International Journal of Fluid Power 4 (2003) No. 1 pp. 41-48

DEVELOPMENT OF A PNEUMATIC FORCE-DISPLAY (APPLICATION TO A MASTER-SLAVE SYSTEM)

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Abstract

In this study we deal with a bilateral master-slave system composed of a pneumatic force-display as the master and a hydraulic servo system as the slave. In such systems the force-display must play two roles as master: first as a reference input device to the slave and second as a force-display device. The first purpose of this study is to develop a pneumatic force-display that consists of a pneumatic servo system. To achieve this, it is necessary to solve a problem called back-drivability, a characteristic of pneumatic servo systems. The second purpose is to investigate the compatibility of our thusly developed force-display with some representative methods of bilateral master-slave control systems in conventional use. In experiments to confirm such compatibility, the sensibility of load forces is estimated based on a master-slave system equipped with a spring to serve as a load. The experiments confirm that the developed force-display would be applicable to conventional methods of bilateral master-slave systems.

Keywords: manipulator, pneumatic servo-system, master-slave system, force-display, bilateral control, remote control

This manuscript was received on 07 October 2002 and was accepted after revision for publication on 06 March 2003

1 Introduction

A teleoperating system for machining tasks such as grinding, or for construction tasks such as drilling, must provide the operator with not only an accurate visual representation of the working conditions but also an faithful sense of the reaction force at the work head. To achieve this, various haptic force-displays, which are master devices that provide a machine operator with a sense of the force being applied, have been developed (Burdea, 1996). Existing models of haptic force-display devices, however, still have a number of inadequacies, such as insufficient force range, small loading capacity, complicated structure and high cost; they are therefore not ideal for actual production environments. In addition, the majority of existing haptic displays are the electric-drive type (Massie, 1994), and there are few pneumatic-drive models (Takaiwa, 1999). A wider range of display types, utilizing different control and/or driving methods, needs to be developed.

From this perspective, in a previous study we developed a hydraulic-type haptic force-display that incorporates a novel drive-control method and offers a number of advantages (Kudomi, 2000):

(1) The new display is driven hydraulically and so is expected to be free from the drawbacks of existing displays, such as limited force range, small loading capacity, and structural complexity.
(2) The display is constructed on a hydraulic servo control system. By adopting the display as the master, it is possible to create a master-slave system that is well matched to other slave systems constructed on a similar hydraulic servo system.
(3) Experimental investigations regarding the reaction force reveal a fairly precise response which corresponds to a spring load directly. It is also confirmed that the system has excellent controllability when applied to various master-slave control systems.

In this study, we develop a similar force-display that is pneumatically driven instead of hydraulically driven. The new display is expected to offer the following advantages:

(1) As it is similar to a fluid control device, the pneumatic display should offer the same advantages shown by the hydraulic display in the previous study (Kudomi, 2000). There is a similarity in the valve or cylinder mechanisms between hydraulic and pneumatic

system. Therefore it is expected that the similar control algorithms, which is adopted in the previous paper, are applicable into the present research. On the other hand, there is a problem that rigidity of pneumatic system is much lower than that of hydraulic system. However, the pneumatic system is used as the force display in this study. Therefore, the characteristics of low rigidity can be admitted practically. Additionally, although the pneumatic method causes an inevitable decrease of the deliverable force, the cost is much lower.

(2) If both hydraulic and pneumatic displays of similar structure are marketed, the customer can choose the display that is appropriate for an application.

Therefore, as the first step in this study the authors investigate experimentally the haptic function of the developed pneumatic display. We construct a single-degree-of-freedom master-slave control system consisting of a pneumatic display as the master and a hydraulic servo system as the slave. We then conduct a series of contact experiments against a spring load and investigate the adaptability of the display to a number of typical bilateral control methods.

2 Pneumatic Force-display

2.1 System Structure

The present study investigates the pneumatic servo system shown in Fig. 1 and its application to a haptic force-display. The servo system is composed mainly of a nozzle-flapper type servo-solenoid valve ("servo valve"), and a low-friction pneumatic cylinder. The servo valve is a non-symmetrical component. Therefore, when the input signal is zero, the flapper displacement is held at zero, and in this state the piston continues to pull back (moves to the left in the figure). To stop this piston movement, it is necessary to add a constant voltage (offset signal) to the input signal, thereby keeping the piston in the neutral condition (zero drive force). The pneumatic cylinder has no packing or seal, minimizing the piston's frictional resistance. This reduces the occurrence of stick-slip even at low velocity, but it also increases internal air leakage in the cylinder.

