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Compound Velocity Synchronizing Control Strategy for Electro-Hydraulic Load Simulator and Its Engineering Application

An electro-hydraulic load simulator (EHLS) is a typical case of torque systems with strong external disturbances from hydraulic motion systems. A new velocity synchronizing compensation strategy is proposed in this paper to eliminate motion disturbances, based on theoretical and experimental analysis of a structure invariance method and traditional velocity synchronizing compensation controller (TVSM). This strategy only uses the servo-valve's control signal of motion system and torque feedback of torque system, which could avoid the requirement on the velocity and acceleration signal in the structure invariance method, and effectively achieve a more accurate velocity synchronizing compensation in large loading conditions than a TVSM. In order to facilitate the implementation of this strategy in engineering cases, the selection rules for compensation parameters are proposed. It does not rely on any accurate information of structure parameters. This paper presents the comparison data of an EHLS with various typical operating conditions using three controllers, i.e., closed loop proportional integral derivative (PID) controller, TVSM, and the proposed improved velocity synchronizing controller. Experiments are conducted to confirm that the new strategy performs well against motion disturbances. It is more effective to improve the tracking accuracy and is a more appropriate choice for engineering applications. [DOI: 10.1115/1.4026921]

1 Introduction

An EHLS is essential equipment for hardware-in-the-loop experiments in the aviation and aerospace field [1]. An EHLS simulates the air load on a positioning actuation system (actuator). This allows a designer to test the actual performance of actuators and predict or detect potential problems related to the mechanics and the flight control system under laboratory conditions. As a typical electro-hydraulic force/torque system is strongly coupled with motion disturbances, elimination of motion disturbances is the key problem to achieve accurate load tracking in an EHLS. The solutions for load tracking given in literature can be divided into four types:

(1) Optimization of mechanical parameters

The basic idea is to reduce the impact of motion disturbances through optimization of mechanical parameters such as internal leakage coefficient of hydraulic motor, connecting stiffness. This solution is easy to apply, but its ability to suppress motion disturbances is usually limited.

(2) Mechanical structure compensation

Its purpose is to employ an auxiliary device to eliminate motion disturbances [2,3], such as an interconnected force of a hydraulic motor, oil container, buffer spring adjustments, "double-valve scheme," and "double-motor scheme" that keeps the EHLS synchronization with actuators. Overall, this kind of method is more costly and complex.

(3) Feed-forward decoupling control

Its purpose is to let the EHLS track the motion of actuator systems by using a feed-forward compensation controller. Liu proposed the structure invariance method that uses

actuator's velocity as compensation input to suppress the surplus torque [4–6]. Jiao et al. investigated the disturbances root of an EHLS and presented a velocity synchronizing control method through importing the servo-valve's control signal of the actuator system [7]. Yao et al. proposed an optimal compensation control strategy and designed an adaptive robust torque control strategy to solve the uncertainty of the system model, which needs actuator velocity and acceleration signals [8,9].

(4) Antidisturbances robust control

Taking the coupling motion as external disturbances, an EHLS can adopt a robust method to improve its antidisturbances capability, such as H_∞ mixed sensitivity theory [10], quantitative feedback theory [11–13], and fuzzy technology.

Currently, the structure invariance method based on the actuator's velocity [4] and TVSM [7] are widely used in EHLSs to eliminate disturbances in the engineering practice due to advantages of low cost, convenience, and low computation effort. However, structure invariance control method requires a high quality signal of the actuator's velocity and acceleration in real time, which are always hard to obtain from the actuators. They also have high-frequency noise or phase-lag due to the high-order derivative of the actuator's discrete-angle signal or filtering in actual test conditions. On the other hand, this paper finds that the TVSM becomes poor in large or medium load tracking conditions.

Based on in-depth analysis of the above two controllers, this paper proposes a more effective and convenient compensation strategy to solve the above-mentioned issues. The proposed strategy uses the control signal of actuators and torque feedback signal of an EHLS to compensate for motion disturbances. The system does not require any additional sensors (e.g., angle/velocity/acceleration sensors or pressure difference sensor). This new controller is also convenient for realization in engineering applications because

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Fig. 1 The structure diagram (left) of EHLS and actuator and hydraulic rotary motor (right)

the servo-valve's control signal can be obtained from actuators and every EHLS must have a torque feedback signal.

2 Mathematical Model of EHLS and Actuator

The structure of an EHLS and actuator system is shown in Fig. 1. The left part is the actuator, which consists of a servo-valve, hydraulic motor, angle sensor, and inertia load. The EHLS is on the right side and consists of a hydraulic loading motor, servo-valve, and torque sensor. It is obvious that the EHLS and actuator exist inherent coupling and interacting. Motion disturbances are the main problem in an EHLS. To begin the controller design, practical and reasonable assumptions of the hydraulic systems have to be made.

ASSUMPTION 1. *The servo-valve is matched symmetrically with ideal zero opening and zero lapping, radial-clearance leakage of spool valve and external leakage of motor are negligible, return pressure is zero, and the effective bulk modulus is constant.*

2.1 Modeling of EHLS

- (1) The flow equation of servo-valve [14] in an EHLS is given by

$$Q_L = C_{vL} W_L x_L \sqrt{\frac{1}{\rho_L} [P_{sL} - \text{sign}(x_L) P_{fL}]} \quad (1)$$

where Q_L is the load flow rate (m^3/s), C_{vL} is the discharge coefficient of valve, W_L is the area gradient of valve (m), x_L is the spool valve displacement (m), ρ_L is the density of oil (kg/m^3), P_{sL} is the supply pressure of the fluid (N/m^2), P_{fL} is the pressure drop across hydraulic motor (N/m^2), and $\text{sgn}(\cdot)$ denotes the discontinuous sign function as

$$\text{sgn}(\cdot) = \begin{cases} 1 & \text{if } \cdot > 0 \\ 0 & \text{if } \cdot = 0 \\ -1 & \text{if } \cdot < 0 \end{cases} \quad (2)$$

x_L is related to the control input u_L given by

$$x_L = K_{iL} G_{svL}(s) u_L \quad (3)$$

where K_{iL} is the gain of servo-valve dynamics (m/V), $G_{svL}(s)$ is the valve dynamic function, s the Laplace-transform operator, and u_L is the control signal of servo-valve (V). Substituting Eq. (3) into Eq. (1) yields

$$\begin{aligned} Q_L &= C_{vL} W_L K_{iL} G_{svL}(s) u_L \sqrt{\frac{1}{\rho_L} [P_{sL} - \text{sgn}(x_L) P_{fL}]} \\ &= K_{qL} G_{svL}(s) R_L(P_{sL}, P_{fL}, x_L) u_L \end{aligned} \quad (4)$$

