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STATIC AND DYNAMIC CHARACTERISTICS OF A CLASS OF PRESSURE COMPENSATED FLOW CONTROL VALVES

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ABSTRACT

This paper deals with the static and dynamic characteristics of a pressure compensated flow control valve. A nonlinear mathematical model of the valve is developed. The simulation of the valve is carried out using a digital computer on the basis of the developed mathematical model. The validity of the models is verified in the steady state by comparing simulation and experimental results. The dynamic characteristics of the valve are predicted theoretically using the simulation program. The influence of some constructional parameters on the valve static and dynamic characteristics is investigated and optimum values of these parameters are determined.

NOMENCLATURE

A_3	Area of the 1st damping orifice, m^2.
A_5	Area of the 2nd damping orifice, m^2
A_p	Spool area subjected to pressure, m^2.
A_t	Throttling area of the valve, m^2.
b	Compensating stage opening, m.
B	Bulk modulus, MPa.
C_c	Contraction coefficient.
C_d	Discharge coefficient.
d_1	1st damping orifice diameter, m.
d_2	2nd damping orifice diameter, m.
d_p	Spool diameter, m.
f	Coefficient of viscous friction, Ns/m.
k	Spring stiffness, N/m.
m	Spool and spring reduced mass, kg.
P_1	Input pressure, MPa.
P_2	Pressure in the 1st chamber, MPa.
P_3	Pressure in the 2nd chamber, MPa.

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P_5 Pressure in the 3rd chamber, MPa.
P_T Tank pressure, MPa.
Q_2 Flow rate through the upstream opening, m^3/s.
Q_3 Flow rate through the 1st damping orifice, m^3/s.
Q_5 Flow rate through the 2nd damping orifice, m^3/s.
Q_6 Flow rate through the main valve restriction, m^3/s.
Q_7 Outlet flow rate, m^3/s.
R_1 Resistance element representing the 1st damping orifice.
R_2 Resistance element representing the 2nd damping orifice.
V_1 Volume of valve central chamber, m^3
V_2 Volume of the spool upper chamber, m^3.
V_3 Volume of the spool lower chamber, m^3.
x Spool displacement, m.
x_0 Initial spring compression, m.
ρ Fluid density, kg/m^3.
ϑ Jet angle, rad.

INTRODUCTION

The hydraulic power systems are widely applied for the control and power transmission in the different engineering fields. The control of the transmitted power and, consequently, the function of the controlled system is insured by the different hydraulic control valves. Among these valves one can distinguish the flow control valves, controlling the flow rates and the speeds of the moving organs. The investigation of the performance of these valves and their effect on the system behavior has been the subject of several publications. Takenaka and Urata [1] reported the results of a study of the dynamic characteristics of a pressure compensated flow control valve. Their study is based upon the development of a linearized model of the valve. They reached concrete conclusions concerning the effect of pressure compensator and the valve transient behavior. In his survey of the studies of dynamic characteristics of the hydraulic control valves in Japan ,Takinaka [2] pointed out the importance of the investigation of the valve stability and speed of response. Kassem and Rabie [3] deduced a nonlinear mathematical model of a flow control valve with series pressure compensator. The model is based upon the development of an augmented bond graph of the valve. The valve dynamic behavior is investigated theoretically on the basis of this model. The study pointed out the important effect of the damping orifices on the valve transient response.

Herein, the performance of a two way series pressure compensated flow control valve with pressure compensator prior to the throttling section is investigated. The valve is assumed to be arranged in a meter out mode of speed control. A nonlinear mathematical model of the valve is deduced. Alternatively, a block bond graph is developed. The simulation is realized on a digital computer using TUTSIM program. The static characteristics of the valve are evaluated experimentally and theoretically while the valve transient response is investigated theoretically. The effect of some of the valve constructional and operational parameters on the response is investigated.

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DESCRIPTION AND MODELLING OF THE FLOW CONTROL VALVE.

The studied pressure compensated flow control valve is drawn schematically in Fig. 1. The valve consists of a pressure compensating stage prior to the throttle element. The pressure difference across the sharp edged throttling element acts on the spool of the pressure compensating stage against the action of a built in spring. The upper chamber of the spool is connected to that above the annular part of the spool by means of radial and axial holes. These holes, being of relatively large diameter, are found to be of negligible hydraulic resistance. The spool moves in the direction to compensate the effect of the pressure variation across the valve.

