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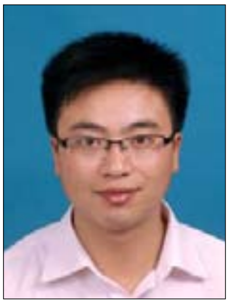

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ANALYSIS OF THE INFLUENCE OF POWER FACTOR ANGLE PARAMETERS ON RESONANCE IN PURE ELECTRIC TRACTORS

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Abstract. One of the major causes of damage to the transmission system is torsional vibration. It leads to the fatigue damage of the transmission parts, reducing their service life. In response to this problem, the torsional vibration model of the transmission system of the electromechanical coupling of the pure electric tractor is established, and the influence of internal power factor angle on the natural frequency of the transmission system under electromechanical coupling is analyzed. The results show that when the power train is designed, if the internal power factor angle is increased from 0 to π/2, the change of the electromagnetic parameters leads to the resonance of the transmission system and the increase of the unstable area. The power train may also have saddle components, fork, jump and other phenomena, resulting in fatigue damage to the transmission components.

Keywords: pure electric tractor; power transmission system; internal power factor angle; resonance frequency.

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Problem statement. China is a big agricultural country, and the new type of agricultural equipment is an important part of the national manufacturing industry. Tractors as the main power plant of agricultural machinery have developed rapidly in recent years. Taking into account the development of new energy vehicles and the state's strong support for this venture, pure electric tractors will inevitably replace the traditional fuel tractor. Thus, the problem of vibration in pure electric tractors is bound to be an important research topic. Vibration will affect the service life of the tractor's structural components, causing fatigue damage and thus leading to accidents. Vibration and resonance are major factors in the electromagnetic effect as well [1]. Therefore, it is of great practical significance and applied value to study the torsional vibration of the system under the coupled electromagnetic excitation of the pure electric tractor.

At present, the study on torsional vibration of automobile power trains is conducted both in China and abroad. Regarding the transmission system of pure electric tractors, it is still in its initial stage. Aimed at the problem of vibration and noise in hybrid vehicle drive systems [2], Wang Kai et al. [3] established a torsional vibration mechanical model of compound planetary gear train and complete vehicle transmission system. It was found that the transmission system resonates at a certain speed ratio; the effect increases with the increase of the torsional vibration damper stiffness. Based on the Lagrange equation, the torsional vibration model of the vehicle power train is established to study the similarity of natural characteristics under different operating conditions. The distribution of natural frequency and the influence of physical parameters on natural frequency are obtained. Then, there has been developed a transmission mass concentration equivalent model, a transmission system for forced vibration simulation, a resonance speed point [5].

With regard to the torsional vibration problem of the transmission system of pure electric tractors, this paper derives the electromagnetic torque expression from the energy point of view based on the principle of electromechanics. With the use of the kinetic theory of electromechanical analysis, we develop the torsional vibration model of the electromechanical coupling of the pure electric tractor transmission system and establish the influence of the internal power factor angle on the natural frequency of transmission system under the action of electromechanical coupling.

1. Electromechanical coupled torsional vibration model

1.1. Development of the motor model

According to the theory of motor double reaction, the permanent magnet synchronous motor phasor diagram can be drawn when the motor is running steadily. The diagram is shown in Fig. 1.

In the diagram, E_0 represents the electromotive force generated by the air gap fundamental magnetic field, which is called the no-load back electromotive force (*V*).

