LOAD SIMULATOR BASED ON THE PRINCIPLE OF HYDROSTATIC SECONDARY CONTROL

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Abstract

A new concept for a load simulator based on the principle of hydrostatic secondary control is put forward, which are the characteristics of energy saving, energy recovery and four quadrant working. The main problems are the influence of the friction, power supply system and torque noise of itself. Some control schemas are put forward and related simulation work is carried out. Especially a fuzzy controller is designed and applied to overcome the friction. Proving there are better control performances. This principle can be used in swing loading and spinning loading systems. Its advantage is that there is no extraneous torque and energy saving comparing with traditional electric-hydraulic servo control load simulator. So it is very useful in high speed motion loading.

1 Introduction

The electro-hydraulic-servo controlled load simulator systems are widely used in aerospace, automobile, shipbuilding industry etc. which technique characteristics are plant complex, difficult to control, especially on velocity disturbance, Another problem is the larger energy exhausting. In most of the situation, the loading system is same as adding a resistance to the systems, that is equivalent to pump situation. But regard as to the traditional load simulator, This part of energy must be exhausted and consequently converted into thermal energy, which need a complex thermal dissipation system[6][7][8][9].

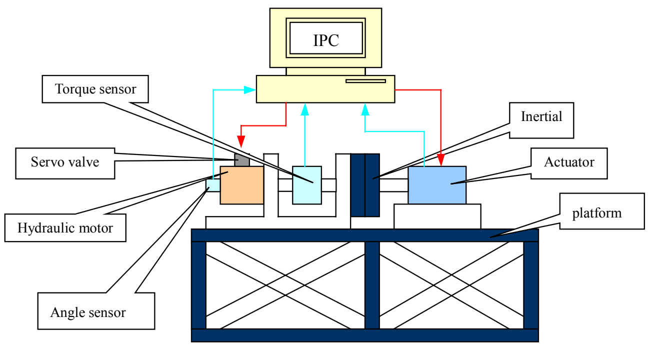


Fig.1 Loading system on the actuator

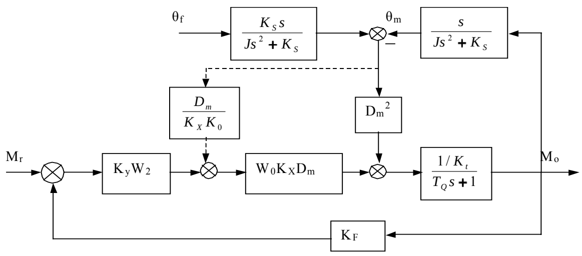


Fig .2 The transfer function of loading system with positioning disturbance

Fig.1 is a typical loading system on the actuator. The control block diagram is shown in Fig .2(See reference[7][11]).

Its characteristics are as follows:

- The mechanical natural frequency is the limit of loading system frequency band;
- The velocity disturbance makes a great extraneous force. It is not easy to be overcome;
- The power is related to the load gradient and frequency.

Dynamic performance and disturbance influence are two key problems in the design of loading system. The velocity disturbance can bring up extraneous force, which is related to the velocity and some uncertainty factors. An effective method is to add a feedforward link, see Fig.2, dot line, whereas the dynamic performance need to be improved by PID, adaptive control etc.

This paper presents a new concept load simulator system based on the hydrostatic secondary control principle[1][2][3][4][5]. In the rotating load system, replace the servo control motor with servo controlled variable displacement motor, realized the torque control by torque feedback. And connect all sub swashplate angle and system pressure. The torque, velocity and position control can be obtained by related feedback.

2 Fundamental Principle

Fig.3 illustrates the basic schema of the load simulator, see control block diagram in Fig.4. It is a kind of volume control. If it is in pump situation (that is to exert resistance), it can feedback some energy to whole system while exerting force. The energy can be made use of again and result in a simpler thermal dissipation system.

