STUDY OF VIBRATION REGIME TO CONTROL NOISE PRODUCED BY BELT CONVEYORS

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Abstract: Vibrations are dynamic phenomena that are found both in the human body through the heartbeat, the activities of humans and animals through running and walking, various natural phenomena such as swaying trees in the wind and buildings shocks to earthquakes, vibrations produced by musical instruments, pneumatic drills, reciprocating belt conveyors etc. The vibrations are often referred to shake that produce relatively high noise or mechanical stress. To reduce and control the level of noise pollution produced by conveyor is necessary to determine the vibration regime. In this paper, we propose to study the vibration mode of belt conveyors to control noise from the belt conveyors.

Keywords: vibrations, noise, belt conveyors, screen

1. INTRODUCTION

Most often "vibrations" are defined as unwanted movements that produce relatively high noise or mechanical stress. In this case, particularly we are interested in the effect of vibrations on humans, machines and buildings. Modeling involves defining the structure and parameters of vibratory bodies in vibration excitation functions and levels that describe the dynamic response.

Control the frequency and amplitude of vibration are necessary because they are harmful to machines and systems, particularly affecting the proper functioning of machines, wear their precision and their foundations, and human health.

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Exceeding the limits of tolerance of vibrations have negative influence on human health and on labor productivity, producing physical trouble and intellectual activity, damage to parts of the body, as subjective phenomena: the perception of vibration, discomfort, pain and fear.

To reduce noise produced by conveyer belts and sound-absorbing acoustic panels we realized the polycarbonate plates mounted directly on the metal structure of the conveyor belts. Their behavior is influenced by regime vibration that is transmitted from their belts.

Vibration from conveyors are generated by structural vibration effect creating waves that propagate in the metal structure with low rigidity, in which case antinode areas are sources of radiating noise.

2. THEORETICAL CONSIDERATIONS

Soundproofed and sound-absorbing panels located on conveyor belts metal structure is attached to a height of 0.5 m from the ground. This height was chosen to be screened all the moving parts of the conveyor belt.

These soundproofing panels used in noise shielding acts as a monopartition.

Isolator panel can be modeled as a homogeneous tile referring to a system of axes Oxyz and normal wave radiation directly enters a vibratory motion in the direction Ox simultaneously bending wave propagation.

Considering excitement bending waves in one direction, the bending moment $M = M_Z$ to give the appearance of a strong variable in the direction Ox:

$$F = \frac{\partial M}{\partial y}; \frac{\partial^2 x}{\partial y^2} = -\frac{M}{B}, \tag{1}$$

in which:

$$B = \frac{El_y}{1 - v^2} = \frac{Eh^3}{12(1 - v^2)}$$

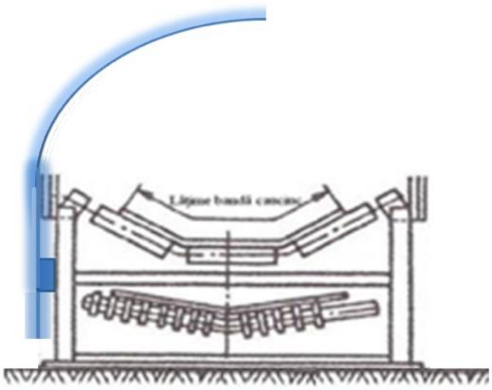

Fig. 1. Curved screen for noise protection

It is the plate rigidity per surface unit where: E - Young module's; v – coefficient of Poisson.

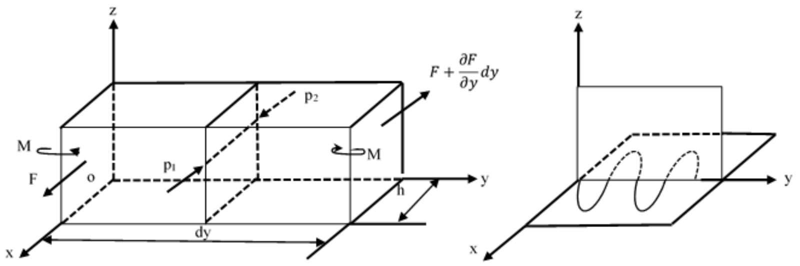

Fig. 2. Graphical representation of the forces occurring in a homogeneous plate

The equation of motion of the unitary wall element after the division with surface plan yOz (dA = l dy) is:

$$m\ddot{x} = p_1 - p_2 + \frac{\partial F}{\partial y}, \quad (2)$$

Where m=l·l·h, and after using the expression (1), becomes:

$$m\ddot{x} = p_1 - p_2 - B\frac{\partial^2 x}{\partial y^2}, \quad (3)$$

Provided that particular air wall surface sp to not detach is:

$$\frac{dx}{dt} = v = Ve^{i\omega t}, \quad (4)$$

Integrated which leads to expression:

$$x = \int v dt = \frac{1}{i\omega}Ve^{i\omega t} = \frac{v}{i\omega} \quad (5)$$

