THE DRILLING DRAW-WORKS BAND BRAKES THERMAL REGIME

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ABSTRACT

Operation systems of the drilling rigs are equipped with drilling draw-workses, which are provided both with mechanical and hydraulic brakes that can run separately or simultaneously. The mechanical brakes are band brake type, and they have the mission to take the biggest part of the braking moment, which appear during the rig operation. Because of the very high loads (up to 500 tf) the thermal stresses which appear in the adjacent zone of the friction surface are big, and the temperature values reach values in the range of 300…800^oC.

The aim of the paper is the friction surface thermal regime modelling for the band brake drums which functioning in real conditions of exploitation. The obtained results can be used directly both in design calculus and in band brake exploitation.

KEYWORDS: band brake drums, thermal regime.

1. CONSTRUCTIVE AND OPERATING ASPECTS OF DRILLING RIGS MECHANICAL BRAKING SYSTEM

Like mechanical brake of drilling draw-workses hoisting drum it was imposed in majority of the cases the double and equilibrated band brake. This kind of brake permits the braking with a relative reduce force which is manually applied at its lever, and it is self-detaching when the hoisting drum begin to rotate in an inverse sense. Usually, the entire weight of the unload travelling block can be braked only by the lever weight.

Generally for a mechanical brake, there appear two opposed requirements which are resolved using a compromise between the necessity of a high multiplication ratio existence and the necessity of a long time functioning between two readjustments (for ferrodo wear compensation).

When the lever is manipulated, there appears a resistant moment, which is the result of the friction forces that appear on the contact surfaces between the drum and the brake shoes.

In the brake band it appear a tensile force which varies on entire periphery between a maximum value F_2, corresponding of the fixed extremity, and a minimum value F_1, corresponding of the mobile ending of the band.

The friction force difference between the band extremities it equilibrates the tangential force F_p from brake drum periphery:

$$F_p = F_2 - F_1 \tag{1}$$

The relation between band extremities tensile forces is done by Euler formula for elastic wires:

$$\frac{F_2}{F_1} = e^{\mu\Psi} \tag{2}$$

where: e is the natural logarithm base; μ - friction coefficient; Ψ - grip hold angle of the brake. The friction coefficient μ has usually values in the range of 0.35…0.65, but its values depends on many factors, and the grip hold angle has values in the range of 270^o…325^o [1, 2].

The braking couple moment M_f, which is done by tangential forces F_p from the two drums brakes, like resultants of the friction forces, is done by the relation:

$$M_f = 2 \cdot F_p \cdot \frac{D_f}{2} = 2 \cdot F_1 \left(e^{\mu\Psi} - 1\right)\frac{D_f}{2} \tag{3}$$

where D_f is drum brake diameter.

The friction coefficient of brake shoes material has variations, which depend on working conditions: temperatures on the friction surface, contamination with wear particles, oil or drilling mud. The braking moment can be expressed depending on the applied force on the braking lever F_m and on the brake multiplication ratio, i:

$$M_f = F_m \cdot i \cdot \left(e^{\mu\Psi} - 1\right)\frac{D_f}{2} \tag{4}$$

The obtained braking moment has to be bigger than static forces moment corresponding to the hook load M_{st} and to the forces of inertia moment of the movement weights, M_d.

$$M_f \geq M_{st} + M_d \geq K_d \cdot M_{st} \tag{5}$$

If it considering:

$$M_{st} = F_1 \frac{D_{tu}}{2} \tag{6}$$

where: F_1 is the cable traction force of the hoisting drum; D_{tu} – the hoisting drum diameter. Using the relations (4) and (5) it can be writing:

$$F_m = \frac{F_t \cdot D_{tu}}{i \cdot \left(e^{\mu \Psi} - 1\right) \cdot D_f} \tag{7}$$

Analyzing the relation (7) it results the following conclusions:

- the increase of multiplication ratio implies the decrease of the lever force for the same braking load;
- the increase of the grip hold angle has the same influence [3];
- the braking force, for a giving hook load, is proportional with the ratio D_{tu}/D_f. So, it is better to increase the drums diameter and to decrease the hoisting drum diameter [1];
- in the relation (7) the brake drum width and the number of brake drums do not appear (the brake drums width is determined only from wear considerations).