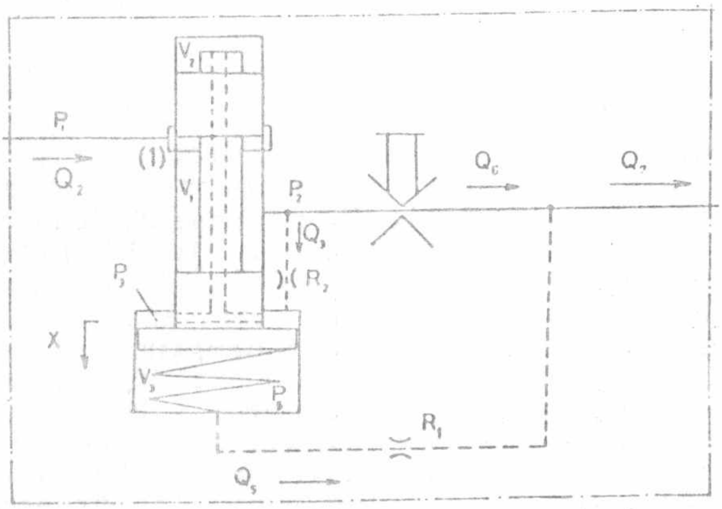

Fig. 1. Scheme of the series pressure compensated flow control valve

MATHEMATICAL MODEL

The studied flow control valve can be described mathematically by the following equations, considering the following assumptions:

1. The internal leakage through the spool radial clearance is negligible.
2. The valve is installed in a meter-out arrangement and its exit pressure is atmospheric.
3. The spool and main valve restrictions are of sharp edges.

$$Q_2 = C_d \pi d_p (b-x) \sqrt{\frac{2}{\rho}(P_1-P_2)} \tag{1}$$

$$Q_6 = C_d A_t \sqrt{\frac{2}{\rho}(P_2)} \tag{2}$$

$$Q_3 = C_{d3} A_3 \sqrt{\frac{2}{\rho}(P_2-P_3)} \tag{3}$$

$$Q_5 = C_{d5} A_5 \sqrt{\frac{2}{\rho}(P_5)} \tag{4}$$

$$Q_7 = Q_5 + Q_6 \tag{5}$$

$$Q_2 = Q_3 + Q_6 + \frac{V_1}{B}\frac{dP_2}{dt} \tag{6}$$

$$Q_3 = A_p \frac{dx}{dt} + \frac{V_2 + A_p x}{B}\frac{dP_3}{dt} \tag{7}$$

$$Q_5 = A_p \frac{dx}{dt} - \frac{V_3 - A_p x}{B}\frac{dP_5}{dt} \tag{8}$$

$$A_p(P_3 - P_5) - m\frac{d^2x}{dt^2} - f\frac{dx}{dt} - k(x + x_o) + \frac{\rho Q_2^2 \cos(\theta)}{C_d \pi d_p (b-x)} = 0 \tag{9}$$

The term ($A_p x$) is found to be neglegible compared with V_2 and V_3, in equations 7 and 8; therefore it is neglected.

BOND GRAPH

A bond graph model of the valve is developed, fig.2. The input pressure P_1 and exit pressure P_T are imposed by the sources SE3 and SE31. The two port resistor R_4 represents the effect of the spool valve restriction. The effect of fluid compressibility in the valve chambers of volumes V_1, V_2 and V_3 is taken into consideration by the capacitors C_9, C_{28} and C_{12} respectively. The energy dissipating effect of the damping orifices R_1 and R_2 and that of the main restriction is introduced by the resistances R_{30}, R_{11} and R_{10} respectively. The inertia element I_{19}, the resistance R_{22} and the capacitance C_{23} represent the effect of the inertia of the moving parts, the viscous friction resisting the spool motion and the spring stiffness respectively.

