On diagnosis of brake mechanism of hoisting machines

VILHELM ITU, MIHAI CARMELO RIDZI, IOSIF DUMITRESCU
Department of Mechanics and Machine Design
University of Petroșani
University 20-22, Petroșani 332006
ROMANIA

Abstract: - 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 theoretical and real safety coefficients in the use of safety and maneuver brakes and the numerical analysis of the components of brake mechanism. Experimental measurements were made in Shaft with skip in Livezeni Mine in view of examination and adjustment of the winding engine.

Key-Words: - hoisting installation, hoisting machines, diagnosis brake mechanism

1 Introduction

The fundamental elements of hoisting installation placed on the mining surface (Fig.1) are: the shaft tower 1; the counterfort 2; the pulleys 6; the rope 7; the hoisting vessels 8 and the winding machine consisting of the wrapping devices of the rope 3 (in given case the drums); the reducing-gearbox 4 and the engine 5.

The hoisting facility works as follows: when the wrapping-device is actuated by the engine, one branch of the rope is wrapped up on the drum raising the loaded hoisting vessel from underground, and the other branch unfolding itself, the empty extraction vessel hanging on this rope goes downwards [3].

The two hoisting vessels reaching the level ramps, the loading and unloading operations are taking place simultaneously and after that the entire cycle is repeated.

2 Problem Formulation

Every extraction machine is endowed with a braking system (Fig.2) while ensure the right movement of the hoisting vessels, or allows to stop the machine in a certain position of the vessels (brake tests) and the automatic brake device, independently of the operator will, in one of the following 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 ) [3].

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.

2.1 The working mechanism

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.2) 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) [3].

Fig.2. Braking–device extraction – machine with wheel–engine, type „MK 5x2 Shaft skip” Petrila Mining Plant.

2.2 Operating conditions required for the braking device

Braking momentums, both for manoeuvre and for safety braking should be at least three times the static momentum:

\[ M_{fr} \geq 3M_{st} \ [Nm] \],

(1)

In case of an unbalanced winding engines (no compensation cable (balance)), static momentum is:

\[ M_{st} = g (Q_u + qH) R \ [Nm] \],

(2)

Where \( g \) is gravitational acceleration, \( g = 9.81 [m/s^2] \); \( Q_u \) useful mass of extraction vessel [kg], \( q \) weight per linear meter of extraction cable [kg/m], \( H \) extraction depth [m]; \( R \) is radius of the winding part [m].

For a statically or dynamically balanced installation (with compensation cable):

\[ M_{st} = g [Q_u + (q - q_1) H] R \ [Nm] \],

(3)

where \( q_1 \) is mass per linear meter of compensation cable, kg/m. In case of adjusting drum position as to another, in changing the hoisting level,

\[ M_{fr} \geq 1.2M_{st} \ [Nm] \],

(4)

braking momentum will be developed on the fixed drum rim, where \( M_{st} \) is static momentum of a cable branch, generated by the weight of the empty extraction vessel and the extraction cable,

\[ M_{1st} = g (Q_u + qH) R \ [Nm] \],

(5)

where \( Q_u \) is mass of the empty extraction vessel, kg.

Maximum distance between shoes and braking rim should be no more than 2 mm. A deceleration of at least 1.5 m/s\(^2\) and at most 4-5 m/s\(^2\) is also required during braking, but the critical magnitude when driving wheel winding installation cables slide shall not be exceeded.

As mentioned above, movement of the cable and of the load is initiated due to the force of friction between the cable and the driving wheel. Therefore, the deeper the extraction, the higher the safety for cable sliding on the driving wheel.

To avoid cable sliding on the driving wheel, a certain ratio of the cable winding and unwinding tensions will be maintained. In this sense, both starting acceleration and delay in case of braking are limited.

To appraise the cable sliding possibility, companies apply the ratio:

\[ k_{st} = \frac{T_u}{T_d} \],

(6)

where:

\( T_u \) is static tension in the cable branches going up, [N];
\( T_d \) is static tension in the cable branches going down, [N].

Critical deceleration, depending on the installation working regime, is determined by the formulae:

For installations with no deviation pulley:

\[ a_{st} = g \frac{k_{st}e^{mu} - 1}{k_{st}e^{mu} + 1} \ [m/s^2] \],

(7)

when loads are going down:

\[ a_{st} = g \frac{e^{mu} - k_{st}}{e^{mu} + k_{st}} \ [m/s^2] \],

(8)

- for installations with deviation pulley:

  when lifting the load (load on the opposite branch of the deviation pulley position):

\[ a_{st} = g \frac{k_{st}e^{mu} - 1}{k_{st}e^{mu} + 1 + \frac{M_{st} + g}{T_d}} \ [m/s^2] \],

(9)

when lifting the load (load on the side of the deviation pulley):

\[ a_{st} = g \frac{k_{st}e^{mu} - 1}{e^{mu}(k_{st} + \frac{M_{st} + g}{T_d}) + 1} \ [m/s^2] \],

(10)

when load goes down (load on the opposite branch of the deviation pulley position):

\[ a_{st} = g \frac{e^{mu} - k_{st}}{e^{mu}(1 + \frac{M_{st} + g}{T_d}) + k_{st}} \ [m/s^2] \],

(11)

when load goes down (load on the side of the deviation pulley):
\[ a_{3cr} = g \frac{e^{\mu a} - k_{st}}{e^{\mu a} + k_{st} + \frac{M_m}{T_d} g} \text{ [m/s}^2]\], \quad (12)

where \( M_m \) is mass of deviation pulleys.

