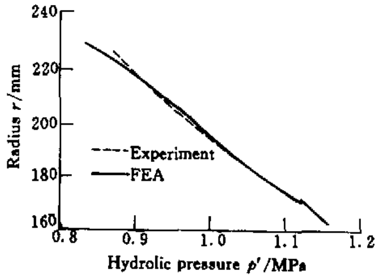


Fig.6 Dimensionless lining pressure distributions

In actual brakes, geometry of the piston and the opposing plate are complex. Methodology for designing the modulus of rigidity distributions of the piston and the opposing plate is the key in establishing ideal initial pressure distribution.

3 DESIGN METHOD

In the operation of a wet brake, two distinct modes may be identified. In emergency braking mode, the friction plate and the steel plate slip with respect to each other for a very short time. It usually becomes from 0.2 to 2 s. The frictional heat generated during an engagement period is mostly absorbed by the steel plate with the oil flowing through the grooves playing a secondary role in heat removal. In continual braking mode, the two plates may slip for periods of time as long as 10~20 s. In this mode, the temperatures at various points in the plates reach final steady state values in a few seconds. After which all of the generated heat must be removed by the oil flowing in the grooves of the friction plate. The value of the heat transfer coefficient between the steel plate and the groove oil will determine the steady state temperature level in the brake.

The isolines of temperatures t of a frictional pair and tangential thermal stresses σ_t of the same steel plate are shown as Fig.7 in a continual braking, respectively.

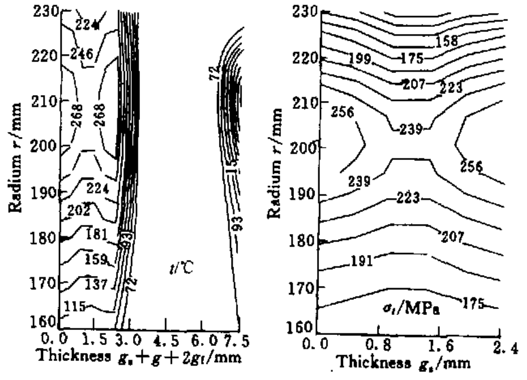


Fig.7 Isoclines of temperature and tangential stress in a continual braking

In an emergency braking, surface temperatures and tangential stresses of the steel plate were measured by thermal couples and strain gauges in an LBA0049 inertia dynamometer. The thermal couples and strain gauges were uniformly distribution along the radial direction. The comparison between experimental and calculated results are shown in Fig.8.

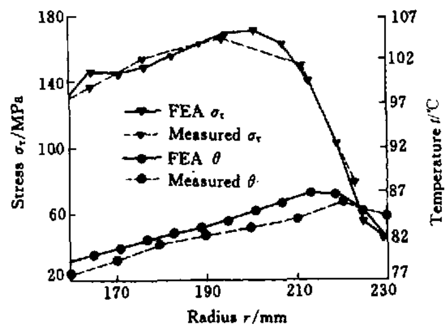


Fig.8 Comparison of measured and calculated results

Although the initial lining pressure is higher near the inside radius (shown in Fig.6), Fig.7 and Fig.8 show that high dynamic stress and high temperature are near the outside radius. This is why the local heat flux input at any point is a function of normal pressure, friction factor and sliding velocity. Although a wet brake may be designed to the low average energy per unit lining area, local high lining pressure spots may cause surface burning of friction lining due to thermomechanical instability between the friction pairs.

The steel plate dishing is usually found at the first steel plate near the piston and the last steel plate near the supporting plate. The main cause of the failure is due to great temperature difference and dishing thermal deformations. Because the two steel plates only have a single friction surface, the dishing failure may be more generated than other plates.

The crazing failure of steel plate is caused by fatigue damage in repeated braking cycle. During an engagement period, the surface temperature rises much more rapidly than the main body of the steel plate dose. It induces compressive stresses at the outer of the steel plate, balanced by tensile stresses in the cooler inner of the steel plate. When this braking cycle is over, the outer temperature of the steel plate will become cooler than the inner temperature of the steel plate due to the effect of cooling oil. The compressive stresses at the outer of the steel plate become tensile stresses, balanced by compressive stresses in the hotter inner of the steel plate. So the crazing may occur in an unacceptably low number of cycles.

To summarize these analyses, in order to prevent brake damage caused by high local temperatures and stresses, it is necessary to perform the finite element calculations of the dynamic stress and surface temperature. The proper design procedure for a wet brake can be described in the following steps: First, the initial lining pressure distribution in the radial direction is estimated under the condition of uniform heat flux. Second, considering the influence of non-uniform distributions of heat transfer coefficient and friction factor, the initial lining pressure distribution is revised on the basis of the dynamic stress and friction temperature. Third, in order to realize the uniform distributions of surface temperature and thermoelastic deformation, the structures of the piston and opposing plates are optimized. This is an isothermal design method. It will reduce the unfavorable thermomechanical phenomena.

4 CONCLUSIONS

(1) Thermomechanical phenomena are caused by the nonrational initial lining pressure and the non-uniform distributions of heat transfer coefficient and friction factor. Thermomechanical phenomena leading to high local temperature and high stress are the main failure cause of a multiple disc wet brake.

(2) The tangential stresses on the surface of steel plates are greater than the radial stresses. So the surface cracking of steel plates is usually generated in the radial direction.

(3) How to design the geometry of a piston is the key point in a multiple disc wet brake. The modulus of rigidity distribution of the piston has a great influence on the initial lining pressure distributions between the friction pairs.

(4) In order to avoid the failure of a multiple disc wet brake, the isothermal design method is proposed. In other words, a wet brake designer should try his best to achieve the uniform temperature distributions along the radial direction of steel plate.

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