satisfactory accuracy, we can start creation of DTM and ortorectification process. In additional, we must set other parameter – spatial resolution of DTM. This resolution is limited by resolution of aerial photographs. The accuracy obtained by the digital terrain model can be estimated on the basis of large-scale topographic maps, aerial photographs or other more detailed digital model. At small areas we can verify quality of DTM using high-precision GPS receivers or using way of tacheometric removal of known binding. Based on digital terrain models created for each pair of images were executed orthorectification. Such images can be used for precise measurements of position, the creation of various GIS systems, since they account for the relief of the territory and have a spatial geographic location. In the specialized GIS software it is easy to perform modeling and forecasting of various phenomena basing on orthorectificated images, significantly increases the accuracy of various objects spaces measuring after decryption.

Keywords: Digital Terrain Model, coordinate system, Leica Photogrammetry Suite, Ground Control Points, aero photo image, orthorectification.

УДК 621.86.065 ***Prof. M.P. Martyntsiv; assoc. prof. O.M. Udovytskyy – NUFWT of Ukraine***

DESIGN AND ANALYSIS OF WORK OF TRANSPORT SYSTEMS WITH A FLEXIBLE HAULAGE ELEMENT

The mathematical model of transport systems with a flexible element has been developed. Analytical dependences for determination of efforts in the branches of flexible haulage element taking into account the dynamic loadings have been obtained. The analysis of the obtained dependences, which allowed to substantiate optimum rates of movement and parameters of drive of such transport systems, has been carried out.

Conveyers and suspended systems with a flexible reserved element are widely used in different branches of industry and agriculture. They have similar kinematics charts, that is why it is possible for them to develop a general mathematical model. Their basic elements are occasions, transmissions, sending blocks (drums) and haulage element, loaded with external forces, which leans against certain supporting surfaces. Between supports a haulage element sags due to loads and it sown weight. Conveyers of forest and woodworking industry differ in complication of construction, power intensity, considerable inertia of hauling organs and work in heavy operating conditions [1, 2].

The suspended systems of ropes with the mobile reserved rope are widely used for the development of mountain masses with the purpose of their recreation use and wood transportation. Thus they provide conservation of ecological systems which have been formed for many ages [3, 4]. However transport and suspended systems of ropes according to the known methods are calculated for static loadings, and the calculation comes to determination of power of drive and parameters of haulage element, which are determined from the terms of durability. In the process of work of such systems, especially at starting and braking, the considerable dynamic loadings appear both in haulage elements and in the elements of drive, and also in such mode of operation drives pass a resonance area [5]. In the known works [1, 5, 11-17] the models for the research of the phenomena in the machines of continuous transport have the appearance of the open systems with concentrated masses. But a haulage element is the closed circuit. It substantially influences the changing of efforts in the elements of the system, especially at the unstated mode of operation. Oscillation in the mechanical system of drive has been considered in a number of works [6, 7, 8]. For the development of mathematical model the calculation chart of fig.1 has been made.

Equation of motion of the system can be written down as Lagrangian's equations of the second type:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_j}\right)-\frac{\partial T}{\partial q_j}+\frac{\partial \Pi}{\partial q_j}\frac{\partial \Phi_i}{\partial \dot{q}_j}=Q_j,\ (j=1,2,3,4,5,6), \tag{1}$$

where: q_j – generalized co-ordinates of the system (it is accepted for the generalized co-ordinates: $q_1=\square\varphi_1$; $q_2=\square\varphi_2$; $q_3=\square\varphi_3$; $q_4=\square\varphi_4$, $q_5=x_к$, $q_6=y_к$); φ_i – turning angles of the mobile masses; j – an amount of the generalized co-ordinates; $x_к, y_к$ – linear co-ordinates of characteristic points; T, Π – accordingly kinetic and potential energies of the system; Φ_i – Relay's dissipative function; i – an amount of mobile masses of the system; Q_j – generalized external force.