Working deceleration is adopted based on the formula [3]:
\[ a_3 = (0.7 - 0.8) \ a_{3cr} \text{ [m/s}^2]\]. \quad (13)

2.3 The installation taken into study

Winding installation of Livezeni Mine skip shaft is intended [4] to underground extraction. Extraction depth is 361 m. Shaft kip elevations (skip charge and discharge) is 260.1 and 6585, respectively. Shaft winding installation is balanced (with balance cable) and is equipped with an MK 2.25 x 4 type winding engine (Fig.3), made in the former USSR and had been commissioned in the year 1971. The Engine is fitted with two asynchronous AKH-2-16-39-12YXN4 type motors of 500 kW nominal power and 490 rpm nominal rotation speed and a 2TD-14 type reduction gear of 1:11.5 transmission ratio. Friction drum diameter (actuation) is 2.25 m.

The four winding cables of \( \Phi \) 40 mm maximum admitted diameter and weight (per linear meter) of 3.22 kg/m go over the \( \Phi \) 2.25 m driving wheel (Fig.3), the other cable ends being caught by the extraction vessels by the cable balancing device (CBD) and cable linking device(CLD). Extraction vessels are four 8000 kg extraction cables skips. Extraction vessels are fastened to cables by CBDs and four CLDs.

Maximum static charge is 277400 N, maximum balancing charge 72170 N. The two compensation cables’ weight (per linear meter) is 5.447 kg/m. Maximum speed is 7.85 m/s. Braking time with going up full skip at 7.85 m/s speed is 2.44 s, and braking space is 7 m.

Maximum braking momentum is 269000 Nm.

Adjustable distance between \( \Phi \)2.4 m braking disk and shoes is 1.2÷1.5 mm.

Another main component of the winding installation is the 68.708 m high tower (Fig.4), to the shaft of the tower’s winding engine [6].

3 Problem Solution

3.1 Experimental verification of effective forces in tie bars

To determine the stretching forces in tie bars (Figs.5 and 6) two tensiometric marks were glued together on each tie bar (Fig.7), 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 [1],[2].

The Wheatstone bridge was balanced with a compensator, in various states of the brake, and specific deformation of the material was determined [1],[2], [4].
MM-SUA made EA-06-250BG-120 type tension meters marks were applied, nominal resistance 120 ohms, actual sensitivity factor 2.06 and SPIDER 8 type measuring amplifier.

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. 8 and 9 [3].
3.2 Determination of braking momentums and safety coefficients

Indication of compensator:
Left brake \( \Delta m V_s / V = 0.118 \),
Right brake \( \Delta m V_d / V = 0.118 \).

\[ e_s = \frac{4000 \Delta m V_s}{2 \times 2.06}, \quad e_d = \frac{4000 \Delta m V_d}{2 \times 2.06}. \]  

Table 1. Magnitudes of actual decelerations in safety braking.

<table>
<thead>
<tr>
<th>Nr.</th>
<th>( dQ ) [N]</th>
<th>( V_e ) [m/s]</th>
<th>( t_1 ) [s]</th>
<th>( t_2 ) [s]</th>
<th>( t_3 ) [s]</th>
<th>( c_t ) [s]</th>
<th>( a ) [m/s²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
<td>8</td>
<td>27.4</td>
<td>28</td>
<td>30.8</td>
<td>0.6</td>
<td>2.86</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>7.66</td>
<td>19.6</td>
<td>20.2</td>
<td>23</td>
<td>0.6</td>
<td>2.74</td>
</tr>
<tr>
<td>3</td>
<td>0</td>
<td>6.72</td>
<td>26.6</td>
<td>27.2</td>
<td>29.8</td>
<td>0.6</td>
<td>2.58</td>
</tr>
<tr>
<td>4</td>
<td>0</td>
<td>8</td>
<td>24.6</td>
<td>25.2</td>
<td>27.8</td>
<td>0.6</td>
<td>3.08</td>
</tr>
</tbody>
</table>

Actual decelerations according to table 1 are lower than those admitted [3].

3.4 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 stress from their elements.

\[ k_s = \frac{T_i}{T_2}, \quad k_s = 1.352, \]  

- maximum admitted deceleration \([\text{m/s}^2]\):
\[ a_{\text{max}} = 0.85 a_{\text{cr}}, \quad a_{\text{cr}} = 4.605 \text{ m/s}^2, \]  

- load lowering
\[ a_{\text{cr}} = g \frac{e^{\mu_1} k_d}{\mu_1 + k_d} a_{\text{cr}} = 3.04 \text{ m/s}^2, \]  

- maximum admitted deceleration \([\text{m/s}^2]\)
\[ a_{\text{max}} = 0.85 a_{\text{cr}} a_{\text{cr}} = 2.584 \text{ m/s}^2, \]  

- empty skips: \( k_d = 1 \)
\[ a_{\text{cr}} = g \frac{e^{\mu_1} k_d}{k_d e^{\mu_1} + 1} a_{\text{cr}} = 4.307 \text{ m/s}^2, \]  

\[ a_{\text{max}} = 0.85 a_{\text{cr}} a_{\text{max}} = 3.661 \text{ m/s}^2. \]
In order to realize an analysis with finite element of the cable connecting device, a 3D geometrical modelling of it was needed. Modelling the elements of the device were made due to the Solid Edge software, and the analysis with finite elements was made with the COSMOS Design STAR software, as it follows: lever (Fig.10), tie bar (Fig.11) and support beam with support of the brake shoes (Fig.12).

4 Conclusion

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.

With the experimental determinated real load, was computed all the forces necessary for Finite Element Analysis.

Acknowledgements. Supports for this work by the SC TECHNOSAM S.R.L. Satu Mare are gratefully acknowledged.

References:
[6] * * * Technical documentation, Lonea Mining Plant 2008.