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PROPOSALS FOR A CARGO ANTI-SWAY METHOD USING MOTOR TORQUE CONTROL

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Abstract. The paper discusses the cargo anti-sway method based on the induction motor torque control using modern variable frequency drives. The main equations relating the tractive force and the cargo sway angle, on the basis of which the motor torque control law is formulated for zero cargo sway at the end of accelerating and braking, are written. The results of simulating the behaviour of the two-mass trolley-cargo system are presented for the typical ratios of the cargo weight to the rope length, which support the assumption about the feasibility of cargo anti-sway control by direct motor torque control.

Key words: variable frequency drive, anti-sway control, container crane, induction motor torque control.

1. Introduction

One of the topical issues of automation and raising the efficiency of loading operations is cargo anti-sway control. Today, there are certain approaches to solving this task, ranging from mechanical solutions (reeving the ropes of the lifting device, pulling up the ropes for the spreaders [2, 8]) to electrical ones (setting up the container crane trolley speed or the boom swing speed with the zero vertical deviation of the cargo at the end of acceleration/braking). The electrical influence on the travel drive (swing drive) was as a rule implemented by applying the sign-reversing total motor torque to the travel gear [1, 3, 4], which caused accelerated wear both of the mechanical gears and the motor.

It should be noted that modern variable frequency drives (VFD) provide for control modes both for the speed and the torque. The control task for the torque can be fed through the analogue input, RS-485 (Modbus-RTU; Modbus-ASCII), CANopen or in other standards via COMM Card, as shown in Fig.1 on the example of modern VFD, C2000 from Delta Electronics Inc. Thereat, toque control can be implemented both by using the encoder and without it, in the sensorless mode.

2. The main ratios and control law for the moment for obtaining the expected character of the cargo sway angle

Let us consider the oscillations of the travel gear in the two-mass trolley-cargo system ensuring the movement along the boom of the container crane. In such a gear, all the processes are presented in a two-dimensional Cartesian coordinate system.

Considering the onset and damping of the oscillations in one plane makes it possible to formulate the fundamental principles of the engineering approach to the sway control problem using the available technical capabilities offered by VFD, as well as to develop the main VFD control algorithms for this process. For swing cranes, these principles must be supplemented with mathematical expressions enabling the transfer from the Cartesian coordinate system to the polar one.

Hereinafter, the movement of the two-mass trolley-cargo system will be divided into two stages: the dynamical stage, during which the speed of both masses changes due to the motor torque control, and the steady-state one, which is characterized by equal speeds of both masses and movement at a stabilized speed.

The specific feature of the first stage is VFD operation in the torque control mode (for the method IM TQC for Delta C2000 VFD, Fig. 1), when formulating the required law of the cargo's vertical deviation during the acceleration and braking stages of the travel gear, which is acceptable in terms of its safe operation.

The second stage takes place in the Induction motor field oriented sensorless vector control (IM FOC Sensorless) mode, whose specificity is the solution of the stabilization problem with a high accuracy of the motor shaft speed by the sensorless method using the data measured earlier for the equivalent circuit of the motor.

The two-mass trolley-cargo system is primarily driven by applying the tractive force F (i.e. the motor torque) to the first mass of the trolley, m_1. The shift of the trolley relative to the second mass (resulting from the applied force F) is accompanied by the vertical deviation of the rope having the length l by the angle α due to the cargo inertia (Fig. 2). This process is aracterized by the ratio of the trolley weight m_1 and the cargo plus the locking mechanism weight m_2.

At a deviation by the angle α, the gravity force $G_2 = m_2 g$ acting on the cargo can be resolved into the elastic force F_s, directed along the rope and the force F_t, directed along the tangent to the cargo sway trajectory, which is a circle with the radius l).