The contact pressure between the friction material (ferrodo like) of the brake shoes and the drums, p, has a continuous exponential variation between the mobile ending where is minimum and the immobile ending where is maximum (fig. 1).

$$p_{max} = \frac{2F_2}{BD_f} = \frac{2F_1 e^{\mu \Psi}}{BD_f} = F_t \frac{D_{tu}}{BD_f^2} \frac{e^{\mu \Psi}}{e^{\mu \Psi} - 1} \tag{8}$$

$$p_{min} = \frac{2F_1}{BD_f} = F_t \frac{D_{tu}}{BD_f^2} \frac{1}{e^{\mu \Psi} - 1} \tag{9}$$

In relations (8) and (9) B is the width of the brakes shoes.

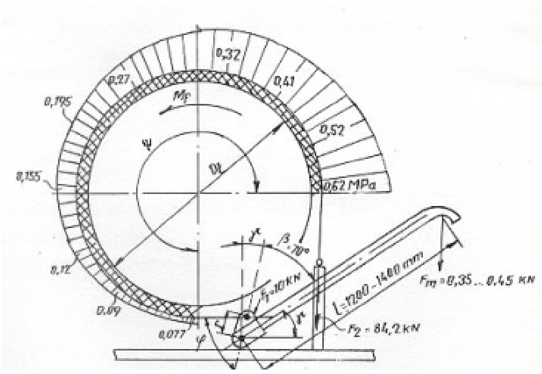

Fig. 1 The variation of the contact pressure between the friction material and the band brake drum

The mechanical and thermal stresses, which are supported by the band brake drum, depend on the relation between the following functional parameters: the friction coefficient, the specific contact pressure and the relative sliding speed between the friction couple elements.

As it is shown in figure 1, the contact pressure has a non-uniform distribution on the brake drum circum-ference, with values in the range of 0.077…0.62MPa. These values are usual in situations when the auxiliary brake ("slowly brake") of the draw-works functioning. In extreme situations the nominal contact pressure can reach values up to 2MPa [4]. The sliding speed varies in large limits, which are generally difficult to define. From constructive characteristics of the drilling draw-works it can be deduce that these functioning with rotational speed in the range of 300…400rpm and linear velocities of the cable in the range of 2…24m/s [5]. If it takes account on the drum diameters which is in the range of 900…2000mm, then the sliding speed values are in the range of 0.1…12m/s.

About the friction coefficient, it can be done the following remarks:

- for mechanical band brake of the draw-works, the friction coefficient value decreases with the increase of the nominal contact pressure (fig. 2) [4];
- for sliding speed values up to 1.5m/s, and for the same nominal contact value, the friction coefficient increases when the sliding speed increases;
- when the sliding speed has values bigger than 2.0 m/s the friction coefficient decreases both with the increase of the sliding speed and of the nominal contact pressure (fig. 2) because of the temperature increase;

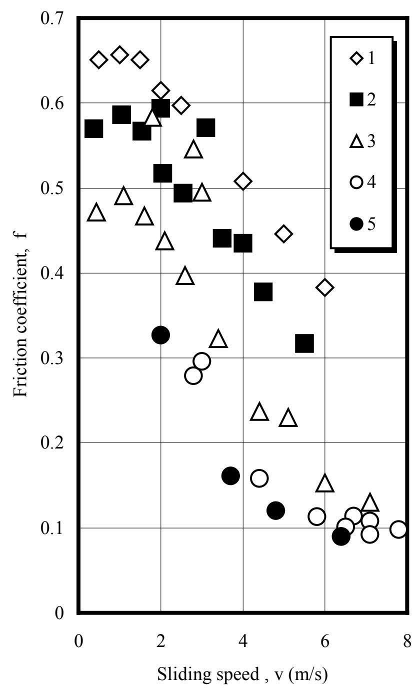

Fig. 2 Friction coefficient vs. sliding speed for different nominal contact pressure values:
1) 0-0.5MPa; 2) 0.5-1.0MPa; 3) 1.0-1.5MPa; 4) 1.5-2.0MPa; 5) 2.0-2.5MPa.

