THE VERIFICATION BRAKE MECHANISM OF WINDING MACHINES WITH SINGLE CABLE DRIVING WHEELS ON

PhD, Eng., Assoc. Lecturer, Răzvan Bogdan ITU, Department of Industrial Mechanical Engineering and Transport, University of Petroșani, raz.van4u@yahoo.com
PhD, Eng, Lecturer, Vilhelm ITU, Department of Industrial Mechanical Engineering and Transport, University of Petroşani, drituv@yahoo.com

Abstract: The development in safe conditions of the extracting process continuously imposes the need of optimal functioning of the extracting installations as important links in the transport flow. Diagnosis of winding engine brake mechanism in mines is important to provide normal extraction vessel movement in the shaft, or stopping machines in a certain position of the vessels in disturbances or failures. The paper presents the calculus of safety coefficients in the use of safety and maneuver brakes. Mine winding engines brake mechanisms is important to provide normal extraction vessel movement along the shaft, or stopping the engine in a certain position of the vessel in disturbances or failures. To assess the real safety coefficient, results obtained by tensiometric measurements were used. After diagnosis, necessary information is obtained to improve present maintenance system and repair this category of machines in view of increasing safety in use of winding installations, with possibility of monitoring brake mechanism...

Keywords: Diagnosis, Braking mechanism, Winding engine, Hoisting installation;

INTRODUCTION

The fundamental elements of an extraction installation placed on the mining surface are: the extraction tower, the countrafort, the extraction pulleys, the extraction cable, the extraction vessels and the extraction-machine consisting of the wrapping device of the cable, the reducing-gear and the action engine. If the installation is meant for a blind shaft, the extraction vessels are lifted from the inferior ramp of the lower level to the ramp level of a superior level. The upper part of the shaft, over the ramp of the superior level, has the role of a winding tower.

The development in safe conditions of the extracting process continuously imposes the need of optimal functioning of the extracting installations as important links in the transport flow. Every extraction machine is foreseen with a stop–gear whide ensure the right movement of the extraction vessels, or allows to stop the machine in a certain position of the vessels (brake tests) and the automatic stopping machine, independently of the operator will, in one of the situations, considered to be perturbations or damages: tension absence, pressure drop of the working fluid in the braking system circuit, the overraising of the extraction vessels, exceeding the limit speed, overload etc. (safety–braking) [1].

Speed decrease made by the brake system must be between 1,5–5 m/s2 and the delay length of the brake (from the action release till the effective application) at the most 0,7 s.

THE WORKING MECHANISM OF IMPLEMENTATION

Actuation of braking mechanisms for maneuver braking and for safety braking is done by independent devices. Safety braking may be willingly performed by the operator and is accompanied by decoupling the actuating engine from the mains, in view of a rapid and safe stop of the engine, while maneuver braking is adjustable and provides a real braking momentum, function of the working forces.

The braking mechanism is designed in two parts: the execution mechanism and the actuation mechanism.

Depending on the execution mechanism, the common design brakes can be with band and with shoes, and actuation wise, through weights and arches (Fig. 1), pneumatic (Fig. 2), hydraulic (Fig. 3) and combined.

Fig. 1. NKMZ 2x2, winding engine

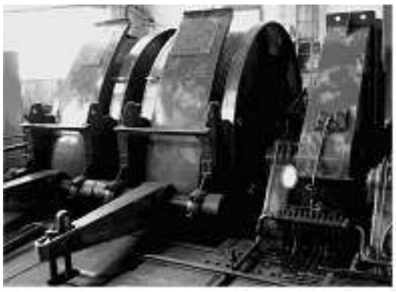

Fig. 2. BAMERT winding engine

Fig. 3. 2 BM 3000x900 winding engine

Brake execution mechanisms are made up of braking shoes and transmission by levers connecting shoes to the actuation device. Execution mechanisms are common (in most of the cases) for maneuver braking and safety braking. When execution parts of the brakes are two shoes, their movement and transmission of forces to those is done by various kinematical schemes.