Replacing (3) in (2), we obtain:

$$m\frac{dv}{dt} = p_1 - p_2 - \frac{B}{i\omega}\frac{\partial^4 v}{\partial y^4} \quad (6)$$

from where,

$$p_1 - p_2 = i\omega m v + \frac{B}{i\omega}\frac{\partial^4 v}{\partial y^4}. \quad (7)$$

This raises a bending wave which is transmitted along the axis Oy.

These bending waves occur due to vibrations make sound-insulating screen where they fall on different angles of incidence θ, which make it appear a phase shift in the direction Oy. The phase angle difference is due to the road $Dr = ysin\theta$.

Speed v of air particle in contact with points will be expressed by Dr as follows:

$$v = Ve^{i\omega(t-\frac{\Delta r}{c})} = Ve^{i\omega(t-\frac{y\sin\theta}{c})}$$

In this case results:

$$\frac{\partial^4 v}{\partial y^4} = \left(\frac{\omega}{c}\sin\theta\right)^4 v,$$

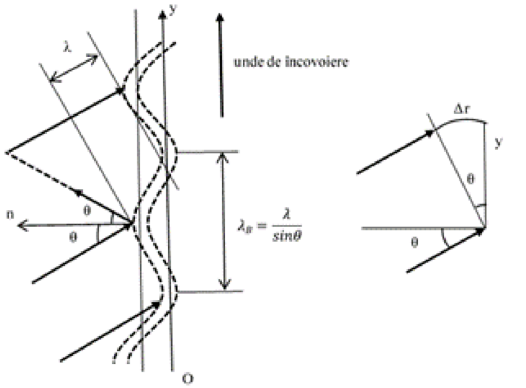

Fig. 3. Transmission of bending wave in a homogeneous plate

Expression (6) become

$$p_1 - p_2 = \left(i\omega m - i\frac{B\omega^3\sin^4\theta}{c^4}\right)v. \tag{8}$$

It has specific nature wave impedance containing reactant:

- Inertion ω m
- Mechanical compliance $\left(\frac{B\omega^3\sin^4\theta}{c^4}\right)$

It will be considered critical if the pressure is transmitted entirely through wall, so $p_1=p_2$.

So:

$$\frac{\partial^4 v}{\partial y^4} = \omega^2\frac{m}{B}v \tag{9}$$

Wave velocity bending the wall is:

$$C_B^4 = \omega^2\frac{B}{m};\ C_B = \sqrt{\omega}\sqrt[4]{\frac{B}{m}} = \sqrt{\omega}\sqrt[4]{\frac{EIy}{m(1-\vartheta^2)}} \tag{10}$$

Introducing (9) in (10) equation become:

$$\frac{\partial^4 v}{\partial y^4} - \left(\frac{\omega}{C_B}\right)^4 v = 0 \tag{11}$$

Which leads to integrated overall solution:

$$v = C_1 e^{\alpha y} + C_2 e^{-\alpha y} + C_3 e^{i\alpha y} + C_4 e^{-i\alpha y}, \alpha = \frac{\omega}{C_B} \tag{12}$$

From expression (10) results that speed CB flexural wave is dependent upon the angular frequency ω, and condition $p_1=p_2$ imposed equation (8), result:

$$1 - \frac{B\omega^2 mc sin^4\theta}{mc} = 0 \implies C_B = \frac{c}{sin\theta} \tag{13}$$

So, for the critic case will result critical velocity of pressure waves which induce wall where bending, forming integral to transmit sound pressure through the wall:

$$\frac{c^2}{sin^2\theta} = 2\pi f_{cr}\sqrt{\frac{B}{m}}, \Longrightarrow f_{cr} = \frac{c^2}{2\pi sin^2\theta}\sqrt{\frac{m}{B}} \tag{14}$$

The lower critical frequency is obtained for θ=π/2 (waves parallel with the wall)

$$(f_{cr})_{min} = \frac{c^2}{2\pi}\sqrt{\frac{m}{B}}. \tag{15}$$

In case of wall h=d,

$$(f_{cr})_{min} = \frac{c^2}{2\pi d}\sqrt{\frac{12\rho(1-\vartheta^2)}{E}}. \tag{16}$$

To avoid sound transmission through the wall without damping aims throwing minimum frequency within the audible frequencies.

