At the same time, the packet delay in the switching node should not exceed the allowable value. Excessive packet delay results in the loss of relevance of transmitted messages and, consequently, in the loss of intelligence data. The proposed method of eliminating excessive delay involves the formation of two queues of packets in the switching node. Withdrawal of service packages from two queues is carried out using a new adaptive procedure.

An adaptive switch has been introduced into the structure of the switching node, designed to withdraw packets from queues to the packet service system. When selecting a packet queue, the adaptive switch compares the number of service requests from each queue. The method assumes that the number of service requests from the fast packet flow queue determines the number of packets that are in the queue. The number of requests that are formed from the queue of slow-flow packets has two components: the number of real and the number of conditional requests. Actual applications take into account the number of packets in the queue.

Real requests take into account the number of packets in the queue. Conditional requests take into account the delay time, since for each missed cycle of the packet's withdrawal from the slow flow queue, the adaptive switch generates a conditional request. Therefore, the number of requests for a slow stream grows even without packets entering the queue. A comparative study of options for implementing the adaptive procedure has been carried out. The proposed method for eliminating excessive packet delay in the switching node provides an increase in the efficiency of the functioning of a special-purpose information network as a whole.

**Keywords:** switching node, information communication system for special purposes, probability of packet loss, adaptive decommissioning of packets for service.

UDC 621.867.52 DOI: https://doi.org/10.33577/2312-4458.26.2022.12-16

I. Vrublevskyi

*Hetman Petro Sahaidachnyi National Army Academy, Lviv*

**Article history:** Received 10 March 2022; Revised 11 April 2022; Accepted 2 May 2022

**PROSPECTS OF APPLICATION OF VIBRATORY DEVICES IN MILITARY EQUIPMENT**

Possibilities of employing vibratory devices as conveying mechanisms in some models of military equipment, in particular in engineering weapons, are considered. Vibratory conveyors, which are widely used in industry for moving products over short distances, are mainly devices with one-component rectilinear oscillations, in which at the high conveying velocity the modes of movement with hopping are implemented, that is unacceptable for conveying many military facilities, including weapons and explosive objects. Vibratory conveying in non-hopping modes (without tossing) at high speed is possible when using independent (two-component) horizontal and vertical oscillations of the working body of the vibratory conveyor. Based on the analysis of the influence of oscillation parameters on the velocity and stability of vibratory movement of objects in non-hopping modes, a conclusion is made about the feasibility of using horizontal harmonic oscillations in near-resonant mode and vertical oscillations with piecewise constant acceleration. The equations describing the optimal law of vertical oscillations, which consists of two segments of parabolas during the period of oscillations, are derived. A constructive scheme of the vibratory conveyor based on a three-mass oscillating system with an unbalanced vibratory drive of horizontal oscillations and eccentric vibratory drive of vertical oscillations is proposed. The working body of the vibratory conveyor is connected to the intermediate frame by an elastic system of vertical oscillations, the stiffness of which is adjusted to a mode far from resonance. The reactive frame is connected to the intermediate frame by an elastic system of horizontal oscillations, the stiffness of which provides near-resonant mode. The presented vibratory conveyor significantly enhances the conveying velocity in non-hopping modes, which are required for conveying explosive objects, as well as increases the lifting angles and reduces the vibration transmitted to the environment, which will facilitate its use in military machines. It also outlines a number of issues that need to be addressed in the design and implementation of such devices in military equipment.

**Keywords:** vibratory conveyor, independent horizontal and vertical oscillations

**Statement of problem**

Military engineering equipment often uses mechanisms for conveying the samples of weapons and other military loads over relatively short distances. They include, in particular, the mechanism for moving mines in the tracked barrier, means of cargo transportation in mechanized warehouses, etc. [1,2]. As conveying devices, they often use belt or chain conveyors, which are equipped with an electric motor. They are quite bulky and energy-intensive, require quite high maintenance and repair costs.

Vibratory conveyors widely used in industrial enterprises [3,4] are devoid of such shortcomings. However, available mass-produced conveyors are rarely suitable for
exploitation in military equipment, especially for the transportation of explosive cargo. Since a large horizontal amplitude of oscillations of the conveyor’s working body is required to achieve high conveying velocity, in such devices its increase leads to the vertical component of the amplitude increase, which leads to moving modes with hopping, that is often unacceptable for conveying military cargo. For their conveying it is advisable to use vibratory conveyors with independent drives of horizontal and vertical oscillations of the working body. They allow one to obtain high conveying velocity with a low amplitude of vertical oscillations, which provides a non-hopping mode of transported objects. It also significantly reduces the transfer of the conveyor’s vibration to the components of the machine on which it is installed. Therefore, the development of the design of vibratory conveying devices, suitable for transporting explosive and other military cargos at high velocity is greatly demanded.

Analysis of recent research and publications

The process of vibratory conveying and devices, which perform it, is considered in many scientific papers. In particular, theoretically and practically proved that the use of independent horizontal and vertical oscillations of the conveying surface, for example, elliptical oscillations, at certain ratios, can significantly increase conveying velocity, especially in non-hopping modes of movement [4-6]. The author’s works, devoted to the study of vibratory conveying devices with independent horizontal and vertical oscillations of the working body, show the reasonability of using the vertical oscillations with piecewise constant acceleration (linear piecewise velocity) [7,8]. However, when developing the design of conveying devices that implement such oscillations, there are multiple problems to be solved.