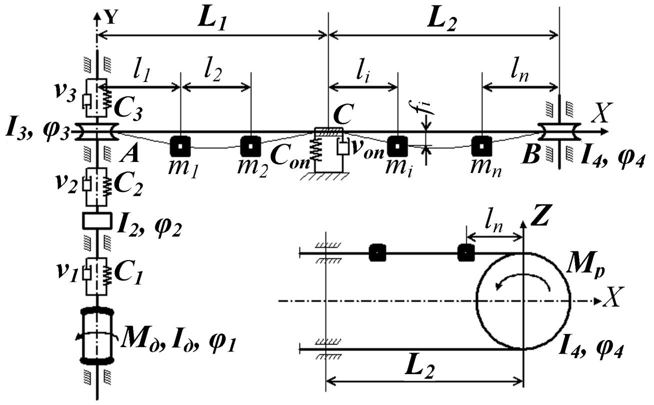


Fig. 1. Calculation chart of the transport system with a flexible haulage element

For determination of kinetic energy of the system we will use the dependence:

$$T=\frac{1}{2}\sum_{i=1}^{n}\left(I_i\omega_i^2+m_i\dot{x}_i^2\right)\int_l F_x\cdot d_x+T_{to} \tag{2}$$

where: m_i – masses of loads which are moved by a transport system; I_i – moments of inertia of the proper masses; F_x – tension of the haulage element; x – moving co-ordinate; T_{to} – kinetic energy of haulage element.

We will define the magnitude of kinetic energy of the haulage element from the dependence:

$$T_{to}=\frac{m_t\cdot L}{4}\sum_{i=1}^{\infty}\left(\dot{x}_i\right)^2 \tag{3}$$

where: m_t – mass of running meter of the haulage element; L – length of the haulage element.

Tension of the haulage element F_x is variable in time. For its determination a calculation chart has been considered (fig. 2).

Taking into account the weight of haulage element and possibility of the origin of both transversal and longitudinal vibrations, equation of motion, which can be presented in the following way, has been obtained:

$$\int_x^{x+dx}\frac{m_t}{g}\left[\frac{d}{dt}z_x(x,t)-\frac{d}{dt}z_t(x,t)\right]\cdot dx=\int_t^{t+dt}F_{(x)}\left[z_x(x,t)\right]\cdot dt+\int_x^{x+dx}\int_t^{t+dt}F_{(x)}\left[z(x,t),z_x(x,t),z_t(x,t),t\right]\cdot dxdt \tag{4}$$

where: g – acceleration of the free fall; $z_x\ (x,\ t)$ – deviation from the position of intersection equilibrium with the co-ordinate of x in the arbitrary moment of time of t.

If we marc the rate of movement of haulage element as ϑ, it is possible to write down:

$$\frac{d_x}{dt}=\vartheta\frac{\partial}{\partial x}+\frac{\partial}{\partial t},\ \frac{dx^2}{dt^2}=\vartheta^2\frac{\partial^2}{\partial x^2}+2\vartheta\frac{\partial^2}{\partial x\partial t}+\frac{\partial^2}{\partial t^2}. \tag{5}$$

Then equation (4) can be presented as equation: $z_{tt}-2\vartheta\cdot z_{xt}-\left(\frac{F_k}{m_t}-\vartheta^2\right)z_{xx}=0$ (6)

We will accept the followings statements:

$$z(x,t)_{|x=0} = z(x,t)_{|x=l} = 0;\ z_x(x,t)_{|x=0} = z_x(x,t)_{|x=l} = 0. \quad (7)$$

Then the solution of equation (6) can be presented in the following way:

$$z(x,t) = C_1 \cos(kx + \omega t + \varphi) + C_2 \cos(\chi x - \omega t - \psi), \quad (8)$$

where; C_1, C_2 – amplitudes of oscillation of the hauling element; φ, ψ – initial phases of oscillation; k, χ – wave numbers; ω – frequency of oscillation of the haulage element.

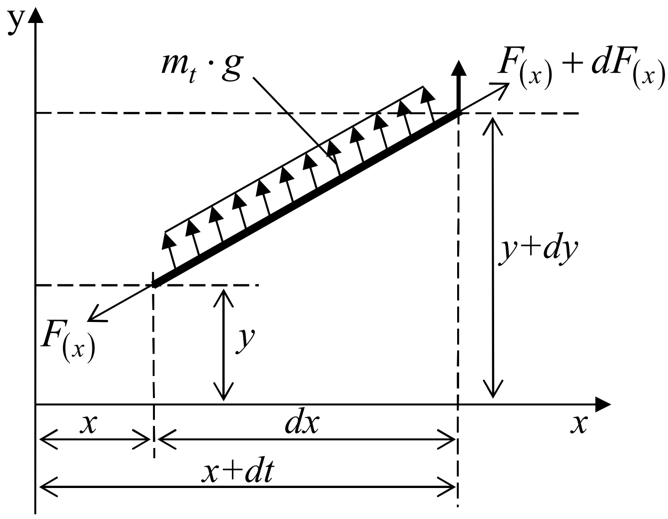


Fig. 2. Chart of the efforts having affect on a haulage element

To get the solution we will use the method of D'alamber of the construction of equations solutions of hyperbolical type [9] in obedience to which there is a certain dependence between frequency of vibrations and wave numbers [10]:

