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Engineering Notes

Static Characteristics of a Flow Regulator for a Liquid Rocket Engine

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Nomenclature

A_p	=	piston area of spool, m^2
A_t	=	end-tip area of spool, m^2
A_x	=	geometric flow area of slot, m^2
C_d	=	discharge coefficient
D	=	inlet, outlet port diameter, m
d	=	diameter of sliding rod, m
h	=	height of slot, m
k	=	spring constant of spring, N/m
N	=	number of slot
P_1, P_3	=	inlet, outlet pressure, bar
P_2	=	pressure in spool, bar
Q	=	flow rate, m^3/s
t_1	=	thickness of spool guide, m
t_2	=	thickness of spool, m
w	=	width of slot, m
x	=	spool position, m
x_o	=	precompression length of spring at $x = 0$, m
α	=	correction coefficient
ξ_1	=	hydraulic resistance of throttle, $1/m^4$
ξ_2	=	total hydraulic resistance of the lines and valves after flow regulator, $1/m^4$
μ	=	absolute viscosity, $N\,s/m^2$
ρ	=	propellant density, kg/m^3

I. Introduction

A FLOW regulator is one of the main control units for liquid rocket engines (LREs). It fulfills the following two functions: changing the flow rate to a gas generator or preburner for the thrust adjustment of the LRE and the automatically maintaining the flow rate to keep the LRE thrust within a specified range despite the change of pressure difference across the flow regulator. It is known that flow regulators have been widely used in Russian LREs. However, the literature related to flow regulators is not readily available, due to restrictions on the dissemination of information on rockets.

A flow regulator of the direct-acting type was introduced and only the operating mechanism was briefly explained by Kozlov [1]. Also, Lebedinsky et al. [2] simulated numerically the steady-state performance of a similar flow regulator. However, they neither performed experiments nor analyses of the design parameters affecting the performance of their flow regulators.

In the present Note, a new flow regulator of the direct-acting type shown in Fig. 1 is proposed. It is similar to the flow regulator proposed by Lebedinsky et al. [2], except an inlet pressure probe is added. The pressure probe makes the mathematical modeling of the flow regulator easier. It also plays the role of a damping unit, which can control transient response of the flow regulator although the dynamics of the flow regulator will not be considered in this Note. Design parameters, which affect the performance of the flow regulator, were identified through mathematical modeling and theoretical analysis. Experiments were also carried out. For the experiments, the flow regulator proposed in the present study was designed and fabricated. Experimental results of the flow regulator were compared with the simulation results of a mathematical model under steady-state conditions.

II. Design of a Flow Regulator

A flow regulator of direct-acting type shown in Fig. 1 is proposed and fabricated. The flow regulator consists of two parts to perform the two functions, which are changing the flow rate for thrust adjustment and maintaining thrust constant. The throttling parts (rotating shaft and sleeve) control the flow rate for a specified thrust, and the regulating parts (spool and spring) maintain the flow rate.

The operating mechanism for the maintenance of flow rate is as follows. From Fig. 1, when the inlet pressure P_1 increases, the spool moves to the right. This movement decreases the flow area of the slots. As a result, the flow rate can be kept constant within a specified range.

III. Experimental Setup and Test Method

For the derivation of the design parameters and the performance validation of the flow regulator, a test facility was equipped. Water was pressurized by pressure-regulated nitrogen gas and supplied to the flow regulator. There is a manual throttle valve after the flow regulator to modulate the hydraulic resistance. For flow rate measurements, turbine flow meters are installed. For static pressure measurements, static pressure transducers are installed. Water is expelled to a drain tank at atmospheric pressure. Pressure and flow rate are measured at a 100 Hz sampling rate.

IV. Mathematical Analysis

A. Mathematical Model

First, flow equations for the flow regulator and system will be established. Next, a force equation will be obtained by applying the principle of force equilibrium. This force equation along with the flow equations will lead to the characterization of the static behavior of the flow regulator. The leakage between the spool, the guides, and the sliding rod are ignored, because the clearance is very small. The fluid is also assumed to be incompressible.

