Characteristics of a Precise Pressure Control Valve for the Force Control System

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Pressure control valves are principal components used for controlling hydraulic fluid pressure, and therefore important for force control system. The operating principle of this valve is simple, that is, the control pressure is balanced with the restorative force of a spring compressed by displacement.

This paper concerns the characteristics of newly developed pressure control valve, designed to overcome drawbacks of conventional pressure control valves, that is;

i) large drain volume, ii) poor noise immunity, reproducigility, and hysteresis, iii) narrow output control range and low resolution, iv) poor responsiveness (Fig. 1).

Test runs revealed the outstanding performance of the prototype offered by its structural and operational features ; 1) changing the spring enhances flexibility is the ratio of control pressure to displacement, ii) null drain is necessarily negligible and pressured hydraulic fluid consumption is as little as the compensation volume for the spool travel, iii) significant linearity and resolution, iv) output pressure is controllable down to nearly zero from supply pressure with a negligible degree of hysteresis, and v) enough response.

The proposed pressure control valve can be used in force control servo systems instead of the ordinary pressure control, especially in fields where the capacity of power source is limited, such as for car brakes, and attitude control systems of aircrafts and space vehicles.

Key words : Pressure control/Servomechanism/Hydraulic control value.

1. Introduction

There are numerous applications in which control of force is required to drive devices. They include, for example, brake systems for high speed trains, drive mechanisms for machine tools, attitude control systems for aircrafts, rockets, and space vehicles. In these applications, hydraulic servomechanisms are often used. But the servomechanisms are the position control system in principle, consequently, from the standpoint of controlling force, the use of a pressure control valve is better suited, bcause in car brakes, for example, only force is to be controlled precicely regardless of the dispqacement of the brake shoes.

Conventional pressure control valves dictates, from its operation principle, that a considerable amount of pressured hydraulic fluid should always escape through the valve. For example, when a relief valve is used, the amount of pressured fluid required by the main cylinder of brakes can be zero or very small to compensate for leakage in static conditions, but pracfical valves can not operate in such conditions unless a considerable relief flow to the valve can be ensured. None the less, the rated control range of many valves of this type is relatively narrow. The presence of this relief flow posess a number of problems, such as, waste of the drive energy of hydraulic power sources, and the overheat of hydraulic fluid and systems. This paper concerns a novel concept of pressure control valve with rapid response without relief flow rate. Its development, experiments, analytical results, and improvements to be made for practical use are described. It can set an arbitrary control pressure with high accuracy over a wide range, being suitably used for force control systems. Considerations given in the development of this valve include ; i) the relief flow rate mentioned above, and leakage flow rate, ii) a change in the control pressure with change in the load flow rate and supply pressure, iii) a controllable operation range,

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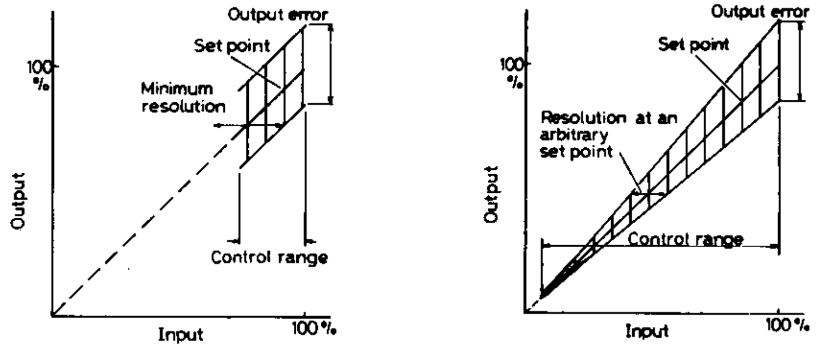


(a) Ordinary pressure control valve. (b) Expected and proposed pressure control valve.

Fig. 1 Comparison of the characteristics of two type pressure control valve.

and iv) responsiveness.

Fig. 1 shows a comparison of the characteristics of an ordinary pressure control valve and the proposed one.

