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Analysis of Hoisting Electric Drive Systems in Braking Modes

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Abstract

Braking in hoisting drives is used to reduce the speed of the load (cargo) or to stop it completely. Braking can be realized using electrical and mechanical means; mechanical brake is used to hold the suspended cargo after disconnecting the driving motor from the supply. In modern hoisting electric drive systems, in addition to mechanical braking, electric braking is also used. Electric braking can be realized by transforming the driving electrical motor into the braking mode. The dynamic loads created in the mechanical parts of the hoisting drive depend on the magnitude of the braking torque, load torque, motor speed and other parameters. In this paper a mathematical analysis of the factors on which the braking process depend are analyzed. Recommendations about the appropriate braking methodology by which dynamic loads can be minimized are brought out. A mathematical model of the proposed hoisting electric drive is developed.

Keywords: Hoisting drives; Braking mode; Dynamic loads; Induction motor

1. Introduction

Safety and control is crucial in all crane applications. Operating cranes demand precision and leave no margin for errors. The critical starting and stopping can lead to harmful jerks and false tripping, not to mention accidental dropping of the load. Safety for both people and load is of the outmost importance in crane operation.

Optimizing the operation of the crane is also essential. Minimizing cycle time improves productivity. There are also great savings to be made from extended equipment lifetime and more reliable operation without unplanned stops and downtime. In hoisting motor-drive systems, a torsional vibration is often generated due to the presence of an elastic element in torque transmission. Such a mechanical system is modeled as a two-mass system. In such system, vibration suppression of a low inertia ratio, i.e. when the motor inertia is larger than the load inertia is very difficult. Such a problem was and still a crucial issue for many researchers [1-14]. In this research, an approach to find the maximum loads (torque) created in the elastic elements (ropes) of the electromechanical hoisting system at braking and give recommendations for minimizing these dynamic loads.

2. Design Considerations And Mathematical Representation

In cranes, the braking torque required for holding the suspended nominal cargo $T_{br(nom)}$ can be calculated as follows:

$$T_{br(nom)} = K_s \cdot T^{\downarrow}_{L(nom)} \tag{1}$$

Where;

K_s- Safety coefficient, K_s=1.5 - 2.5,

$T^{\downarrow}_{L(nom)}$ – Nominal load (cargo) torque at lowering, referred to the motor's shaft,

$$T^{\downarrow}_{L(nom)} = T_{L(nom)} \cdot \eta_{nom} \tag{2}$$

$T_{L(nom)}$ - Nominal load (cargo) torque, referred to the motor's shaft.

η_{nom} - Nominal efficiency coefficient of the mechanism.

Typically, hoisting mechanisms can be physically modeled as two-mass electromechanical systems. Here the first mass (inertia) J_1 represents the equivalent inertia of the motor rotor and all elements rotating on the motor shaft with speed ω_1 including the brake drum, coupling, etc. On the other hand the second inertia J_2 represents the equivalent inertia of the suspended cargo and all elements moving with and at common speed ω_2 to the cargo. The motion is transferred from J_1 to J_2 via ropes with a stiffness coefficient C_{12}. It is obvious, for analysis, that both J_2 and C_{12} are referred to the motor shaft.

The braking torque T_{br} in general is a function of time, motor speed ω_1 and other parameters; it can be realized by mechanical brake or electrically by transferring the motor into braking mode.

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The basic equations describing the braking process at lowering:

$$T_{\mathrm{br}} - T_{\mathrm{el}} = J_1 \frac{d^2\varphi_1}{dt^2} \qquad (3.1.)$$

$$T_{\mathrm{el}} - T_{\mathrm{L}}^{\downarrow} = J_2 \frac{d^2\varphi_2}{dt^2} \qquad (3.2.)$$

Where

$T_{\mathrm{el}} = C_{12}(\varphi_1 - \varphi_2)$ - Elasticity torque induced in the elastic elements,

φ_1, φ_2 - The angle of rotation of the first mass and the second, respectively.

