Nat. Volatiles & Essent. Oils, 2021; 8(5): 10611 - 10620

NVEO
Natural Volatiles & Essential Oils

The Effect of the Stiffness of a Compound Roller Chain Drive Tension of the Driven Branch

A'zam A. Mamakhonov

Namangan institute of engineering and technology, Uzbekistan

Abstract

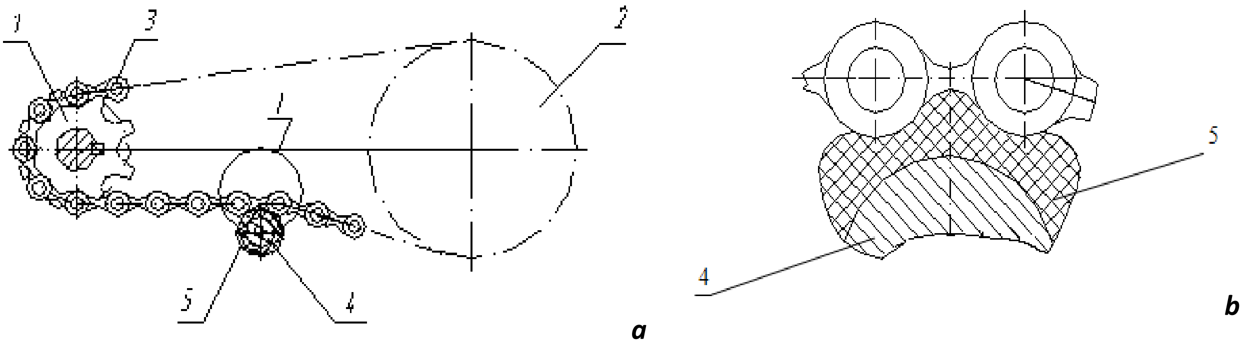
The article proposes a new design of a chain drive with the flexible element tensioning device. The formula for calculating the stiffness of the tension bushing of the tension roller of the chain drive with the compressive tensioning device is obtained. The dependences of the elastic bushing coefficient on the radius of the tensioning roller and the change in the coefficient of friction are obtained. The laws of the chain vibration, velocity and acceleration with respect to time depend on the angular velocity of the tension roller, the mass of the chain interacting with the tension roller and the radius of the tension roller. Recommended parameters have been developed.

Keywords: chain drive, tension roller, flexible bushing, tension, stiffness, rubber, vibration, angular velocity, angular acceleration, friction, coverage angle, mathematical model, vibration frequency, vibration amplitude, chain mass

Introduction

The main disadvantage of the existing chain drive is the rapid wear and noisy operation of the working profiles during the operation. This leads to a decrease in transmission reliability and efficiency. When the chain drive operates at high speeds, the transverse and split vibrations of the chain increase. This increases the unevenness of the output shaft movement. Therefore, the main task of the research was to reduce the unevenness of the oscillating motion of the extension lead network and at the same time improve the adhesion of the chain links with the star teeth. When the chain drive is used without a tensioning device, noise and chain breakage are observed due to the impact of the chain rollers and sprocket teeth[1-7].

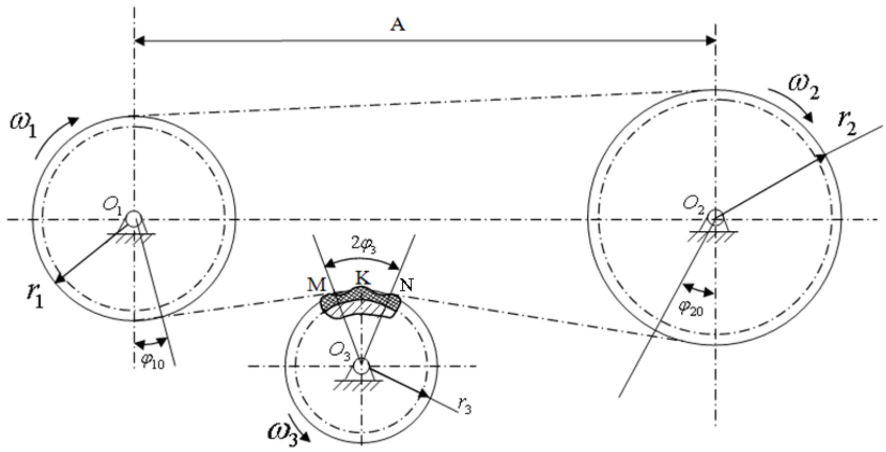
Development of the effective tensioning device of the chain drive design. In order to ensure the smooth operation of the chain drive, increase its durability, a new tensioning device transmission design was recommended. A schematic diagram of the chain drive is shown in picture 1. The structure consists of a drive 1 and driven sprocket 2, a chain 3 attached to them, and a structural tension roller 4. The compression tension roller consists of 4 main shafts and a bushing 5 made of elastic (rubber) material covered on it [1]. The surface of the rubber bushing 5, i.e. the part affected by the chain 3, forms a curved surface in sinusoidal law. This curved surface pitch is taken to be equal to the pitch between the joints of chain 3. During chain drive operation, the motion is transmitted from the drive sprocket 1 to the sprocket 2 driven by the chain 3. In this case, the flexible bushing 5 of the tension roller 4 provides the tension of the guide arm of the chain 3. This increases the angle of coverage of the sprockets 1 and 2 with the chain 3, regulates the displacement (coolness) of the chain 3 and provides the desired tension of the chain 3. Also, in the interaction of the rubber bushing 5 and the chain 3, its transverse vibrations are extinguished and their wear is reduced. Since the sinusoidal curved surface of the rubber bushing 5 coincides with the step of the chain 3 joints, it provides the required friction between them. It is possible to replace the tension roller when the existing chain drive is needed.