Nomenclature

A	:	section area
a	:	area of the port opening
C_t	:	coefficient of viscus friction
C_t	:	coefficient of flow rate
f_R	:	Coulomb friction
D	:	amplitude
f	:	frequency
F_S	:	spring force
F_C	:	control force
G	:	gain
h	:	thickness of clearance
k	:	spring constant
K	:	bulk modulus coefficient of hydraulic fluid
l	:	length of clearance width
m	:	mass of sleeve and piston mechanism
P_B	:	pressure at the reverse side of the spring support
P_C	:	control pressure
$P_{D/R}$	:	drain/return pressure
P_S	:	supply pressure
$q_{1,2}$	:	flow rate
q_L	:	total leakage flow rate
$q_{l1,2}$	:	flow rate of port
r	:	radius of port
s	:	Laplace operator
t	:	time
U	:	dz/dt
V	:	volume of the chamber
x	:	displacement of spool
x_{set}	:	setting point of operation input
y	:	dispalcement of the sleeve
z	:	opening of the port $(=x-y)$
Δ, δ	:	difference
ρ	:	density of hydraulic fluid
μ	:	viscosity of hydraulic fluid
Θ	:	phase lag

2. Pressure Control Valve Based on New Operating Principle

2.1 Principle of operation

The basic operating principle of the pressure control valve discussed here is such that the control pressure P_C is balanced with spring force F_S $(=ky)$, and it uses two two-way-valves with integrated construction (Fig. 2). Hereafter, marks and symbols are as per Fig. 2.

Now, if the spool is displaced rightward (+) from some settled position x, y, P_C $(=ky/A=F_S/A)$ to $x'=x+\Delta x$, by Δx, shows in the illustration. Then the No. 1 port between the spool and sleeve on the pressure supply side opens, and the pressured oil is fed to the chamber, thereby increasing P_C to $P_C'=P_C+\Delta P_C$. With this, the piston which forms an integrated part with sleeve moves rightward (+) to $y'=y+\Delta y$, compressing the spring, F_S'. This behavior is fed back so as to close the port No. 1, that

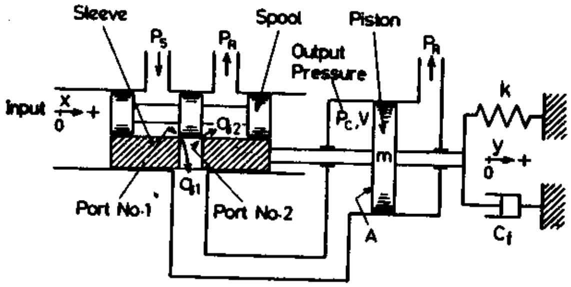


Fig. 2 Structural model of the pressure control valve.

is $(\Delta x-\Delta y)$ to zero. Finally it balances at $F_S'=ky'=k(y+\Delta x)=(P_C+\Delta P_C)A=P_C'A$, thereby shifting to a new settled position, x', y', P_C'. Displacing the spool leftward opens the port No. 2 on the pressure return side and discharges the pressured oil, thereby the same behavior being carried out.

This can be given by the following equation ;

$$F_C=AP_C$$
$$F_S=kx \qquad (1)$$

If we consider the balanced state in which F_C equals F_S, the following equqtion derives :

$$P_C=(k/A)x \qquad (2)$$

From the above discussion, it can be said that this pressure control valve , in principle ;

i) the valve needs no leakage flow rate,

ii) pressured oil in consumped only when Δx is positive direction, and the consumption is as little as $A\cdot\Delta y$ (finally, $\Delta y=\Delta x$),

iii) the control range extends over the entire rated range, (P_C equals (0 to 100 %) $\cdot P_S$), within the range of mechanical constants, such as spring constant, etc.,

iv) almost no force is needed for spool driving,

v) the relationship of P_C/x can be set arbitrarily by changing the spring.

These matters are most desirable features for force/pressure control systems, when we put it into practical use.

2.2 Primitive equation of pressure control valve

This pressure control valve employs two two-way-valves and is constructed as dual-sleeve port valves (refer Fig. 4), so that it connot be helped that leakage flow occures between the spool and sleeves, in the land, and others.

Here, primitive equqtions are derived for analyzing the behavior of the pressure control valve and numerical analysis, and moreover in order to examine the influence of the leakage flow rate. In this section, the following assumptions are made : i) both No. 1 and No. 2 ports are zero lapped and are completely symmetrical, ii) the conduit connecting the port and chamber is sufficiently short and thick, iii) no cavitation occures, iv) the leakage flow rate is produced by clearance flow at all times.