The purpose of article

The purpose of the article is to substantiate the new design of the vibratory conveying device with independent drives of horizontal and vertical oscillations, which provide a high velocity in non-hopping modes and large angles of lifting loads. The design of the vibratory conveyor should provide the ability to move explosive cargos with high productivity.

The main material presentation

Consider the process of vibratory conveying with independent harmonic horizontal oscillations (coordinate x) and vertical oscillations (coordinate y) with piecewise constant acceleration (Fig. 1, a), the law of their variation is described as follows [7]

\[ x = A_x \cdot \sin(\omega t + \varepsilon), \quad (1) \]

where \( A_x \) is the amplitude of horizontal oscillations, \( \varepsilon \) is phase difference angle, \( \omega \) is frequency of forced oscillations, \( A_y \) is amplitude of vertical oscillations, \( g \) is gravitational acceleration, \( t \) is time, \( n \) is parameter of vertical oscillations asymmetry, i.e. the ratio of the acceleration amplitudes when moving up to that moving down. Condition for ensuring the non-hopping mode of the vibratory conveying \( \xi \leq 1 \), and the maximum velocity is reached at \( \xi = 1 \). By integrating equations (2) twice, we obtain the equations for the variation of oscillation velocity (Fig. 1, b) and displacement (Fig. 1, c) in the vertical direction in a form of two segments of parabolas during the period of oscillations:

\[
\eta = \frac{y}{A_y}, \quad \xi = \frac{\omega_y^2 t^2}{g \cos \alpha}, \quad \tau = \frac{\pi}{n + 1},
\]

and

\[
\eta = \frac{y}{A_y}, \quad \xi = \frac{\omega_y^2 t^2}{g \cos \alpha}, \quad \tau = \frac{\pi}{n + 1},
\]

where \( A_x \) is amplitude of horizontal oscillations, \( \varepsilon \) is phase difference angle, \( \alpha \) is inclination angle of conveying plane to the horizon, \( \omega \) is frequency of forced oscillations, \( A_y \) is amplitude of vertical oscillations, \( g \) is gravitational acceleration, \( t \) is time, \( n \) is parameter of vertical oscillations asymmetry, i.e. the ratio of the acceleration amplitudes when moving up to that moving down. Condition for ensuring the non-hopping mode of the vibratory conveying \( \xi \leq 1 \), and the maximum velocity is reached at \( \xi = 1 \). By integrating equations (2) twice, we obtain the equations for the variation of oscillation velocity (Fig. 1, b) and displacement (Fig. 1, c) in the vertical direction in a form of two segments of parabolas during the period of oscillations:

\[
y = \begin{cases} 
-\frac{\omega^2 t^2}{2} - C_1(\omega t), & -\frac{\pi}{2} - \tau < \omega t < \frac{3\pi}{2} - \tau \\
\frac{n \omega^2 t^2}{2} + (C_1 + C_2) \omega t + C_1, & \frac{3\pi}{2} - \tau < \omega t < \frac{3\pi}{2} + \tau 
\end{cases}
\]

where \( w = \frac{\omega^2 \cos \alpha}{2} \).

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The values of the integration constants are determined from the condition of continuity of the function \( y(t) \), namely:

\[
C_1 = \frac{\pi}{2}, \quad C_2 = \frac{\pi}{2} (3n + 1), \quad C_3 = \frac{\pi^2}{8} \frac{(3n + 1)^2}{n + 1}.
\]

The optimal value of the phase difference angle \( \alpha \) depends on several dimensionless parameters [9]. In high-speed vibratory conveying, when the amplitude of horizontal oscillations is much larger than that of vertical oscillations, the optimal values of \( \alpha \) are in the range of 80°-90°.

The velocity of vibratory conveying \( v \) can be found by the formula [4]:

\[
V = A_{x} \omega K,
\]

where \( K \) – dimensionless coefficient of velocity, which depends on several dimensionless parameters, and first of all on the law of conveying surface oscillations. Obviously, the velocity decreases when angle \( \alpha \) increases. The dependence of dimensionless velocity \( K \) on dimensionless inclination angle parameter \( a = \frac{\tan \alpha}{\mu} \), (where \( \mu \) – friction coefficient of conveying loads) for the law of oscillations, described by equations (1)-(2), for different values of parameter \( n \) from 1 to 9, is shown in Fig. 2 [7].

\[Fig. 2. \text{Dependence of dimensionless velocity } K \text{ on inclination angle parameter } a \text{ for different values of } n.\]

The larger is the value of \( n \), i.e. the higher the asymmetry of the vertical oscillations, the higher velocity can be obtained. For comparison, in Fig. 2 the dashed line shows the corresponding dependence with the harmonic law of vertical oscillations. Therefore, the advantages of vertical oscillations with piecewise constant acceleration are obvious, especially at large angles of elevation \( \alpha \). The value of the parameter \( n \) is limited by the design of the elastic system of normal vibrations, the springs of which must be ensured for strength at maximum acceleration.

