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Ensuring of the industrial robot link motion accuracy parameters in the low-speed zone[*]

N. F. Karnaukhov[1], D. A. Smyatsky[2], M. N. Filimonov[3], K. I. Rudnev[4]**
[1, 2, 3, 4]Don State Technical University, Rostov-on-Don, Russian Federation

Обеспечение точностных параметров движения звена промышленного робота в зоне малой скорости[***]

Н. Ф. Карнаухов[1], Д. А. Смяцкий[2], М. Н. Филимонов[3], К. И. Руднев[4]**
[1, 2, 3, 4]Донской государственный технический университет, Ростов-на-Дону, Российская Федерация

Introduction. The article is devoted to the search for a method of reducing the effect of friction links mobility industrial robot PR (production mechanism PM) with frequency-controlled electric drive (FCED) on the static error (accuracy) positioning of the working body (WB) when moving in a low speed zone. The random character of friction forces changes in the implementation process start-braking modes of induction motor (IM) operation create difficulties in performance specified technological process parameters. The formation of pulsating moments on the IM shaft, due to the stator IM current harmonics, combined with the friction torque of the moving parts in the IR (IM) guide, accompanied by a deterioration of the WB (IM output link) movement dynamics.
Materials and Methods. To correction the motion of the WB (PM output link) in the IM start-brake modes, the authors proposed to use dual-mode control of the Autonomous voltage inverter (AVI), providing software control of the IM stator current harmonics amplitudes and the corresponding regulation of the pulsating moments on the IM shaft by introducing a multiple "m — submodulation" of the AVI carrier frequency (CF).
Results. The simplified representation of the FCED block diagram with local and global negative feedbacks (GNF) allowed to reveal the features of specific parameters influence on the static error δ and the FCED dynamic stability with the limiting gain of the corrective amplifier K1.
Discussion and Conclusions. The introduction of rotor "micro-vibration", allows the IM "conditionally reduce" the moving link friction force in the guides, reduce the drive power to overcome the friction forces. The use of dual-mode control AVI expands the scope of use of scalar control in process equipment, multi-link mechanisms of automated production, operating in the zone of "low and creeping speeds".

Введение. Рассматривается трение звеньев подвижности промышленного робота (ПР) производственного механизма (ПМ) с частотно-управляемым электроприводом (ЧУЭП). Цель работы — поиск метода снижения влияния такого трения на статическую погрешность (точность) позиционирования рабочего органа (РО) при движении в зоне малой скорости. Случайный характер изменения сил трения при реализации пуско-тормозных режимов работы асинхронного двигателя (АД) создают трудности в достижении заданных параметров технологического процесса. На валу АД формируются пульсирующие моменты, обусловленные гармониками тока статора АД. Это явление в сочетании с моментом трения подвижных звеньев в направляющих ПР (ПМ) сопровождается ухудшением динамики движения РО (выходного звена ПМ).
Материалы и методы. Для корректировки движения РО (выходного звена ПМ) в пуско-тормозных режимах работы АД авторы статьи предлагают использовать двухрежимное управление автономным инвертором напряжения (АИН), обеспечивающее программное управление амплитудами гармоник тока статора АД и соответствующее регулирование пульсирующих моментов на валу АД посредством введения кратной m-подмодуляции несущей частоты (НЧ) АИН.
Результаты исследования. Упрощенное представление структурной схемы ЧУЭП с местной и глобальной отрицательными обратными связями (ООС) позволило выявить особенности влияния конкретных параметров на статическую погрешность δ и динамическую устойчивость работы ЧУЭП с предельным коэффициентом усиления корректирующего усилителя К1.
Обсуждение и заключения. Введение микровибрации ротора АД позволяет условно уменьшать силу трения движущегося звена в направляющих, снижать мощность привода на преодоление сил трения. Применение АИН с двухрежимным управлением расширяет сферу использования ЧУЭП скалярного управления в технологическом оборудовании, многозвенных механизмах автоматизированных производств, работающих в зоне малых и ползучих скоростей.

[*] The research is done within the frame of the independent R&D.
[**] E-mail: Nfk_609@mail.ru, aid219@mail.ru, maxfil2006@yandex.ru, gloom_z@mail.ru
[***] Работа выполнена в рамках инициативной НИР.

