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Investigation of a nonlinear dynamic hydraulic system model through the energy analysis approach†

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Abstract

The dynamics of a pressure regulator valve have been studied using the through Bondgraph simulation technique. This valve consists of several elements that can transmit, transform, store, and consume hydraulic energy. The governing equations of the system have been derived from the dynamic model. In solving system equations numerically, various pressure-flow characteristics across the regulator ports and orifices have been taken into consideration. This simulation study identifies some critical parameters that have significant effects on the transient response of the system. The results have been obtained using the MATLAB-SIMULINK environment. The main advantage of the proposed methodology is its ability to model the nonlinear behavior of the hydraulic resistance of system elements as a function of the fluid flow rate.

Keywords: Bondgraph technique; Pressure regulator valve; Hydraulic system

1. Introduction

Almost every hydraulic system is equipped with a regulator valve system to maintain the working pressure of the system at a pre-determined level. Under ideal conditions, the pressure regulator valve should provide an alternative flow path to the tank for system fluids while keeping the system pressure constant. The Bondgraph simulation technique and its applications in modeling the dynamic system are discussed extensively by Karnopp and Rosenburg [1]. Thoma also presented a detailed approach for modeling the system performed through the Bondgraph simulation technique [2]. The dynamics of a pilot-operated pressure relief valve have also been studied using the Bondgraph simulation technique by Dasgupta and Karmakar [3]. By solving the system equations numerically, the various pressure-flow characteristics of the valve ports have been taken into consideration. They also investigated the dynamics of a direct-operated relief valve with directional damping using the same technique. Consequently, significant parameters of the valve response have been identified, some of which can be modified by improving the dynamic characteristics of the regulator valve [4]. Dasgupta and Watton further studied the dynamics of a proportional-controlled piloted relief valve using the same simulation technique [5]. In this study, the simulation results can help identify some critical parameters that may have significant effects on the transient response of a hydraulic system. Ray [6] and Watton [7] have discussed several simplifications related to the use of the transfer function formulation technique for analyzing a single-stage pressure regulator valve. Likewise, Borutzky analyzed the dynamics of the spool valve controlling the orifices using the Bondgraph simulation technique [8].

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The current work concentrates on identifying the nonlinear dynamic model of a pressure regulator using the Bondgraph simulation technique. An accurate non-linear model of the system which explains the physical characteristics of each element of the valve assembly based on the transmission, transformation, and consumption of hydraulic energy is hereby presented. Finally, the effects of the various significant design parameters of the valve that influence its dynamic response are also investigated.

2. The physical model of the system

The physical characteristics of a pressure regulator are presented in this section. The main application for regulator valves is in the automobile LPG system. A simplified representation of a pressure regulator valve is depicted in Fig. 1. This regulator is basically composed of two dynamical subsystems, namely, the mass-spring and hydraulic subsystems. The mass-spring subsystem consists of two governors denoted by m_1 and m_2. As depicted in Fig. 1, these governors are connected to each other using the two lever arms presented by a and b. The second governor is limited by a diaphragm on its upside. The diaphragm, which does not possess any elastic property, is connected to the governor via a powerful spring that is manually depressed by a preload. Normally, it is only used to divide the chamber of the valve into two separate parts. The outer part is affected by the atmospheric pressure P_{atm}, and the inner part is influenced by hydraulic fluid flows in the chamber of the regulator valve. The other side of the spring is affected by an adjusting screw which is manually regulated. By twisting the screw, the initial depression of the spring, specified as l_0, can control further adjustments to the regulator valve.

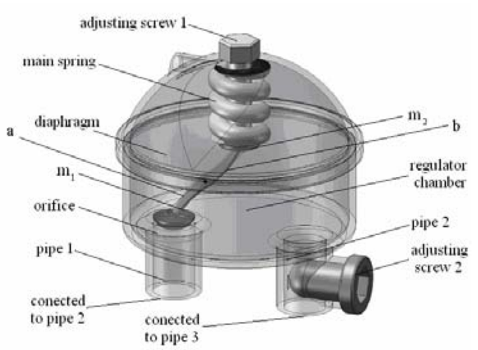

Fig. 1. A simplified model of a pressure regulator valve.