Fig. 3 shows the design scheme of a vibratory conveyor with independent drives of horizontal and vertical oscillations, which are implemented according to the law (1), (3). Conveyor consists of a working body 1 of the mass \( m_1 \), which is connected to the intermediate frame 3 of the mass \( m_3 \) by means of an elastic system 5 with sufficient flexibility only in the vertical direction, and the stiffness \( c_y \) of which is adjusted such that the natural oscillation frequency is 40-60% higher than the forced frequency. The reactive frame 2 of the mass \( m_2 \) is connected to the intermediate frame 3 by means of an elastic system 4 with sufficient flexibility only in the horizontal direction, and stiffness \( c_x \) of which provides near-resonant mode (natural frequency should be 5-6% higher than a forced frequency). Between the reactive frame 2 and intermediate frame 3, the unbalanced vibratory exciter 6 of horizontal oscillations is located, the applied force of which varies according to the harmonic law. The eccentric vibratory exciter 7 of vertical oscillations, the force of which \( f_y \) should provide the law of vertical oscillations (Fig. 1, c) with a piecewise constant acceleration described by equations (3), is located between the working body 1 and frame 3. The accuracy of this law is achieved by appropriate calculation and manufacture of the cam of the eccentric drive. Frames 2 and 3 are installed at the destination by means of vibration isolators 8, which minimize the transmission of vibration to the environment or elements of military devices on which the conveyor is mounted. Stiffness \( c_1 \) should be much lower than stiffness \( c_x \) and \( c_y \). For the elastic systems 4 and 5, it is advisable to use the leaf springs with rectangular cross-sections. The natural frequencies of horizontal and vertical oscillations are provided by the corresponding calculation of stiffness \( c_x \) and \( c_y \) [10].

\[Fig. 3. \text{Design scheme of the vibratory conveyor with independent horizontal and vertical oscillations of the working body.}\]
The considered vibratory conveyor will provide optimal horizontal and vertical oscillations of the working body, which will significantly increase its productivity, as well as the angles of lifting loads comparable to available designs of commercially produced vibratory conveyors. However, its implementation in engineering weapons will require a number of issues and implies practical challenges. First, one has to ensure the same frequency of forced oscillations when using essentially different types of vibratory drives to implement the optimal laws of oscillation of the working body. When using such a vibratory conveyor, for example, in a mine tracked barrier, it is necessary to provide the synchronization of the speed of mine movement with the speed of the barrier itself, i.e. the amplitude of horizontal oscillations should vary according to the variation of the machine speed. The rotational motion of unbalanced 6 and eccentric 7 drivers must be applied at the same angular velocity. It is also necessary to provide the variabilities of the natural frequency by adjusting the stiffness of the elastic system of the conveyor. 

Conclusions

The process of vibratory conveying with independent harmonic horizontal oscillations and vertical oscillations with piecewise constant acceleration and maximum asymmetry has been studied. Equations describing the parameters of the law of vertical oscillations, in the form of two segments of parabolas during the period of oscillation, that provide the maximum conveying velocity in non-hopping modes of movement, are derived. The design scheme of a three-mass vibratory conveyor with the independent unbalanced drive of the harmonic horizontal oscillations and eccentric drive of vertical oscillations, varying according to the optimal law with piecewise constant acceleration, has been developed. This allows for increasing the conveying velocity and lifting angles.

We expect that after further optimization the proposed vibratory conveyor can be implemented in mechanisms for transportation of machine parts, weapons, or explosive materials in military depots as well as in engineering equipment.

References


Вступ

Постановка проблеми в загальному вигляді та аналіз літератури. Перспективним напрямом визначення складових сили опору повітря є підхід, заснований на відновленні їх аеродинамічних коефіцієнтів на основі експериментальних даних балістичних стрільб, який полягає в тому, що значення коефіцієнтів визначається на основі наблідуваного та/або точного рішення рівнянь просторового руху снаряда за результатами вимірювання параметрів польоту снаряда [1, 2]. При проведенні балістичних стрільб застосовуються як універсальні, так і спеціальні методи вимірювань, які використовують обладнання, спеціально розроблене для вимірювання параметрів польоту снаряда із заданими технічними, метрологічними та експлуатаційними характеристиками [3, 4]. Визначення параметрів польоту снаряду, а саме координат і швидкості польоту снаряда, розробка методів приведення швидкості до дульного зрізу ствола гармати та методів її нормалізації, являє собою дозвол відому задачу зовнішньої балістики, який присвячена значна кількість підходів, починаючи від класичного методу вимірювання часу проходження снарядом ділянки траєкторії визначеної довжини до сучасних методів, які базуються на рентгенометричних, радіо- і лазеролокаційних методах [2-5]. Водночас провідними науковцями балістичної дослідної лабораторії США оприлюднені точності показники, які досягнути при визначенні аеродинамічних коефіцієнтів, які не повною мірою відповідають вимогам до точності розрахунку траекторії польоту снаряда і, відповідно, точності складання Таблиць стрільби [3].