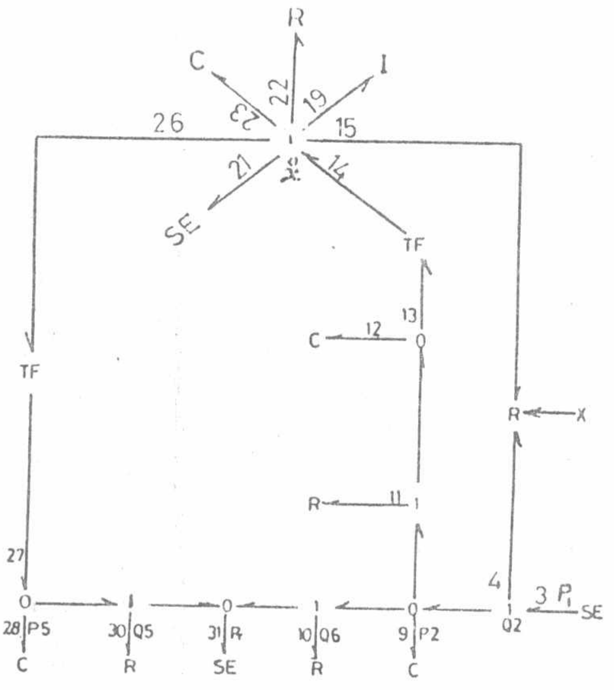

Fig.1. Bond graph.

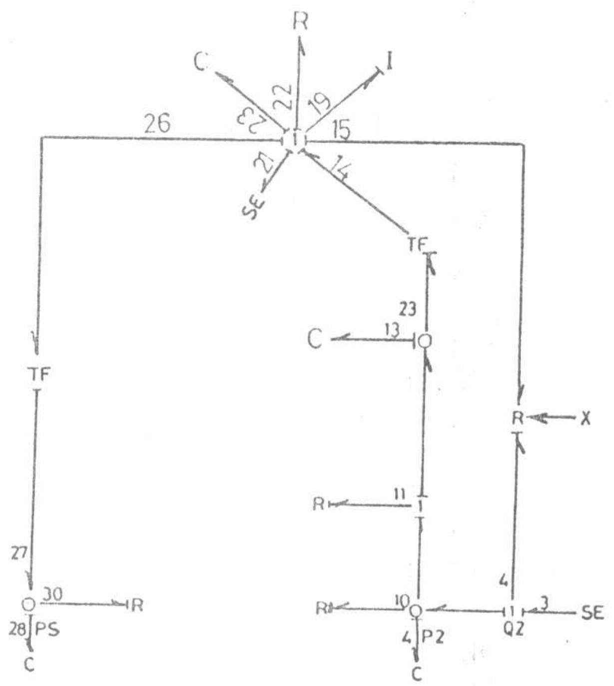

Fig.3. Augmented bond graph.

