

Table of contents

Volume 865

2020

◀ Previous issue Next issue ▶

XII All-Russian Scientific and Practical Conference (with international participation) on Automation systems in education, science and production, 2019 28-30 November 2019, SibSIU, Russia, Novokuznetsk

Accepted papers received: 14 May 2020

Published online: 12 August 2020

Open all abstracts

Preface

OPEN ACCESS	011001
-------------	--------

Preface

+ Open abstract  View article  PDF

OPEN ACCESS	011002
-------------	--------

Peer review statement

+ Open abstract  View article  PDF

Papers

OPEN ACCESS	012001
-------------	--------

Control systems for complex technological facilities with the accumulation of experience in the development of control actions

S M Kulakov, M V Lyakhovets, R S Koynov and E N Taraborina

+ Open abstract  View article  PDF

OPEN ACCESS	012002
-------------	--------

Modern concept of technological processes modal control in a state space combined with wavelet monitoring of current modes

D B Fedosenkov, A A Simikova and B A Fedosenkov

+ Open abstract  View article  PDF

OPEN ACCESS	012003
-------------	--------

Structural and parametric non-stationary modal control systems

Functional status technological equipments modelling

G V Makarov, E V Tamarkina, M V Lyakhovets and A S Salamatina

[+ Open abstract](#) [View article](#) [PDF](#)

The mechanism for the formation of a diversified production program of an enterprise, taking into account environmental and economic restrictions

T V Kiseleva, V G Mikhailov and G S Mikhailov

[+ Open abstract](#) [View article](#) [PDF](#)

Flotation reagent dosing control system at the coil preparation plant LLC JV "Barsasskoye Tovarischestvo"

L P Myshlyaev, M V Lyakhovets, G V Makarov, D E Korovin and G A Kulyushin

[+ Open abstract](#) [View article](#) [PDF](#)

Creation of a control automation system on the example of the coal processing plant

M V Shipunov, V V Grachev, L P Myshlyaev, K A Ivushkin, Sh A Fayrushin and G V Makarov

[+ Open abstract](#) [View article](#) [PDF](#)

Geotechnological aspects of mining processes automation

S A Kizilov, S M Nikitenko and M S Nikitenko

[+ Open abstract](#) [View article](#) [PDF](#)

Mirror symmetry in transverse oscillations of ropes and shafts in electromechanical systems

M Yu Borshchinsky

[+ Open abstract](#) [View article](#) [PDF](#)

Models and mechanisms for planning service improvements

A V Zymin, I A Zolin, I V Burkova and V V Zimin

[+ Open abstract](#) [View article](#) [PDF](#)

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To cite this article: M Yu Borshchinsky 2020 *IOP Conf. Ser.: Mater. Sci. Eng.* **865** 012015

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Mirror symmetry in transverse oscillations of ropes and shafts in electromechanical systems

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Abstract: The article explores the possibility of developing a mathematical description of the bending of a round beam, based on end supports, to construct a theory of oscillations of a curved shaft of a mine hoist. The results of the study allow to develop a model of transverse oscillations of the elastic shaft and a long rope in order to synthesize a microcontroller control system.

1. Introduction

The transverse bending of the shaft is closest to that of the round beam (figure 1). But the beam rests freely on the supports, its ends bending upwards. Unlike the beam, the shaft is fixed in bearings, connected to other devices that prevent bending, which excludes the possibility to use theory [1, 2, 3, 4, 5] for the shaft.



Figure 1. Experimental bending of a thin elastic round shaft.

The rope is pulled by a load and approaches the string in flexibility. However, according to Figure 2, the string is broken at the points of attachment and retraction, which is physically impossible for the rope. At the same time, the attachment of the rope ends is closer to the attachment of the shaft ends. So fastening the rope to the vessel with the load also does not allow a kink, and the rope runs onto the hoist drum and the pulley of the headframe tangentially to the circle, which physically excludes bending at the end point of tangency of the pulley circumference. It follows that the rope and shaft are subject to the same theory. However, the additional accounting of rope tension will increase the accuracy of the mathematical model.

