

A PRECISION DRIVING SYSTEM COMPOSED OF A HYDRAULIC CYLINDER AND HIGH-SPEED ON/OFF VALVES

Sojiro Tsuchiya¹, Hironao Yamada² and Takayoshi Muto²

¹DENSO Corporation, 1 shindo, takatana-cho, Anjyo, Aichi 446-8507, Japan

²Department of Mechanical & Systems Engineering, Gifu University, 1-1 Yanagido, Gifu, Japan
muto@cc.gifu-u.ac.jp

Abstract

In manufacturing technology, the predominant tendency in recent years has been for machine tools, for example, turning-, milling-, and drilling-machines, to employ electrically operated actuators such as a servo-motor equipped with a ball screw. There are, however, various problems with these electric driving systems; they are excessively large-sized with complex machinery, and their application is expensive, as seen, for example, in the case of the NC-machine. In order to solve these problems, this study aims to develop a precision driving system actuated by a hydraulic cylinder. The hydraulic driving system consists of a cylinder and four ON/OFF solenoid valves. The valves are the same as those used in a fuel injector of an automobile, which are capable of high speed switching, as fast as 1.5 ms. It was confirmed in experiments that the developed system had a moving resolution of 1.2 μm and, as a result, was applicable to a precision driving table for micro-processing.

Keywords: hydraulic driving table, on/off control system, on/off valve, precision processing, micro-processing machines

1 Introduction

In manufacturing technology, the predominant tendency in recent years has been for machine tools, for example, turning-, milling-, and drilling-machines, to employ electric operated actuators such as a servo-motor equipped with a ball screw. There are, however, various problems with these electric driving systems; they are excessively large-sized with complex machinery, and their application is expensive, as seen, for example, in the case of the NC-machine. In order to solve these problems, this study aims to develop a precision driving system actuated by a hydraulic cylinder. The advantages of hydraulic actuators are, in general, that they offer high power-density, high system-rigidity, high speed, and have a simple constitution. Incorporation of the hydraulic actuator is expected to produce a system not only of high resolution and low cost, but one of small size, adequate for application to micro-processing machines. To develop such a new table driving system is the objective of this study.

The simplest type of ON/OFF valve was adopted in this study for controlling the driving system. The valve expresses a 2/2 function that opens or closes a hydraulic line. Such ON/OFF function as a discrete

electro-hydraulic converters is robust, very simple in design and operation and therefore provides a valve which is cheaper and more reliable than servo valves (Mansfeld, 1981). In order to apply this kind of ON/OFF valve to the present table driving system, however, the valves need to be capable of fast switching and precise flow control. An appropriate valve that meets such requirements was selected in this study from those used in a particular application: as fuel injectors for automobile engines.

The hydraulic driving system to be developed consists of a cylinder and four ON/OFF valves. The basic design of this system is the same as the representative one used in conventional applications (Wennmacher, 1993). In this system, furthermore, as a method for driving the ON/OFF valves, a pulse width modulation (PWM) method similar to those used in conventional applications (Hesse, 1972) was adopted.

There are serious problems to be solved in a micro-displacement table system operated by ON/OFF valves. The first problem is how to minimize the moving resolution of the table (Mansfeld, 1978); here, the moving resolution means the table displacement per one pulse of input signal. The second is how to realize a smooth waveform for the displacement, under the operating condition of the ON/OFF control (Muto, 1990). The third is to reduce the oscillations and noises induced in

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the system to the lowest possible levels. To solve the first problem, a fast switching ON/OFF valve is installed in the system. Resolution of the second problem is essential in dealing with an ON/OFF control system, and, up to the present, it has generally been perceived as inevitable that the system lapses into a stepwise or discontinuous motion. In order to overcome this problem, a method for generating a smooth waveform of motion is proposed in this study.

2 System Constitution and ON/OFF Valves

2.1 System constitution

The system for the present study is, as shown in Fig. 1, composed primarily of four ON/OFF solenoid valves, a hydraulic cylinder, and a driving table. The constitution of this system is the same as one in conventional applications (Wennmacher, 1993). In this study, however, the open-loop type without a feedback loop is applied to the basic investigation of a hydraulic micro-displacement table system.

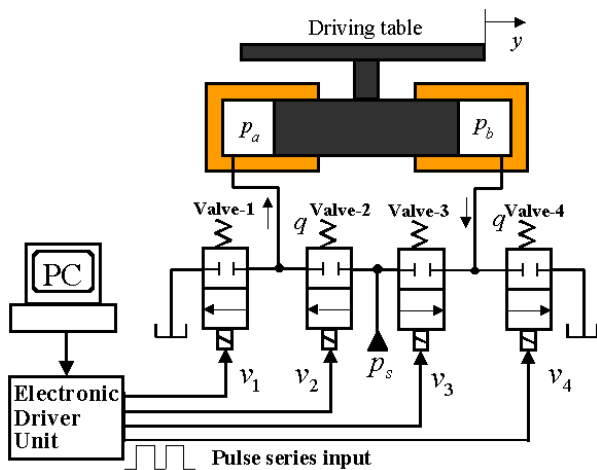


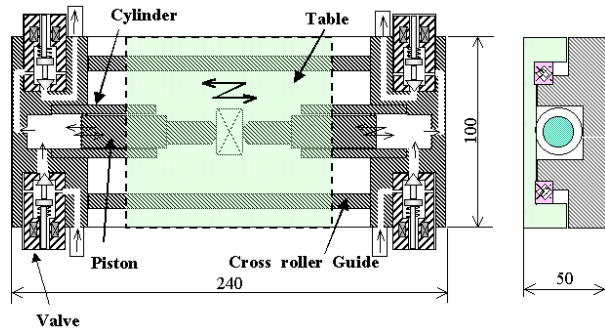
Fig. 1: Schematic diagram of hydraulic driving system

Four ON/OFF valves, valve-1 to valve-4, are driven by electric signals v_1 to v_4 . Each signal of v_1 to v_4 is generated by a personal computer (PC) as a waveform of a pulse-series, then sent to the EDU (Electronic Driver Unit). A driving method (proposed by Sato, 1993) for improving the switching speed of the valves was installed in the EDU. Here, the width of the pulse-series, and each phase difference among the four signals, can be changed arbitrarily by the PC.