The second subsystem of the valve is the hydraulic subsystem; it is composed of three pipes, two reversible elbows, two orifices, and two adjusting screws. The two screws are twisted manually to regulate the flow rate of the hydraulic fluid. At the end of the first pipe, there is an orifice which has two types of hydraulic resistance. One type is a constant resistance related to the ratio of the orifice diameter to the pipe diameter. The other resistance is related to the variable distance between the orifice and the first governor, specified as x. Meanwhile, there is an adjusting screw at the end of the second pipe, and the distance between the adjusting screw and the inner surface of the pipe, y, is the second parameter which controls the flow rate. The hydraulic resistance presented by these two adjusting valves refers to the nonlinear function of the flow rate discussed by Karnopp [1]. When the inlet pressure of the valve is changed, the first governor moves up and down as result of the difference produced between the atmospheric and motivated fluid pressures. Considering the fact that the connected arm is controlled by the main spring oscillations, moving the diaphragm causes a hydraulic balance between the inner pressure and the atmospheric pressure. The spring preload further affects the oscillation amplitude and other properties of the dynamic behaviors of the system. It has been demonstrated that the up-and-down motions of the diaphragm can control the flow rate of a hydraulic fluid. As long as the system pressure does not exceed the setting pressure of the valve, the spring motion stops, after which the dynamic system parts achieve hydraulic balance.

3. The Bondgraph model of the pressure regulator valve

To determine the dynamic model of the system, the following considerations are included:

- The resistive and capacitive effects are consolidated wherever appropriate.
- The capacitive characteristics of the diaphragm are not considered.
- The outlet pressure is assumed to be equal to the atmospheric pressure.
- A stable supply to the regulator valve inlet port is ensured.
- The masses of the spring and diaphragm are neglected, but the masses of the two arms are considered.
- The fluid considered for the analysis has the

characteristics of a Newtonian fluid and is an incompressible fluid with a computable inertia.

- The spring depression module is constant.
- All pipes are long, thin tubes wherein the laminar flow of an incompressible fluid develops.

The Bondgraph model of the system is shown in Fig. 2, wherein the geometry parameters of the pressure regulator valve are separately marked for clarity. In this model, the element SE_1 represents the inlet pressure, and *P(t)* is supplied to the valve from a stable source. In addition, SE_2 and SE_3 represent two constant and stable sources of pressure. The atmospheric pressure and the desirable outlet pressure are denoted by P_{atm} and P_0, respectively. The parameters I_{h1}, I_{h2}, and I_{h3} indicate the respective fluid inertia in the associated pipes. In this way, a linear inertia coefficient of the *I*-element is formulated using the Newton principle in the following form:

$$I_h = \frac{\rho l}{A}. \tag{1}$$

The parameters, R_{h1}, R_{h2}, and R_{h3} indicate the hydraulic resistance of fluid in each pipe of the valve. For a long, thin tube with laminar flow of an incompressible fluid, the hydraulic resistance is formulated as [1]:

$$R_h = 128\frac{\mu l}{\pi d^4}. \tag{2}$$

The R_1 element connected to the 0-Junction indicates the hydraulic resistance in Orifice 1, which is composed of two parameters. The first parameter is a constant hydraulic resistance related to the differences in the pipe cross section, A_1, which is relative to the orifice area, A. The other parameter affecting the orifice resistance is a time-variant motion of the first governor with respect to the pipe orifice. This resistance is a nonlinear time-variant function of the distance, *x*, and can be written as:

$$R_1 = R(x) + R_{Orifice} \tag{3}$$

where the parameters *R(x)* and $R_{Orifice}$ are in the following form [1]:

