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A new design solution for reducing the impact of transported rock on belt conveyors in mining

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Abstract:

As we are increasingly focusing on the economy of mining, researchers are constantly trying to find new technologies to improve existing machines to be more efficient and less perishable. Such actions allow mining companies to save money in all of the mining processes. The authors focused on the transport of materials and how to improve it. The most frequent problems in this matter are caused by the destruction of conveyor belts. The cost of repairs is estimated at about half a billion crowns in the Czech Republic. That is why engineers are working on reduction of the rock impact force on the conveyor belt. The paper deals with the issue of reducing the passage of conveyor belts by proposing a new solution to the impact stand, which will considerably reduce the number of conveyor belt breaks.

Streszczenie:

Zwracając uwagę na ekonomikę wydobycia, nieustannie trwają badania poszukujące nowych technologii ulepszających istniejące maszyny i urządzenia, by były one bardziej wydajne i mniej wadliwe. Takie działania pozwalają firmom górniczym oszczędzać pieniądze we wszystkich procesach wydobywczych. Autorzy skupili się na transporcie materiałów i tym, jak go poprawić. Najczęstsze problemy w tej kwestii są spowodowane zniszczeniem taśm przenośnikowych. W Czechach koszt napraw szacowany jest na około pół miliarda koron rocznie. Dlatego inżynierowie pracują nad zmniejszeniem siły uderzenia skały w taśmę przenośnika. W artykule poruszono kwestię zmniejszenia awaryjności taśm przenośnikowych w miejscu przesypu, proponując nowe rozwiązanie stołu udarowego pochłaniającego energię uderzenia, który znacznie zmniejszy liczbę zerwań taśmy przenośnikowej.

1. Introduction

From the results of the measurements [1, 2, 3, 4] and the verification of the dynamics of impact bed with impact bars follow the result that for the further improvement of the impact bed must be rearranged at least one supporting element and on the frame stand of the impact bed must be assigned at least one connection piece which one end is connected to the support part by the holder and the other end is movably connected to the frame stand of the impact bar. This increases the efficiency of the impact bar and at the same time converts the impact bed construction into multiple springing with an effort to achieve a flexible attachment at all points of the fixed attachment parts of impact bed, including the attachment itself to the conveyor frame. This solution is illustrated in fig.1, 2, 3 and 4.

2. Description of a proposed impact bed

The new impact bed [4] is composed of an impact bed frame containing a vertical, to the belt movement cross-established frame stands which contain a device for flexible attachment to the belt conveyor frame. Stands are set approximately upwardly in the form of an open gutter which concave side contains a plurality of support parts. There is at least one flexible bar attached to each support part. The improvement is based on the fact that between at least one supporting part and the impact bed frame stand is associated a minimum of one connecting part, the first end of which is secured by the holder of the support part to the support part and the second part, which is opposite to the first end of the fastener part, is movably attached to the frame stand of the impact bed. The second part of the fastener part is then movably connected to the stand through with it connected top plate and bottom plate, which are positioned one above the other with a gap and at the same time, the connecting part is connected through holes in the upper and lower plates of the stand. To the first end of connecting part is assigned at least one (first) springing tool which is set up so it is in touch with firstly its adjacent side with an inner surface of support frame holder and secondly its opposite side with the top surface of support frame top desk. To the second end of connecting part is assigned at least one (second) springing tool which is set up so it is in touch firstly its adjacent side with supporting surface second end of connecting part and secondly its opposite side with a lower surface of impact bed support frame´s lower desk. The connecting piece is connected to the support part so, between the inner surface of the holder, at least one first springing tool and the upper surface of impact bed support frame´s upper desk and also between supporting surface of the second end of connecting part, at least one-second springing tool and the lower surface of impact bed support frame´s lower desk is created connection with preload.

In the new design is a support frame of impact bed formed by two sidewalls which are established parallelly beside each other with a gap which allows established connecting part between sidewalls. To the stable securing of support parts establishing, carrying impact bars, are sidewalls of the support frame in a perpendicular direction to the upper desk mounted with a group of cross members whose number is one more than several assigned support pieces.