When a hydraulic servo system is used as a force-display, the well-known problem of "back-drivability" is expected to arise, owing to the inherent dynamic characteristics of the hydraulic servo system. Here, definition of back-drivability is the characteristics of drivability whether the operator's input force can drive the hydraulic piston. Generally, hydraulic or pneumatic piston cannot be operated freely by means of manual force because of the operating principle of the servo-valve or frictional resistance acting on the cylinder. Due to this problem, the operator has such difficulty that he is unable to drive the system manually by operating, for example, a cylinder or a servo-valve. Our previous paper reports an effective solution to this problem (Kudomi, 2000). Since back-drivability can be expected to occur also in the pneumatic servo system, here we examine experimentally the drivability of the system shown in Fig. 1, where the operator's force is applied directly to the piston. Fig. 2 shows the relation between the operating force f_{op} and piston displacement y obtained in the experiment. It can be seen that a considerable force is required to move the piston. From a practical point of view, the operating force should be as small as possible. Hence the system in this form would not be practical. We therefore introduce a cylinder drive method with a force sensor similar to the one applied to the hydraulic type display in the previous study (Kudomi, 2000). The proposed system is based on a pneumatic servo system, and incorporates a driving table (connected directly to the piston rod) equipped with a force sensor (a strain gauge pasted on a plate spring, see Fig. 4). Using the force sensor as an input device, a manual cylinder drive method can be achieved as follows. By touching the plate spring ("control plate") shown in Fig. 4, the operator can input a force signal to the system. First, the force sensor on the driving table measures the force applied by the operator and then generates a force signal. This signal drives the servo valve, moving the piston in the direction of the input force. Fig. 3 shows the result obtained by driving a pneumatic piston with this method. In comparison with Fig. 2, we can see that a small force can cause a large piston movement, so this method greatly improves drivability.

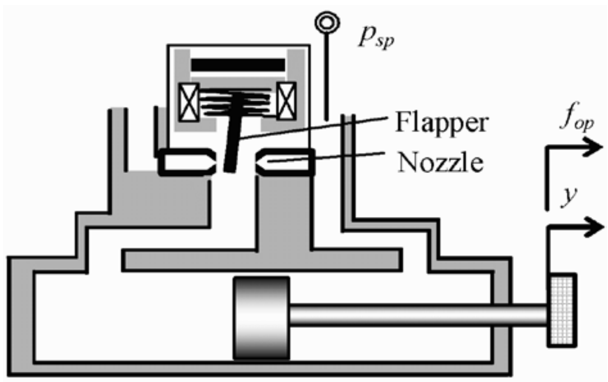


Fig.1: *Pneumatic servo system*

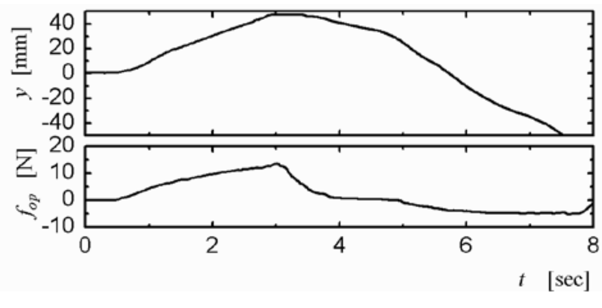


Fig. 2: *Back-drivability of the pneumatic servo system*

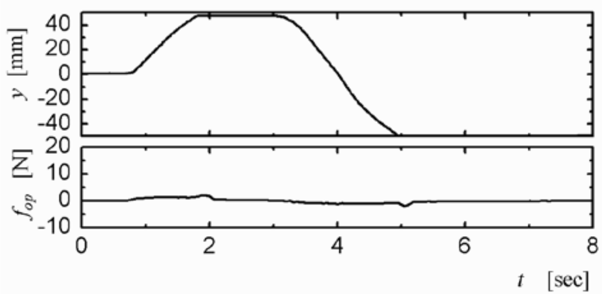


Fig. 3: *Driving method using a force sensor*

2.2 Force Display with a Virtual Spring

We investigate the haptic force-display with a virtual spring to confirm that the force is displayed to the operator. Figure 4 illustrates the configuration of the system with a virtual spring. As shown, the virtual spring is set at a distance "*a*" from the neutral position on the piston's retreat side. The haptic force-display is realized as follows. First, a control force f_{op} generates the piston movement in accordance with the aforementioned drive control method. Next, when the piston has traveled 10 mm, it makes contact with the virtual spring, whereupon a reaction force f_e is generated. The reaction force is equal to the piston displacement Δy multiplied by the spring constant k. By controlling the piston movement in such a way that f_e follows f_{op}, the system can deliver to the operator the reaction force from the virtual spring. In the following experiment, we investigate the display function of the reaction force from the virtual spring (simulated by a computer). Two different spring constants are chosen for comparison.