$$R(x) = \frac{\rho |Q|}{2{C_d}^2(x)A^2(x)}, \text{ and} \tag{4}$$

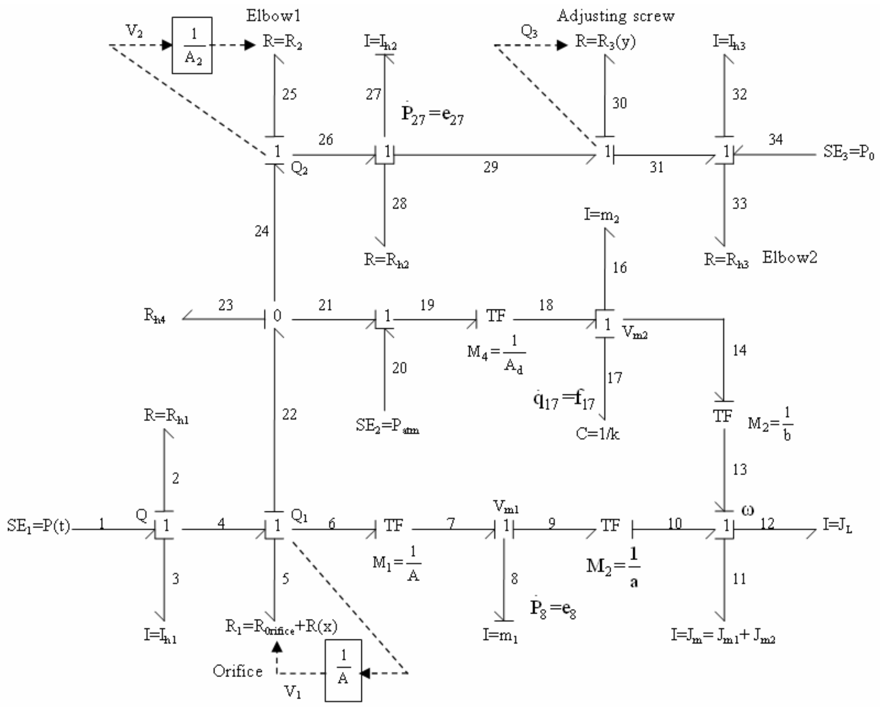

Fig. 2. The Bondgraph model of a pressure regulator valve.

$$R_{Orifice} = k_1 v_1^2 \,. \tag{5}$$

The constant coefficient, k_1, is determined from hydraulic handbooks. Moreover, $A(x)$ is the variable area of fluid passage, C_d is a constant coefficient related to the geometric properties of the orifice, and Q denotes the fluid flow rate. By applying these considerations and substituting the numeric values, Eq. (3) is simplified in the following nonlinear relation:

$$R_1 = 112 P_8^2 + 131.49 \frac{P_8}{x^2} \tag{6}$$

After this orifice, the fluid hydraulic energy is then divided into two sections. The first section involves the mechanical energy consumed by the mass-spring subsystem. The associated consumption bond is connected to a 1-Junction and is depicted by Bond 6. The second section is the hydraulic subsystem which starts with Bond 22.

In the mechanical subsystem, after Bond 6, a transfer function is situated to demonstrate the first governor velocity. The inertia characteristics of the governor and the two connected arms in the up-and-down and rotational motions are then simulated. In the hydraulic subsystem, a 0-Junction is employed after Bond 22 as result of changing the flow rate with the diaphragm oscillations. Meanwhile, the hydraulic resistance of this chamber is indicated by R_{h4}. The rest of the energy is consumed and transmitted by this subsystem because of the existing connection between the diaphragm and the mechanical subsystem. Finally, using a transformer and Bond 13, the transmitted kinetic energy can be added to the rotational elements. Using Bond 24, the simulation process of the hydraulic subsystems is thereby continued. At first, the resistance of Elbow 1 is computed using a similar formulation presented in Eq. (4). The corresponding constant coefficient can be selected from hydraulic handbooks. The required magnitude of velocity is determined using an information bond drown from 1-Junction, which also provides information about the fluid flow rate in the pipe. After Bond 29, the resistance R_3 appears and yields to the resistance of an adjusting screw formulated as a nonlinear function of parameter y. It is possible to control the fluid flow rate when this screw is twisted manually. To find the nonlinear resistance of this adjusting screw, Eq. (5) is employed:

$$R(y) = \frac{\rho\,|Q|}{2 C_d^2(y) A^2(y)} \tag{7}$$

Finally, by modeling the inertia and resistance of the fluid, and applying the constant and stable source of effort specified as P_0, the Bondgraph simulation is completed. It will be possible to derive the state space equations of the system afterwards by adding the causality significant to the Bondgraph simulation. As shown in Fig. 2, there are three integrated elements in this Bondgraph simulation and are determined by Bonds 8, 17, and 27. Having been introduced to the causality sign in deriving the state equations, we can conclude that there are three nonlinear state space equations with state variables defined as:

$$x = [p_8 \quad p_{27} \quad q_{17}]^T \tag{8}$$

In the equation above, p_8 is the momentum of the first governor, p_{27} is the fluid momentum in the Pipe 3, and q_{17} is the displacement of the spring located in the second governor. The first step to deriving the state equations can be accomplished by defining the basic equation of each integrating element determined by an integrating causality sign as:

$$\begin{cases} \dot{p}_8 = e_8, \\ \dot{p}_{27} = e_{27}, \text{ and} \\ \dot{q}_{17} = f_{17}. \end{cases} \tag{9}$$