1. Flow Equations

In general, the flow rate through an orifice or a valve is given by Eq. (1). Therefore, the flow rate through the throttle of the flow regulator can be expressed as

$$P_1 - P_2 = \tfrac{1}{2}\xi_1 \rho Q^2 \tag{1}$$

The flow rate through the slots of the flow regulator can also be expressed as

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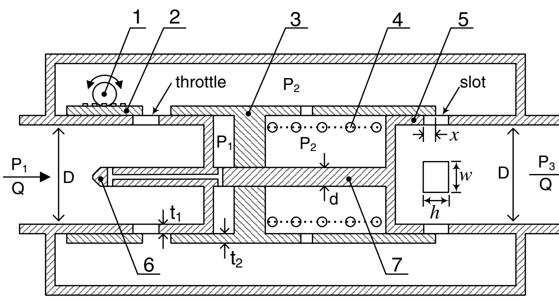

Fig. 1 Cross-sectional view of a flow regulator: 1, rotating shaft; 2, sleeve; 3, spool; 4, spring; 5, guide; 6, pressure probe; and 7, sliding rod.

$$Q = C_d N A_x \sqrt{\frac{2(P_2 - P_3)}{\rho}} \tag{2}$$

Since the slots are square, each flow area A_x is linearly proportional to the spool position x:

$$A_x = w(h - x) \tag{3}$$

The flow rate Q can also be expressed as follows.

$$P_3 = \tfrac{1}{2}\xi_2 \rho Q^2 \tag{4}$$

2. Force Equations

Forces acting on the spool under steady-state condition are listed as follows:

1) $F_p = (P_1 - P_2)A_p$ is the pressure force acting on the piston area of the spool.
2) $F_s = k(x_o + x)$ is the force by spring elasticity.
3) $F_t = \alpha(P_2 - P_3)A_t$ is the pressure force acting on the end tips of the spool.

Among the above forces, F_t should be compensated by a correction coefficient, α which can be found by experiment. The correction coefficient, α is considered constant regardless of spool position x, as proposed by Lebedinsky et al. [2].

By applying the principle of force equilibrium to the spool, one can establish a force-balance equation as follows:

$$(P_1 - P_2)A_p + \alpha A_t (P_2 - P_3) - k(x_o + x) = 0 \tag{5}$$

where A_p and A_t are given by

$$A_p = \frac{\pi}{4}[(D + 2t_1)^2 - d^2]$$
$$A_t = \frac{\pi}{4}[(D + 2t_1 + 2t_2)^2 - (D + 2t_1)^2]$$

3. Governing Equation

By combining Eqs. (1–5), one can construct a governing equation of the flow regulator system as follows:

$$\rho[\xi_1 A_p - (\xi_1 + \xi_2)\alpha A_t]Q^2 + 2k\beta + 2\alpha A_t P_1 - 2k(x_o + h) = 0 \tag{6}$$

where β is given by

$$\beta = \frac{1}{C_d N w}\sqrt{\frac{\rho Q^2}{2P_1 - (\xi_1 + \xi_2)\rho Q^2}} = h - x \tag{7}$$

From Eq. (6), Q can be calculated using numerical methods such as the Newton–Raphson or bisection methods.

B. Theoretical Analysis

In this section, an ideal design criterion and design parameters of the flow regulator are extracted from Eq. (6). Also, the mechanism of maintaining constant flow rate regardless of pressure change across the flow regulator is proven by theoretical analysis. As an ideal case, the correction coefficient α in Eq. (5) is set to zero in this section. If the thickness t_2 of the spool is very small, the effect of the second force term in Eq. (5) can be ignored.

1. In the Case of Inlet Pressure Change of the Flow Regulator

Equation (6) is rearranged as follows, when α is set to zero.

$$2P_1[\xi_1 A_p \rho Q^2 - 2k(h + x_o)] - \left[(\xi_1 + \xi_2)\rho Q^2\{\xi_1 A_p \rho Q^2 - 2k(h + x_o)\} + \frac{4k^2 \rho Q^2}{C_d^2 N^2 w^2}\right] = 0 \tag{8}$$

From Eq. (8), one can deduce that the terms in each square bracket should be equal to zero in order to meet the constant flow rate regardless of the change of inlet pressure, P_1. From the terms in the 1st set of square brackets, one can calculate the flow rate as follows:

$$\rho Q^2 = \frac{2k(h + x_o)}{\xi_1 A_p} \tag{9}$$

Combining Eqs. (1), (5), and (9) gives the following equation:

$$\frac{k(h + x_o)}{A} = \frac{k(x + x_o)}{A} \tag{10}$$

From Eq. (10), an ideal design criterion of the flow regulator is given by

$$x = h \tag{11}$$

Under the condition such as Eq. (11), the flow rate of the flow regulator is always constant, regardless of the change of inlet pressure. However, if x were equal to h, then the flow rate would be zero. Therefore, x should be as close to h as possible. From Eq. (7), one can easily deduce that the design parameters of the flow regulator are only N and w, which are related to the slot area of the flow regulator. Therefore, the larger N and w are, the closer x is to h. However, the larger N and w are, the larger the flow regulator becomes. This results in a weight increase of the flow regulator. Therefore, tradeoffs are required in designing the flow regulator.