***a* is the general scheme chain drive, and *b* is the interaction scheme of the tension roller with the chain**

Pic. 1. Composite tension roller chain extension scheme

The design works in the following order: The rotational motion is transmitted from the drive sprocket 1 to the driven sprocket 2 via chain 3. The motion is then transmitted to the shaft 7 by means of an asterisk 2 to 8 rubber bushing 5 and a metal bushing 6. In this case, the complex oscillations of the resistance torque in the working body affect the rubber bushing 5 through the shaft 7 and the bushing 6. Depending on the elastic-dissipative properties of the rubber 5, the complex oscillations of the resistance moment are sufficiently extinguished, i.e. the effect on the asterisk 1 through the asterisk 2 and the chain 3 is reduced sufficiently. It is recommended to use this chain drive at large range speeds. The design extends slightly the life of the transmission. During the operation, the flexible bushing in the tension roller is deformed, which slightly flattens the loads in the vibration of the chain [8-16]. This degree of the adjustment depends mainly on the coefficient of elasticity of the flexible bushing [11]. However, the increase in the deformation of the flexible bushing leads to failure of the supports, increasing the friction between the chain elements and the flexible bushing, which leads to a decrease in the tensile strength. For medium linear velocities ($v \leq 10\ m/s$), it is recommended not to exceed the deformation (*2mm*) of the flexible bushing. To do this, it is necessary to determine the virginity of the flexible bushing. The scheme for determining the stiffness of the tension roller flexible bushing is shown in pic. 2.



Pic. 2. Schematic diagram of a chain extension with a compressive tensioning device

The method of the calculation of the parameters of the compound roller of chain drives.

From this pic. 2: ΔMKO_3:

$$MK = r_3 \sin\varphi_3, \tag{1}$$

from which it is possible to determine the length of the cover with the tension roller of the chain:

$$MN = 2r_3 \sin\varphi_3 \, , \tag{2}$$

where, r_3 is the radius of the tension roller; φ_3 is ½ of the coverage angle.

Relative weight of the chain in the zone of impact with the tension roller:

$$q_{MN} = 2q \cdot r_3 \sin\varphi_3 \, , \tag{3}$$

where, q is the weight per unit length of the chain is N/m.

In this case, to estimate the fact that the tension force in the leading link of the chain is formed mainly by the friction force between the tension roller and the chain, and estimating the results of research in [8,9] we obtain the following expression:

$$Q_T = 2 f_u q r_3 \cos\varphi_3 \sqrt{2\left[1 - \cos\left(\varphi_3 + \frac{\Delta_3}{l_3}\right)\right]} \, , \tag{4}$$

where, f_u is the coefficient of friction with the tension roller of the chain, Δ_3 is the sagging chain, l_3 is the length of the chain-led (free) chain.

Then natural frequency:

$$\rho \; = k \cdot z_3 n_3 / 60 \, , \tag{5}$$

where, z_3 is the number of elements affected by the tension roller of the chain, n_3 is the frequency of rotation of the tension roller, k is the coefficient of proportionality.

Natural frequency:

$$\rho_x = \sqrt{\frac{C_3 g}{ql}} \, , \tag{6}$$

where, C_3 is the coefficient of elasticity given by the elongation of the chain, l is the length of the part which is affected by the tension roller of the chain.