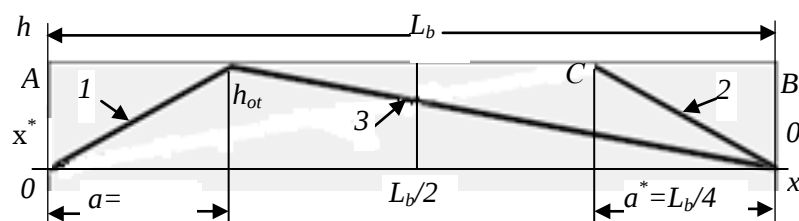


Figure 2. Bend string and mirror line 1 to line 2.



2. Results and discussion

Figure 3 shows the rope of the physical model of the mine hoist. At the pull point, the rope bends without a break, just like the shaft, but unlike a shaft with a smaller radius of curvature.



Figure 3. Rope, hoist drum and headframe of the laboratory stand of mine hoist.

The theory of bending in [4, 5] is built using bending moments M (Figure 4) applied to the beam ends, which made it possible to solve the problem of bending a freely supported beam, but it is impossible to fulfill the condition of pinching the shaft ends.

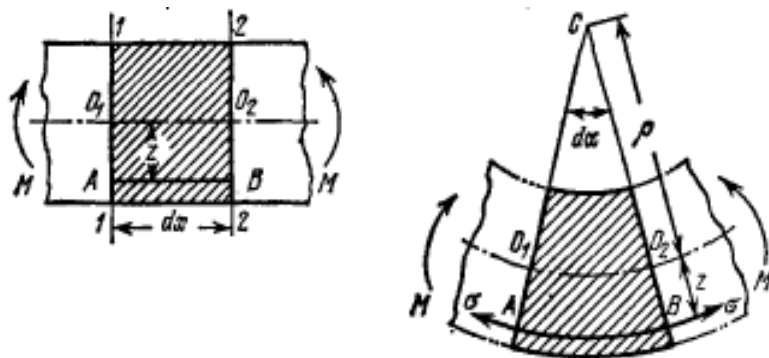


Figure 4. To the theory of bending of the beam freely supported at the ends.

The solution of this problem with respect to the shaft is based on Figure 5, which shows the concentrated force P , which is applied to the right face $x = a$ of the selected element of infinitely small length dx . The view of the element before and after bending is shown in Figure 5.

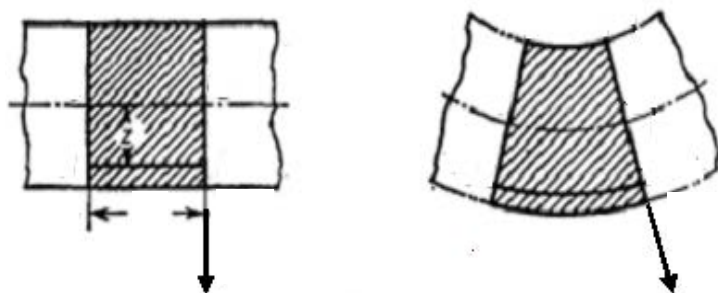


Figure 5. Bending of rope and shaft due to concentrated force.

Applying force P causes the shaft and rope to bend. When bent, the length of the axis O_1-O_2 (Figure 4) of the shaft and rope remains unchanged, and the fibres are compressed above the axis and stretched below. Compressive and stretching forces according to Hooke's law grow when bending increases, compensating for the bending action of force P , the bending process will stop only when the sum of compressive and stretching forces is equal to force P .