where $K_{qL} = C_{vL} W_L K_{iL} \sqrt{1/\rho_L}$ is the flow gain of valve, and $R_L(P_{sL}, x_L, P_{fL}) = \sqrt{P_{sL} - \text{sgn}(x_L) P_{fL}}$ is the nonlinear part of valve flow, which is related to P_{sL} , x_L , and P_{fL} . Because of the nonlinear property in $R_L(P_{sL}, x_L, P_{fL})$, the flow equation of the servo-valve is always difficult to handle in linear control theory. Based on the back-stepping design method, instead of the actual control input u_L , we take u_{rL} as the virtual control input, which is defined as

$$u_{rL} = R_L(P_{sL}, P_{fL}, x_L) u_L \quad (5)$$

The first step is to synthesize a virtual control law u_{rL} , and the next step is to calculate an actual control input u_L from Eq. (5), given by

$$u_L = u_{rL} / R_L(P_{sL}, P_{fL}, x_L) \quad (6)$$

So the problem to obtain u_L is converted to the problem to synthesize u_{rL} . Noting Eqs. (4) and (6), Eq. (1) can be rewritten as

$$Q_L = K_{qL} G_{svL}(s) u_{rL} \quad (7)$$

- (2) The continuity equation for the loading motor is given as

$$Q_L = D_L \frac{d\theta_L}{dt} + \frac{V_L}{4\beta_e} \frac{dP_{fL}}{dt} + C_{sL} P_{fL} \quad (8)$$

where D_L is the displacement of hydraulic motor (m^3/rad), θ_L is the angle of the EHLS (rad), V_L is the total control volume of the EHLS (m^3), C_{sL} is the total leakage coefficient of the hydraulic motor ($\text{m}^3/(\text{N s})$), and β_e is the effective bulk modulus.

- (3) The torque balance equation for the motor is given by

$$D_L P_{fL} = J_L \frac{d^2\theta_L}{dt^2} + B_L \frac{d\theta_L}{dt} + T \quad (9)$$

where J_L is the inertia of the motor (kg m^2), B_L is the viscous damping coefficient (Nm/rad), G_s is the stiffness of the torque sensor (Nm/rad), and T is the torque output (Nm).

- (4) The torque output of the EHLS is

$$T = G_s(\theta_L - \theta_a) \quad (10)$$

where θ_a is the angle of the actuator system (rad).

- (5) The output tracking error e is defined as

$$e = T - T_r \quad (11)$$

where T_r is the torque command (Nm).

Combining Eqs. (7)–(9), and using Laplace-transform technique, the mathematical model is

$$T = \frac{G_{L1}(s)u_{rL} - G_{L2}(s)s\theta_L}{G_{L3}(s)} \quad (12)$$

where

$$\begin{aligned} G_{L1}(s) &= D_L K_{qL} G_{svL}(s) \\ G_{L2}(s) &= \frac{J_L V_L}{4\beta_e} s^2 + \left(J_L K_{tmL} + \frac{B_L V_L}{4\beta_e} \right) s + B_L K_{tmL} + D_L^2 \\ G_{L3}(s) &= \frac{V_L}{4\beta_e} s + K_{tmL} \end{aligned}$$

$K_{tmL} = C_{sIL}$ is the flow-pressure coefficient ($m^5/(N \cdot s)$). Substituting Eq. (10) into Eq. (12) yields

$$\begin{aligned} T &= \frac{G_{L1}(s)u_{rL} - G_{L2}(s)s(\theta_a + T/G_s)}{G_{L3}(s)} \\ \Rightarrow G_{L3}(s)T &= G_{L1}(s)u_{rL} - G_{L2}(s)s\theta_a - G_{L2}(s)sT/G_s \\ \Rightarrow (G_{L3}(s) + G_{L2}(s)s/G_s)T &= G_{L1}(s)u_{rL} - G_{L2}(s)s\theta_a \\ \Rightarrow T &= \frac{G_{L1}(s)u_{rL} - G_{L2}(s)s\theta_a}{G_{L4}(s)} \end{aligned} \quad (13)$$

where

$$\begin{aligned} G_{L4}(s) &= G_{L3}(s) + \frac{G_{L2}(s)s}{G_s} = \frac{J_L V_L}{4\beta_e G_s} s^3 \\ &+ \left(\frac{J_L K_{tmL}}{G_s} + \frac{B_L V_L}{4\beta_e G_s} \right) s^2 \\ &+ \left(\frac{B_L K_{tmL}}{G_s} + \frac{V_L}{4\beta_e} + \frac{D_L^2}{G_s} \right) s + K_{tmL} \end{aligned}$$

2.2 Mathematic Model of Actuator

(1) The flow equation of the actuator valve [14] is given by

$$Q_a = C_{va} W_a x_a \sqrt{\frac{1}{\rho_a} [P_{sa} - \text{sgn}(x_a) P_{fa}]} \quad (14)$$

where Q_a is the load flow rate (m^3/s), C_{va} is the discharge coefficient, W_a is the area gradient of valve (m), x_a is the spool valve displacement (m), ρ_a is the density of oil (kg/m^3), P_{sa} is the supply pressure of the fluid (N/m^2), and P_{fa} is the pressure drop across hydraulic motor (N/m^2). x_a is related to the control input u_a given by

$$x_a = K_{ia} G_{sva}(s) u_a \quad (15)$$

where K_{ia} is the gain of servo-valve dynamics (m/V), $G_{sva}(s)$ is the valve dynamic function, and u_a is the control input of the servo-valve (V).

Substituting Eq. (15) into Eq. (14) yields

$$\begin{aligned} Q_a &= C_{va} W_a K_{ia} G_{sva}(s) u_a \sqrt{\frac{1}{\rho_a} [P_{sa} - \text{sgn}(x_a) P_{fa}]} \\ &= K_{qa} G_{sva}(s) R_a(P_{sa}, P_{fa}, x_a) u_a \end{aligned} \quad (16)$$

where $K_{qa} = C_{va} W_a K_{ia} \sqrt{1/\rho_a}$ is the flow gain of the servo-valve, and $R_a(P_{sa}, P_{fa}, x_a) = \sqrt{P_{sa} - \text{sgn}(x_a) P_{fa}}$ is the non-linear part, which is related to P_{sa} , x_a , and P_{fa} [15,16].

Similar to Eq. (7), the servo-valve flow equation is given by

$$Q_a = K_{qa} G_{sva}(s) u_{ra} \quad (17)$$

where $u_{ra} = R_a(P_{sa}, P_{fa}, x_a) u_a$ is the virtual control input.

(2) The continuity equation for the hydraulic motor is given as

$$Q_a = D_a \frac{d\theta_a}{dt} + \frac{V_a}{4\beta_e} \frac{dP_{fa}}{dt} + C_{sla} P_{fa} \quad (18)$$

where D_a is displacement of the hydraulic motor (m^3/rad), θ_a is angle output (rad), V_a is the total control volume (m^3), and C_{sla} is total leakage coefficient of the hydraulic motor ($m^5/(N \cdot s)$).

