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The Self-Excite Vibrations of Multi-Scoop Digger Machine

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In his work energetic method Has been tested. The conditions connected with creation of the self-excite vibrations in mutli-tools digger machine. The safety conditions to avoid dangerous work conditions have been presented. This article is a continuation and extension of the work [3].

Keywords: kinematics, dynamics, vibrations, digger machine

1. Introduction

The device able to produce undamping vibrations and characterize that have: source of energy, regulation valve (regulate the energy in the system) and feedback loop control. Is called the self-excite vibration system. For the self-excite vibration systems the most significant case is the mode of energy consumption. This mode enable to recognize autonomous self-excited system from nonautonomous systems, which energy flow is connected with outer forces (depending on time) acting.

In the equation which describe self-excite system, the time is not presented in apparent way, the source of energy is constant (do not depends on time) and the energy flow is regulated by it-self by vibrating system. When we look at self-excite systems with 1 DOF we can enumerate the following elements Fig. 1. I – permanent source of energy, II vibrating system, III regulators, IV feedback between vibrating system and regulator with the help of which the vibrating system controls the energy supply. The elements I and II are linear, elements III should be nonlinear.

As a result of existing feedback there is mutual influence between the regulator and the vibrating system, which allows self-excite system to control its own energetic levels, which may result in (despite the losses in the system) undamping periodical vibrations.

Technology know many self-excite systems which have been described in scientific works [1, 2, 3]. This system include also building machines (Fig. 2)

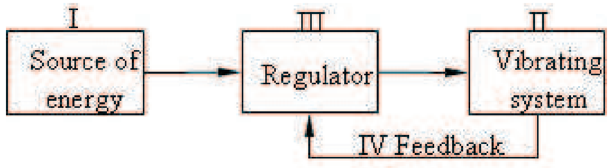

Figure 1

The multi-scoop digger machine – a type of machine characterized by the constant digging process, equipped witch many digging tools, called scoops, placed in equal distances on a chain or a wheel. Machines used in mining are used in multidimensional excavation and in the process of storing materials. There are one of the biggest machines designed for earth excavations, their dimensions exceed 200 m length and 100 m height, the mass is greater than 13.000 tons and efficiency is greater than 200.000 m^3 excavated materials per hour. The smaller machines of this type are used to dig holes and are called trench excavator.

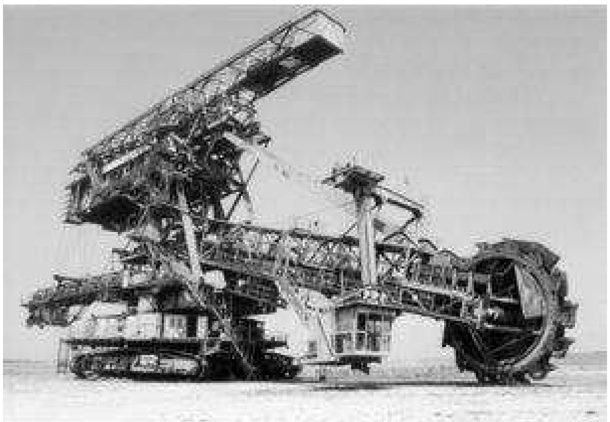

Figure 2 The multi-scoop digger machine

2. The power engineering of self-excite vibrations

One of the most general points of view is the power engineering point of view. We take in to consideration the energy balance of desired phenomena. The idea of self-excite vibration system can be described in the following way: Lets assume that vibration exist. The energy of those vibrations should decrease as a result of: loss or energy transfer as vibration to the user. The loss of energy is supplemented by source of energy from which some part of energy is directed by valve to the vibrating system in each period.

Certainly the constant vibrations are possible just in one case where energy inflow from source of energy in period equals the loss of energy with in this time.

This condition of the energy balance is a condition of growing vibrations. It is obvious that if the energy balance is not in equilibrium (it means the loss of energy in the system is not reduced) the vibration will be disappearing. In the opposite situation the vibrations will be growing.

