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DEVELOPMENT OF AGRICULTURAL HYDRAULIC ROBOT (Part II)
--- Dynamic Characteristic of Hydraulic System

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ABSTRACT

Agricultural hydraulic robot which was reported in Part I had been developed. The robot was satisfied performance to intend before development. For actual use, however, it have been necessary to reduce manipulator weight and to simplify construction of hydraulic control valve. Then, working stress of manipulator link and pressure fluctuation of hydraulic circuit were measured. Step and frequency response tests were done subject to amplitude of reference voltage of 0.1, 0.3, 0.5 and 1.0V, and delivery pressure of 3.5 and 5.0 MPa.

Working stress were about 25% comparing with fatigue strength. Thus, mass of manipulator might be reduce to 30%.

In hydraulic control system, virtual natural frequency of 6.5Hz is produced from the combination of drain passage area shortage of servovalve. Further, because of passage area shortage, working pressure at both side of cylinder was acted on. This phenomenon prevent to utilize effectively engine power. Then, control valve for new model was proposed.

Key Word : Robot, Servovalve, Stress, Virtual natural frequency

INTRODUCTION

In Part I, necessity of agricultural robot was reported, which could handle heavy vegetable or fertilizer baggage and move in field. Further, advantages of hydraulic drive system for agricultural robot comparing with electric one was reported. Parameter identification tests using developing agricultural hydraulic robot. It was clarified that the robot had good responsibility and accurate positioning ability. However, servovalve capacity was too large and manipulator strength was excess. To use as agricultural robot, it was necessary to reduce manipulator weight and simplify construction of hydraulic control valve.

In this study, working stress of manipulator link and pressure fluctuation of hydraulic circuit were measured and analyzed. On the basis of experimental fact, hydraulic control valve and system for next model are taken into consideration.

CONSTRUCTION OF HYDRAULIC CONTROL VALVE AND ITS ANALYSIS

Manipulator and hydraulic circuit is shown as Fig. 2 and 5 in Part I, but prime mover is changed from engine to electric motor in order to measure accurately high frequency response. Construction of servovalve is illustrated in Fig. 1. The spool move until the force due to return spring balances the force due to input current to solenoid. The input current is proportional to reference voltage to driver. The spool take four basic position due to reference voltage "off", "negative", "zero" and "positive". As a result of fluid flowing, steady and unsteady flow force are produced on the spool. Thus, spool position indicated by reference voltage is not always kept. In this servovalve, detecting spool position by differential transformer, error between spool position and reference voltage is compensated with feedback control. Since spool position i.e. fluid passage area of spool is controlled by reference voltage, fluid flow and supply direction are selected by reference voltage to change from 5V to -5V. Block diagram of servovalve control system is shown in Fig.2. Where u(s) is reference voltage input, $x_v(s)$ is spool valve stroke, spool mass m is 0.027 kg. spring constant k is 9800 N/m. Thus, undamped natural frequency with no feedback is 95.9Hz. Viscous damping coefficient c should be obtained from experimental data. On the other hand, proportional constant between voltage and current G_1 is 0.04667 A/V. Proportional constant between current and force G_2 is 42.0 N/A. Proportional constant at feedback loop H is 5000 V/m.

Hydraulic pump is a variable displacement axial plunger type with pressure compensator system. Schematic diagram of pump swash plate control system is shown in Fig. 1. The number of plunger is seven and theoretical delivery volume Vth is 8×10^{-6} m^3/rev (8 cc/rev). If delivery pressure exceed set value, selector valve is operated and pressure fluid send to servo cylinder to return the angle of swash plate. The incline of the angle is decreased and delivery flow is decreased. Therefore, delivery pressure is decreased and pressure is kept on set point. Parameters of the control system should be obtained from experimental data.

Similarly, construction of proportional control valve is illustrated in Fig. 3. The proportional valve has two solenoid and two return spring at both side, respectively. When input current is "off", the force due to return spring at right hand side balances the force due to spring at left hand side so that spool is kept at neutral position. This valve has overlap of 2.5mm and no feedback control to detect spool position. Thus accuracy of spool position is inferior to compare with servovalve. Hydraulic cylinder force is given by

$$F=p_A A_A - p_B A_B \qquad (1)$$

where p is working pressure of cylinder, A is working area of cylinder. Subscript $_A$ and $_B$ are indicated bottom and rod side of the cylinder, respectively. In case of Link 3, working area A_A is 39.2 cm^2 and A_B is 31.6 cm^2. Relation between cylinder force F and joint torque τ is obtained using theory of cosine. Although the relation between F and τ is nonliner, it is possible to treat approximately as liner in short range. The relation between angular acceleration $\ddot{\theta}$ and torque of Joint 3 is given by Eq.(4) in Part I. Relation between joint torque and strain of Link 3 is given by

$$\varepsilon = s\tau/ZE\ell \qquad (2)$$

where s is distance from end of link to strain measured point, Z is section modules, E is modules of elasticity and ℓ is length of link. Strain ε is proportional to joint torque τ. Also, acceleration of end of manipulator is approximately proportional to joint torque.

