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DYNAMIC LOADING OF MINING HOIST ELEMENTS IN THE CONDITION OF EMERGENCY BRAKING

The advance of technical state criteria for elements of mining hoists demands a basic strength-fatigue analysis where the real values of loads and the real time function of the load variability could be used. That problem concerns also the suspension gear of skip and balance ropes, where fatigue durability should be considered as time function related to the hoist facility type. Such objective can not be achieved without comprehensive study of the dynamics of processes both in the regular operation of the facility and in its emergency states. In this work the author presents some considerations, that are however, limited to the analysis of dynamic phenomena observed in the condition of the emergency braking of the hoist facility. The results were verified by load measurements taken for some elements of the analysed real object system.

1. Introduction

Supporting assemblies of the mining hoist (suspensions, main shank, head, bottom frame) are spatial structures consisting of beams, shields and plates designed in complex forms, having many, mostly circular, passages where ropes and guiding elements are fixed. Determination of the stress distribution and concentration coefficients in these elements is a complex problem of elasticity theory. According to the legal mining regulations [11] dimensions of elements are determined by the method of allowable stresses with regard to the maximal operational static load.

Due to this method, stresses are calculated for characteristic sections on the basis of relations known as the fundamental equations of strength of materials, and just compared with permissible stresses. As usual, the simplest calculation schemes of structures (tie bars, free supported beams) are assumed.

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In such circumstances, when the information on the stress state in the mining hoist elements is far from satisfactory, the prediction of the safe operational period seems to be highly uncertain, even impossible, neither in the design stage nor in the operational use.

The following considerations are required for application of the finite elements method and the method of boundary states for dimensioning and estimation of the structure safety as well as application of fatigue strength methods for the determination of its safe operation period:

1. Analysis of dynamic operation of suspensions for every possible operational and emergency state.
2. Determination of optimal loads of suspension elements as a function of the design and operational parameters of the system.
3. Strength – fatigue analysis of suspension elements as their operational time function relevant to the type of the hoist facility.

Results of the analysis of problems shown in points 1÷3 will form the basis for "Elaboration of criteria for the estimation of technological condition of suspensions with special regard to the fatigue durability as the function of operational use period and the type of hoist facility".

Real values of the loads in some elements of suspension gears were obtained from a dynamic analysis of the regular operation of the facility [6], [7], [10], in the emergency braking [7] and in the condition of operational braking. Results of this analysis have been verified by the measurement of loads in the skip suspension gear. However, in this paper, only emergency braking is considered.

Presently, the tower multi-rope hoist facilities with guiding wheels are in common use, thus further considerations will be limited to this typical winding gear used in Poland.

Figure 1 shows the scheme of the analysed machinery.

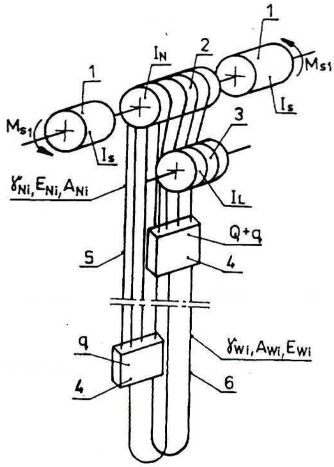

Fig.1. Scheme of the hoist facility

The technological structure of the presented hoist consists of the following components (the abbreviation: *M.I.* – denotes the Moment of Inertia of an element or assembly):

1. slow running DC motors with *M.I.* of the armature – I_s,
2. multi-rope driving wheel of diameter D and M.I. – I_N,
3. the set of deflecting wheels of *M.I.* – I_L,
4. skip tubs of mass q and the load capacity Q with filled upper tub,
5. parallel lines of winding ropes of linear density γ_N and tension rigidity $A_N E_N$,

6 – parallel lines of balance ropes of linear density γ_W and tension rigidity $A_W E_W$.

The rotors of driving motors and the drive wheel connected by the short rigid shaft are in rotary motion. In the rotary motion are also these parts of hoisting ropes, which at that moment are in contact with the driving wheel. The skip, as well as the hoisting and balance ropes, are in reciprocal motion.

The mechanical model of the hoist has been initially constructed for theoretical consideration of the process of operational braking of the facility, then verified by real time measurements of loads in some elements of the system done on the real object, discussed in subsequent part of the paper.

The fundamental issue for the solution of the considered problem is the selection of the model, which should be as much as possible to adequate to the real conditions.

Some simplifications shown in the analysis presented in [2] and [4] could be also applied in the present model of the hoist. The equivalent mechanical arrangement of the hoist may be considered as a composition of two mechanical systems, because shafts connecting rotors of the motors with the driving wheel are very rigid, and discrete masses of the drive assembly elements and skip are comparable. These systems are:

– drive assembly and the exciting force imposed to the drive wheel, which represents vibration of ropes,
– hoisting ropes, skips and the drive wheel where the reduced moment of inertia of the drive assembly is concentrated, its flexibility (deformability) being neglected.

The separated mechanical subassembly will be called the mechanical system because further considerations do not concern the dynamics of the drive assembly. The system is composed of the following mechanical elements of the hoist: skips with the suspension gears and the material, hoisting and balance ropes, the friction drive wheel with the main shaft of the machine, the rope wheel and the armature of the electric motor. The characteristic feature of the shaft hoists is distribution of mechanical components' masses over the long distance between the winding machine and the vicinity of the shaft's sump. Furthermore, these masses are distributed unevenly along the whole winding length. Rotating masses that significantly participate in the total mass of the hoist are located in the tower, when masses of both lines of hoisting and balance ropes are distributed within the depth of the shaft. Disproportion in the mass

distribution will be highly increased when the loaded mining skip reaches the surface vicinity. Therefore, for the considered case of winding gear and high or medium extraction depth, it would be reasonable to consider ropes as distributed masses and the both tubs as concentrated masses.
On the other hand, some mechanical properties of the assembly e.g. the rigidity of particular elements are considerably different.
Finally, it has been assumed that rotating masses and tubs are ideal rigid bodies, while supporting and balance ropes are the flexible parts of the system.