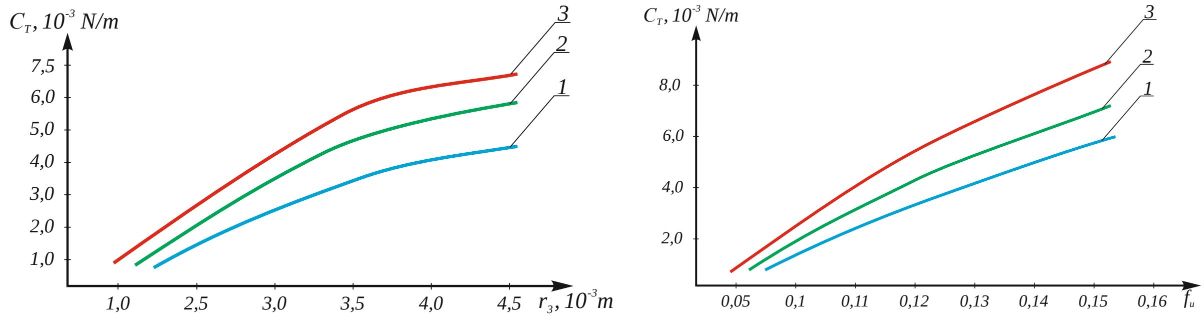
To estimate the above, we create a formula for calculating the stiffness of the tension roller bushing:

$$C_T = \frac{2}{\delta_m} f_u q r_3 \cos\varphi_3 \sqrt{2\left[1 - \cos 2(\varphi_3 + \frac{4\Delta_3}{l_э})\right]} \, , \tag{7}$$

where, $\delta_m$is the deformation value of the elastic (rubber) bushing of the tension roller.

In the case of the chain drive operation, the stiffness of the tension element of the tension roller depends on mainly the rubber structure, the impact force of the chain, the coefficient of friction, the radius of the roller and the angle of coverage [11-16]. As a result of the research, graphs of the dependence of the radius of tension tension of the elastic bushing and the coefficient of friction with the chain were obtained (see pic. 3). The analysis of the obtained graphs showed that as the radius of the tension roller increases, the coefficient of elasticity of the flexible bushing increases in a nonlinear pattern (pic. 3*a*). In particular, when the radius of the tension roller increases the stiffness of the bushing without line. This can be explained by the fact that as the radius of the tension roller increases, the effect of the chain on the flexible bushing also increases, its deformation increases. The coverage angle also increases accordingly. As a result of the research, the increase in friction between the chain and the tension roller and the elastic bushing leads to an increase in the linear regularity of the tension. Pic. 3b shows these graphical links. The coefficient of friction 0,10 and position of the stiffness flexible bushing is equal to the $1{,}85 \cdot 10^3 N/m$. In the case of the friction, the stiffness flexible bushing is equal $7{,}95 \cdot 10^3 N/m$.

As mentioned above, it is necessary to increase the coverage angle, reduce the coefficient of friction and prepare a tension roller with a small radius, assuming that the chain does not exceed vibrations [8-11]. Therefore, it is important that the recommended parameters for the chain drive under consideration are in the following range: $f_u = 0{,}1 \div 0{,}13;\ \varphi_3 = 0{,}3 \div 0{,}45\, rad,\ r_3 = (2{,}5 \div 4{,}0) \cdot 10^{-2} m.$



a *б*

$1 - \varphi_3 = 0{,}425;\ 2 - \varphi_3 = 0{,}34;\ 3 - \varphi_3 = 0{,}17$

a is the dependence of the tension of the tension roller on the stiffness elastic bushing, the radius of the roller, and ***b*** is the coefficient of friction between the chain and the roller.

Pic. 3. Graphs of the dependence of the tension of the tension bushing of the chain drive on the tensioning device, the radius of the roller and the coefficient of the friction between the chain and the roller.

The calculation scheme and mathematical model of the chain oscillations. The chain drive with a compressive tensioning device can be widely used in accordance with the loads of technological machines. High performance was achieved as a result of the application of the recommended chain drive on the Redler conveyor [15,16]. The service life of a chain drive depends on whether the sprockets, chain and tension roller are made. In many cases, the performance of these elements, the long-term performance depends on the vibrations of the chain cooling network. Pic.4 shows the proposed circuit (*a*) and the calculation scheme (*b*) of the oscillations of this part of the chain with the chain tension roller of the proposed chain drive. To develop a mathematical model of the chain oscillation, we use Langraj's second-order equation [8,9]:

$$\frac{d}{dt} \cdot \left(\frac{\partial T}{\partial \dot{x}} \right) - \frac{dT}{dx} = -\frac{d\Pi}{dx} - \frac{dR}{d\dot{x}} + Q_y , \qquad (8)$$

where T, Π is the kinetic and potential energy of the system, x is the generalized coordinate, R is the dissipative function of the Reley, Q_y is the generalized external force.

$$T_3 = \frac{m_3}{2} \left(\frac{dx_3}{dt} \right)^2 . \qquad (9)$$

Potential energy:

$$\Pi_3 = \frac{C_T x_3^2}{2}. \quad (10)$$

Dissipative function

$$R_3 = \frac{b_T}{2}\left(\frac{dx}{dt}\right)^2 \quad (11)$$

where, x_3 is the displacement of the chain part along the transverse oscillations, C_T is the flexible bushing stiffness, b_T is flexible bushing dissipation coefficient, m_3 is the mass of the part of the chain affected by the tension roller.