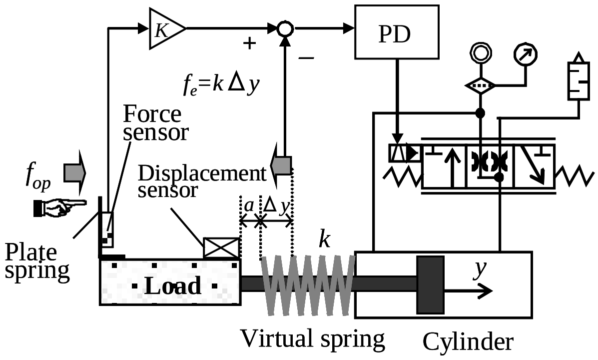
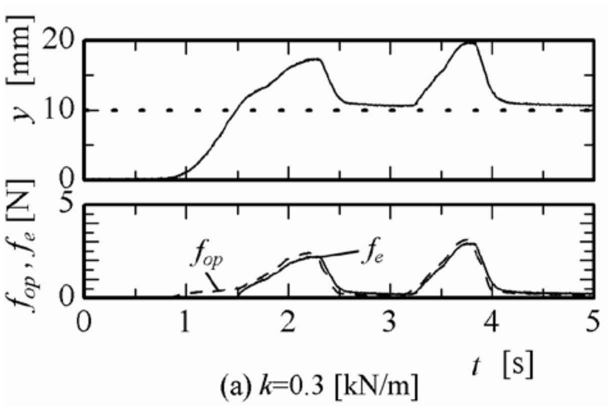


Fig. 4: *Pneumatic force-display*



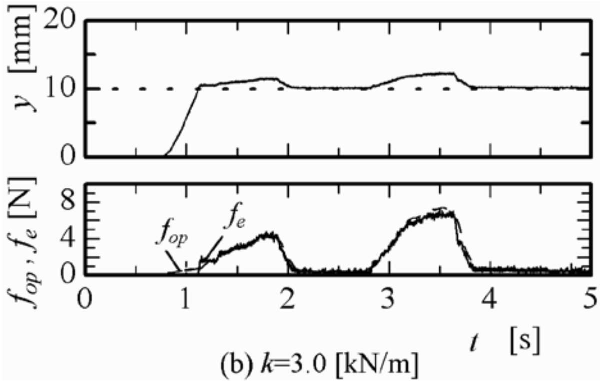


Fig. 5: *Force feedback from a virtual spring*

The results are shown in Fig. 5(a) and (b). Figure 5(a) shows the results when the spring constant is set to $k = 0.3$ kN/m. After the piston makes contact with the spring, f_{op} and f_e agree very well (the two curves overlap in the figure). Thus, the spring reaction force is accurately represented. Furthermore, the developed piston displacement is proportional to and much higher than the applied force, starting from the point of contact (10[mm] point) indicated by the broken line in the figure. This means that the operator would feel he is touching a soft spring. Fig. 5(b) shows the results when the spring constant is set to $k = 3.0$ kN/m. As in Fig. 5 (a), f_{op} and f_e agree with each other very well. The piston displacement in this case is very small beyond the contact point, which means that the operator would feel he is touching a very stiff spring. These experimental results confirm that the present system functions well as a haptic force-display.

Next, we construct a master-slave system composed of the above force-display as the master and a hydraulic servo system as the slave. With this system, a series of contact experiments is conducted to evaluate the adaptability of the force-display to different teleoperating systems, using four typical bilateral control methods for comparison.

3 Pneumatic-hydraulic Type Master-slave Control System

3.1 System Configuration

The configuration illustrated in Fig. 6 shows that, on the master side, the force sensor measures the force applied by the operator and converts it into an electric signal that activates the piston motion. In the slave system, a force sensor is mounted on the end of the piston rod.