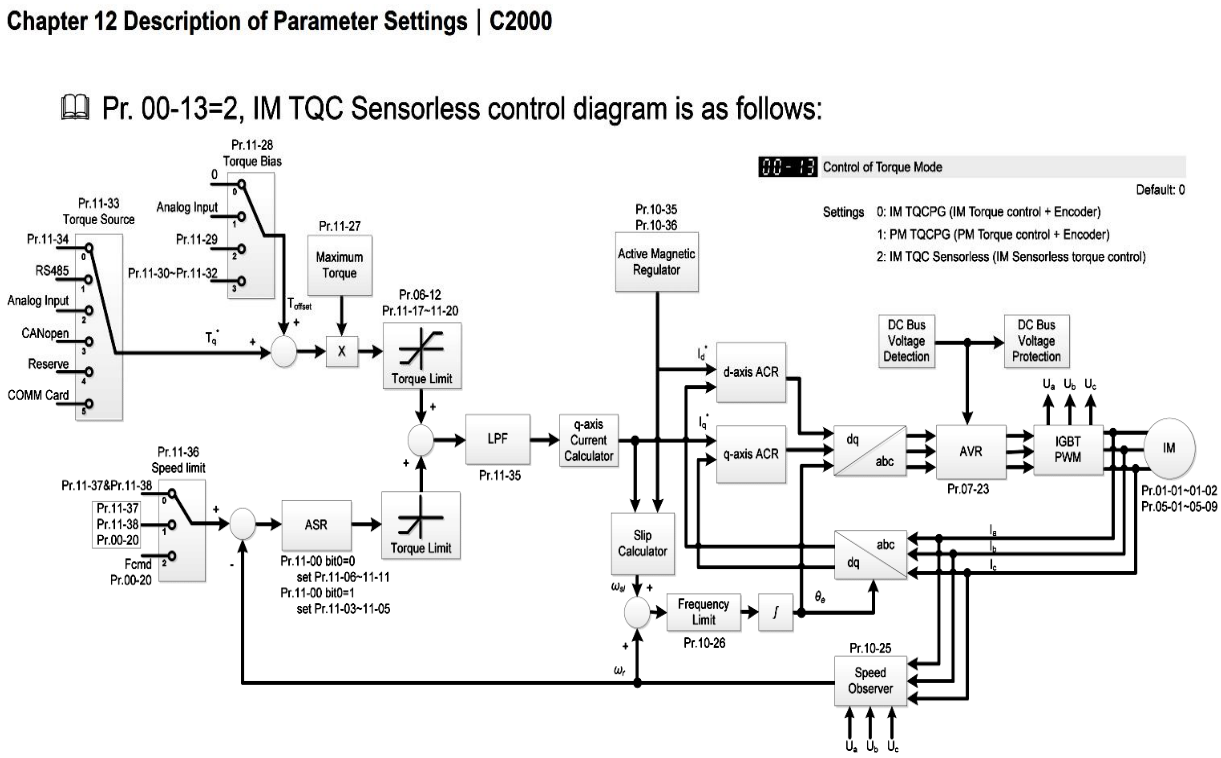


Fig.1. Induction motor in sensorless torque control mode for Delta C2000 VFD.

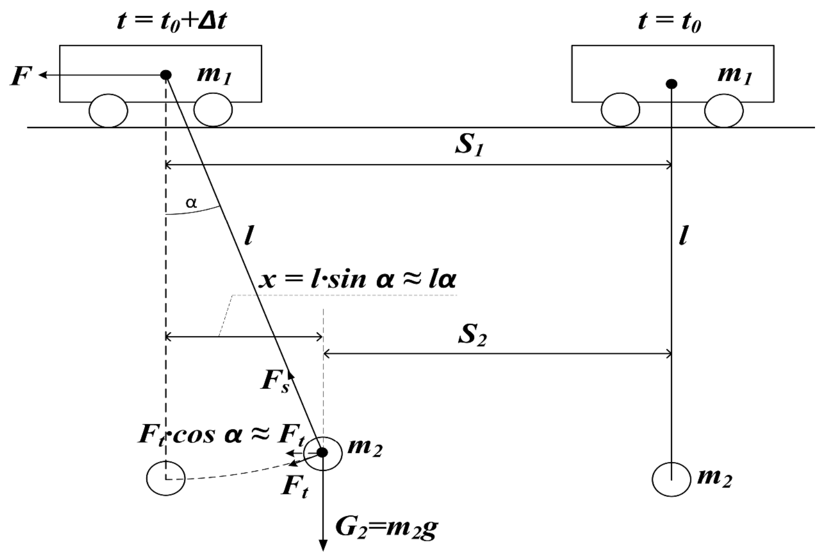


Fig. 2. Trolley-cargo system

Obviously:

$$F_s = m_2 g \cos\alpha ;$$

$$F_t = m_2 g \sin\alpha ,$$

where $g = 9{,}81\ m/s^2$ is the gravity of Earth.