For calculation, the speed of both masses is assumed to be negative at lowering, and positive at lifting respectively, i.e.:

$\omega_1 = \frac{d\varphi_1}{dt}$ <0 and $\omega_2 = \frac{d\varphi_2}{dt}$ <0 at lowering,

$\omega_1 = \frac{d\varphi_1}{dt}$ >0 and $\omega_2 = \frac{d\varphi_2}{dt}$ >0 at lifting.

Assuming the braking process at lowering can be accomplished only when $T_{br} > T_L^{\downarrow}$, equations 3 can be used for the given process until the motor's speed $\omega_1 \neq 0$, the braking torque is also assumed constant over the braking process and does not depend on the motor's speed (Figure 1).

Dividing (3.1) by J_1 and (3.2) by J_2 and subtracting 3.2 from 3.1 after division we get:

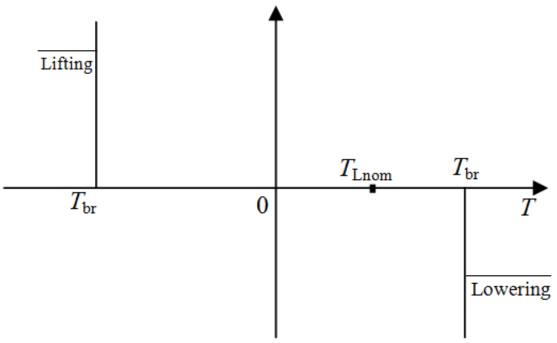


Figure 1: Speed-torque diagram of the hoisting drive with constant braking torque.

$$\frac{1}{J_1}T_{\mathrm{br}} - \left(\frac{1}{J_1}T_{\mathrm{el}} + \frac{1}{J_2}T_{\mathrm{el}}\right) + \frac{1}{J_2}T_{\mathrm{L}} = \frac{d^2(\varphi_1 - \varphi_2)}{dt^2} \qquad (4)$$

Multiplying (4) by the stiffness coefficient C_{12} we get

$$T^2 \frac{d^2 T_{el}}{dt^2} + T_{el} = \frac{T_{br}J_2 + T_L J_1}{J} \qquad (5)$$

Where

$J = J_1 + J_2,$

$T = \frac{1}{\Omega}$, Ω - The frequency of oscillations of the second-order system.

If the losses in the ropes due to deformation are neglected, then the ropes elasticity torque is

$$T_{\mathrm{el}} = C_{12}(\varphi_1 - \varphi_2).$$

The solution of (5) can be found in the following form

$$T_{\mathrm{el}} = \frac{T_{\mathrm{br}}}{\gamma_2} + \frac{T_{\mathrm{L}}}{\gamma_1} + A\sin\Omega t + B\cos\Omega t \qquad (6)$$

Where

$$\gamma_1 = \frac{J}{J_1}, \quad \gamma_2 = \frac{J}{J_2}.$$

A, *B*- constants that can be found from the initial conditions of braking, the speed of both masses is equal, i.e. $\omega_1 = \omega_2$, and the elasticity torque equals the load torque, thus substituting in (6), (when t = 0, $\omega_1 = \omega_2$ and $T_{\mathrm{el}} = T_{\mathrm{L}}$) we get

$$T_{el} = T_{\mathrm{L}} + \frac{T_{\mathrm{br}} - T_{\mathrm{L}}}{\gamma_2}(1 - \cos\Omega t) \qquad (7)$$

The time at which the periodic oscillations of the elasticity torque will have maximum value is

$$t_{\mathrm{m}} = (2n-1)\frac{\pi}{\Omega} \qquad (8)$$

Where n=1, 2, 3 … - the number of the period.

