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Dynamic characteristics and optimization design for disc-shape electro-rheological actuator

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Abstract: A disc-shape electro-rheological actuator was developed. The torque generated by a pair of rotary discs consists of two components: the torque from the viscosity of electro-rheological fluid(ERF) and the torque owing to the field-dependent yield shear stress. The former contributes to the viscous power loss. Changing the application of electric field between the two discs, the output torque of ER actuator can be controlled to amplify the torque of input signal to overcome the load and friction. Relationships between the main structure parameters, the dynamic characteristics of the device, the physical properties of ERF and the torque transmission, range of speed regulation, power loss, stiffness of system were discussed. A multi-object programming model was set up for the optimization design. Experimental simulations with step and pulse signals were presented.

Key words: electro-rheological actuator; structure parameters; optimization design; power loss

1 Introduction

Most devices which use controllable fluids can be classified as pressure driving flow mode, direct-shear mode and squeeze mode. Direct-shear mode devices include actuators, clutches, brakes, dampers, etc. Actuators of direct-shear mode include discs-shape and cylinder-shape.

2 Dynamic characteristics of ER actuator

In this paper, a disc-shape actuator of direct-shear mode was developed, shown in Fig.1. ERF fulfills the gap between active and slaving discs, an angular velocity difference between the two discs generates the shear rate $\dot{\gamma}$. With the application of electric field, an ER torque can be controlled to enhance the input torque from slaving disc for overcoming the loads and frictions.

2.1 Power loss of ER actuator

The shear stress of ERF can be described by the general equation:

$$\tau = \tau_y + \eta \dot{\gamma} \tag{1}$$

where τ_y is the yield stress, η is the viscosity of ERF without electric field, and the shear rate $\dot{\gamma}$ is given by

$$\dot{\gamma} = \frac{\mathrm{d}u}{\mathrm{d}x} = \frac{\mathrm{d}(r\omega_x)}{\mathrm{d}x} = \frac{r}{h}(\Omega - \omega) = \frac{r}{h}\Delta\Omega \tag{2}$$

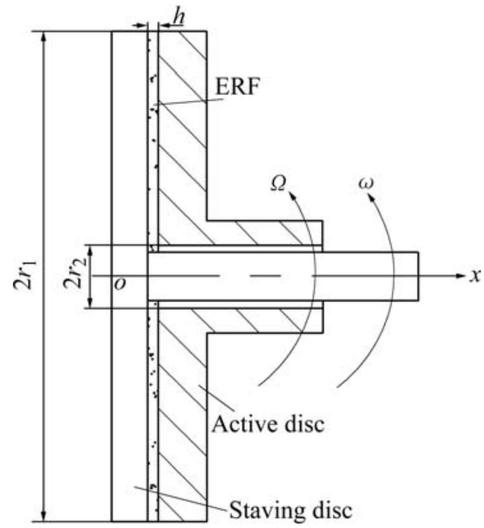

Fig.1 Principle of disc-shape electro-rheological actuator

The total torque generated by a pair of ER disc is

$$T = \int_{r_1}^{r_2} \tau \cdot 2\pi r^2 \mathrm{d}r = \frac{2\pi\tau_y}{3}(r_1^3 - r_2^3) + \frac{\pi\eta\Delta\Omega}{2h}(r_1^4 - r_2^4) \tag{3}$$

where r_1 and r_2 are the outer and inner radius of the two discs, h is the electrode space, and $\Delta\Omega$ is the angular velocity difference.

ER torque described by Eqn.(3) consists of two components, left part, the torque owing to the field dependent yield shear stress τ_y, and the right part, the

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torque generated by the viscosity of ERF which is the main cause of power loss. The viscous power loss W_v is given by

$$W_v = \frac{\pi \eta \tau_y \Delta\Omega^2}{2h}(r_1^4 - r_2^4) \tag{4}$$

Defining W_r as the reference power can be transmitted by field dependent yield shear stress at the angular velocity of $\Delta\Omega$:

$$W_r = \frac{2\pi \tau_y \Delta\Omega}{3}(r_1^3 - r_2^3) \tag{5}$$

The rate of viscous power loss can be described by a dimensionless factor, F_{los}:

$$F_{los} = \frac{W_v}{W_r} = \frac{\frac{\pi \eta \Delta\Omega}{2h}(r_1^4 - r_2^4)}{\frac{2\pi \tau_y}{3}(r_1^3 - r_2^3)} \tag{6}$$

Because of r_2/r_1<<1, we have

$$F_{los} = \frac{3\eta\Delta\Omega}{4\tau_y}\frac{r_1}{h}(1 - \frac{r_2^4}{r_1^4})(1 + \frac{r_2^3}{r_1^3} + \frac{r_2^6}{r_1^6}) \tag{7}$$

The relationships between F_{los} and radius r_1, r_2, velocity difference $\Delta\Omega$, viscosity of ERF η is illustrated in Fig.2. The F_{los} goes up with the increase of η and $\Delta\Omega$, has the maximum of 0.123 at r_1=42, r_2=28.