To simplify the further presentation, let us consider the following assumptions. The projection of the cargo deviation onto the X axis is equal:

$$x = l \cdot \sin\alpha .$$

Taking into account that for small deviation angles (in radians) $\alpha \approx \sin\alpha$, we obtain:

$$x \approx l\alpha .$$

In practice, real cargo oscillations do not exceed 4^0. For such an angle, the relative error is 0.025 % for the assumption $\sin\alpha \approx \alpha$ and 0.24 % for $\cos\alpha \approx 1$. Therefore, we will further take with a high accuracy that *sin α = α,* and $\cos\alpha = 1$ for the above-mentioned range of the deviation angle.

The important outcome of this assumption is that the F_t component can be located not along the tangent to the segment of the circle, but along the horizontal axis, as the accurate value of the tangent component of the force G_2 equals:

$$F_t = m_2 g \sin\alpha \approx m_2 g\alpha ,$$

and the horizontal projection of the tangent component of F_t can be with a high accuracy equalled to F_t itself, as:

$$F_t \cos\alpha = F_t ,$$

which enables a further significant simplification of the mathematical description of the process.

It should be noted that the normal component F_s of the resolution of G_2, which is equal to the rope's funicular force, has no effect on the movement of m_2. However, it is one of the causes of the oscillations damping (due to the elasticity and flexibility of the rope), along with the frictional forces in places of the cargo suspension and rope and air resistance. The oscillations are caused by the tangent component F_t. It is actually the load force for the motor, indirectly via the first mass of the trolley m_1 in the two-mass trolley-cargo system.

Let us write the main equations describing the movement of the mechanical system being considered (Fig. 2) from the standstill at a moment of time $t_0 = 0$ at applying the tractive force F to the first mass m_1 resulting in a change of the speeds of the first and second masses and in the deviation of the rope by an angle α:

$$F - F_t = m_1 \frac{d^2 S_1}{dt^2} ; \tag{1}$$

$$F_t = m_2 \frac{d^2 S_2}{dt^2} . \tag{2}$$

Considering that the difference $\Delta S = S_1 - S_2$ in the distances covered by the trolley S_1 and the cargo S_2 during the acceleration stage, taking into account the assumption made that $\alpha = \sin\alpha$, can be defined as

$$\Delta S = S_1 - S_2 = l \sin\alpha \approx l\alpha , \tag{3}$$

and

$$F_t = m_2 g\alpha = m_2 a_2 , \tag{4}$$

where a_2 is the linear acceleration of the cargo, we obtain from (2) that:

$$a_2 = \frac{d^2 S_2}{dt^2} = g\alpha . \tag{5}$$

From the equation (3) we derive that

$$S_1 = \Delta S + S_2 = l\alpha + S_2 .$$

Then the linear speed V_1 of the trolley equals:

$$V_1 = \frac{dS_1}{dt} = l\frac{d\alpha}{dt} + \frac{dS_2}{dt} , \tag{6}$$

and the linear acceleration a_1 of the trolley is

$$a_1 = \frac{d^2 S_1}{dt^2} = l\frac{d^2\alpha}{dt^2} + \frac{d^2 S_2}{dt^2} , \tag{7}$$

which, based on (5), can be written as:

$$a_1 = \frac{d^2 S_1}{dt^2} = l\frac{d^2\alpha}{dt^2} + g\alpha . \tag{8}$$

Then the equation (1) will have the form:

$$F = F_t + m_1 \frac{d^2 S_1}{dt^2} ;$$

$$F = m_2 g\alpha + m_1 l \frac{d^2\alpha}{dt^2} + m_1 g\alpha ,$$

or

$$F = m_1 l \frac{d^2\alpha}{dt^2} + (m_1 + m_2) g\alpha . \tag{9}$$

The last equation relates the tractive force (i.e. the torque on the motor shaft) with the rope's deviation angle α and the second derivative of α using the ratio of the trolley weight to the cargo weight and the rope length.