The second set of square brackets in Eq. (8) should be equal to zero. When Eq. (9) is applied to the second set of square brackets, the remainder is as follows:

$$\frac{8k^3(x_o + h)}{C_d^2 N^2 w^2 \xi_1 A} \tag{12}$$

Although the remainder is not really zero, if N and w were increased, the remainder would be closer to zero.

2. In the Case of Outlet Pressure Change of the Flow Regulator

One can establish Eq. (13) by combining Eqs. (4) and (6), when α is set to zero:

$$\left[(2P_1 - \xi_1 \rho Q^2)\{\xi_1 A_p \rho Q^2 - 2k(x_o + h)\} + \frac{4k^2 \rho Q^2}{C_d^2 N^2 w^2}\right] - 2P_3[\xi_1 A_p \rho Q^2 - 2k(x_o + h)] = 0 \tag{13}$$

Through the same procedure in Sec. IV.B.1, one can obtain the same results.

Therefore, the ideal design criterion and the critical design parameters for maintaining a constant flow rate regardless of the inlet and outlet pressure changes of the flow regulator have been established.

V. Simulation and Experimental Results

The flow regulator shown in Fig. 1 has been fabricated for experiments. The specific value of design parameters are given in Table 1. Experiments were performed to verify the performance and

Table 1 Specifications of the flow regulator

Parameter	Value
w	8 mm
D	14.4 mm
N	4
h	8 mm
k	12,021 N/m
d	5 mm
x_o	1.8 mm
t_1	1.3 mm
t_2	2.0 mm

the mathematical model of the flow regulator by comparing them with the results of the numerical simulation.

A. Simulation Method

The bisection method was used to calculate Q from the nonlinear algebraic Eq. (6). P_1, ξ_1, and ξ_2 acquired from experiments are used as input data. In Eq. (7), C_d was measured along x and expressed by curve fitting as follows:

$$C_d = 5476.2x^2 + 34.1x + 0.3825 \tag{14}$$

Equations (6), (7), and (14) were calculated repeatedly until x converges, and then P_2 and P_3 were calculated using Eqs. (1) and (4) respectively.

B. Comparison of Simulation and Experimental Results

Experiments were performed along the pressure difference, $P_1 - P_3$ and compared with simulation results. Figure 2 shows the simulation and experimental results under the condition of $\xi_1 =$

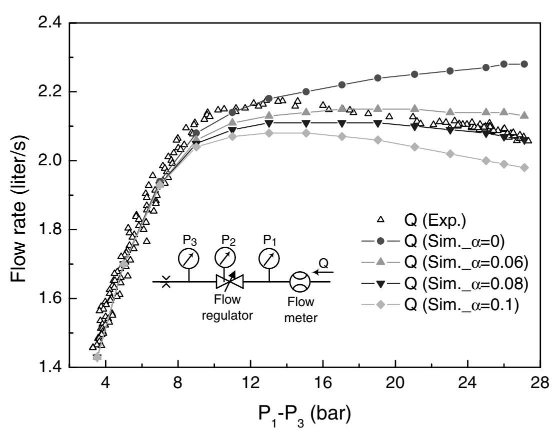

Fig. 2 Simulation and experimental results of flow rate along α.

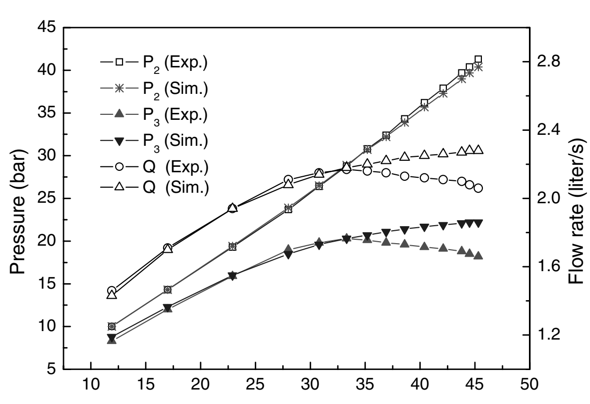

Fig. 3 Simulation and experimental results with $\alpha = 0$.

