The dynamic control method of the weight cargo transported lifting machinery

A V Egorov¹, S V Dorohin², A V Lysyannikov³, Yu F Kaizer³, A V Kuznetsov⁴, N N Lysyannikova³, V L Tyukanov³ and V G Shram³

¹ Volga State University of Technology, 424000, Yoshkar-Ola, 3, Lenin Square, Russia
² Voronezh State University of Forestry and Technologies Named after G.F. Morozov, 394087, Voronezh, 8, Timiryazeva str., Russia
³ Siberian Federal University, 660041, Krasnoyarsk, 82/6, Svobodny Avenu, Russia
⁴ Krasnoyarsk State Agrarian University, 660049, Krasnoyarsk, 2 Kirenskogo, Russia

E-mail: kaiser171074@mail.ru

Abstract. The problem of improving the safety of the use of lifting machinery is particularly relevant for regions where intensive industrial and civil construction is carried out. To date, a significant amount of work has been done, designed at the theoretical level to justify the design and safe operation of lifting machinery. However, the existing methods of control of the total weight of the cargo transported by the lifting machinery are based mainly on the strain measurement methods developed in the second half of the twentieth century. The main disadvantage of systems that implement the strain measurement method is the need to integrate the weighing module directly into the design of the lifting machinery. A significant number of accidents of hoisting machines indicates the fuzzy operation of the protection systems of hoisting machines from overload, made on the basis of tensometric methods. As shown by the results of studies in the operation of various types of electric motors under load torque fluctuations occur at the rated speed. The use of this pattern allows weighing the cargo by the dynamic method transported by a lifting machinery. This article is devoted to the development and scientific and technical substantiation of the method of dynamic control of the weight of the cargo transported by the lifting machinery, based on the analysis of the dynamics of the angular accelerations of the rotor of the electric motor of the lifting drive.

1. Introduction
The problem of increasing the safety of using lifting machinery is particularly relevant for regions in which intensive industrial and civil construction is carried out [1, 2].

To date, a significant amount of work has been carried out, designed at the theoretical level to substantiate the design and safe operating modes of lifting machinery [3, 4].

However, the existing methods of controlling the total weight of cargo transported by a lifting machinery are based mainly on strain gauge measurement methods [5, 6], developed in the second half of the twentieth century. The main disadvantage of systems that implement strain gauge measurement is the need to integrate the weighing module directly into the design of a lifting machinery. A significant number of accidents of lifting machines indicates a lack of clarity in the operation of systems for protecting lifting machines against overload, based on strain gauge methods.
Thus, to date, the actual task is the development of methods for controlling the weight of the cargo transported by a lifting machine, including those development on the basis of new physical principles.

2. The development dynamic method of controlling the weight of the cargo transported lifting machinery

As the results of studies [7-11] show, when various types of electric motors are operating under load, torque oscillations occur at the nominal speed. The use of this pattern allows weighing of the cargo by the dynamic method transported by a lifting machinery.

Consider a system of lifting cargo with a lifting device (figure 1).

According to [12], the reduced moment of inertia of the system of rotating loads is the moment of inertia of the system, consisting only of elements rotating with the angular velocity of the electric motor shaft but having a kinetic energy reserve equal to the kinetic energy reserve of the actual system.

From the condition of invariance of the kinetic energy it follows that for a system consisting of an electric motor 1, a clutch with an electromagnetic brake 2, a reducer 3, a gear coupling 4, a drum 5, assuming a small resistance to movement from the air in comparison with other components, we obtain

\[
J (\omega) \frac{\omega_{\text{motor}}^2}{2} = J_{e,\text{motor}} (\omega) \frac{\omega_{\text{motor}}^2}{2} + J_{\text{coupling}} (\omega) \frac{\omega_{\text{motor}}^2}{2} + J_{\text{reduction}} (\omega) \frac{\omega_{\text{motor}}^2}{2} + J_{\text{gear coupling}} (\omega) \frac{\omega_{\text{drum}}^2}{2} + \\
+ J_{\text{drum}} (\omega) \frac{\omega_{\text{drum}}^2}{2} + m \frac{V_{\text{drum}}^2}{2},
\]