Therefore, by setting up a certain law of the angle variation at the acceleration (braking) stage we obtain a certain character of the tractive force F for ensuring this required law of variation of the angle α.

The control aims at forming such a signal on the input of the VFD torque setting, which would ensure the absence of the cargo oscillations at the end of the change of the cargo speed from the starting value to the set-up one.

Apparently, the angle variation law is to be characterized by a differentiable continuous function with zero initial and final conditions, including its first derivative. Such requirements are satisfied by a cosine wave shifted by the value of its own amplitude:

$$\left.\begin{aligned}
&\omega_\alpha = 2\pi f\alpha = 2\pi\frac{1}{T\alpha} = \frac{2\pi}{t_{acc}} \\
&T\alpha = t_{acc} = t_{dcc} \\
&\alpha(t) = \alpha_m\left(1 - \cos\omega_\alpha t\right) = \alpha_m\left(1 - \sin\left(\frac{2\pi}{T\alpha}t + \frac{\pi}{2}\right)\right) \\
&\alpha(t) = \alpha_m\left(1 - \cos\frac{2}{3}\pi t\right) = \alpha_m\left(1 - \sin\left(\frac{2}{3}\pi t + \frac{\pi}{2}\right)\right)
\end{aligned}\right\} \tag{10}$$

where α_m is half the angle allowable at rope vertical deviation; t_{acc} is the time allotted for acceleration (braking, t_{dcc}) of the cargo.

As a reminder, the maximum deviation of the cargo should not exceed 4°. The real acceleration/braking time in practice is about 3 sec, which is why we assume t_{acc} = 3 s. The graph of the required deviation angle α is presented in Fig. 3.

3. Mathematical modelling

In order to verify the proposals made, a simulation of the behaviour of the two-mass trolley-cargo system was implemented in the MATLAB environment for the case of direct control of the motor torque for the characteristic values of $l(10,30,50\text{m})$ and $m_2(10,30,50\text{t})$ and constant value $m_1 = 10\text{t}$, which corresponds to the real conditions of the crane operation. For selecting these values and calculating the respective coefficients, the real data of the container crane Liebherr P-179 L Super were used.

In each simulation problem, the start of the motor and acceleration of the gear from zero to the speed $V_1 = V_2 = 1\ \text{m/s}$ was implemented for the time $T_\alpha = t_{acc} = 3\text{s}$ on condition that the character of the cargo deviation angle corresponds to the expressions (10).

For each combination of l and m_2 we obtained functions vs. time (on the acceleration interval) for the following characteristic values:

- stator current (root-mean-square value in phase), A;
- rotor speed, RPM;
- electromagnetic torque, Nm, and the task to form the torque (Electromagnetic Torque 2);
- deviation angle α and its derivatives $\ddot{\alpha}$, $\dot{\alpha}$;
- linear speed of the trolley V_1 and the cargo V_2, m/s;
- distances travelled by the trolley, S_1, and the cargo, S_2, m;
- the difference in the distances travelled by the trolley and the cargo $S_1 = \Delta S + S_2$.

The obtained results for the characteristic ratios between the cargo weight 30 t and the rope length 50 m are shown in Fig. 4.

The presented simulation results fully confirm the feasibility of the controlled and predictable anti-sway method in the two-mass trolley-cargo system at driving motor torque control.

It should be noted that:

1) the motor torque is none the less sign-reversing but it is applied not abruptly, as it is proposed in most conventional methods, but it changes smoothly according to the task;

2) the deviation angle does not depend on the ratio between m_1, m_2, l, it is set up according to (10), and is controlled by the drive;

3) at the end of the acceleration period, the speeds of the trolley and the cargo become equal at the set-up value 1 m/s. Thereat, the trolley speed reverses the sign in the middle of the acceleration period;

4) the distance travelled by the trolley and the cargo by the end of the acceleration is expectedly 1.5 m, as it is set up using (10); this distance is also independent of the ratio between m_1, m_2, l.

Therefore, control using the proposed method at the stage of varying speed of the two-mass system ensures an accurate set deviation of the cargo in the middle of the stage and zero deviation at its end. Thereat, the distance travelled by the cargo is expected and predictable, which automatically solves the problem of the cargo positioning.