2. Emergency braking of the mining hoist facility

Emergency braking of the hoist facility has the opinion of highly hazardous operation. Even the loss of frictional coupling between the rope and the drive wheel could happen in this stage [2]. The hoisting rope being in motion is rewinded through the drive wheel, so the lengths of parts of this rope on both sides of the wheel and the relevant parts of the balance rope change. Both the change of ropes lengths as well as its elastic slip affect the virtual values of forces in ropes what complicates both the form of boundary conditions, and mathematical equations [2].
The elastic slip on the drive wheel is included into the boundary conditions. However, it gives an insignificant difference in results, because the change of dynamic force caused by the change of rope length is inversely proportional to the fourth root of the virtual length of the rope [2]. Then, it could be easily neglected due to the fact that the time between the initiation of braking and the growth of the forces in the rope to the maximum value which decides on the upset of the frictional coupling is short. Therefore, each part of the hoisting rope in both sides of the wheel could be considered as an independent element.

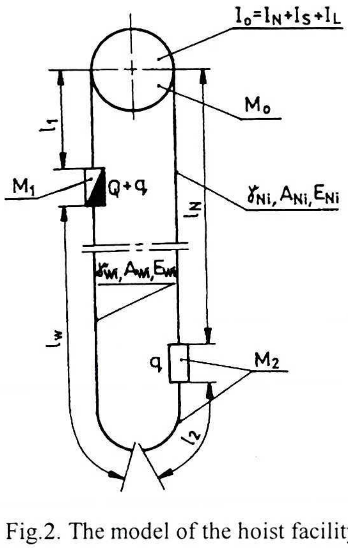

Fig.2. The model of the hoist facility

Fig. 1 shows the case when one tub is near the shaft top and the other is approaching the shaft's bottom (start from the shaft bottom and braking at the approach of the shaft's top) is the most interesting for practical purposes in the operation of the skip tubs. This situation could be represented by the model of the hoist shown in Fig. 2.

We have in the model:

$$M_0 = \frac{G_0}{g}, \; M_i = \frac{1}{g}\left(G_1 + q_1 \cdot l_i\right) \; \left(i = 1, 2\right) \tag{1}$$

where: G_1, G_2, G_0 – weights of tubs and the reduced weigh of rotating elements including the guide wheels. Masses of short parts of ropes l_1 (between the upper tub and the drive wheel) and l_2 (below the bottom tub near the reverse in the sump) are included into the masses of the tubs.

The model shown in Fig. 2 has been simplified in the following way:

- according to the analysis in [2], the drive wheel, rope wheels and armatures of electric motors are considered as a single rigid mass of the moment of inertia $I_0 = I_L + I_N + I_S$ because the short driving shaft is an element of high torsion rigidity,
- both tubs are considered as the rigid bodies, too,
- structural damping in ropes is neglected due to the short time of the operational braking process,
- vibrations from one side do not transfer across the balance rope loop onto the other side; thus, the closed assembly of model masses (Fig. 2) could be separated in this point.

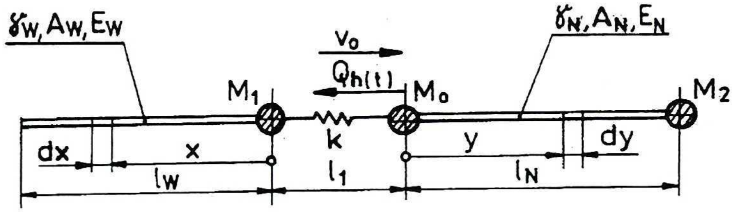

Fig. 3. Straightened model of the hoist facility in the case of emergency braking: $Q_{h(t)}$ – braking force in the drive wheel, M_1 – mass of the tub with the material, M_0 – reduced masses rotating in the tower, M_2 – reduced masses of the hoist in the sump, l_1 – length of supporting ropes between the upper tub and the drive wheel, l_w – length of balance ropes, l_N – length of winding ropes, $A_W E_W$, $A_N E_N$ – tension rigidity of balance ropes and winding ropes, γ_W, γ_N – linear density of balance ropes and winding ropes

After straightening the assembly from Fig. 2, takes the form shown in Fig. 3, i.e. it is thought as one-dimensional inertial system of the finite number of rigid concentrated masses and elastic masses distributed continuously, all of them located along the straight line.

In such a model – due to negligibly small reaction of the short element of the balance rope l_2 from the reverse to the bottom tub – the mass of balance rope is included into the mass of the tub.

$$M_2 = q + \gamma_w \cdot l_2 \tag{2}$$

The model structured in such a way is valid till the total axial force in any cross-section of the rope exceeds zero. Masses M_0 and M_1 are related with the weightless spring of the elasticity coefficient:

$$k = \frac{A_N E_N}{l_1} \tag{3}$$

That coefficient represents the elasticity of the section of hoisting ropes contained between the upper tub and the drive wheel.

The effect of emergency braking of the mining hoist is the result of the action of the force exerted on the drive wheel by the brake assembly (in Fig. 3 it is the force $Q_{h(t)}$, which is applied to the mass M_0).

Determination of displacements and strains in cross-sections of hoisting and balance ropes in the time since the initiation of emergency braking, requires for the solution of the following equations be solved:

$$\begin{aligned} \frac{\partial^2 u(x,t)}{\partial t^2} - a_w^2 \frac{\partial^2 u(x,t)}{\partial x^2} = 0, \\ \frac{\partial^2 v(y,t)}{\partial t^2} - a_N^2 \frac{\partial^2 v(y,t)}{\partial y^2} = 0, \end{aligned} \tag{4}$$

with the boundary conditions:

$$x = 0,\ M_1 \frac{\partial^2 u}{\partial t^2} = A_w E_w \frac{\partial u}{\partial x} - k\left[u(x=0,t) + v(y=0,t)\right], \tag{5a}$$

$$x = l_W,\ \frac{\partial u}{\partial x} = 0 \tag{5b}$$

$$y = 0,\ M \ \frac{\partial^2 v}{\partial t^2} = A_N E_N \frac{\partial v}{\partial y} - k\left[u(x=0,t) + v(y=0,t)\right] - Q_h(t) \tag{5c}$$

$$y = l_N,\ M_2 \frac{\partial^2 \theta v}{\partial t^2} = -A_N E_N \frac{\partial v}{\partial y} \tag{5d}$$

The zero initial condition is assumed.