- the above statements are in accordance with the Kragelski conclusion which it has mentioned that for different friction couples, the friction coefficient reach a maximum value when the sliding speed increases and the nominal contact pressure has small values.

For emphasize the influence of the sliding speed, the nominal contact pressure, and the temperature on the friction coefficient, it can be used the regression relation (10) [6]:

$$f = 0.794 - 0.214\,p - 0.071\,v + 1.753 \cdot 10^{-4}\,t \qquad (10)$$

where: p is the nominal contact pressure, MPa; v – the sliding speed, m/s; t – temperature, °C.

In relation (10) it observes that among the regime parameters taken in consideration the nominal contact pressure has the major influence on the friction coefficient.

The friction surface temperature has a minor influence on the friction surface, its increase implies the friction coefficient increase. However, at its side the contact pressure determines the friction temperature.

Therefore, it can be said that the main parameters, which influence the friction temperature and the friction coefficient, are the contact pressure and the sliding speed.

2. THE BRAKING SURFACES THERMAL REGIME

During the braking process the contact surface of the shoes and drum, presents roughness, undulations, indents. This fact implies the decrease of the real contact area, which it reaches smaller values than nominal contact area. Therefore, during the braking, the specific contact pressure increases very much and it reaches veritable temperature explosions, when the temperature values are in the range of 800-1000°C. The heat quantity, which is generated on the friction surface, is shared between the shoes and drum, and it diffuses in the environment. Because of the extremely low thermal conductibility of the friction material (usually ferrodo type), the most part of the heat quantity is taken by the brake drum.

The result of the cyclic heating and cooling is the thermal stresses and strains appearance, which finally implies the cutting out of function of the brake drum. As it was presented before, the band brake functions simultaneously with the hydraulic brake.

Thus in the final period of the braking the braking moment of the hydraulic brake is null, and the band brake is subjected at a braking moment which is much bigger than the statical moment. In this way, it can say that the effects of the variations of the friction coefficient value are take by the hydraulic brake, and the calculus of the friction surface temperature it can take into consideration an average value of the friction coefficient.

Starting of these facts, it was developed an argument for determine a calculus relation of the drum friction surface temperature when the band brake functions simultaneously with the hydraulic (auxiliary) brake of the draw-works.

The mechanical brake running is characterized by an elementary mechanical work, which is calculated with the relation:

$$dL = l \cdot dF \qquad (11)$$

where dF is the elementary friction force, and l is the displacement, which is made by a point that is considered on the brake drum that is subjected on this force.

For determine the displacement l it is used the following differential movement equation:

$$I \cdot \omega \frac{d\omega}{d\varphi} = M_{st}(a+b) \cdot \varphi - M_h \qquad (12)$$

where: I is the average value of the launching system inertia moment; ω - rotational speed of the hoisting drum axle; φ - rotation angle of the hoisting drum axle during the mechanical braking; a – coefficient which includes statical moment reduction during the drill-pipe stand launching; b – coefficient which characterizes the increase of the hydraulic brake braking moment, and it depends on hoisting drum rotational angle during the hydraulic brake functioning; M_{st} – the statical moment value at the mechanical braking beginning; M_h – braking moment of the hydraulic brake which is calculated with the relation [1]:

$$M_h = k_{mf} \cdot D_h^5 \cdot n^2 = A \cdot \omega^2 \qquad (13)$$

where: k_{mf} is moment capacity coefficient of the hydraulic brake; n – the rotational speed of the brake; D_h – active diameter of the hydraulic brake; A – coefficient which depends on the constructive characteristics of the hydraulic brake.