Based on these assumpitons and from Fig. 2, and moreover taking into considearation the structure of the prototype valve (refer Fig. 4), that is, the spring is put in a chamber for damping and reducing friction, two ports are provided symmetrically, the difference in displacement between the spool and sleeve, that is the opening of the port z, is given by,

$$z=x-y \qquad (3)$$

If oil flows in and out through this opening, the flow rate, q_1 and q_2, of the oil that flows into, and out of, the chamber through the port is given by the following equations ;

$$q_1=A\dot{y}+V\dot{P}_c/K-q_{l_1}+q_{l_2} \qquad (z\geqq 0)$$
$$q_2=-A\dot{y}-V\dot{P}_c/K+q_{l_1}-q_{l_2} \qquad (z<0) \qquad (4)$$

The motion equqtion is expressee by,

$$m\ddot{y}+C_f\dot{y}+f_R(\mathrm{sgn}\,\dot{y})=AP_C \qquad (5)$$

Generally, $q_{1,2}$, are represented by;

$$q_1=2C_i(z)a(z)\sqrt{2(P_S-P_C)/\rho} \qquad (z\geqq 0)$$
$$q_2=2C_i(z)a(z)\sqrt{2(P_C-P_R)/\rho} \qquad (z<0) \qquad (6)$$
$$f_R(\dot{y})=+f_R\sim -f_R \qquad (\dot{y}=0)$$
$$=f_R\mathrm{sgn}(\dot{y}) \qquad (\dot{y}\neq 0)$$
$$C_i^2(z)=C_K|z|$$
$$a(z)=r^2(2|z|/r)^{3/2}/12$$

Assuming that $ql_{1,2}$ are the leakage flow rate and that both the flows are produced in the lap portion in the proximity of the port, as shows in Fig. 3 and can be regarded as the Quette's flow, q_l is given by the following equqtion ;

$$q_l=-Uhb/2+bh^3(P_1-P_2)/(12\mu l) \qquad (7)$$
$$(P_1\geqq P_2)$$

Here, the deviation δx, from the design value of x is obtained from the static input/output charcteristics (Fig. 7) when P_C is 0 and 100 %, and the equivalent clearence length l_1 at zero lapping is set more than 10th, while q_1 equivalently concentrates to the port

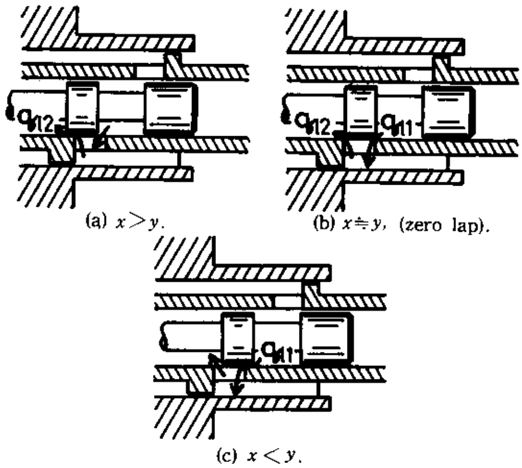


Fig. 3 Thought of leakage flow rate.

width in the circumferential direction to produce a laminar flow. Thus the equivalent clearence length $l=l_1+|z|_1$, when an arbitrary control force P_C is set, can be obtained from the leakage characteristics (Fig. 6).

$$|z|_1=|(1.56\times10^{-4})P_C-0.005| \quad (8)$$

Since the influence of friction in Eq. 5 is expected approximately 2 % and the first term in the left side of Eq. 7 several percent of the second term, both could be ignored.

3. Structure of The Prototype Pressure Control Valve and Experimental Setup

Fig. 4 shows the schematic diagram of the prototype pressure control valve set up at the author's laboratory, based on the aforementioned operating principle. To determine the performance limit of this valve; i) no initial load was applied to the spring to use it from its free length and the oil seal was deleted to cope with Coulomb's friction, ii) the spring was put the chamber (filled with hydraulic fluid) so that variable damping can be applied to the system without using any dashpot or other, and the number of the holes provided in the spring clamp was changed to attain the purpose. Also the spring clamp and spring receiver were subjected to baking finish Teflon coating so as to minimize friction.

Table 1 shows the principal design data on the prototype valve. Photo shows the overall view of the experimental setup used for measuring its frequency response, Fig.5 the measuring system, and Table 2 the experimental conditions.