Obtained (9), (10), (11) We determine the additions of the Lagrange equation on:

$$\frac{d}{dt}\frac{dT_3}{d\dot{x}_3} = m_3\ddot{x}_3;\ \frac{d\Pi_3}{dx_3} = C_T x_3; \frac{dR_3}{d\dot{x}} = в_T \dot{x}_3 \quad (12)$$

Substituting (12) into (8), we obtain a differential equation representing the transverse oscillations of the part of the chain drive affected by the tension roller.

$$m_3 \frac{d^2 x_3}{dt^2} + b_T \frac{dx_3}{dt} + c_T x_3 = Q_y \quad (13)$$

Considering the expression for determining the coefficient of elasticity of the tension roller bushing (7), the mathematical model representing the oscillation of the chain is as follows:

$$m_3 \frac{d^2 x_3}{dt^2} + b_T \frac{dx_3}{dt} + \frac{2x_3}{\delta_m} f_u \cdot q r_3 \cos\varphi_3 \sqrt{2\left[1-\cos 2\left(\varphi_3 + \frac{4\Delta_3}{l_э}\right)\right]} = Q_y \quad (14)$$

On the basis of pic. 4, the regularities of the change in the influencing force:

$$Q_y = Q_j + Q_1|sin\omega t| \quad (15)$$

We extend the obtained expression (15) to the Fure series according to [9,10]:

$$Q_y(t) = Q_1 + Q_0\left[\frac{2}{\pi} - \frac{4}{\pi}\left(\frac{\cos 2\omega_3 t}{1\cdot 3} + \frac{\cos 4\omega 3t}{3\cdot 5} + \frac{\cos 6\omega_3 t}{5\cdot 7} + \ldots + \frac{\cos 2n\omega_3 t}{(2n-1)\cdot(2n+1)}\right)\right] \quad (16)$$

where, n is equal to 1,2,3…

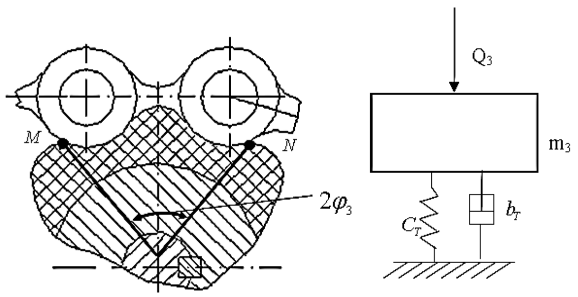
It is known [14-16] that the dissipation coefficient in oscillating moving systems mainly allows to reduce the amplitude of vibration, accelerating processes. However, it does not affect the vibration frequency. It is therefore advisable to see the state of the maximum amplitude in the oscillations of the tensioning roller of the chain saw chain under the influence of the elastic bushing. Therefore, the solution of the problem was carried out without taking into account the dissipation coefficient in (15). As a result, taking into account (16), the equation of transverse oscillation of the chain is as follows:

$$m_3\frac{d^2x_3}{dt^2}+\frac{2x_3}{\delta_m}f_u q r_3\cos\varphi_3\sqrt{2\left[1-\cos2\left(\varphi_3+\frac{4\Delta_3}{l_э}\right)\right]}=Q_1+$$
$$+Q_0\left[\frac{2}{\pi}-\frac{4}{\pi}\left(\frac{\cos2\omega_3t}{1\cdot3}+\frac{\cos4\omega3t}{3\cdot5}+\frac{\cos6\omega_3t}{5\cdot7}+...+\frac{\cos2n\omega_3t}{(2n-1)\cdot(2n+1)}\right)\right] \quad (17)$$

The solution of the given differential equation (17) was carried out using the solution method given in [10]. The solution of Equation (17) consists of the sum of the solutions for each additive given in (16), i.e.
$$x_з=\frac{1}{C_T}(Q_1+\frac{2Q_0}{\pi})-\frac{4Q_0}{\pi m_з}\sum_{n=1}^{\infty}\frac{\cos2n\omega t}{(2n-1)(2n+1)(p_0^2-4n^2\omega_3^2)} \quad (18)$$

From the solution of the problem it can be seen that the transverse oscillations are in a harmonic law when the tension roller of the chain drive chain is affected by the elastic bushing, and this law shifts from the state of static equilibrium to the following value:

$$X_{зCT}=\frac{Q_1+\frac{2Q_0}{2\pi}}{\frac{2}{\delta m}f_u q r_3\cos\varphi_3\sqrt{2\left[1-\cos2(\varphi_3+\frac{4\Delta}{l_э})\right]}} \quad (19)$$