Applying the Bondgraph rules for both 0-Junction and 1-Junction, and considering the numeric values of the geometrical parameters tabulated in Table 1, the nonlinear state space equations can thus be derived in the following form:

$$\dot{p}_{27} = 1.559 p_{27} - 35043.775 p_8 - 1.074 \times 10^{-9} p_{27}^3, \tag{10}$$

$$\dot{q}_{17} = 20 p_8, \text{ and} \tag{11}$$

$$\begin{aligned} \dot{p}_8 = {} & 1.441 \times 10^{-4} p_{27} + \frac{3.089 \times 10^{-10} p_{27}^2}{y^2} \\ & - 1.306 \times 10^{-7} [112 p_8^3 + \frac{131.491 p_8^2}{x^2}] - 176.362 q_{17} \\ & + 1.385 \times 10^{-5} P_{atm} + 1.697 \times 10^{-6} P(t) - 28.512 p_8 \\ & - 6.531 \times 10^{-14} p_{27}^3 - 1.215 \times 10^{-5} P_0. \end{aligned} \tag{12}$$

Table 1. Numeric values of geometrical parameters.

Parameter	Value	Parameter	Value	Parameter	Value
a, b, D	0.10 m	k_1, k_2	0.28	P_0	1.813 Pa
A	$\pi/4\times0.07^2$ m^2	K_s	400000 Kg/m	P_{atm}	1.013×10^5 Pa
A_1, A_d	$\pi/4\times0.02^2$ m^2	L	0.6 m	R_{h1}	1.987×10^8
A_2, A_3	$\pi/4\times0.05^2$ m^2	L_0, L_2	0.01 m	R_{h2}	1.696×10^4
$C_d(x)$, $C_d(y)$	0.62	L_1	3.0 m	R_{h3}	8.480×10^5
I_{h1}	7.640×10^6	K_s	400000 Kg/m	R_{h4}	6.356×10^4
I_{h2}	4.076×10^4	L_3	0.50 m	x	0.03 m
I_{h3}	2.040×10^5	M_1	0.050 Kg	y	0.02 m
J_L	1.0×10^{-4} Kgm^2	M_2	0.070 Kg	μ	0.26 Kg/m.sec
J_M	1.2×10^{-3} Kgm^2	$P(t)_{max}$	1.0×10^6 Pa	ρ	800 Kg/m^3

Substituting the following unknown parameters in the state Eqs. (10)-(12) yields:

$$P_{atm}=1.013\times10^5\,pa,\quad P_0=1.813\times10^5\,pa,\quad P(t)=20\times10^5\,pa,\quad x=0.03\,m,\quad y=0.02\,m. \tag{13}$$

Assuming that the initial conditions are:

$$p_8(t=0)=0,\ p_{27}(t=0)=0,\ q_{17}(t=0)=0.01 \tag{14}$$

then the state equations can be determined as:

$$\begin{aligned}\dot p_8 &= -28.512p_8+1.441\times10^{-4}p_{27}-176.362q_{17}\\ &\quad -1.463\times10^{-5}p_8^{\ 3}+0.897+1.0908\times10^{-2}p_8^{\ 2}\\ &\quad +7.723\times10^{-5}p_{27}^{\ 2}-6.531\times10^{-14}p_{27}^{\ 3},\end{aligned} \tag{15}$$

$$\dot p_{27}=1.559p_{27}-35043.775p_8-1.074\times10^{-9}p_{27}^{\ 3}, \tag{16}$$

and

$$\dot q_{17}=20p_8. \tag{17}$$

To specify the behavior of the regulator, it is necessary to numerically solve the determined differential equations with their initial conditions. To achieve this purpose, and to determine the transient response of the regulator dynamic system, the MATLAB-SIMULINK environment is employed. The profiles of the first governor velocity, main spring displacement, flow rate in Pipe 2, outlet pressure of the regulator valve, and flow rate in the regulator chamber are depicted in Figs. 3-7, respectively. Meanwhile, Figs. 8, 9, and 10 show that an increase in the screw adjustment (y) yields a corresponding increase in the Pipe 2 flow rate and regulator chamber, as well as in the system outlet pressure. By comparing these behaviors with the expected results, the accuracy of the proposed Bondgraph model could thus be concluded.