4. Experimental Result and its Consideration

4.1 Experimental result

As for static characterintics, the leakage characteristics, $(x-q_L)$ (q_L is the total quantity of leakage flow rate), and input/output characteristics, $(x-P_C)$, were measured. Fig. 6 and Fig. 7 show examples.

Fig. 8 shows an oscillograph record of P_C for a sinusoidal input, $(x_s+D\sin2\pi ft)$. Incidentally, P_B is the pressure on the reverse side of the spring clamp measured for reference. As the waveform of P_C is

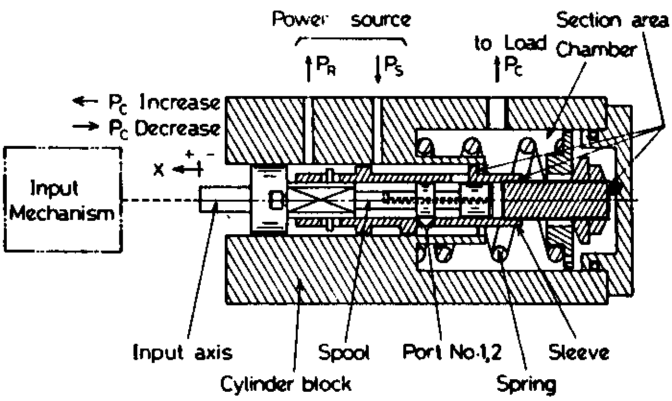


Fig. 4 Schematie diagram of the prototype pressure control valve.

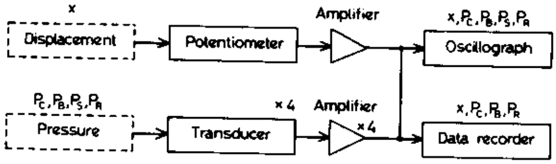


Fig. 5 Flow chart for experimentation.

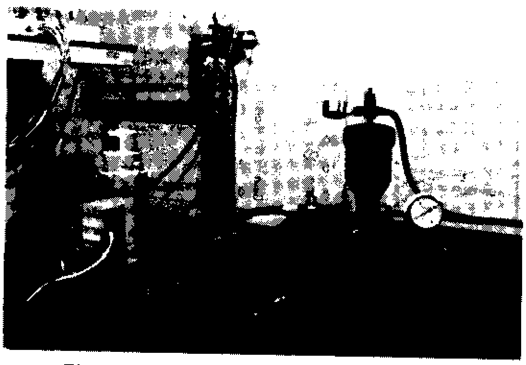


Photo 1 Overall view of the experimental setup, used to measure frequency response.

considerably disturbed by the influence of nonlinearity because of the featured of the port valve, the analog data was subjected to the data processing shown in Fig. 9, in obtaining the frequency response of the valve from this figures, and the digital output was further subjected to Fourier's expansion to obtain the fundamental wave, Fig. 10 shows an example. Regarding gain G, however, $f=0.5$Hz was used as a reference and the phase lag Θ was compared with x.

4.2 Consideration of experimental result

The result of the experiment conducted here on the static characterintics indicates that q_L is determined by (P_S-P_C), as shown in Fig. 6. It reveals that q_{Lmax} is approximately 14 cm³/s when P_S is 16 Mpa, which in considerably smaller than the relief flow rate required for stable operation of an ordinary pressure control valve on the market. Furthermore, q_L is not originally a relief flow rate but a leakage flow from the clearance in the sliding protion and others, witch is essentially unnecessary in the light of the operating principle. Fig. 7 also indicates, that; i) the input/output characteristics are good in both linearity and reproducibility, ii) the control range can be satisfied over the rated range from 10 % to 100 % despite the use of the spring from its free length, and iii) the resolution obtained from the followup error is held within ± 5 % as against an arbitrary set value. Fig. 1 showed a comparison for the concept of the characteristics of this new pressure control valve

Table 1 Main data of the pressure control valve.

Item			Data
Section area	(A)	〔cm²〕	1.54
Input (piston stroke)	(x)	〔mm〕	max 10.5
Pressure controlable region	(P_C)	〔MPa〕	0 to P_s
Hydraulic fluid			MIL H-5606B

Table 2 Experimental conditions.