Keywords: industrial robot, link mobility, dry friction, frequency asynchronous electric drive, harmonics of the stator current, microvibration of the rotor, the transmission link, the automatic control system.

Ключевые слова: промышленный робот, звено подвижности, сухое трение, частотный асинхронный электропривод, гармоники тока статора, микровибрация ротора, передаточное звено, автоматическая система управления.

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Introduction. When selecting an industrial robot (IR), the most important parameters are its possible technical characteristics for carrying out transport and technological operations with programmed precision in movement of the working body (WB) desired when assembling the joint, processing parts, and implementing special work under the production conditions and in emergency situations. Such a task is usually considered when designing the kinematic chain (KC) of the IR (or other production mechanism of the PM) designed to perform complex movements of the WB in the shortest, most rational way (for example, during processing, assembly, painting of the product in a closed space and other special cases including technogenic catastrophes). Such capabilities can be expected from IR with enhanced manoeuvrability at seven or more degrees of mobility provided by various combinations of links and kinematic pairs [1-3]. At the same time, components of the reaction force vector and moments for coordinate systems rigidly connected with movable links are changed in the KC. Here, it is necessary to consider the impact of friction forces in rotating and sliding kinematic pairs, both of an individual link and of the multilink mechanism as a whole, which determine the total resistance to the movement of the WB of the IR (or the production mechanism of the PM). Since friction forces in kinematic pairs are of random nature, depend on a number of factors and determine the mechanical efficiency, and the dynamics of the movement of the multilink mechanism, the search for a way to reduce the effect of friction on the fundamental losses in the mechanism and to improve the quality of movement of the WB remains in demand. The need for solving this problem is also due to the fact that these factors impede stability in the development of the programmable operating mode of the IR (or of the law of WB motion) by the automatic control system in the low-speed zone. A special impact of the friction forces is manifested in a decrease in the positional accuracy of the WB, both in discrete and continuous IR operation in the form of an error in motion path following of the multilink mechanism. In addition, it is also necessary to consider the components of the angular and linear errors of motion of the links of the mechanism depending on the actual values of the forces (moments) of dry friction in the joints and of random nature. Unpredictability of the friction effect on the programmable result of the the output link of the IR (PM) motion significantly worsens the dynamic processes in the FCD and complicates the task of correcting the nonlinearities of its structural scheme of the automatic control system.

To reduce the static error in the drive operation due to the dry friction, the following is used: local negative feedbacks (NFB), increase in the gain constant of the pre-amplifier (to a critical value of the entire system considering the system order) and other known approaches [4,5]. A real change in the forces (moments) of friction in the joints of the multilink IR mechanism determines the search for the best decision to reduce the effect of dry friction on the most important indicators of the WB motion, to increase the energy efficiency of electric energy conversion in the frequency drive system during the process.

The key objective of the paper, according to the authors, is to familiarize a reader with the possibilities of using controlled “micro-vibration” in the joints of the rubbing parts of the IR under special control of an autonomous voltage inverter in a frequency electric drive based on a three-phase squirrel-cage induction motor (IM). The possibility of programmatic influence on the level of the oscillatory IM shaft torque (under the introduction of the “*m*–submodulation” mode of the carrier frequency of the AVI) provides changing $M_{ТР}$ value, improving the static and dynamic indices of the FCD.

Materials and Methods. The authors of the paper see a solution to the problem in the method of using oscillatory rotor shaft torques formed by the stator current harmonics in the FCD to introduce controlled “micro-vibration” of the rubbing movable links of the IR mechanism under consideration. At that, in the low-speed zones, the breakaway torque of the movable IR link can be programmatically changed to the torque creep, which is due to a decrease in the effective friction coefficient $K_{эф}$ to the minimum level [5, 6]. Such a state between the rubbing surfaces of the mechanism links can be created through controlling the amplitude of oscillatory IM rotor shaft torque and changing the effective coefficient of friction $K_{ЭФ}$ under the bimodal control of the AVI [7,8].

