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INVESTIGATION METHODS OF THE INFLUENCE OF THE TIMES IN TRANSITIONAL PROCESSES ON THE DYNAMIC LOAD FOR LIFTING DEVICE

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ABSTRACT:
Theoretical dependences for studying the influence of the times in transitional processes on the dynamic load of the elements of lifting device and supporting metal structure are worked out. Dependencies for finding of the influence of the start / stop time ratio and the characteristic oscillations of the load after the completion of the transitional processes on the minimum and maximum amplitude values for the acceleration and the dynamic force are received.
KEYWORDS:
Lifting device, oscillations, dynamic load, dynamic processes

1. INTRODUCTION

The work of the lifting devices is characterized by frequent starts and stops in both moving directions which is related to the arising of accelerations, dynamic forces and oscillations with values and durations that have an influence on the duration of the transitional processes, the precision of positioning and comfort work. The presence of various elastic links between components of the device results in vibration processes during the start and stops that have fading behaviour and also continue after the acceleration or deceleration of the load.

The limitations of the amplitudes and durations of the oscillations are desirable not only because of the process requirements of the handling process but also to reduce the dynamic loads of the device components and supporting metal structure.

The lifting devices are built using components with various elasticity degrees, for example rope or chain, clutch, shafts, etc. The most important component is the flexible element (rope or chain) that together with the load forms a system with comparatively low frequency of characteristic oscillations and its half-cycles is commensurable with the acceleration time of the motor [1]. The characteristic oscillations' frequencies caused by the elasticity of other elements of the lifting device, are considerable higher.

The subject of this work is to establish methods for investigation of the dynamic loads of the components of a lifting device depending on the transitional processes times and the durations of the characteristic oscillations for the load.

2. EXPOSITION

A general view of a rope and drum lifting device may be illustrated with the kinematical scheme shown on Fig. 1. This device is substituted with an equivalent 2-mass system shown on Fig. 2 that describes the dynamic processes of a tackle system [1], [2] with sufficient precision factor. Using the dynamic model the dynamic loads of the device during the transitional processes may be investigated and the quality of the drive system to be analysed during the selection of the motor output, start and stop controls, the brake and the related elements.

During the acceleration at the motor start the differential equations are obtainable, from the requirement for equilibrium of the torques, respectively the forces, respectively (1) for the motor rotor rotation caused by continuous torque, and (2) of the load movement:

$$\frac{d^4\varphi}{dt^4}+2k\frac{d^3\varphi}{dt^3}+p_o^2\frac{d^2\varphi}{dt^2}=p_o^2\bar{\varepsilon}\,,\qquad \frac{d^4x}{dt^4}+2k\frac{d^3x}{dt^3}+p_o^2\frac{d^2x}{dt^2}=p_o^2\bar{a}\qquad (1)\text{- }(2)$$

where p_o is the frequency of the characteristic cycle of the system, s^{-1}; k - fading factor, s^{-1}; $\bar{\varepsilon}$ and $\bar{a}$ - average acceleration values, respectively angular for the motor rotor and lineal for the load in case of sturdy link between the two masses.

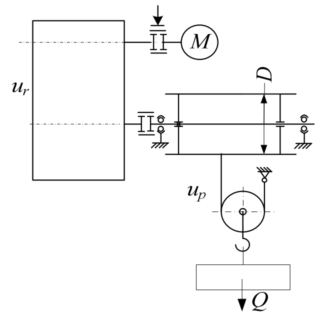

Figure 1. Rope and drum lifting device

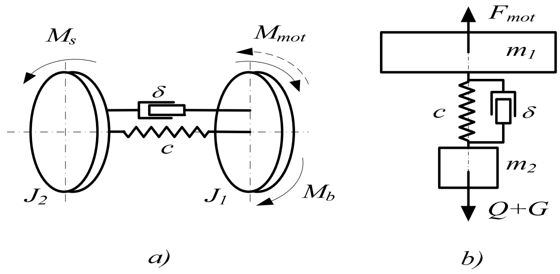

Figure 2. 2-mass dynamic model of rope and drum lifting system
a) adjusted to the motor shaft;
b) adjusted to the straight-forward movement

The acceleration values are calculated by the following expressions:

$$\bar{\varepsilon} = \frac{\pm M_{mot} - M_s}{J_1 + J_2}; \quad \bar{a} = \frac{\pm F_{mot} - (Q+G)}{m_1 + m_2} = \frac{D}{2u_r u_p}\bar{\varepsilon} \qquad (3)-(4)$$

where

M_{mot} is the average starting torque of the motor, Nm;
M_s – resisting torque caused by the load weight adjusted to the motor shaft, Nm;
F_{mot} – the driving force adjusted to the load, N;
Q and G – the weights of the load, resp. load-catching device, N;
D – the drum diameter where the axes of the rope are located, m;
u_r and u_p – the gear ratios of the gear unit, resp. the tackle.

