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Development of Hydraulically Driven Fatigue Testing Machine for Insulators

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ABSTRACT The hydraulically driven fatigue testing machine is studied to simulate the vibration in the fatigue experiments for the insulators. First, hydraulic scheme and control principle are introduced. A special composite cylinder is designed, which is controlled by a servo valve and a proportional relief valve to generate static and dynamic tensional force. Then, models of electro-hydraulic subsystems are analyzed. For the actual needs of the dynamic tensional force system, a high-fidelity force source is of great significance. Considering parameter uncertainty and a high bandwidth of the servo valve pressure dynamics, there is a need to improve the bandwidth of the flow type servo valve as far as possible. Taking into account the limitations of general approaches, this paper proposes load velocity feedback to solve the challenging control issues, which mainly solves the problem of response bandwidth. Finally, a new system controller is designed, and some experiments are developed to verify the feasibility of system principle and the effectiveness of control algorithm. The hydraulically driven fatigue testing machine based on the discussed electro-hydraulic technology has been developed, with satisfactory technical specifications.

INDEX TERMS Insulators, vibration, electro-hydraulic control, fatigue test, response bandwidth, load velocity.

I. INTRODUCTION

The insulator is a kind of special insulation device, widely used in overhead transmission lines. Due to the aerodynamic disturbance of winds, the lines are subject to various kinds of vibrations, among which the aeolian vibration caused by steady laminar flow wind happens most frequently [1], with high vibrating frequency and small amplitude. The insulators suffer from different kinds of forces, which may greatly affect the operating life of the insulators. Therefore, the fatigue issue is paid great attention during design, manufacturing and inspection stages of the insulators. The fatigue test is an important experimental approach to simulate the vibration in the accelerated condition and to research vibration performance and fatigue characteristics of the insulators, and in many countries, such as China, Japan, India etc, the fatigue test for the insulators are compulsively required, especially for new models [2].

The fatigue testing machine, as an important experimental apparatus, is widely used in engineering research to produce periodical or random mechanical movement for structural components so as to measure their fatigue limit and operating life [3]. For the insulators, the fatigue test machine is usually driven by hydraulic actuator or electric-motor to obtain different kinds of vibration forces [4]. Compared with other types of drives, a hydraulic drive has many distinct advantages, such as high power-mass ratio and stiffness, fast response, high control precision, compactness and high payload capability [5]. Driven by technology development of the insulators, the fatigue testing machine with higher vibration power and control performance are demandingly required. Therefore, the hydraulic drive, especially electro-hydraulic drive, has been widely applied in the fatigue test for the insulators [6].

The present study comes from a research project of fatigue testing machine for new types of composite insulators, which have advantages of small mass, high stiffness and long operating life [7]. The special characteristics of such insulators have put forward higher requirements for the fatigue test, with technical specifications of 150kN maximum tensioning force, 15° maximum torsion angle, 100Hz maximum vibration frequency, 1kN amplitude error and more than 72 hours continuous duration. Therefore, the electro-hydraulic control technology is applied to such fatigue testing machine to meet demanding requirements in drive power and control performance.

However, the dynamic performance of control valve is often influenced by supply pressure, fluid and ambient temperature, and so on. Apart from the nonlinear nature of hydraulic dynamics, the performance of the electro-hydraulic force control system doesn't only depend on the characteristics of valve drivers, but also depend on the load stiffness and other load characteristics. Therefore, it is a more challenging issue for the control of such hydraulically driven fatigue testing system [8].

Conventional proportional-integral-derivative (PID) control has been widely used in hydraulic system for its simplicity, clear functionality and easy implementation [9], but it is difficult to achieve satisfactory control performance because of the inherent frequency characteristic of the tensional force system. Therefore new researches are mainly focused on some frequency domain analysis methods of system mathematical model, which is designed to change the shear frequency in order to achieve a desired performance.

The adaptive control method is equally a research hotspot [10], [11], employing adaptation laws to compensate for uncertain parameters. For the mismatched modeling uncertainties and unmeasured system states of hydraulic servo system, an extended state observer and a nonlinear robust controller are synthesized via the back-stepping method [12]. Other control strategies, such as back stepping control [13], feedback linearization control [14], fuzzy PID control [15], robust integral control [16], quantitative feedback control [17], neural network control [18], etc, are also applied in hydraulic control systems, but their limitations couldn't be ignored. For example, complicated calculation may be required, which cannot meet the real-time requirement in motion control systems.

Since the development of early robotic technology, robot force control was of fundamental interest [19]. The force control includes many different forms, such as model-based control, virtual model control [20], operational space control [21] and so on. Researches on the force-controlled system find that advanced control approaches, such as the model-based control method, can be successfully applied in the low-level force control to reach a good tracking performance without having to give up on the bandwidth [22]. Considering the load characteristics, load motion compensation method was initially discussed for hydraulic actuators in [23], which is expected to provide a feasible solution for control issue to the dynamic tensional force system.

