A R C H I V E S O F M E T A L L U R G Y A N D M A T E R I A L S
Volume 52 2007 Issue 3

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GANTRY BEAMS VIBRATION CONTROL USING CAST IRON CUSHIONS

TŁUMIENIE DRGAŃ BELEK PODSUWNICOWYCH Z WYKORZYSTANIEM ODLEWANYCH PODKŁADEK ŻELIWNYCH

The paper explores the potentials of harnessing the structural friction in the system comprising a gantry beam, cast iron cushion of requiret form and rail to the vibration control in this system.

Theoretical studies yield an equivalent logarithmic decrement of vibration reduction in the system, expressed as a function of rail pressure against a crane beam through a cast iron cushion and hence the function of bolt tightness as well as other parameters.

Outlined are practical applications of a cast iron cushion to vibration control of a gantry rail in a charging car installation.

W opracowaniu przedstawiono możliwość wykorzystania zjawiska tarcia konstrukcyjnego w układzie: belka, odlewana podkładka żeliwna o wymaganym kształcie, szyna podsuwnicowa na tłumienie drgań tego układu.

Na podstawie rozwiązań teoretycznych wyznaczono zastępczy – logarytmiczny dekrement tłumienia drgań rozpatrywanego układu, jako funkcję siły docisku szyny do belki poprzez podkładkę żeliwną, a co za tym idzie i momentu dokręcenia śrub mocujących te elementy oraz pozostałych parametrów.

W zakończeniu przedstawiono praktyczny aspekt wykorzystania żeliwnej podkładki wibroizolacyjnej do tłumienia drgań szyny jezdnej na przykładzie torowiska wozu zasypowego.

1. Introduction

Vibration reduction in gantry crane uprights and beams in lifting installations has received a great deal of attention so far [1], [2].

The authors of the present study investigate the potentials of harnessing the structural friction on the beam-cushion and rail-cushion interface to vibration reduction. Incorporation of a cushion between the gantry beam and rail increases the number of friction faces, which improves energy dissipation, leading to an increase of an equivalent logarithmic decrement of vibration reduction. An increase of a logarithmic decrement of vibration damping might be achieved by increasing the cushion thickness or by adjusting the cushion's pressure against the rail and beam through the control of torque applied to the fixing bolts.

The study investigates the influence of geometric parameters and physical properties of the cushion material on the actual value of the logarithmic decrement of vibration damping in the analysed system, for cushions made from the material with the elasticity modulus $E_p < E_s$ expressed as the function of rail pressure against the beam (E_s – modulus of elasticity of steel). The actual damping performance of a beam in a charging car installation is explored, in a system complete with a cast iron cushion. The practical analysis is supported by theoretical data.

2. Harnessing of structural friction to the vibration damping

The effects of structural friction in a system comprising a beam – cast iron cushion – gantry rail on vibration damping performance are investigated. A freely supported I-beam is considered, with the length L. Cross-section of the beam-cushion-rail system with designations of relevant parameters is shown in Fig 1.

Designation of parameters in the theoretical analysis:

A_s, A_p, A_b – respective cross-sections of the rail, cushion and beam

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C_1, C_2, C_3 – centres of gravity (cog) of the beam, rail and cushion, respectively
C_4 – cog of the whole system
C_5 – cog of the rail with the cushion
y_b – distance between cogs C_4 and C_5
y_s – distance between cogs C_4 and C_5
y_{mb} – distance between cogs C_1 and C_5
y_{ms} – distance between cogs C_1 and C_2
e_b – distance between the system's cog C_4 and the upper beam flange
y_{es} – distance between cogs C_5 and C_2
y_{ch} – distance between cogs C_1 and C_4
E_p – modulus of elasticity of the cushion
E_s – modulus of elasticity of steel.

