On-lineMonitoringofTribologyParametersandFaultDiagnosisforDiscBrake System

Zhao-Jian Yang[1]**Seock-Sam Kim**[2]

[1] *College of Mechanical Engineering, Taiyuan University of Technology, Taiyuan, 030024, China*

[2]*School of Mechanical Engineering, Kyungpook National University, Daegu, 702-701, Korea*

Abstract: The basic principles and methods of the on-line monitoring of tribology parameters (friction coefficient and wear allowance) and fault diagnosis for the hoist disc brake system were introduced, the method were based on the spring force and oil pressure of the brake system and the hoist kinematics parameters. The experiment on the monitoring and diagnosis of hoist brake system were carried out. The research results showed: the monitoring and diagnosis methods are feasible.

Key words: Tribology parameter, Brake system, Monitoring, Fault diagnosis

1. Introduction

Disc brakes are widely used in cars, trains, hoisting transportation equipment, so as to guarantee safety of theequipmentinoperation.Themoderntribologytheory and many correlation researches showed that the tribology parameter changes of the brake system is a very complex process, the quantitative analysis of the parameters including friction coefficient and wear is difficult,becausetheparameterscanbealteredbyother parameters for instance friction velocity, temperature, pressure, medium, environment, and system property where friction material works and so on. [1][2][3][4][5] The studyonon-linemonitoringoftribologyparametersand fault diagnosis for disc brake system becomes more important.Thispapertakesthediscbrakesystemofmine hoistastheinstance,themonitoringandfaultdiagnosis ofthebrakesystemwerecarriedoutbasedonthespring force and oil pressure of brake system and the hoist kinematics parameters.

2. Monitoring principle and method

2.1 Friction coefficient monitoring

2.1.1 Monitoring principle

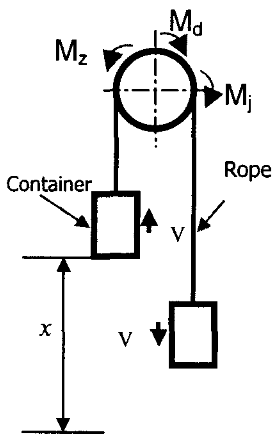

Fig.1 Hoist working principle diagram

1-brake disk 2-brake shoe 3-piston 4-disk spring 5-load cell

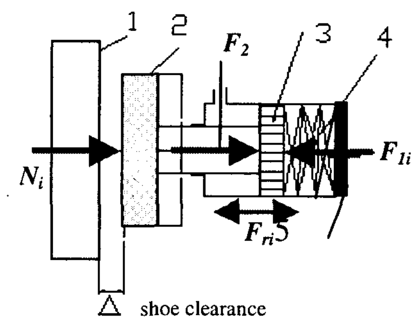

Fig.2 Disc brake working principle diagram

The Hoist working principle is shown as Fig. 1, the disc brake working principle of the hoist is shown as Fig.2. From Fig.1 and Fig.2, the equilibrium kinetics equation of the hoist on mechanical braking is

$$M_d = M_z \pm M_j \quad (1)$$

Where,

M_d is the dynamic moment of the hoisting system,

$$M_d = R\sum ma_z \quad (2)$$

M_j is the static moment of resistance of the hoisting system, $M_j = RF_j$ (3)

M_z is the braking moment of the hoisting system,

$$\begin{aligned} M_z &= \mu R_m \sum_{i=1}^{2n} N_i \\ &= \mu R_m \sum_{i=1}^{2n} (F_{1i} - F_2) \end{aligned} \quad (4)$$

Note: The running resistance of the brake F_{ri} is leaved out in formula (4), and hoisting load uses the plus sign "+", dropping load uses the minus sign"-" in formula (1). By formulas (1), (2), (3), (4), the braking moment can be shown as

$$\begin{aligned} M_Z &= M_d \mp M_j \\ &= R\left(a_z \sum m \mp F_j\right) \end{aligned} \quad (5)$$