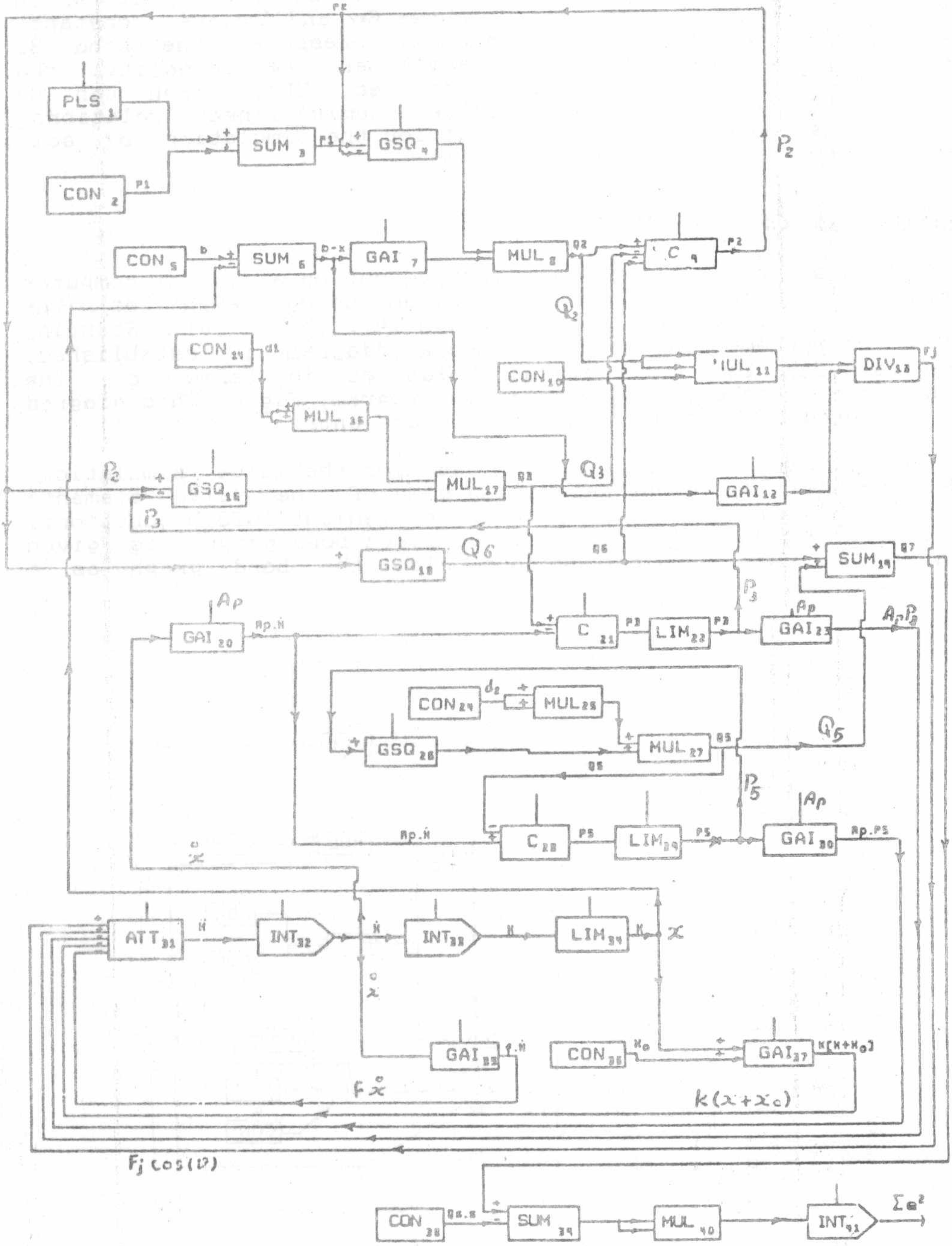

Fig. 4. TUTSIM block diagram of the valve

In this model, the variation of the volume of the spool chambers due to the spool displacement is found to be negligible which allows to assume that the capacitances C_{12} and C_{28} are constant. Considering the assumption of zero exit pressure, the bond 31 can be eliminated. Then by the assignment of causality, the augmented bond graph of fig.3 is obtained. This graph can be used for the simulation of the valve assuming linear relations. It is accepted as a structure input for the simulation by some digital computer simulation programs.

SIMULATION OF THE VALVE

The simulation of the valve is carried out on a digital computer using the TUTSIM simulation program using each of the mathematical model and the bond graph [4], [5] & [6]. Starting with the mathematical model, a block diagram is established, representing the mathematical relations in terms of the functional blocks of the simulation program, Fig.4. This diagram is accepted by the TUTSIM as a structure entry.

Alternatively, the bond graph is used for the valve simulation. The nonlinear relations describing some of the valve elements are introduced in the bond graph by convenient TUTSIM functional blocks. The resulting graph, called block bond graph, is given in Fig. 5, [7].The TUTSIM accepts also the bond graph as a structure entry.