In this paper, the hydraulically driven fatigue testing machine is developed for the insulator fatigue test. A special composite cylinder controlled by servo valve and proportional relief valve is designed, to generate static and dynamic tensional force. Load velocity feedback has been proposed to achieve high-precision control performance. The rest of the paper is organized as follows. Section 2 introduces system principle of fatigue testing machine for the insulators, including analysis of loading forces, hydraulic control principles. In Section 3, models of electro-hydraulic subsystems are derived, and their characteristics are also analyzed. In Section 4, the fundamental principle of load velocity feedback is introduced, which the PI controller and the load velocity compensator are designed for electro-hydraulic control process in the fatigue testing machine. In Section 5, the physical fatigue testing machine for the insulators is described, and the experimental results are also presented. Lastly, conclusions are made to summarize the main points of this paper.

FIGURE 1. Force analysis diagram of the insulator.

II. TEST PRINCIPLES FOR THE INSULATORS

A. ANALYSIS FOR LOADING FORCES

Actual loading forces endured by the insulator are very complicated, but according to force direction, they can be simplified to tensional force and torsional force, which can be described by Fig.1. The tensional force behaves continuously, which may reach to more than 15 tons and produce small tensile deformation of the insulator. The torsional force with small amplitude happens to the insulator intermittently, causing considerable torsional deformation to the insulator. Considering the tensional force is the main factor affecting the operating life of the insulator, it is paid more attention to in the following discussion.

According to the varying speed of the controlled force, the tensional force can be decomposed to static force and dynamic force. The static force caused by the weight of the overhead conductors which are connected with the insulators, normally keeps unchangeable or changes slowly, but with larger amplitude. Compared with the static force, the dynamic force is often with small amplitude, less than one fifth of the static force, but with rapid alternating frequency.

Fig.2 describes actual vibrating curve of the tensional force, which can be expanded to a serial of sine waves and constant offset. Therefore, after some simplifications, the tensional force F_{ten} can be expressed by

$$F_{ten} = A_s + A_d \sin(2\pi f_d t), \quad A_s \geq 5A_d \tag{1}$$

Where A_s is amplitude of static force, and A_d and f_d are amplitude and frequency of dynamic force respectively, and A_s is much bigger than A_d numerically.

The torsional force F_{tor} is caused by slow swing movement of transmission lines, and similarly it also can be described by

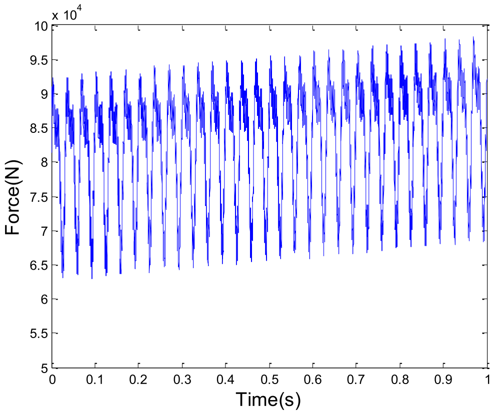


FIGURE 2. Actual curve of the tensional force of the insulator.

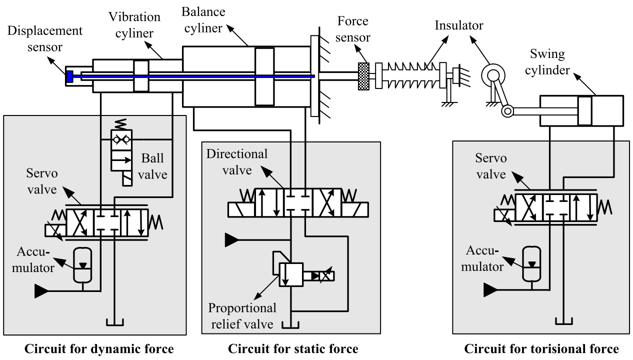


FIGURE 3. Hydraulic principle for the insulator fatigue test.

sine regularity, as shown by

$$F_{tor} = A_t \sin(2\pi f_t t) \tag{2}$$

where A_t and f_t are amplitude and frequency of torsional force respectively.

B. HYDRAULIC CONTROL PRINCIPLE

A compound electro-hydraulic loading principle for the servo valve and proportional valve has been designed and illustrated integrally in [5], so an improved hydraulic principle for the insulator fatigue test is developed by Fig.3 on this basis, including a special composite cylinder, a swing cylinder and corresponding hydraulic control circuit. According to the control object, the hydraulic circuit can be divided to three parts, circuit for dynamic tensional force, circuit for static tensional force and circuit for torsional force.

Then according to presented hydraulic principle, the corresponding control procedure can be divided into four steps, initial connection, static tensional loading, dynamic tensional loading and torsional loading. The static and dynamic tensional loading procedures are shown in Fig.4. Firstly, the proportional hydraulic circuit is used to gradually adjust static load force in static loading process with the servo circuit in a dormant state. Then when the static load error is less than a certain threshold, the power of electromagnetic valve is removed which corresponds to close the switch K, so the

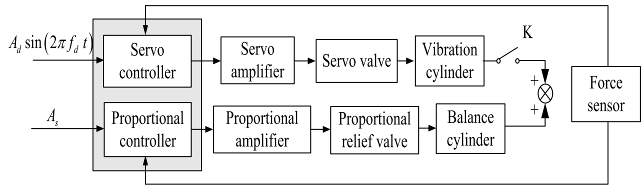


FIGURE 4. Principle diagram of tensioning force control.