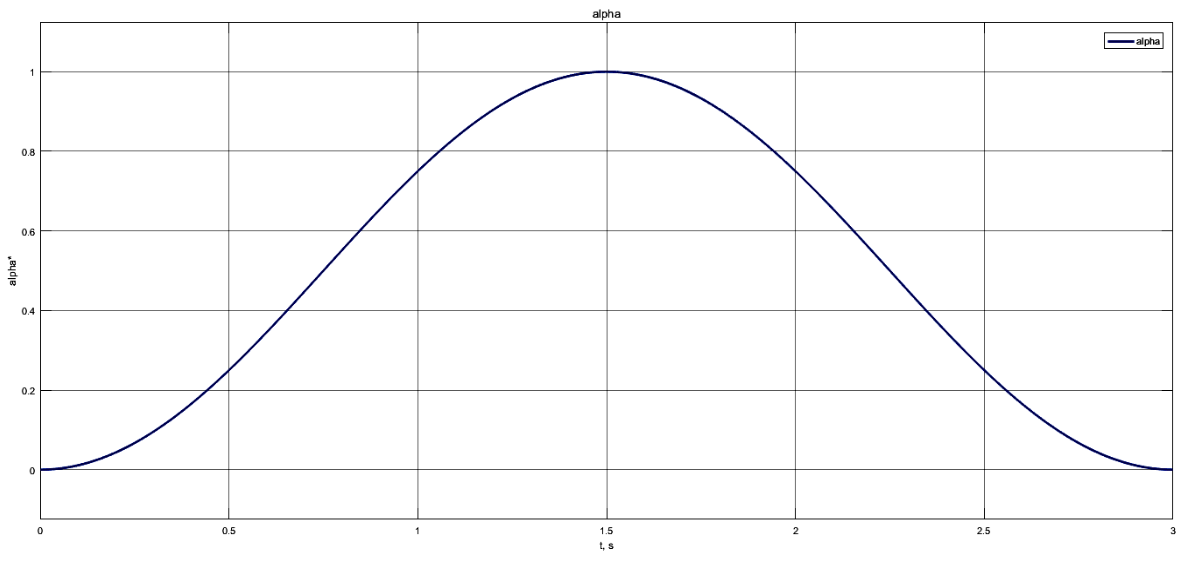
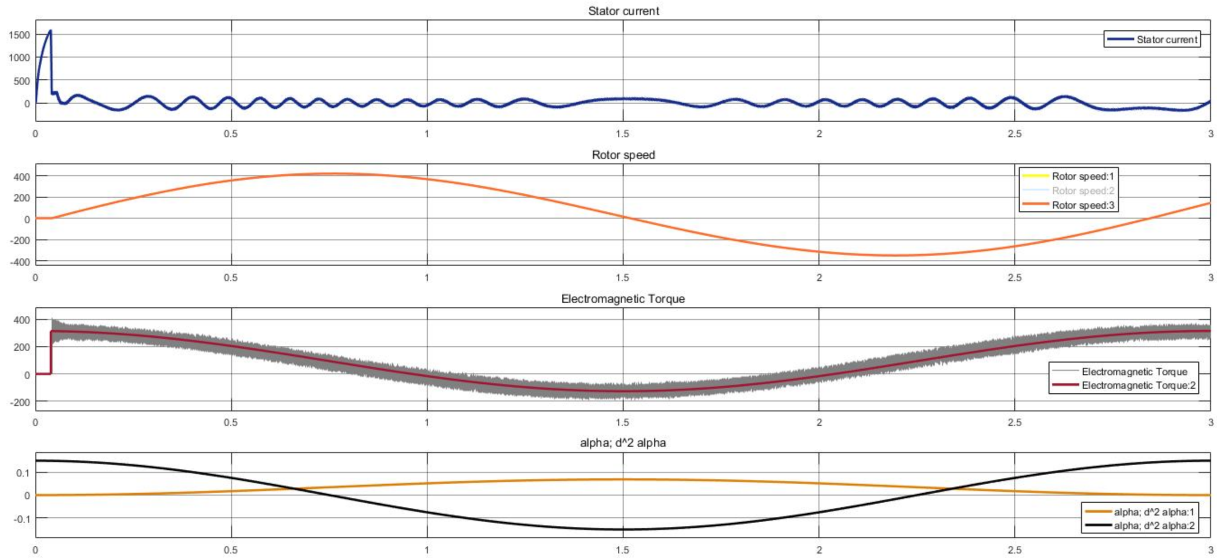
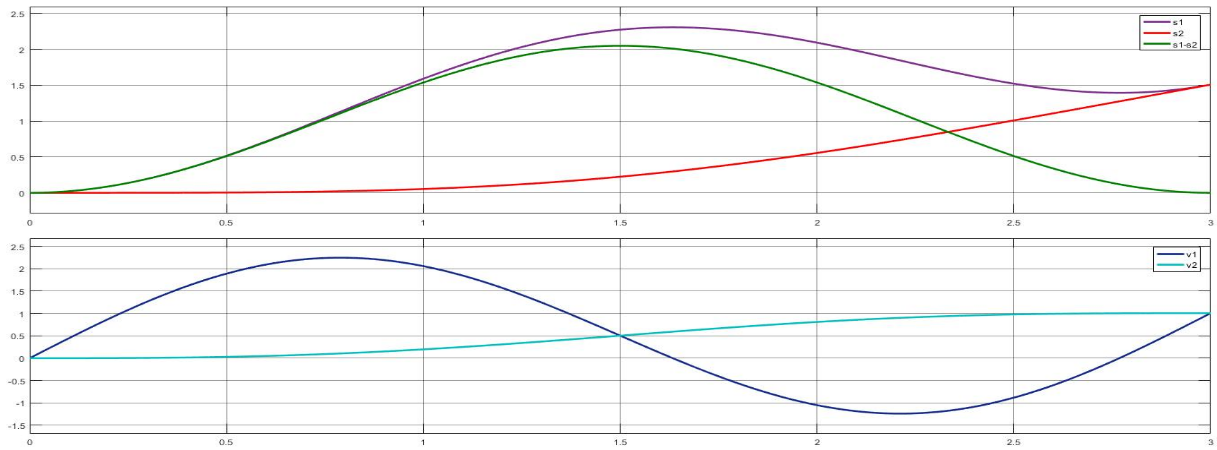