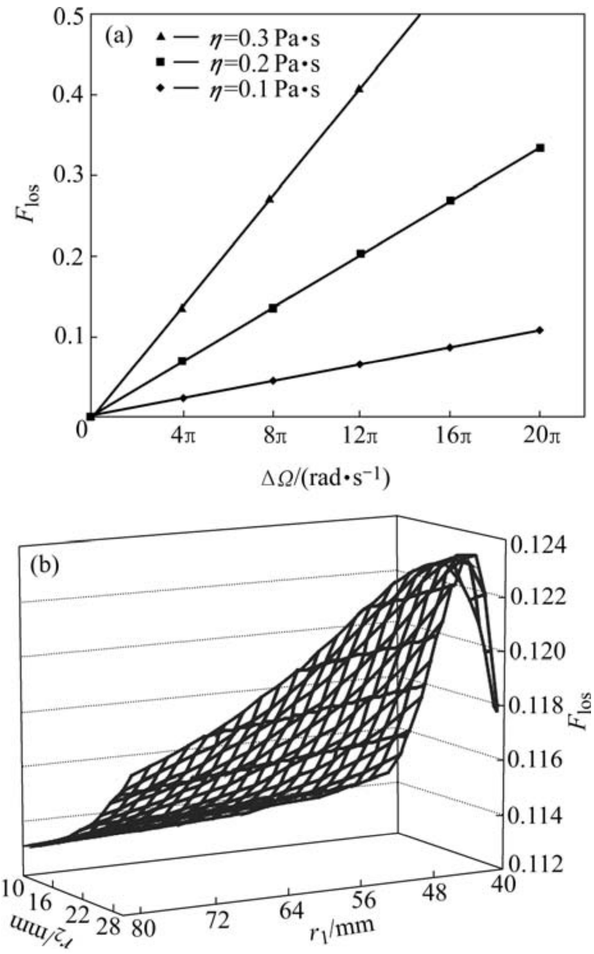

Fig.2 Relationship between F_{los} and $\Delta\Omega$, η (a) and r_1, r_2 (b)

2.2 Range of velocity regulation

From Eqn.(3), the velocity difference $\Delta\Omega$ can be given by

$$\Delta\Omega = \frac{2hT}{\pi\eta r_1^4}\left(1 + \frac{r_2^4}{r_1^4} + \frac{r_2^8}{r_1^8}\right) - \frac{4kU^2}{3h\eta r_1^4}\left(r_1^3 - r_2^3\right)\left(1 + \frac{r_2^4}{r_1^4} + \frac{r_2^8}{r_1^8}\right) \tag{8}$$

The angular velocity difference $\Delta\Omega$ increases with the gap space h while decreases as the radius ratio α increases for certain ERF. The applied electric field and viscosity of ERF can also alter the $\Delta\Omega$. The relationships between h, α and U, η are shown in Fig.3.

From Eqn.(8), it can be expressed as

$$\mathrm{d}\Omega = k_{\Omega T}\mathrm{d}T + k_{\Omega U}\mathrm{d}U \tag{9}$$

where $k_{\Omega T}$ is factor of velocity-torque, and $k_{\Omega U}$ is factor of velocity-voltage,

$$k_{\Omega T} = \frac{2}{\pi\eta}\frac{h}{r_1^4}\left(1 + \alpha^4 + \alpha^8\right)$$

$$k_{\Omega U} = -\frac{8}{3\eta}\frac{kU}{r_1 h}\left(1 - \alpha^3\right)\left(1 + \alpha^4 + \alpha^8\right)$$

It is obviously that reciprocal of the factor of velocity-torque describes the transmission stiffness of the actuator,

$$k_{T\Omega} = \frac{1}{k_{\Omega T}} = \frac{\pi\eta r_1^4}{2h\left(1 + \alpha^4 + \alpha^8\right)} \tag{10}$$

The larger the viscosity η or the smaller the gap space h is, the larger the stiffness is. The transmitted torque lies on the outer radius mostly.