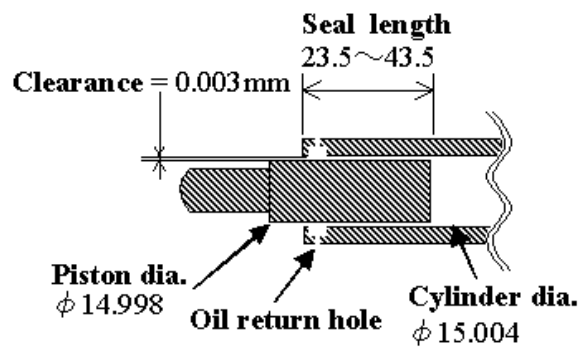
The drawings of the hydraulic driving system are shown in Fig. 2(a) and (b), where Fig. (a) is a general view, and Fig. (b) is a detailed drawing of the piston and cylinder. As is shown in Fig. 2(a), the table attains a dimension about the size of type A-4 paper.

The cylinder employed in this study is of a plunger type produced by a high precision technique of metal cutting, which was newly designed so that a leakage flow could be fully shielded without using any sealing materials. The other approach for designing this system is that two plungers are set facing each other, in such a

manner that the table stay is inserted between them, as illustrated in Fig. 2(a). The reason for this design is mainly to exclude O-rings, since this type of sealing can be an impediment to develop a micro-displacement table.



(a) General view of the driven table



(b) Detailed drawing of piston and cylinder

Fig. 2: Drawings of the hydraulic driving table system

Next, the principle of the system operation is demonstrated for the case in which the driving table moves in the positive direction of the y -coordinate. Initially, the system is at rest at the position $y = 0$, with all the valves closed. Subsequently, two input signals v_2 and v_4 are sent to valve-2 and valve-4, respectively, in order to charge and discharge flow rates to/from the cylinder. Here the definition of angle θ is introduced as the phase difference between the two phase angles of the signals v_2 and v_4 . This parameter θ plays an important role in this study for attaining a smooth feed of the table system, as discussed later. After receiving the input signals, the valves start to act in an ON/OFF-motion, then the piston displaces in accordance with the number of ON-pulses. The main parameters of the system are given in Table 1.

Table 1: System parameters

Table and cylinder	Diameter of cylinder = 15 mm Stroke-length of table = 66 mm Mass of cylinder = 1.32 kg Viscous damping coefficient of piston = 99.6 kNs/m Table size = 100 × 100 mm
Other parameters	$\rho = 850 \text{ kg/m}^3$, $\nu = 7.5 \cdot 10^{-5} \text{ m}^2/\text{s}$ Bulk modulus of fluid = $1.5 \cdot 10^9 \text{ Pa}$ Sampling time = 0.1 ms

2.2 Fast switching ON/OFF valves

A fuel injector (produced by Denso Co. Ltd.; Type D) for use in an automobile is employed in this study as a fast switching ON/OFF valve. The schematic diagrams of the injector are shown in Fig. 3(a) and (b), respectively, to illustrate the constitution and the physical model of the valve.

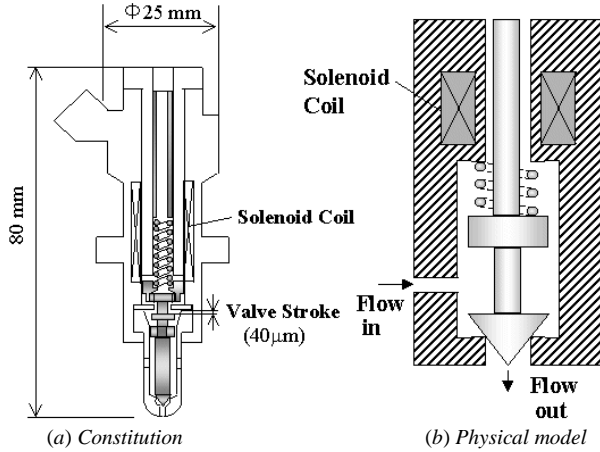


Fig. 3: Schematic diagram of fuel injector of automobile

This injector was selected because conventional ON/OFF valves are not necessarily capable of high speed switching, and precise flow control is not guaranteed. The injector, however, is capable of high speed switching, attaining a speed of about 1.5 ms, and precise flow control is highly guaranteed. In addition to those advantages, the injector is also guaranteed to be hard-wearing and shows no leakage, as demonstrated in its practical use in an automobile engine. The main dimensions of the injector are given in Table 2.

Table 2: Dimensions of ON/OFF valves

Maximum supply pressure = 20 Mpa
Static flow rate = $0.33 \sim 3.3 \cdot 10^{-6} \text{ m}^3/\text{s}$
Diameter of valve seat = 1.5 mm
Valve stroke = 40 μm
Inductance = 4.6 mH
Coil resistance = 2.8 Ω
$t_{\text{on}} = 1.4 \text{ ms}$, $t_{\text{off}} = 1.6 \text{ ms}$

This injector can act as a fast-switching ON/OFF valve, since it injects precisely a required amount of fuel into the cylinders of an engine over a time period of 5 ms for a compression stroke, during high speed rotation of engine at more than 6000 rpm. In order to modify the injector to a hydraulic ON/OFF valve for the present study, a spray nozzle mounted in the injector was removed. In the following, the static and dynamic characteristics of the ON/OFF valve is discussed based on experimental investigations.

Figure 4 shows a transient response of the valve when measured in atmospheric air under condition of no hydraulic fluid. The curve in the figure represents the response of the valve stroke x to a stepwise change in input v . In the response curve, the switching times from the OFF- to ON-state and vice versa are estimated as 1.1 ms and 0.8 ms, respectively; that is, about 1 ms in average.

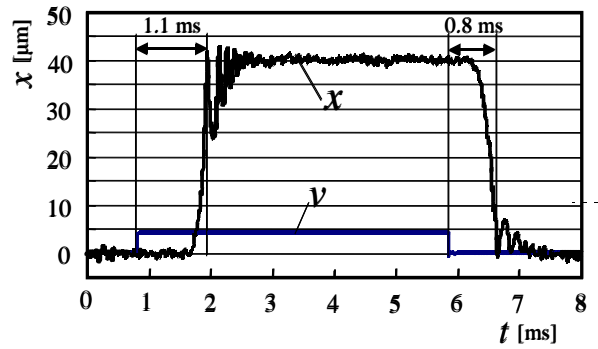


Fig. 4: Transient response of the ON/OFF valve (in air)

A transient response was measured under condition of supplying hydraulic fluid to the valve; the result obtained is presented in Fig. 5.