By formula (4), (5), the friction coefficient can be shown as

$$\mu = \frac{R}{R_m} \cdot \frac{a_z \sum m \mp F_j}{\sum_{i=1}^{2n} (F_{1i} - F_2)} \quad (6)$$

Where,

μ is the mean friction coefficient of the brake system,

R is the radius of the hoisting drum,

R_m is the friction radius of the brake disk,

a_z is the deceleration of the hoisting system,

F_{1i} is the spring force caused by the brake No.i,

F_2 is the oil pressure of the brake system,

N_i is the vertical pressure of the brake No.i,

n is the amount of the brake in the brake system,

$m\Sigma$ is the equivalent mass of the all moving components in the hoisting system on the hoisting drum circumference,

F_j is the static resistance of the hoisting system,

For the shaft:

$$F_j = \pm Q + (n_1 p - n_2 q)\cdot(H - 2x) + \omega_1 + \omega_2 \quad (7)$$

For the inclined well:

$$\begin{aligned} F_j &= \pm n_3 q_1 \sin\alpha + n_3(q_1 + 2q_c) f_1 \cos\alpha \\ &+ p(L - 2x)\sin\alpha + pLf_2 \cos\alpha + \omega_1 + \omega_2 \end{aligned} \quad (8)$$

Note: Hoisting load uses the plus sign "+", dropping load uses the minus sign"-" in formula (7), (8).

Where,

Q is the hoisting load,

p is the hoisting rope weight per unit meter,

q is the balancing rope weight per unit meter,

n_1 is the number of the hoisting ropes,

n_2 is the number of the balancing ropes,

H is the hoisting altitude,

x is the hoisting container place,

n_3 is the number of the tubs,

q_1 is the load of the tub,

q_c is the self-weight of the tub,

α is the slanting angle of the pit shaft,

ω_1 and ω_2 are the hoisting and dropping running resistance separately,

$\omega_1 + \omega_2 = (K-1)Q$,

K=1.15 (skip hoisting); K=1.2 (cage hoisting),

f_1 is the resistance coefficient of the tub,

f_2 is the resistance coefficient of the hoisting rope,

L is the hoisting length.

By means of the calculation or testing of the equivalent mass $m\Sigma$[6], monitoring the deceleration a_z, the spring force $F_{1i} (i=1,2,\ldots,n)$, the oil pressure F_2, the on-line monitoring for the friction coefficient of the brake system can be achieved by formula (6). Namely, the on-line monitoring for the friction coefficient is the on-line monitoring for the spring force F_{1i}, the oil pressure F_2, the deceleration a_z and the change mass of position $m\Sigma$

2.1.2 Monitoring method

The equivalent mass $\bar{m}$ can be calculated with the technical description of the hoisting system, and can be tested by the on-site testing method introduced in the reference [5].

When the hoist is running in full velocity and the hoisting container is about at $x=H/2$ position, the brake system brakes at once, the velocity change ΔV and the brake time T are collected at the same time, the deceleration a_z can be obtained by the formula (9).

$$a_z = \Delta V / T \text{ (m/s}^2\text{)} \quad (9)$$

The oil pressure F_2 can be obtained by an oil pressure sensor, which is fixed in hydraulic pressure circuit system of the brakes.

The spring force $F_{1i}(i=1,2,...,2n)$ can be obtained by the special load cell[7], which are fixed in the brakes as shown Fig. 2.

2.2 Allowable wear monitoring

2.2.1 Monitoring principle

The spring force F_1 of the disc brake is decided by the linear deformation of the spring, as shown Fig3. namely,

$$\begin{cases} F_{1a} = k \cdot (l_{\max} - l_{\max}) = 0 \\ F_{1b} = k \cdot (l_{\max} - l_1) \\ F_{1c} = k \cdot (l_{\max} - l_2) = k \cdot (l_{\max} - l_1 - \delta) \\ F_{1d} = F_{1e} = k \cdot (l_{\max} - l_{\min}) = F_{1\max} \end{cases} \quad (10)$$

Where,

k is the spring stiffness;