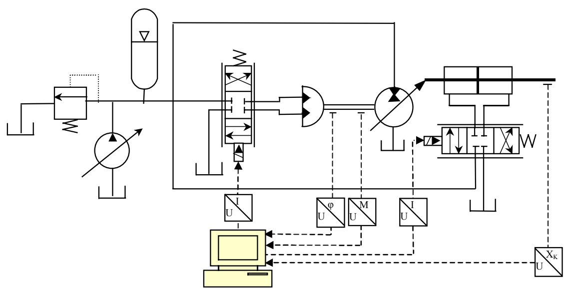


Fig.3 The Diagram of the load simulator based on the secondary control principle

Why does we use the torque feedback rather than swashplate angle feedback? Because there are the disturbances from power supply unit

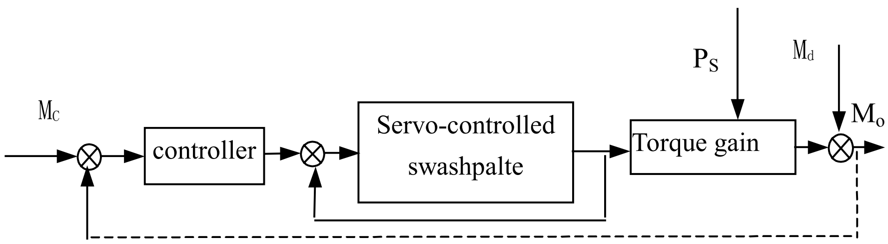


Fig.3 The servo control principle of hydrostatic secondary control

hydraulic system into a same constant pressure supply system, in order to make use of the energy for the best.

Fig.3 illustrates the control block diagram of the secondary control system. The open loop output is the torque, which is determined by and torque contamination from itself and friction. Only if put it into closed loop, it is possible to reduce the influence of them. These disturbances are mainly affects the forward gain of the system. Therefore, there need a torque sensor to mounted on the output axial of the motor. The paper put forward different compensating method according to different applications. There are two kinds of application.

- Swing loading system. The swing angle less than 180°. The motion frequency is not very high, less than a couple of Hz, but must trace the load spectrum accurately. Its key problem is robustness to the disturbance of power supply unit and torque noise itself;
- Spinning loading system. Loading while motor is in high speed rotation. There is a high frequency and low amplitude torque vibration which is related to the speed and the number of the pistons. In generally the speed is constant or changes slowly.

3 Modeling, Simulation and Analysis

3.1 Modeling

Fig.5 is the transfer function block diagram of the load simulator. There are two influence factors, one is pressure of power supply unit, another is torque noise itself, which related to the structure of motor. If discard the influence of rotating angle φ of the motor to the torque, $f_m = 1$, otherwise:

$$f_m = 1 + \begin{cases} \cos(\phi - \pi/2z), & 0 \le \phi \le \pi/z \\ \cos(\phi - 3\pi/2z), & \pi/z \le \phi \le 2\pi/z \end{cases} \tag{1}$$

This is a part-cosine function factor with cycle π/z, which influences the gain of the systems. z is the motor piston number.

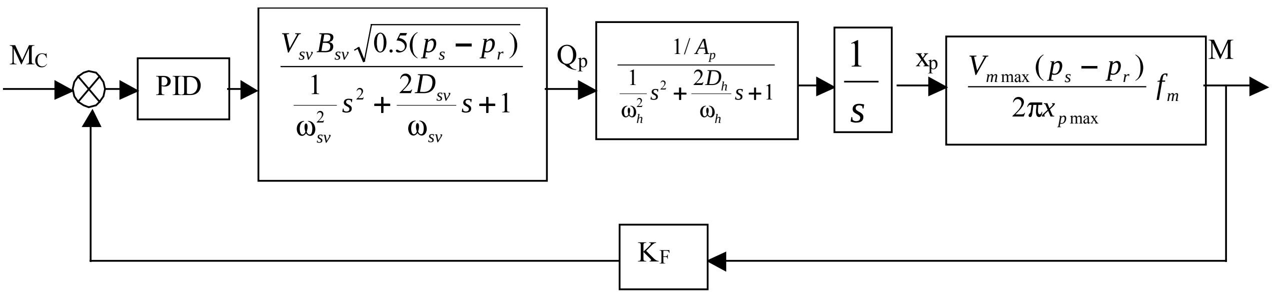


Fig.5 Transfer function diagram of the load simulator based on the secondary control