Respectively the maximum value of the elasticity torque at the half of the first period (n=1) is

$$T_{\mathrm{el\ max}} = T_{\mathrm{L}} + \frac{2(T_{\mathrm{br}} - T_{\mathrm{L}})}{\gamma_2} \qquad (9)$$

The ratio between the maximum instantaneous $T_{\mathrm{el\ max}}$ and the maximum calculated $T_{\mathrm{el\ mc}}$ (by design) elasticity torque value is expressed by the coefficient K_{d}

$$K_{\mathrm{d}} = \frac{T_{\mathrm{el\ max}}}{\mathrm{T}_{\mathrm{el\ mc}}},$$

Where

$$T_{\mathrm{el\ mc}} = \frac{T_{\mathrm{m}} - T_{\mathrm{L}}^{\uparrow}}{\gamma_2} + T_{\mathrm{L}}^{\uparrow},$$

Where T_{m} -The motor's torque.

Usually in cranes design $T_{\text{m}} = 2T^{\uparrow}_{\text{L nom}}$ and $T^{\uparrow}_{\text{L}} = T^{\uparrow}_{\text{L nom}}$, thus;

$$T_{\text{el mc}} = \frac{T^{\uparrow}_{\text{L}}}{\gamma_2}(\gamma_2 + 1) \tag{10}$$

Substituting (9) and (10) in K_{d}, we get

$$K_{\text{d}} = \left[1 + \frac{2K_{\text{br}} - 3}{\gamma_2 + 1}\right]\eta^2_{\text{nom}} \tag{11}$$

Where

$$K_{\text{br}} = \frac{T_{\text{br}}}{T^{\downarrow}_{\text{L}}}$$

Equation (11) shows that when $K_{\text{br}} \leq 1.5$, Kd<1. This means that at breaking, the maximum load created in the rope at lowering will be less than that at lifting. It shows also, that when $K_{\text{br}}=1.5$, K_{d} does not depend on γ_2 and the less is K_{d}, the less will be η_{nom}.

Coefficient K_{OL} shows the ratio between the maximum instantaneous elasticity torque value and the load torque value:

$$K_{\text{OL}} = \frac{T_{\text{el max}}}{T^{\downarrow}_{\text{L}}} = 1 + \frac{2(K_{\text{br}} - 1)}{\gamma_2} \tag{12}$$

Equations (9-12) are used to estimate the dynamic loads created in the links of the mechanical part of the hoisting drive, that should not exceed 150% of the steady-state load value. On the other hand K_{d} practically should not exceed 1.25.

3. Simulation and Results

In order to find expressions for the speed $\omega_1(t)$ and $\omega_2(t)$, (7) should be substituted in (3.1) and 3.2, then integrating both equations and substituting the value of the initial speed for both masses to be the nominal value ω_{nom} we get:

$$\omega_1 = -\omega_{\text{nom}} + \frac{T_{\text{br}} - T_{\text{L}}}{J}t + \frac{T_{\text{br}} - T_{\text{L}}}{J\Omega}\frac{J_2}{J_1}\sin\Omega t \tag{13}$$

$$\omega_2 = -\omega_{\text{nom}} + \frac{T_{\text{br}} - T_{\text{L}}}{J}t - \frac{T_{\text{br}} - T_{\text{L}}}{J\Omega}\sin\Omega t \tag{14}$$

As in hoisting drives $J_1 > J_2$ then the amplitude of the speed $\omega_1(t)$ should be less that one of $\omega_2(t)$. After braking $\omega_1(t)$ will reach zero after a finite time which can be found from (13) by substituting $\omega_1(t) = 0$ we get:

$$\frac{T_{\text{br}} - T_{\text{L}}}{J}\left(t + \frac{J_2}{J_1\Omega}\sin\Omega t\right) = \omega_{\text{nom}}.$$

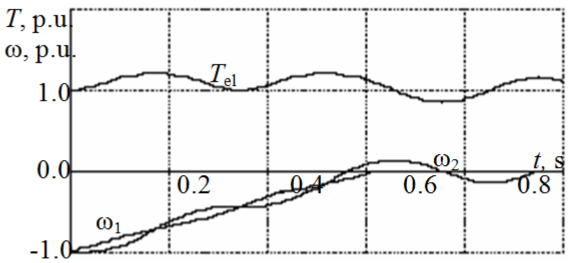
After the motors speed reaches zero, the braking torque will be assumed to be infinity ($T_{\text{br}} = \infty$) and the motor's speed will remain zero. The results of simulation for the braking process at lowering the nominal load (3 ton) for a particular crane are depicted in Fig. 2 (a and b).