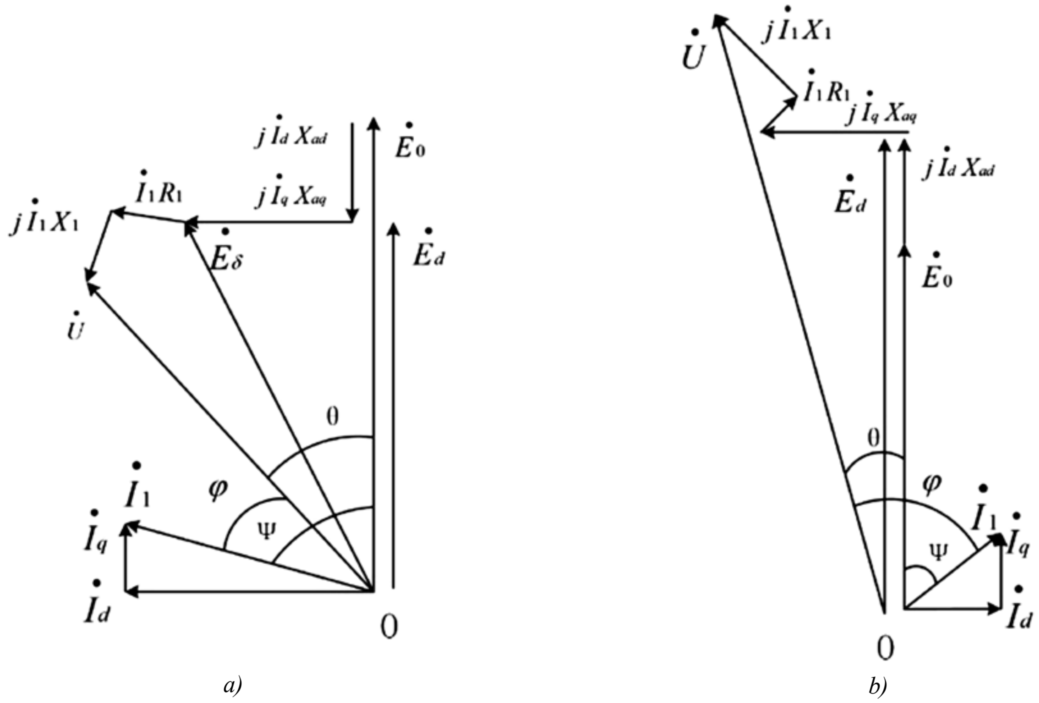

Fig. 1. Permanent magnet synchronous motor phasor diagram
a — Ψ > 0, *b* — Ψ < 0

E_d is the electromotive force generated by the direct-axis component of the fundamental magnetic field of air-gap synthesis and is called the in-axis electromotive force (V). E_δ is the electromotive force generated by the air-gap synthetic fundamental magnetic field, which is called the air-gap combined electromotive force (V). Further on, θ is the power angle, φ is the power factor angle, Ψ is the internal power factor angle. In Fig. 1, *a*, current $\dot{I}_1$ ahead of the no-load back-EMF $\dot{E}_0$, $\Psi > 0$; straight-axis armature reaction is demagnetization. The EMF E_d within the direct axis of the motor is less than the no-load back-EMF E_0. In Fig. 1, *b*, current $\dot{I}_1$ is lagging behind the no-load back-EMF, $\Psi < 0$. Axial armature reactions are all of magnetic properties, resulting in the motor shaft EMF E_d being greater than the no-load back-EMF E_0.

In our study, ω is the electrical angular velocity of the motor; Ω is the mechanical angular velocity, $\omega = p\Omega$; m is the motor phase number; p is the pole pair number of the motor; E_0 is the no-load back-EMF. As evidenced by Fig. 1, the current stator of the straight cross-axis shafts can be expressed with the help of the following relationship:

$$\begin{cases} I_d = \dfrac{R_1 U\sin\theta + X_q\ \ E_0 - U\cos\theta}{R_1^2 + X_d X_q}, \\ I_q = \dfrac{X_d U\sin\theta - R_1\ \ E_0 - U\cos\theta}{R_1^2 + X_d X_q}. \end{cases} \qquad (1)$$