In case of drum shaped winding parts, braking rims are on their inner sides (Fig. 3) or on their outer sides. Braking rims are located on the inner sides of the drums when both pairs of shoes have the same actuating device. They are located on the outside in case of simple drums, or in case of large winding engines, when each pair of shoe is individually actuated. With mono-cable driving wheels (Fig. 4), braking rims are located on the two sides of the cable channel. With multi-cable wheels, braking rims can be located on the outer sides or between the channels of the cables, when several pairs of shoes will be used.

Depending on the suspension of the shoes, executing part are of two types: with shoes fixed on supports of fixed seats and with shoes on supports of articulated seats.

Fig. 4. DEMAG winding engine

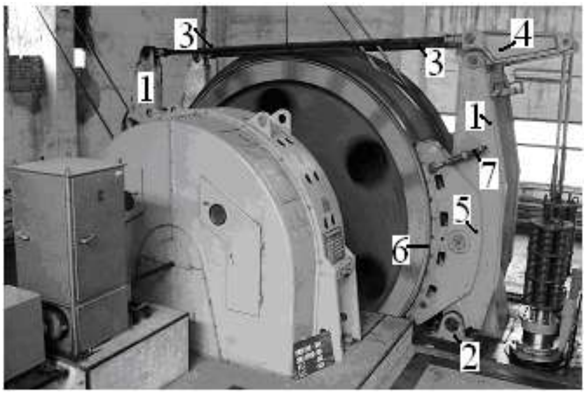

Fig. 5. MK 2,25 x 4 type winding engine

Constructively, the brake system consists of two components: the working mechanism and the actuating system. Upon the working system, the common brakes can be with disk or with shoes, and from the point of view of actuation, can be with weights and, spring assembly (Fig. 2), pneumatics, hydraulics and combined.

The working mechanism of the brakes with shoes and levers (Fig. 5) consists of two support beams (1), articulated in joints (2) connected each other through the rod (3) actuated by raising or lowering the lever (4). On the support bars there are fixed the supports (5) of the brake shoes (rigid in case of angular movement and articulated in case of parallel motion).

On the inner side surface of the supports the shoes are fixed (6) whit action straight about the brake system. The shoes motion during the braking time is stopped by the joints (7) at the ends of the supports (5).

WINDING INSTALLATION IN STUDY

Winding installation (Fig. 6) is intended [4] to underground extraction. Extraction depth is 566 m. Shaft winding installation is balanced (with balance cable) and is equipped with an K 6080-SR29A type winding engine (Fig. 7). Maximum static charge is 214.900 N, maximum balancing charge 42.280 N.

The compensation cables' weight (per linear meter) is 12,5 kg/m and 6,8 kg/m. Φ 55 mm diameter and 11,05 kg/m weight (per linear meter) winding cables wind over the two Φ 6000 mm extraction pulleys, at a 34,30 m and 27,30 m height (pulley axes). The extracting vessels are untipping cages with trei level, with two trolleys per isch level having a mass (own mass plus D.L.C.-3) of 6500 kg. The mass of a trolley is of 435 kg, and the effective load is 1067 kg/trolley. Maximum braking momentum is 555.800 Nm. Another main component of the winding installation is the 55,75 m high tower (Fig. 6).

Fig. 6. Winding installation

Fig. 7. K 6080-SR29A type winding engine

EXPERIMENTAL VERIFICATION OF EFFECTIVE FORCES IN TIE BARS

To determine the stretching forces in tie bars (Figs. 8 and 9) two tensiometric marks were glued together on each tie bar (Fig. 5), diametrically opposite, to remove the bending effect , and with two additional compensation marks a Wheatstone bridge was created with two active and two passive branches [2], [3] The Wheatstone bridge was balanced with a compensator, in various states of the brake, and specific deformation of the material was determined. MM-SUA made EA-06-250BG-120 type tensiometric marks were applied, nominal resistance 120 ohms, actual sensitivity factor 2,06 and SPIDER 8 type measuring amplifier.