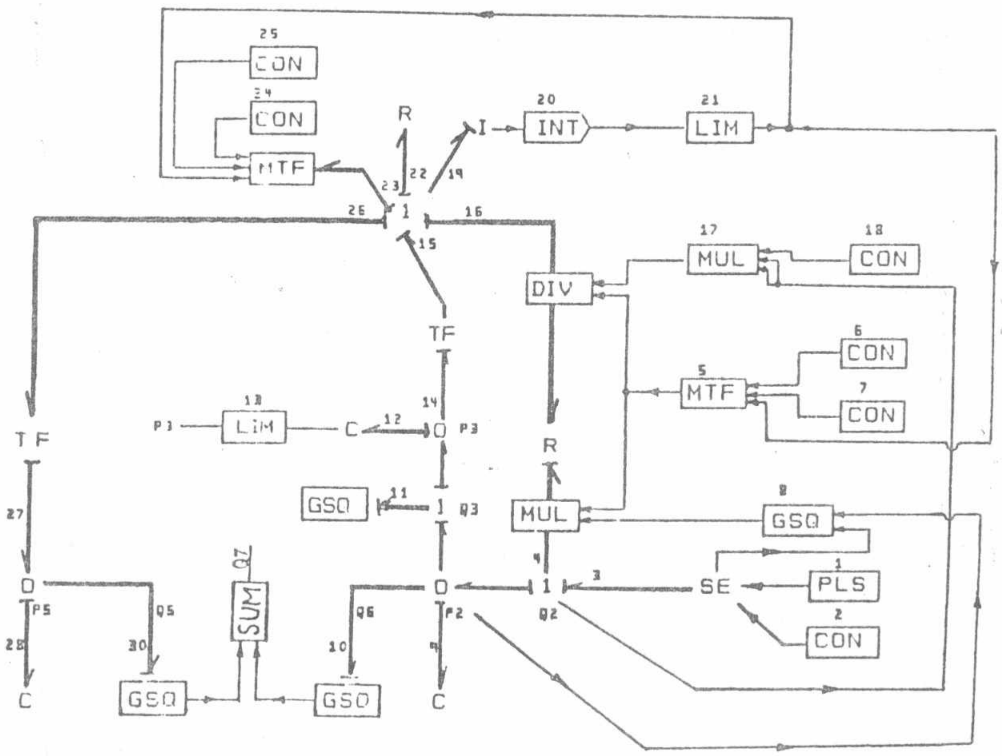

Fig. 5. Block bond graph of the flow control valve

This study is applied to a series pressure compensated flow control valve type 2FRM size 10 produced by Rexroth, Germany. The parameters of this valve were found out from the manufacturers data sheets and by direct measurements on the valve elements. The valve has the following parameters.

$d_p = 12$ mm	$b = 5.5$ mm	$A_p = 706.5$ mm^2
$k = 20$ kN/m	$x_o = 10$ mm	$A_t = 0$ to 20 mm^2

Two different simulation programs of the valve were developed on the basis of the mathematical model and the block bond graph. These two simulation programs gave identical results.

VALVE STATIC CHARACTERISTICS

The valve behavior in the steady state have been evaluated by finding out the relation between the valve flow rate and the input pressure. Theoretically, this relation is calculated by the simulation program for different values of valve restriction area A_t. The same dependence is evaluated experimentally. Fig.6 carries the theoretical and experimental results. The figure shows remarkable agreement between the experimental and simulation results. It is obvious that there is a minimum value of the input pressure, in order to attain a nearly constant flow. This value is found to be about 0.5 MPa.

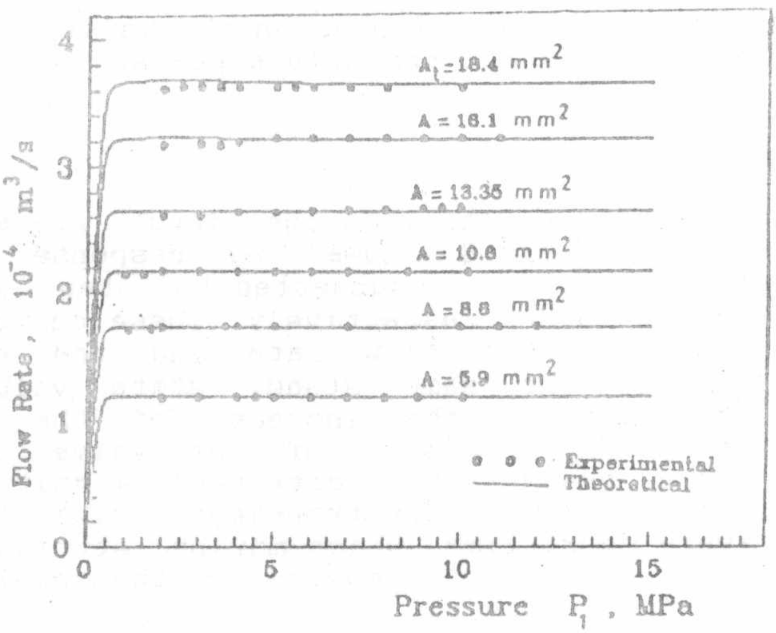

Fig. 6. Steady state relation between the valve flow and inlet pressure, for different throttling areas.