Item			Condition
Static characteristics			
Supply pressure	(P_S)	〔MPa〕	3.5, 10.0, 16.0
Spring constant	(k)	〔k N/cm〕	0.5, 1.54, 2.50
Input displacement	(x)	〔mm〕	0 to 10
Dynamic characteristics :			
Supply pressure	(P_S)	〔MPa〕	3.5, 10.0, 16.0
Spring constant	(k)	〔kN/cm〕	0.5, 1.54, 2.50
Piston set point (neutral)	(X_s)	〔mm〕	5, (2)
Amplitude	(D)	〔mm〕	±1, ±2, ±4
Working frequency	(f)	〔Hz〕	0 to 30
Hydraulic fluid temperature (T_H)		〔℃〕	20 to 30

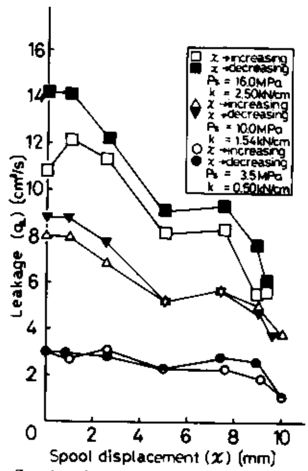


Fig. 6 Static characteristics ($x-q_L$ diagram)

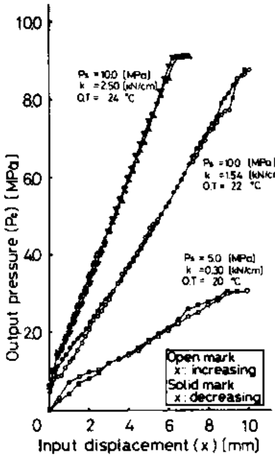


Fig. 7 Static characteristics ($x - P_C$ diagram.)

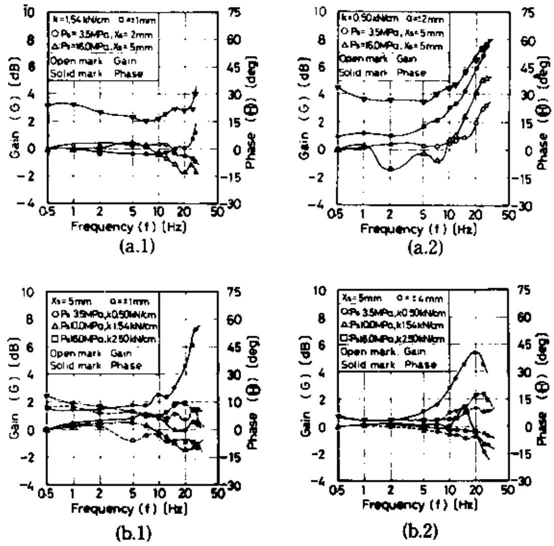


Fig. 10 Dynamic characteristics (frequency response).

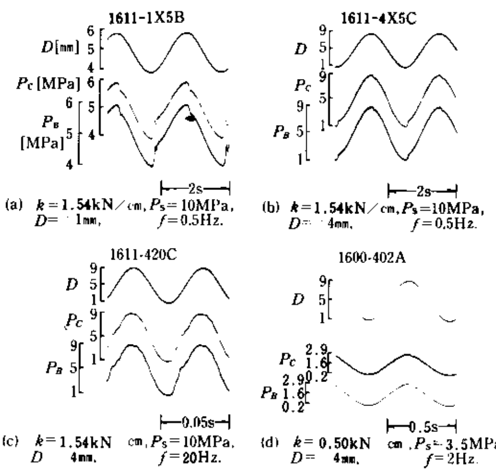


Fig. 8 Analog data.

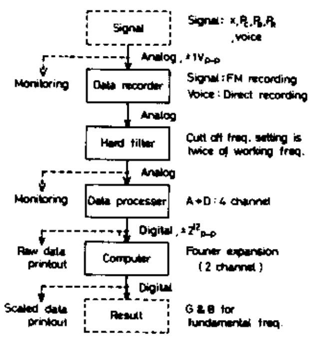


Fig. 9 Data processing system.

with those of the conventional pressure control valve.

It can be held that principal factors that limit these characteristics are leakage flow and Coulomb's friction, and appropriate measures and methods are to be taken in designing the valve for practical use of commercial product.