EXPERIMENTAL METHOD

Step and frequency response tests were done subject to amplitude of reference voltage of 0.1, 0.3, 0.5 and 1.0V, and delivery pressure of 3.5 and 5.0 MPa. Since maximum input voltage of the servovalve is 5.0V, input voltage was too low comparing with ordinary operating condition. In step response test, input voltage was changed submit to rectangular wave. Joint 3 and Joint 2 were operated individually. Table 1 shows experimental conditions. Measured items are strain at two points of Link 3, pressures of hydraulic cylinders, acceleration of end of Link 3 and reference input voltage. Measured points and cross section at strain measured point are shown in Fig. 4.

RESULTS AND DISCUSSION

Transient response of Link 3 due to step voltage input to change from -0.3 to 0.3V is shown in Fig. 5. Fluctuation curve of delivery pressure of pump i.e. main pressure p_M is obtained to superpose four type of waveform. When reference voltage input is changed from negative to positive or from positive to negative, spool of servovalve is moved from right to left or from left to light. In this case, spool is always passed through neutral position. At the neutral position, delivery flow from pump is locked and delivery pressure p_M is increased. Therefore, pressure compensator system is acted immediately and the angle of swash plate is rapidly decreased. However, this fluctuation is faded away in 0.05s.

Servovalve is vibrating system with feedback control which consist of mass of spool and return spring. Right after reference voltage input is changed, spool of servovalve is vibrated and delivery pressure is fluctuated by vibrating spool. This fluctuation can be observed distinctly in Fig. 6. From these results, damped natural frequency with feedback control is 125Hz and damping coefficient c is obtained as 17.83 Ns/m. To compensate by feedback control, also, this fluctuation is faded away in 0.03s

Third wave is locking phenomenon with plungers of hydraulic pump. The pump has seven plungers and rotated at 1800rpm in this test. Therefore, the frequency became 210Hz. Amplitude of this vibration may be neglected to be too small comparing with other wave.

Last wave is low frequency comparing with other wave, but this wave is given significant effect. The frequency of 12Hz is generated by combination of shortage of drain flow passage area and closed loop natural frequency of pressure compensator system. Transient response due to rectangular waveform voltage input to change from -0.3V to 0.3V is shown in Fig. 7. Nevertheless Link 3 is lifting up or going down, both pressure p_A and p_B are acted on both cylinder side. As hydraulic cylinder force is given by Eq.(1), this phenomenon reduce cylinder force per working pressure. Both pressure pA and pB are fluctuated at 12Hz. However, difference of phase shift between pressure p_A and p_B is about 180 . Therefore, cylinder force is vibrated at the frequency of 6.5Hz. Although, both pressure p_A and p_B act simultaneously on cylinder, Link 3 is moved at required speed and is generated required force. If drain flow passage area were opened sufficiently, pressure pB would nearly equal zero, and 200N payload could be lifted up provided pressure pA is 0.5 MPa.

Static load - strain diagram is shown in Fig. 8. Strain i.e. stress under arbitrary load can be calculated using equations given by Strength of materials. Strain under rating payload 200N is 70×10^{-6} (stress 4.9MPa). As fatigue strength of aluminum alloy is about 35MPa, stress of this link is one sventh of fatigue. Transient response of Link 3 due to step reference input to change from -0.3V to 0.3V is shown in Fig. 9. Strain is not steady state before reference voltage input is changed in Fig. 9. Because rectangular function is used instead of step function so that fluctuation has already begun. Strain is vibrated at frequency of 71.4Hz. this frequency is natural frequency of Link 3. However, amplitude of this frequency is small. If frequency of input to servovalve is not 71.4Hz, this can be neglected. In part I, we reported that manipulator link can be treated as rigid body. This fact is proved in this results. Considering, thus, only static load and inertia force, working stress can be calculated. Strain and acceleration are proportional to joint torque. Then joint torque is nearly proportional to cylinder force. Strain and acceleration which are fluctuated proportional to cylinder force are observed in Fig. 8.