From Figure 5 we have $dM(x) = Pdx$ and integrating we get the moment-of-force function P $M(x) = Px + C_1$. But force P creates not one, but two moments with shoulders a and $L_b - a$, and the distribution of force P for two moments is determined by the reaction of supports P_1 and P_2 (Figure 6), which are inversely proportional to the lengths of shoulders [1, 3, 8], in the consequence of which for the left support we have

$$\frac{P_1}{P} = \frac{L_b - a}{L_b},$$

from where

$$P_1 = \frac{P(L_b - a)}{L_b}.$$

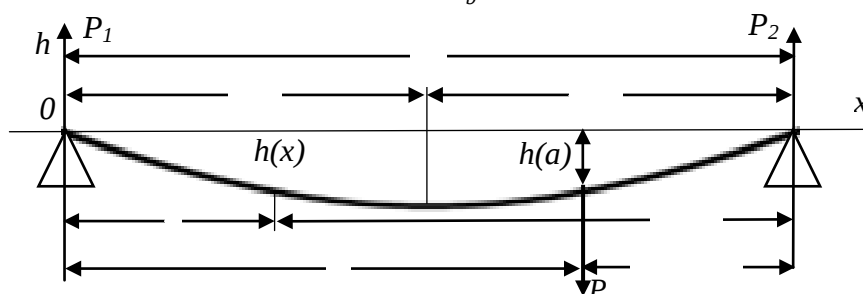


Figure 6. Design diagram of curved round beam.

By substituting P_1 in the moment function, we get

$$M(x) = \frac{P(L_b - a)}{L_b}x + C_1, \quad 0 \leq x \leq a. \quad (1)$$

The integration in Figure 6 extends only to the left side of the shaft, the solution of the problem with respect to the right side remains open. To solve the problem, we will use the same modes of fixing the shaft ends and ideas of mirror symmetry. The demonstration of the idea will be performed on a simple experiment with a string fixed on two supports A and B . Figure 2 shows the string in a pulled-off state and its mathematical description, borrowed from [1] has the form

$$h_1(x) = h_{ot} \cdot \frac{x}{a}, \quad 0 \leq x \leq a \quad (2,a)$$

$$h_2(x) = h_{ot} \cdot \frac{L_b - x}{L_b - a}, \quad a \leq x \leq L_b \quad (2,b)$$

In Figure 2, we introduce the second coordinate system, the origin $x^* = 0^*$ of which is located on the support B , and the x -axis x^* is directed from right to left, opposite the x -axis (we change the sign to the opposite) with the origin on the support O . Based on the similarity of triangles we write

$$\frac{h_1(x)}{h_{ot}} = \frac{x}{a}, \quad \frac{h_2(x)}{h_{ot}} = \frac{x^*}{a^*}, \quad \text{откуда}$$

$$h_1(x) = h_{ot} \frac{x}{a}, \quad 0 \leq x \leq a, \quad (3,a)$$

$$h_2(x^*) = h_{ot} \frac{x^*}{a}, \quad a \leq x^* \leq L_b. \quad (3,b)$$

Figure 2 plots the functions (3, a) line 1 and the functions (3, b) - line 2. The plots are constructed in different coordinate systems and illustrate mirror symmetry, confirming the same operation modes of the supports. To construct the bending functions, the function (3, b) from the coordinate system x^* must be transformed into the system x .

According to [7] subject to conversion

$$x^* = L_b - x, \quad a^* = L_b - a \quad (4)$$

as a result (3) will take the final form

$$h_1(x) = h_{ot} \frac{x}{a}, \quad 0 \leq x \leq a, \quad (5,a)$$

$$h_2(x) = h_{ot} \frac{L_b - x}{L_b - a}, \quad a \leq x \leq L_b, \quad (5,b)$$

which corresponds to the graph of lines 1 and 3 in figure 2.