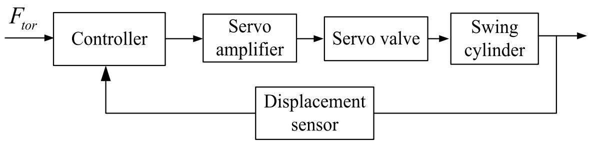


FIGURE 5. Principle diagram of torsional force control.

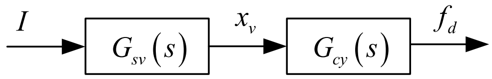


FIGURE 6. Model block diagram of two-stage servo valve controlled symmetric cylinder.

servo loading circuit composed of the servo valve and the vibration cylinder starts to produce the dynamic load force according to the setting rule in dynamic loading process, simultaneously, the proportional circuit is controlled with a low gain to prevent the static force from drifting under the disturbance of the dynamic loading procedure.

At the same time, the torsional force is produced by a swing cylinder, connecting one extremity of tested insulator with rocker. The swing cylinder is controlled by a servo valve, which forms a conventional position control system. And within a certain angle range, the torsional force has a linear relationship with the torsional angle approximately, and therefore, the torsional force control is essentially a typical position control, regarding torsional angle as control object, then its control principle can be expressed by Fig.5.

Generally, the dynamic loading progress is the key to ensure the dynamic characteristics and load accuracy of the whole system. So this paper mainly focuses on the control method of dynamic loading circuit.

III. MODEL ANALYSIS OF HYDRAULIC SYSTEM

A. MODEL OF DYNAMIC TENSIONAL FORCE

The subsystem for dynamic tensional force in the fatigue test machine is a typical force servo system containing a symmetric servo cylinder (vibration cylinder) and a two-stage servo valve in dual nozzle flapper. Therefore, the model block of this subsystem can be described by Fig.6, where $G_{sv}(s)$ is transfer function of servo valve with current I as input and spool displacement x_v as output, and $G_{cy}(s)$ is transfer function of spool valve controlled symmetric cylinder with spool displacement x_v as input and dynamic force f_d as output.

Then according to the classical flow linear, flow continuity and force balance equation, models of the subsystems can be

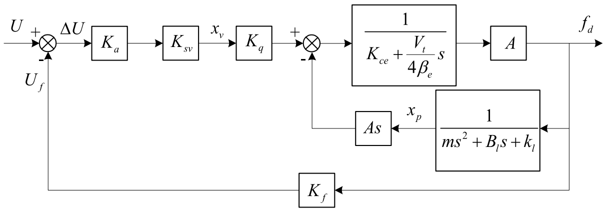


FIGURE 7. **The block diagram of the electro-hydraulic servo force control system.**

get to made a great help on the deduction process of the whole block diagram of the electro-hydraulic servo force control system, which can be shown in Fig. 7, where U is the input for the control command, f_d is the output of the loading force, x_p is the load displacement, x_v is the servo valve spool displacement, K_a is gain amplifier, K_{sv} is a gain without considering servo valve dynamics, K_q is the servo valve flow gain, K_f is the force sensor gain, A is the effective area of the small cylinder, V_t is effective volume of the small cylinder, K_{ce} is leakage coefficient, β_e is hydraulic bulk modulus, m is the mass of moving parts, B_l is load damping coefficient and k_l is the load elastic coefficient.

The open loop transfer function of the force control system is shown by

$$\frac{f_d(s)}{U(s)} = \frac{\frac{K_f K_a K_{sv} K_q A}{K_{ce}}\left(\frac{m}{k_l}s^2+\frac{B_l}{k_l}s+1\right)}{\frac{mV_t s^3}{4\beta_e K_{ce} k_l}+\left(\frac{V_t B_l}{4\beta_e K_{ce} k_l}+\frac{m}{k_l}\right)s^2+\left(\frac{V_t}{4\beta_e K_{ce}}+\frac{B_l}{k_l}+\frac{A^2}{K_{ce} k_l}\right)s+1}. \tag{3}$$

Then it can be transformed to

$$\frac{f_d(s)}{U(s)} = \frac{K_v\left(\frac{s^2}{\omega_m^2}+\frac{2\xi_m}{\omega_m}s+1\right)}{(\frac{s}{\omega_r}+1)\left(\frac{s^2}{\omega_0^2}+\frac{2\xi_0}{\omega_0}s+1\right)} \tag{4}$$

where ω_m is the inherent mechanical frequency, ω_r is the turning frequency, ω_0 is the natural frequency of the hydraulic spring and the mechanical spring, ξ_m is the mechanical damping ratio, ξ_0 is the overall damping ratio of the hydraulics and mechanics, $K_\varpi = K_\phi K_\alpha K_{\sigma\varpi} K_\theta A/K_{\chi\varepsilon}$ is the system whole open loop gain.