The stator phase current, input power, electromagnetic torque and other basic parameters of the motor can be derived as follows from formula 2:

$$\begin{cases} E_0 = 4.44 f K_{dp} N \Phi_{10}, \\ \Phi_{10} = K_\Phi \dfrac{b_{m0} B_r A_m}{\sigma_0} \times 10^{-4} \\ K_\Phi = \dfrac{8}{\alpha_i \pi^2} \sin\dfrac{\alpha_i \pi}{2}, \\ \alpha_i = \alpha_p + \dfrac{4}{\frac{\pi D}{2p\delta} + \frac{6}{1-\alpha_p}}. \end{cases} \qquad (2)$$

where N is the number of turns per phase in series; Φ_{10} is the fundamental flux of the permanent magnet; A_m is the area of the permanent magnet; f is the power frequency; K_{dp} is the fundamental winding coefficient; B_r is the permanent magnet remanence ;σ_0 is the no-load magnetic-leakage coefficient; b_{m0} is the permanent magnet no-load operating point assumed value; K_Φ is the air gap flux waveform coefficient; α_i is the pole arc coefficient; α_p is the arc coefficient; δ is the air gap length; D is the stator diameter.

Thinking that $b_1 = \frac{mpE_0U}{\omega X_d}$, $b_2 = \frac{mpU^2}{2\omega}\left(\frac{1}{X_d} - \frac{1}{X_q}\right)$, it follows that

$$T_e = b_1 \sin\theta + b_2 \sin\theta, \qquad (3)$$

As can be seen from the vector diagram 1, $\theta = \varphi + \psi$, the $\sin\theta$ and $\sin 2\theta$ trigonometric functions are expanded. Using the approximation of the Taylor expansion, the trigonometric functions containing φ are expanded and simplified by using $\sin x \approx x^{-1}/6x^3$, $\cos x \approx 1^{-1}/2x^2$. It enables calculating the electromagnetic torque acting on the rotor:

$$T_e = k_0 + k_1\varphi + k_2\varphi^2 + k_3\varphi^3, \qquad (4)$$

where $k_0 = b_1\sin\psi + b_2\cos 2\psi$; $k_1 = b_1\cos\psi + 2b_2\sin\psi$; $k_2 = 1/2b_1\sin\psi + 2b_2\sin 2\psi$; $k_3 = 1/6b_1\cos\psi + 4/3b_2\cos 2\psi$.

1.2. Torsional vibration model of the transmission system and solution of the mechanical natural frequency

Let us establish the electromechanical mass transfer model shown in Fig. 2 [6–7]. The torsional vibration characteristics of the electromechanical coupling of the pure electric tractor's drive system are analyzed. In the diagram, T_e is the electromagnetic torque, T_L is the load torque.

According to the Newton's law, the transmission system's torsional vibration kinetic equation is expressed as follows [8]:

$$\begin{cases} J_1\ddot{\varphi}_1 + C\left(\dot{\varphi}_1 - \dot{\varphi}_2\right) + K(\varphi_1 - \varphi_2) = T_e \\ J_2\ddot{\varphi}_2 - C\left(\dot{\varphi}_1 - \dot{\varphi}_2\right) - K(\varphi_1 - \varphi_2) = -T_L \end{cases} \qquad (5)$$

Let us define the following quantities: $v = J_1/J_2$; $r = 1/(1+v)$; $\eta^2 = \omega_2 0 - k_1 r/J_1$; $\mu = (1/J_1 + 1/J_2)C$; $\beta = k_2 r^2/J_1$; $\gamma = k_3 r^3/J_1$. Thus, the simplified equation (5) can be converted to the following form:

$$\ddot{x} + \eta^2 x + \mu\dot{x} + \beta x^2 + \gamma x^3 = f\cos\omega t, \qquad (6)$$