2.3 Transmission performance

In order to evaluate the performance of the actuator, torque transmitted by unit rotational inertia of slaving disc is measured. The rotational inertia of slaving disc is represented by

$$J = \frac{1}{2}\pi b \rho r_1^4$$

Thus the torque transmitted by unit rotational inertia is given by

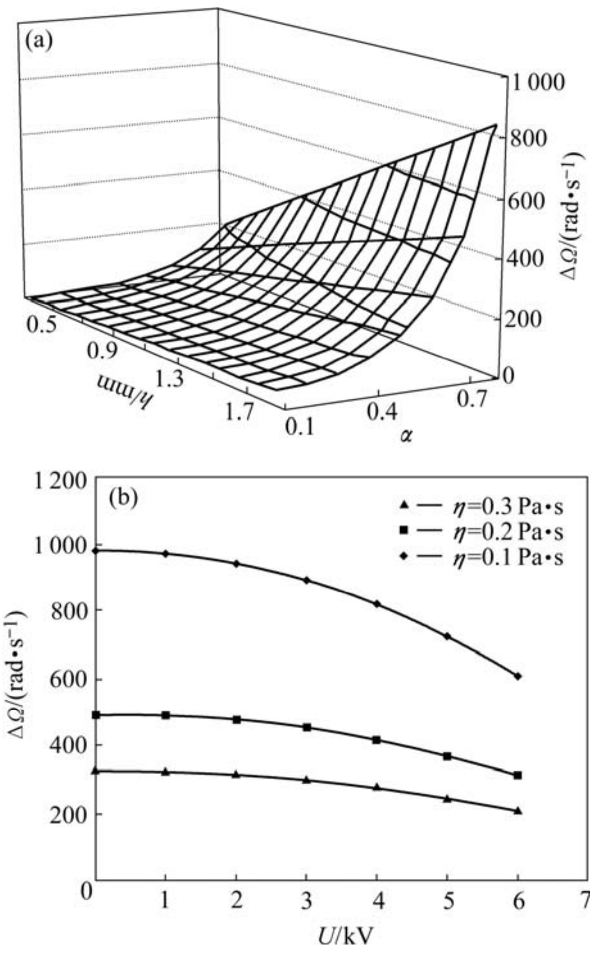

Fig.3 Relationships between $\Delta\Omega$ and h, α (a) and U, η(b)

$$F_{TJ} = \frac{T}{J} = \frac{1}{b\rho}\left[\frac{4kU^2}{3r_1h^2}\left(1-\alpha^3\right)+\frac{\eta_{\rho l}\Delta\Omega}{h}(1-\alpha^4)\right] \quad (11)$$

Fig.4 represents the transmission performance and the relationship between F_{TJ} and radius ratio α, electrodes gap space h.

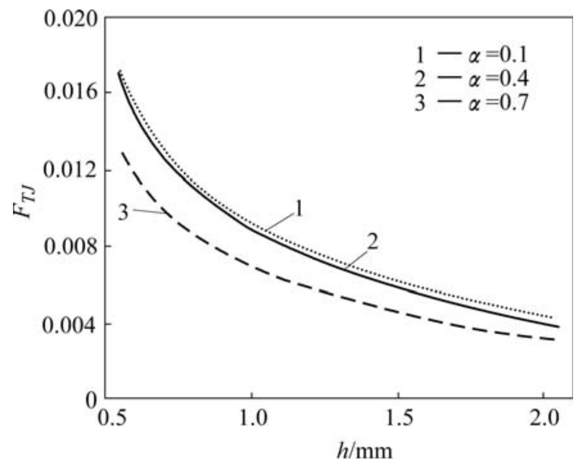

Fig.4 F_{TJ} vs h and a

2.4 Dynamic response

Fig.5 compares the transmitted torque and response time between step and pulse signals under discretional loads. The device has a fast response speed within 0.01 s. The output torque is amplified and has an agreement with that under input discretional loads. It is also observed that the output torque lags behind the input electric signals, this is because of the elastic couples in the device and inertia of discs.