In this work the energetic method of measuring the self-excided vibrations has been presented. The vibrating system of bodies is a system where dissipation is positive, if we have disappearing vibrations. Those Conditions for 1 DOF are described by inequalities (1) where firs means that coefficient of elasticity is greater than 0 and second means that system have positive dissipation [4].

$$\frac{\partial f\left(x\right)}{\partial x} > 0, \qquad R\left(\dot{x}\right) \geq 0 \tag{1}$$

Where:

$f(x)$ – elasticity characteristic,

$R(x)$ – damping characteristic.

3. The multi-scoop wheel vibrations examination

The examination was leaded on the example of self-excided vibrations connected with multi-scoop wheel of digging machine. In this case scoop wheel with extension arm in horizontal surface has been taken in to consideration. In this surface vibration frequency of the extension arm is much lower than in vertical surface.

In some conditions the rotational motion of the extension arm can be the source of not beneficial vibrations in the machine. This type of vibrations is classified as self-excided. Those vibrations are created in non preserved system where the loss of energy is compensated by the source of energy which is included in the vibrating system.

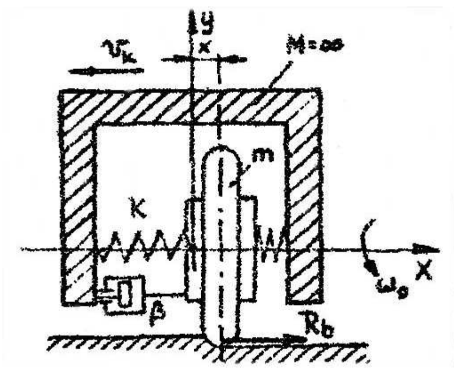

Figure 3 The vibrating system with 1 DOF

The diagram scheme for his self-excite system has been presented in Fig. 1. The source of energy is a rotational motion of the extension arm with constant speed. The energy regulator is moment from frictional force and the vibrating system is scoop wheel. In this complex system the frictional force is a feedback.

The purpose of his analyze is to give the reasons of self-excided vibrations creation on scoop wheel and give also conditions where we can eliminate those vibrations. Because of that the vibrating system has been substituted by the model based on I DOF (Fig. 3)

The following signs have been assumed on the Fig. 3:

m – reduced mass of digging tool,
M – the mass of rotary construction,
k – substitional elasticity coefficient for the extension arm,
β – the substitional coefficient of construction and inside friction.

In his work there is assumption that the velocity of rotary construction v_k and circular velocity v_o of wheel is constant. In this case the resistance forces which are generated by ground we can describe as follow:

value of circular force $R_r = \sum R_{ri}$
value of normal force $R_n = \sum R_{ni}$
value of side force $R_b = \sum R_{bi}$

Under influence of force R_b the tool is moving according to rotary construction.

The dynamic motion equations for the taken model hale been presented below:

$$m\ddot{x} + kx + \beta\dot{x} = R_b \tag{2}$$

Where:

x – The Wheel displacement with tool.

The value of the side force acting on one scoop can be expressed in the form:

$$R_{bi} = R_{ri}\frac{v_k}{v_o} - R_{ni}\sin\varphi \tag{3}$$

Where:

R_{ri} – the value of tangent force on scoop,
R_{ni} – the value of normal force on scoop,
φ – the angle of cutting tool setting on scoop.

The values of R_{ri} i R_{ni} are equal:

$$\begin{aligned} R_{ri} &= k_r ab\sin\alpha \\ R_{ni} &= R_i ctg\left(\delta_o+\mu\right) - R_{ni}\left[ctg\left(\delta_o+\mu\right)+ctg\left(\delta_1+\mu\right)\right]\sin\varphi \end{aligned} \tag{4}$$

Where:

k_r – coefficient of the digging resistance in N/cm^2,
ab – thickness and width of tool (Fig. 4),
δ_0 – the angle of slice,
δ_1 – the angle between cutting trajectory and frictional surface,
μ – the angle of friction between the ground and Edg of cutting,
R_{ni} – The additional resistance from the cutting tool.

The ab product on the base of work [2] equal:

$$ab = \frac{q}{rk_p\left(1-\cos\alpha_n\right)} \tag{5}$$

Where:

q – the capacity of one scoop,
r – the scoop wheel radius,
k_p – the ground coefficient,
α_n – the full angle of cutting wheel.