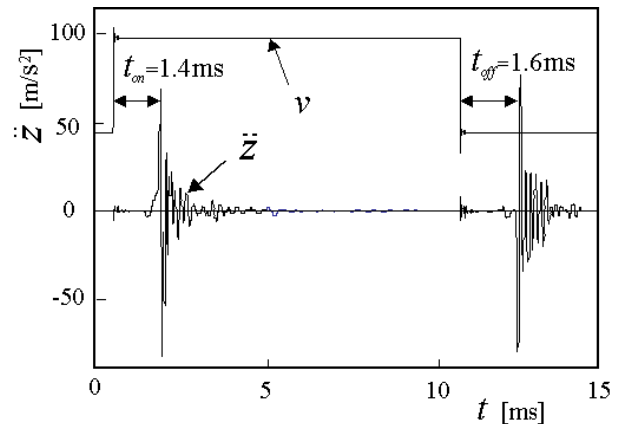


Fig. 5: Transient response of the ON/OFF valve (in oil)

Figure 5 is distinguished from Fig. 4 by the fact that the response was measured in terms of an acceleration sensor, and the acceleration of valve body \ddot{z} is plotted in the figure. The first peak appearing in the response curve corresponds to the instance at which the valve stroke touched the lift-stopper, after the ON-signal was received by the valve. The second peak corresponds to the instance at which the valve stroke touched the valve seat, after the OFF-signal was given. The two response times t_{on} and t_{off} , which are defined as designated in the figure, are measured to be 1.4 ms and 1.6 ms, respectively. This result of response times, about 1.5 ms in average, verifies that the present valve is superior to the conventional valves (Tanaka, 1985).

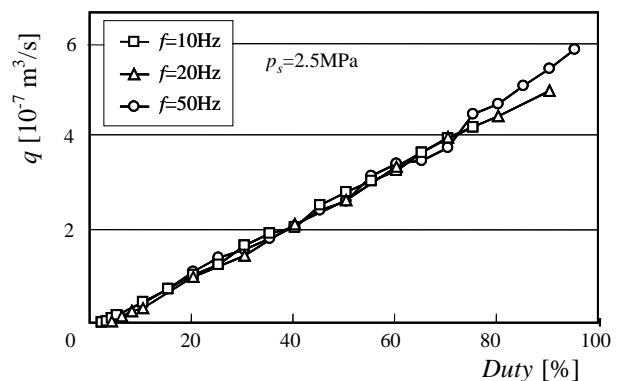


Fig. 6: Valve characteristics of Duty vs. flow rate. A result of further measurement of the static characteristics is presented in Fig. 6.

In the figure, the curves of the Duty ($=T_{on}/T$) vs. discharge flow rate q are plotted, taking the frequency of carrier wave f [Hz] as a parameter. As seen in the figure, the characteristic of Duty-flow rate behaves with a good linearity over a frequency range of 10 Hz to 50 Hz.

3 Dynamic characteristics of the developed table system

The ultimate purpose of this study is to develop a micro-processing machine as an application of the driving table system shown in Fig. 1. Taking this purpose into consideration, this chapter deals with the dynamic characteristics of the system in terms of system performance, including the aspects of feed resolution, smoothness of the table displacement, and the influence of payload on the system. For the sake of simplicity, hereafter, the simplified-model system shown in Fig. 7 is treated, rather than the system shown in Fig. 1. Here, the simplified system is capable of displacing only toward the positive direction of displacement y .

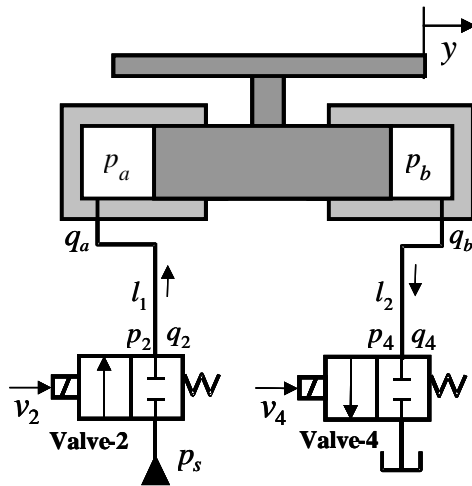


Fig. 7: System constitution for positive displacement of y

3.1 Mathematical model for simulation analysis

In this section, in preparation for the simulation analysis discussed in the next section, we derive a mathematical model of the system. The following assumptions are introduced:

- The supply pressure p_s is constant.
- A functional relationship of the valve between the displacement x and time t is described by the equation comprised of a time lag and a linear stroking (Muto, 1990).
- Each equation of two pipelines (l_1, l_2) for connecting the valves and hydraulic cylinder is described by the equation of a lumped-parameter-system model with one-DOF.
- Leakage flows from each valve can be neglected.

The discharge flow rate q_2 from the valve-2 and the inlet flow rate q_4 to the valve-4 are expressed as Eq. 1 and Eq. 2, respectively:

$$q_2 = c a_{v2} \sqrt{\frac{2|p_s - p_2|}{\rho}} \quad (1)$$

$$q_4 = c a_{v4} \sqrt{\frac{2|p_4|}{\rho}} \quad (2)$$

where the opening area a_{vi} ($i = 2, 4$) is (Merrit, 1967):

$$a_{vi} = \pi d x_i \sin \alpha \quad (3)$$

The equations of motion for the oil-columns in two pipelines l_1, l_2 are written by four equations as follows:

$$\frac{dq_a}{dt} = \frac{a_1}{\rho l_1} (p_2 - p_a) - R_1 q_a \quad (4)$$

$$q_2 - q_a = \frac{a_1 l_1}{K} \frac{dp_2}{dt} \quad (5)$$

$$\frac{dq_b}{dt} = \frac{a_2}{\rho l_2} (p_b - p_4) - R_1 q_b \quad (6)$$

$$q_b - q_4 = \frac{a_2 l_2}{K} \frac{dp_4}{dt} \quad (7)$$

where the frictional coefficient of viscosity R_i ($i = 1, 2$) is given by the equation:

$$R_i = \frac{8\nu}{d_i^2} \quad (8)$$

From the equation of continuity for describing the flow rates in two chambers of the cylinder, the following two equations are obtained:

$$q_a = a_p \frac{dy}{dt} + \frac{(v_{a0} + a_p y)}{K} \frac{dp_a}{dt} \quad (9)$$

$$q_b = a_p \frac{dy}{dt} - \frac{(v_{b0} - a_p y)}{K} \frac{dp_b}{dt} \quad (10)$$

The equation of motion for the piston can be obtained by Eq. 11:

$$m \frac{d^2 y}{dt^2} + b_p \frac{dy}{dt} + f_p = a_p (p_a - p_b) \quad (11)$$

where frictional force f_p is represented by (Muto, 1992):

$$f_p = \begin{cases} f_d \operatorname{sgn}(\dot{y}) & ; \dot{y} \neq 0 \\ f_s \operatorname{sgn}(f_a) & ; \dot{y} = 0, |f_a| > f_s \\ f_a & ; \dot{y} = 0, |f_a| \leq f_s \end{cases} \quad (12)$$

and f_a is given by:

$$f_a = a_p (p_a - p_b) \quad (13)$$

The frictional model expressed by Eq. 12 is illustrated in Fig. 8.