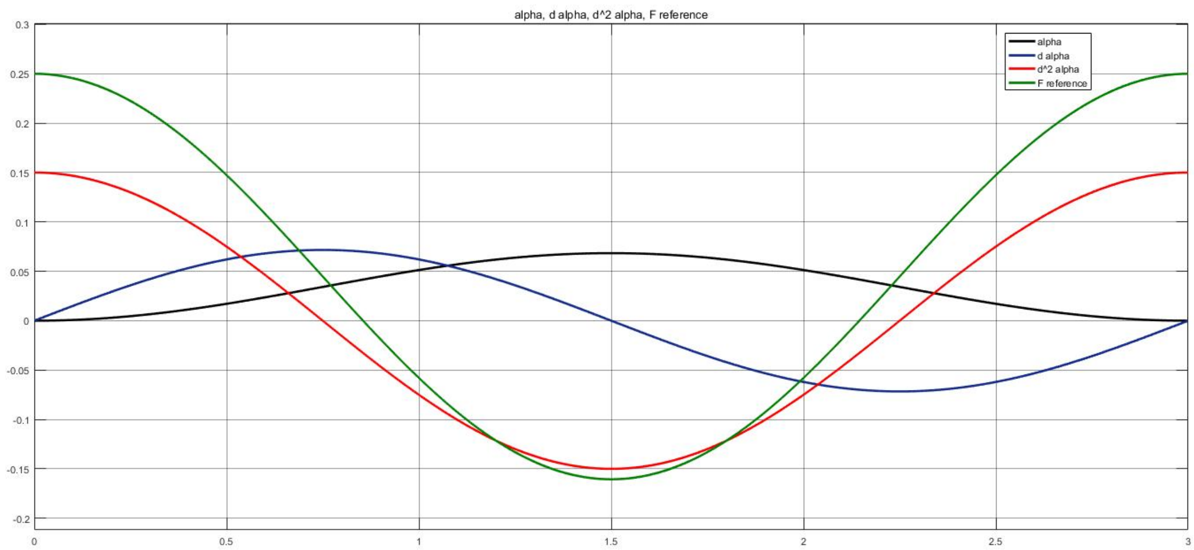
Fig. 3 Rope deviation angle



a



b



c

Fig.4. Simulation results for 30 t – 50 m:
a – stator current, torque and speed of the motor; b – speed and distances of the trolley and the cargo;
c – deviation angle, its derivatives and current set-up task