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The above guidelines are considered in the paper in relation to the structural diagram of the multilink IR mechanism (in any coordinate system), where it is possible to distinguish a kinematic pair of horizontal displacement of the movable output WB link, for example, an extended arm with a gripping device (GD). The link is moved by an individual FCD running on the "AVI-IM" system with bimodal control of the AVI [9]. Validation of the possibility of "changing" the friction force in the joint of the links (guides) can be obtained through analysing the design FCD model with the output mobile link IR (or PM) WB by indicating the forces acting on it (Fig. 1). The following notation is introduced in the model: $M_Д$ is engine torque; $\Omega_Д$ is rotational speed of the IM rotor; c is stiffness of the connection PM with IM; $M_{ЗП}$ is moving torque at the link input (resistance torque "minus" $M_{ЗП}$) at the IM shaft; φ_1 is the angle of rotation of the PM (IR) input link; $\varphi_{ЗП}$ is output IM shaft angle; $F(t)$ is total (external) driving force; F_{TC} is friction force in the guides; F_T is process resistance force; N is normal pressure force of WB with mass m, Φ is vibrational component of the total driving force $F(t)$.

The equation of the IR (PM) link motion along the longitudinal axis x can be written in the following form:

$$m\ddot{x} = F(t) - F_T - F_{TC}\,. \qquad (1)$$

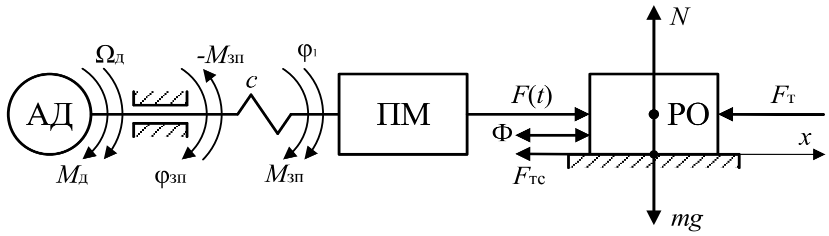
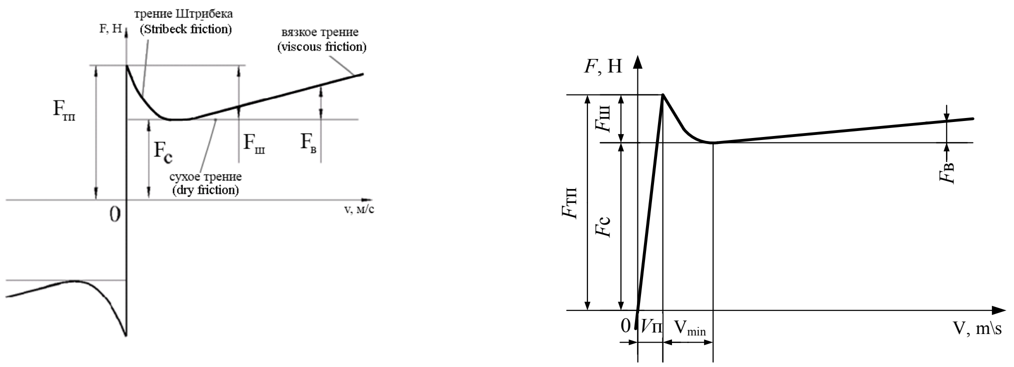


Fig. 1. Calculation model of the production mechanism mobility link

The friction force F_{TC} in the expression (1) can be represented by the components:

$$F_{TC} = F_{TП} = F_{Ш} + F_C\,, \qquad (2)$$

where $F_{TП}$ is static friction force, $F_{Ш}$ is Stribeck friction force [10], F_C is Stribeck friction force. Graphical dependences corresponding to these concepts are shown in Fig. 2.



a) *b*)

Fig. 2. Friction characteristics: *a*) with a break near zero speed of the link in the guides, *b*) without a break near zero speed