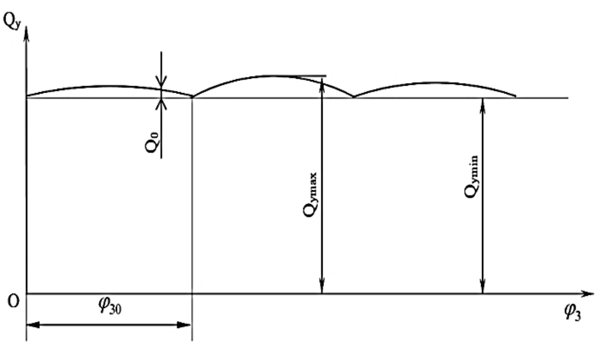


a is the diagram of the part of the chain affected by the tension roller,

b is the diagram of the calculation of transverse vibrations affected by the tension roller.

Pic. 4. The scheme of calculation of impact and transverse oscillations with a chain tension roller

a b



where, Q_1 is the constant component of the force of action, Q_0 *is* the amplitude of the vibration of the force, φ_{30} *is* the angle of coverage of the tension roller on the complete change of force

Pic. 5. Graph of the force change acting on the part of the chain affected by the tension roller

To determine the transverse displacement of the chain, an approximate result can be obtained by considering two additives in solution (18). In this case (18) can be written as follows:

$$X_з = \frac{\delta_m (Q_1 + \frac{2Q_o}{\pi})}{2 f_u q r_3 \cos\varphi_3 \sqrt{2\left[1-\cos 2(\varphi_3 + \frac{4\Delta}{l_э})\right]}} - \frac{4Q_o}{\pi m_3} \cdot$$

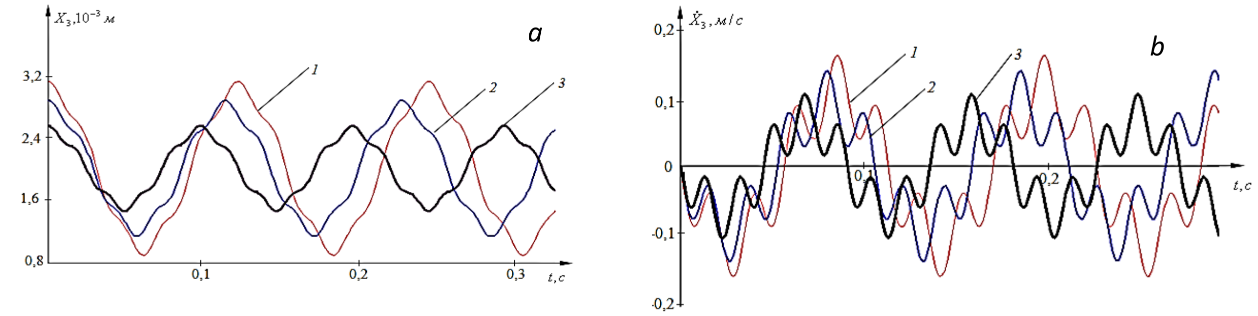
$$\cdot \frac{\cos 2\omega_3 t}{3\left(\frac{2 f_u q r_3 \cos\varphi_3}{\delta_m m_3} - \sqrt{2\left[1-\cos 2(\varphi_3 + \frac{4\Delta}{l_э})\right]} - 4\omega_3^2\right)} +$$

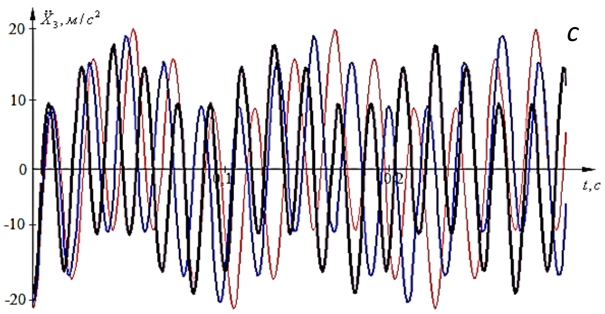
$$+ \frac{\cos 4\omega 3t}{15\left(\frac{2 f_u q r_3 \cos\varphi_3}{\delta_m m_3} \sqrt{2\left[1-\cos 2(\varphi_3 + \frac{4\Delta}{l_э})\right]} - 16\omega_3^2\right)} \quad (20)$$