The p_o and k for the equation (1) are calculated by the expressions:

$$p_o = \sqrt{c\left(\frac{J_1 + J_2}{J_1 J_2}\right)}; \qquad k = \frac{d(J_1 + J_2)}{2J_1 J_2} \qquad (5)-(6)$$

and for the equation (2) according to the formulas:

$$p_o = \sqrt{c\left(\frac{m_1 + m_2}{m_1 m_2}\right)}; \qquad k = \frac{d(m_1 + m_2)}{2m_1 m_2} \qquad (7)-(8)$$

where

J_1 is the reduced mass moment of inertia of all rotating weights on the motor rotor, kgm²;
J_2 - the reduced mass moment of inertia of straight-forward moving weights on the motor rotor, kgm²;
m_1 – reduced rotating weights to straight-forward moving ones on the load, kg;
m_2 – the weights of the load and load-catching device, kg;
c – the hardness factor of the system;
d – the damper factor of the system

The "+" and "-" signs in the equations (3) and (4) are valid for the acceleration processes of the load in upward, resp. downward directions.

The solution of the differential equations for the accelerations at starting of the motor from the state of static pendant load with initial conditions $t=0$, $d^2\varphi/dt^2 = 0$ and $d^3\varphi/dt^3 = 0$ for the motor rotor is the following:

$$\varepsilon = \bar{\varepsilon}\left[1 - \frac{p_o}{p}e^{-kt}\cos(pt - \alpha)\right], \qquad (9)$$

and for the load is

$$a = \bar{a}\left[1 - \frac{p_o}{p}e^{-kt}\cos(pt - \alpha)\right], \qquad (10)$$

where

$$p = \sqrt{p_o^2 - k^2} \qquad (11)$$

is the frequency of the fading oscillations of the system $\quad \alpha = \mathrm{arctg}\frac{k}{p}$. (12)

During the acceleration, the force F is a sum of the gravitational force of the load Q, the weight of the load-catching device G and the inertia force m_2a that is the disturbance force for the system:

$$F = Q + G + m_2 a = m_2\left[g + \bar{a} - \bar{a}\frac{p_o}{p}e^{-kt}\cos(pt - \alpha)\right], \qquad (13)$$

where $g = 9.81$ m/s² is the gravity factor.

The inertia force m_2a consists of one constant component corresponding to the $\overline{a}$ and one variable. From (13) follows that in case of acceleration with lifting load, the greatest dynamic load appears with some delay equal to the half-cycle *T/2* of the fading oscillations.

At the completion of the acceleration process, the system performs free fading oscillations. The differential equation of the movement of the load after the acceleration process is received from (2) with $\overline{a} = 0$

$$\frac{d^2x}{dt^2} + 2k\frac{dx}{dt} + p_o^2 = 0 . \tag{14}$$

The initial conditions for solving the equation (14) are determined by the acceleration value and the first derivative of the acceleration in the end of the acceleration process with duration t_m. From (10) for $t = t_m$:

$$a_{п} = \overline{a}\left[1 - \frac{p_o}{p} e^{-kt_m} \cos(pt_m - \alpha)\right], \tag{15}$$

$$\dot{a}_{п} = \overline{a}\frac{p_o^2}{p} e^{-kt_m} \sin(pt_m - \alpha) . \tag{16}$$

The solution of (14) with initial conditions (15) and (16) is

$$a = A\overline{a}\frac{p_o}{p} e^{-kt} \cos(pt - \alpha_o) , \tag{17}$$

where $A = \sqrt{1 - 2e^{-kt_m} \cos pt_m + e^{-2kt_m}}$, $\alpha_o = \mathrm{arctg}\dfrac{\dfrac{k}{p} + \dfrac{p_o}{p} e^{-kt_m} \sin(pt_m - \alpha)}{1 - \dfrac{p_o}{p} e^{-kt_m} \cos(pt_m - \alpha)}$ (18) - (19)

The force in the flexible link rope – motor in the process of free fading oscillations is

$$F = Q + G + m_2 a = m_2\left[g + A\overline{a}\frac{p_o}{p} e^{-kt} \cos(pt - \alpha_o)\right]. \tag{20}$$

From (20) follows that the force consists of the weights *Q* and *G* and one variable fading component and the *A* factor determines the acceleration amplitude and the dynamic load depending on the duration of the initial rotation and the frequency of the fading oscillations. The acceleration amplitudes and the maximum value of the dynamic force after the completion of the acceleration process depend on the ratio of acceleration time t_m and the fading oscillations period for the load *T*.