The analysis of the obtained simulation results for different hoisting drives conducted elsewhere but omitted for brevity, revealed that the sudden intensive braking of hoisting drives at lowering the nominal load with the nominal speed does not lead to impermissible loading in the elastic links of the mechanism like ropes.

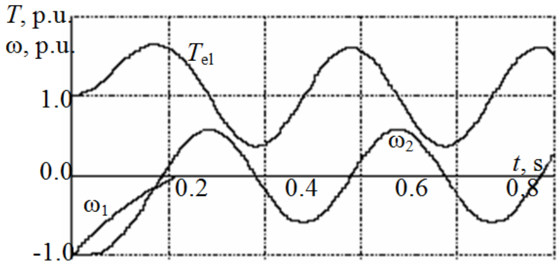
The braking process at lifting with a constant braking torque (T_{br}) is also simulated using the following equations:

$$-T_{\text{br}} - T_{\text{el}} = J_1\frac{d^2\varphi_1}{dt^2} \tag{15.1}$$

$$T_{\text{el}} - T^{\uparrow}_{\text{L}} = J_2\frac{d^2\varphi_2}{dt^2} \tag{15.2}$$



a) $K_{\text{br}} = 1.5$



b) $K_{\text{br}} = 2.5$

Figure 2: Braking transient at lowering the nominal load with two different braking torque values.

At lowering the nominal load which corresponds to T_{Lnom}, the motor's torque required for lowering the nominal load $T^{\downarrow}_{\text{L(nom)}}$ was calculated using (2), while at lifting the same nominal load, the motor's torque is calculated using the following equation:

$$T^{\uparrow}_{\text{L nom}} = \frac{T_{\text{L}}}{\eta_{\text{nom}}} \tag{16}$$

The solution for the differential equations (15.1) and (15.2) substituting the initial conditions (when $t = 0$, $\omega_1 = \omega_2$, $T_{\text{el}} = T_{\text{L}}$

and $\frac{dT_{\text{el}}}{dt} = 0$) is:

$$T_{\text{el}} = T_{\text{L}} - \frac{T_{\text{br}} + T_{\text{L}}}{\gamma_2}(1 - \cos\Omega t) \tag{17}$$

The minimum value of the elasticity torque at lifting can be found by substituting $t = t_m = (2n-1)\frac{\pi}{\Omega}$, (found in (8) in (17)):

$$T_{\text{el min}} = T_L - \frac{2(T_{br} + T_L)}{\gamma_2} \quad (18)$$

An expression for the speed of the first mass can be found by solving (15.1) taking into account (17):

$$\omega_1 = \omega_{nom} - \frac{T_{br} + T_L}{J} t - \frac{T_{br} + T_L}{J\Omega} \frac{J_2}{J_1} \sin \Omega t \quad (19)$$

Respectively for the speed of the second mass

$$\omega_2 = \omega_{nom} - \frac{T_{br} + T_L}{J} t + \frac{T_{br} + T_L}{J\Omega} \sin \Omega t \quad (20)$$

At the first stage of braking, when the motor's speed $\omega_1 = 0$, the mechanism is to be considered as a one-mass system, thus

$$T_{el} - T_L = J_2 \frac{d^2 \varphi_2}{dt^2} \quad (21)$$

After a complete stop of the motor, the initial conditions for the formed single-mass system are:
$T_{el} = T_{\text{el min}}$ defined in (18), $\omega_2 = \omega_1 = 0$.