Transient response due to sinusoidal input voltage at frequency of 1Hz is shown in Fig.10. Pressure p_A and p_B, strain force and acceleration consist of frequency of 6.5Hz and 1Hz. 1Hz is frequency of input voltage and 6.5Hz is above mentioned virtual natural frequency. Transient response due to sinusoidal input voltage at frequency of 6.5Hz is shown in Fig.11. In this test, Link 3 was vibrated with large amplitude. Before this analysis, we supposed that resonance occurred for voltage input frequency went near natural frequency of Link 3. However, cause of this phenomenon was virtual natural frequency of 6.5Hz.

Transient response of proportional control valve was shown as Fig. 11 in part I. Since the valve had overlap of 2.5mm, time delay of 0.1s occurred. On account of time delay, overlapped valve can not use as control device of robot manipulator.

HYDRAULIC CONTROL VALVE FOR NEXT MODEL

Robot manipulator to harvest vegetable is required to lift up 200N payload. If the manipulator can lift up 400N, we supposed that the manipulator can treat almost farm operation. If cylinder of current model may be used, working pressure of 0.5MPa is enough to lift up 200N. Therefore, required pressure for agricultural robot is 1MPa. It is important to be able to operate at low working pressure. Also hydraulic control valve for robot is required quick response. Thus overlapped valve can not be used. To use effectively engine power, opposite side pressure of hydraulic cylinder shall be kept nearly equal zero. Thus underlapped valve can not be used, too. On the other hand, it is effective for accurate positioning and speed control that working pressure act on both bottom and rod side of hydraulic cylinder according as necessity. To satisfy such a condition, we propose hydraulic system as shown in Fig. 12. In proposed system, pressure of hydraulic cylinder is controlled individually by each solenoid valve. In automobile industry, low pressure solenoid valves are mass produced for automatic transmission control. If we will select 1MPa as working pressure, we can use such a low cost solenoid valve for agricultural robot Satisfying, however, operating speed for agricultural use, delivery flow from pump may be required $0.2\times10^{-3}m^3/s$ (12l/min). Such a large flow can not usually pass through the solenoid valve. Therefore, pilot-actuate valve method should be adopted. As working pressure of 1MPa is, this method may be used easily as agricultural robot.

CONCLUSION

Agricultural hydraulic robot had been developed to harvest watermelon. The robot had good performance about operational space, positioning accuracy and responsibility. For actual use, however, it have been necessary to reduce manipulator weight and to simplify construction of hydraulic control valve. To study manipulator and hydraulic valve for next model, working stress of manipulator link and pressure fluctuation of hydraulic circuit were measured.

Working stress were about 25% comparing with fatigue strength due to aluminum alloy metal when rating lifting force 200N was acted on. If steel was adopted instead of aluminum, massof manipulator might be reduce to 30%, inversely.

In hydraulic control system, by the reason of the combination of drain passage area shortage of servovalve and natural frequency of pressure compensator system of pump, vibration at frequency of 12Hz was generated. Further, because of passage area shortage, working pressure at both side of cylinder is acted on. Phase shift between both pressure were difference of 180°. Therefore, virtual natural frequency of 6.5Hz become to be in pressure compensator system of pump. This phenomenon prevent to utilize effectively engine power. By this phenomenon, however, position and speed control became easy. Since overlapped valve had time delay, it was not used as control valve for robot manipulator.

If cylinder size is equal to current model, working pressure of 1MPa and delivery flow of pump is $0.2\times10^{-3}m^3/s$ is required at the most. On the condition, low cost solenoide valve for automatic transmission control of automobile. we proposed hydraulic control valve provide using low cost solenoide valve after this, we will take into considration details moreover and we wish to develop hydraulic cotrol valve and agricultural robot.

Table 1. Experimental condition

Reference voltage input waveform	Rectangualar, Sinusoidal
Input amplitude	0.1, 1.3, 0.5, 1.0 V
Pump delivery Pressure	3.5, 5.0 MPa

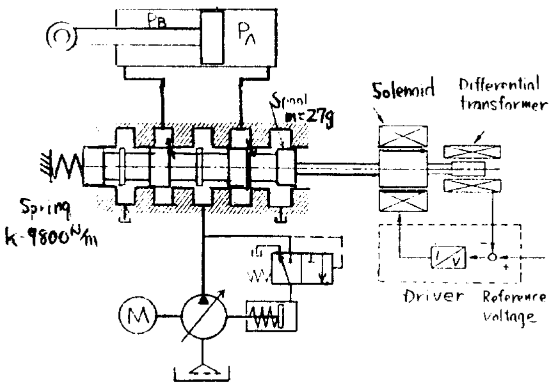

Fig. 1. Construction of servovalve and schematic diagram of pump pressure compensator system