In the above relations there are: $u(x,t)$, $v(y,t)$ – displacements of any cross-section of ropes within distance x, y (for $t = 0$) from origins of movable co-ordinate systems connected with masses M_1 or M_0; displacements are calculated for systems whose origins at the moment $t=0$ were identical with masses M_1 and M_0 and they move with the velocity $v_0 = const$, which is assumed

to be the same as the velocity of all elements of the hoist at the initial moment; $Q_{h(t)}$ – is the braking force applied to the brake assembly.
It is expected that the solution of equations (3) will take the form:

$$u(x,t) = \varphi\left(t - \frac{x}{a_w}\right) + \psi\left(t + \frac{x}{a_w}\right) \tag{6}$$

$$v(y,t) = f\left(t - \frac{y}{a_N}\right) + g\left(t + \frac{y}{a_N}\right) \tag{7}$$

Relations (6) and (7) were regarded in equations (4) and in boundary conditions (5), then the forms of functions φ, ψ, f, and g for different intervals of variables y,t and x,t were determined. On this basis, one obtained, the general analytical formulae that determine displacements and stresses in hoisting and balance ropes.
Displacements are described by the formulae:
– for the balance rope

$$u(x,t) = q_0\left(t - \frac{x}{a_w}\right) + q_{01} + \sum_{i=1}^{3} q_1 e^{a_i\left(t - \frac{x}{a_w}\right)}, \tag{8}$$

– for the winding rope

$$v(y,t) = W_0\left(t - \frac{y}{a_N}\right) + W_{01} + \sum_{i=1}^{3} W_i e^{a_i\left(t - \frac{y}{a_N}\right)}, \tag{9}$$

where:

$$\overset{*}{q}_{01} = -\frac{M_0 M_1 l_1}{A_N E_N\left(\dfrac{A_N E_N}{a_N} + \dfrac{A_W E_W}{a_W}\right)}, \tag{10}$$

$$\overset{*}{q}_{02} = \frac{-M_0 L_1}{A_N E_N\left(\dfrac{A_N E_N}{a_N} + \dfrac{A_W E_W}{a_N}\right)}\left[\frac{1}{a_0} + \frac{M_0 + M_1 + \dfrac{A_W E_W}{a_N a_W} L_1}{\dfrac{A_N E_N}{a_N} + \dfrac{A_W E_W}{a_W}}\right],$$

$$\overset{*}{q}_0 = \frac{-1}{a_0(a_0^3 - Aa_0^2 + Ba_0 - D)}, \tag{11}$$

$$\overset{*}{q}_i = \frac{a_0(a_i - a_i)}{a_i^2(a_i + a_0)(a_i - a_1)(a_i - a_2)(a_i - a_3)} \quad (i = 1,2,3),$$

a_i (i=1,2,3) – are the roots of the equation,

$$a^3 + Aa^2 + Ba + C = 0\,,$$

$$A = \frac{A_N E_N}{M_0 a_N} + \frac{A_W E_W}{M_1 a_W}\,, \tag{12a}$$

$$q_0^* = \frac{-1}{a_0(a_0^3 - Aa_0^2 + Ba_0 - D)}\,, \tag{12b}$$

$$q_i^* = \frac{a_0(a_i - a_i)}{a_i^2(a_i + a_0)(a_i - a_1)(a_i - a_2)(a_i - a_3)} \quad (i = 1,2,3), \tag{13}$$

and

$$W_{01}^* = \frac{-M_0 M_1}{\dfrac{A_N E_N}{a_N} + \dfrac{A_W E_W}{a_N}}\,, \tag{14}$$

$$W_0^* = \frac{M_0 M_1 L_1}{\left(\dfrac{A_N E_N}{a_N} + \dfrac{A_W E_W}{a_W}\right)} \left[\frac{\dfrac{M_0 M_1}{L_1} + \dfrac{A_W E_W}{a_N a_W}}{\dfrac{A_N E_N}{a_N} + \dfrac{A_W E_W}{a_W}} + \frac{1}{a_0 L_1} - \frac{1}{a_W} \frac{A_W E_W}{A_N E_N} \right], \tag{15}$$

$$W_0^* = \frac{M_1 a_0^2 - \dfrac{A_W E_W}{a_W} a_0 + \dfrac{A_N E_N}{L_1}}{a_0\left(a_0^3 - Aa_0^2 + Ba_0 - C\right)}\,, \tag{16}$$

$$W_i^* = \frac{-a_0\left(M_1 a_i^2 + \dfrac{A_W E_W}{a_W} a_i + \dfrac{A_N E_N}{L_1}\right)\left(a_i - a_i\right)}{a_i^2\left(a_i + a_0\right)\left(a_i - a_1\right)\left(a_i - a_2\right)\left(a_i - a_3\right)}\,, \quad \left(i = 1,2,3\right) \tag{17}$$

Analytical formulae (8) and (9) determining displacements in any cross-section of hoisting ropes and balance ropes made it possible to formulate relations that resulted in determination of the load in any cross-section of ropes, including loads of suspensions gears and balance ropes:

a) suspension gear of the loaded skip in the highest position

$$S_{LN}^* = k\left[u\left(x = 0,t\right) + v\left(y = 0,t\right)\right] =$$

$$= \frac{A_N E_N}{L_1}\left\{ \frac{2Q_h A_N E_N}{M_0 M_1 L_1}\left[q_{01}^* t + q_{02}^* + \sum_{i=0}^{3} q_i^* e^{a_i t} \right] + \frac{2Q_h}{M_0 M_1}\left[W_{01}^* t + W_{02}^* + \sum_{i=0}^{3} W_i^* e^{a_i t} \right] \right\}, \tag{18}$$

b) suspension gear of the balance rope

$$S_{L_W} = A_W E_W \frac{\partial u(x=0,t)}{\partial x} = -\frac{2Q_h}{M_0 M_1}\frac{A_N E_N}{L_1}\frac{A_W E_W}{a_W}\left[q_{01}^{*} + \sum_{i=1}^{3} a_i q_i^{*} e^{a_i t}\right], \quad (19)$$

c) suspension gear of the unloaded skip in the lowest position

$$S_{L_{ND}} = A_N E_N \frac{\partial v(y=l_N,t)}{\partial y} = -\frac{2Q_h}{M_0 M_1}\frac{A_N E_N}{a_N}\left[W_{01}^{*} + \sum_{i=0}^{3} a_i W_i^{*} e^{a_i\left(t-\frac{l_N}{a_N}\right)}\right]. \quad (20)$$

The above relations have been derived under the assumption that the braking process is realised by the brake assembly of the dynamic characteristic shown in Fig. 4.