The coefficients a and b are calculate with the relations:

$$a = \frac{M_{st_i} - M_{st_f}}{\varphi_f} \qquad (14)$$

$$b = \frac{(M_h)_{max}}{\varphi_v} \qquad (15)$$

where: M_{st_i}, M_{st_f} are the static moment values corresponding to the beginning and the end of the drill-pipe stand input; φ_f – rotational angle of the hoisting drum axle during the braking; φ_v - rotational angle of the hoisting drum axle during hydraulic brake running.

If it is resolved the equation (12) using the equation (13) and the initial conditions $\omega|_{\varphi=0} = \omega_h$, it is obtained:

$$\omega^2 = \Phi\left(1 - e^{-B\varphi}\right) - \frac{a+b}{A}\varphi + \omega_h^2 \cdot e^{-B\varphi} \qquad (16)$$

where: $\Phi = \frac{1}{A}\left[M_{st} + \frac{a+b}{2A} I\right]$ and $B = \frac{2A}{I}$; ω_h – the rotational speed of the hoisting drum axle at the control ending of the hydraulic brake.

If it is considered that at the end of mechanical braking the launching instrument speed is, zero, and $\varphi=\varphi_1$, the equation (16) has the form:

$$\Phi - \left(\Phi - \omega_h^2\right) e^{-B\varphi_1} - \frac{a+b}{A}\varphi_1 = 0 \qquad (17)$$

From eq. (17) it results φ_1 which it is used for distance l determination:

$$l = R \cdot \varphi_1 \qquad (18)$$

where R is the drum brake radius.

The friction force is determined with the relation:

$$F = F_1 \cdot \left(e^{\mu\alpha} - 1\right) \qquad (19)$$

where: μ is the friction coefficient; F_1 – the tensile force of the band brake free end (fig. 3).

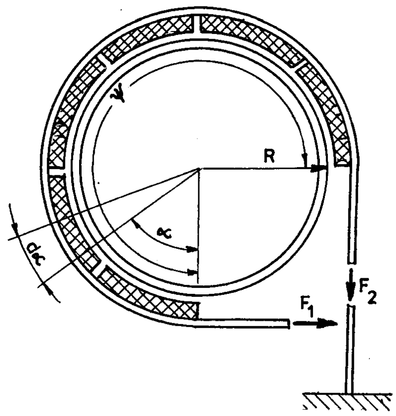

Fig. 3 Calculus model for Euler formula.

The elementary friction force is:

$$dF = \mu \cdot F_1 \cdot e^{\mu\alpha} \cdot d\alpha \qquad (20)$$

Using Euler formula:

$$F_2 = F_1 \cdot e^{\mu\alpha} \qquad (21)$$

where F_2 is the tensile force of the immobile band ending (fig. 3).

The relation (21) can take the form:

$$dF = \mu \cdot F_2 \cdot d\alpha \qquad (22)$$

Replacing in relation (21) the value of l from equation (18) and dF from equation (22) it results:

$$dL = R \cdot \varphi_1 \cdot \mu \cdot F_2 \cdot d\alpha \qquad (23)$$

Because of the friction force on the contact surface during the braking it is generated the heat quantity:

$$dQ = dL \qquad (24)$$

The generated heat it is shared between the friction couple elements in the following way: dQ_1 for drum; dQ_2 for shoe ($dQ=dQ_1+dQ_2$).

The dissipated heat quantities for each couple element are [7]:

$$dQ_1 = \frac{\lambda_1}{\lambda_1 + \lambda_2} dQ \qquad (25)$$

$$dQ_2 = \frac{\lambda_2}{\lambda_1 + \lambda_2} dQ \qquad (26)$$

where λ_1 and λ_2 are the thermal conductivities of the drum and shoes materials.