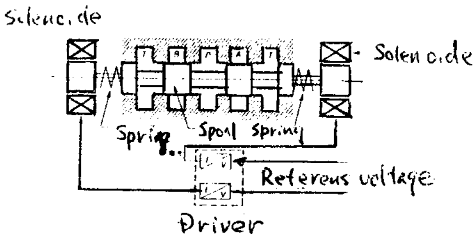

Fig. 3. Construction of proportional control valve

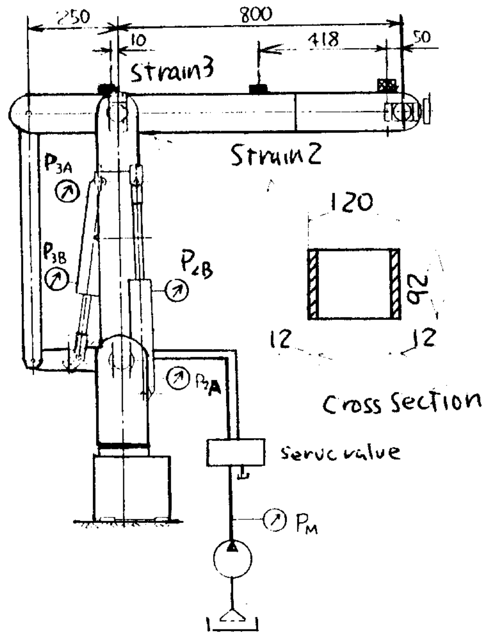

Fig. 4. Measured points and cross section at strain measured point

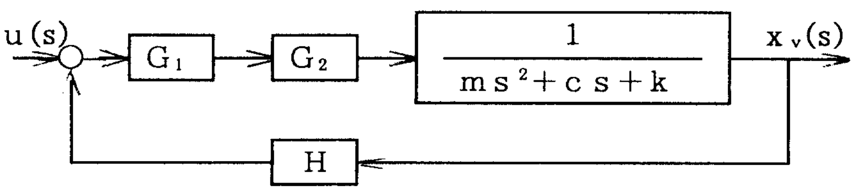

Fig. 2. Block diagram of servovalve

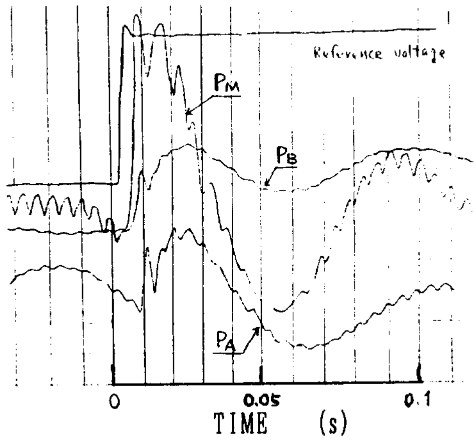

Fig. 5. Transient response due to step voltage input

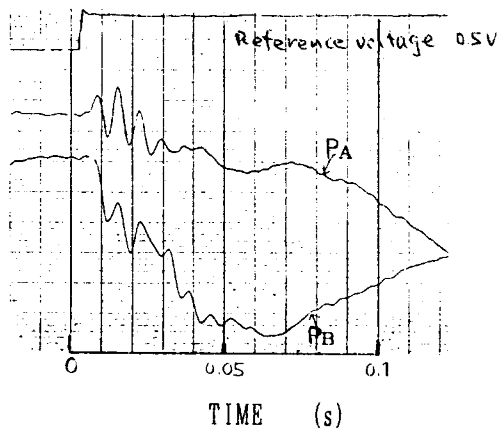

Fig. 6. Vibration of servovalve spool with feedback cotrol

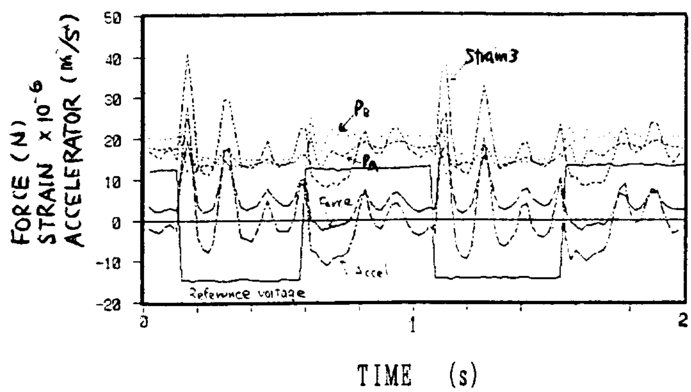