The following calculated values of the parameters were obtained to determine the regularity of the transverse oscillation of the transmission chain in numerical values:

$$\delta_m = (2{,}0 \div 3{,}0)\cdot 10^{-3} m;\ \ Q_1 = 2{,}0 \div 2{,}5\, N;\ \ Q_o = 0{,}15 \div 0{,}25\, N;\ \ f_u = 0{,}1 \div 0{,}13;$$
$$q = (20 \div 25) N/m;\ r_3 = (2{,}4 \div 4{,}0)\cdot 10^{-2} m;\ m_3 = (0.015 \div 0.045) kg,$$
$$\omega_3 = 30 \div 35 s^{-1};\ \varphi_3 = 0{,}3 \div 0{,}45 rad,\ \ \Delta = (2.0 \div 10{,}0)\cdot 10^{-3} m.$$

The results of the analysis of the dependence of chain vibrations on the transmission parameters. Considering the analysis of the oscillations of the proposed chain drive parameters when they are affected by the chain tension roller. Based on the numerical solution of the obtained expressed (20), the law of transverse oscillation of the chain was determined. Pic. 6. shows the change in transverse shear, velocity, and acceleration under the influence of a chain tension roller as a function of the tension roller at different angular velocities over time. The oscillation frequency of the chain mainly varies depending on the angular velocity of the tension roller. In this case, the vibration amplitude is $0.74mm$ when the angular velocity is $25s^{-1}$, while the amplitude decreases to $0.62mm$ when the angular velocity is $35s^{-1}$. That is, the increase in the speed of impact reduces the time of deformation of the elastic roller (shell) of the tension roller and increases the angle of impact. Therefore, the vibration amplitude decreases (see pic. 6. *a, b, c*). Also, when the rotational speed is $25s^{-1}$, the lower component of the oscillation frequency of the chain is $12.3Hz$, while the upper frequency is $45.65Hz$. Similarly, if the angular velocity of the tension roller is $35s^{-1}$, the frequency of the upper component of the chain oscillation is $42.4Hz$, while the lower component is $8.2Hz$. It should be noted that the displacement of the chain from the static position with the tension roller (19) is an average of $2.4\ mm$, respectively (see pic. 6.). It is known that in oscillating motion, the change in the oscillating mass has a direct effect on its amplitude [9].





Where, $1-\omega_3=25s^{-1}$

$2-\omega_3=30s^{-1}1-\omega_3=35s^{-1}$

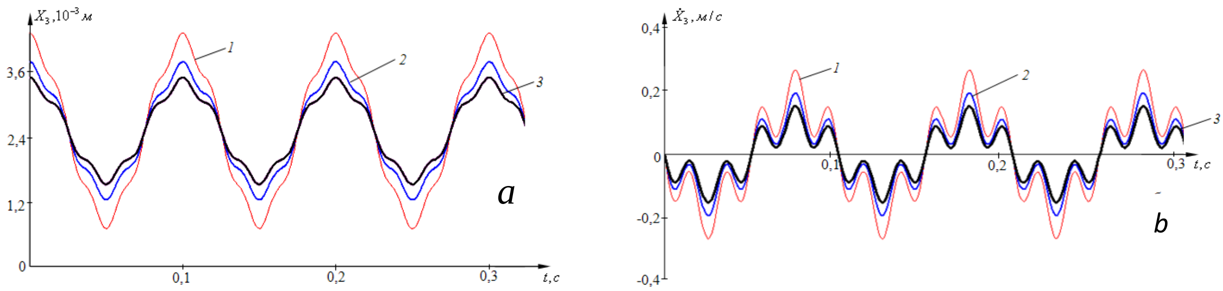
$\boldsymbol{a}$ is the law of oscillation of a-chain,

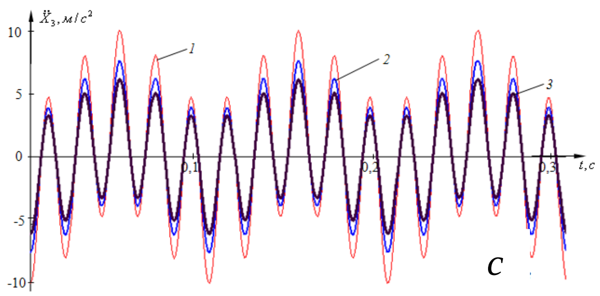
$\boldsymbol{b}$ is the velocity of oscillation of the chain, $\boldsymbol{c}$ is the acceleration of oscillation of the chain with respect to time relative to its stellar angular velocity.