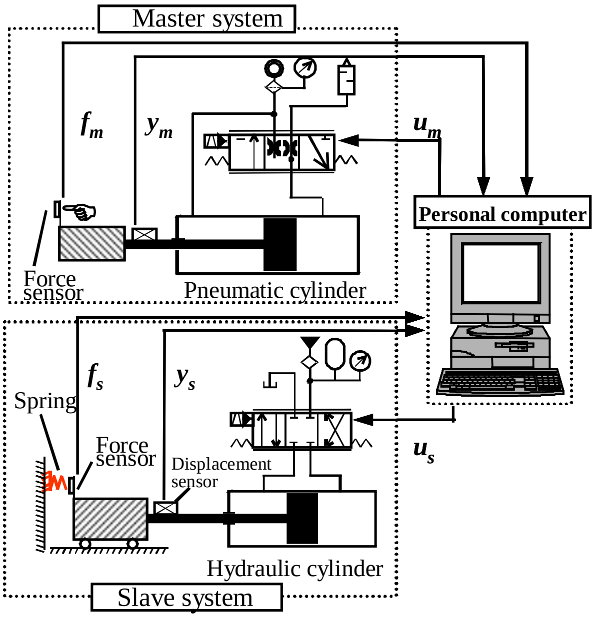


Fig. 6: *Diagram of experimental apparatus*

Next, the computer processes two piston-displacement signals (y_m, y_s) in the master and two force-sensor signals ($f_{op} = f_m$, f_s) in the slave. Using a bilateral control algorithm, the computer generates control signals (u_m, u_s) that are fed into the master and slave, respectively. The PD type control method is used for both the master and slave systems, and the chosen sampling time is 1 ms. And suitable feedback gain K from the force sensor is adjusted up through trial and error. As a controller for the servo valve, we adopt PD controller as a simple and effective controller similarly as in the previous paper (Kudomi, 2000). The open-loop transfer function of the pneumatic system becomes 1-type controller which include an integrator (1/s). Thus, we adopt PD controller because an integrator is not required in the system. Here, a derivative control "D" is adopted in order to stabilize the system. As experimental parameters, two different spring constants are chosen, k = 0.4 and 3.0 kN/m. Table 1 shows the major parameters for the experiments.

Table 1: System parameters

Master system	ω_{np} = 800 rad/s, ζ_p = 0.4, $p_{sp} = 0.35\times10^6$ Pa, k_{fm} = 0.12 V/N
Slave system	ω_{nh} = 623 rad/s, ζ_h = 3.0, $p_{sh} = 3.5\times10^6$ Pa, k_{fs} = 0.16 V/N

3.2 Bilateral Control

In this study, a series of comparative experiments is conducted on the control performance of the systems. Among different bilateral control methods currently in use, we choose four typical ones for this study: the force-reflecting servo type (Kato, 1990), the symmetric force feedback S type, the symmetric force feedback M type (Sato, 1993), and the parallel control type (Miyazaki, 1988). These are denoted as FRS, SFF-S, SFF-M, and PC, respectively. Figure 7(a) to (d) shows block diagrams of these methods.

The FRS system has a force sensor on the master and another on the slave. The master is driven by a servo system with reference to force, the slave by a servo system with reference to position. This configuration makes the operation of the master easier, because the master develops the driving force in the direction the operator applies to the control force. However, because the master and slave are connected in series, the phase lag in their motions is likely to increase, making the system unstable.

In symmetric force feedback systems, the master and slave are connected in parallel, their driving forces are generated to compensate for the difference in the forces, and position deviation between the master and slave is used as a feedback parameter. Depending on the difference in feedback methods using position deviation, two types of control systems—SFF-S and SFF-M—may be applied, as illustrated in Fig. 7(b) and (c). In SFF-M, the feedback is fed to the master, while in SFF-S, it is fed to the slave. With either system, the phase lag is expected to be smaller than that of FRS, and so better system stability is expected.

In the PC type system shown in Fig. 7(d), a position command is generated using the control force of the master and the reaction force of the slave. The two control systems are connected in parallel so that they both follow this position command, thus improving system stability.