At this time, a simulation model can be used to get the system frequency response characteristic in Simulink when $K_v = 30$, $\omega_m = 90$ rad/s, $\omega_0 = 350$ rad/s, $\omega_r = 1$ rad/s, $\xi_m = 0.15$, $\xi_0 = 0.1$.

From the open-loop transfer function in (4), it can be found that the subsystem for dynamic tensional force is a O-type system, and through the frequency characteristics analysis in Fig. 8, the amplitude margin of the open-loop system is at a relatively high level yet with the shear frequency $\omega_c = 4.356$Hz. According to the three bands theory in automatic control principles, the bigger ω_c in the intermediate frequency band, the faster response speed of the servo hydraulic circuit.

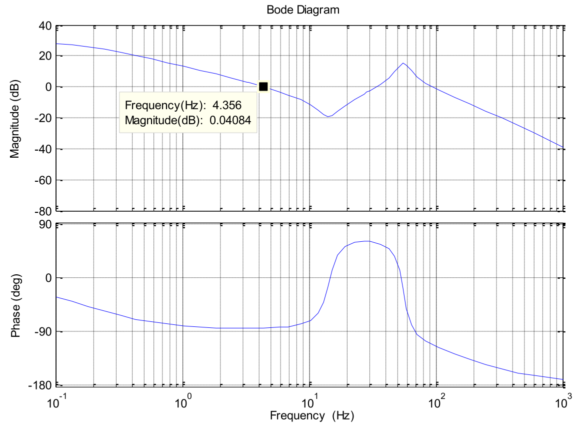


FIGURE 8. **Open-loop frequency characteristics of the system.**

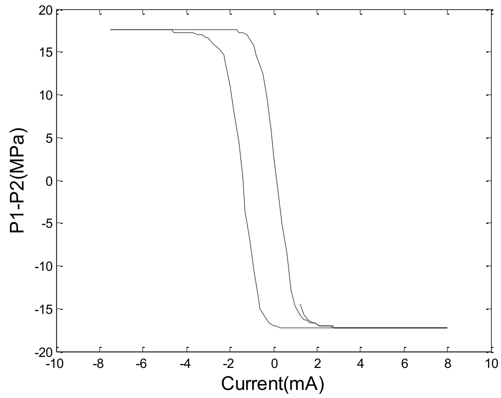


FIGURE 9. **Current-pressure curve of the servo valve.**

Because the shear frequency of the force control system is quite small in this paper, the whole dynamic characteristics are not ideal for control needs.

At the same time, the open-loop gain K_v includes the pressure gain of servo valve $K_p = K_q/K_{ce}$. The actual pressure characteristic curve of the servo valve applied in the fatigue testing machine is described by Fig.9 with a large value for a flow-type servo valve, therefore, the design for control gain should be conservative as to keep system from instability. As a result, the way that improves the system response by conventional PID feedback controller is restricted for such dynamic force control subsystem.

B. MODEL OF STATIC TENSIONAL FORCE CONTROL

In the fatigue testing machine, a pilot operated proportional relief valve is used to regulate static tensional force, which can control output pressure P proportionally according to input signal U_g. The actual static characteristic of the proportional relief valve used in the fatigue testing machine are shown by Fig.10 respectively, indicating that such valve is severely nonlinear control component, for the existence of hysteresis, dead-zone, saturation, nonlinearity and time-delay. Therefore, the proportional relief valve can just be used to generate static force with constant value or slow changing regularity.

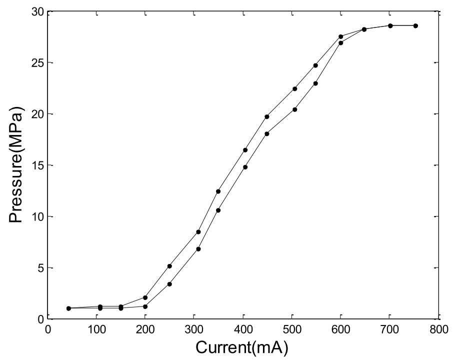


FIGURE 10. Static characteristic of the proportional relief valve.

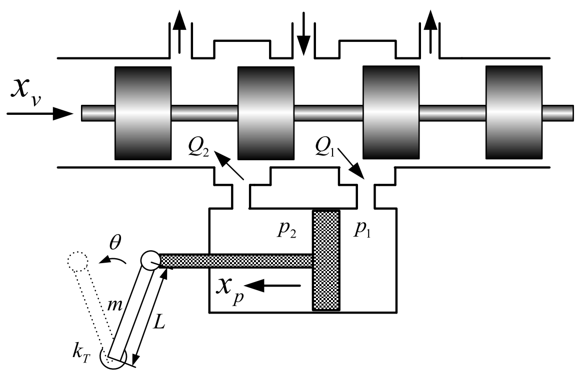


FIGURE 11. Principle diagram of servo valve controlled asymmetric cylinder.