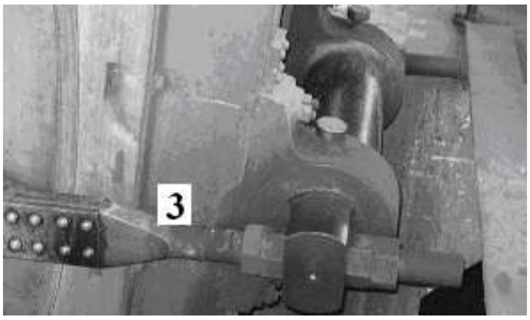

Fig. 8. Left tie bar

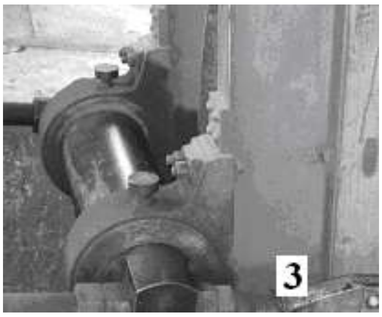

Fig. 9. Right tie bar

Measurements were effected in a static regime to determine absolute magnitudes. To find the dynamics of the phenomena, output signal from the amplifier was recorded with a data acquisition system. Tie bar forces magnitudes, with which safety coefficients were calculated, found as a result of measurements performed during an extraction cycle, together with kinematical movement of vessels in the shaft, are given in Figs. 10 and 11 [2].

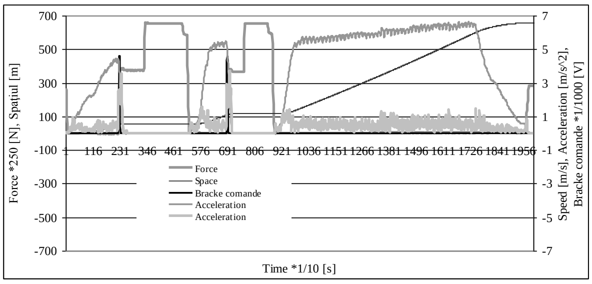

Fig. 10. Left tyrant, left cage goes downwards

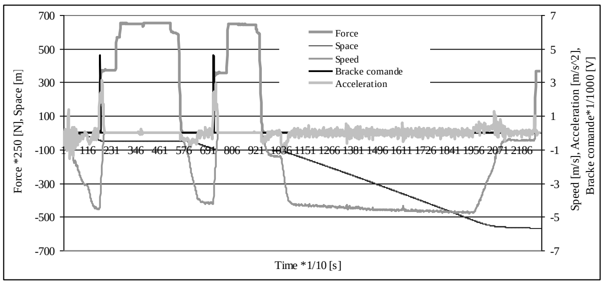

Fig. 11. Left tyrant, right cage rises

DETERMINATION OF BRAKING MOMENTUMS AND SAFETY COEFFICIENTS

Left brake	î – Closed	Right brake
$F_{sî}$		$F_{dî}$
$\Delta mV_s = 0{,}085$		$\Delta mV_d = 0{,}085$
$FM_{î}$.		$FM_{î}$.

ε – Specific measured deformation µm/m

$$\varepsilon_s = \frac{4000\Delta mV_s}{2.2{,}06} \qquad \varepsilon_d = \frac{4000\Delta mV_d}{2.2{,}06} \tag{1}$$

Stretching forces in tie bars F (daN)

$$F_s = \varepsilon_s.E_o.S_s.10^{-6} \qquad F_s = 1{,}17x10^4 \tag{2}$$

$$F_d = \varepsilon_d.E_o.S_d.10^{-6} \qquad F_d = 1,17x10^4 \qquad (3)$$

D_j – Diameter of braking rim (m) $D_j = 6$
i_2 – Partial amplification ratio (post shoe holder) $i_2 = 2,5$
Left brake braking momentum (daNm)

$$M_{Fs} = \frac{F_s.i_2 D_j 2.\mu.\eta.}{2} \qquad M_{Fs} = 1,378x10^4 \qquad (4)$$

Right brake braking momentum (daNm)

$$M_{Fd} = \frac{F_d i_2 D_j 2\mu.\eta}{2} \qquad M_{Fd} = 1,378x10^4 \qquad (5)$$

Total braking momentum: M_t (daN•m)

$$M_t = M_{Fs} + M_{Fd} \qquad M_t = 2,756x104$$

c_s – Experimentally determined actual safety coefficient
M_{st1} – Maximum static momentum (daNm) $M_{st1} = 3,046$

$$c_s = \frac{M_t}{M_{st1}} \qquad c_s = 4,14 \qquad (6)$$

CALCULATION OF CRITICAL DECELERATIONS IN CABLE SLIDING. CALCULATION OF THEORETICAL CRITICAL DECELERATIONS FOR NON SLIP OF THE CABLE.