$$\left.\begin{aligned}\omega^2 + 2\vartheta k\omega - (\alpha^2 - \vartheta^2)\cdot k^2 = 0\\ \omega^2 - 2\vartheta\chi\omega - (\alpha^2 - \vartheta^2)\cdot \chi^2 = 0\end{aligned}\right\} \text{ where: } \alpha \text{ - phase of vibrations; } \alpha^2 = \frac{F_{(x)}}{m_t} \quad (9)$$

Taking into account the accepted boundary conditions, unknown parameters can be defined from the followings expressions:

$$k = \frac{k\pi}{\alpha L}(\alpha + \vartheta);\ \chi = \frac{k\pi}{\alpha L}(\alpha - \vartheta);\ \omega = \frac{k\pi}{\alpha L}(\alpha^2 - \vartheta^2);\ \varphi = -\psi;\ C_1 = -C_2 = \alpha;\ C_1 = \frac{k}{\chi}, C_2 = \alpha. \quad (10)$$

Knowing the indicated coefficients for the concrete transport systems and using equations (7) and (8), it is possible to find the laws of change of force F_x and its value.

The laws of change of vibration frequency and wave numbers of the mobile haulage elements depending on the rate of movement are shown on fig.3.

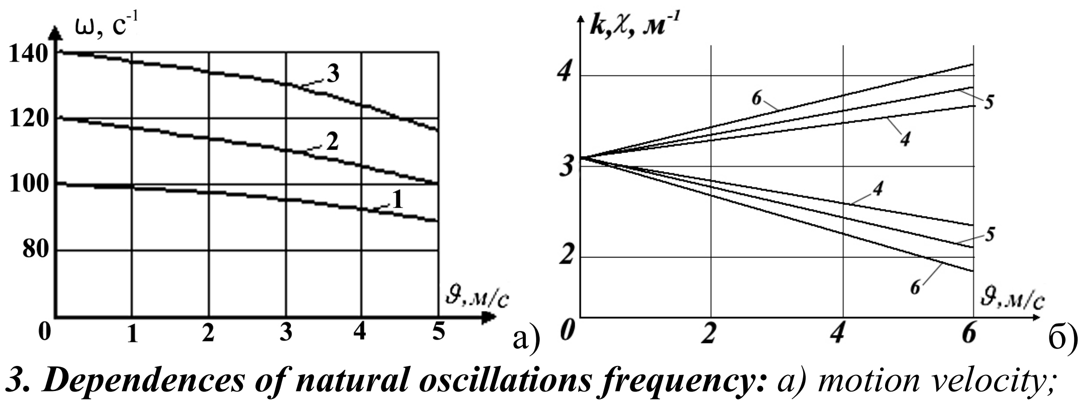


Fig. 3. Dependences of natural oscillations frequency: *а) motion velocity; б) wave numbers: 1, 6 – at α^2=1000; 2,5 – at α^2=1500; 3, 4 – at α^2=2000*

Expression for potential energy of the system looks like:

$$\Pi = \sum_{i=1}^{n} C_i(\varphi_i - \varphi_{(i+1)})^2 + \frac{1}{2}C_t(\varphi_n \cdot r_n - x)^2 \quad (11)$$

where: C_i, φ_i – accordingly inflexibility and turning angle of i-element of the system; C_t – inflexibility of the haulage element,

$$C_t = \frac{E_t \cdot A_t}{l_t} = \frac{E_t \cdot A_t}{l_{ot} + \varphi_n \cdot r_n} \tag{12}$$

where: E_t, A_t – accordingly modulus of elasticity and cross-section area of the haulage element; l_t – length of the haulage element; φ_n, r_n- accordingly turning angle and radius of drive drum (pulley, sprockets).