δ is the wear thickness of brake shoe;

l_{max} is the spring length and the F_{1a} is spring force when the spring is free, as shown Fig.3(a);

l_1 is the spring length and the F_{1b} is spring force when the brake is full closed, as shown Fig.3(b);

l_2 is the spring length and the F_{1c} is spring force when the brake is full closed and the wear thickness of brake shoe is the δ, as shown Fig.3(c);

l_{min} is the spring length and the F_{1d} is spring force when the brake is full opened and the brake clearances are the Δ and $\Delta+\delta$ separately, as shown Fig.3 (d) and (e)

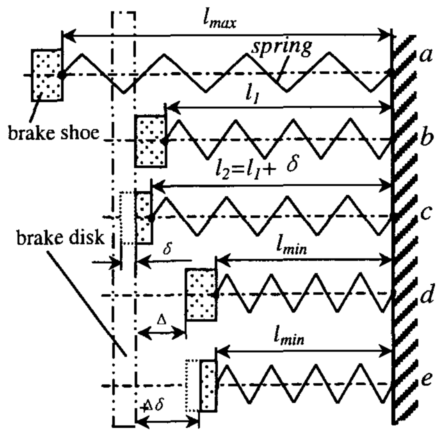

Fig.3 Wear monitoring principle

a—spring free; b—braking condition; c—braking condition (brake shoe has worn thin) δ; d—brake shoe clearance Δ condition; e—brake shoe clearance $\Delta+\delta$ condition.

By the formula (10), we can got the wear formula as follows,

$$\delta = \frac{1}{k}(F_{1c} - F_{1b}) \quad (11)$$

By means of testing of the F_{1b} and on-line monitoring the F_{1c}, the on-line monitoring for the brake shoe wear of the brake system can be achieved by formula (10). Namely, the on-line monitoring for the brake shoe wear is the on-line monitoring for the spring force F_{1i}.

2.2.2 Monitoring method

The special load cells [7] are fixed in the brakes of the brake system as shown Fig. 2, the spring force F_{1bi} $(i=1,2,...,2n)$ can be obtained by the load cell when the brake shoes have no wear, afterwards we can monitor the brake shoe wear of every brake in the system with monitoring the spring force F_{1i} method. The formula of the wear calculation as follows:

$$\delta_i = \frac{1}{k}(F_{1i} - F_{1ib}) \quad (i=1,2,...,2n) \quad (12)$$

3. Fault Diagnosis

3.1 Basic principles

The fault diagnosis of the brake system can be carried out by the information of the spring force F_{1i} and the oil pressure F_2 and hoist kinematics parameters above-mentioned. The fault diagnoses include diagnosing spring force F_{1i}, the oil pressure F_2, the braking moment, the friction coefficient, and the brake shoe wear and clearance, the idle acting time period of brake.

3.2 Fault diagnosis method

1) If $F_2 < F_{2max}$ (maximum designed), the oil pressure is too low, accordingly it belongs to hydraulic system fault.
2) If $F_2 < F_{2min}$ (minimum designed), the residual voltage of the hydraulic system is higher.
3) If the mean friction coefficient of the brake system < or > the value designed, a serious fault warning is sounded.
4) If the wear thickness of some brake shoe > the value designed, a fault warning is sounded.
5) If F_2 is in normal limit but F_{1i} decreases heavily in volume, the spring may be fatigued or broken; if F_{1i} decreases or increases a little, the number i brake clearance i may be bigger or smaller.
6) If the number i brake shoe in the idle acting time period is greater than designed volume, the brake clearance may be larger or system resistance larger.
7) If F_{1i} cannot vary with the increase of the F_2, it shows the hydro-cylinder may be blocked by piston.

So, though obtaining the two types of the F_{1i} *and* F_2, the disc brake system can be monitored on line and diagnosed.

4. Experiment

Above all, the feasibility of monitoring on line and fault has already been proved in theories. The next we will prove the feasibility and the reliability of the method through the experiment. The experiments were carried out with the 2JTP-1.2 hoist rig in the lab. The experiments of the shoe clearance, the idle acting time period and the braking process are merely introduced in the paper. Recording the output of oil pressure sensor, load cell and hoist drum rotation velocity, while alternating the brake-off-brake test; adjusting the brake clearance, alternating the test again and recording the output curves of the sensors.