Tensions in the cable (daN):
- loaded branch:

$$Q_1 = 21490 \qquad T_1 = Q_1 \qquad (7)$$

- unloaded branch:

$$Q_2 = 16660 \qquad T_2 = Q_2 \qquad (8)$$

Angle of cable winding α (rad):

$$\alpha = \pi \qquad (9)$$

Critical decelerations (m/s^2):
- load lifting

friction coefficient between cable and lining: μ = 0,3

$$k_{st} = \frac{T_1}{T_2} \qquad kst = 1,29 \qquad (10)$$

$$a_{crr} = g\frac{k_{st}.e^{\mu.\alpha} \quad 1}{k_{st}.e^{\mu.\alpha} + 1} \qquad a_{crr} = 5,782\ m/s^2 \qquad (11)$$

- maximum admitted deceleration (m/s^2):

$$a_{maxr} = 0,85.a_{crr} \qquad a_{maxr} = 4,915m/sec^2 \qquad (12)$$

- load lowering

$$a_{crc} = g\frac{e^{\mu.\alpha} \quad k_{st}}{e^{\mu.\alpha} + k_{st}} \qquad a_{crc} = 3,913\ m/sec^2 \qquad (13)$$

- maximum admitted deceleration (m/s^2)

$$a_{maxc} = 0,85 \bullet a_{\ crc} \qquad a_{crc} = 3,326 \qquad (14)$$

- empty skips: kst = 1

$$a_{crr} = g\frac{k_{st}.e^{\mu.\alpha} - 1}{k_{st}.e^{\mu.\alpha} + 1} \quad a_{crr} = 4,512\ m/sec^2 \tag{15}$$

$$a_{maxr} = 0,85 \bullet a_{crr} \quad a_{maxr} = 3,835 m/sec^2 \tag{16}$$

Table 1. Magnitudes of actual decelerations in safety braking [2]

Nr. crt.	dQ [daN]	$V_{ef.}$ [m/s]	t_1 [s]	t_2 [s]	t_3 [s]	t_m [s]	a [m/s^2]
1	0	8,57	22,4	22,9	25,8	0,5	2,96
2	0	8,5	48,9	49,4	52,3	0,5	2,93
3	0	8,46	24,8	25,5	28,3	0,7	3,02
4	0	8,47	49,2	49,6	52,4	0,4	3,02

VERIFICATION OF THE WORKING MECHANISM

Due to the safety coefficient that has to be applied to these mechanism system elements it is necessary a more precise determination of the tension from their elements. In order to realise an analysis with finit element of the cable connecting device, a 3D geometrical modelling of it was needed. Modelling the elements of the device where made due to the Solid Edge software, and the analysis with finite elements was made with the COSMOS Design STAR software, as it follows (Figs. 12, 13 and 14):

Fig. 12. Left support beams with support of the brake shoes

Fig.13. Right support beams with support of the brake shoes

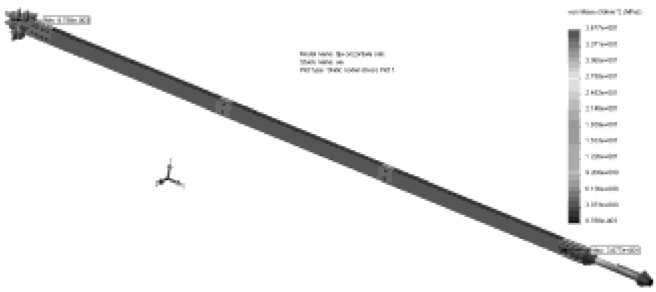

Fig. 14. The tie bar

CONCLUSIONS

The real safety factor calculated with the effective force from the tyrant, obtained as followed the experimental measurement results, when the safety–brake was applied, and the operating–brake was also applied, is according to the admitted limits. Deceleration/speed–reducing, delay–times and dead–times at the application of the safety–brake and operating–brake, have been in accordance with the calculated values from the real ones [tahograms] recorded after the measurements performed in admitted limits. After diagnosis, necessary information is obtained to improve present maintenance system and repair this category of machines in view of increasing safety in use of winding installations, with possibility of monitoring brake mechanism..

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