Where

V_{sv} the static gain of servo-valve(m/V);
B_{sv} flow coefficient of ervo-valve($m^3/sN^{1/2}$);
P_r return pressure(Pa);
P_S supply pressure(Pa);
ω_{sv} the natural frequency of ervo-valve(rad/s);
D_{sv} damping factor of the servo-valve;
ω_h the natural frequency of dynamics(rad/s);
D_h damping factor of the hydraulic dynamics;
A_p aera of the swashplate operating piston(m^2); V_{mmax} maximum instantaneous motor displacement(m^3/rev);
x_{pmax} maximum position of the swashplate operating position(m);
K_F feedback factor(V/Nm).

Another problem is that the output torque is related to the system pressure which is influenced by flowrate (Refer to P-Q curve of the Fig.7). And in the process of exerting force, the flowrate requirement is varing occasionally.

In order to study this effective, the simple modeling of power supply unit is suggested as follows. See Fig.6, the constant pressure is adjusted by relief valve (similar modeling with variable pump). V is the control volume between pump and motor. The accumulator can be considered in as an equivalent liquid compactor approximately. The equation is

$$\frac{dp}{dt} = \frac{E_y}{V}(Q_S - Q_L) \tag{2}$$

Fig.7 is the P-Q characteristics of the constant pressure power supply unit. In the period B, pressure and flowrate meet the relation of rate K. and assuming dynamic characteristic is a typical second order system：

$$\frac{Q_S}{P_S} = -\frac{K}{s^2 + 2\xi\omega_h s + \omega_h^2} \tag{3}$$

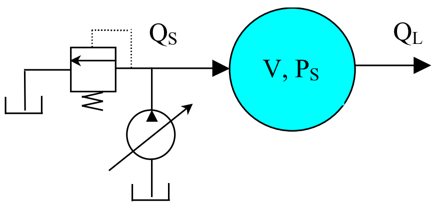


Fig.6 The Power Supply System(PSU)

E_y is oil bulk module, about 10000baror so. E_y/V is equivalent to liquid compactor. The accumulator can be accounted as some liquid compactor values. K is the rate of P-Q, ω_h and ξ are the natural frequency and dump of the second order control system.

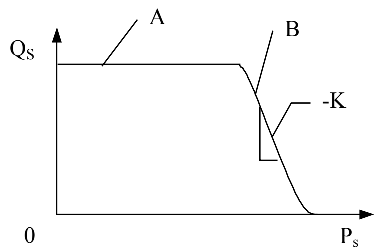


Fig.7 The pressure adjustment curve of PSU

Therefore, the transfer function of PSU is:

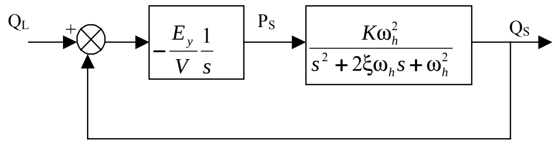


Fig.8 The transfer function of PSU

The combination model is as Fig.9.