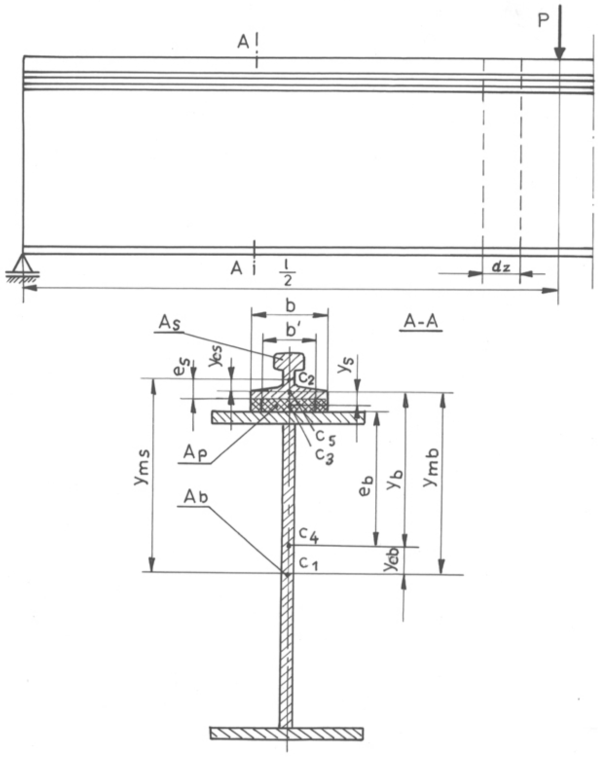

Fig. 1. Cross-section of the beam-cushion-rail system

Underlying the theoretical analysis are the following assumptions:

1. The rail and beam will deform with respect to an axis that is neutral until the tangent stress at the cushion – rail interface exceeds the dry friction force.
2. Beam – rail interactions acting through the cushion (pressing force) are taken as uniform along the whole length, with the intensity q (interaction forces were controlled by varying the torque applied to the fixing bolts).

$$q = \frac{2P_s}{S}\,[\mathrm{N/m}], \tag{1}$$

where: P_s – pressing force in each pair, S – bolt or clamp spacing.

3. The limit value of dry friction per unit surface equals:

$$\tau = \frac{q}{b}\mu\left[\mathrm{N/m^2}\right], \tag{2}$$

where: b – rail toe width, μ – coefficient of friction on the cushion- beam and rail interface.

The hysteresis loops were obtained for the two friction faces by the energy method proposed by I.G Panovko and the envelopes of natural vibrations to be damped were determined accordingly.

The differential equation of the envelope is written as:

$$-T\frac{dA}{dt}(\gamma' A + \lambda\tau_0) = 4x\tau_a A - 8\eta\tau_0^2, \tag{3}$$

where: A – amplitude of vibrations, T – period, γ, λ, χ, η – constants, $\gamma' = 48E\left(I_b + I_s + I_p\right)/L^3$, $\lambda = 2b'y_{ms}$ $x = 2b'y_{mb}\left[1 + \frac{y_s^2 A_b A_p}{y_b^2(A_b+A_s+A_p)A}\right]$, $\eta = b'^2L^3y_{ms}/24A_{s+p}y_b$, τ_0 – value of tangent stress at which slipping occurs: $\tau_0 = \mu q/b$.

Solving Eq (3) with respect to t by grouping of variables and integrating from A_0 to A_n yields:

$$t = Tn, \tag{4}$$

where: n – number of vibration cycles in the amplitude range from A_0 to A_n.

$$n = \frac{\gamma'}{4\chi\tau_0}(A_0 - A_n) + \left(\frac{\gamma\eta}{2\chi^2} + \frac{1}{4}\right)\ln\frac{\chi A_0 - 2\eta\tau_0}{\chi A_n - 2\eta\tau_0}. \tag{5}$$

In order to find the minimal damping time, the minimum of the expression (5) is sought.

$$\frac{dn}{d\tau_{0(A_0\to A_n)}} = 0 \tag{6}$$

Finally, we get:

$$n_{\min} = C\left[A^* - \frac{1}{A^*} + \left(2 + \frac{1}{2C}\right)\ln A^*\right], \tag{7}$$

where:

$$C = \frac{\gamma'\eta}{2\chi^2} = \frac{\left(I_b + I_p + I_p\right)y_{ms}}{4\left(A_S + A_p\right)y_b y_{mb}^2\left[1 + \frac{A_bA_py_s^2}{y_b^2(A_b+A_s+A_p)A_s}\right]^2}$$

A^* – dimensionless amplitude $A^* = \frac{A_0}{A_n}$.