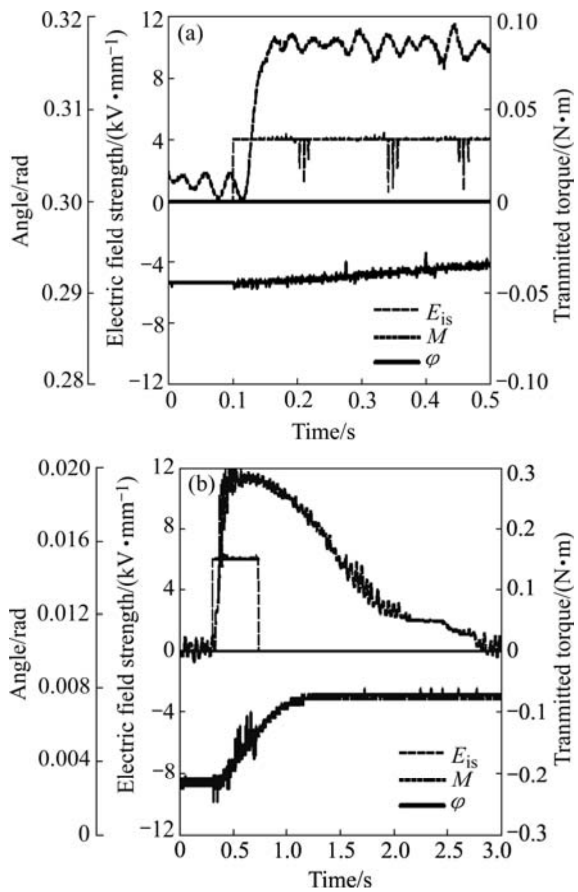

Fig.5 Transmitted torque and responsible time under step signals (a) and pulse signals (b)

3 Optimization

The field dependent ER torque given by Eqn.(3) and torque transmitted by unit rotational inertia given by Eqn.(11) were selected as the objective functions. Considering the precision and machining process, gap space h was defined as h=0.01r_1. The outer radius r_1, inner radius r_2 were the variables to be designed.

The problem of the actuator structure optimization was expressed as

$$\min_{r_i \in D} F = \sum_{i=1}^{2} \lambda_i \left| f_i \right|$$

$$\text{s.t} \quad r_{i,\min} \leqslant r_i \leqslant r_{i,\max},\ \alpha_i \leqslant \dot{f}_i \leqslant \beta_i,\ i=1,\ 2 \quad (12)$$

where f_i is non-dimensional relative departure each rational value respect to their optimum $\dot{f}$, is described by

$$f_i = \frac{F_i - \dot{f}_i}{\beta_i - \alpha_i}$$

where α_i and β_i are the lower and upper limit for feasible region of each objective function; λ is the weight.

Table 1 lists the result of optimation design, Fig.6 compares the results under different weights. When the transmitted torque is the main object, large discs are recommended; relatively small discs can fulfill both torque amplify factor and transmission performance.

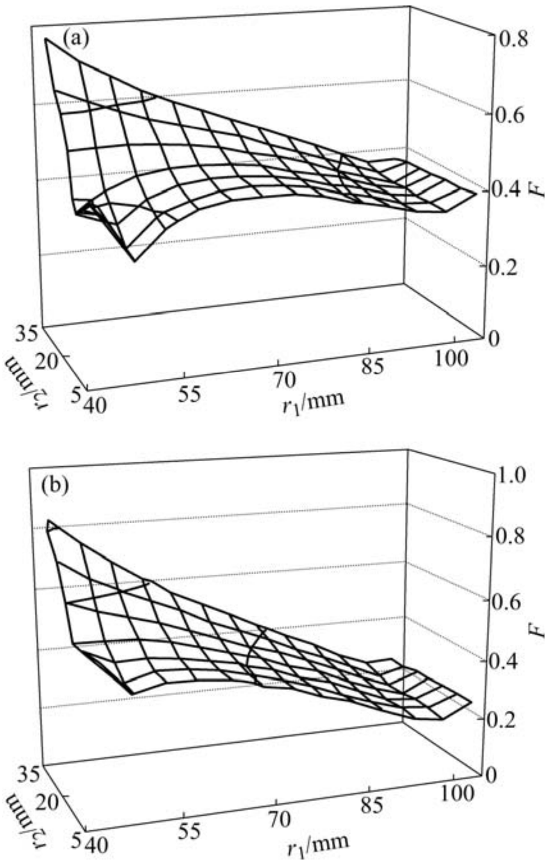

Fig.6 Optimization results under different weights
(a) λ_1=0.5, λ_2=0.5; (b) λ_1=0.7, λ_2=0.3

Table 1 Optimization results

λ_1	λ_2	ς_1	ς_2	r^*	F^*
0.5	0.5	0.7	0.7	[44, 5]	0.289
0.5	0.5	0.6	0.6	[46, 5]	0.175
0.7	0.3	0.7	0.7	[94, 10]	0.209
0.7	0.3	0.6	0.6	[76, 15]	0.159

Note: ς is sub-object factor.

4 Conclusions

1) Structural parameter has great influence to the actuator. Increasing the outer radius of the discs can improve the transmitted torque, range of velocity regulation, velocity stiffness as well as viscous power loss but decrease the torque transmitted by unit ratational inertia. The transmission stiffness, viscous power loss and torque transmitted by unit rotational inertia can be improved by decreasing the electrode gap space.

2) ERF viscosity without electric field also affects the performance of the actuator.

3) Operating conditions such as velocity difference and applied electric field contribute to the transmitted torque in like manner.

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