Within the formula, do not consider the role of mechanical and electrical coupling when the mechanical natural frequency ω0 is as follows:

$$\omega_0 = \sqrt{\left(\frac{1}{J_1} + \frac{1}{J_2}\right)K}, \qquad (7)$$

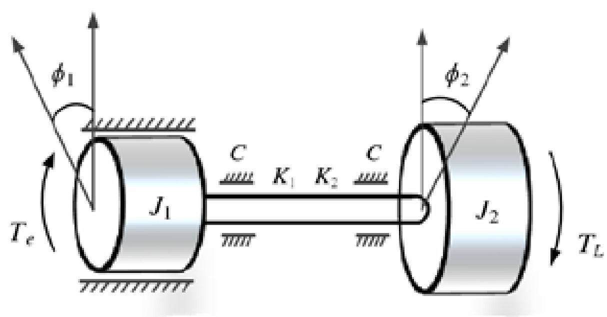

Fig. 2. Electromechanical and mechanical coupling transmission system

1.3. Natural frequency analysis

In this calculation, motor rotor inertia $J_1 = 0.067$ kg·m², mechanical rotor inertia $J_2 = 0.067$ kg·m², damping term μ = 1.03, incentive amplitude $f = 1.5$. According to equation (7), the natural frequency of the torsional vibration system of the transmission system when electromechanically coupled is obtained as follows:

$$\eta = \omega_0 \sqrt{1 - \frac{k_1 r}{J_1 \omega_0^2}}, \quad (8)$$

The permanent magnet synchronous motor is applied a wide range of vehicles and has high requirements on power density, design, and weak magnetic space. Since weak magnetic motor control is used here, the motor's power factor angle is ψ > 0.

It can be seen that when Ψ > 0, the natural frequency is coupled mechanically and the transmission system has as a soft characteristic. Under electromechanical coupling, the natural frequency can drop by 40% under overload conditions. The design and control of the permanent magnet synchronous motor needs to take into account the influence of the internal power factor angle, since it is an important factor affecting the natural frequency of torsional vibration of a transmission system. Therefore, the permanent magnet synchronous motor requires a rational control of the range of internal power factor angles.

2. Main Resonance Analysis of Electromechanical Coupling Torsion

Main resonance refers to the resonance when the external excitation frequency ω is close to the natural frequency η of the derived system. If it is a damping system, a small excitation amplitude f stimulates strong resonance, and there is a step-by-step multi-scale derivation of the following autonomous differential equations [9; 10]:

$$\begin{cases} D_1 a = -\frac{1}{2}\mu a + \frac{f}{2\eta}\sin\varphi \\ aD_1\varphi = \sigma a - \frac{3\gamma}{8\eta}a^3 + \frac{f}{2\eta}\cos\varphi \end{cases}, \quad (9)$$

To determine the amplitude α and phase φ of a stationary solution corresponding to the steady-state motion, let us accept $D_1\alpha = 0$, $D_1\varphi = 0$ for equation (9) to obtain an algebraic equation satisfying the amplitude α and φ:

$$\begin{cases} \frac{1}{2}\mu a = \frac{f}{2\eta}\sin\varphi \\ \sigma a - \frac{3\gamma}{8\eta}a^3 = -\frac{f}{2\eta}\cos\varphi \end{cases}, \quad (10)$$

The sum of two squares is added to eliminate φ. Thus, the equations for the reduced amplitude frequency response and the phase frequency response are simplified, with the following expressions obtained, respectively:

$$[(\frac{1}{2}\varepsilon\mu)^2 + (\omega - \eta - \frac{3\varepsilon\gamma}{8\eta}a^2)^2]a^2 = (\frac{\varepsilon f}{2\eta})^2, \quad (11)$$

$$\varphi = \tan^{-1}\left(\frac{-\varepsilon\mu}{2(\omega-\eta) - \frac{3\varepsilon\gamma}{4\eta}a^2}\right), \quad (12)$$