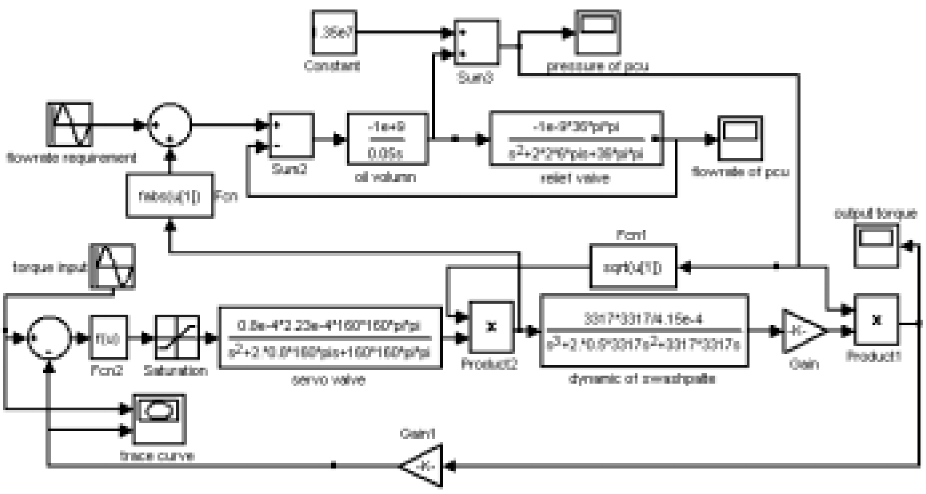


Fig.9 Transfer function of load simulator with PSU mdel

Gain=40e-6/2/pi/14.2e-3,Gain1=10*2*pi/40e-6/1.35e7

In this model, there is no swashplate angle feedback, the coupling of PSU is gave out, but no contamination torque attached which can be got by multiply pulse factor shown on the above. Next is the simulation according to variable situation.

3.2 Static Loading

There are no disturbance from the PSU and torque noise for static loading. Therefore, a good performance can be got. Fig.10 is step response, Fig.11 is the trace curve between input and output, the input is a 1Hz sine signal. There is a little of errors due to dynamic response.

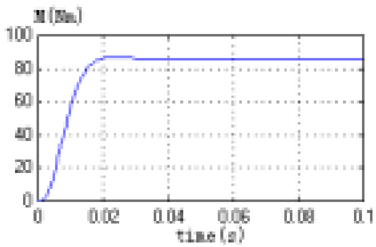


Fig.10 Step response of static loading

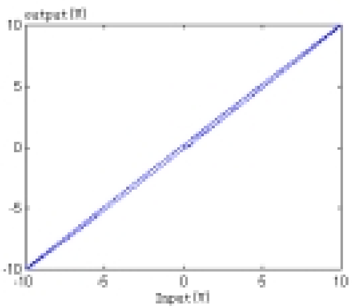


Fig.11 Trace graph of Sine input torque(1Hz)

3.3 Motion loading

The extraneous force problem is the key to the traditional electro-hydraulic servo loading system in the motion loading. Up to now there is not a special effective method to resolve it thoroughly. The current method can only reach some degree, for example, reduce 90% of its value which maybe also too big for some applications. But the new concept load simulator has not this problem at all.

In generally, there exists the motion while loading. It will make the system pressure wave, which will result in the torque wave. Another influence is the torque noise of itself. Because the swing loading frequency is not very high, this torque noise is not high also, which is related to the swing angle, swing frequency and piston number. This can be improved by robustness design. Fig.12 is the step response with torque wave. Fig.13 are the simulation results with PID controller.

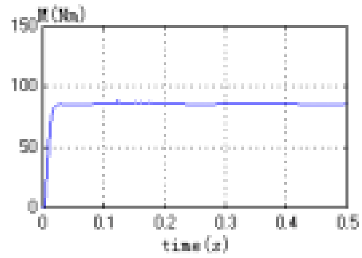


Fig.12 The step response with PSU influence

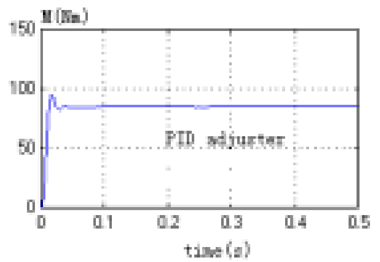


Fig.13 The step response with PID controller

The simulation results show that proper controller can improve system performance and robustness to the PSU disturbance