The friction surface temperature is determined with the relation:

$$dT = \frac{e^{-\frac{r^2}{4a_1 \cdot t}} dQ_1}{4\pi \cdot R \cdot S\sqrt{\pi \cdot \lambda_1 \cdot c_1 \cdot \rho_1 \cdot t}} \qquad (27)$$

where: c_1 is thermal capacity of the brake drum capacity; S – the drum width; ρ_1 – the density of the brake drum material; t – the braking time; a_1 – thermal diffusivity of the brake drum material; r – distance between two points placed on the friction surface, which is calculated with the relation:

$$r = R \cdot \alpha \qquad (28)$$

Replacing in eq. (27) value dQ_1 from relation (26) and eq. (23) it obtains:

$$dT = \frac{\varphi_1 \mu F_2 \lambda_1 e^{-\frac{R^2\alpha^2}{\pi a_1 t}}}{4\pi S\sqrt{\pi\lambda_1 c_1 \rho_1}\left(\lambda_1 + \lambda_2\right)} d\alpha \qquad (29)$$

If it takes account that $\lambda_1 = a_1\rho_1 c_1$ then the relation (29) can be written:

$$dT = \frac{\varphi_1 \mu F_2 c_1 e^{-\frac{R^2\alpha^2}{\pi a_1 t}}}{4\pi S\sqrt{\pi a_1 t}\left(\lambda_1 - \lambda_2\right)} d\alpha \qquad (30)$$

The braking moment M_f works out by the two drums which equip the draw-works, is:

$$M_f \cong M_{st_f} \qquad (31)$$

where M_{st_f} is the final statical moment which corresponding to braking end.

For one brake drum the braking moment M will be:

$$M = \frac{M_{st_f}}{2} = 2RF_1\left(e^{\mu\Psi} - 1\right) \qquad (32)$$

where F_1 is the immobile band brake ending tensile force. If it takes account on relation (21), then:

$$F_2 = \frac{Me^{\mu\Psi}}{2R\left(e^{\mu\Psi} - 1\right)} \qquad (33)$$

The relation (30) can be written like:

$$dT = \frac{M\varphi_1 \mu a_1 e^{\mu\Psi} e^{-\frac{R^2\alpha^2}{4a_1 t}}}{8\pi RS\sqrt{\pi a_1 t}\left(\lambda_1 + \lambda_2\right)\left(e^{\mu\Psi} - 1\right)} \qquad (34)$$

If relation (34) is integrated, than it obtains:

$$T = T_0 + \\ + \frac{M\varphi_1 \mu a_1 e^{\mu\Psi}}{8\pi RS\sqrt{\pi a_1 t}(\lambda_1+\lambda_2)\left(e^{\mu\Psi}-1\right)} \int_0^{\Psi} e^{-\frac{R^2\alpha^2}{4a_1 t}} d\alpha \qquad (35)$$

where Ψ is grip hold angle (fig. 4), and T_0 is the environment temperature.

For computing the integral $\int_0^{\Psi} e^{-\frac{R^2\alpha^2}{4a_1 t}} d\alpha$, it is used the substitution $z = \frac{R\alpha}{2a_1 t}$, and it obtains

$$\int_0^{\Psi} e^{-\frac{R^2\alpha^2}{4a_1 t}} d\alpha = \frac{\sqrt{2a_1 t}}{R} \int_0^{\frac{R\Psi}{2a_1 t}} e^{-\frac{z^2}{2}} dz \qquad (36)$$

When the Gauss integral (36) is calculated it is considered usually its values corresponding to maximum argument $z = 3$,

$$\int_0^{\frac{R\Psi}{2a_1 t}} e^{-\frac{z^2}{2}} dz = \int_0^{\infty} e^{-\frac{z^2}{2}} dz = \sqrt{\frac{\pi}{2}} \qquad (37)$$

In this case:

$$\int_0^{\Psi} e^{-\frac{R^2\alpha^2}{4a_1 t}} d\alpha = \frac{\sqrt{\pi a_1 t}}{R} \qquad (38)$$

If the value of integral (38) is substituted in equation (35), then it obtains:

$$T = T_0 + \frac{M\varphi_1 \mu a_1}{8\pi R^2 S(\lambda_1+\lambda_2)} \frac{e^{\mu\Psi}}{\left(e^{\mu\Psi}-1\right)} \qquad (39)$$

The relation (39) permits friction surface temperature determination for the mechanical brake when it functions simultaneously with the hydraulic brake.