Considering the expression (2), when the static friction force $F_{TП}$ under the operating conditions of the IR can vary over a wide range, i.e. $F_{TП}/F_C \approx (0.1 \ldots 2.5)$, the required value of the external (total driving) force $F(t)$ (for displacement of WB) should correspond to the relation $F(t) > F_{TП} + F_T$ [10]. In reality, the nature of the static friction force $F_{TП}$ dependence on the speed (V) of the mobility link displacement (in the low-speed zone) is different, and there is a good reason to consider it as an example of a continuous function (Fig. 2, *b*). Here, the friction force $F_{TП}$ law completely reflects physical processes in the mechanical contact distributed in the plane between the bodies – movable links of the IR in a small area of speed where $V \to 0$. The Stribeck friction characteristic $F_{Ш}$ has a negative slope in the low-speed zone ($V_{МИН} - V_П$) relative to the displacement of rubbing bodies. The sum of the forces $F_{Ш} + F_C$ at the boundary of the low-speed range ($0 - V_П$) (or near linear zero speed) forms the static friction force $F_{TП}$ ("stall friction force" [11-13]. To disturb the state of relative rest of a solid body (in the motion guiding connections), "vibrations" of the body are often used. They are provided by an external force $F(t) > F_{TП}$ due to oscillatory torques from the IM shaft harmonics [14-17]. If the bimodal control of the AVI in the simplified block diagram of the FCD (Fig. 3) with speed feedback is used, then the controlled "micro-vibration" of the IM rotor can be programmatically specified, and the longitudinal (transverse) pulsating force Φ (or vibrational component) (Fig. 1) of the total driving force $F(t)$ can be generated.

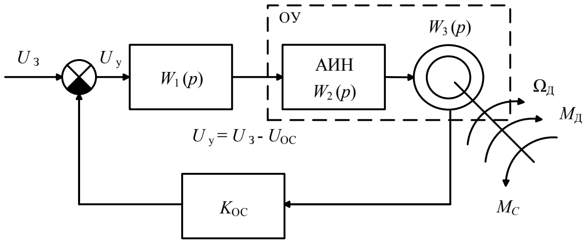


Fig. 3. Simplified block diagram of FCD movable PM link

In Fig. 3, the following notation is used: $U_З$ is the task control signal for the output coordinate; $W_1(p)$, $W_2(p)$, $W_3(p)$ are transfer functions of the compensating amplifying link (device), autonomous inverter AVI, asynchronous motor IM, respectively; $K_{ОС}$ is the gear ratio of the speed feedback link; $U_{ОС}$ is the speed feedback signal; $Ω_Д$ is the angular IM rotor speed; $M_Д$, $M_С$ are moments of propulsive forces and resistance forces on the IM shaft, respectively, including the friction torque $M_{ТР}$.

Research Results. It is known that closed-loop automatic control systems (ACS) covered by the feedback (FB) are prone to unstable operation [18], which requires a careful analysis of the parameters of each element of the control loop when searching for the motion control law of the WB as a transmission link including mechanical movable links considering various components of the friction torque $M_{ТР}$ (from the rest force $F_{ТП}$ and dry friction force $F_С$) [19]. In the multilink mechanisms with an individual drive, $M_{ТР}$ can vary over a wide range, which generally complicates the system. It is possible to improve the quality of the transient process in the drive through introducing a local NFB from the inertial link output (IM) to its input (Fig. 4) by switching on the link with the transfer function $W_0(p)$ [4]. In the global NFB, the signal is fed by speed to the input of the compensating amplifier with the transfer function $W_1(p)$ through the comparator (node $Σ_1$).

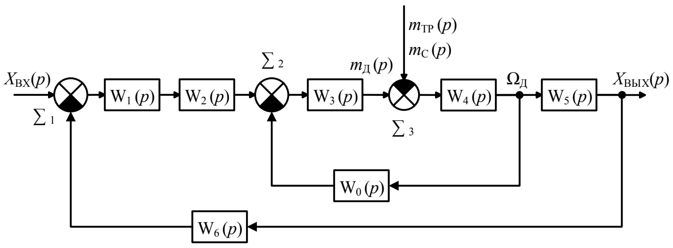


Fig. 4. FCD transformed block diagram of PM mobility link

Having selected a straight section of the control system circuit with IM covered by the link $W_0(p)$ of the local FB in the block diagram, we will replace it with an equivalent link with the transfer function [4.19] of the form:

$$W_Э(p) = \frac{W_4(p)\cdot m_{ТР}(p)}{1+W'(p)} = \frac{W_4(p)\cdot m_{ТР}(p)}{1+W_3(p)\cdot W_4(p)\cdot W_0(p)},$$

where $W'(p)=W_3(p)\cdot W_4(p)\cdot W_0(p)$ is the open-loop transfer function of the inner circuit.