The solution of (21) is:

$$T_{el} = T_L - 2\frac{T_L - T_{br}}{\gamma_2} \cos \Omega_2 t \quad (22)$$

The maximum value of T_{el} will occur when $\Omega_2 t = \pi$, thus

$$T_{\text{el max}} = T_L + \frac{2(T_{br} + T_L)}{\gamma_2} \quad (23)$$

For this case $K_d = \dfrac{2\left(T_{br} + 2T_L^{\uparrow}\right) + \gamma_2 T_L^{\uparrow}}{T_L^{\uparrow}(\gamma_2 + 1)}$.

After affecting some mathematical rearrangements and substituting $T_L^{\downarrow} = T_L^{\uparrow}\eta^2$, K_d can be rewritten in the following form

$$K_d = 1 + \frac{2K_{br}\eta_{nom}^2 + 1}{\gamma_2 + 1} \quad (24)$$

K_{OL} for this case will be

$$K_{OL} = \frac{T_{\text{el max}}}{T_L^{\uparrow}} = 1 + \frac{2\left(K_{br}\eta_{nom}^2 + 1\right)}{\gamma_2} \quad (25)$$

Simulations of the braking process at lifting were made for the same 3-ton hoisting capacity crane. The results of simulation revealed that assuming similar conditions (the load weight, braking torque, motor's speed, etc.) at lifting and lowering, the dynamic loads created in the elastic elements (ropes) are greater at lifting.

As seen from (17), (19) and (20) the maximum value of the elasticity torque can occur at finite initial conditions of the braking mode at lifting, particularly, there is a finite value of the oscillations frequency at which the elasticity torque has maximum value. Substituting the time at which T_{el} will have maximum magnitude $t_m = \frac{\pi}{\Omega}$ in (19) and equating (19) to zero, the frequency at which T_{el} will have maximum can be found as follows:

$$\Omega_m = \frac{T_{br} + T_L}{\omega_{nom} J} \pi \quad (26.1)$$

The frequency of the vibrations induced in the two-mass system depends on the stiffness coefficient C_{12} of the ropes and the moment of inertia of the first and the second mass (J_1 and J_2):

$$\Omega = \sqrt{\frac{C_{12} J}{J_1 J_2}} \quad (26.2)$$

Assuming J_1 and J_2 to be constant for the hoisting drive, a value of C_{12m} (which corresponds to the maximum frequency found in (26)) can be found by substituting $\Omega = \Omega_m$ in (26.2):

$$C_{12m} = \frac{\pi^2 (T_{br} + T_L)^2}{J\gamma_1\gamma_2\omega_{nom}^2} \quad (27)$$

As the stiffness coefficient of the ropes $C_{12} = \frac{a}{l}$, where a- is a constant that depends on the material and the cross-sectional area of the rope, l- is the length of the rope, therefore a finite value of the rope length l_m that corresponds to C_{12m} as follows: (Assuming $\pi^2 \cong 10$)

$$l_m = 0{,}1\, a \frac{J\gamma_1\gamma_2\omega_{nom}^2}{(T_{br} + T_L)^2} \quad (28)$$

If the braking process with finite J, T_{br} and T_L occurs at a ropes length l_m, then the ropes will be exposed to maximum elasticity torque, but l_m can have values which maybe unreal for a certain drive.
The change in the cargo (load) weight will result in a change in the moment of inertia of the second mass of the two-mass mechanical system, thus γ_2 will also vary. The effect of the change of γ_2 on the dynamic loads induced in the ropes at braking at lifting the nominal load (at the nominal hoisting speed assuming critical ropes length l_m) is shown in Fig. 3.

Analysis of the curves depicted in Fig. 3 shows that for mechanisms with relatively low J_2 ($\gamma_2 > 20$), braking can be accomplished using pure mechanical brake without exposing the ropes to impermissible elasticity torque values (K_d =1.15-1.2). On the other hand, mechanical brake can cause harmful deformation in ropes for mechanisms with $\gamma_2 < 10$ at possible ropes length value (K_d >1.3 at K_{br} = 1.5). Higher values of K_d are possible at K_{br} = 2.5 but with unreal ropes length values.