(3) The torque balance equation for the motor is given by

$$D_a P_{fa} = J_a \frac{d^2\theta_a}{dt^2} + B_a \frac{d\theta_a}{dt} - T \quad (19)$$

where J_a is the inertia of actuator motor and rudder ($kg \cdot m^2$) and B_a is the viscous damping coefficient (Nm/rad).

Combining Eqs. (17)–(19), the mathematical model of the actuator is described as

$$\dot{\theta}_a = \frac{G_{a1}(s)u_{ra} + G_{a3}(s)T}{G_{a2}(s)} \quad (20)$$

where

$$\begin{aligned} G_{a1}(s) &= D_a K_{qa} G_{sva}(s) \\ G_{a2}(s) &= \frac{J_a V_a}{4\beta_e} s^2 + \left(J_a K_{tma} + \frac{B_a V_a}{4\beta_e} \right) s + B_a K_{tma} + D_a^2 \\ G_{a3}(s) &= \frac{V_a}{4\beta_e} s + K_{tma} \end{aligned}$$

$K_{tma} = C_{sla}$ is the flow-pressure coefficient ($m^5/(N \cdot s)$).

3 Theory and Experiment Analysis on Structure Invariance Method and TVSM

It can be seen from Eq. (13), the motion coupling disturbances in an EHLS is related to the velocity, acceleration, and jerk of the actuator system, whose block diagram is shown in Fig. 2.

If we only use a PID torque controller, the effect of torque tracking will be seriously affected by the actuator velocity $\dot{\theta}_a$. Therefore, the structure invariance method adopted the actuator velocity signal to achieve the goal of eliminating motion disturbances, whose principle diagram is in Fig. 3.

In theory, the compensation unit $G_v(s)$ in Fig. 3, which satisfies the equation: $G_{L2}(s) = G_v(s) \cdot G_{L1}(s)$, can entirely eliminate disturbances in the EHLS. Hence, $G_v(s)$ should satisfy

Fig. 2 The block diagram of EHLS with motion disturbances

Fig. 3 The principle diagram of the structure invariance method

Fig. 4 The principle diagram of TVSM

$$G_v(s) = \frac{G_{L2}(s)}{G_{L1}(s)} = \frac{J_L V_L}{4\beta_e} s^2 + \left(\frac{J_L K_{tmL}}{4\beta_e} + \frac{B_L V_L}{4\beta_e} \right) s + B_L K_{tmL} + D_L^2 \quad (21)$$

$$= \frac{G_{svL}(s) D_L K_{qL}}{G_{svL}(s) D_L K_{qL}}$$

However, practical design and engineering implementation of the $G_v(s)$ is difficult because of the high-order differentials of the actuator angle, model error, and dynamic characteristic of the servo-valve, nonlinear, and time-varying parameters in $G_v(s)$. Hence, $G_v(s)$ is usually simplified as constant. In this case, the phase advance compensation effect of $\ddot{\theta}_a$, $\dot{\theta}_a$, and θ_a in $G_v(s)$ will be ignored. Thus, the phase-lag compensation effect will be caused for only using actuator velocity signal, which will greatly influence the ability of eliminating motion disturbances in medium-high frequency motion disturbances. Further, owing to large installation clearance, low precision of angle sensor, and long sampling time of flight actuator, the derivative of the angle sensor always has high-frequency noise. After filtering, it will have larger phase-lag effect, which further limits the ability to eliminate disturbances in medium-high frequency motion disturbances.

Because a high quality signal of velocity/acceleration/jerk is difficult to obtain, the TVSM proposes to use the control signal of the actuator's servo-valve (u_a), which is approximately proportional to the actuator's velocity. u_a is not noisier than velocity signal. What's more, the phase of u_a is more advanced than velocity signal, which can have a better prediction effect.

In engineering practice, it has been shown that TVSM based on u_a is better in eliminating motion disturbances than the structure invariance method, especially for motion disturbances in medium-high frequency. The basic idea of the TVSM can be understood as: due to similarity of the mathematical models between an EHLS and actuator, the synchronization of the valve signal can ensure the velocity synchronization [7]. Its principle diagram is shown in Fig. 4.

Fig. 5 Comparison between u_L and u_{com} with 0 Nm torque command

Fig. 6 Comparison between u_L and u_{com} with 500 Nm at 1 Hz torque command

But in this paper, it has been found based on lots of experiments that TVSM can generate good compensation effect only under small load, but in large and medium torque tracking, the compensation will be badly affected. To analyze this phenomenon, comparative experiments have been made in certain load simulator, whose maximum dynamic output torque is 650 Nm. Figure 5 denotes the comparison between the total control signal u_L of an EHLS's servo-valve and feed-forward synchronous compensation input u_{com} (proportional to valve signal of actuator system), which is controlled by TVSM with 0 Nm torque command under sinusoidal motion disturbances (1 deg at 5 Hz).

Figure 5 shows under 0 Nm tracking condition, u_{com} and u_L are basically the same. So the closed-loop controller output u_c based on torque error will be very small, which means that TVSM has very good feed-forward compensation effect.

However, under large or medium load conditions, u_{com} cannot have very good feed-forward compensation effect. Figure 6 is the comparison between u_L and u_{com} , when the same load simulator tracks 500 Nm at 1 Hz sinusoidal torque command with the same motion disturbances (1 deg at 5 Hz).

Figure 6 indicates that under large or medium loading conditions, TVSM is not very good. This also means that TVSM, based on valve signal of actuator system, needs further improvement.

To further analyze ways to improve TVSM, u_c from Fig. 7, which represents difference between u_L and u_{com} from Fig. 6. The torque output T is shown in Fig. 8. From Figs. 7 and 8, it can be found that both u_c and T have the similar phase and waveform. It means that the rational use of torque T could further improve compensation effect of TVSM.

4 New Velocity Synchronizing Control Strategy

It can be found from Figs. 6–8 that velocity compensation effect is not only related to u_a but also affected by T . This phenomenon could be explained as follows: when the actuator and EHLS have the similar velocity, the interaction torque T will make that P_{fa} and P_{fL} equal approximately. However, P_{fa} and P_{fL}

Fig. 7 The closed-loop controller output u_c

Fig. 8 The torque output T of loading simulator

do not always have the same direction. So TVSM, only using u_a to compensate the motion disturbances, may result in two adverse effects:

- (1) With the different direction between P_{fa} and P_{fl} , the same valve input will cause the valve flow rate Q_L of the EHLS and Q_a of the actuator to be very different. In this case, traditional method only based on u_a will not get good velocity synchronizing compensation effect.
For example: under the condition of maximum dynamic torque of the EHLS, we can get $P_{fl} = (2/3)P_{sL}$. Assuming both the EHLS and actuator have the same mechanical structure and oil supply pressure, with the constant velocity between EHLS and actuator, we can get $P_{fa} = (-2/3)P_{sa}$. So with Eqs. (1) and (14), we can get

$$\begin{aligned} \frac{Q_a}{Q_L} &= \frac{C_{va}W_a x_a \sqrt{\frac{1}{\rho_a}(P_{sa} - \text{sgn}(x_a)P_{fa})}}{C_{vL}W_L x_L \sqrt{\frac{1}{\rho_L}(P_{sL} - \text{sgn}(x_L)P_{fl})}} \\ &= \frac{\sqrt{5/3}P_{sa}}{\sqrt{1/3}P_{sL}} = \sqrt{5} \end{aligned} \quad (22)$$

- (2) With the different direction between P_{fa} and P_{fl} , the leakage flow of the EHLS and the leakage flow of the actuator will have the opposite direction. If we only use feed-forward of u_a and do not consider load effects, the compensation of leakage flow will cause the opposite effect.
The actuator's velocity has relationship with the valve flow rate, leakage flow, and oil elasticity, which depend both on P_{fa} and u_a . Hence, in order to completely eliminate motion disturbances, we should introduce P_{fa} and u_a .