With the simultaneous action of all disturbances (according to the input and load control when $m_{ТР}$ changes), the resulting value $W_{ВЫХ}(p)$ in the linear control system can be represented (by the superposition method) as the sum of individual reactions from external actions separately. If we accept the correspondence of the input signal $X_{ВХ}(p) \rightarrow f_1(p)$ as the specification of the frequency of the IM stator current, and $X_{ВЫХ}(p) \rightarrow n_Д(p)$ is the output frequency of the mechanism shaft speed, then the resulting operator expression with respect to the output quantity can be written in the form

$$X_{ВЫХ}(p) = \frac{W_1(p)\cdot W_2(p)\cdot W_Э(p)\cdot W_5(p)\cdot W_6(p)}{1+W(p)}\cdot f_1(p) + \frac{W_4(p)\cdot m_{ТР}(p)}{1+W_3(p)\cdot W_4(p)\cdot W_0(p)},$$

where $W(p)=W_1(p)\cdot W_2(p)\cdot W_Э(p)\cdot W_5(p)\cdot W_6(p)$ is the open-loop transfer function of the outer duct.

While investigating the control block diagram with the given parameters of the power circuit elements, simplified relations can be used for the transmission links [20, 21]:

– AVI transfer function

$$W_2(p) = \frac{k_{ПР}}{T_{ПР}\cdot p+1},$$

where $k_{ПР}$ is the transmission coefficient of the frequency converter, $k_{ПР} = \frac{U_{1ф}}{f1}$; $T_{ПР} = \frac{1}{f_К}$ is the time constant of the transistor frequency converter (AVI); $f_К$ is the value of the modulation carrier frequency (3-50 kHz); $U_{1ф}$, f_1 are the voltage and frequency of the IM current phase, respectively;

– IM transfer function by control action

$$W_3(p)=\frac{n_Д}{f_1(p)}=\frac{k_Д}{T_{МЭ}T_{ЭМ}p^2+T_{МЭ}p+1},$$

where the parameter f_1(p) is determined from the ratio $f_1=\frac{f_{1н}}{D}$ Hz for the calculated range of D – regulation of the IM, $f_{1Н}$ is the nominal voltage frequency of the IM; $k_Д$ is the transmission coefficient of the IM; $k_Д$ = 60/p , where p is the number of pairs of the IM poles;

– transfer function of an induction motor in a disturbing action from a static load moment $m_С$

$$W_3(p)=-\frac{n_Д(p)}{m_С(p)}=-\frac{\frac{60}{2\pi}\cdot\frac{1}{k_{ЖЭ}}(T_{эм}p+1)}{T_{МЭ}T_{ЭМ}p^2+T_{МЭ}p+1},$$

where

$$k_{ЖЭ}=\frac{3U_{1\phi}^2p^2\cdot\frac{R_2^{'}}{S_2}}{4f_1^2\pi^2[(R_1+R_{1доб}+\frac{R_2^{'}}{S_2})+X_к^2]},$$

where R'_2 is the reduced real resistance of the rotor to the stator, S is the design slip, $X_К$ is the total inductive resistance of the stator and rotor, R_1 is the nominal real resistance of the IM stator winding phase, $R_{1доб}$ is the design real resistance of the IM stator winding in the voltage-frequency correction mode characteristics [18,19]; $T_{МЭ}=\frac{4\pi^2f_1^2J_DR'_2}{3U_{1ф}^2p^2}$ is the electromechanical time constant of the IM, J_D is the reduced moment of inertia of the drive; $T_{ЭМ}=\frac{X_к}{2\pi f_1R'_2}$

– electromagnetic time constant of the IM.