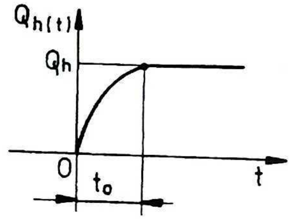

Fig. 4. Characteristic of the braking assembly: Q_h – braking force on the circumference of the drive wheel

A number of measurements taken in the real object [2], as well as measurements of the pressure fluctuations in the brake cylinder, gave good reasons for approximating the curve in Fig. 4 by an equation of the type:

$$Q_{h(t)} = 2Q_h\left(1-e^{-a_0 t}\right) \quad \text{for} \quad t \in (0, t_0), \quad (21a)$$

and

$$Q_{h(t)} = Q_h \quad \text{for} \quad t \in (t_0, \infty). \quad (21b)$$

Fig. 5 shows an exemplary diagram illustrating the change of load of the upper tub suspension (loaded skip approaching the upper loading level) at the stage of emergency braking for different masses of the skip: M_1=30000 [kg], 35000 [kg], 40000 [kg], 45000 [kg], 50000 [kg]. The winding machinery parameters are: M_0 = 40000 [kg], E_W = $1{,}1\cdot10^{11}$ [N/m^2], A_W = 0,0028 [m^2], l_1 = 100 [m]. Braking process is realized by the brake assembly of the characteristic shown in Fig. 4 The brake assembly parameters are: Q_h = 250 [kN], t_0 = 0.8 [S], a_0 = 2.37 [1/s]. Curves in Fig. 5 illustrate the change of the load in the skip suspension gear different values of the ratio $c = M_0/M_1$: 1.33, 1.14, 1.00, 0.89. 0.80.

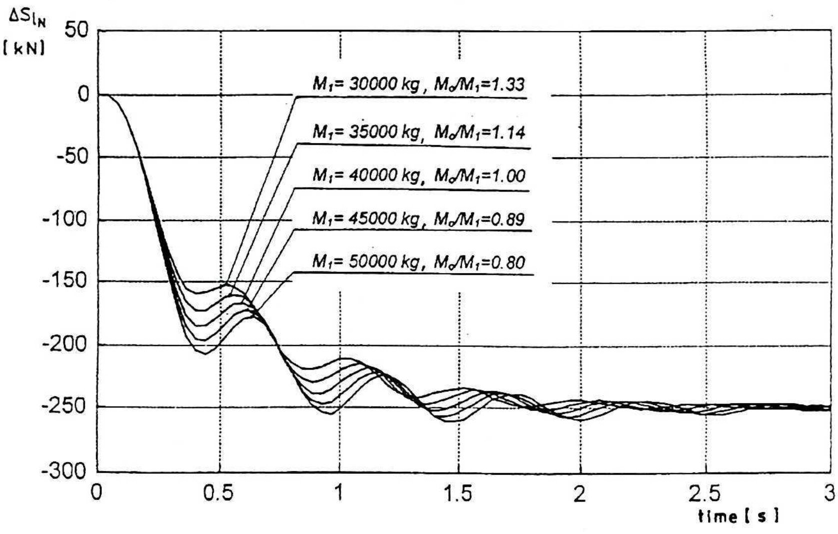

Fig.5. Change of the load of the upper skip suspension gear (loaded) reaching the upper unloading level at emergency braking for a steady braking force: Q_h = 250 kN, a_0 = 2,37 [1/s], t_0 = 0,8[s]

Moreover, the application of simplifying assumptions [2] (valid for the tower hoist type where the experiment have been carried out)

$$\frac{A_W E_W}{a_W} \cong \frac{A_N E_N}{a_N} = \frac{AE}{a}, \quad \text{and} \quad M_0 \cong M_1 = M, \tag{22}$$

results in following solutions:

a) displacement of the cross-sections of balance ropes

$$u^*(x,t) = \frac{2Q_h}{M^2}\frac{AE}{L_1}\left\{\frac{M^2 L_1 a}{2(AE)^2}\left(t - \frac{x}{a_W}\right) - \frac{M^2 L_2}{2(AE)^2}\left[\frac{a}{a_0} + \frac{L_1}{2} + \frac{a^2 M}{AE}\right] - \right.$$

$$- \frac{e^{-a_0\left(t - \frac{x}{a_W}\right)}}{a_0\left\{a_0^3 - 2\frac{AE}{Ma}a_0^2 + \left[\left(\frac{AE}{Ma}\right)^2 + \frac{2AE}{ML_1}\right]a_0 - 2\frac{(AE)^2}{M^2 L_1 a}\right\}} -$$

$$- \frac{a_0 e^{-\frac{AE}{Ma}\left(t - \frac{x}{a_W}\right)}}{2\left(\frac{AE}{Ma}\right)^2\left(\frac{AE}{Ma} - a_0\right)\frac{AE}{ML_1}} - \tag{23}$$

$$
-\frac{a_0\sqrt{2\frac{AE}{ML_1}}e^{-\frac{AE}{2Ma}\left(t-\frac{x}{a_W}\right)}}{4\left(\frac{AE}{ML_1}\right)^2\sqrt{2\frac{AE}{ML_1}-a_0\left(\frac{AE}{Ma}-a_0\right)}\sqrt{2\frac{AE}{ML_1}-\left(\frac{AE}{2Ma}\right)^2}}
$$