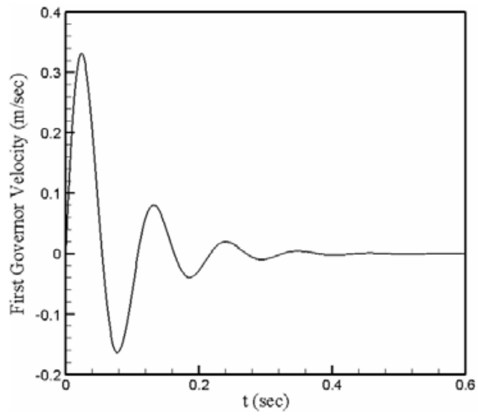

Fig. 3. Profile of the first governor velocity.

Considering the resulting figures, it can be concluded that the pressure regulator has an excellent ability to reduce the inlet pressure to a desirable one. The settling time is defined as the time the system takes to reach a steady state condition, while the peak time refers to the time the system takes to achieve its peak condition after the first overshoot with the same input. The transient response indicates a condition where both the settling and peak times are in the appropriate range. According to Fig. 3, the governor motion stopped after the 0.4-second mark. This result could be attributed to good regulator performance. Observing Fig. 6, it can be concluded that the outlet pressure has been regulated appropriately. The spring oscillations also stopped after 0.6 second and thus, the first overshoot amplitude did not exceed 2.2 cm. This kind of main spring performance is very desirable in industrial applications.

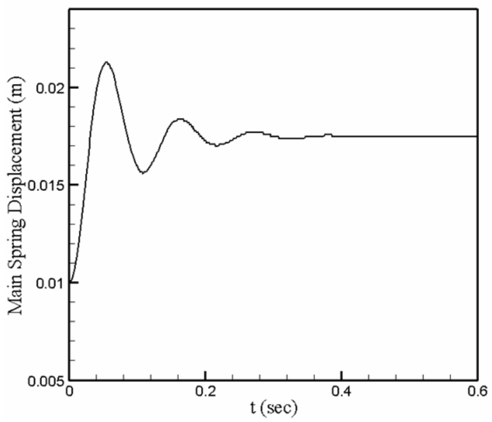

Fig. 4. Profile of the main spring displacement.

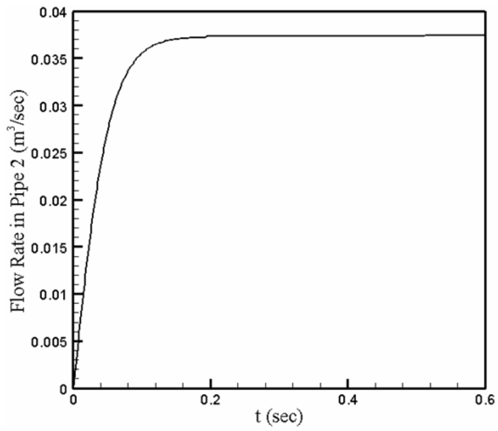

Fig. 5. Profile of the flow rate in Pipe 2.

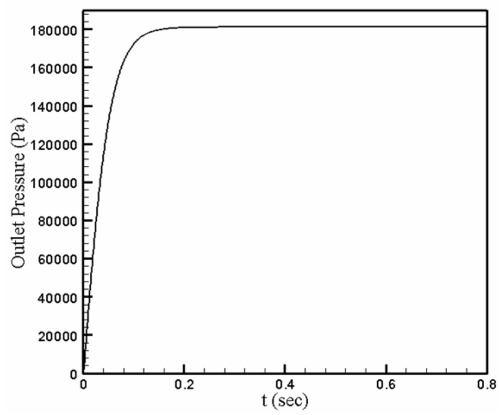

Fig. 6. Profile of the outlet pressure.