The constant C is associated with geometric characteristic of the beam, cushion and rail. Knowing $n_{\min}$, the equivalent logarithmic damping decrement is derived from the formula:

$$\delta = \frac{1}{n_{\min}} \ln A^*. \tag{8}$$

Eq (7) and (8) were employed to graph the plots of $n_{\min} = f(A^*)$ and $\delta = f(A^*)$, which is shown in Fig 2 and 3.

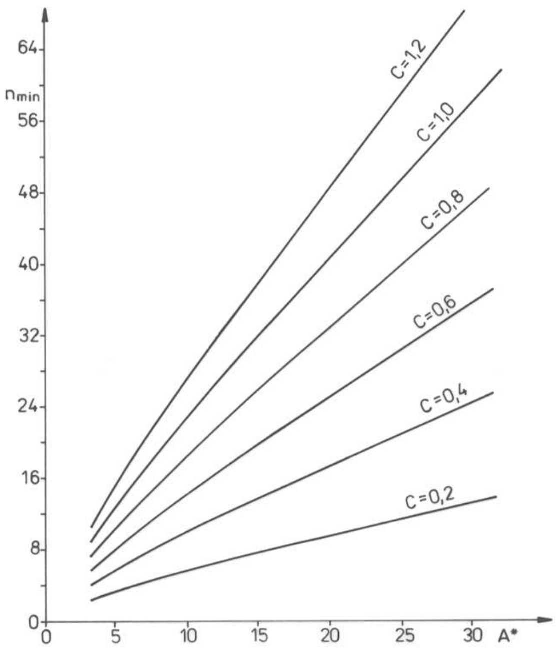

Fig. 2. Plot of $n_{\min} = f(A^*)$ for the constants C

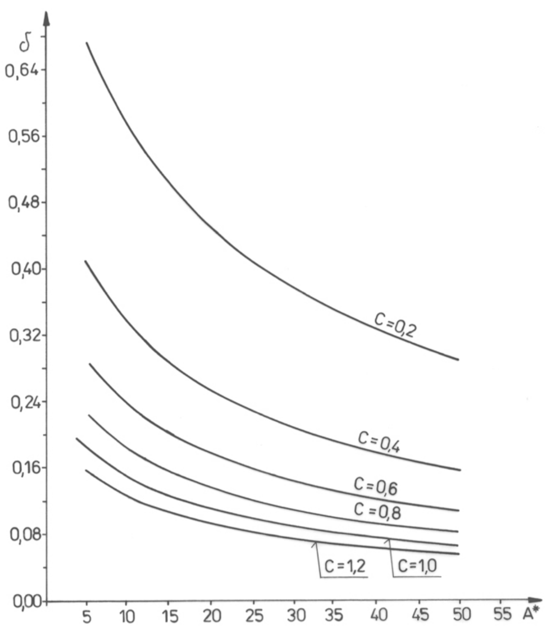

Fig. 3. Plot of $\delta = f(A^*)$ for the constants C

In order to ensure the optimal damping performance in the amplitude range A0-An, the rail should be pressed against to the beam such that the generated tangent stress should be large enough to cause the rail slipping on the cushion and the cushion slipping on the beam.

Transforming Eq (3) yields:

$$P_s = \frac{Sb'\chi A_0}{4\mu\eta\,(1 + A^*)}, \tag{9}$$

where: b' – equivalent cushion width, $b' = bE_p/E_s$.

The required value of force P_s is obtained by applying controlled torque obtained from the formula:

$$M_c = \frac{1}{2} P_s \left[d_s tg\,(\gamma + \rho') + D_s\mu \right], \tag{10}$$

where: d_s – working diameter of the threaded connection, D_s – working diameter of the nut, γ – inclination angle of the helix line, ρ' – apparent friction angle.