Pic. 6. Dependency graphs

Pic. 7 shows the laws of transverse oscillation (*a*), velocity (*b*), and acceleration (*c*) of a chain depending on the mass of the chain. It can be seen from the obtained oscillation laws (see pic. 7) that is an increase in the mass of the part of the chain affected by the tension roller also leads to a decrease in the amplitude of the vibration amplitude, velocity and acceleration. But in this case the oscillation frequencies of displacement, velocity and acceleration are almost unchanged. In particular, when the mass is $0{,}015\ kg$, the oscillation amplitude is $1{,}36mm$, respectively, the velocity oscillation amplitude is $0{,}242m/s$, the acceleration oscillation amplitude is $8{,}84m/s$. When the mass of the transverse oscillating part of the chain increases by 0.045kg, its vibration amplitude decreases to $0{,}87mm$, the velocity amplitude decreases to $0{,}168m/s$, and the acceleration amplitude decreases to $3{,}15m/s$. It should be noted here that the amplitude of the high-frequency component of the chain oscillation also decreases accordingly.

As a result of the research, the laws of change of velocity and acceleration of the leading network oscillations were determined at different values of the roller radius (see pic. 8). Based on the analysis of the obtained vibration laws, it can be said that the oscillation frequency, velocity and frequency of change of accelerations do not change with increasing radius of the tension roller. In this case, only the amplitudes of vibration displacement, velocity and acceleration increase. However, it is noteworthy that the effect of increasing the radius of the tension roller on the vibration amplitude decreases with decreasing the value of the length of the chain guide network.

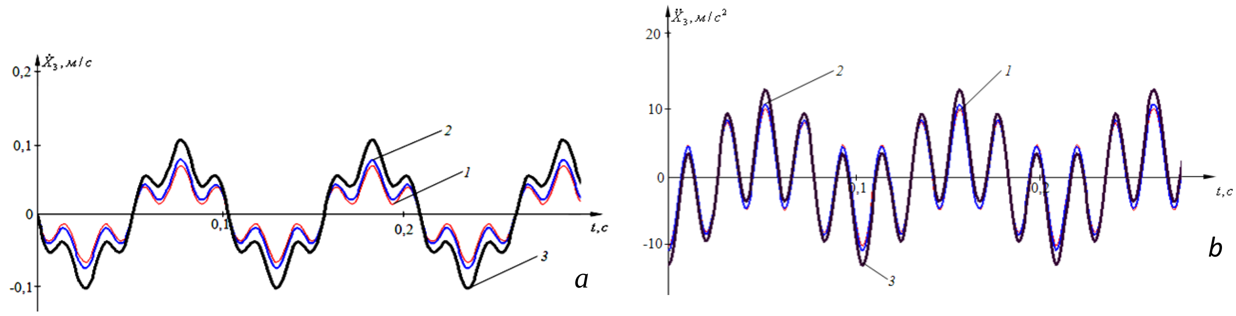




a *is* the change in transverse oscillation of chain, b *is* the transverse oscillation velocity of chain, c *is* the transverse oscillation acceleration of chain in time and the values of the mass of the chain in the affected area; where

$$1-m_3=0{,}015kg;$$
$$2-m_3=0{,}030kg;$$
$$1-m_3=0{,}045kg.$$

pic. 7.



Where: $1-r_3=25\cdot10^{-3};\ 2-r_3=30\cdot10^{-3};\ 3-r_3=40\cdot10^{-3};$

a is the effect of the radius of the tension roller on the oscillation rate of the chain, and b-acceleration on the laws of change.

Pic. 8. Dependency graphs

The discussion of the research. The radius of the tension roller in the chain drive in question $(3{,}5\div4{,}0)\cdot10^{-2}m$. In this case, the amplitude of vibration of the chain $0{,}8\div1{,}5$ *mm* does not exceed. Hence, the recommended values of the exposure angle should be obtained in the range: $0{,}8\div1{,}5mm$. According to the analysis, the vibration $\varphi_3=0{,}4\div0{,}5rad$ speed of the chain is from $3{,}0\cdot10^{-3}m$ to $9{,}0\cdot10^{-3}m$ chain coolness. $20\div25$ %the amplitude of the acceleration decreases to $10\div15$ %.

Conclusion: The formula for calculating the stiffness of a chain $0{,}8\div1{,}5mm$ extension tension roller is obtained. It is recommended that the coefficient of friction, the value of the angle of impact and the parameters of the radius of the roller between the chain and the tension roller are as follows to ensure that the vibration amplitude of the chain suspension is in the range:

$$f=0{,}1\div0{,}13;\ \varphi_3=0{,}3\div0{,}45rad;\ r_3=(2{,}5\div4{,}0)\cdot10^{-2}m$$

REFERENCES

Berhanu, Admassu ; & Yoseph, Shiferaw. (2011).Donkeys, horses and mules-their contribution to people's livelihoods in Ethiopia, The Brooke, Addis Ababa, Ethiopia

Blench, R. (2001). You can't go home again: Pastoralism in the new millennium, Overseas Development Institute, London, UK.