4.1 New Control Strategy Based on u_a and P_{fa} . In this paper, we put forward an improved velocity synchronizing method based on u_a and P_{fa} to eliminate motion disturbances.

From Eqs. (17) and (18), we can get

$$\dot{\theta}_a = [k_{qa}G_{sva}(s)u_{ra} - k_{tmsa}(s)P_{fa}]/D_a \quad (23)$$

where

$$K_{tmsa}(s) = \frac{V_a}{4\beta_e}s + K_{tma} \quad (24)$$

Substituting Eq. (23) into Eq. (13) yields

$$T(s) = \frac{G_{L1}(s)u_{rL} - \{G_{L2}(s)[k_{qa}G_{sva}(s)u_{ra} - k_{tmsa}(s)P_{fa}]/D_a\}}{G_{L4}(s)} \quad (25)$$

So the new virtual controller law can be synthesized as

$$\begin{aligned} u_{rL} &= u'_s + u'_{com} \\ u'_{com} &= G_v \cdot \dot{\theta}_a = \frac{G_{L2}(s)}{G_{L1}(s)} \cdot \frac{[k_{qa}G_{sva}(s)u_{ra} - k_{tmsa}(s)P_{fa}]}{D_a} \\ &= G'_{com1} \cdot u_{ra} + G'_{com2} \cdot P_{fa} \\ G'_{com1} &= \frac{J_L K_{tmsL}(s)s + B_L K_{tmsL}(s) + D_L^2}{D_L D_a} \cdot \frac{k_{qa}G_{sva}(s)}{K_{qL}G_{svL}(s)} \\ G'_{com2} &= \frac{J_L K_{tmsL}(s)s + B_L K_{tmsL}(s) + D_L^2}{D_L K_{qL}G_{svL}(s)D_a} \cdot k_{tmsa}(s) \\ K_{tmsL}(s) &= K_{tmL} + \frac{V_L}{4\beta_e}s \end{aligned} \quad (26)$$

where u'_{com} is the feed-forward compensation input to eliminate motion disturbance, G'_{com1} is the compensation function of u_{ra} , G'_{com2} is the compensation function of P_{fa} , and u'_s is the closed-loop robust input. Substituting Eq. (26) into Eq. (25) yields

$$T(s) = \frac{G_{L1}(s)}{G_{L4}(s)} u'_s \quad (27)$$

4.2 New Control Strategy Based on u_a and T . Because not all the actuators could provide pressure drop signal P_{fa} , in order to more conveniently use in practical engineering, we further introduce the torque output T to achieve the same functionality in this paper. The new controller law can be synthesized as

$$u_{rL} = u_s + u_{comL} \quad (28)$$

where u_{comL} is the feed-forward compensation term to eliminate motion disturbance and u_s is the closed-loop robust term. From Eqs. (13) and (20), we can get

$$T(s) = \frac{G_{L1}(s)u_{rL} - G_{L2}(s) \left[\frac{G_{a1}(s)u_{ra} + G_{a3}(s)T}{G_{a2}(s)} \right]}{G_{L4}(s)} \quad (29)$$

From Eq. (29), the compensation term u_{comL} is given by

$$\begin{aligned} u_{comL} &= \frac{G_{L2}(s)}{G_{L1}(s)} \cdot \frac{G_{a1}(s)u_{ra} + G_{a3}(s)T}{G_{a2}(s)} \\ &= G_{com1}(s) \cdot u_{ra} + G_{com2}(s) \cdot T \\ &= u_1 + u_2 \end{aligned} \quad (30)$$

where

$$\begin{aligned} u_1 &= \frac{G_{a1}(s)G_{L2}(s)}{G_{L1}(s)G_{a2}(s)} u_{r2} = G_{com1}(s)u_{r2} \\ u_2 &= \frac{G_{a3}(s)G_{L2}(s)}{G_{L1}(s)G_{a2}(s)} T = G_{com2}(s)T \\ G_{com1}(s) &= \frac{D_a K_{qa} G_{sva}(s)(J_L K_{tmsL}(s)s + B_L K_{tmsL}(s) + D_L^2)}{D_L K_{qL} G_{svL}(s)(J_a K_{tmsa}(s)s + B_a K_{tmsa}(s) + D_a^2)} \\ G_{com2}(s) &= \frac{K_{tmsa}(s)(J_L K_{tmsL}(s)s + B_L K_{tmsL}(s) + D_L^2)}{D_L K_{qL} G_{svL}(s)(J_a K_{tmsa}(s)s + B_a K_{tmsa}(s) + D_a^2)} \end{aligned} \quad (31)$$

Substituting Eq. (31) into Eq. (29) yields

$$T(s) = \frac{G_{L1}(s)(u_s + u_{comL}) - G_{L2}(s) \frac{G_{a1}(s)u_{ra} + G_{a3}(s)T}{G_{a2}(s)}}{G_{L4}(s)} \quad (32)$$

$$= \frac{G_{L1}(s)}{G_{L4}(s)} u_s$$

From Eq. (32), it can be found that the torque output will not be affected by the actuator's motion disturbances.

Remark 1. The primary advantage of the velocity synchronizing method in Eq. (31) is that the compensation term G_{com1} is more operable in eliminating motion disturbances than the compensation term G_v of the structure invariance method in Eq. (21) in practical engineering.

In order to eliminate high-frequency disturbances motion, if using the structure invariance method in Eq. (21), higher differentiation of the actuator velocity signal is needed to obtain the effect of advanced phase. But if adopting the velocity synchronizing method in Eq. (31), the compensation term G_{com1} does not need to take the derivative of valve signal, because G_{com1} does not need to obtain advanced phase by reasonably designing mechanical parameters of the EHLS in Eq. (31), such as higher response servo-valve $G_{svL}(s)$ and lower inertia $J_L (J_L \leq J_a)$, which also explains that the EHLS should have the faster response than the actuator system. On the other hand, assuming both actuator and EHLS have the same structure parameters, from Eqs. (21) and (31), we can easily get

$$G_v(s) \cdot \dot{\theta}_a = \frac{J_L V_L s^2 + \left(J_L K_{tmL} + \frac{B_L V_L}{4\beta_e} \right) s + B_L K_{tmL} + D_L^2}{G_{svL}(s) D_L K_{qL}} \cdot \dot{\theta}_a$$

$$G_{com1}(s) \cdot u_{ra} = u_{ra} \quad (33)$$

Obviously, in order to fully eliminate motion disturbances, the former should take third-order differential of the actuator velocity, while the latter just needs the const gain, which is more welcome in engineering application.