Conclusions

1. The anti-sway method proposed in the paper by means of direct torque control makes the considered system to some extent invariant in relation to the rope length l and cargo weight m_2: the acceleration time (braking time), deviation angle and distance travelled by the trolley and the cargo at the stage of varying speed are set up by the equations (10). The method allows for a simple solution of the positioning problems at braking, as the distance that the cargo travels at this stage is predictable: at a distance 1,5 m from a required point (at the speed of 1 m/s) a braking command is sent, and in 3 s without oscillations the cargo stops in a needed point.

2. An important feature of the method is the absence of the pulse signals of the sign-reversing torque on the shaft. As seen from the figures, the torque reverses sign but this occurs smoothly and under a complete control in a strict correspondence to the task.

3. The figures show that the acceleration with damping the oscillations is a fully controllable process, due to which the trolley speed and motor torque comply with the established technological and engineering constraints.

4. The character of the force (motor torque) depends on the ratio between the weights of the trolley and the cargo and the rope length. These variables are included in (9) as coefficients: $m_1 = const$, and m_2 and l are determined at the beginning and at the end of the operation. In modern loading works, the weight of each specific container is a value known in advance and is included in the operator's task for processing certain containers in the order planned by the logisticians. The data on the effective rope length can be obtained from the encoder of the lifting device.

5. The method is simple to implement. The cosine wave generator and the force evaluator can be included in the algorithms of a particular controlling PLC (VFD Delta C2000 comprises a modern powerful PLC) along with the algorithm for switching VFD modes from the torque regulation at the acceleration-braking stage to the automatic speed stabilization at the stage of moving the cargo at a constant speed.

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ПРОПОЗИЦІЇ ЩОДО МЕТОДУ ГАСІННЯ КОЛИВАНЬ ВАНТАЖУ ШЛЯХОМ УПРАВЛІННЯ МОМЕНТОМ ДВИГУНА

Анатолій Шестака, Любов Мельнікова

У роботі розглянуто метод гасіння коливань вантажу шляхом управління моментом приводного асинхронного двигуна з використанням сучасних частотно-керованих електроприводів. Детально розглянуто процес коливань у двомасовій системі “візок-вантаж” механізму пересування вздовж консолі причального контейнерного перевантажувача. Записані основні рівняння, що пов’язують тягове зусилля і кут відхилення вантажу від вертикалі, на основі яких формується закон управління моментом приводного двигуна за умови нульового відхилення вантажу від вертикалі наприкінці етапів розгону і гальмування. Сформульовані базові принципи інженерного підходу до вирішення задачі гасіння коливань доступними сьогодні день технічними можливостями частотно-керованого електроприводу, а також до розробки основних алгоритмів керування VFD в цьому процесі.

У статті наведено результати моделювання поведінки в режимах розгону-гальмування двомасової системи “візок-вантаж” для характерних співвідношень маси вантажу та довжини канату, які підтверджують можливість гасіння коливань вантажу шляхом безпосереднього управління моментом приводного двигуна. Модель базується на реальних технічних даних контейнерного перевантажувача Liebherr P-179 L Super.

Важливою особливістю методу є відсутність імпульсів знакозмінного моменту на валу: момент хоч і змінює знак, але цей процес відбувається плавно і повністю керовано, в точній відповідності до завдання, а процес розгону з гасінням коливань є повністю керованим, наслідком чого є відповідність швидкості візка та моменту двигуна до закладених технологічних і технічних обмежень.

Метод є простим у технічній реалізації: генератор косинусоїди і обчислювач зусилля можуть бути закладені в алгоритми окремого керуючого PLC разом з алгоритмом перемикання режимів VFD з управління моментом на стадії розгону-гальмування на автоматичну стабілізацію швидкості на стадії пересування вантажу із сталою швидкістю.

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