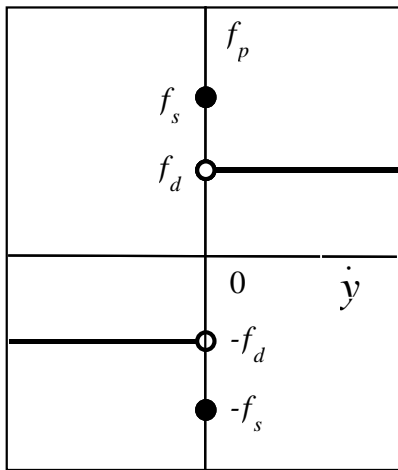


Fig. 8: Frictional model of Eq. 12

3.2 Results of experiments and simulations

First, the dynamic characteristics of the driving table were investigated experimentally under two operating conditions of the start and the stop. In this experiment, the two ON/OFF-signals v_2 and v_4 were given to valve-2 and valve-4, respectively, with the equivalent phase angle; that is, $\theta = 0^\circ$. Two response curves for the table displacement y are shown in Fig. 9(a) and (b) corresponding to the start- and stop-operations, respectively.

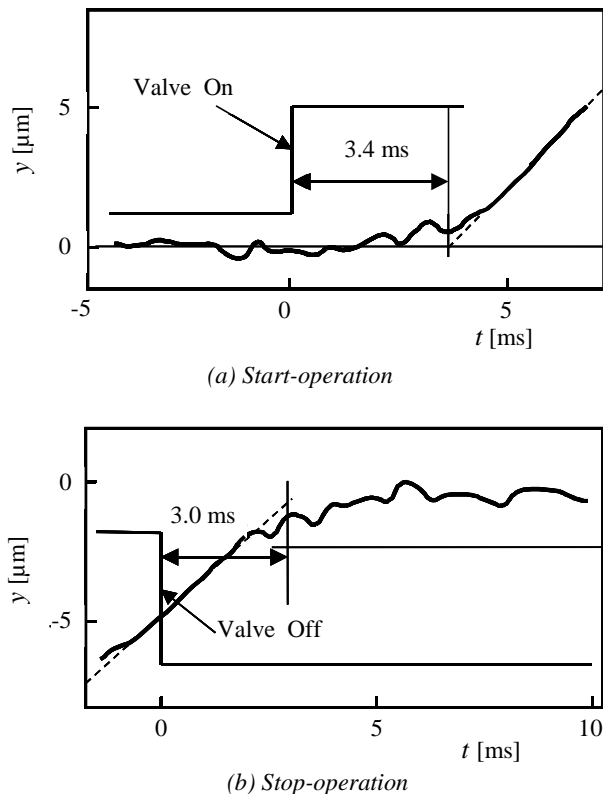


Fig. 9: Response curves y in start/stop-operations

As seen in the figures, each response curve shows a ripple motion with amplitude of about $9 \mu\text{m}$. This ripple was confirmed as a noise generated in the measuring system for sensing the displacement y . Since this

magnitude of noise is of the same order as those frequently encountered in the electric driving systems typically employed (e.g., in NC-machines), the noise can be regarded as of negligible order for the present study. For this reason, in the remainder of this paper, the existence of the ripple observed in Fig. 9 is neglected. In Fig. 9(a) and (b), the delay times of the system response are estimated as about 3.4 ms and about 3.0 ms for the start- and the stop-operation, respectively. These delay times are considered to be the sum of the delay time of the valves (about 1.5 ms), and that of the remaining parts composing the present system. As a consequence of the results presented in Fig. 9, it can be concluded that the system developed has a high-speed response for applying to a micro-processing machine appropriately.

Next, the system was investigated experimentally in terms of the dynamic characteristic of micro-displacement, and the result obtained is shown in Fig. 10.

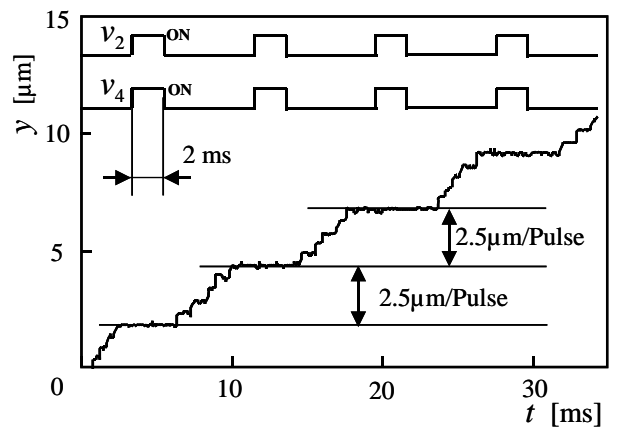


Fig. 10: Table displacement y by ON/OFF pulse wave

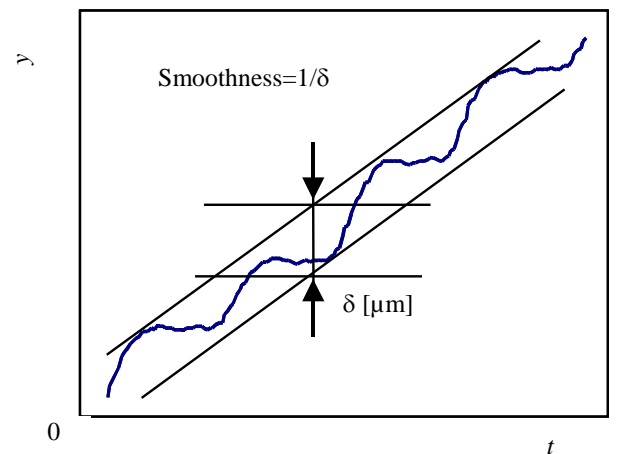
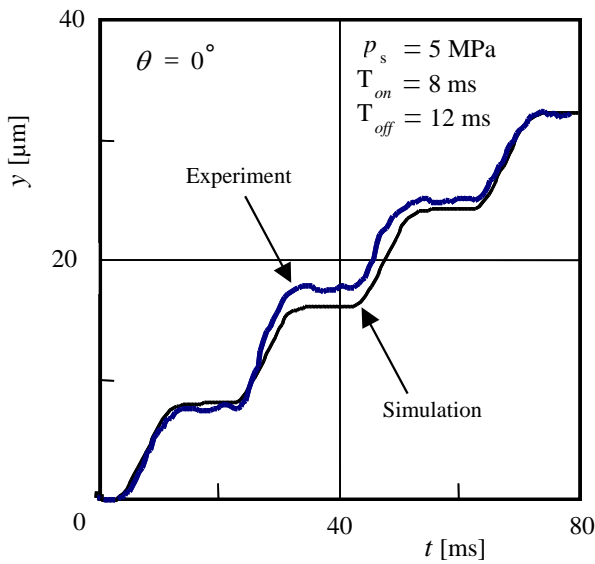