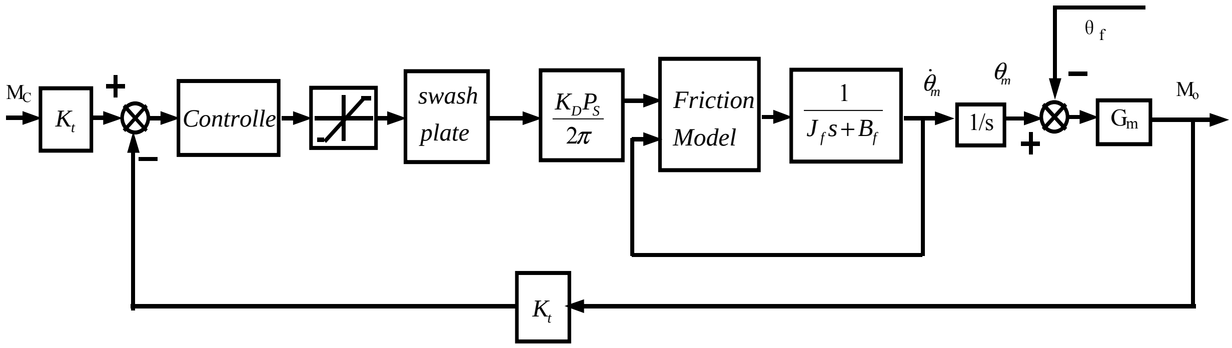


Fig. 14 Mathematical model of the total secondary unit with friction

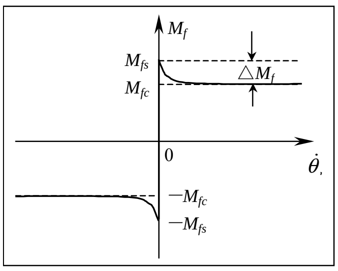
4 Friction Model and Fuzzy Control

4.1 Friction Model

Because of the special structure of the secondary unit, the friction is setup inevitably by the complex movement of the parts. It sometimes decreases the performance greatly. The complexity of the friction phenomenon makes itself difficult to be grasped clearly and totally. It depends not only on the movement velocity but also on the temperature, position, lubrication, and even the history of motion [10][12]. It is desirable to compensate frictions directly via an accurate model.

It is useful to set up these models to investigate its characteristics and its compensating methods. Here, in order to simplify the model without losing much accuracy, static friction, Coulomb friction, viscous friction and Stribeck effects are considered. In this condition, the friction M_f can be calculated by Equ.(4). Fig.15 shows the curve of friction model.

$$M_f = \mathrm{sgn}(\dot{\theta})[M_{fc} + \Delta M_f e^{-\alpha|\dot{\theta}|}] \tag{4}$$



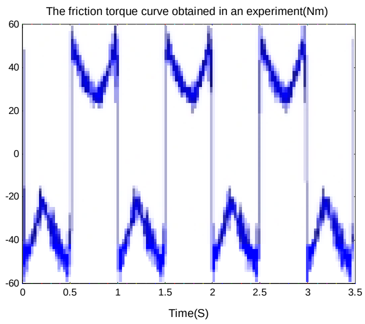


Fig. 15 Friction model and experimental

The experimental curve section was obtained while the load simulator moved with the tested object to trace a sine order.

Actually, there are two main frictions in the secondary unit. When the swash plate is adjusted, it pulls or pushes the pistons in the cylinders to produce friction. This friction creates the static error of the swash plate angle and decreases the dynamic performance. Furthermore, when secondary unit rotates, all the related motion in secondary unit can also generate friction. This friction affects the accuracy of the force tracing directly.

Fig.16 shows the influence of the friction in the swash plate angle control. The solid curve is the step response when considering the friction, and the dotted line is the step response without considering the friction. It can be concluded that the friction can bring more lag, more damp, and static error.

Fig.17 shows the influence of the friction in the torque tracing control. The solid curve is the step response without considering any friction. The dash curve is the step response with considering the friction in torque control but not the swash plate angle control. In this condition, dead zone, crawling and steady-state oscillation appear. And when all the frictions are included in the model, the performance becomes worse because the factors of lag and static error take effect.