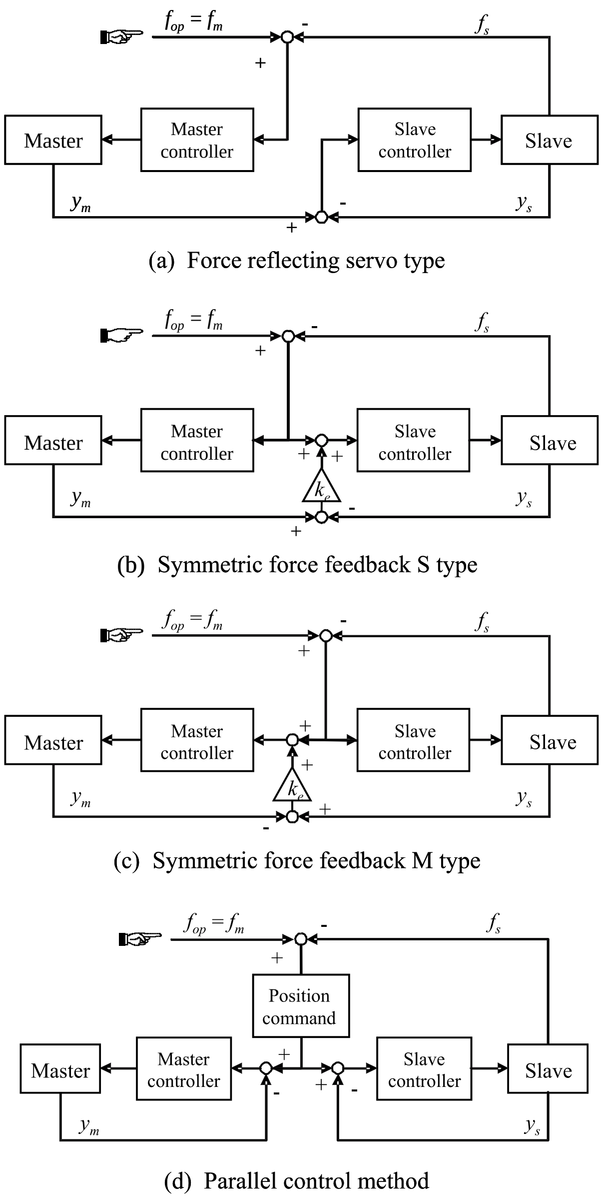


Fig. 7: *Bilateral control methods applied for pneumatic force-display*

4 Results and Discussion

A master-slave control system based on each of the four bilateral control methods described in Section 3.2 was manufactured and used in experiments in which contact tasks were applied to spring loads. The results were compared to evaluate the control performance, focusing on control force, the follow-up performance of the piston (displacement) and system stability.

For each control method, two PD-control formulae were adopted:

- Master controller: $K_{pm}\,(1 + K_{dm}\, s)$
- Slave controller: $K_{ps}\,(1 + K_{ds}\, s)$

The position command generator, which is required in PC type controllers, uses the following PID control formula, based on the study reported in our previous paper (Kudomi, 2000):