Each support piece is established in the area between two cross members. For the secure possibility of linear sliding movement of support piece in the area between two cross members there and back, is the inner distance between two beside cross members minimum or more than is the width of support piece in its cross-section. Frame stand construction´s stiffness is preferably secured by that both sidewalls, upper desk of frame stand, lower desk of frame stand and the cross beams are together connected, so they form welding piece. For the right function of an impact, the bed is also advantageous when the holes in the upper frame stand desk and the lower frame stand desk between two sidewalls are situated in one axis above each other.

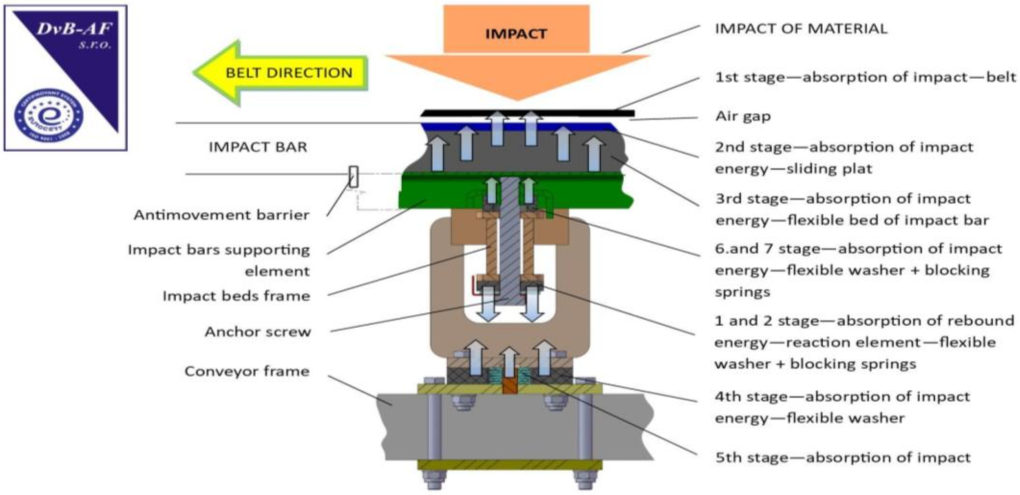

Fig. 1. Force load scheme acting on the impact bed [1]

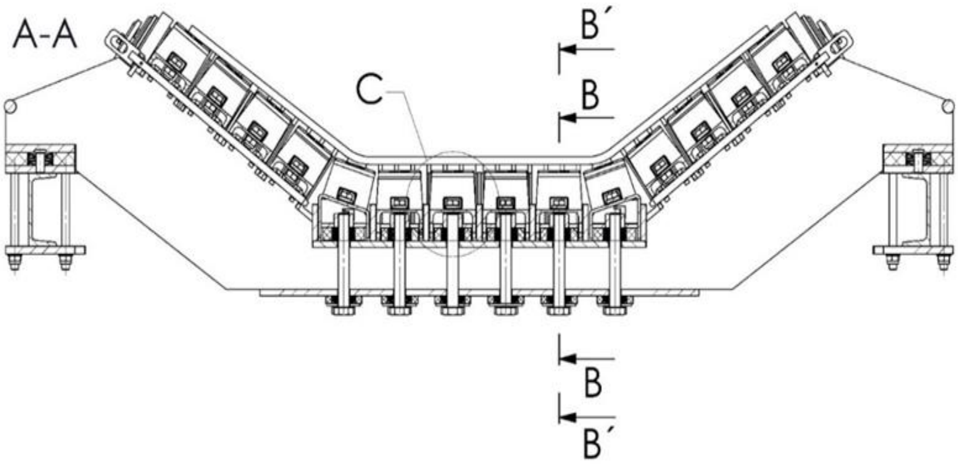

Fig. 2. Cut of impact bed [4]