The effect of the spring stiffness on the valve static characteristics is estimated by calculating the relation between the valve flow and input pressure for different values of the spring stiffness. The calculation results are given in figure 7. The figure gives a nondimensional plot of the flow rate versus the input pressure. The reference flow Q_r is the flow rate corresponding to a chosen throttle area, A_t, of 18.4 mm^2, an input pressure of 5 MPa and a spring stiffness of 20 kN/m. The figure shows that the minimum pressure required to reach the constant flow rate increases with the spring stiffness.

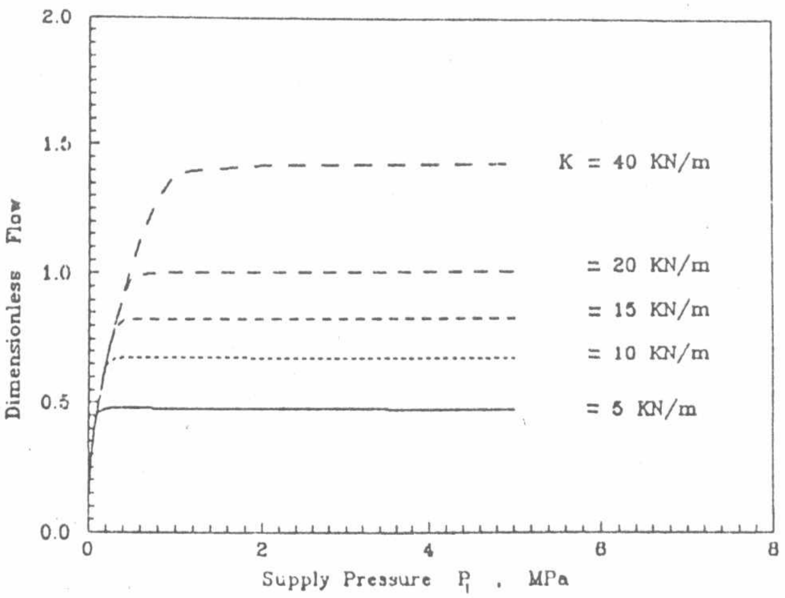

Fig. 7. Effect of the spring stifness on the valve static characteristics, for constant throttle area; A_t= 18.4 mm^2

VALVE DYNAMIC CHARACTERISTICS

The valve response to step variation in the input pressure is investigated theoretically. Fig.8 shows the response of the valve when the input pressure P_i is subjected to step increase from 5 MPa to 6, 7.5 and 10 MPa respectively. These curves show considerable overshoots in the flow rate and the response decrease rapidly to almost the same steady state value. The maximum overshoot increases with the increase of the applied step pressure variation. The response of the valve to step reduction of the input pressure of different magnitudes is calculated and plotted in Fig. 9. The transient period is seen to exhibit a relatively short time, which may be attributed to to the very small spool displacement covered by the spool in the transient period as shown in Fig. 10.

The valve includes two damping orifices. These orifices are to be sized to give the best transient behavior, according to a selected criterion.