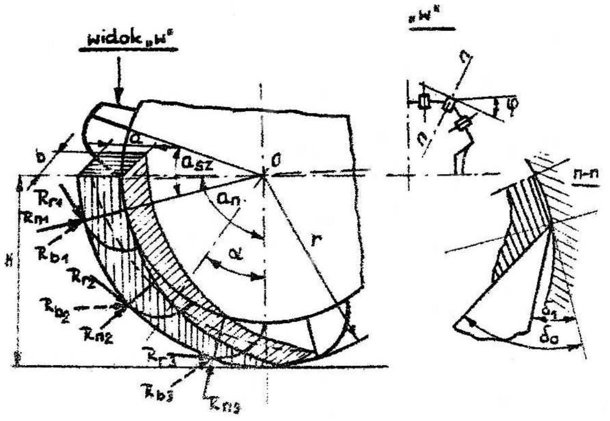

Figure 4 The cutting tool

The v_k and b because of vibrations are not constant and it is possible to express it in the form:

$$v_k^* = v_k - \dot{x}, \qquad b^* = b\left(1 - \frac{\dot{x}}{v_o}\right) \tag{6}$$

The value of the force R_b acting on the wheel, on the bases of equations (3) - (6) and necessary mathematical operations can be expressed in the following form:

$$R_b = -\frac{A}{v_k}\left[\frac{2v_k}{v_o} - ctg\left(\delta_o + \mu\right)\sin\varphi\right]\dot{x} + \frac{A}{v_k v_o}\dot{x}^2 + B \tag{7}$$

The quantity A and B in the equation (7) have the form presented below:

$$\begin{aligned} A &= \frac{zk_F\alpha_n q}{rk_p 2\pi} \\ B &= A\left[\frac{v_k}{v_o} - ctg\left(\delta_o + \mu\right)\sin\varphi\right] + \sum R_{ni}\left[ctg\left(\delta_o + \mu\right) + ctg\left(\delta_1 + \mu\right)\sin\varphi\right] \end{aligned} \tag{8}$$

Where:

n – the quantity of the scoops working simultaneously.

The equation (2) after the substitution (7) and transformation application $x_1 = x + \dfrac{b}{k}$ we can describe in the following form:

$$m\ddot{x}_1 + kx_1 + \left\{\alpha_o + \frac{A}{v_k}\left[\frac{2v_k}{v_o} - ctg\left(\delta_o + \mu\right)\sin\varphi\right]\right\}\dot{x}_1 - \frac{A}{v_k x_o}\dot{x}_1^2 = 0 \tag{9}$$

Equation (9) will be describing damping vibrations if the system will be dissipatic. It means where the formula (1) will be fulfilled.

$$f\left(x\right) = kx_1, \quad R = \left\{\beta + \frac{A}{v_k}\left[\frac{2v_k}{v_o} - ctg\left(\delta_o + \mu\right)\sin\varphi\right]\right\}\dot{x} - \frac{A}{v_k x_o}\dot{x}^2 \tag{10}$$

After substitution (10) to (1) we received:

$$k > 0, \quad \left\{ \left[\alpha_o + \frac{A}{v_k} \left(\frac{2v_k}{v_o} - ctg \right) (\delta_o + \mu) \sin\varphi \right] \right\} \dot{x}_1 - \frac{A}{v_k x_o} \dot{x}_1^2 \geq 0 \tag{11}$$

The first inequality In (11) is always fulfilled, the second inequality will be fulfilled for the small $\dot{x}_1$ and for:

$$\alpha_o + \frac{A}{v_k} \left[\frac{2v}{v_o} - ctg\,(\delta_o + \mu) \sin\varphi \right] > 0 \tag{12}$$

If we assume condition connected with energy In the system it means where $\alpha_o = 0$, the vibration will be damping when:

$$\frac{v_k}{v_o} > \frac{1}{2} ctg\,(\delta_o + \mu) \sin\varphi \tag{13}$$

In Fig. 5 has been presented quantity v_k/v_o in dependence with angle of cutting tool setting on the scoop by the assumed angle $\delta_o + \mu$.. The angle φ In the process of cutting is in range $30^0 \leq \varphi \leq 90^0$.