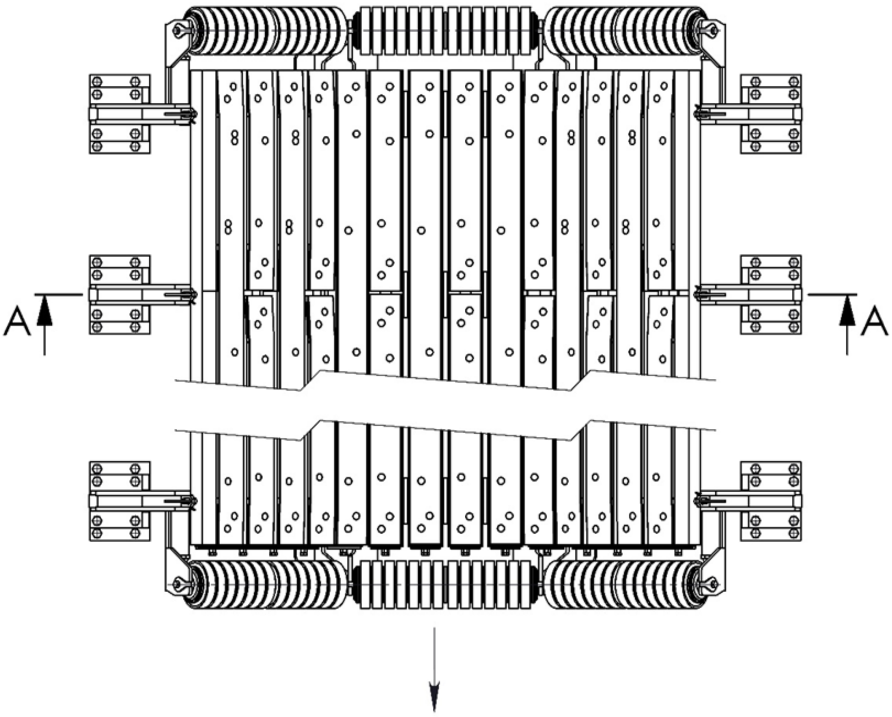

Fig. 3. Ground plan of impact bed [4]

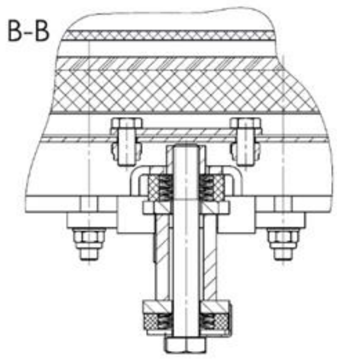

Fig. 4. Cut of impact bed ´s absorbing part [4]

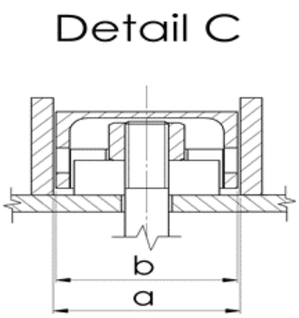

Fig. 5. Cut of holder [4]

3. Determination of the damping coefficient of absorbing block

As it can be seen from the theory of material impact on the impacted place, for determining of damping force the damping coefficient is deciding [2]. This coefficient shows us the size of damping for impact bed. For a quality appraisal of the impact bed, it is necessary to decide this figure.

3.1. Measuring of damping coefficient on impact bed absorber [2]

The proposed procedure for determining the damping coefficient is based on the deformation of the block of material induced by the fall of the weights (fig. 6). The block is connected on the bottom side with a massive base plate and on the top side is connected with the desk where the weights fall. The weight of the desk is smaller than the weight of the weights.