Bourguignon, F. (2004). The Poverty-Growth-Inequality Triangle. Washington DC: World Bank

Balakrishnan, R., Steinberg, C., & Syed, M. (2013). The Elusive Quest for Inclusive Growth: Growth, Poverty, and Inequality in Asia? Washington DC: IMF

CSA, (2007). National Population Statistics. Federal Democratic Republic of Ethiopia, Central Statistical Authority, Addis Ababa. Ethiopia

Coppock, D.L., (1994). The Borana Plateau of Southern Ethiopia: Synthesis of Pastoral Cossins, N. J, & Upton,M. (1988a). Options for improvement of theBorana pastoral system. Agric. Systems *(27)* .251-278.

Dreyfus, F, (1976). Contribution a l'étude de la zootechnic et de la pathologie des equides domestique en Ethiopie. Thèse pour le doctorat vet. Ecole Nationale Vétérinaire d'Alfort (ENVA), Paris, France , 122

Delgado, C., M, Rossegrant., H, Steinfeld .,& S, Ehui. (1999). Livestock to 2020: The Next Food Revolution. IFPRI , 2020 Brief 61.

Desta, S.(1996). Past and present pastoral development in Ethiopia. *In*S. Edwards and T. Mesfin (Ed)., *Proceedings of the conference on pastoralism in Ethiopia*, Addis Ababa. p. 26-28

Desta, S., & Coppock, D.L. (2002) Cattle Population Dynamics in the Southern Ethiopian Rangelands, 1980-97 Journal of Range Management (55): 439-451.

Desta, S., & Coppock, D.L. (2004) Pastoralism under pressure: Tracking System change in southern Ethiopia. Human Ecology, 32(4), 465-486.

Devereux, S. (2006). Vulnerabl Livelihoods in Somali Region, Ethiopia. Research Report Desai , Vandana., & Robert, B. Potter; Robert, B. (2014).*The Companion to development Studies(Ed),* Routledge, London and New York,3rd Edition, pp.75-77, pp.87-89

Desai, Vandana.,& Potter,Robert,B.(2014) .*The Companion to Development Studies(Ed), 3rd ed.* Routledge, London, New York p 31, 63, 499, pp. 500-501 ,pp.513-515

Daron, Acemoglu., Aghion, Philippe; et.al;(2012). "The Environment and Directed Technical Change." American Economic Review 102 (1).131–66.

Escobar, Arturo.,(1995). *Encountering Development*: *The Making and unmaking of the Third World,* Princeton University Press, Princeton, New Jersey, USA ,P.23

Escobar, A.,. (1995). *Encountering Development: The Making and Unmaking of the Third World,* Princeton, NJ: Princeton University Press.

Fankhauser, Samuel., & Stern, Nicholas(2016) .Climate Change, Development, Poverty and Economics, Granthan Research Institute on Climate Change, the Environment

Goulet, D., (1971). The Cruel Choice: A New Concept on the Theory of Development, New York: Atheneum.

Gupta, Joyeeta.,& Vegelin, Courtney(2016).Sustainable Development Goals and Inclusive Development, International Environment Agreements .433-448

Hodge, J. M., (2007).Triumph of the Expert: Agrarian Doctrines of Development and the Legacies of British Colonialism. Athens: Ohio University Press.

IPCC. (2014). *Climate Change 2014: Impacts, Adaptation and Vulnerability*. Contribution of Working Group II to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change (IPCC), Geneva, Switzerland. https://www.ipcc.ch/report/ar5/wg2/

Justin, Ginnetti., & Travis, Franck.(2014). Assessing Drought Displacement Risk for Kenyan, Ethiopian and Somali Pastoralists, Technical Paper, Internal Displacement Monitoring Centre, Norwegian Refugee Council, Geneva, Switzerland, 61

Kothari, U. (2006a). 'From Colonialism to Development: Reflections of Former Colonial Officers' Commonwealth and Comparative Politics 44(1), pp.118–136.

Marx, K.(1976)(Ed), *Kapital(1)* , Harmondsworth, UK, Penguin.

Newsham, Andrew., Daley, Ben; et.al;(2018). International Programs for Climate Change and Development: Centre for Development, Environment and Policy, University of London pp5-7

Raniere, R., & Ramos, R. (2013). *Inclusive Growth: Building up a Concept*. Brasilia: International Policy Centre for Inclusive Growth

Shalal, Andrea. (2021) , Environment, Reuters, Washington

Sen, A. (1999). *Development as Freedom: Human Capability and Global Need*, New York: Knopf.

Sen, A. (1984). *Poverty and Famines: An Essay in Entitlement and Deprivation*, Oxford: Clarendon Press.

World Bank. (2009). *What is Inclusive Growth?* Washington DC: World Bank in http://siteresources.worldbank.org/INTDEBTDEPT/Resources/468980-1218567884549/WhatIsInclusiveGrowth20081230.pdf