Fig. 11: Definition of smoothness

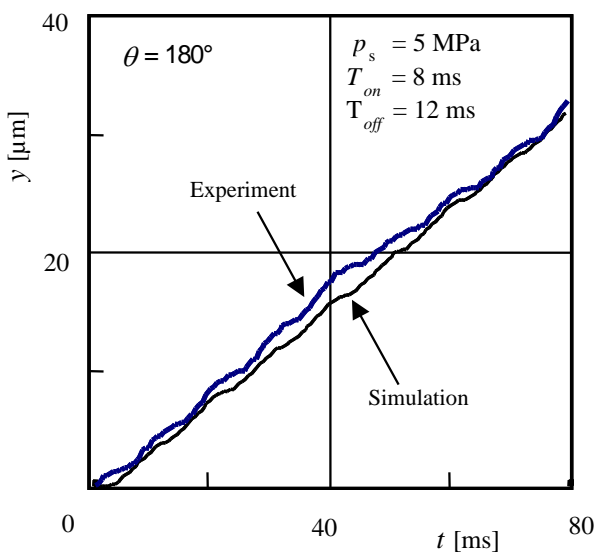
In this experiment, as shown by the waveforms of input signals v_2 and v_4 in the figure, the system was driven by a pulse series with the equivalent width of the ON-pulse ($T_{\text{on}} = 2 \text{ ms}$) at pulse-period $T = 16 \text{ ms}$ and phase difference $\theta = 0^\circ$. The response to this input is plotted in the figure as a response curve of the table displacement y against time t . In this experiment, because the magnitude of T_{on} is set at a comparatively small value

such as $T_{on} = 2$ ms, the displacement y per one pulse has a minute value such as about $2.5 \mu\text{m}$. From the observation of Fig. 10, it can be said that the waveform of the y - t characteristic, if the ripple, as stated earlier, is neglected, approximately reveals a stepwise change. These results indicate that the system has a good feed resolution, of about $2.5 \mu\text{m}$, which is sufficient for the micro-displacement table in the present study. On the other hand, the stepwise waveform seen in the figure can be an undesirable factor in realizing a smooth feed of the table, which is usually important in the case of the cutting machine. In the following, therefore, a method for realizing a smooth feed of the table is investigated. Here the smoothness of displacement y is defined by the factor $1/\delta$, as depicted on the y - t characteristic shown in Fig. 11, and δ is called hereafter the "fluctuation".

It was found in this study that the stepwise waveform as seen in Fig. 10 can be remarkably smoothed out.



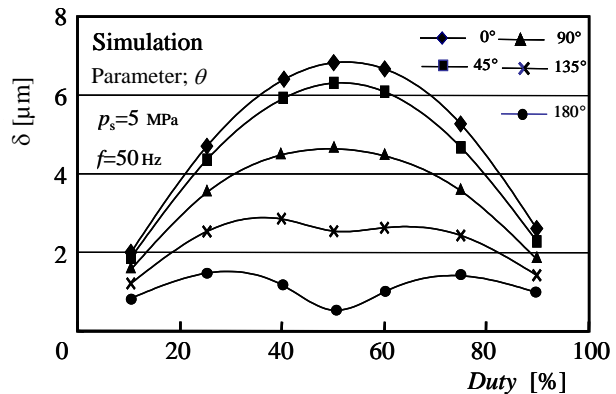
(a) $\theta = 0^\circ$



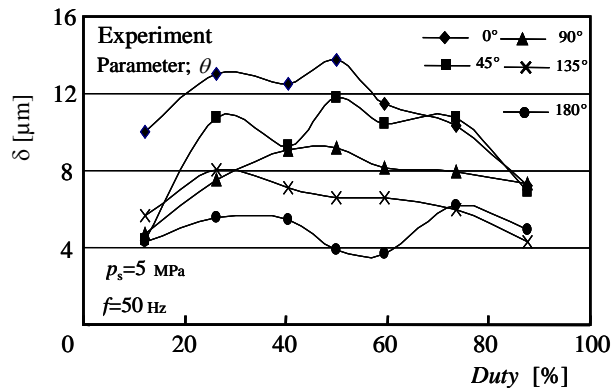
(b) $\theta = 180^\circ$

Fig. 12: Time response y with phase shift θ was clarified by experiments and simulations that the stepwise waveform can be smoothed out by moderately adjusting the phase difference θ , as shown in Fig. 12. In Fig. 12(a) and (b), two relationships of y - t characteristics are plotted; for $\theta = 0^\circ$ and $\theta = 180^\circ$, respectively. From the comparison of these two figures, it is clear that a stepwise waveform as seen in Fig. 12(a) can be remarkably smoothed out by changing the phase difference from $\theta = 0^\circ$ to $\theta = 180^\circ$. Further, given that the simulated model properly represented the experimental results, the validity of the simulation analysis can be confirmed.

Subsequent discussion is made on how to optimize the phase difference θ in order to minimize the fluctuation δ . A variation of the fluctuation δ with respect to $Duty$ were investigated by simulation, and the result obtained is presented in Fig. 13(a) by taking the phase difference θ as a parameter.



(a) Simulation



(b) Experiment

Fig. 13: Characteristics of fluctuating width δ vs. Duty

The figure clearly shows that the fluctuation δ takes the minimum value at $\theta = 180^\circ$. The experimental result corresponding to this simulation is presented in Fig. 13(b). By comparing Fig. 13(a) and Fig. 13(b), it can be seen that each experimental curve is nearly twice the simulated one, though the curves agree qualitatively. The main reason for this disagreement seems to be that the experimental result of displacement y was, as shown, for examples, in Fig. 12(b) and Fig. 20(b), measured as a waveform including a slight degree of unsteadiness, influenced by the following three factors. The first is electric or optical noises infiltrating

the measuring system. The second is a pulsating disturbance that was generated in the pump and, as a result, the disturbance was interfered with the valve motion. The third is that two frictional forces acting on the table and piston are not necessarily constant with the variation of displacement y .