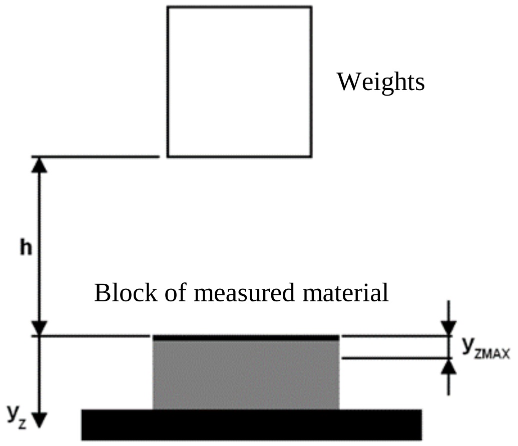

Fig. 6. *Scheme of measuring systems*

After falling off the weights from known heights the measured block will be pressed. When will be the maximum deformation is reached the deformation stops and then the weights will go up. If we know the stiffness of the material of the block, the damping parameters (coefficient of linear damping, coefficient of viscous damping) will be given from measuring of geometric variable: the start height of weights above the desk and maximum compression of the block.

In the phase of material compression the movement of the weight is described by motion equation:

$$(m_z + m_M) \cdot \ddot{y}_z + b_M \cdot \dot{y}_z + k_M \cdot y_z = (m_z + m_M) \cdot g \tag{1}$$

m_z is the weight of weights, m_M is the weight of the measured block, b_M is coefficient of material linear damping, k_M is stiffness, g is gravitational acceleration, y_z is a shift of the weights (compression of a block) and ($\dot{y}_z$), ($\ddot{y}_z$) means the first and the second derivative.

The solution of the motion equation (1) in prerequisite under critical damping is:

$$y_z = C_1 \cdot e^{-\delta t} \cdot \sin(\Omega_t \cdot t + C_2) + y_G \tag{2}$$

C_1, C_2 are integration constants, δ is subsided coefficient, Ω_t is own circular frequency of subdued oscillation, t is time and y_G is particular integral. It can be set an estimate by the form of the right side of the motion equation (1):

$$y_G = \frac{(m_Z + m_M) \cdot g}{k_M} \tag{3}$$

For own circular frequency of subdued oscillation is:

$$\Omega_t = \sqrt{\Omega^2 - \delta^2} \tag{4}$$

where Ω is own circular frequency of unsubdued oscillation of the system

$$\Omega^2 = \frac{k_M}{m_Z + m_M} \tag{5}$$

A relation for movement speed of the weights is set by a derivative of relationship (2) for deviation by time

$$\dot{y}_Z = -\delta \cdot C_1 \cdot e^{-\delta\, t} \cdot \sin(\Omega_t \cdot t + C_2) + C_1 \cdot \Omega_t \cdot e^{-\delta\, t} \cdot \cos(\Omega_t \cdot t + C_2) \tag{6}$$

Integration constants C_1, C_2, coefficient of desaturation δ and stopping time of the movement is set from additional conditions:

$$y_Z(0) = 0 \tag{7}$$

$$\dot{y}_Z(0) = v_0 \tag{8}$$

$$y_Z(t_Z) = y_{ZMAX} \tag{9}$$

$$\dot{y}_Z(t_Z) = 0 \tag{10}$$

v_0 is impact speed of the weights to the block of measured material, y_{ZMAX} marks its maximum compression.

After being put into relationships (6) – (9) to (2) and (5) is obtained set of four non-linear algebraic equations, where the unknowns are C_1, C_2, δ and t_Z

$$C_1 \cdot \sin(C_2) + y_G = 0 \tag{11}$$

$$-\delta \cdot C_1 \cdot \sin(C_2) + C_1 \cdot \Omega_t \cdot \cos(C_2) = v_0 \tag{12}$$

$$C_1 \cdot e^{-\delta\, t_Z} \cdot \sin(\Omega_t \cdot t_Z + C_2) + y_G = y_{ZMAX} \tag{13}$$

$$-\delta \cdot \sin(\Omega_t \cdot t_Z + C_2) + \Omega_t \cdot \cos(\Omega_t \cdot t + C_2) = 0 \tag{14}$$