The characteristic of the proportional relief valve has been analyzed and described by means of methods of classical control theory, but the derived model is complicated, which causes inconvenience to system analysis and simulation. Therefore, in present study the model of the proportional relief valve is simplified to a second order oscillation system with time-delay element, as shown by the following equation, where K_{pv} is pressure gain of valve, ω_{pv} and ξ_{pv} nature frequency and damping coefficient of valve:

$$G_{pv}(s) = \frac{P(s)}{U_g(s)} = \frac{K_{pv}e^{-\tau s}}{\frac{s^2}{\omega_{pv}^2} + \frac{2\xi_{pv}}{\omega_{pv}}s + 1}. \tag{5}$$

Due to bad dynamic characteristics of the proportional relief valve, as well as its server nonlinearity, the PID control is applied to complete stable control for the static control subsystem.

C. MODEL OF TORIONAL FORCE CONTROL

Regarding swing angle as control object, the subsystem for torsional force control is a typical position servo system, composing of asymmetric cylinder (the swing cylinder) and the two-stage servo valve. Fig.11 displays a principle diagram of servo valve controlled asymmetric cylinder, where servo valve is simplified to spool valve, damping loading is neglected, and elastic stress is the main loading.

In Fig.11, k_T is torsional elastic coefficient of tested insulator, m and L mass and length of swing arm, θ rotation angle of tested insulator. Within certain small angle range, the relationship between θ and x_p as well as the relationship between spool displacement x_v and input current I may be expressed by

$$\theta = x_p/L, \quad x_v = K_m I \tag{6}$$

where K_m is the gain of torque motor.

The transfer function between x_v and x_p can be expressed by [24]

$$x_p(s) = \frac{\frac{K_q A_p}{K_{ce} k_T} x_v(s)}{\left(\frac{s}{\omega_v} + 1\right)\left(\frac{s}{\omega_0^2} + \frac{2\zeta_0 s}{\omega_0} + 1\right)} \tag{7}$$

where ω_v, ω_0 and ξ_0 are given by

$$\omega_v = \frac{K_{ce} k_T}{A_p^2 (2k_T/k_h + 1)},$$
$$\omega_0 = 2\sqrt{\frac{\beta_e A_p^2}{m V_t}\left(1 + \frac{k_T}{k_h}\right)}$$
$$\xi_0 = \frac{1}{2\omega_0}\left(\frac{\beta_e K_{ce}}{V_t} - \omega_r\right). \tag{8}$$

The transfer function of torsional force control subsystem can be derived from (6) and (7), as shown by

$$G_{tor}(s) = \frac{\theta(s)}{I(s)} = \frac{\frac{K_q A_p}{K_{ce} k_T} L K_m x_v(s)}{\left(\frac{s}{\omega_v} + 1\right)\left(\frac{s}{\omega_0^2} + \frac{2\zeta_0 s}{\omega_0} + 1\right)}. \tag{9}$$

Equation (9) indicates that such position control subsystem under torsional elastic loading is also a special O-type system with poor stability, compared with common S-type position control system. Furthermore, because of the asymmetrical structure of the cylinder, many parameters in the discussed model, such as K_q, K_{ce}, A_p and ω_r, would change greatly when the rod moving in different directions, among these parameters, the variation of K_q has a significant influence on the dynamic performance of such system.

Because of asymmetry, nonlinearity and time-variation of the torsional force control subsystem, its high precision control becomes difficult. Therefore, a classical PID control scheme is also applied to figure out this challenging issue.

IV. DESIGN OF THE CONTROLLER FOR DYNAMIC LOADING PROGRESS

Generally, the load accuracy of dynamic loading progress is the core and difficulty to ensure the dynamic characteristics of the overall system. So this paper mainly focuses on the control method of dynamic circuit.

For the dynamic loading circuit, some general compensation methods are rather difficult to increase the system response speed. As a result, this paper uses load velocity feedback to eliminate dynamics of the load and improve tracking performance of the force tracking system. Through the compensator $G_b(s) = \frac{As}{K_q K_{sv}}$, an additional control signal $u' = \frac{A}{K_q K_{sv}}\dot{x}$ can be added to cancel out the influence of the

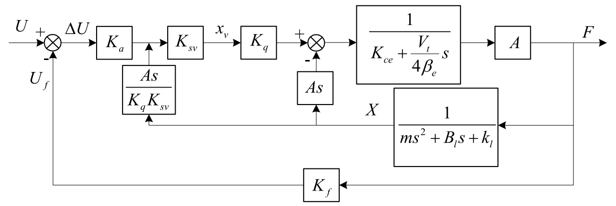


FIGURE 12. The block diagram of the force control system corrected by the structural invariance principle.

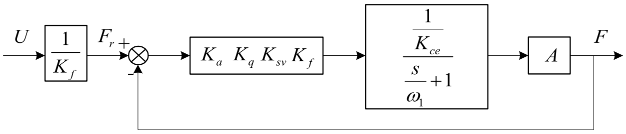


FIGURE 13. The block diagram of the dynamic loading circuit after compensated and simplification.