$$
\sin\left[\sqrt{2\frac{AE}{ML_1}-\left(\frac{AE}{2Ma}\right)^2}\left(t-\frac{x}{a_W}\right)+\Phi\right]\Bigg\},
$$

$$
\Phi=\frac{\pi}{2}-arctg\left\{\frac{\frac{AE}{2Ma}\left(\frac{AE}{Ma}-a_0\right)-2\frac{AE}{ML_1}}{\left(\frac{AE}{Ma}-a_0\right)\sqrt{2\frac{AE}{ML_1}-\left(\frac{AE}{2Ma}\right)^2}}\right\},
$$

b) displacement of the cross-sections of hoisting ropes

$$
v^*(y,t)=\frac{2Q_h}{M^2}\left\{-\frac{M^2a}{2AE}\left(t-\frac{y}{a_N}\right)+\frac{M^2L_1a}{2AE}\left(\frac{Ma}{L_1AE}-\frac{1}{2a}+\frac{1}{a_0L_1}\right)+1\right.
$$

$$
+\frac{Ma_0^2+AE\left(\frac{1}{L_1}-\frac{a_0}{a}\right)}{a_0\left[a_0^3-2\frac{AE}{Ma}a_0^2+\left[\left(\frac{AE}{Ma}\right)^2+2\frac{AE}{ML_1}\right]a_0-2\left(\frac{AE}{Ma}\right)^2\frac{a}{L_1}\right]}e^{-a_0\left(t-\frac{y}{a_N}\right)}+
$$

$$
-\frac{Ma_0}{2\left(\frac{AE}{Ma}\right)^2\left(\frac{AE}{Ma}-a_0\right)}e^{-\frac{}{a}\left(t-\frac{y}{a_N}\right)}-
$$

$$
-\frac{Ma_0\sqrt{2\frac{AE}{ML_1}}e^{-\frac{AE}{2Ma}\left(t-\frac{y}{a_N}\right)}}{4\frac{AE}{ML_1}\sqrt{2\frac{AE}{ML_1}-a_0\left(\frac{AE}{Ma}-a_0\right)}\sqrt{2\frac{AE}{ML_1}-\left(\frac{AE}{2Ma}\right)^2}}.
$$

$$
\cdot\sin\left[\sqrt{2\frac{AE}{ML_1}-\left(\frac{AE}{2Ma}\right)^2}\left(t-\frac{y}{a_N}\right)+\Phi\right]\Bigg\}. \tag{24}
$$

The stress in any cross-section of ropes could be determined from the following relationships:

a) for the balance rope

$$\sigma^{*}\left(x,t\right)=-\frac{2EQ_h}{a_W M^2}\left\{\frac{M^2 a}{2AE}+\frac{\frac{AE}{L_1}e^{-a_0\left(t-\frac{x}{a_W}\right)}}{\left[a_0^3-2\frac{AE}{Ma}a_0^2+\left[\left(\frac{AE}{Ma}\right)^2+\frac{2AE}{ML_1}\right]a_0-2\frac{\left(AE\right)^2}{M^2L_1a}\right]}+\right.$$

$$+\frac{Ma_0e^{-\frac{AE}{Ma}\left(t-\frac{x}{a_W}\right)}}{2\frac{AE}{Ma}\left(\frac{AE}{Ma}-a_0\right)}+\frac{Ma_0e^{-\frac{AE}{2Ma}\left(t-\frac{x}{a_W}\right)}}{2\sqrt{2\frac{AE}{ML_1}-a_0\left(\frac{AE}{Ma}-a_0\right)}\sqrt{2\frac{AE}{ML_1}-\left(\frac{AE}{2Ma}\right)^2}}\cdot$$

$$\left.\cdot\sin\left[\sqrt{2\frac{AE}{ML_1}-\left(\frac{AE}{2Ma}\right)^2}\left(t-\frac{x}{a_W}\right)+\Phi_1\right]\right\}, \tag{25}$$

where:

$$\Phi_1=\frac{\pi}{2}-\operatorname{arctg}\left\{\frac{\frac{AE}{2Ma}-a_0}{\sqrt{2\frac{AE}{ML_1}-\left(\frac{AE}{2Ma}\right)^2}}\right\},$$

b) for the hoisting rope

$$\sigma^{*}\left(y,t\right)=\frac{2EQ_h}{M^2a_N}\cdot$$

$$\left\{\frac{M^2a}{2AE}+\frac{\left[Ma_0^2+AE\left(\frac{1}{L_1}-\frac{a_0}{a}\right)\right]}{\left[a_0^3-2\frac{AE}{Ma}a_0^2+\left[\left(\frac{AE}{Ma}\right)^2+2\frac{AE}{ML_1}\right]\right]a_0-2\left(\frac{AE}{Ma}\right)^2\frac{a}{L_1}}e^{-a_0\left(t-\frac{y}{a_N}\right)}+\right.$$

$$+\frac{Ma_0}{2\left(\frac{AE}{Ma}\right)\left(\frac{AE}{Ma}-a_0\right)}e^{-\frac{AE}{Ma}\left(t-\frac{y}{a_N}\right)}-\frac{Ma_0e^{-\frac{AE}{2Ma}\left(t-\frac{y}{a_N}\right)}}{2\sqrt{2\frac{AE}{ML_1}-a_0\left(\frac{AE}{Ma}-a_0\right)}\sqrt{2\frac{AE}{ML_1}-\left(\frac{AE}{2Ma}\right)}}\cdot$$

$$\left.\cdot\sin\left[\sqrt{2\frac{AE}{ML_1}-\left(\frac{AE}{2Ma}\right)^2}\left(t-\frac{y}{a_N}+\Phi_1\right)\right]\right\} \tag{26}$$

The load of the suspension of the loaded upper skip in the highest position could be determined from the relationship:

$$S^*_{L_N} = k\left[u^*(x=0,t) + v^*(y=0,t)\right] =$$

$$= Q_h \left\{ \frac{-\sqrt{2\frac{AE}{ML_1}}a_0 e^{-\frac{AE}{2Ma}t}}{\sqrt{2\frac{AE}{ML_1} - a_0\left(\frac{AE}{Ma} - a_0\right)}\sqrt{2\frac{AE}{ML_1} - \left(\frac{AE}{2Ma}\right)^2}} \sin\left(\sqrt{2\frac{AE}{ML_1} - \left(\frac{AE}{2Ma}\right)^2}\, t + \Phi\right) + \right.$$

$$\left. + \frac{2\frac{AE}{ML_1}e^{-a_0 t}}{a_0^2 - \frac{AE}{Ma}a_0 + 2\frac{AE}{ML_1}} - 1 \right\} \qquad (27)$$

Fig. 6 shows the example of the load variation in the suspension gear for the emergency braking when the loaded skip comes to its highest position. The curves concern different distances between the skip and the drive wheel (l_1=100 m, 300 m, 500 m) with the assumption (22).