where \( J (\omega) \) - the moment of inertia brought to the axis of rotation of the rotor of the electric motor of rotating and moving progressively loads of the entire lifting mechanism, taking into account mechanical losses; \( J_{e,\text{motor}} (\omega) \) - the moment of inertia of the rotor of the electric motor, taking into account mechanical losses; \( J_{\text{coupling}} (\omega) \) - the moment of inertia of the coupling with an electromagnetic brake, taking into account mechanical losses; \( J_{\text{reduction}} (\omega) \) - the moment of inertia of the reduction gear, taking into account mechanical losses; \( J_{\text{drum}} (\omega) \) - the moment of inertia
of the drum, taking into account mechanical losses; $J_{\text{gear.coupl.}}(\omega)$ - the moment of inertia of the gear safety coupling; $\omega_{\text{motor}}$ - angular velocity of the rotor of the electric motor; $\omega_{\text{drum}}$ - angular velocity of the drum; $V_{\text{drum}}$ - linear speed moving of the drum surface; $m$ - the weight of the cargo moved by the lifting gear.

From here the required reduced moment of inertia of the system is

$$J(\omega) = J_{\text{e.motor}}(\omega) + J_{\text{coupling}}(\omega) + J_{\text{reduction}}(\omega) + J_{\text{gear.coupl.}}(\omega) \left( \frac{\omega_{\text{drum}}}{\omega_{\text{motor}}} \right)^2 + J_{\text{drum}}(\omega) \left( \frac{\omega_{\text{drum}}}{\omega_{\text{motor}}} \right)^2 + m \frac{V_{\text{drum}}^2}{\omega_{\text{motor}}^2};$$

(2)

The gear ratio between the motor and the drum is $i$. Then (2) can be represented as

$$J(\omega) = J_{\text{e.motor}}(\omega) + J_{\text{coupling}}(\omega) + J_{\text{reduction}}(\omega) + J_{\text{gear.coupl.}}(\omega) \frac{1}{i^2} + J_{\text{drum}}(\omega) \frac{1}{i^2} + m \frac{r_{\text{drum}}^2}{i^2}.$$  

(3)

The dependence of the motor torque on the rotor speed of the motor, expressed through the angular acceleration of the rotor of the electric motor during acceleration of the lifting machine, is calculated using equation (4).

$$M(\omega) = J(\omega) \varepsilon(\omega) = \left[ J_{\text{e.motor}}(\omega) + J_{\text{coupling}}(\omega) + J_{\text{reduction}}(\omega) + J_{\text{gear.coupl.}}(\omega) \frac{1}{i^2} + J_{\text{drum}}(\omega) \frac{1}{i^2} \right] \varepsilon_1(\omega),$$

(4)

where $\varepsilon_1(\omega)$ - angular acceleration of the rotor of the electric motor during acceleration of the electric motor 1 of the lifting machinery without cargo 6.

The calculated equation of torque during acceleration of the electric motor 1 of the lifting machinery with a cargo of 6 weight $m$:

$$M(\omega) = \left[ J_{\text{e.motor}}(\omega) + J_{\text{coupling}}(\omega) + J_{\text{reduction}}(\omega) + J_{\text{gear.coupl.}}(\omega) \frac{1}{i^2} + J_{\text{drum}}(\omega) \frac{1}{i^2} + m_{\text{cargo}} \frac{r_{\text{drum}}^2}{i^2} \right] \varepsilon_2(\omega),$$

(5)

where $\varepsilon_2(\omega)$ - angular acceleration of the rotor of the electric motor during acceleration of the electric motor 1 of the lifting machinery with a cargo of 6 weight $m_{\text{cargo}}$.

Equating (5) and (6), we determine the weight of the cargo $m_{\text{cargo}}$. 

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\[ m_{\text{cargo}} = \frac{i^2}{J_{\text{drum}}} \left( J_{\text{motor}}(\omega) + J_{\text{coupling}}(\omega) + J_{\text{reduction}}(\omega) + J_{\text{gear.coupl.}}(\omega) \frac{1}{i^2} + J_{\text{drum}}(\omega) \frac{1}{i^2} \right) \left( \varepsilon_1(\omega) - \varepsilon_2(\omega) \right) \] (7)

Similarly, the dynamic method of controlling the weight of the cargo transported by the lifting machinery, based on the equality of the traction forces developed by the drum during acceleration without cargo and with the cargo, can be solved on the basis of the analysis of the dynamics of linear acceleration.

Figure 2 shows the implementation of the proposed dynamic method of controlling the weight of the cargo lifting machinery.

Figure 2. Dynamic method of controlling the weight of the cargo lifting machinery: 1 - Lifting machinery; 2 - Cargo securing device; 3 - Cargo.