Position command generator: $K_{p}\left(1+\dfrac{K_{i}}{s}+K_{d}s\right)$

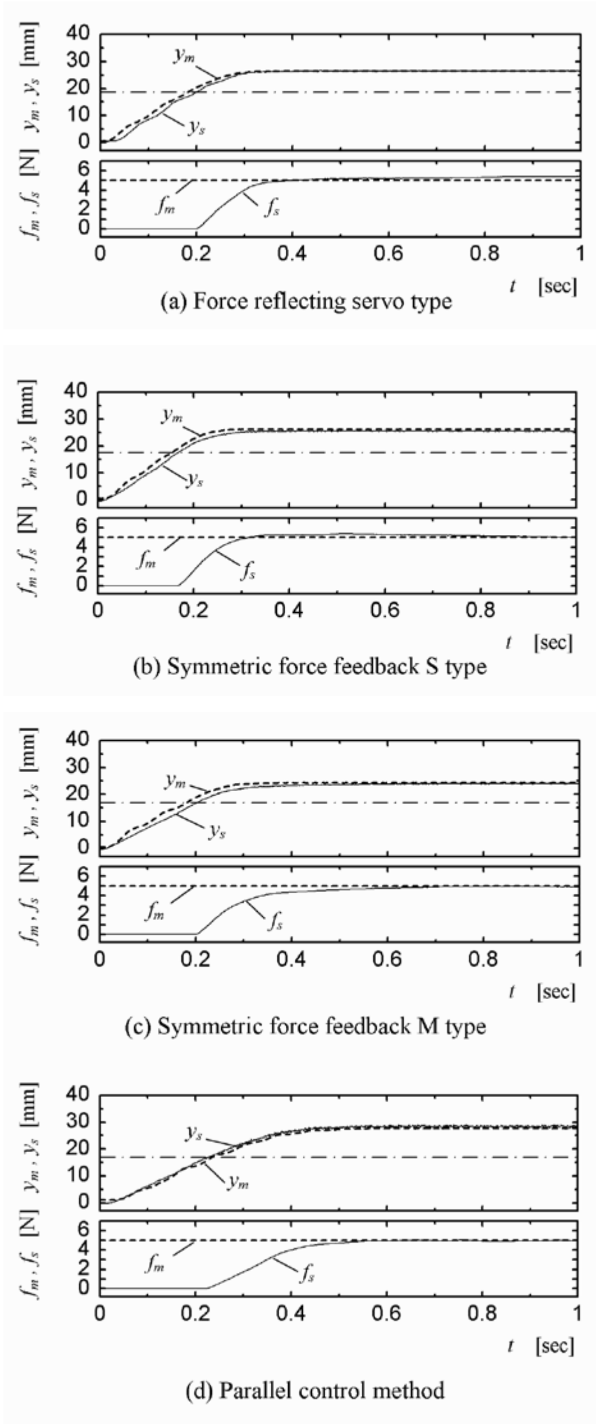


Fig. 8: *Experimental results of contacting a soft spring (k = 0.4 kN/m)*

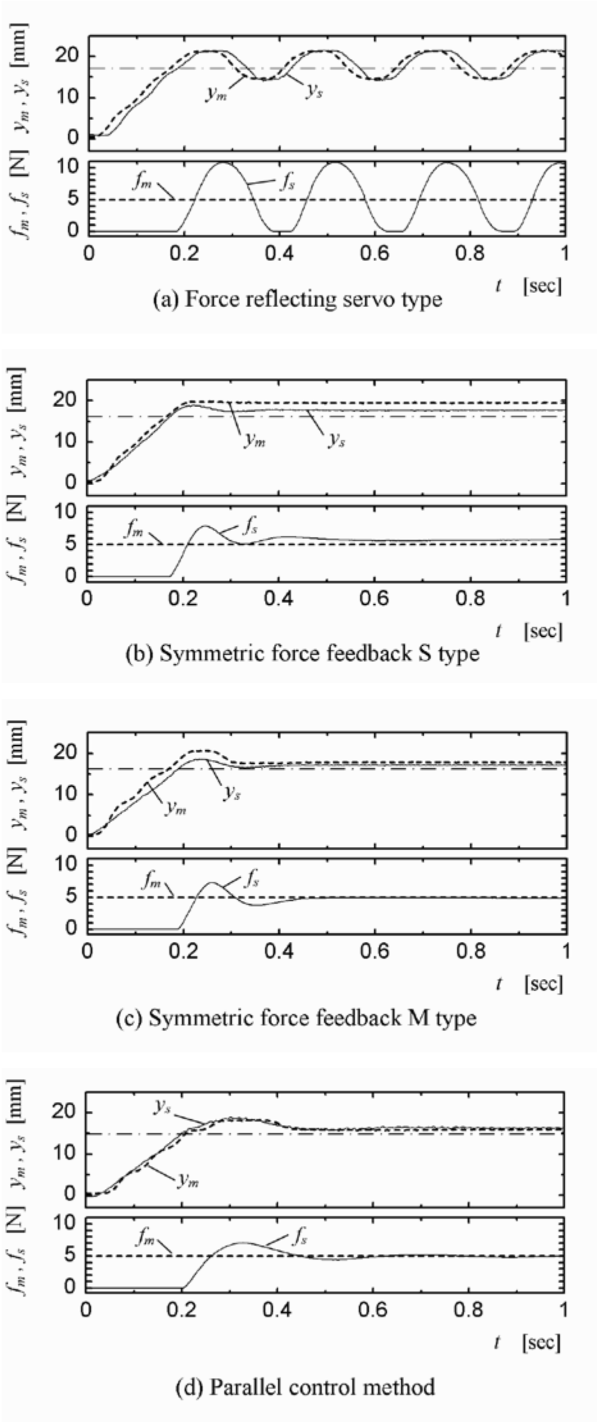


Fig.9: *Experimental results of contacting a hard spring (k = 3.0 kN/m)*

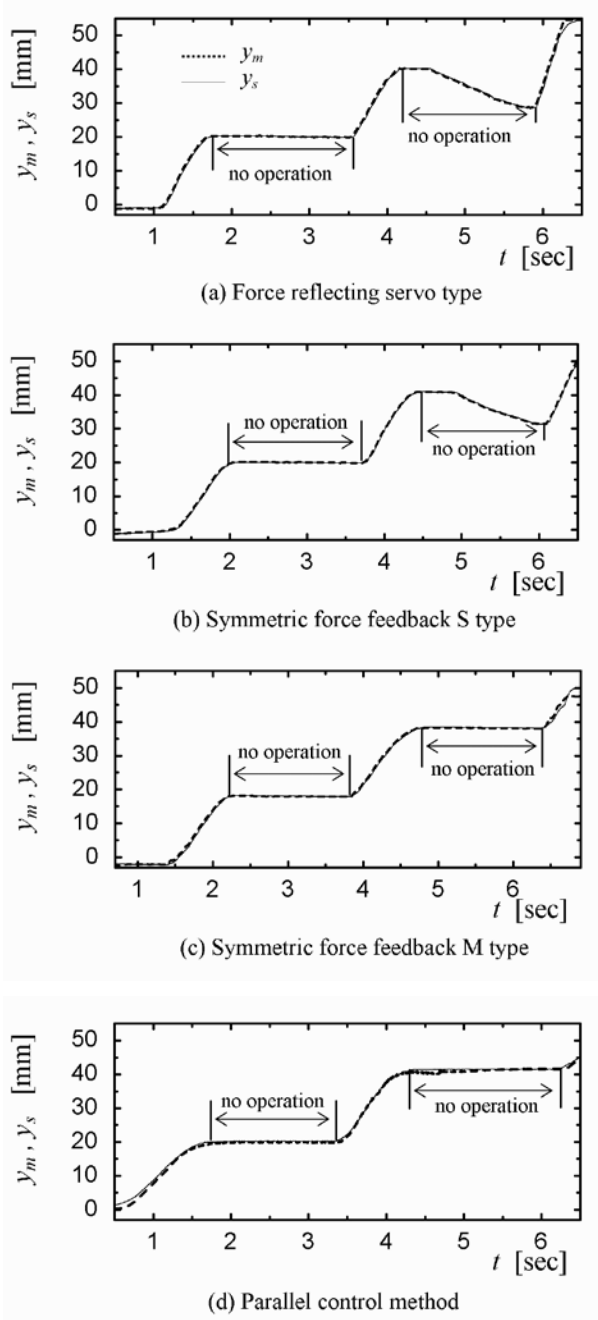


Fig.10: *Stationary performance test*

In the case of FRS, SFF-S and SFF-M, the open-loop transfer functions from the input f_{m} to the output f_{s} become 1-type controller with an integrator (1/*s*). Thus it can be considered that f_{s} follows f_{m} without any steady-state error. Therefore we adopt PD controller because an integrator is not required in the system. Here, a derivative control “D” is added in order to stabilize the system. On the other hand, open-loop transfer function of PC does not include an integrator. For this reason, PID controller is adopted as the position command generator.

When comparing the control performance of the four control systems, it is desirable to standardize the control force to the master. Therefore, instead of the operator's control force, a step signal of 0.6 V (equivalent to 5 N) is input to the master controller. In order to standardize the controllability of each system, this step signal is fed to the slave without mounting the spring load, and the gains of each controller are adjusted in such a way that the piston speed is the same for all of the systems. That is, for a comparison of the drivability of each system, the gains are adjusted in an equal piston speed under the same operational force. In the experiment, however, it is difficult to adjust gains of each controller in such a way that the piston speed is precisely the same. The major cause for it is estimated that the frictional force acting on the cylinder is not constant.

Thus we adjust the gains to equalize the piston speed as precisely as possible. The results of the step response experiments with a soft spring (k = 0.4 kN/m) and with a hard spring (k = 3.0 kN/m) are shown in Fig. 8 and 9, respectively. The sub figures (a) to (d) correspond to the FRS, SFF-S, SFF-M and PC methods, respectively. In the sub figures, the solid lines represent the piston displacement of the master and slave, and the broken lines indicate the slave piston displacement at the moment the slave makes contact with the spring.

First, the results for the soft spring, as shown in Fig. 8, look very similar. This means that stable contact task is generally achieved. Note that the slave force f_s shows a rather good follow-up to the command force of the master f_m, and the follow-up of the slave displacement y_s to the master displacement y_m is also good.