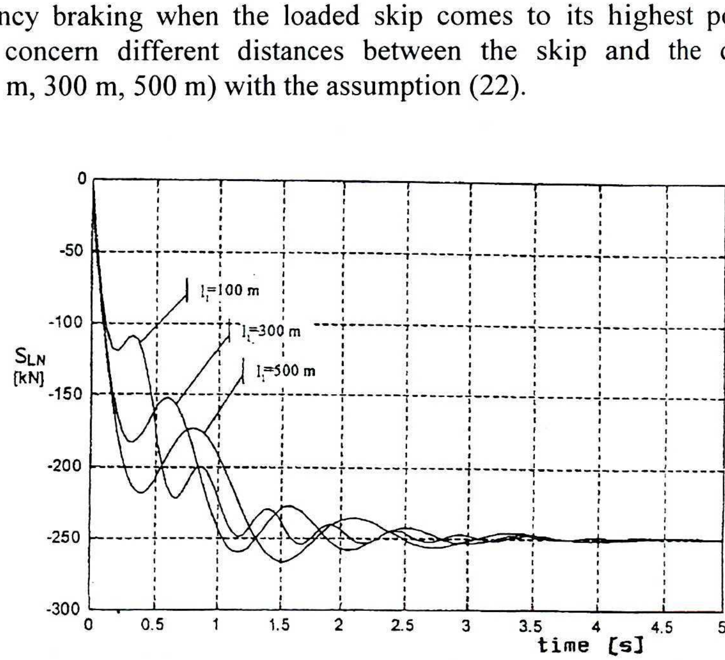

Fig.6. Variations of the load in the skip suspension gear for emergency braking and different distances between the skip and the drive wheel (l_1 = 100 m, 300 m, 500 m)

The presented diagrams were drawn for the winding machinery of the above parameters for $M_0=M_1=M=40000$ [kg], and the braking assembly characteristic parameters: $Q_h = 250$ [kN], $t_0 = 0,8$ [s], $a_0 = -2,37$ [1/s].
The results obtained in theoretical considerations, describing the process of operational braking, were practically verified in the object by measurements of selected parameters. Owing to some technological difficulties, the experiment was limited to the measurement of forces in the suspension of the loaded skip coming to its highest position and retarded by the emergency brake.

3. Measurements of forces in the tub suspension done in practical conditions during emergency braking of the real object

Results of theoretical analysis were practically verified by measurement of forces in some elements of the mining hoist facility during emergency braking in the real object experiment in one of mining shafts.

Fig. 7 shows the scheme of the hoist facility where the experiment was carried out.

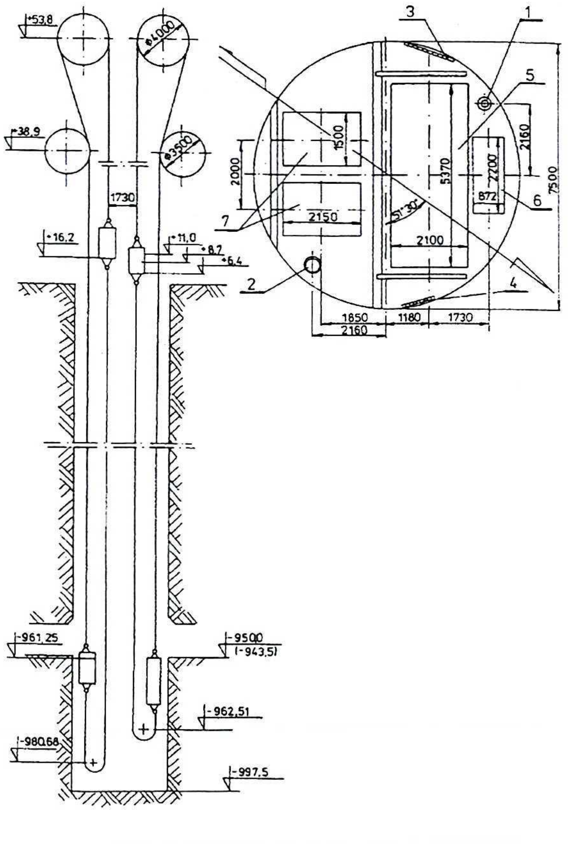

Fig.7 The scheme of hoist facility.

3.1. General technological data of the hoist where the experiment was done

General technological data of the hoist facility:

Type of the machine	4*L*-4000/2900.
Drive	D/C motor 2900 kW.
Maximum rotational speed	77 rpm.
Maximum velocity of the skip hoist	$v = 16$ [m/s].
Mass of the assembled empty tub with the suspension	$m_{ku} = 16500$ [kg].
Load	$m_u = 17000$ [kg].

Moments of inertia of rotating elements (GD^2)	
a) flywheel effect of the drum	$GD_B^{\ 2} = 1868.8$ [kNm^2],
b) flywheel effect of the motor rotor	$GD_s^{\ 2} = 1275.3$ [kNm^2],
c) flywheel effect of the guide wheels	$GD_K^{\ 2} = 427$ [kNm^2].