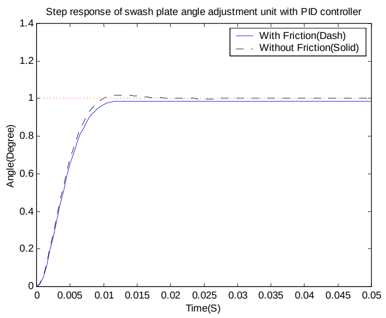


Fig.16 Step response of the swash plate

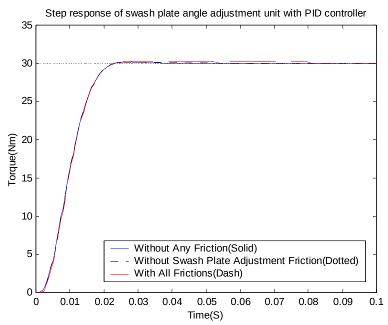


Fig. 17 Step response of the torque tracing

4.2 Fuzzy Controller

In order to compensate the nonlinearity and improved the system performance, a fuzzy controller is utilized. Generally, the fuzzy controller has four main components: (1) The “rule-base” holds the knowledge, in the form of a set of rules, of how best to control the system. (2) The inference mechanism evaluates which control rules are relevant at the current time and then decides what the input to plant should be. (3) The fuzzification interface simply modifies the inputs so that they can be interpreted and compared to the rules in the rule-base. And (4) the defuzzification interface converts the conclusions reached by the interface mechanism into the inputs to the plant. Fig.18 shows the principle of the fuzzy controller.

Here, the error of the expected torque and actual one is used as a reference input, while the derivation of the error is used as another one. In order to get better performance, a rule table, which is given by table 1, is created.

Table 1: Rule table for the torque tracing

u		$\dot{e}$						
		NL	NM	NS	ZR	PS	PM	PL
e	NL	PL	PL	PL	PM	PM	PS	ZR
	NM	PL	PL	PM	PM	PS	ZR	NS
	NS	PL	PM	PM	PS	ZR	NS	NM
	ZR	PM	PM	PS	ZR	NS	NM	NM
	PS	PM	PS	ZR	NS	NM	NM	NL
	PM	PS	ZR	NS	NM	NM	NL	NL
	PL	ZR	NS	NM	NM	NL	NL	NL

In this table, NL are used as an abbreviation for “negative large in size”, PS as “positive small in size”, ZR as “zero”, NM as “negative middle in size”, and so on. During the fuzzification and defuzzification, $e(t)$, $\dot{e}(t)$ and $u(t)$ use the same membership functions showed in Fig.19. The inference and the defuzzification are processed using the weighed average method

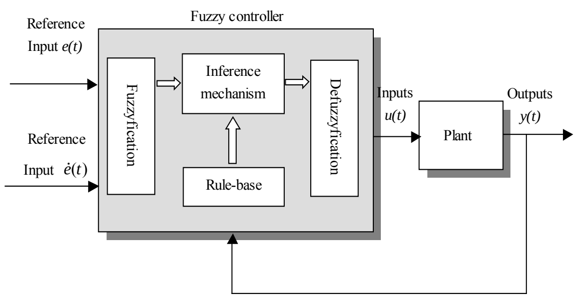


Fig. 18 Fuzzy controller

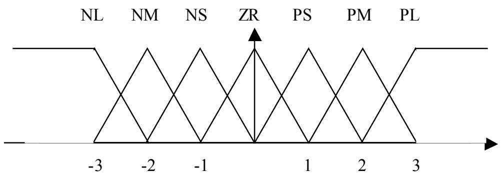


Fig. 19 Membership functions for torque tracing

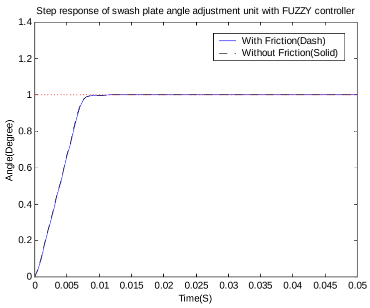


Fig. 20 step response of swash plate unit

Fig.20 shows the step response of swash plate based on fuzzy control. The dotted curve is obtained when considering the friction, and the solid one is obtained without considering the friction model. The curves fit each other well. It proves that the fuzzy controller can compensate the friction effectively. Figure 21 provides another proof of the fuzzy controller's effect. It shows the step response of torque tracing.