3. RESULTS AND DISCUSSIONS

To study the regime vibration of conveyor belts we used MSR145. This is a device that perished measurement of parameters such as temperature, humidity, pressure and acceleration on three axes X, Y, Z. The acceleration can be measured both in 2G and in 10 G.

For the study we conducted measurements on a conveyor belt, which operate unloaded careers tape used to open pit, but also before starting it, to be able to see the differences between the two situations.

Following measurements made can be seen as relative stretching values when the conveyor belt does not work in all three areas are: ACCx - -0.313, ACCY - 0049, ACCz - from 0.873 to 0905 (Figure 4).

From measurements made after starting the conveyor belt can be seen that there are significant differences to relative stretching on all three axes (Figure 5).

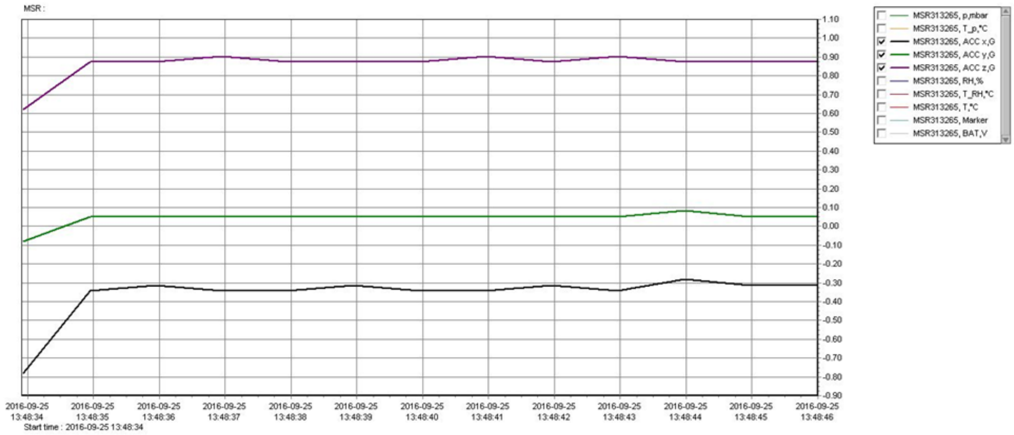

Fig. 4. Relative elongation waves on three axes before starting the conveyor belt

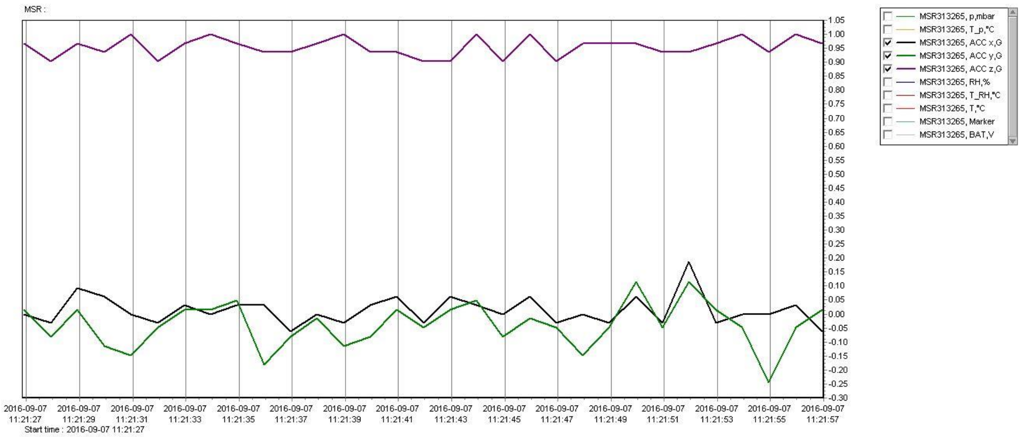

Fig. 5. Relative elongation waves on three axes after starting the conveyor belt

From measurements recorded large variations observed structural vibrations generated by creating waves that propagate effect fraught metal structure on all three axes.

Can be observed that the relative elongation on the x axis between -0.031 which represents the value rest on this axis and 0.188. Y-axis variations are targets for 0049 (the value rest) and -0.148. Z axis shows values between 0.905 (the maximum amount awarded for conveyor belt at value rest) and 1.

Conveyor band operation may influence performance and sound absorbing of soundproofed panels by structural vibration transmission.

4. CONCLUSION

Vibration from conveyors can influence the reduction of noise by absorbing panels and soundproofing panels.

Following measurements made on the conveyor belt which operate without load can be seen this vibration on three axes x, y and z.

Vibration produced by conveyor band lead to degraded performance soundproofing and sound absorbing panels.

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