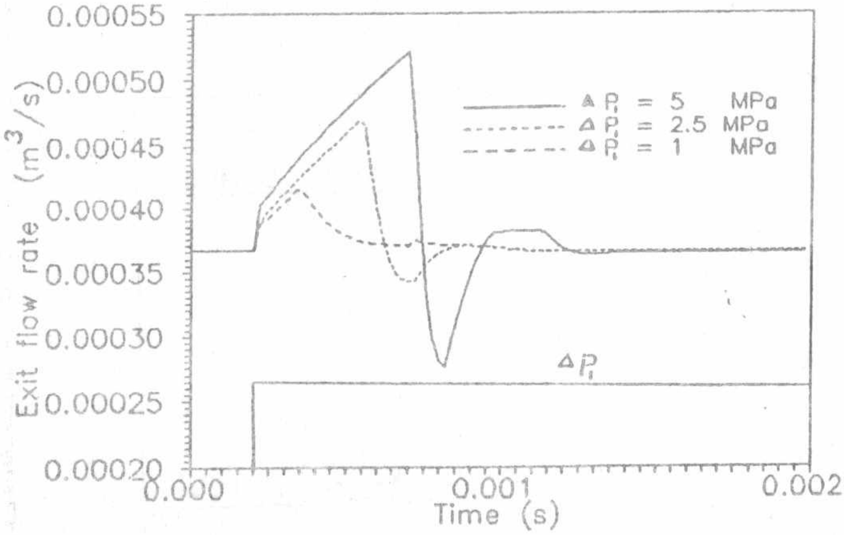

Fig. 8. Transient response of the valve exit flow rate to step increase of input pressure of different magnitudes.

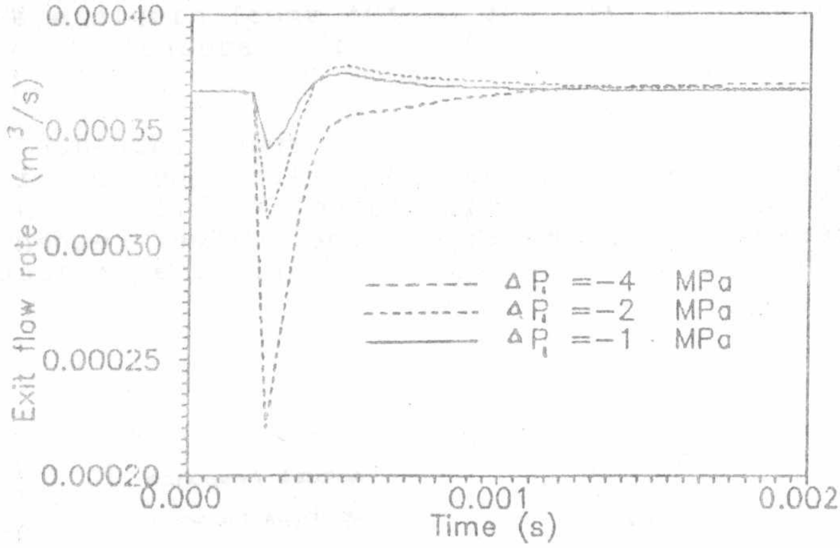

Fig. 9. Transient response of the valve exit flow rate to step decrease of input pressure of different magnitudes.

The used simulation program gives the possibility of parameter estimation using the simplex method. Up to eight parameters can be estimated to give the minimum value of the selected error function. Using the parameter estimation command, PE, the optimum diameters of the damping orifices could be predicted, minimizing the integral square error IES of the response, where:

$$IES = \int (Q_7 - Q_{ss})^2 \, dt \qquad (10)$$

The optimum values of these diameters are found to be:

d_1 = 5.9 mm and d_2 = 1.2 mm

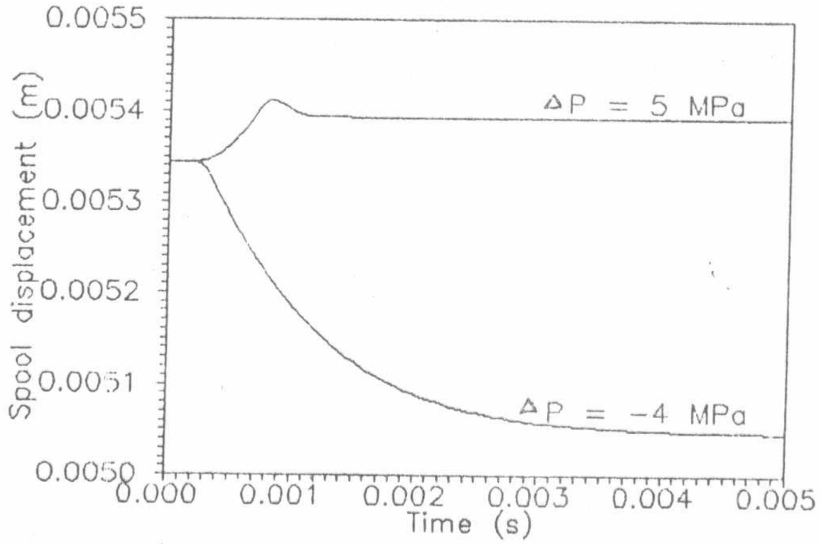

Fig.10. Transient response of the spool displacement to step variation of input pressure.