Remark 2. In comparison to TVSM, the new compensation term u_{comL} , which considers the load effect, can greatly improve the ability to eliminate motion disturbances under large-medium torque tracking conditions. In the meantime, the new method just uses torque output feedback signal T and does not need to add new sensors, which is more practical.

5 Design of the Compound Control Strategy

Using the velocity synchronizing method above, from Eqs. (12), (13), and (32), we can get

$$T(s) = \frac{G_{L1}(s)}{G_{L4}(s)} u_s$$

$$= \frac{D_L K_{qL} G_{svL}(s) u_s}{\frac{J_L V_L}{4\beta_e G_s} s^3 + \left(\frac{J_L K_{tmL}}{G_s} + \frac{B_L V_L}{4\beta_e G_s} \right) s^2 + \left(\frac{B_L K_{tmL}}{G_s} + \frac{V_L}{4\beta_e} + \frac{D_L^2}{G_s} \right) s + K_{tmL}} \quad (34)$$

It could be concluded that T will not be affected by the actuator's motion disturbances, but it is hard to find further compensation method just from Eq. (34). So in this paper, substituting Eq. (34) into Fig. 2 yields Fig. 9, which is the block diagram of the EHLS without motion disturbances. In this case, the flow of loading system consists of three parts indicated in Fig. 9, which satisfy

$$Q_1 = Q_2 + Q_3 \quad (35)$$

where

$$Q_1 = G_{svL}(s) K_{qL} u_s$$

$$Q_2 = \frac{K_{tmL}(s)}{D_L} T = \frac{K_{tmL} + \frac{V_L}{4\beta_e} s}{D_L} T$$

$$Q_3 = \frac{J_L K_{tmL}(s) s + B_L K_{tmL}(s) + D_L^2}{G_s D_L} T_s$$

Q_1 is servo-valve flow related to control input u_s , Q_2 is the sum of leakage flow and elastic compression flow, and Q_3 is the flow for stiffness deformation of torque sensor. From the above definitions, Q_1 can be decomposed into

$$Q_1 = K_{qL} G_{svL}(s) u_s = K_{qL} G_{svL}(s) (u_3 + u_4 + u_c)$$

$$= K_{qL} G_{svL}(s) u_3 + K_{qL} G_{svL}(s) u_4 + K_{qL} G_{svL}(s) u_c \quad (36)$$

where $u_c = u_3 + u_4 + u_s$, $K_{qL} G_{svL}(s) u_3$ is used to compensate Q_2 (the sum of leakage flow and elastic compression flow), $K_{qL} G_{svL}(s) u_4$ is used to compensate Q_3 (the flow for stiffness deformation of torque sensor), and $K_{qL} G_{svL}(s) u_c$ is robust control item of closed-loop system. Then, u_3 should satisfy

$$K_{qL} G_{svL}(s) u_3 = Q_2 \Rightarrow u_3 = \frac{Q_2}{G_{svL}(s) K_{qL}} = \frac{K_{tmL}(s)}{D_L K_{qL} G_{svL}(s)} T \quad \text{or}$$

$$K_{qL} G_{svL}(s) u_3 = Q_2 \Rightarrow u_3 = \frac{Q_2}{G_{svL}(s) K_{qL}} \quad (37)$$

u_4 can be chosen as

$$K_{qL} G_{svL}(s) u_4 = Q_3 \Rightarrow u_4 = \frac{Q_3}{K_{qL} G_{svL}(s)}$$

$$= \frac{J_L K_{tmL}(s) s + B_L K_{tmL}(s) + D_L^2}{G_s D_L K_{qL} G_{svL}(s)} T_s \quad \text{or} \quad (38)$$

$$K_{qL} G_{svL}(s) u_4 = Q_3 \Rightarrow u_4 = \frac{Q_3}{K_{qL} G_{svL}(s)}$$

High-order differentials of T in u_4 is hard to obtain in an EHLS. On the other hand, compared with Q_1 and Q_2 , the flow for stiffness deformation of torque sensor Q_3 is smaller. So we use T_r instead of T in

$$u_4 = \frac{J_L K_{tmL}(s) s + B_L K_{tmL}(s) + D_L^2}{G_s D_L K_{qL} G_{svL}(s)} T_{rs} \quad (39)$$

u_c can be chosen as robust control input of the closed-loop system based on torque errors given by

Fig. 9 The block diagram of EHLS without motion disturbances

$$u_c = G_{com5}(s)e \quad (40)$$

where the most common expression of G_{com5} is the PID controller given by

$$G_{com5}(s) = T_p + \frac{1}{T_i s} + T_d s \quad (41)$$

in which T_p , T_i , and T_d are proportional gain, integral gain, and derivative gain, respectively. Noting Eqs. (37), (39), and (40)

$$\begin{aligned} u_s &= u_3 + u_4 + u_c \\ &= G_{com3}(s)T + G_{com4}(s)\dot{T}_r + G_{com5}(s)e \end{aligned} \quad (42)$$

where

$$\begin{aligned} u_3 &= G_{com3}(s)T \\ u_4 &= G_{com4}(s)\dot{T}_r \\ G_{com3}(s) &= \frac{K_{tmsL}(s)}{D_L K_{qL} G_{svL}(s)} \\ G_{com4}(s) &= \frac{J_L K_{tmsL}(s)s + B_L K_{tmsL}(s) + D_L^2}{G_s D_L K_{qL} G_{svL}(s)} \end{aligned}$$

So the virtual control law consists of five terms as

$$\begin{aligned} u_{rL} &= u_{comL} + u_s \\ &= u_1 + u_2 + u_3 + u_4 + u_c = u_{com} + u_c \\ &= G_{com1}(s)u_{ra} + G_{com2}(s)T \\ &\quad + G_{com3}(s)T + G_{com4}(s)\dot{T}_r + G_{com5}(s)e \end{aligned} \quad (43)$$

where $u_{com} = u_1 + u_2 + u_3 + u_4$ represents all the feed-forward compensation inputs. From Eq. (43), the block diagram of the new compound controller is illustrated in Fig. 10.

Remark 3. From Eq. (43) and Fig. 10, the proposed controller consists of five parts: part 1 is u_1 , which compensates servo-valve flow of the actuator system; part 2 is u_2 , which compensates leakage flow and elastic compression of the actuator caused by T ; part 3 is u_3 , which compensates leakage flow and elastic compression of the EHLS caused by T ; part 4 is u_4 , which compensates the flow for stiffness deformation of the torque sensor; the fifth part is robust control item of the closed-loop system based on torque error, in which u_1 and u_2 compose the velocity synchronizing compensation network.