3. NUMERICAL APPLICATION

It consider a drilling rig F 320E type, which is equipped with a draw-works T 38 type, and hydraulic brake FH 560 type that has the braking moment M_h = 17000 Nm.

The total length of the drilling pipes column is 6000 m, and its structure is: 2000 m drilling pipes with 140 mm diameter; 4000 m drilling pipes with 115 mm diameter.

The wire number of traveling block is 12; the production line diameter is 31mm; the drilling mud density – 1200Kg/m^3; the friction coefficient of the mechanical brake – 0.35; the grip hold angle of the brake band – 325^0; the drill-pipe stand length – 37.5m; the maximum force at the band brake lever – 50daN; the braking shoes material is WM 17-15; the drum brake material is T35Mn14 steel.

The thermal conductivities of the braking couples materials are for the brake drum λ_1=45.3 W/m^{o}C, and for the shoe λ_2=0.66 W/m^{o}C; the rotation angle of the hoisting drum axle during the hydraulic braking, φ_v = 180^o.

Starting from the above-presented characteristics, and using the relation (39) it was calculated the temperature on the friction surface. It has to be mentioned the fact that the presented characteristics were used at the calculus of the parameters which are in the relation (39) composition, and, so, indirectly at the temperature on the friction surface.

In Table 1 there are shown the temperature values reach on the drum-braking surface, which was calculated depending on the braking system parameters.

Table 1 Temperature values reach on the braking surface of the brake drum calculated for real conditions with relation (39).

Para-meter	Number of drill-pipe stands launched in well				
	80	100	120	140	160
I	190	194	197.5	201.1	204.7
F_t	7145	8309	9473	10637	11801
M_{sti}	4001	4653	5305	5957	6609
M_{stf}	3172	3689	4206	4723	5240
ω_h	34.93	38.47	41.96	45.49	49.05
φ_I	97	110	122.8	136.8	149
T	340	440	550	690	830

The significance of the parameters is: I – average moment of inertia of the launching system, daNm/s^2; F_t – hoisting drum line load, daN; M_{sti} – initial static moment, daN; M_{stf} – final static moment, daN; ω_h – hoisting drum axle rotation speed at the final of hydraulic brake running, 1/s; φ_I – hoisting drum axle rotation angle at the braking end, rad; T – temperature on the brake drum braking surface at the braking end, ^{o}C.

4. CONCLUSIONS

The band brakes, which equip the drilling draw-works, are subjected during the braking to high thermal stresses. Because of the statical moments, which have very high value during the rig operations, the brake drums get hot very much (Table 1). This fact imposes their water cooling. It has to be mentioned the fact that almost entirely of heat quantity (approximately 98%) [2, 3] is taken by the brake drum.

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The brake drum material is subjected to the heat shocks, which are characterized by temperature peaks that reach usually 800^{o}C, even in the situation considered "normal" when the mechanical brake functions simultaneously with the hydraulic brake.

The brake drums durability is influenced in a decisive way by the cyclic thermal stresses, which appear during the mechanical brakes running. The cyclic or quasi-cyclic of the heatings and coolings is determined by the fact that at the rig operations the differences between the drill-pipe column weights launching in well, from a drill-pipe stand to another, can be considered insignificant. The forced cooling of the brake drums implies that every braking cycle is characterized to the same brake drum temperature.

The relation (39) can be used both in design and exploitation calculus of the brake drums.

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