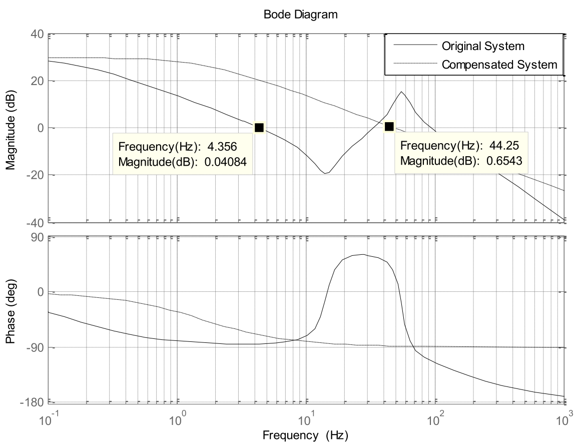


FIGURE 14. The block diagram of the dynamic loading circuit after compensated and simplification.

load stiffness and mass. After the velocity compensation is added to the system, the system block diagram is shown by Fig. 12 [8].

Then the Fig. 12 can be simplified to the block diagram in Fig. 13 with $\omega_1 = 4\beta_e K_{ce}/V_t$.

So the open loop transfer function of the compensated system is shown by

$$G(s) = \frac{\frac{K_f K_a K_{sv} K_q A}{K_{ce}}}{(\frac{s}{\omega_1}+1)} = \frac{K}{(\frac{s}{\omega_1}+1)}. \tag{10}$$

When $\omega_1 = 10$ rad/s, $K = 30$, the frequency characteristics of the compensated system can be compared with the original system in Fig. 14.

As shown in Fig. 14, the compensated system is similar to the standard inertial element, and as an important parameter of the system, the crossing frequency is improved from the original 4Hz to current 40Hz by the compensation method used in this paper, and the bandwidth is raised greatly at last. So as long as the bandwidth of the electro hydraulic servo valve is large enough, the shear frequency of the system can be raised by increasing the open-loop gain of the dynamic loading circuit, and it will not make the system unstable even though there is a very large pressure gain of valve in the circuit. Ultimately, the speed and control precision of the system can be significantly improved.

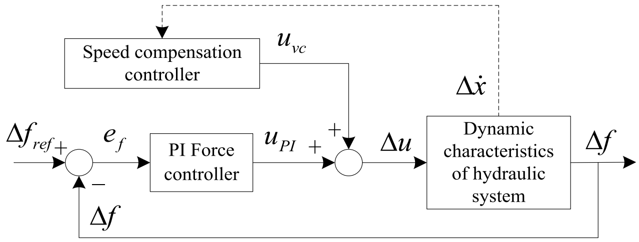


FIGURE 15. The principle of the whole force controller.

For the hydraulic control system in this paper, the dynamic characteristics of the system are mainly determined by the slowest component and lots of previous studies have proved that the valve spool dynamics is several times faster than the pressure and load motion dynamics [22]. Therefore, there is a good application significance that load motion compensation is the main factor that needs to be considered in this paper.

Then when the dynamic force control sub circuit is taken into account, the extra velocity compensation represents an extra flow that has to be supplied by the valve, and the extra flow is used to overcome the effect of friction and load movement. Since the conventional PID regulator has many advantages including mature technology, simple control structure and better control effect, this paper considers using the PID controller which is based on the principle of structural invariance to get a great control response. The principle of the whole controller is shown in Fig. 15 [25].

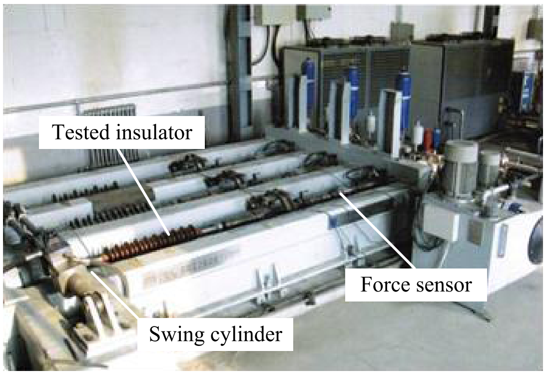
Considering the influence of the noise from the force sensor and the pressure sensor, the differential term may cause the system instability, so the feedback controller is simplified to a PI instead of the classic PID. Meantime, as shown in Fig. 15, a compensator is set up according to the velocity compensation principle and the final servo valve control signal Δu is the sum of u_{vc} and u_{PI}. Then by designing the parameters of the PI controller and the velocity compensator, a desired force control effect can be obtained.

V. EXPERIMENTS

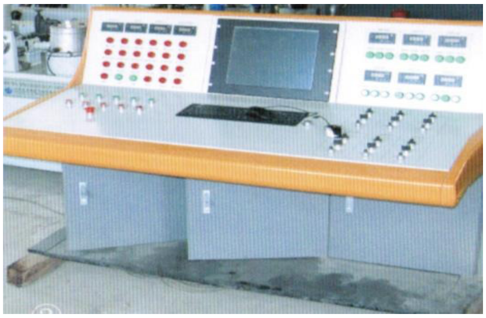
A. CONSTITUTE OF THE FATIGUE TEST MACHINE

According to the loading principle and control method in this paper, testing machine has been developed by Beijing Institute of Technology, including three parts physically, test bench, computer control & test system and hydraulic pump station, Fig. 16 displays physical photos of key components in this fatigue testing machine.