From the Fig. 5 we see that vibration is damping when, for assumed $\alpha_o = 0$, the value v_k/v_o for assumed angle $\delta_o + \mu$, should be on the appropriate curve or above it. From the analysis it results that the vibrations are damping if the inequality (12) is fulfilled also if the inequality (13) is fulfilled vibrations are damping. The self-excite vibrations are possible if the condition (12) is not fulfilled and also if the condition (13) is not fulfilled. The system in this case have negative dissipation for the arbitrary $\dot{y}$.

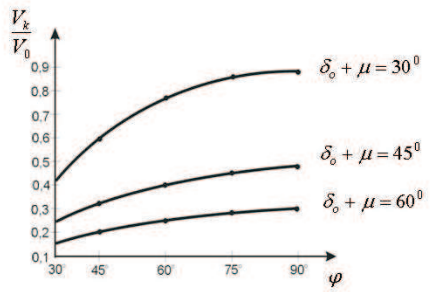

Figure 5.

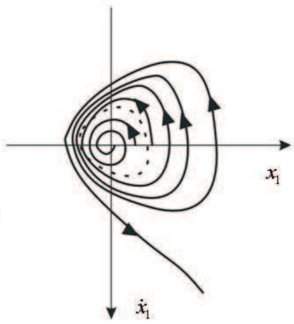

Figure 6.

If the condition (12) or (13) for the the arbitrary $\dot{y}$is not meet the dissipation of the system (8) is also negative, it means that it can exist self-excided vibrations for the scoop wheel with extension arm. The equation (9) has been used in simulation for the input data presented belowe:

$m = 25\text{T}, \quad k = 100\text{kN/m}, \quad \beta = 0.0096, \quad \varphi = 90^o, \quad \delta_o + \mu = 450.$

The results of those calculations have been presented on the scope in Fig. 6 to Fig. 8 for the different values of v_k/v_o.

$m = 25\text{T}, \quad k = 100\text{kN/m}, \quad \beta = 0.0096, \quad \varphi = 90^o, \quad \delta_o + \mu = 450.$

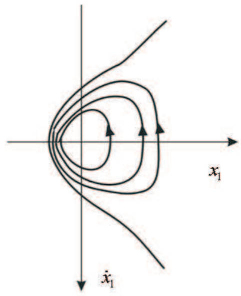

Figure 7.

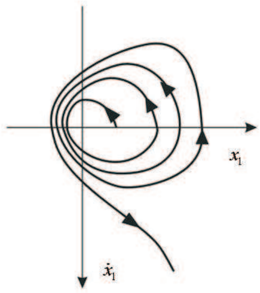

Figure 8.

In the Fig. 6 the courses of phase trajectory Has been presented. In this case the coefficient connected with $\dot{y}$ is positive. The vibrations have dual characteristic in accordance to initial conditions. In the range of point of balance the vibrations are disappearing and point (0, 0) is statically asymptotic. After some range of point of balance vibration can growing. In Fig. 7 the courses of phase trajectory for the condition $y = 0$ has been presented. The trajectories are divided by function described by equation $x_1 = \frac{A}{kv_k v_o}, \dot{x}_1 = \frac{mv_k v_o}{2A}$. Inside of this function trajectories are closed and vibrations are periodic. The amplitude of those vibrations depends on initial conditions. Outside of this function trajectories are open and closed to this function, there are oscillations and the velocity and displacement are going in to infinity. The theoretical and practical results are the some. In the Fig. 8 the courses of phase trajectory for coefficient connected with $\dot{y}$ is negative. The vibrations are growing and point of balance is not stable.

4. Conclusions

The presented theoretical analysis of undamping vibrations of the digging machine results in the conclusion the when you design and exploit it it is necessary to apply the above mentioned dependencies in order to avoid dangerous undamped vibrations. An improper assumption of the construction parameters may cause failure and what fallows to significante financiale costs.

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