The lifting machinery 1 is equipped with a cargo fixing device 2, on which a cargo of unknown weight 3 is fixed.

Implemented the proposed dynamic method of controlling the weight of the cargo transported lifting machinery as follows.

At the initial stage, the cargo securing device 2 does not fix the cargo of an unknown weight 3.

The cargo securing device 2 weight \( m_{\text{CSD}} \) when lifting in a given altitude interval with a specific law of change in the frequency of the power supply drive motor voltage rise is reported acceleration \( a \).

Then the cargo securing device 2 with weight \( m_{\text{CSD}} \) is returned to the height of the beginning of the implementation of the initial stage and the cargo 3 of unknown weight \( m_{\text{cargo}} \) is fixed on it. The device for fixing the cargo 2 and a cargo of unknown weight 3 rigidly fixed on it when lifting in the same predetermined height range with the same specific law of changing the frequency of the power supply motor driving the voltage increase is reported by acceleration \( a_i \).

The force applied to the cargo securing device 2 and the cargo of an unknown weight 3, from the side of the lifting motor drive, we denote by \( F \).

We write the projections of the acting forces on the axis Oy for the first and second lifts:

\[ F = m_{\text{CSD}} (a + g), \] (8)

\[ F = (m_{\text{CSD}} + m_{\text{cargo}})(a_i + g), \] (9)
where $g$ - acceleration of gravity.

Since the law of voltage variation in the supply network during the first and second lifts was identical, the force applied by the sides of the drive electric motor of lift F developed during the first and second lift the same.

Equating (8) to (9), we determine the unknown weight $m_{\text{cargo}}$:

$$m_{\text{cargo}} = m_{\text{CSD}} \frac{(a - a_c)}{(a_1 + g)},$$

(10)

3. Conclusion

The dynamic method of controlling the weight of the cargo lifting device will allow you to signal the excess of the permissible weight of the cargo directly at the beginning of the lifting of the cargo. An application for the invention of the Russian Federation was submitted to the presented dynamic method for controlling the weight of the cargo on a lifting device based on the analysis of vertical accelerations [13].

References

[1] Huang Y and Huang J 2018 Study on safety monitoring system of construction cranes in Guangzhou 39th International Annual Conference of the American Society for Engineering Management Bridging the Gap Between Engineering and Business pp 816–23
[2] Sun G-F and Huang J-X 2006 Research on dynamic behavior of construction crane during luffing motion with payload Journal of Shenyang Jianzhu University (Natural Science) 22 524–8
[3] Sun G and Liu J 2006 Dynamic responses of hydraulic crane during luffing motion Mechanism and Machine Theory 41 1273–88
[4] Sun G and Kleeberger M 2003 Dynamic responses of hydraulic mobile crane with consideration of the drive system Mechanism and Machine Theory 38 1489–508
[5] Jones W H 1971 Electronic overload indicating system for mobile construction cranes SAE Technical Papers (DOI: 10.4271/710705)
[6] Rosenfield M J and Wendler B H 1975 Evaluation of load-indicating devices (LIDS) for mobile construction cranes Constr Eng Res Lab Tech Rep M 188
[7] Zhongbin W, Bin X, Zhen L, Ruijuan C, Zhiyong R, Yuefeng D, Eiji I, Muneshi M, Takashi O and Yasumaru H 2019 Modelling and verification of driving torque management for electric tractor: Dual-mode driving intention interpretation with torque demand restriction Biosystems Engineering 182 65–83
[8] Bowen N, Shanmei C, Baokang Y and Fengxing Z 2018 Examination and implementation for direct torque controlled permanent magnet synchronous motor with space vector modulation First Published 233 153–63
[9] Ortega R, Barabanov N and Valderrama G 2001 Direct torque control of induction motors: Stability analysis and performance improvement IEEE Transactions on Automatic Control 46 1209–22
[10] Buja, Giuseppe, Casadei, Domenico, Serra and Giovanni 1998 Direct stator flux and torque control of an induction motor: Theoretical analysis and experimental results IECON Proceedings (Industrial Electronics Conference) 1 pp 50–64
[11] Cecati C and Rotondale N 1999 Torque and speed regulation of induction motors using the passivity theory approach IEEE Transactions on Industrial Electronics 46 119–27
[12] Kasatkin A S 1983 Electrical Engineering 440 p
[13] Egorov A V 2018 The method of determining the weight of the cargo transported by the vehicle Application for the patent of the RF 2018139179