Regarding dynamic characteristics, Fig 10 indicates that the frequency response in held within -2 dB in gain decrease (ΔG) and within -5 deg in phase lag ($\Delta\Theta$), so that this new pressure control valve may have enough characteristics for practical use as a pressure control valve to be used in a force control system.

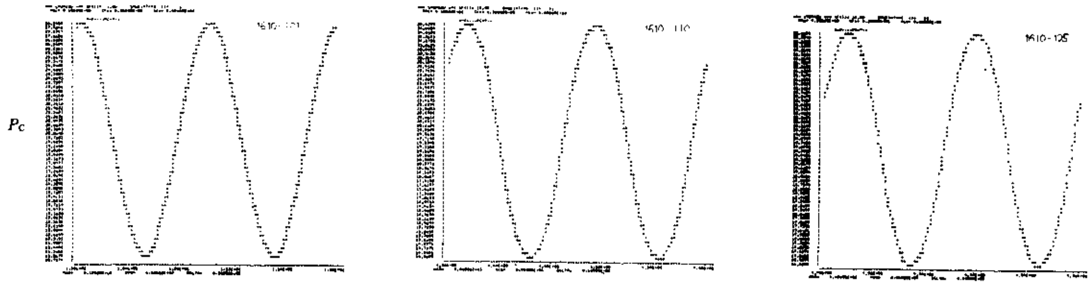
5. Result of Numerical Calculation and its Consideration

The numerical calculation/analysis of primitive equations derived in Section 2.2 was carried out to verify the evaluation of experimental result.

The calculation constants to be used in the numerical calculation were determined from the experimental results (Table 3).

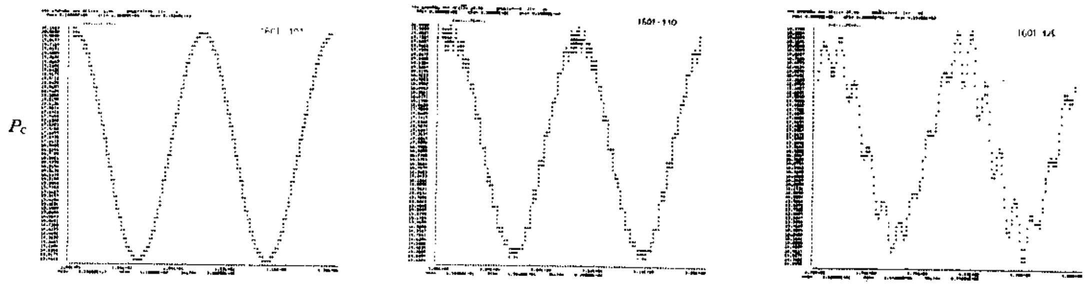
Fig. 11 and Fig. 12 show some examples of the results of calculation obtained by using f as a parameter.

The calculation results of the characteristics of this pressure control valve are in good agreement with the



(a) $f=1$Hz. (b) $f=10$Hz. (c) $f=25$Hz.

Fig. 11 Calculation of frequency response ; Case I K/P_s are comparatively large. $K=1.54$KN/cm, $P_s=3.5$Mpa.



(a) $f=1$Hz. (b) $f=10$Hz. (c) $f=25$Hz.

Fig. 12 Calculation of frequency response ; Case II K/P_s are comparatively small. $K=0.5$KN/cm, $P_s=10$Mpa.

Table 3 Calculation constants.

Item \ k (kN/cm)		0.50	1.54	2.50
V	(cm³)	52	133	129
K	(kN/cm³)	1.0×10^2	0.9×10^2	0.9×10^2
m	(kg)	0.248	0.283	0.301
P_S	(MPa)	3.5, 10.0, 16.0		
f_R	(N)	1.0		
C_f	(N/s)	0.3		
C_l		0.1		
ρ	(kg/cm³)	0.867×10^{-6}		
μ	(Pa·s)	$0.1\times0.867\times10^{-6}$	common in all	
υ	(cSt)	10		
l_1	(cm)	0.01		
b	(cm)	0.8		
h	(cm)	9×10^{-4}		
A	(cm²)	1.54		