Dissipative function is presented in the following way:

$$\Phi = \frac{1}{2}\left[\sum_{i=1}^{3} \nu_i\left(\omega_i - \omega_{(i+1)}\right)^2 + \nu_б\left(\omega_3 \cdot r_3 - \vartheta_t\right)^2 + \nu_{on}\left(\omega_n \cdot r_n - \vartheta_t\right)^2\right] + \sum_{i=1}^{n}\left(m_i + m_t \cdot l_i\right)\eta_k \tag{13}$$

where: ν_i, ω_i – accordingly linear resistance of motion and angular velocity of the movement of separate elements; $\nu_б$, ν_{on} – coefficients of linear resistance of the driving element and supports; r – radius of the driving element; η_k – friction coefficient.

Generalized external force is expressed by the dependence:

$$Q_j = \sum_{i=1}^{n} m_i\left(g + \frac{d\vartheta_t}{dt}\right) + m_t l_t \cdot \frac{d\vartheta_t}{dt}. \tag{14}$$

We will accept the generalized speeds: $\omega_1 = \dot{\varphi}_1;\ \omega_2 = \dot{\varphi}_2;\ \omega_3 = \dot{\varphi}_3;\ \vartheta = \dot{x}$.

Then: $$\frac{\partial T}{\partial \dot{\varphi}_1} = 0;\ \frac{\partial T}{\partial \dot{\varphi}_2} = 0;\ \frac{\partial T}{\partial \dot{\varphi}_3} = \frac{1}{2}\frac{dI_3}{d\varphi_3} \cdot \omega_3^2.$$

The rate of movement of the haulage element can be defined, proceeding from the hypothesis about the linear law of velocity distribution: $\vartheta_t = a + b\xi.$ (15)

For determination of two unknown quantities ***a*** and ***b*** there are two initial conditions: $\xi = 0;\ \vartheta_t = 0; \xi = x(t);\ \vartheta_t = \dot{x}.$

After transformations we will have: $$\vartheta_t = \frac{\dot{x}_{(t)}}{x}\xi \tag{16}$$

The study of the unsteady modes of operations of transport systems has practical value. In the case when at the start of the drive speed of rotation of engine operating shaft changes in a narrow scope, it is possible to apply linearized characteristics of engines, that substantially simplifies the research task. Then the moment of asynchronous electric engine within nominal speed of rotation of a rotor is determined from the differential equation [6]: $$\frac{dM_\partial}{dt} + \frac{1}{\partial\tau}M_\partial + \frac{1}{\upsilon \cdot \omega_0 \cdot \partial\tau}\dot{\varphi}_1 = \frac{1}{\upsilon \cdot \partial\tau} \tag{17}$$

where; M_∂ – a circulating moment of the engine; $\partial\tau$ – a component of time which takes into account transitional processes in an engine; υ – a coefficient of steepness of static characteristic; ω_0 – angular speed of the idle running.

The moment of an engine at every step of motion equation integration is determined according to the values of the mechanical system characteristics. In order to get time dependences of change of efforts in the haulage element, movements and rates of movement the rate of relative deformation will be determined from the expression:

$$\xi = \left(\eta \cdot L - r_в \cdot \delta \cdot \omega\right)/L^2 \tag{18}$$

where: η – speed of absolute deformation of haulage element; $r_в$ – a radius of driving drum (block, sprockets, pulley). Using the given method, it is possible to carry out numerical calculations. With this aim, having defined kinetic and potential energies, dissipative function and potential force, Lagrangion's equation (1) can be presented as a system of Koshi's equations. Such system can be solved by the numerical method of Hause with the help of the package of software «Mathematics for Windows».

As an example we will show a solution, when the initial conditions are:

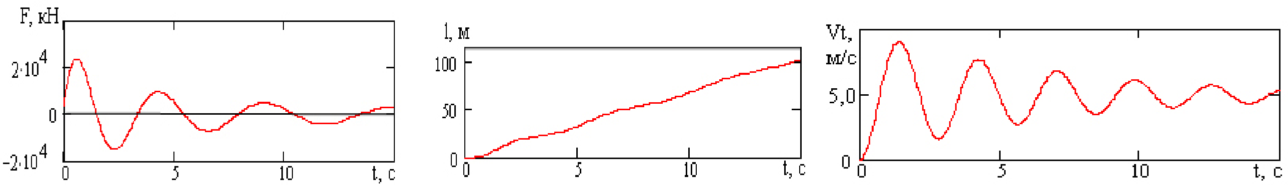
$$x(0)=0;\ \vartheta(0)=\vartheta_0;\ \ \omega_1=\dot{\varphi}_1;\ \omega_2=\dot{\varphi}_2;\ \omega_3=\dot{\varphi}_3;\ \vartheta=\dot{x};\ \ \frac{d\vartheta}{dt}=\frac{1}{m}(F_1-F(x)).$$