As $k << p$ for performing analysis of the cases considered one may assume $p \approx p_o$ and $\alpha \approx \alpha_o \approx 0$. In such case the lowest values of *F* are received at $pt_m = 2\pi, 4\pi, \ldots$, i. e. when $t_m = T, 2T, \ldots$ and the dynamic loads are highest at $pt_m = \pi, 3\pi, \ldots$, when $t_m = T/2, 3T/2, \ldots$.

During the lifting, the maximum dynamic load is normally shown at acceleration. This time is close to and most frequently higher than the period of the fading oscillations of the load. In case of lowering, the maximum dynamic load is read after the acceleration. In such case the acceleration time may be close to the cycle or the half-cycle of the fading oscillations of the load. This means that for given lifting devices and load according to the rope length (the period of the fading oscillations), both very high and very low values of the maximum dynamic load may be received. In this case, in the most unfavourable case (t_m=T/2) in lowering, the dynamic load is greater than the load in lifting which is confirmed by the results attained in tests of dynamic loads of lifting devices of electric hoist [2].

During the processes of deceleration, the load oscillations and dynamic loads are described by differential equations (1) and (2) and in equation (3) the average starting moment of the motor M_{mot} will be replaced by the braking moment M_b taking into account the respective sign. At deducing of the equation for delays, the same initial conditions are used ($t = 0$, $d^2\varphi/dt^2 = 0$ и $d^3\varphi/dt^3 = 0$) that correspond to movement with stable speed and for the delays and the dynamic loads during the stoppage the equations (9), (10) and (13) may be used.

For analysis and estimation of the dynamic loads of the components of the lifting device and metal structure supports, the dynamic factor k_d is used determined as a ratio of the maximum force F_{max} to the weight of rated load Q_n and the weight of the load-catching device *G*

$$k_d = \frac{F_{max}}{Q_n + G} . \tag{21}$$

The dynamic overload is calculated by the factor

$$\Delta k_d = \frac{F_{max} - (Q_n + G)}{Q_n + G}. \tag{22}$$

In case of lifting, the maximum dynamic overload arises during the acceleration in the time $t_m = T/2$ and according to (13)

$$F_{max} = Q_n + G + m_{2n}\bar{a}\left(1 + \frac{p_o}{p}e^{-\frac{kT}{2}}\right), \tag{23}$$

whence for the dynamic overload the following equation is received

$$\Delta k_d = A_1\bar{a} = A_1\frac{D}{2u_r u_p}\bar{\varepsilon} = A_1\frac{D(M_{mot} - M_{sn})}{2u_r u_p J_o}, \tag{24}$$

where $A_1 = \left(1 + \frac{p_o}{p}e^{-\frac{kT}{2}}\right)/g$; $J_o = J_1 + J_2$, *kgm²*; $M_{sn} = \frac{D(Q_n + G)}{2u_r u_p \eta}$, *Nm*.

During deceleration, after lowering with braking moment M_D, the dynamic overload will be:

$$\Delta k_d = A_1\frac{D(M_b - M'_{sn})}{2u_r u_p J_o}, \tag{25}$$

where $M'_{sn} = \frac{D(Q_n + G)}{2u_r u_p}\eta$, *Nm*.

In case of lowering and deceleration after lifting, the maximum dynamic overload in most unfavourable case arises after the transitional process with duration $t_m = T/2$ and the factor A according to the (18) will be $A = 1 + e^{-\frac{kT}{2}}$. (26)

The maximum dynamic overload after completion of the transitional process arises after some new time interval $t = T/2$ where from (20) follows that the maximum possible dynamic load is

$$F_{max} = Q_n + G + m_{2n}|\bar{a}|\left(1 + e^{-\frac{kT}{2}}\right)\frac{p_o}{p}e^{-\frac{kT}{2}}. \tag{27}$$

In case of lowering with average starting moment M_{mot} the dynamic overload factor will be

$$\Delta k_d = A_2|\bar{a}| = A_2\frac{D}{2u_r u_p}|\bar{\varepsilon}| = A_2\frac{D(M_{mot} + M_{sn})}{2u_r u_p J_o}, \tag{28}$$

where $A_2 = \left(1 + e^{-\frac{kT}{2}}\right)\frac{p_o}{p}e^{-\frac{kT}{2}}/g$.

In case of stopping after lifting with braking moment M_b

$$\Delta k_d = A_2\frac{D(M_b + M'_{sn})}{2u_r u_p J_o}. \tag{29}$$

3. CONCLUSIONS

1. Methods for investigation of the influence of the start / stop time ratio towards the duration of the fading oscillations of the load onto dynamic load of the elements of lifting device, are established.
2. Dependencies for maximum dynamic overload for most unfavourable case of the ratio of the durations of the transitional processes and the half-cycle of the fading vertical oscillations of the load are received.
3. Methods for calculation of rational durations of transitional processes on a base of the dependencies received according to the rope length may be established in order to decrease the dynamic load of the elements of the lifting device.

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