Using mirror symmetry, we write the bending function in the coordinate system x^*

$$M(x^*) = \frac{P(L_b - a^*)}{L_b} x^* + C_2, \quad a \leq x^* \leq L_b. \quad (5,c)$$

Applying the transformation (4) to (5, c), we obtain

$$M(x) = \frac{P \cdot a}{L_b} (L_b - x) + C_2, \quad a \leq x \leq L_b \quad (5,d)$$

Thus, for (1) and (5, d), we have

$$M(x) = \begin{cases} \frac{P(L_b - a)}{L_b} x + C_1, & 0 \leq x \leq a, \\ -\frac{P \cdot a}{L_b} x + P \cdot a + C_2, & a \leq x \leq L_b. \end{cases} \quad (6)$$

In sources [4, 5], only bending moments are taken into account, taking into account the additional dynamic moment and the moment of internal viscous friction $M_i(t)$, we obtain

$$EJ \frac{d^2 h(x,t)}{dx^2} = M(x) + M(t) - M_c(t) - M_i(t), \quad (7)$$

where according to the source [8] we have

$$M(t) = \begin{cases} (L_b - a)m \frac{d^2 h(t)}{dt^2}, & 0 \leq x \leq a, \\ am \frac{d^2 h(t)}{dt^2}, & a \leq x \leq L_b. \end{cases} \quad (8)$$

$$M_t(t) = C_t L_b \eta D \frac{dh(t)}{dt}, \quad C_t L_b = c_t \quad (9)$$

By substituting (6), (8) and (9) in (7), we obtain a differential equation in the second-order partial derivatives describing the damping oscillations of the shaft and rope

$$EJ \frac{\partial^2 h(x,t)}{\partial x^2} - Am \frac{\partial^2 h_1(x,t)}{\partial t^2} - B \frac{\partial h_1(x,t)}{\partial t} = \begin{cases} \frac{P(L_b - a)}{L_b} x + C_1, & A = L_b - a, \quad 0 \leq x \leq a, \\ -\frac{P \cdot a}{L_b} x + Pa + C_2, & A = a, \quad a \leq x \leq L_b. \end{cases} \quad (10)$$

Constant integration is determined from steady-state mode, when

$$\frac{\partial^2 h(x,t)}{dt^2} = \frac{\partial h(x,t)}{dt} = 0,$$

in this case, the partial differential equation (10) is transformed into an ordinary differential equation

$$EJ \frac{d^2 h(x,t)}{dx^2} = \begin{cases} \frac{P(L_b - a)}{L_b} x + C_1, & 0 \leq x \leq a, \\ \frac{Pa}{L_b} (L_b - x) + C_2, & a \leq x \leq L_b. \end{cases} \quad (11)$$

Integrating twice, we get

$$EJ \frac{dh(x,t)}{dx} = \begin{cases} \frac{P(L_b - a)}{2L_b} x^2 + C_1 x + C_3, & 0 \leq x \leq a, \\ -\frac{Pa}{2L_b} (L_b - x)^2 - C_2 (L_b - x) + C_4, & a \leq x \leq L_b. \end{cases}$$

$$EJh(x) = \begin{cases} \frac{P(L_b - a)}{6L_b} x^3 + C_1 \frac{x^2}{2} + C_3 x + C_5, & 0 \leq x \leq a, \\ \frac{Pa}{6L_b} (L_b - x)^3 + C_2 \frac{(L_b - x)^2}{2} - C_4 (L_b - x) + C_6, & a \leq x \leq L_b. \end{cases}$$

Fulfilling the conditions for fixing the ends of the shaft in the bearings, we obtain the boundary conditions

$$h(0) = h'(0) = h(L_b) = h'(L_b) = 0 \quad (12)$$

we get $C_3 = C_4 = C_5 = C_6 = 0$, the substitution of which gives

$$EJ \frac{dh(x,t)}{dx} = \begin{cases} \frac{P(L_b-a)}{2L_b} x^2 + C_1 x, & 0 \leq x \leq a, \\ -\frac{P \cdot a}{2L_b} (L_b-x)^2 - C_2 (L_b-x), & a \leq x \leq L_b. \end{cases} \quad (13)$$

$$EJh(x) = \begin{cases} \frac{P(L_b-a)}{6L_b} x^3 + C_1 \frac{x^2}{2}, & 0 \leq x \leq a, \\ \frac{P \cdot a}{6L_b} (L_b-x)^3 + C_2 \frac{(L_b-x)^2}{2}, & a \leq x \leq L_b. \end{cases} \quad (14)$$