Find force F_1 from the expression:

$$F_1=F_{1c}+F_{1\mu} \qquad (19)$$

where: $F_{1c}=C_0\delta_i$ – elastic force; $F_{1\mu}=\mu_0\cdot\upsilon_i$ – force of linear resistance; C_0, μ_0 – inflexibility and linear resistance coefficient of haulage element of unit length; δ_i, υ_i – relative deformation of branch and rate of its variation in time.

Standard cable of such parameters: diameter $d_к$ = 12,0 mm; length L=500 m; linear mass $m_к$ = 2 kg/m; rate of movement ϑ=5 m/c; is accepted as a haulage element. Graphs of change of forces, movement and velocity in a branch of the cable is shown on fig. 4.



а) *б)* *в)*

***Fig. 4. Time dependences of deformations and speed change in a branch of the cable:** а) forces; б) movement; в) velocity*

From the graphs one can see, that at the beginning of motion a dynamic component of an effort in a rope is the most intensive. During small time interval this component disappears, and the effort in the cable is set. Thus, the greater weight of loads, the greater period of vibrations of efforts in the cable and the slower vibration damping. The danger of resonance emergence does not exist for such systems. The above-mentioned methods allow to substantiate optimum parameters of transport systems depending on their operating conditions and structural features.

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Моделювання та аналіз роботи транспортних систем з гнучким тяговим елементом

Конвеєри та підвісні системи з гнучким замкнутим елементом широко використовуються в різних галузях промисловості та сільського господарства. Вони мають аналогічні кінематичні схеми, тому для них можна розробити спільну математичну модель. Конвеєри лісової та деревообробної промисловості відрізняються складністю конструкції, енергоємністю, значною інерційністю тягових органів і працюють у важких експлуатаційних умовах. Однак транспортні та підвісні канатні системи за відомими методиками розраховуються на статичні навантаження, а розрахунок зводиться до визначення потужності привода і параметрів тягового органа, які визначаються з умов міцності. В процесі роботи такі системи, особливо при пуску і гальмуванні, виникають значні динамічні навантаження як в тягових органах, так і в елементах приводів, а також в таких режимах роботи приводи проходять резонансну зону. Для дослідження явищ у машинах безперервного транспорту слід враховувати, що тяговий орган представляє собою замкнений контур. Математична модель для вивчення транспортних систем з гнучким елементом пропонується на основі складеної розрахункової схеми та застосування рівнянь Лагранжа. Отримано аналітичні залежності для визначення зусиль у вітках гнучкого тягового органу з врахуванням динамічних навантажень. Виконано аналіз отриманих залежностей, який дозволив обґрунтувати оптимальні швидкості руху та параметри привода таких транспортних систем.

УДК 630*165 ***Prof. Dr. M.P. Horoshko – NUFWT of Ukraine***

METHODICAL PRECONDITIONS FOR EVALUATION OF GROWING PROCESS OF MIXED SPRUCE FORESTS IN CARPATHIANS

It should be stressed that majority of forest assessment researches concern studies the pure forest stands. They do not take into account to full extent the structure of mountain landscape, estimation of homogeneity of research material was not always presented. It is proposed to use the type of the forest in the research process for growth of mixed stands. Additionally of large-sized stems growth was analyzed. Two categories of stands were selected depending on tending intensity: normal and modal. Growing process was analyzed according to the forest elements by the analytical smoothing of results for trees measuring on sample plots and by using of plural regressive analysis methods. It is proposed to use logistic equation for estimation of forest assessment indexes dynamics. This equation was proved to represent biological essence for natural process of growth to fuller extent when considering these two forest measurement indexes. Correspondence between regression equations and experimental data was estimated by the determination coefficient (R^2), for additional estimation the Fisher criterion (F) under probability P=0,95 was used. Absolute values of logistic equation coefficients and biometrical evaluations of connection were presented.

Keywords: Carpathians, mixed spruce, normal and modal stands.