The testing result of brake shoe clearance and error analysis is shown as table 1. The maximum output error is 1.14%; it has good test accuracy.

Table 1 Brake shoe clearance testing

brake shoe clearance/mm	output of load cell/mV		
	s	a	(s-a)/s%
3.0	3.5	3.5	0.00
2.6	3.72	3.7	0.50
2.2	3.94	3.9	1.00
1.8	4.16	4.13	0.72
1.4	4.38	4.33	1.14
1.0	4.6	4.58	0.43

Note: *s* is standard corresponding brake shoe clearance, *a* is average of four times.

The idle acting time period can be calculated according to the testing curves, as shown table 2, the data are accurate with the higher reliability.

Table 2 Idle acting time period testing

brake shoe clearance/mm	Idle acting time period /s			
	1	2	3	average
3.0	0.567	0.600	0.567	0.578
1.5	0.450	0.450	0.400	0.433
1.0	0.383	0.400	0.417	0.400
0.5	0.333	0.333	0.333	0.333

The braking process testing curves are shown Fig. 4, the whole process falls into four steps.

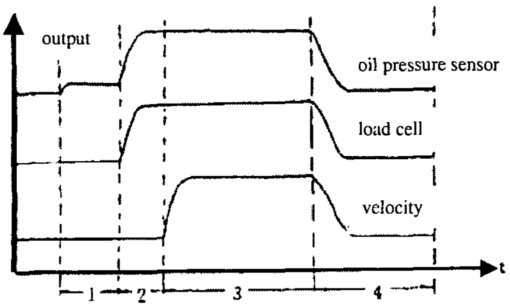

Fig. 4 braking process curves

(1) From zero pressure to a little pressure, the oil pressure sensor indicates the residual pressure of the brake system.

(2) Off brake, the output of the load cell and oil pressure sensor increase from zero to maximum.

(3) Drum rotation, velocity output increase from zero to maximum.

(4) Brake, the all output curves gradually return to home position.

5. Discussions and Conclusions

The friction coefficient in formula (6) is a approximate mean value, which usually denotes the entire tribology property (friction coefficient) of the $2n$ brakes in the brake system, so the entire tribology property of the brake system can be monitored on-line.

The wear value$_i$ (i=1,2,...,2n) in formula (12) denotes the wear condition of every brake in the brake system, so the tribology property (wear) of every brake can be monitored on-line separately.

The on-line monitoring and fault diagnosis method based on brake spring load cells and oil pressure sensor has higher accuracy and reliability, the key technique is the special brake spring load cell fitted for the brake structure, and the method is feasible for engineering application of the hoist and elevator equipment.

Reference

[1] Yang Z.J., Ren F., Kim S.S. Study on Tribology Characteristics of Friction Material Based on Tribo-Systems, KSTLE International Journal, June 2003, Vol. 4, No. 1, pp27-30.

[2] Luise G.H., Allan B., Georg T.N., Per M., Tribological properties of automotive disc brakes with solid lubricants, Wear Vol. 232, 1999, pp168–175.

[3] Bettge D., Starcevic J., Topographic properties of the contact zones of wear surfaces in disc brakes, Wear Vol.254, 2003, pp195–202.

[4] Severin D., Dörsch S., Friction mechanism in industrial brakes, Wear Vol.249, 2001, pp771–779.

[5] Ostermeyer G.P., On the dynamics of the friction coefficient, Wear, Vol. 254, 2003, pp852–858.

[6] Yang Z.J., Actually Measuring Equivalent Mass of Mine Hoist, Mining & Processing Equipment (China), Vol.22, (serial No.208), No.4 April 1994, pp34-37.

[7] Ren F., Yang Z.J., Research on the Hole Spoke sensor, Instrument Technique & Sensor (China), 2000, Vol.37 No.10, pp4-5.