Fig. 7. Transient response due to rectangular voltage input

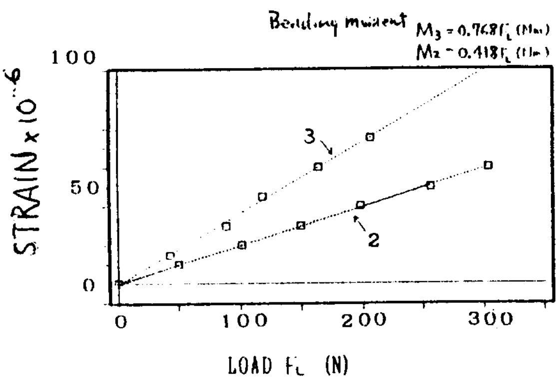

Fig. 8. Static load -link strain diagram

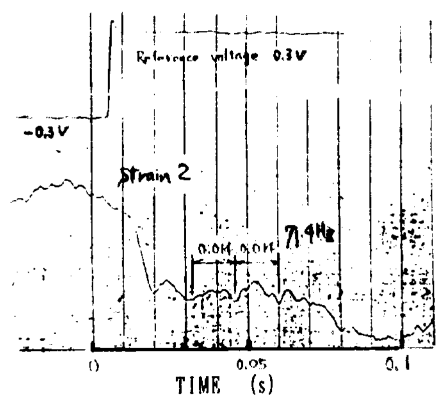

Fig. 9. Link strain transient response

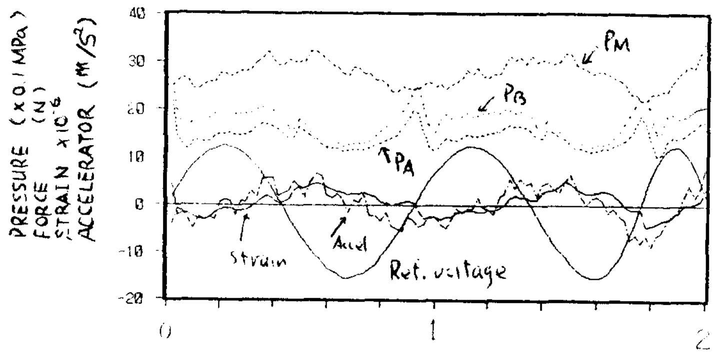

Fig.10. Transient response due to sinusoidal input at frequency of 1Hz

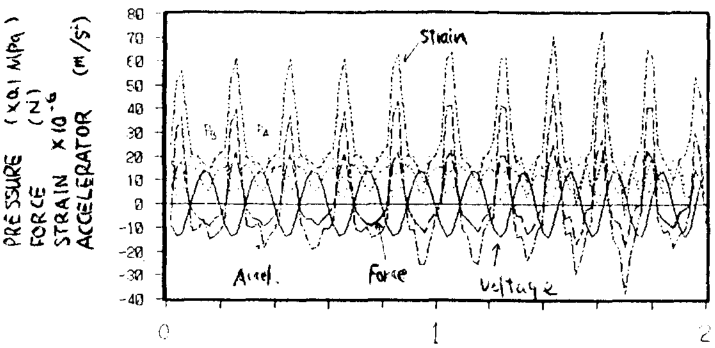

Fig.11. Transient response due to sinusoidal input at frequency of 6Hz

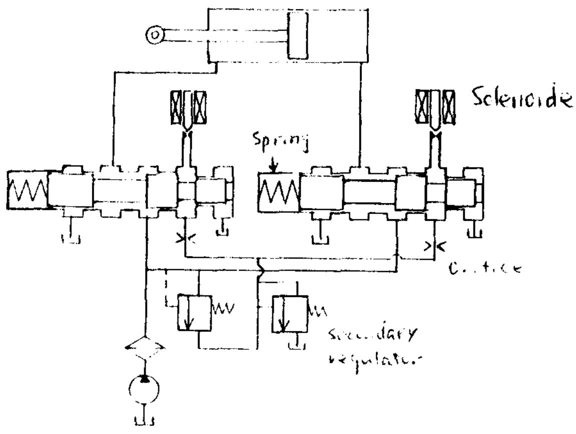

Fig.12. Hydraulic control system for next model