In order to investigate for a mechanism of forming the smooth waveform of displacement y , the pressure variations of p_a , p_b in two cylinder chambers were measured together with the displacement y . The results obtained for the two cases $\theta = 0^\circ$ and $\theta = 180^\circ$ are presented in Fig. 14 and Fig. 15, respectively.

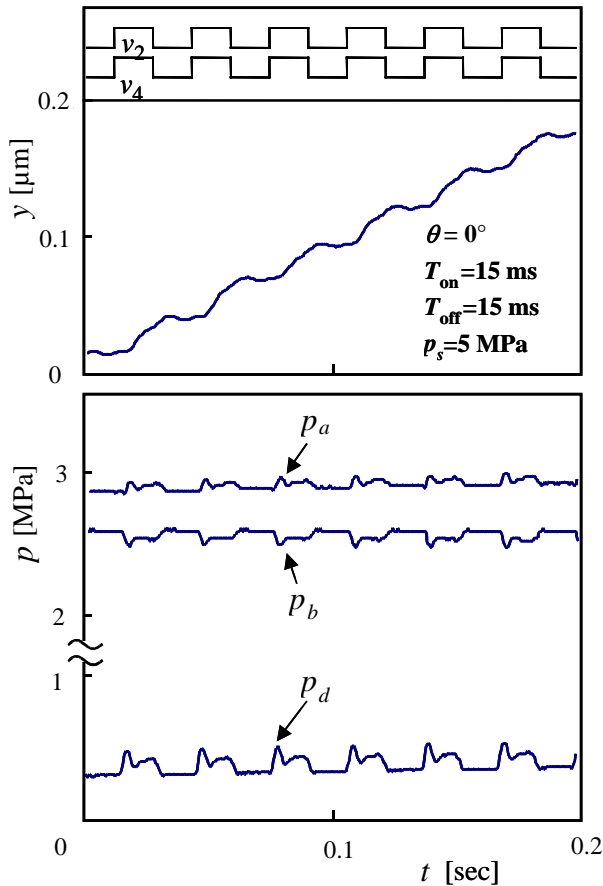


Fig. 14: Time response of displacement y and pressures p_a , p_b , p_d in cylinder ($\theta = 0^\circ$)

In the figures, the curves of y - t characteristics are plotted, along with the pressure variations of p_a , p_b and the pressure difference $p_d = p_a - p_b$. As seen in Fig. 14 for the case $\theta = 0^\circ$, the waveform of the pressure p_d changes its shape significantly with time t , in such a manner to cause a stepwise waveform of the displacement y . On the other hand, in Fig. 15 for the case $\theta = 180^\circ$, the change in the pressure p_d remains at a diminutive level, due to the effective cancellation that results from almost the same magnitude of variations of pressures p_a and p_b .

As a consequence of this diminutive change in the pressure p_d , the smoothed waveform of y as seen in the figure was realized. It can be concluded from this result that the stepwise waveform, previously considered inevitable in ON/OFF control systems, can be smoothed out remarkably by adjusting the phase difference θ at 180° . Such a system set at $\theta = 180^\circ$ is hereaf-

ter called the Out-of-phase system.

Based on the y - t characteristic shown in Fig. 15, which is under the condition of $T_{\text{off}} = T_{\text{on}} = 15$ ms, the fluctuation δ was measured to be $4.6 \mu\text{m}$. From an attempt to further decrease the fluctuation δ , $\delta = 1.2 \mu\text{m}$ was obtained at $T_{\text{off}} = T_{\text{on}} = 2$ ms.

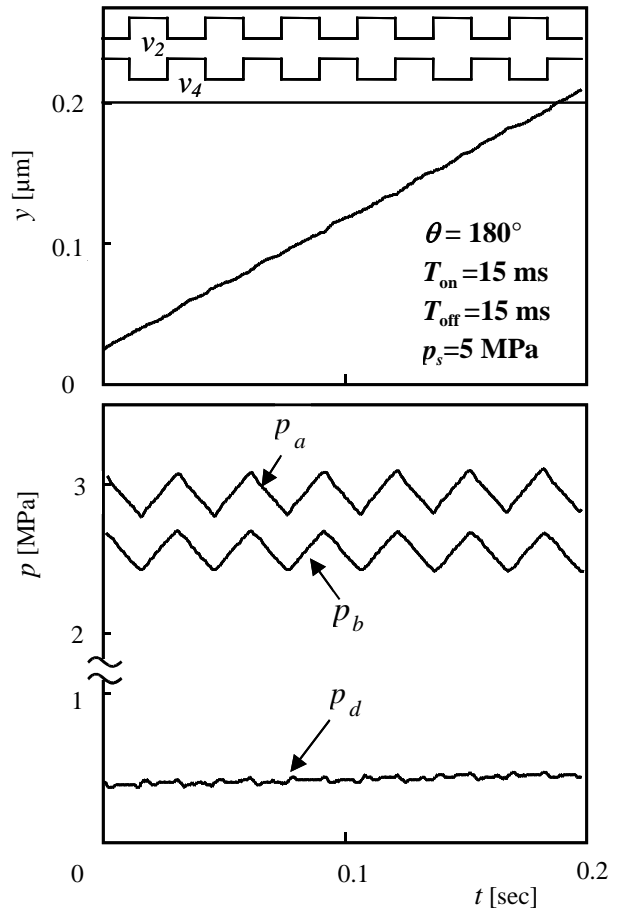


Fig. 15: Time response of displacement y and pressures p_a , p_b , p_d in cylinder ($\theta = 180^\circ$)

A dynamic characteristic of the system, with respect to the displacement y per one-pulse, was measured in relation to the magnitude of T_{on} . Thus obtained curves under two conditions of $\theta = 0^\circ$ and $\theta = 180^\circ$ are plotted in Fig. 16.