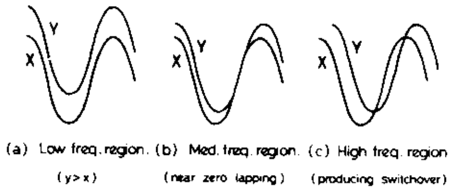
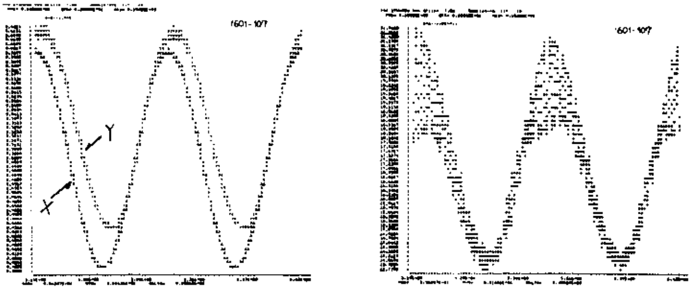


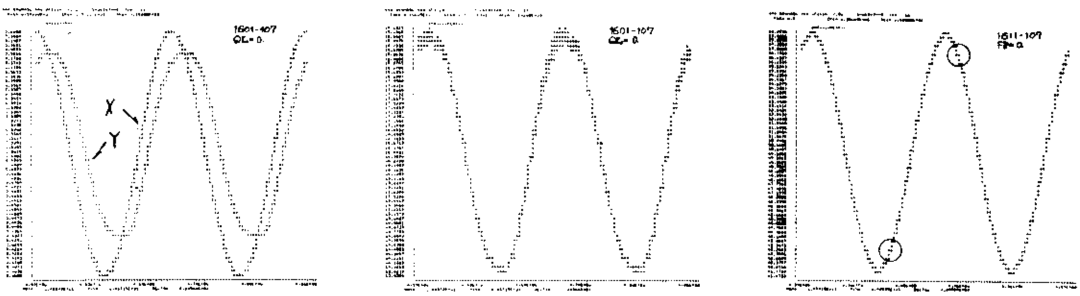
Fig. 13 Behavior of the spool and sleeve.

experimental results, i.e., from the figures we find out the operations sometimes being stable and sometimes high-frequency vibration taking place, depending on the conditions and frequencies (Fig. 11 and 12). Surmising that the principal cause is related to q_1, the behavior of the sleeve with respect to the spool, when there is a flow of q_1, is such that the relationship of $x\sim y$ is as shows in Fig. 13, when Ps is greater than $2P_C$ (and $P_R=0$) for the setted P_C or when k/P_S is small as in the conditions shown in Fig. 12, depending on the operating frequency f. Immediately before $(x-y)$ becomes zero in Fig. 13(b), that is, at near zero lapping and when the system operates at a flow rate of near q_1, high frequency vibration takes place as expected naturally ($f=7$ Hz in Fig. 14), making it necessary to apply appropriate damping. If f becomes more than this, the situation is different and the vibration becomes smaller as shown in Fig. 12. If it is assumed that q_1 equqals zero, the difference in evident as shown in Fig. 15. In this case, y does not cause vibration at a position higher than x. It can be seen that the presence of q_1 considerably affects the behavior of the spool and sleeve and eventually P_C.



(a) x to y relative position. (b) P_C

Fig. 14 Caluclation of frequency response : Case III Same conditions as for Fig.12, but x to y relative position is critical. $f=7$Hz.



(a) Relative position of x to y. (b) P_C, at $q_l=0$. (c) P_C, at $F_R=0$.

Fig. 15 Calculation of frequency response : Case IV Same conditions as for Fig.14, except $q_l=0$ or $F_R=0$.

6. Conclusion

This paper concerns the concept, as well as experimental and analytical results, of a prototype pressure control valve using a novel method for a control pressure feedback mechanism, to be used in a hydraulic pressure/force control system. The results revealed that :

1) This valve requires no null/drain leakage flow in operating principle, and the consumption of pressured hydraulic fluid is neglegible.
2) It has a wide control range and excellent input/output characteristics, hardly requiring any force for driving input systems.
3) It demonstrates a rapid response up to a sufficiently high frequency region for practical use.

Because of the outstanding features, this pressure control valve expected to find a wide range of application, for example, in braking devices for high speed trains which are limited in drive energy and require a wide pressure control range regardless of the displacement, driving servomechanism such as in aircrafts and rockets, and industry in general. The improvement of the machining accuracy and the extensive tests will be necessary before commercialization.

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