So the next step is to calculate an actual control input u_L from Eqs. (6) and (43) as follows:

$$\begin{aligned} u_L &= u_{rL}/R_L(P_{sL}, P_{fL}, x_L) \\ &= (u_{r1} + u_{r2} + u_{r3} + u_{r4} + u_{rs})/R_L(P_{sL}, P_{fL}, x_L) \end{aligned} \quad (44)$$

As mentioned above, we can synthesize an actual control law, which takes the nonlinear flow property of the valve into account. From Eqs. (1) and (14), it can be found that flow equations of the valve need at least six sensors, including spool valve displacement (x_L , x_a), pressure drop (P_{fL} , P_{fa}), and supply oil pressure (P_{sL} , P_{sa}). However, in the actual conditions, most EHLS/actuator systems do not contain above six sensors. The typical design methodology is to locally linearize the nonlinear flow equations around a nominal operation point. In contrast to the changes of P_{fL} and P_{fa} , we can assume that P_{sL} and P_{sa} are const. So linear equations of the valve in Eqs. (1) and (14) are given by Ref. [14]

$$\begin{cases} Q_L = K'_{qL}x_L - K_{cL}P_{fL} \\ Q_a = K'_{qa}x_a - K_{ca}P_{fa} \end{cases} \quad (45)$$

Fig. 10 The block diagram of the new compound controller

where K'_{qL} is the flow coefficient of EHLS valve (m^2/s), K_{cL} is the flow-pressure coefficient of the EHLS valve ($m^5/N s$), K'_{qa} is the flow coefficient of the actuator valve (m^2/s), and K_{ca} is the flow-pressure coefficient of actuator valve ($m^5/N s$). Substituting Eqs. (3) and (15) into Eq. (45) yields

$$\begin{cases} Q_L = K_{qL}G_{svL}(s)u_L - K_{cL}P_{fL} \\ Q_a = K_{qa}G_{sva}(s)u_a - K_{ca}P_{fa} \end{cases} \quad (46)$$

where, $K_{qL} = K'_{qL}K_{iL}$, $K_{qa} = K'_{qa}K_{ia}$. In this case, if we define

$$\begin{cases} u_{rL} = u_L, K_{imL} = K_{cL} + C_{sIL} \\ u_{ra} = u_a, K_{ima} = K_{ca} + C_{sla} \end{cases} \quad (47)$$

the control law will have the same structure as Eq. (43) given by

$$\begin{aligned} u_L &= u_{rL} = u_{comL} + u_s \\ &= u_1 + u_2 + u_3 + u_4 + u_c = u_{com} + u_c \\ &= G_{com1}(s)u_a + G_{com2}(s)T + G_{com3}(s)T + G_{com4}(s)\dot{T}_r \\ &\quad + G_{com5}(s)e \end{aligned} \quad (48)$$

where G_{com1} , G_{com2} , G_{com3} , G_{com4} , and G_{com5} have the same definitions as Eqs. (30) and (42).

So whether we use nonlinear flow equations or linear flow equations of servo-valve, the proposed compound velocity synchronizing methods have the same control structure for the convenience to use the proposed controller. In Sec. 6, we will discuss how to determinate the values of control parameters in G_{com1} , G_{com2} , G_{com3} , G_{com4} , and G_{com5} .

6 Selection Rules of Control Parameters in New Compound Velocity Synchronizing Scheme

ASSUMPTION 2. An EHLS is used to provide high-precision torque tracking under motion disturbances of actuators. Therefore, in the design process of EHLS, high/ultrahigh-response servo-valves are usually employed, whose response bandwidth should be higher than the actuator's servo-valve. In the working frequency of actuators, we suppose

$$K_{qa}G_{sva}(s) \approx \frac{K_{qa}}{T_s + 1} \quad (49)$$

$$K_{qL}G_{svL}(s) \approx K_{qL} \quad (50)$$

where T_s is the response time constant of actuator's valve.

ASSUMPTION 3. To reduce the adverse effect of motion disturbances and improve the stability of the EHLS, hydraulic motors with bypass orifice are usually employed for larger K_{imL} . On the other

hand, β_e is always very large ($> 700 \times 10^6 \text{ Pa}$) and the motor chamber V_L is very small (tens of cm^3) [5]. Therefore, in EHLS we suppose

$$K_{\text{tmsL}}(s) = K_{\text{tmL}} + \frac{V_L}{4\beta_e} s \approx K_{\text{tmL}} \quad (51)$$

The robust control part u_c can be chosen as a PID controller. Based on the above assumptions, Eqs. (43) and (48) can be simplified to

$$\begin{aligned} G_{\text{com1}}(s) &= \frac{D_a K_{\text{qa}} G_{\text{sva}}(s) (J_L K_{\text{tmL}} s + B_L K_{\text{tmL}} + D_L^2)}{D_L K_{\text{qL}} (J_a K_{\text{tmsa}}(s) s + B_a K_{\text{tmsa}}(s) + D_a^2)} \\ &= K_{\text{com1}} \frac{1}{T_s + 1} \frac{(T_{\text{com1}} s + 1)}{T_1 s^2 + T_2 s + 1} \\ G_{\text{com2}}(s) &= \left[\frac{K_{\text{tma}}(s) (J_L K_{\text{tmL}} s + B_L K_{\text{tmL}} + D_L^2)}{D_L K_{\text{qL}} (J_a K_{\text{tmsa}}(s) s + B_a K_{\text{tmsa}}(s) + D_a^2)} \right] \\ &= K_{\text{com2}} \frac{(T_{\text{com2}} s^2 + T_{\text{com3}} s + 1)}{T_1 s^2 + T_2 s + 1} \\ G_{\text{com3}}(s) &= \frac{K_{\text{tmL}}}{D_L K_{\text{qL}}} = K_{\text{com3}} \\ G_{\text{com4}}(s) &= \frac{J_L K_{\text{tmL}} s + B_L K_{\text{tmL}} + D_L^2}{G_s D_L K_{\text{qL}}} = K_{\text{com4}} (T_{\text{com4}} s + 1) \\ G_{\text{com5}}(s) &= T_p + \frac{1}{T_i s} + T_d s \end{aligned} \quad (52)$$

where K_{com1} , K_{com2} , K_{com3} , T_1 , T_2 , T_{com1} , T_{com2} , T_{com3} , and T_{com4} are the compensation parameters, which are given by