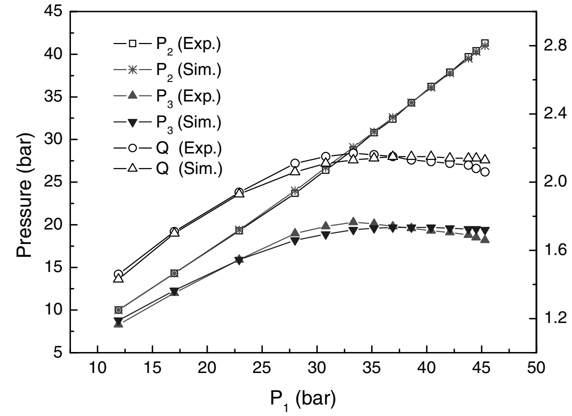

Fig. 4 Simulation and experimental results with $\alpha = 0.06$.

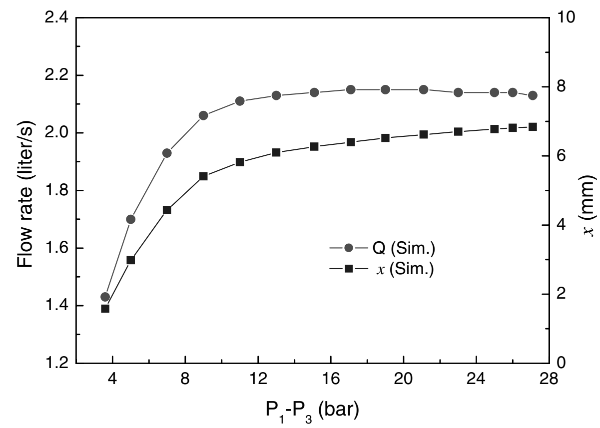

Fig. 5 Simulation results of flow rate, Q and spool position, x with $\alpha = 0.06$.

1.873×10^8 and $\xi_2 = 8.543 \times 10^8$. From the experimental results, it was proven that the flow regulator can maintain the flow rate effectively over about 10 bar of pressure difference. The flow rate changed only 5.3% when the pressure difference changed by 14.6 bar from 12.8 to 27.4 bar. On the other hand, one can see there are quite large discrepancies between the simulated and experimental results with $\alpha = 0$ in the region greater than about 13 bar. For more detailed analysis, see Fig. 3, which shows the results with $\alpha = 0$. One can deduce that the discrepancies of the flow rate result from the discrepancies of pressure, P_3 in the region greater than 35 bar of P_1. On the other hand, P_2 shows good agreement between simulated and experimental results. These results are due to the omission of the second force term in Eq. (5). Therefore, the pressure force acting on the end tip of the spool cannot be ignored in the flow regulator. From Fig. 2, one can deduce that a reasonable correction coefficient, α, is 0.06. On top of that, the flow rate decreases as α increases because the larger α is, the smaller the flow area of the slots is. Figure 4 shows good agreement between simulation and experimental results of pressure and flow rate with $\alpha = 0.06$. In Fig. 5, one can see that the closer x is to h, the better the performance of the flow regulator is. The performance of the flow regulator is drastically lowered in the region of x less than about 6.0 mm.

VI. Conclusions

The static characteristics of a newly proposed flow regulator have been studied. A flow regulator for any cycle of an LRE was designed and fabricated. Also, a test facility was constructed. A mathematical static model of the flow regulator has been developed. It was found from experiments that the flow regulator can effectively maintain a constant flow rate despite the change of pressure difference across the flow regulator.

The ideal design criterion and the main design parameters of the flow regulator were derived by theoretical analysis using a mathematical model. The main design parameters of the flow regulator affecting the performance of the flow rate maintenance are the number (N) and the width (w) of the slots. The performance improves as both N and w increase.

The simulation and experiments were conducted to verify the mathematical model of the flow regulator. The simulation results agreed well with the experimental results, which proved that the established mathematical model of the flow regulator is valid. Although the static characteristics were studied in this Note, a study on dynamic characteristics is also required to evaluate the validity of the flow regulator more fully.

Although the correction coefficient, α is considered constant regardless of spool position x, the correction coefficient depends on the design of the spool and the slots. Therefore, α should be newly defined by experiment when the design of the spool and the slots, such as size and shape, is modified.

The mathematical static model developed in this Note can be used to predict performance and to provide insight for improving the design of the flow regulator. It will also be useful in the design and analysis of LREs that use this flow regulator.

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