The braking process at lifting the nominal load at the nominal speed with different braking torque values and ropes length was simulated; the results of simulation are depicted in Fig. 4 and Fig.5.

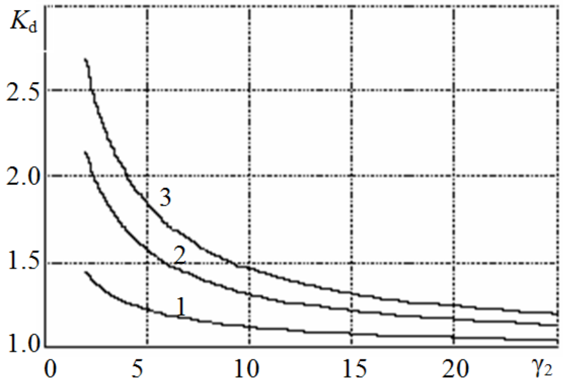


Figure 3: The relationship between K_d and γ_2 for the braking process at lifting the nominal load with different braking torque values (1-K_{br}=0.2; 2-K_{br}=1.5; 3-K_{br}=2.5).

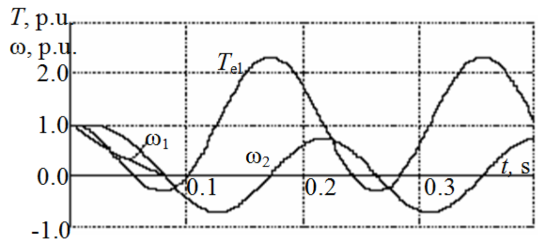


Figure 4: Simulation of the braking process at lifting the nominal load (3 ton). K_{br}=2.5, l=l_m=5.5m.

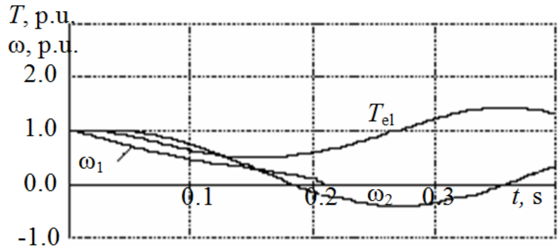


Figure 5: Simulation of the braking process at lifting the nominal load (3 ton). K_{br}=0.2, l=20m, l_m=37m.

The results of simulation of the above mentioned braking transient revealed that the use of the intensive mechanical brake with K_{br} = 2.5 (Fig. 4) will create impermissible dynamic elasticity torque in the ropes (K_d = 1.9), while the braking with a relatively reduced braking torque (Fig. 5) by the use of other braking means like electrical, does not expose ropes to impermissible dynamic elasticity torque (K_d = 1.2), but the transient time will be increased. Mechanical brake is recommended to be used only when the motors speed is decreased to values less than 0.1 of the rated value.

4. Conclusion

In this research study the braking transient of the hoisting electric drive systems was studied, the main parameters on which the elasticity torque induced in the ropes depends are defined. Equations for the calculating the maximum elasticity torque at braking with constant braking torque in lowering and lifting are derived. Analysis revealed that the maximum dynamic elasticity torque at lifting may be of a grater value than at lowering. For mechanisms with γ_2 < 15…20, the magnitude of the induced dynamic elasticity torque at braking is increased. Formulae for calculating the stiffness coefficient of the ropes and respectively the ropes length at which the maximum induced dynamic elasticity torque at braking occur are derived. Mechanical braking of hoisting mechanisms which have K_d>1.5-1.6 is not recommended, it is more rational to brake such mechanisms by transforming the electrical motor into braking mode, mechanical brake is recommended to be used only when the motors speed is less than 0.1 of the rated value.

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