Fig. 16(a) describes the test bench with three channels, which can experiment three insulators simultaneously, and with different vibration parameters. Each channel is to generate independent tensioning force for the insulator, and these three channels share a swing cylinder control by servo valve for torsional loading.



(a)



(b)

(c)

FIGURE 16. Key components in this fatigue testing machine. (a) Test bench with three channels. (b) Computer control & test system. (c) Hydraulic pump station.

As shown in Fig. 17, each driving unit for tensional force is composed of a vibration cylinder, a balance cylinder, a servo valve, a proportional relief valve, a magnetic ball valve and the accumulator, and the specifications of these main components are listed in Tab. 1.

The control system uses the upper-lower computer system, which includes the load controller and the testing computers. The embedded load controller is to complete the close-loop control for tensional force, and torsional force control is executed by one embedded controller. The control system is based on PC104 bus with a 486-level 100MHz embedded CPU board and 16-bit A/D and D/A converters. And the discussed force control algorithm is carried out on the controller, with operation cycle less than 0.2ms, which is less than A/D.

The testing computer, which is an industrial control computer, is to set the force given value, sample data, record running time, show the test curve, analyze

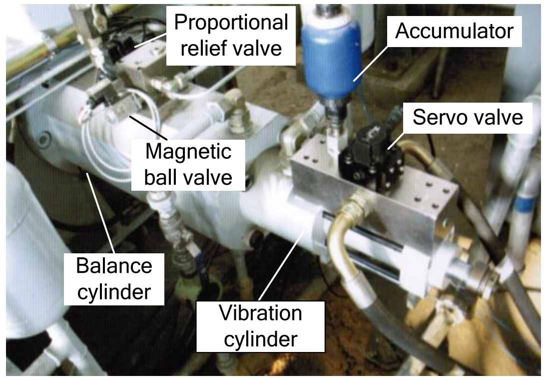


FIGURE 17. Photographs of a driving unit for tensioning force.

TABLE 1. Lists of main component in the compound actuator.

Component	Specifications	Producer
Balance cylinder	Piston diameter 100mm, rod diameter 63mm, stroke 400mm	Beijing Institute of Technology
Vibration cylinder	Piston diameter 63mm, rod diameter 36mm, stroke 50mm	
Servo valve	Rated flow 40L/min, rated current 40mA, frequency bandwidth 120Hz	Moog
Proportional relief valve	Rated pressure 35MPa, rated current 800mA	Rexroth
Accumulator	Operation pressure 35MPa, Nominal volume 2.5L	Hydac

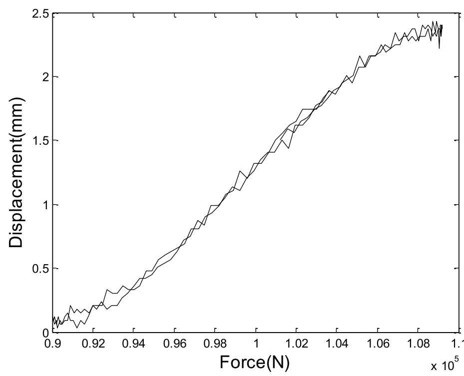


FIGURE 18. Tensional deformation of tested insulator under dynamic vibration force.

vibration performance and communicate with embedded controllers by Ethernet network and TCP/IP protocol.

The purpose of hydraulic pump station is to provide hydraulic flow to the test bench, with maximum pressure 31.5MPa and maximum flow 200L/Min. At the same time, a set of water cooling system is used to control oil temperature for long working hours in the hydraulic pump station.

B. EXPERIMENTAL RESULTS

In this section, a kind of the composite insulator with model FXBW-500/300 is tested by this fatigue test machine. The tested insulators do produce a certain degree of distortion by the tensioning force in Fig. 18. The deformation of the insulator is about 2.4mm when the tensioning force changes

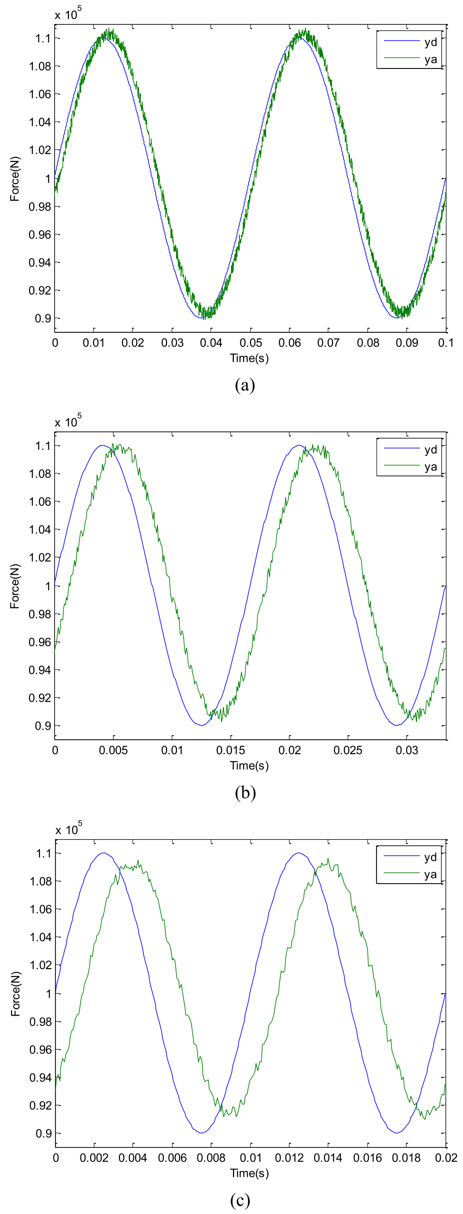


FIGURE 19. Force curves of fatigue testing machine at different vibrating frequencies. (a) 40Hz. (b) 60Hz. (c) 100Hz.