$$\begin{aligned} K_{\text{com1}} &= \frac{D_a K_{\text{qa}} (B_L K_{\text{tmL}} + D_L^2)}{D_L K_{\text{qL}} (B_a K_{\text{tma}} + D_a^2)}; \quad K_{\text{com2}} = \frac{K_{\text{tma}} (B_L K_{\text{tmL}} + D_L^2)}{D_L K_{\text{qL}} (B_a K_{\text{tma}} + D_a^2)} \\ K_{\text{com3}} &= \frac{K_{\text{tmL}}}{D_L K_{\text{qL}}}; \quad K_{\text{com4}} = \frac{B_L K_{\text{tmL}} + D_L^2}{G_s D_L K_{\text{qL}}} \\ T_1 &= \frac{J_a V_a}{4\beta_e (B_a K_{\text{tma}} + D_a^2)}; \quad T_2 = \frac{4\beta_e J_a K_{\text{tma}} + B_a V_a}{4\beta_e (B_a K_{\text{tma}} + D_a^2)} \\ T_{\text{com1}} &= \frac{J_L K_{\text{tmL}}}{B_L K_{\text{tmL}} + D_L^2}; \quad T_{\text{com2}} = \frac{J_L V_a K_{\text{tmL}}}{4\beta_e K_{\text{tma}} (B_L K_{\text{tmL}} + D_L^2)} \\ T_{\text{com3}} &= \frac{4\beta_e J_L K_{\text{tma}} K_{\text{tmL}} + V_a B_L K_{\text{tmL}} + V_a D_L^2}{4\beta_e K_{\text{tma}} (B_L K_{\text{tmL}} + D_L^2)}; \quad T_{\text{com4}} = \frac{J_L K_{\text{tmL}}}{G_s D_L K_{\text{qL}}} \end{aligned} \quad (53)$$

All compensation parameters can be calculated with structural parameters from Eq. (53). But it is usually challenging to obtain all accurate structural parameters of the load simulator and actuator system. Therefore, the determination rules of compensation parameter are proposed, which does not rely on accurate information of structure parameters. According to the actual physical meaning of each compensation term from Remark 3 and engineering practice experience, the rules could be divided into seven steps:

- Step 1: Determine the compensation item $G_{\text{com5}}(s)$. Under the static load conditions (without motion disturbances situation), we can determine the parameters of the PID controller according to the Ziegler Nichols setting method.
- Step 2: Determine the compensation item G_{com3} . Under the *static load conditions* (without motion disturbances situation), the loading system tracks sinusoidal torque command with large amplitude. The torque compensation gain G_{com3} can be determined by using try-and-error method.
- Step 3: Determine the synchronizing compensation gain K_{com1} in G_{com1} . When the actuator tracks sinusoidal angle command with large amplitude and low frequency, loading system tracks 0 Nm torque command. In this case, torque errors

are mainly caused by motion disturbances. Therefore, the velocity compensation gain K_{com1} can be determined with try-and-error method. On the other hand, just like Fig. 5, another standard, judging whether K_{com1} is optimal, could be whether the error between u_L and u_1 is smallest.

- Step 4: Determine the synchronizing compensation dynamic parameter in $G_{\text{com1}}(s)$. When the actuator tracks sinusoidal angle command with large amplitude and high frequency, loading system tracks 0 Nm torque command. The compensation dynamic parameter in $G_{\text{com1}}(s)$ can thus be determined with try-and-error method. $G_{\text{com1}}(s)$ only obtains phase-lag effects, which is easily achieved. On the other hand, as indicated in Fig. 5, an alternative way to judge the optimization of $G_{\text{com1}}(s)$ is based on the error between u_L and u_1 .

- Step 5: Determine the compensation gain K_{com2} in $G_{\text{com2}}(s)$. When the actuator tracks sinusoidal angle command with large amplitude and low frequency, loading system tracks the sinusoidal torque command with large amplitude and low frequency. With try-and-error method based on torque tracking error e , determine the velocity compensation gain K_{com2} . An alternative way to judge the optimization of K_{com2} is based on the error between u_L and $(u_1 + u_2 + u_3)$.

- Step 6: Determine the synchronizing compensation dynamic parameter in $G_{\text{com2}}(s)$. When the actuator tracks sinusoidal angle command with large amplitude and low frequency, loading system tracks the sinusoidal torque command with large amplitude and high-frequency. With try-and-error method based on torque tracking error e , determine the velocity compensation dynamical parameter in $G_{\text{com2}}(s)$. An alternative way to judge the optimization of $G_{\text{com2}}(s)$ is based on the error between u_L and $(u_1 + u_2 + u_3)$.

- Step 7: Determine the compensation item $G_{\text{com4}}(s)$. When the actuator tracks sinusoidal angle command with large amplitude and high frequency, loading system tracks the sinusoidal torque command with large amplitude and high frequency. With try and error method based on torque tracking error e , determine the compensation item $G_{\text{com4}}(s)$. An alternative way to judge the optimization of $G_{\text{com4}}(s)$ is based on the error between u_L and $(u_1 + u_2 + u_3 + u_4)$.

7 Experiment Verification

7.1 Experimental Test Rig. To validate the proposed controller, a platform is set up. As shown in Fig. 11, it consists of bench, load channels (hydraulic rotary motor, torque sensor, servo-valve, and connecting joint), hydraulic supply, and measurement and control system. The EHLS motor is the loading system, whose maximum dynamic torque is 650 Nm. The actuator motor is the positioning system used to produce motion disturbances. Table 1 shows the parameters of the main components.

7.2 Tracking Performance Under Typical Working Conditions. To evaluate the performance of the proposed controller, experiments under typical working conditions are conducted, which can test all the required performance.

The typical working conditions mainly include four cases: (1) static loading, (2) gradient loading, (3) dynamic loading with different frequencies between the EHLS and positioning system, and (4) dynamic loading with arbitrary motion disturbances. The static loading is used to test the actuator's rigidity, and present the tracking performance of the EHLS without motion disturbances. The gradient loading is most commonly used for the test of actuator systems. It can also indicate the synthetical abilities of the EHLS to track desired torque with motion disturbances at the same frequency. The dynamic loading with different frequencies or arbitrary wave motion disturbances can present the EHLS's ability to track random loading under arbitrary motion disturbances.

Fig. 11 Experimental platform

Table 1 Key experimental parameters of the test system

Components	Specification	Num.
Hydraulic motor	58 cm ³ /rad Range: ± 35 deg	2
Servo-valve	MOOG Inc., 19 l/min D765-SHR	2
Torque sensor	± 650 Nm/ ± 9 V $\pm 0.3\%$	2
Digital encoder	HEIDENHAIN ECN413 20"	1
Industrial computer	IEI WS-855GS	1
A/D card	PCI-1716/16 250 KS/s	1
D/A card	PCI-1723/8 16 bit DAC	1
Oil system	P_s : 18 Mpa Maximum flow rate: 120 l/min	1

To verify the tracking performance and the ability of velocity synchronizing compensation, three control strategies are employed for comparison. The first one is non compensation strategy, i.e., only a conventional PID controller is used. The second one is TVSM [7]. And the last one is the new compound velocity synchronizing controller based on Eq. (48) whose control parameters are determined by the rules in Sec. 6. All PID controllers have the same tuning parameters. In most hardware-in-the-loop experiments, the EHLS cannot obtain any information about position/velocity/acceleration of the real actuators. So we will not compare it with the structure invariance control method, which must require a velocity signal of the real actuator.