The weight of guiding wheels ($D_K = 3{,}5$ m) reduced on the rope axis:

$$G_{KKz} = \frac{GD_K^2}{D_K^2} = \frac{474}{3{,}5^2} = 38.68\left[\text{kN}\right].$$

The weight of the drum ($D_B = 4{,}00$ m) reduced on the rope axis:

$$G_{Bz} = \frac{GD_B^2}{D_B^2} = \frac{1868.8}{4^2} = 116.8\left[\text{kN}\right].$$

The weight of the motor rotor ($D_B = 4{,}00$ m) reduced on the rope axis:

$$G_{Wz} = \frac{GD_S^2}{D_B^2} = \frac{1275.3}{4^2} = 79.7\left[\text{kN}\right].$$

Reduced rotating masses in the tower:

$$M_0 = \frac{1}{g}\cdot\left(G_{KKz} + G_{Wz} + G_{Bz}\right) = 1/9{,}81\left(38.68 + 116.8 + 79.7\right) = 23971\left[\text{kg}\right]$$

Hoisting ropes:

– number of ropes	$n_{LN} = 4$,
– length of ropes	$l_N = 1230$ [m],
– diameter of ropes	$\phi = 40$ [mm],
– total section area of ropes wires	$A_N = 710$ [mm^2],
– ultimate strength of ropes	$S_z = 4 \times 1154{,}0 = 4616$ [kN],
– mass of 1 m of ropes	$q_N = 6{,}8$ [kg/mb].

Balance ropes:

– number of ropes	$n_{lw} = 2$,
– length of ropes	$l_w = 1200$ [m],
– diameter of ropes	$\phi_w = 58$ [mm],

– total section area of ropes wires $A_w = 1545$ [mm^2],
– ultimate strength of ropes $S_{zw} = 155$ [kN],
– mass of 1 m of ropes $q_w = 14$ [kg/mb].

The load function of the suspension in the operational cycle was determined with regard to the following parameters of the hoist facility:

– starting acceleration $a_1 = 0,8$ [m/s^2],
– retardation $a_2 = 1,0$ [m/s^2],
– propagation velocity of the elastic deformation wave in hoisting ropes $a_N = 3700$ [m/s].

$$\sum A_N = 4A_N = 4 \cdot 710 = 2840\left[\mathrm{mm}^2\right] = 2840 \cdot 10^{-6}\left[\mathrm{m}^2\right],$$

$l_N = 1230$ [m],
$\mathrm{E}_N = 1{,}1 \cdot 10^{11}$ [N/m^2],
$M_1 = m_u + m_{ku} = 16500 + 17000 = 33500$ [kg],
$l_{N1} = 42{,}8$ [m],

– weight of the part of the balance rope for two positions of the skip:

a) extreme bottom position:
  – the length of balance rope in the shaft sump $l_{wr} = 18$ [m]:
  $$G_{Wr} = 2 \cdot l_{wr} \cdot q_w \cdot g = 2 \cdot 18 \cdot 14 \cdot 9{,}81 = 4{,}94 \text{ [kN]},$$
b) extreme upper position:
  – the length of the balance rope $l_{wr} = 1248$ [m]:
  $$G_{Wh} = 2 l_{wn} \cdot q_w \cdot g = 2 \cdot 1248 \cdot 14 \cdot 9{,}81 = 342{,}80 \text{ [kN]}.$$

3.2. Apparatus for measurement and recording forces in suspension gears

Figure 8 [3] shows the scheme of the apparatus for measurement and recording forces in suspension gears. Force strain gauges WSP of the following technological parameters were used in measurements:

measuring range	100 [kN],
output voltage	5 [V],
accuracy class	0.6,
supply of the bridge	5 [V].

The assembly ZPR-1 supplied by the battery, type HP 2,6–12 V was applied for recording signals from eight sensors (doubled in each string) with the sampling frequency of 40 kHz. Recorded signals were converted into text files, then summed-up, and finally processed with the use of the software Matlab 5.2. Peripheral velocity of the drive wheel was recorded in all measurements.

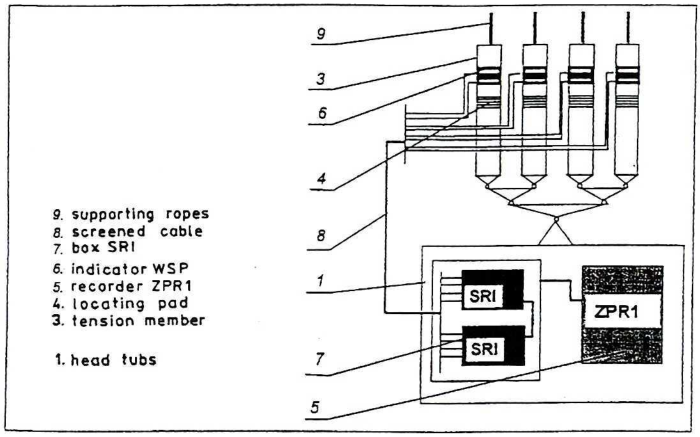

Fig.8 Scheme of the measurement line and location of force sensors

3.3. Results of measurements of forces in tub suspension gears in emergency braking

In this part of the paper the author presents measurement results of forces in the suspension gear of the skip coming to the shaft's top in the condition of emergency braking of the hoist facility whose technological parameters are described above.

Dotted line in Fig. 9 represents the real load of the skip suspension gear at the shaft's top (l_1 = 100 m) at emergency braking. The hoist facility was stopped by the braking assembly of the characteristics shown in Fig. 5 and described by the relation (21). Parameters of the braking assembly were determined on the basis of technological documentation of the facility [2] and the results of pressure measurements in the braking assembly cylinder.

The characteristic parameters of braking assembly, determined by calculations are:
– braking force Q_h = 248 [kN],
– a_0 = 2.11 [1/s],
– t_0 = 0.90 [s].

The change of load of the skip suspension gear resulting from theoretical consideration is plotted with by the continuous line (relation (26)).

Fig. 10 shows the change of the load of the skip suspension gear in the conditions of emergency braking for different starting velocities (operational motion velocities) $V_0 = V$ = 10 [m/s] (dotted line) and $V_0 = V$ = 16 [m/s] (continuous line). Parameters of the hoist facility and the brake assembly were as given above. The braking process was realised for the distance between the filled tub and the drive wheel l_1 = 540 m.