The integration constants C_1 and C_2 are determined from the condition of continuity and smoothness of the function. For this, the first function from (14) must be equated with the second and the first derivative from (13) and the second with $x = a$. Then, grouping the terms in the obtained equalities, we have

$$\begin{aligned} C_1 a + C_2 (L_b - a) &= -\frac{Pa^2(L_b-a)}{2L_b} - \frac{P \cdot a(L_b-a)^2}{2L_b}, \\ C_1 \frac{a^2}{2} - C_2 \frac{(L_b-a)^2}{2} &= \frac{P \cdot a(L_b-a)^3}{6L_b} - \frac{Pa^3(L_b-a)}{6L_b}. \end{aligned}$$

We multiply the first equation by $\frac{L_b-a}{2}$, add it to the second, and solve the result with respect to C_1

$$C_1 = \frac{P(a^2 - L_b^2)}{6L_b}. \quad (15,a)$$

The first equation is multiplied by $\frac{a}{2}$ and subtract the second from it, resolving with respect to C_2 , we obtain

$$C_2 = \frac{P \cdot a(3aL_b - a^2 - 2aL_b^2)}{6(L_b^2 - aL_b)} \quad (15,b)$$

3. Conclusion

The simulation result is shown in Figure 7. The curves were recorded using the support and research system developed by the author. Figure 7 a) shows the bending of the shaft from the action of the force P applied to the middle of the shaft $a = \frac{L_b}{2}$, in figures 7 b) and 7 c) the force P was applied to the points $a = \frac{L_b}{4}$ and $a = \frac{3L_b}{4}$.

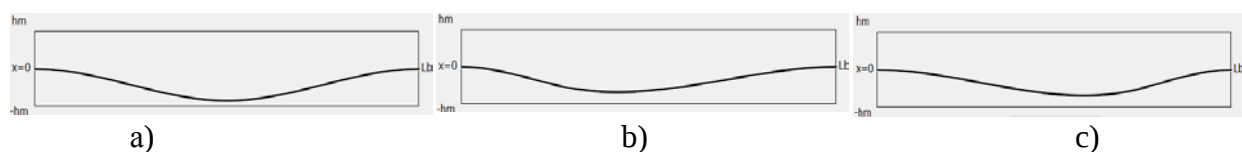


Figure 7. Simulation results of shaft bending from the force P acting at the point $a = \frac{L_b}{2}$ for a), at the point $a = \frac{L_b}{4}$ for b) and at the point $a = \frac{3L_b}{4}$ for c).

Figure 8 shows the experimental curves, and figure 8 a) corresponds to figure 7 a), figure 8 b) to figure 7 b), and figure 8 c) to figure 7 c). A comparison of the figures shows the coincidence of the characteristics of theoretical processes with experimental ones.

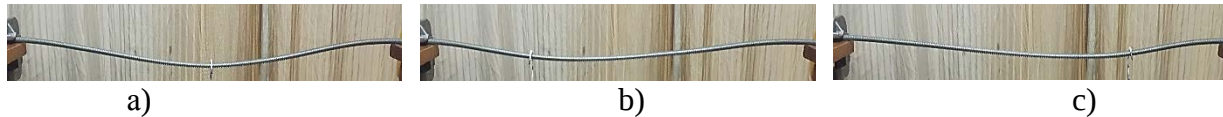


Figure 8. Experimentally obtained bending of the shaft from the force P acting at the point $a = \frac{L_b}{2}$ for a), at the point $a = \frac{L_b}{4}$ for b) and at the point $a = \frac{3L_b}{4}$ for c).

Thus, based on mirror symmetry using two coordinate systems, a universal mathematical model of the transverse oscillations of the shaft and the rope from the action of a concentrated load is created. The model describes forced and free oscillations, as well as a steady state. Accounting for internal viscous friction provides an investigation of damped oscillations.

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