In the range of $0 < a \le f/\eta\mu$, the external excitation frequency ω can be calculated from the following equation:

$$\omega = \eta + \frac{3\varepsilon\gamma}{8\eta}a^2 \pm \sqrt{(\frac{\varepsilon f}{2\eta})^2 - (\frac{\varepsilon\mu}{2})^2}, \quad (13)$$

3. Analysis of the main resonance frequency values

As shown in Fig. 4, three cases of Ψ = 0, Ψ = π/4 and Ψ = π/2 are selected for analysis. When Ψ = 0, the response curve is symmetrical, and the resonance frequency is controlled in a very narrow frequency band, which is a linear forced vibration. When Ψ > 0, the amplitude-

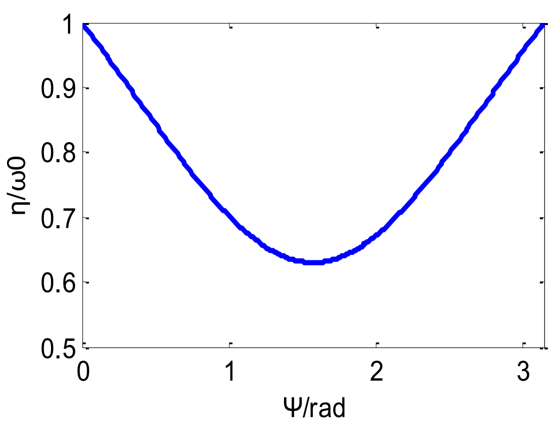

Fig. 3. Frequency ratio changes with the internal power factor angle

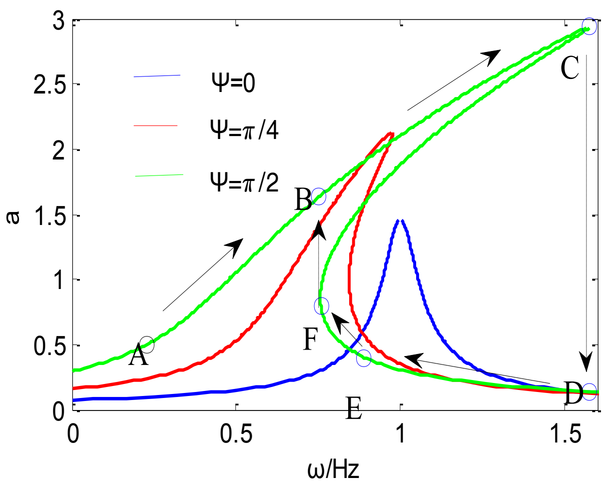

Fig. 4. FEffect of internal power factor angle ψ on the amplitude frequency response of the main resonance

frequency curve is bent to the right, which is a nonlinear forced vibration. As Ψ increases, the degree of the curve's skewing increases with the move to the left, and the amplitude increases. In addition, with the increase of the internal power factor angle, the amplitude of the response curve bending to the right increases, making the resonance and unstable areas larger. This results in the resonance of the transmission system of the pure electric tractor during its normal operation.

CONCLUSIONS. In this paper, the torsional vibration model of the electromechanical coupling of the transmission system of the pure electric tractor driven by PMSM is established by deriving the analytic equation of electromagnetic torque. Based on this, the influence of the internal power factor angle on the natural frequency of the transmission system is studied, and the influence of motor parameters on the torsional vibration characteristics of the system is analyzed. The study allows drawing the following conclusions.

When the internal power factor angle $\psi > 0$, the natural frequency of the torsional vibration of the transmission system declines, which easily makes the natural frequency enter the normal working range of the motor. Considering that the design and control of the automotive permanent magnet synchronous motor are in a weak magnetic field, its size requires a reasonable control.

If the internal power factor angle is increased from 0 to $\pi/2$, the change of the electromagnetic parameters leads to the resonance of the transmission system and the increase of the unstable area. In this case, the transmission system may feature the saddle-junction bifurcation and jump phenomenon, causing fatigue damage to its components, which is detrimental to the stable operation of the transmission system.

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