from 90kN to 110kN, and the elastic modulus of this insulator can be calculated as 83.3kN/mm. Although this tensioning deformation is very slight, it may cause adverse effects to dynamic tensional force control with a rather high frequency.

Firstly, experiment for the tensioning force is developed by this fatigue testing machine. Force control principle and method are experimented for further verification in the paper, and experiment results have verified practical applicability of the fatigue testing machine. Fig.19 gives the force curves of this fatigue testing machine at different vibrating frequencies, where y_d is an expected force curve defined by the following equation and y_a is the actual force curve:

$$y_d = 10 \sin(2\pi f t) + 100 \quad f = 40, 60, 100Hz \text{ unit} : kN. \tag{11}$$

TABLE 2. Amplitude and phase errors between expected and actual force at a serial of different vibrating frequencies.

$f\,/\,Hz$	$e_{amp}\,/\,N$	$e_{pha}\,/\,\circ$
20	40	-10.2
40	150	-20.5
60	360	-30.9
80	680	-41.5
100	1120	-52.1

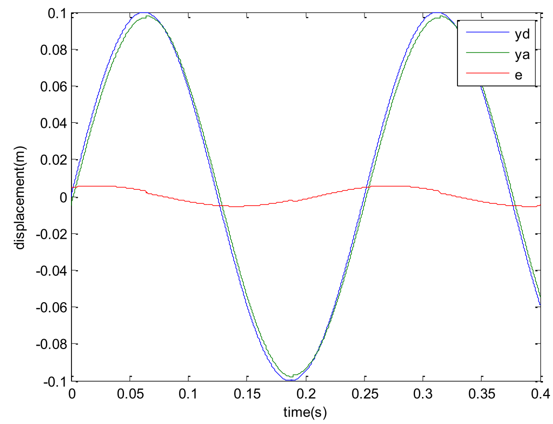


FIGURE 20. Experimental results of valve controlled asymmetrical cylinder for torsional force control.

Obviously, the actual force can track the expected force curve with acceptable errors, especially in amplitude. In engineering application, the amplitude error that is the difference between the amplitude of the expected curve and the actual curve is a main index for the testing machine. When vibrating frequency changes from 20Hz to 100Hz, amplitude errors between expected and actual force are listed in Tab.2. The phase errors are also given in Tab.2, which are not very important in actual applications. The experiment data indicates that amplitude and phase errors become larger with the frequency increase, but they are still tolerated within frequency 100Hz.

The torsional force control system, which is driven by valve controlled asymmetrical cylinder, is controlled by PID control method, and desired motion regularity can be expressed by

$$S = 0.1 \sin(8\pi t) \text{ unit} : m. \tag{12}$$

Experimental result of torsional force control is given by Fig.20, where y_d is expected displacement curve defined by (12), y_a is actual displacement curve and e is error curve. Experimental result demonstrates that the simple PID control can achieve ideal performance for torsional force control.

VI. CONCLUSIONS

In this paper, electro-hydraulic control technology is researched to develop the hydraulically driven fatigue testing machine for the insulators. Hydraulic part and control principles are illustrated, and the models of hydraulic subsystems are analyzed. A method based on the load velocity feedback is proposed for control issues. The experiments have verified the effectiveness of system principle and the control scheme.

Considering complexity of the tensional force endured by the insulators, a special composite cylinder is designed to generate static force with large amplitude and dynamic force with small amplitude but rather high frequency, and its excellent dynamic performance has been proved by multiple experiments.

The control subsystem for dynamic tensional force is an O-type system with a very high open-loop gain and uncertainty of parameters; therefore, through detailed analysis for the frequency characteristics, a typical model-based method is adopted to resolve the complicated control issues, and experiments proof that this load velocity compensation significantly enhances the force tracking capabilities.

Based on the discussed electro-hydraulic control principle, a hydraulically driven fatigue testing machine for the insulators has been developed, and its appearance and composition are also described in detail in this paper. This machine has been applied to the fatigue experiments of a serial of composite insulators in the laboratory of State Grid Corporation of China, and has achieved satisfactory performance, with multiple loading channels, 150KN maximum static tensioning force, 20KN maximum dynamic tensioning force, 0.5KN precision, torsional angle, 100Hz maximum vibration frequency, 72 hours consistent measurement duration.

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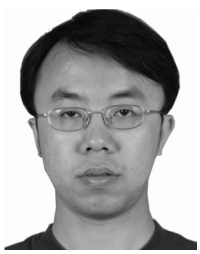

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