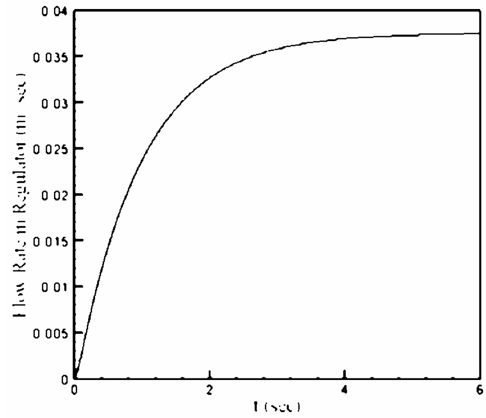

Fig. 7. Profile of the flow rate in the regulator.

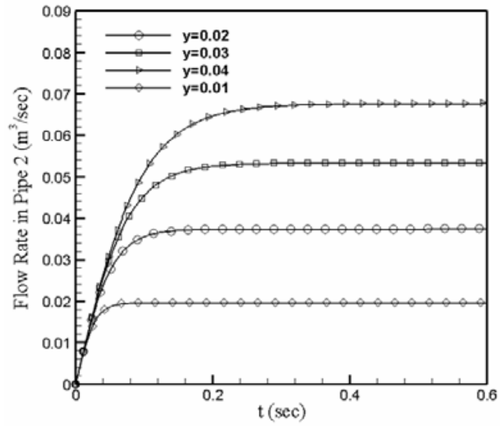

Fig. 8. The effect of screw adjustment (y) variations on the flow rate in Pipe 2.

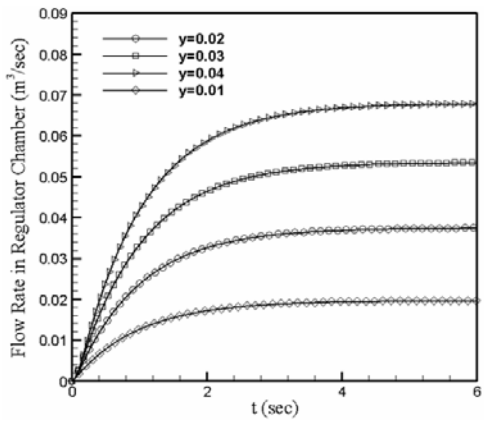

Fig. 9. The effect of screw adjustment (y) variations on the flow rate in the regulator chamber.

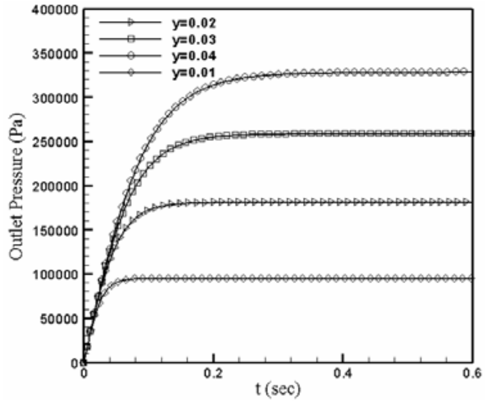

Fig. 10. The effect of screw adjustment (y) variations on the outlet pressure.

4. Conclusion

A dynamic model of a pressure regulator using the Bondgraph simulation method was achieved. The dynamic response of this model under real conditions was investigated using MATLAB-SIMULINK. The presentation of state variable histories contributes to the acceptable operation of the regulator, which decreases the inlet fluid pressure to a desirable pressure at an appropriate time. The nonlinear behavior of this valve is generally brought about by two elements. The first element is an orifice with a hydraulic resistance which varies with the motion of the mechanical subsystem according to a nonlinear function. The other element includes two adjusting screws that are manually controlled. An important advantage of this model is that its simplicity allows it to be used for a wider variety of system parameters. In fact, this model can be used in predicting application trends that may occur under various operating conditions, some of which may be difficult to create experimentally. In this regard, the obtained model takes into account the various nonlinearities of the system that appear in the performance of the two adjusting screws. The effects of various design parameters, such as the adjustment displacement on the overall response of the system, are also investigated through simulation. For example, the depicted plots show that an increase in the second screw displacement results in corresponding increases in the flow rate in Pipe 2 and in the regulator chamber. The obtained results are in agreement with the expected behaviors of pressure regulators.

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