Figure 9 shows the results with a hard spring, revealing marked differences in control performance between the different control methods.

In the FRS method (case (a)), the slave force f_s and displacement of both pistons (y_m, y_s) show heavy oscillation after the contact with the spring. The existence of phase lag or dead time in the pneumatic or hydraulic system is considered as the major cause of the oscillation. In the case of FRS method, the master and the slave are connected in a series. Therefore the phase lag or dead time in the system is amplified. As a result, the system is apt to be unstable. If the system has a high responsiveness it is expected that the system would be stabilized even FRS method is adopted. As a viewpoint of the controller design, a low gain should be chosen in order to stabilize the system. In this case, however, drivability of the system becomes worse. In the SFF-S method (case (b)), although some constant deviation appears in both piston displacement (y_m, y_s) and the slave force f_s, the degree of instability seen in case (a) does not appear. In the SFF-M method (case (c)), the piston displacements and the forces of the master and the slave agree well with each other, although some overshoot can be observed. In the PC method (case (d)), a stabilizing tendency similar to that in case (c) can be observed. A PC system, however, requires a position command generator (a PID controller) in addition to the master and slave units. Therefore, more gain parameters are necessary than with the other systems, and it would be troublesome to adjust the gain for the PC system. All of the above experimental results are obtained by inputting to the master a step signal generated by a computer instead of an operator.

We then conducted a series of contact experiments using a real operator, producing almost the same results. The contact experiments for the FRS method with a hard spring show that the system tends to become unstable. The SFF-S, SFF-M and PC methods produce the same control stability. In these three cases, each haptic display gives the operator a fairly good sense of contact, allowing him to feel the hardness or softness of the spring. In the experiments involving a real operator, the different control methods yield different position-keeping qualities of the pistons during the no-control period. Additional experiments were therefore conducted on the behaviour of both pistons during the no-control period, repeating the control on and off, with the spring load of the slave eliminated.