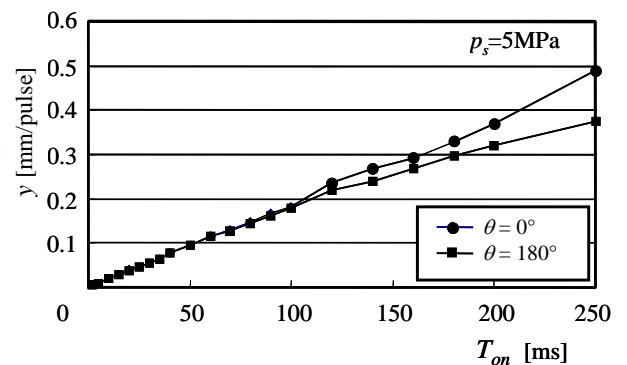


Fig. 16: Table displacement y per one-pulse

The figure shows that the curves for $\theta = 0^\circ$ and $\theta = 180^\circ$ behave with a good linearity, at least in the region of T_{on} smaller than 100 ms, and are plotted by almost the same linear curve. From this result, it was con-

firming that the characteristic of table displacement/pulse vs. pulse width T_{on} behaves with a good linearity in both the In-phase and Out-of-phase systems.

As an experimental result for the In-phase system, the velocity of table displacement \dot{y} vs. *Duty* is presented in Fig. 17.

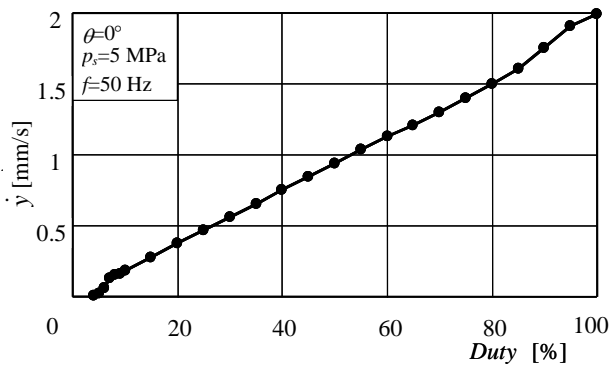


Fig. 17: Characteristics of table velocity \dot{y} vs. *Duty*

As seen in the figure, the obtained curve reveals a good linearity in a fairly wide region of *Duty*.

A characteristic curve similar to those in Fig. 17 was measured (not indicated here) for the Out-of-phase system, and, in this curve, the lower limit of T_{on} up to which the linearity held was estimated as $T_{on} = 2$ ms. Under this pulse condition of $T_{on} = 2$ ms, in addition, the moving resolution of the table per pulse was measured to be $2.5 \mu\text{m}$. Further, by a measurement below the linear region, it was confirmed that the minimum value of T_{on} able to drive the table was 1.6 ms, and the moving resolution was then estimated as $1.2 \mu\text{m}$. Subsequent to the fact that the moving resolution was sustained at such values of $2.5 \mu\text{m}$ and $1.2 \mu\text{m}$, it can be said that the present system is sufficient for application to a micro-processing machine.

Next, the influences of payload on the system performance are discussed based on four experimental results regarding characteristics of payload F_L vs. table-velocity \dot{y} , static rigidity, dynamic compliance, and the influence of phase difference θ on the smoothness of table movement. First, the characteristics of F_L vs. \dot{y} shown in Fig. 18 correspond to two conditions of the In-phase and Out-of-phase system.

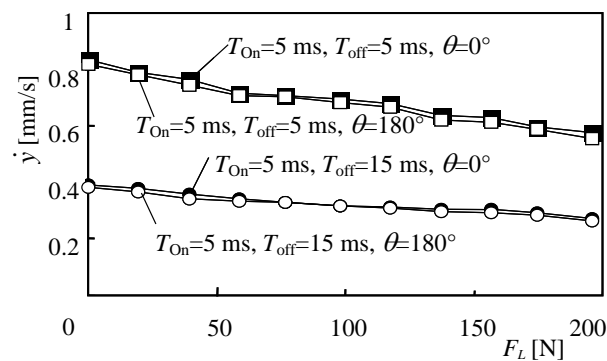


Fig. 18: Characteristics of payload vs. table velocity

In this experiment, the payload F_L was determined by adding weights up to 200 N, taking into consideration that the driving force necessary for the micro-

processing machine is at most 50 N. From the figure, it is observed that the variation of table velocity is less than 10 % in the region of payload between 0 to 50 N. This result indicates that the present system can be driven with a fair degree of stability in spite of the variation of payload. Second, the static rigidity of the system was investigated by measuring the table displacement from the stationary position where all the valves were closed, under the condition that the static payload F_L was provided at various levels. The result showed that the force F_L was approximately proportional to the table displacement. From this relation, the obtained static rigidity was $18.5 \text{ N}/\mu\text{m}$. On the other hand, the static rigidity of an electric table system of the ball screw type with a similar size was measured to be $6.02 \text{ N}/\mu\text{m}$. Therefore, it can be said that the static rigidity of the present system is approximately three times larger than the electric system. Thirdly, the dynamic compliance of the system was measured by applying an oscillatory load with a frequency of f_0 Hz, by using a vibration testing machine. The obtained result is shown in Fig. 19 together with that obtained using the corresponding electric table system.

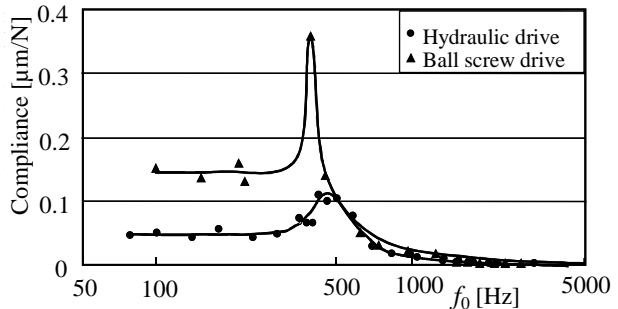


Fig. 19: Dynamic compliance of two driving tables

As demonstrated by the figure, it is estimated that the dynamic rigidity of the present system is, as a whole, superior to that of the electric system.

Finally, the influence of payload on the smoothness of table movement was investigated, and the obtained results are shown in Fig. 20(a) and (b), taking the payload F_L as a parameter. In the figures, the characteristics of displacement y vs. time t are represented in Fig. (a) and Fig. (b), corresponding to two phase angles of $\theta = 0^\circ$ and $\theta = 180^\circ$, respectively. By comparing these two figures, it can be said that the effect of the phase angles θ on the smoothness $1/\delta$ is qualitatively independent of the payload.