7.2.1 Experimental Case 1: Static Loading. Figure 12 illustrates the comparison of torque errors e of three control strategies under sinusoidal tracking torque with amplitude of 400 Nm and frequency of 3 Hz, when the actuator system tracks 0 deg angle command. It shows that the maximum tracking errors are 25 Nm (3.8%F.S.), 36 Nm (5.5%F.S.), and 17 Nm (2.6%F.S.) for PID control, TVSM, and the proposed controller, respectively. "F.S." is the abbreviation of "full scale."

Figure 13 shows the comparison of servo-output voltage. The closed-loop output u_c equals the difference between total output u_L and compensation output u_{com} .

This experiment also indicates that the TVSM is worse than the PID controller under static working conditions, because the TVSM provides completely opposite direction compensation, as shown in Fig. 13. The proposed controller can solve this problem, which means the new controller broadens the applied range of velocity synchronizing controller.

7.2.2 Experimental Case 2: Gradient Loading. Gradient loading means that the loading torque command is proportional to the actuator's position. Figures 14 and 15 show, respectively, the position command and the comparison of torque errors of three control strategies, when the actuator generates sinusoidal movements (10 deg amplitude, 1 Hz), and the tracking gradient of loading system is 40 Nm/deg.

The results show that the maximum tracking errors are 34.5 Nm (5.3%F.S.), 19.5 Nm (3%F.S.), and 11.5 Nm (1.7%F.S.) for PID control, TVSM, and the proposed controller, respectively. Figure 16 shows the comparison of servo control output among three control strategies. For TVSM, the closed-loop output, the differ-

Fig. 12 Comparison of e in static loading situation. (a) PID controller without compensation, (b) TVSM, and (c) the new compound velocity synchronizing controller.

ence between total output and compensation output, is more than 65% of the total control output. But for the new velocity controller, the total output is approximately equal to the compensation output, which means that it can better achieve feed-forward compensation effect and the closed-loop output based on torque errors is relatively small.

To further analyze the impact of motion disturbances on the control performance, the actuator produces high-speed (10 deg at 1.5 Hz), medium-speed (10 deg at 1 Hz), and low-speed command

Fig. 13 Comparison of servo output in static loading situation

Fig. 14 Angle command in gradient loading situation

(10deg at 0.5Hz). The EHLS tracks load gradient from 0 to 60 Nm/deg. Fig. 17 shows the comparison of torque errors among three control strategies.

Fig. 17 indicates that under small load tracking, both TVSM and the proposed controller can eliminate motion disturbances. However, under large or medium load tracking, the performance of the proposed controller is better than TVSM. It can eliminate the surplus torque level by 97%F.S., and dynamic loading accuracy achieves 3%F.S. under arbitrary gradient loading with motion disturbances.

7.2.3 Experimental Case 3: Dynamical Loading at Different Frequencies. In order to further validate the control strategy, experiments are conducted to make actuator and E follow commands at different frequencies. The actuator generates sinusoidal motions (1 deg amplitude, 5 Hz). The torque command is 500 Nm at 1 Hz. Figures 18 and 19 show the angle command and the comparison of e of three control strategies, respectively.

Figure 19 shows that the maximum tracking errors are 42 Nm (6.4%F.S.), 23 Nm (3.5%F.S.), and 15 Nm (2.3%F.S.) for PID control, TVSM, and the proposed controller, respectively. Figure 20 presents the comparison of servo output of these three control strategies. For TVSM, the compensation output cannot offer valid feed-forward compensation. But for the proposed velocity controller, the total output is approximately equal to the compensation output, which means that the compensation output can provide valid feed-forward compensation and the torque error is in a relatively low level.

Fig. 15 Comparison of e in gradient loading situation. (a) PID controller without compensation, (b) TVSM, and (c) the new compound velocity synchronizing controller.

7.2.4 Experimental Case 4: Dynamic Loading With Arbitrary Motion Disturbances. Experiments are also carried out on arbitrary motion disturbances, i.e., making actuator and loading system following different wave commands. The actuator generates triangular wave of 5 deg amplitude and 1 Hz frequency. The EHLS tracks sinusoidal torque of 500 Nm amplitude and frequency of 1 Hz. Figures 21 and 22 show the angle command and the comparison of e of the three control strategies, respectively.

The maximum tracking error are 43 Nm (6.6%F.S.), 28 Nm (4.3%F.S.), and 16 Nm (2.5%F.S.) for PID controller, TVSM, and the proposed controller, respectively.

Fig. 16 Comparison of servo output in gradient loading situation

Fig. 17 Comparison of e for various loading conditions

Fig. 18 Angle command in loading at different frequencies

Fig. 19 Comparison of e in loading at different frequencies. (a) PID controller without compensation, (b) TVSM, and (c) the new compound velocity synchronizing controller.

Fig. 20 Comparison of servo output in loading at different frequencies

Fig. 21 Angle command

(a)

(b)

(c)

Fig. 22 Comparison of e with arbitrary motion disturbances. (a) PID controller without compensation, (b) TVSM, and (c) the new compound velocity synchronizing controller.

Fig. 23 Comparison of servo output with arbitrary motion disturbances

Figure 23 presents the servo output comparison of three control strategies. For TVSM, the compensation output cannot offer valid feed-forward compensation. But for the proposed velocity controller, the total output is approximately equal to the compensation output, which means that the compensation output can provide valid feed-forward compensation and the torque error is low.

The above four experiments on typical conditions shows that the new controller can greatly improve the EHLS's anti-interference ability and dynamic loading precision. It does not require high-quality velocity, acceleration, and differential pressure signal. Its feed-forward compensation can constitute the main part of the total servo output and can eliminate the surplus torque by 97%F.S. and make dynamic loading accuracy achieve 3%F.S. under usual working conditions.

8 Conclusions

To improve the ability of antimotion disturbances, a compound velocity synchronizing strategy is proposed in this paper. It has the following characteristics:

- (1) It only needs u_a and T to eliminate the strong motion disturbances. No additional sensors (e.g., pressure drop sensor, high quality angle sensor, velocity sensor, acceleration sensor) are needed. It can better implement velocity compensation than the structure invariance method in high-frequency motion disturbances, and does not need angle, velocity, and acceleration signal. On the other hand, compared with TVSM, it can better realize velocity synchronization under large and medium loading situations.
- (2) It works under various conditions including static loading, gradient loading with velocity disturbances, dynamic loading with different frequencies, and dynamic loading with different waves. It can eliminate the surplus torque by 97%F.S. and make dynamic loading accuracy achieve 3%F.S. under typical working conditions.
- (3) It reduces the burden of the torque closed-loop controller and provides a sufficient margin for the controller to respond to varieties of sudden motion disturbances.
- (4) An EHLS is a typical case of torque systems strongly coupled with a hydraulic positioning system. It can be referred for other electro-hydraulic servo synchronizing systems.

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