The results are shown in Fig. 10(a) to (d). (Note that in all cases the curves of y_m and y_s are almost overlapping.) Figure 10(a) and (b) refer to the FRS and SFF-S methods. The piston displacement shows some drift in the vicinity of 40 mm. The pneumatic servo system used in the experiment is composed of a low-friction type cylinder. Although this cylinder suffers some air leakage, the leakage rate appears to depend on the position.

On the other hand, as stated in section 2, a constant electric signal (offset voltage) is given to the servo valve. This offset voltage is adjusted according to the origin of the piston (the mid-point of the piston stroke), so the set value of the offset voltage might become inappropriate when the piston is far from the origin, causing the master piston to drift even in the no-control state. The major cause is estimated that leakage rate changes depending on the position because of a processing accuracy of the cylinder. In pneumatic systems generally, there is a difficulty that the offset value changes with the passage of time or disturbances.

In the FRS and SFF-S systems the slave is controlled to follow the master. Therefore, the slave also begins to drift. Figure 10(c) shows that in the SFF-M system, where the master is controlled to follow the slave, the position-keeping quality of both pistons is good. The result for the PC system is shown in Fig. 10(d). In this case, the position command is generated from the force difference between master and slave, and the position is always controlled with regard to this position command. Both the master and the slave are position-controlled to respective position commands generated just an instant before. Hence the pistons remain still, as shown in the figure.

Using the pneumatic-hydraulic master-slave control system developed in this research, we conducted an extensive, comparative experimental investigation of four different drive control methods. The results of the evaluation are summarized as follows.

(1) We evaluated the elements of controllability, such as the follow-up performance of position and force, system stability during contact with a spring load, and the position-keeping of pistons during the no-control period. It is concluded that SFF-M and PC were superior to the other two methods (FRS and SFF-S), and the performance of SFF-M is the same as that of PC.

(2) The PC system is associated with an unavoidable increase in work necessary to adjust the gain.

(3) The symmetrical force feedback M type (SFF-M) bilateral control method is considered the most effective.

5 Conclusion

In the present study, a pneumatic haptic force-display mounted with a piston drive method using a force signal was developed as the first step. We conducted experiments with this display, in which a virtual spring was used as a work load, and confirmed that the display presented the force effectively. We then manufactured four master-slave control systems composed of the present display and a hydraulic servo system. Each system applied one of four typical bilateral control methods (FRS, SFF-S, SFF-M, and PC). We then investigated the force presentation performance of each method by repeating contact trials on a spring load. The SFF-M control method was judged to be the best among the four with respect to control factors such as force presentation performance, system stability, position-keeping performance, and ease of gain adjustment. The pneumatic system, which is adopted in this study, is quite general one and thus it is expected that the same result as the study would be reproducible with the similar pneumatic system.

In conclusion, we have successfully developed a pneumatic force-display that is less expensive than the hydraulic-driven type described in our previous report.

In this study, we conducted a mostly experimental investigation. However, an analytical basis to go with their experimental exposition is quite important. Especially, dynamical passivity is a powerful tool for the analysis of coupled stability problems arising in haptic force displays. And the passivity theorem ensures that a passive system can interact stably with any strictly passive system (Arimoto, 1996). For example, Colgate used passivity techniques in the design of haptic interfaces to virtual environments (Colgate, 1993). Li also proposed a passive control scheme for the bilateral teleoperation of a one degree of freedom electro hydraulic actuator (Li, 1998). As a future work, we plan to apply passivity theorem for the pneumatic force-display, which is developed in this study. And the analysis of coupled stability problems will be conducted. In addition, we will develop a 6-DOF parallel-link type haptic display driven by six parallel-connected pneumatic cylinders.

Nomenclature

f_e	reaction force from virtual spring	[N]
f_m, f_s	force of master and slave	[N]
f_{op}	force from operator	[N]
k	spring coefficient	[N /m]
k_{fm}, k_{fs}	gain for force sensor of master and slave	[V/N]
p_{sp}, p_{sh}	supply pressure of pneumatic and hydraulic system	[Pa]
x_p	flapper displacement	[m]
y	displacement of pneumatic piston	[m]
y_m, y_s	displacement of master and slave	[m]
ζ_p, ζ_h	viscous damping coeffcient of flapper and spool	
ω_{np}, ω_{nh}	natural frequency of flapper and spool	[rad/s]

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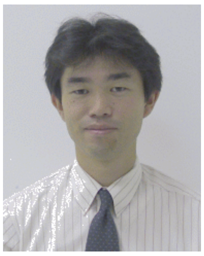

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