The transfer function of the IM under the impact of the (external) frictional torque $M_{ТР}$ can be represented by the conformance of the generation of the electromagnetic torque of the IM to the rotor current (at the constant stator magnetic flux ($\Phi_С$ = const) as the transfer coefficient $k_{МЭ}$ calculated from the relation:

$$W_4(p)=k_{мэ}=\frac{3p\cdot k_{пр}}{2\pi}.$$

Given the reduced moment of load inertia, the developed dynamic moment $M_{ДИН}$ on the IM shaft in the transition mode of the FCD operation can be represented by the relation:

$$J_D\cdot\frac{\partial\Omega_д}{\partial t}=M_д-(M_с+M_{тр}), \tag{3}$$

where $M_Д$ is the IM electromagnetic torque. In the node Σ3, the right-hand side of the equation (3) is conditionally solved through introducing the AVI control in the “*m*– submodulation” mode, which enables to programmatically change the set frictional torque $M_{ДИН}$ (or the breakaway torque) by the amplitude variation of the pulsating moment on the IM shaft. It should be noted that in the multilink IR mechanisms, the frictional (breakaway) torque can vary by 1.5–2.5 or more times in comparison to the moment of dry friction. The conversion of the shaft speed measurement units (from rad/s to rpm) is carried out through the transfer function

$$W_5(p)=k_{ск}=\frac{n_0}{\Omega_0},$$

where n_0, Ω_0 is the IM shaft rotation frequency, rpm, rad/s, respectively.

Link 6 of the structural diagram is an instantaneous link characterizing the natural feedback on induced counter-EMF of the IM. In this case, the design transmission coefficient $k'_Э$ will be

$$W_6(p)=k'_Э,$$

where

$$k'_Э=\frac{f_1}{\frac{60f_1}{p}}=\frac{p}{60}.$$

The transformations performed provide the overall gain of the FCD system $k_У$ through the corresponding coefficients of individual links:

$$k_У = k_1 \cdot k_{ПР} \cdot k_Д \cdot k_{МЭ} \cdot k_{СК} \cdot k'_Э.$$

It is known that the static error δ caused by the dry friction moment $M_{ДИН}$ in closed systems can be reduced by introducing local parallel and sequential high-speed feedbacks into the system which can be seen from the relation [4,5,20]

$$\delta = \frac{M_{тр} k_{мэ}}{k_1 k'_Э},$$

where k_1 is the amplifier gain of the compensating link (for example, when the PID controller is turned on); $k'_Э$ is the speed sensor gain. However, an increase in k_1 greater than a critical value is accompanied by an increase in $k_У$, and at $k_У \geq k_{КР}$, a critical value in the drive, self-oscillations, loss of stability can occur.

In this case, $k_{КР}$ is in (1+ $k_У$) times higher than in a system without feedback, which reduces the static error of the system δ additionally with a decrease in the moment $M_{ДИН}$ when introducing a controlled micro-vibration of the IM shaft in the "*m*–submodulation" mode of the AVI operation.

Discussion and Conclusions. The studies performed allow us to conclude:

1. The application of bimodal control of an autonomous voltage inverter in the FCD provides the improvement of an industrial robot (or RM) in the low-speed zone of the working body.
2. The use of program-controlled "micro-vibration" of the FCD IM rotor provides a predetermined decrease in the frictional ("breakaway") torque in the multilink movable mechanisms, reduces the level of static error of the system by 1.5-2.5 times at an underestimated gain of the compensating amplifier, and increases system stability under the dynamic control modes.

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Authors:

Karnaukhov, Nikolay F.,
professor of the Robotics and Mechatronics Department, Don State Technical University (1, Gagarin sq., Rostov-on-Don, 344000, RF), Cand.Sci. (Eng.), associate professor,
ORCID: https://orcid.org/0000-0003-0625-2973
Nfk_609@mail.ru

Filimonov, Maxim N.,
senior lecturer of the Robotics and Mechatronics Department, Deputy Dean of the Automation, Mechatronics and Control Faculty, Don State Technical University (1, Gagarin sq., Rostov-on-Don, 344000, RF), Cand.Sci. (Eng.),
ORCID: https://orcid.org/0000-0003-2807-9171
maxfil2006@yandex.ru

Smyatsky, Dmitry A.,
postgraduate student, engineer of the Robotics and Mechatronics Department, Don State Technical University (1, Gagarin sq., Rostov-on-Don, 344000, RF),
ORCID: https://orcid.org/0000-0002-7772-1039
aid219@mail.ru

Rudnev, Konstantin I.,
third-year student of the Robotics and Mechatronics Department, Don State Technical University, (1, Gagarin sq., Rostov-on-Don, 344000, RF),
ORCID: http://orcid.org/0000-0002-8485-3873
gloom_z@mail.ru