Impact speed of the weights to the block of measured material from the height h in the absence of air resistance is given by the relationship:

$$v_0 = \sqrt{2\, g \cdot h} \tag{15}$$

For the coefficient of viscous damping η_V and the linear damping factor are valid

$$\eta_V = \frac{2 \cdot (m_Z + m_M) \cdot \delta}{k_M} \tag{16}$$

$$b_M = \eta_v \cdot k_M \tag{17}$$

Block stiffness of measured material is determined from the measurement of its deformation with known weight, for example under a load of own weights which are placed on it:

$$k_M = \frac{m_Z \cdot g}{\Delta_G} \tag{18}$$

The damping viscosity coefficient is then determined from measuring the impact height of the weights h and the maximum compression of the block of the measured material y_{ZMAX}

The solution of the motion equation (3.1) assuming supercritical damping has the form

$$y_Z = C_3 \cdot e^{\lambda_1 t} + C_4 \cdot e^{\lambda_2 t} + y_G \tag{19}$$

where C_3, C_4 are integration constants and λ_1, λ_2 are roots of the characteristic equation

$$\lambda_1 = -\delta - \sqrt{\delta^2 - \Omega^2} \tag{20}$$

$$\lambda_2 = -\delta + \sqrt{\delta^2 - \Omega^2} \tag{21}$$

y_G is the particular integral estimated by the shape of the right-hand side of the motion equation (3.3)

The equation for the speed of movement of the weight is determined by the derivative of the equation (3.19) for deviation by time

$$\dot{y}_Z = \lambda_1 \cdot C_3 \cdot e^{\lambda_1 t} + \lambda_2 \cdot C_4 \cdot e^{\lambda_2 t} \tag{22}$$

The integration constants C_3, C_4, the decay coefficient δ and the stopping time of the weight shall be determined from the additional conditions (7) - (10). Substituting them into formulas (21) and (22), a set of four non-linear algebraic equations is obtained, where they are unknown with respect to (20) and (21) C_3, C_4, δ and t_Z.

$$C_3 + C_4 + y_G = 0 \tag{23}$$

$$\lambda_1 \cdot C_3 + \lambda_2 \cdot C_4 = v_0 \tag{24}$$

$$C_3 \cdot e^{\lambda_1 t_Z} + C_4 \cdot e^{\lambda_2 t_Z} + y_G = y_{ZMAX} \tag{25}$$

$$\lambda_1 \cdot C_3 \cdot e^{\lambda_1 t_Z} + \lambda_2 \cdot C_4 \cdot e^{\lambda_2 t_Z} = 0 \tag{26}$$

The speed of impact of the weight on the measured block is given by (15). The viscosity damping coefficient η_V and the linear damping coefficient b_M are then determined from formulas (16) and (17).

The viscosity damping coefficient is in both cases calculated from the measurement of the height of the weight h and the maximum compression of the y_{ZMAX} material block. For damping to be subcritical (weak), it must apply:

$$\delta < \Omega \tag{27}$$

For supercritical (strong) damping is valid

$$\delta < \Omega \tag{28}$$

4. Conclusion

Problems of mathematical modelling of rock impact on the conveyor belt, are not a simple matter at all. Here we have to make a certain simplification because the issue of dynamics is very complicated and practically we are not able to include all the factors influencing the excited oscillation in the calculations. It is also very problematic to determine the correct modulus of the conveyor belt elasticity, the coefficient of viscous damping of the belt material and other factors. Also, the measurement problem of the impact bed shock absorber coefficient is not as simple as expected, and despite the utmost effort to achieve accurate results, it has not been succeeded. On the other hand, in cooperation with other experts, we measured the impact bed vibration, both on an excavator with a classic impact spot with garlands and a new solution with impact bed with impact bars. Even here we have some things that are directly related to practice. Further progress should go through the improvement of impact silencer even though there have been no defects in this part of the device during the two-year operation. The research should also continue to reduce the weight of the impact bed and by the construction solution of the newly designed absorbing parts.

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