Additional experiments, by means of sensory tests, were performed with respect to noises and oscillations induced in the system. These tests revealed the following clarifications. First, such an oscillation was slightly detected in the In-phase system when a tester touched his hand to the driving table, and also on the pipelines connecting the ON/OFF valves. However, the oscillation disappeared completely when the system was changed to the Out-of-phase system. Second, with respect to audible noises, the tester was able to sense only a faint sound from the valves. Consequently, it can be said that the noises and/or the oscillations induced in the present system remain at a slight level, and the

system is sufficiently applicable for practical use.

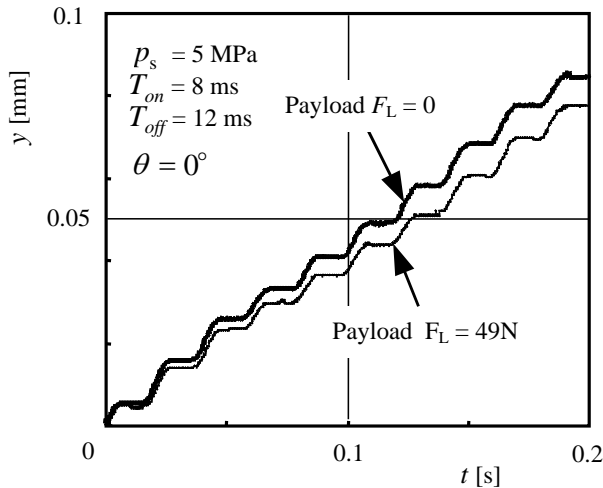


Fig. 20(a): Influence of payload on the smoothness. In-phase system($\theta = 0^\circ$)

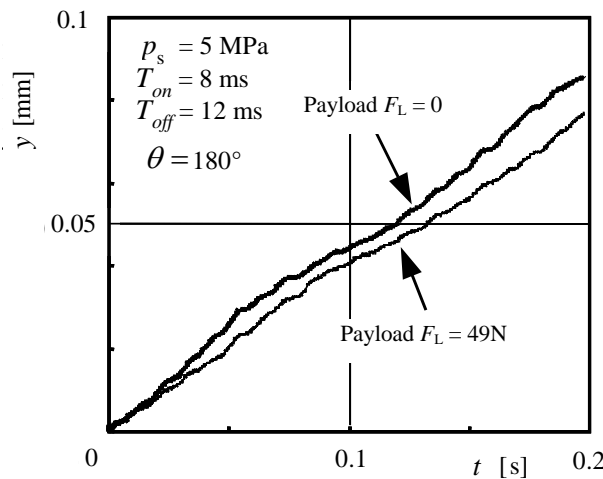


Fig. 20(b): Influence of payload on the smoothness. Out-of-phase system($\theta = 180^\circ$)

4 Conclusions

In manufacturing technology, the predominant tendency in recent years has been for machine tools to employ electrically operated actuators such as a DC servo-motor equipped with a ball screw. There are, however, various problems with these electric driving systems; they are excessively large-sized with complex machinery, and their application is expensive. In order to solve these problems, in this study, a precision driving table system was developed for applying to a micro-processing machine. The developed system consists of four high speed ON/OFF valves and a hydraulic cylinder. In such an ON/OFF-type driving table system, it is essential to realize a smooth waveform of the table displacement. The results of experiments and simulations for investigating the static and dynamic characteristics of the system are summarized as follows:

- The ON/OFF valves employed in the system are the same as those used in a fuel injector of an automobile, and were found to be capable of high speed switching (as fast as 1.5 ms). In the system equipped with these valves, the delay times of the response for the start- and the stop-operations were measured and found to have small values about 3.0 ms. Subsequently, the developed system offers high-speed response, and is applicable to a micro-processing machine appropriately.
- A method for realizing a smooth motion of the system was proposed, by means of adjusting a phase difference between the input signals to the valves. In the system to which this method was applied, it was confirmed that the smoothness of the table displacement was remarkably improved, and the system had a moving resolution of 1.2 μm .
- As a consequence, the developed system can be considered applicable to a precision driving table for micro-processing, and has the advantages of low cost and small size.

Nomenclature

a_i	cross sectional area of pipeline i	[m ²]
a_p	cross sectional area of piston	[m ²]
a_{vi}	opening area of valve i	[m ²]
b_p	viscous damping coefficient of piston	[Ns/m]
c	discharge coefficient of valve	[-]
d	seat diameter of valve	[m]
Duty	input duty to valve(= T_{on}/T)	[%]
f	frequency of PWM-carrier wave ($f = 1/T$)	[Hz]
f_0	frequency of oscillatory load	[Hz]
f_p	dry frictional force of piston	[N]
F_L	payload on the table	[N]
K	bulk modulus of fluid	[Pa]
m	mass of table load	[kg]
p_a, p_b	pressures in two cylinder chambers	[Pa]
p_d	pressure difference (= $p_a - p_b$)	[Pa]
p_s	supply pressure	[Pa]
q_i	flow rate through valve ($i = 1...4$)	[m ³ /s]
R	frictional coefficient of viscosity	[1/s]
t_{on}, t_{off}	response times of valve	[s]
T_{on}	acting time of valve	[s]
v_{a0}, v_{b0}	initial volumes of cylinder chambers a and b	[m ³]
v_i	input signal to valve ($i = 1...4$)	[Volt]
x	valve displacement	[m]
y	displacement of driving table	[m]
\ddot{z}	acceleration of valve body	[m/s ²]
α	half poppet angle	[rad]
δ	magnitude of fluctuation (= $1/(\text{smoothness})$)	[m]
ν	kinematic viscosity of fluid	[m ² /s]
θ	phase difference	[°]
ρ	mass density of fluid	[kg/m ³]

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Sojiro TSUCHIA

(Born May 17, 1949) received the Master of Engineering degree from the Nagoya University Japan, in 1975. He is currently a general manager in Device Product Division, DENSO CORPORATION.



Hironao YAMADA

(Born October 02, 1962) received the Doctor of Engineering degree from the Nagoya University Japan, in 1991. From 1991 to 1994, he worked in Nagoya University. From 1992 to 1993, he was a Visiting Research Fellow at the Aachen Institute of Technology, Germany. He is currently an Associate Professor in the department of mechanical and systems engineering, Gifu University.



Takayoshi MUTO

(Born March 31, 1941) received the Doctor of Engineering degree from the Nagoya University, in 1972. From 1963 to 1972, he worked in the department of mechanical engineering, Nagoya University. From 1981 to 1982, he was a Visiting Research Fellow at the Aachen Institute of Technology, Germany. He is currently a Professor in the department of mechanical and systems engineering, Gifu University.