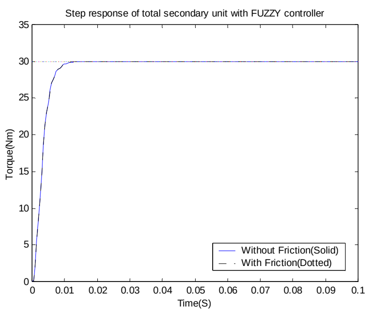


Fig. 21 step response of total secondary unit

Although the pure fuzzy controller can improve the system successfully, it is necessary to improve it by constructing hybrid fuzzy controller or adaptive fuzzy controller that the rule base can be adjusted on-line.

5 Conclusion

The advantage of the load simulator based on the secondary control principle is: Energy saving and recovery; The excellent dynamic performance; The smaller disturbance torque; Swing loading and rotation loading. The disadvantage is the greater of volume to output torque of motor. The main tasks will be low friction and large torque variable motor; Torque noise control; Robustness control of the system.

If the loading system and actuators tested can be powered by a same PSU, the power can be used for the best. The different control methods should be applied according to different applications.

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References

[1] Weishaupt E.Adaptive egelungskonzepte für eine hydraulische verstelleiheit am Netz mit aufgeprägtem versorgungsdruck im Drehzahl-und Drehwinkelregelkreis. *PhD thesis*, Rheinisch-Westfalische Technische Hochschule, Aachen, 1995.

[2] Wüsthof P, Secondary control technology and application, *ASME IMECE'98, FPSTD*, Nov.15-20,1998,Anaheim, CA,USA

[3] M. Ivantysynova, Potetial of Secondary Control for Hydraulic Drives used in Modern Flight Control System, *International conference on Aerospace Advances in Aerospace Hydraulics*, Nov.24-26, 1998, Toulouse, France.

[4] Hans Berg, Robuste Regelung verstellbarer Hydromotoren am Konstantdrucknetz, *PhD thesis*, Technishe Universitat, Harburg-Hmaburg, 1999.

[5] H Berg and M Invantysynova, Design and testing of robust linear controller for secondary controlled hydraulic drive, *IMechE 1999, Proc Instn Mech Engrs* Vol 213 Part I.

[6] Ramachandran S, Dransfield P. Interaction in a 2-channel loaded electro-hydraulic drive. *J of fluid Control*, 1982, 14(3).

[7] Liu Changnian, The Optimum Theory of Hydraulic Servo Systems, *Metallurgy Industry Press*, 1989.

[8] M.Martin, R.Tburton, and G.J.Schoenau., Analysis, design and compensation of computer controlled hydraulic load simulator. *Proceedings of ASME winter meeting 1992*, November, 1992

[9] Ohuchi-H, Jukui-H etc., Model reference adaptive control of a hydraulic load simulator, *ICFP'93,* Sept.14-16, 1993, Hangzhou China.

[10] Cong Shuang., Two Adaptive Friction Compensation for DC Servomotors. *Proceedings of The IEEE International Conference on Industrial Technology*, 1996.

[11] Hua Qing.. Studies on the Key Technology of Electro-hydraulic Load Simulator. *PhD Thesis. Beijing University of Aeronautics and Astronautics*, 2001.

[12] S. S. Ge, T. H. Lee and S. X. Ren. Adaptive Friction Compensation of Servo Mechanisms. *Proceedings of the 1999 IEEE International Conference on Control Applications*. Hawaii, USA, 1999.