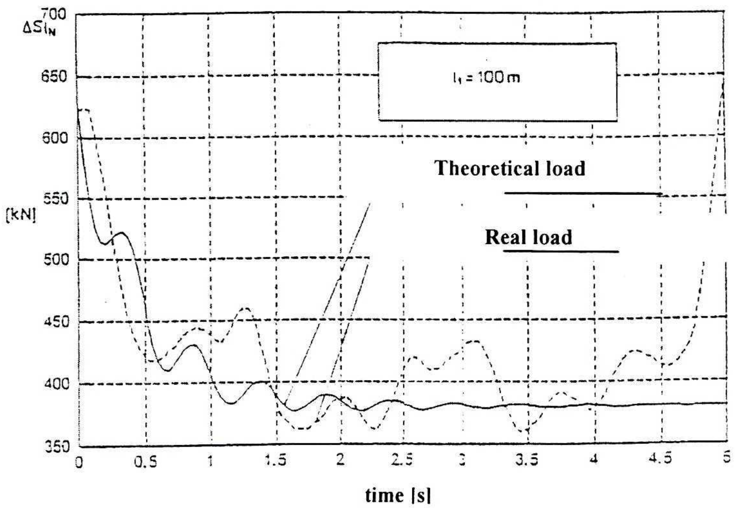

Fig. 9. Diagram of the change of skip load in the condition of emergency braking for the distance between the skip and the drive wheel: l_1= 100 [m]

- load calculated in theoretical consideration (continuous line),
- real load of the tub determined by measurement (dotted line),

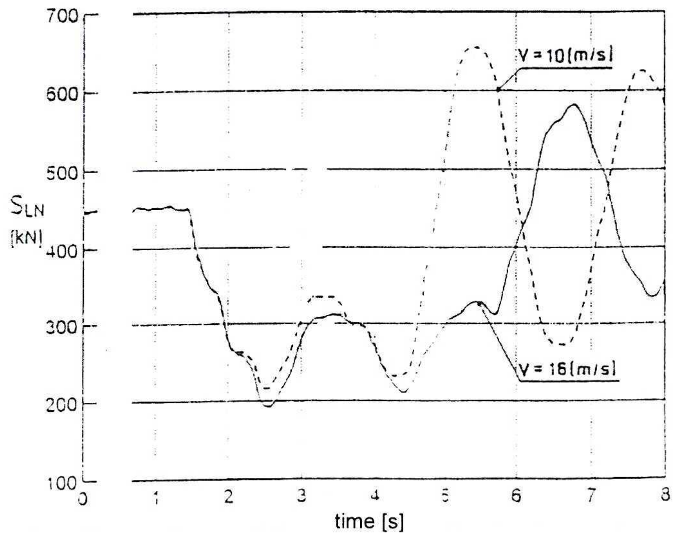

Fig.10. Diagram of the change of the skip load during emergency braking for two different o-erational velocities (l_1=540 [m])

- v_0 = 10 [m/s]; dotted line,
- v_0 = 16 [m/s]; continuous line.

Additionally, Fig. 11 shows the run trace recorded during the tests of the skip suspension gear for emergency braking condition for the steady-state velocity of the tub v = 16 [m/s].

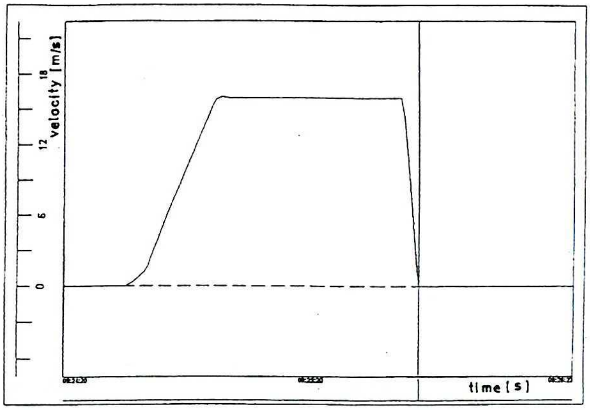

Fig. 11. Trace at run recorded during testing the tub suspension load in condition of emergency braking

4. Concluding remarks

The results of measurements, although limited in this paper only to measurements of loads in suspension gear, are consisted with the theoretical values predicted by the dynamic analysis presented in chapter 2. Differences between characteristic values of measured and theoretically determined parameters describing the process of emergency braking – namely loads in the skip suspension gears – do not exceed, a few percent even in most extreme case (see diagrams Fig. 9, Fig. 10).

The obtained results may be useful as basic data for development of an appropriate design process of suspension elements, but they could also be applied for a specific design of frictional coupling between the rope and the drive wheel (angle of contact, friction coefficient of lining). Moreover, these results, either analytical (given in the general form) or obtained as measurement data (verifying the theoretical solutions) could be useful for the development of criteria for the estimation of technological state of the skip elements with particular regard to fatigue durability as the function of operational time and the type of the hoist facility.

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Obciążenia dynamiczne wybranych elementów górniczego urządzenia wyciągowego w warunkach hamowania manewrowego

Streszczenie

Opracowanie kryteriów oceny stanu technicznego elementów górniczego urządzenia wyciągowego w tym zawieszeń naczyń i lin wyrównawczych, ze szczególnym uwzględnieniem trwałości zmęczeniowej w funkcji czasu eksploatacji i rodzaju urządzenia wyciągowego, wymaga przeprowadzenia gruntownej analizy wytrzymałościowo-zmęczeniowej, uwzględniającej rzeczywiste wartości ich obciążeń oraz zmiany tego obciążenia w czasie. Nie można jednak tego osiągnąć bez wnikliwych studiów nad dynamiką procesów zachodzących w czasie normalnej eksploatacji urządzenia jak również awarii. Rozważania zawarte w pracy ograniczono do analizy zjawisk dynamicznych zachodzących w warunkach hamowania manewrowego urządzenia wyciągowego. Uzyskane rezultaty zweryfikowano pomiarami obciążeń wybranych elementów analizowanego układu na obiekcie rzeczywistym.