3. Application of theoretical data to the vibration damping of a beam in a charging car runway

Theoretical data are corroborated by practical effects of vibration damping strategy utilising a cast iron cushions in a charging car installation. A single beam span is considered with the length $L = 2.5$ m and height $h = 240$ mm.

Cross-section of a beam with the rail and cushion are shown in Fig 4.

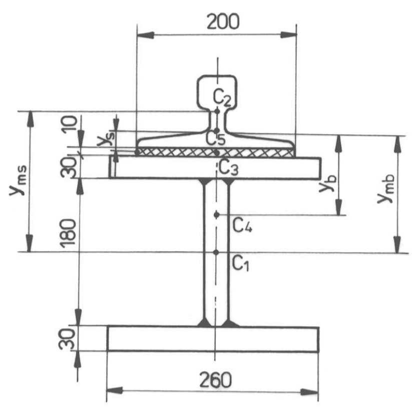

Fig. 4. Cross-section of the charging car runway

Parameters required to determine the equivalent logarithmic decrement of vibration damping are summarised in Table 1.

TABLE 1

Geometric parameters		Constants	Minimal number of cycles	Logarithmic decrement
with a cushion				
$L = 2.5$ m	$b = 20$ cm			
$I_b = 20074$ cm^4	$A_b = 210$ cm^2			
$I_s = 545$ cm^2	$A_s = 72.1$ cm^2	$C = 0.35$	$n_{\min} = 6$	$\delta = 0.38$
$I_p = 1.7$ cm^4	$A_p = 20$ cm^2			
$y_b = 10.8$ cm	$y_{mb} = 15.6$ cm			
$y_s = 3.1$ cm	$y_{ms} = 16.5$ cm			
without a cushion				
$y_{ms} = y_{mb} = y_m = 15.5$ cm		$C' = 0.41$	$n'_{\min} = 8$	$\delta = 0.29$
$y_b = y = 11.3$ cm				
$I_p = A_p = 0$				

TABLE 2

Quantities derived from formulas (9) and (10)		P_s [kN]	M_c [Nm]
$S = 50$ cm	$b' = 10$ cm		
$A^* = 10$	$A_0 = 0.42$ cm		
$\chi = 317$ cm^2	$\eta = 0.514$ cm^2/kG		
$\mu = 0.15$		8.24	71.74
	dla śrub M 20		
$d_s = \frac{d+D_s}{2} = 18.9$ mm			
$D_s = \frac{D_s+D_w}{2} = 27.3$ mm			
$\gamma = 2.68^0$, $p^* = 9.83^0$			

The quantities designated with a prime ’ refer to the system with no cushion. Constant C is derived from the formula:

$$C' = \frac{(I_b + I_s)(A_b + A_s)}{4A_sA_by_m^2}. \qquad (11)$$

Comparison of values of equivalent logarithmic decrements (table 1) reveals that its value is nearly 20% greater in systems complete with cushions in relation to that without a cushion. The damping performance depends on physical and geometric features of the considered system. In the case of a beam in a charging car installation the force of rail pressure against the beam is derived from formula (9) and the torque applied to the fixing bolts is obtained from (10).

Parameters required to find the pressing force and the torque to be applied to bolts are compiled in Table 2.

Quantities derived from formulas (9) and (10).

4. Conclusions

Research data suggest that incorporation of a cast iron cushion between the gantry rail and beam leads to an increase of the equivalent logarithmic decrement of vibration damping in the system. The actual value of the logarithmic decrement depends on mechanical parameters of the cushion (E_p – Young modulus), geometry and interactions (pressure) between the beam, cushion and rail.

Harnessing of structural friction in a system comprising a beam in a charging car installation leads to the increase of the logarithmic decrement of vibration damping. Vibration performance in a system complete with a cushion is improved by 20% in relation to that without a cushion. Improved vibration damping in the analysed system helps to prolong the service life of its major components, particularly the crane beams.

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Received: 18 June 2007.