The transient response of the valve to step increase of the inlet pressure by 10 MPa, when equipped with the original and with the proposed orifices, is calculated, fig 11. This figure shows a considerable improvement of the valve response when equipped with the optimum orifices, specially the reduction of the maximum over shoot.

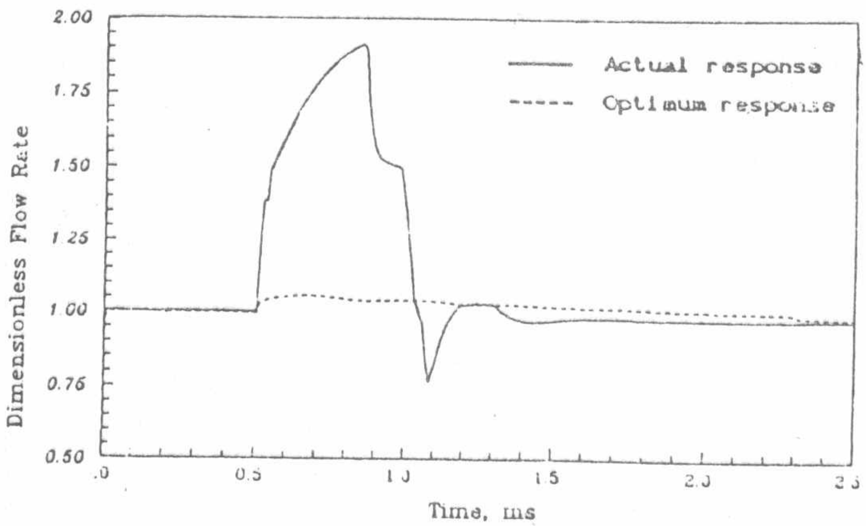

Fig.11 . Effect of the optimum damping orifices on the valve transient response, keeping A_t = 18.4 mm^2 and applying a step increase in the inlet pressure of 10 MPa.

CONCLUSION

The performance of a series pressure compensated flow control valve is investigated in this work. A nonlinear mathematical model of the valve is deduced. Alternatively a block bond graph is developed. The computer simulation of the valve is carried out using the TUTSIM simulation program. The simulation results gave good agreement with the experimental results on the level of the static characteristics. The valve dynamic behavior is studied theoretically. The valve presents very short time of response associated by considerable flow over shoots. The transient response can be improved, on the basis of simulation results, by changing the damping orifices` diameters. The optimum dimensions of these orifices are determined using the parameter estimation facility offered by the simulation program.

REFERENCES

[1] Takenaka T. and Urata E., "Static and Dynamic Characteristics of Oil Hydraulic Control Valve", Fluid Power International Conference, London sept., 1969.

[2] Takenaka T., "A Survey of the Studies of Dynamic Characteristics of Oil Hydraulic Control Values in Japan", Flow Measurement and Control, Vol.1, pp. 1235- 1242, Instrument Soc. America, Pittsburgh, Pa, 1974.

[3] Kassem, S.A., and Rabie, G., "Bond Graph Model and Investigation of Dynamics of A Class of Hydraulic Flow Control Valves", First A.M.E Conf., 19-31, May, Cairo, 1984.

[4] Meerman et al, "TUTSIM Interactive Simulation on 8 Bit Personal Computer", 10^{th} Imacs Con. Montreal.

[5] Meerman, J.W, "TUTSIM User Manual", Meerman Automation, the Netherlands, 1986.

[6] William H.Murray "Mathematical Modeling with TUTSIM", PC Tech. Journal, July, 1984.

[7] M.G. Rabie, SA. Kassem, S.A. Elsayed, M.A. Aziz and O.G. Elsayed, " Block Bond Graph and TUTSIM, A Powerful Tool for Nonlinear Dynamic Systems Modeling and Simulation" Alex. Eng. J., Alex. Univ., Volume 28 No.3, pp. 519-537, 1989.