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Development of a dual-position oxidizer main valve

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Abstract
The supply of liquid oxygen to a rocket combustor is usually controlled using oxidizer shut-off valve with 'on' and 'off' positions. However, the partial opened position of a dual-position valve can control and optimize the engine start transients by regulating the liquid oxygen mass flow during the startup of the engine. In this paper, the design and performances of a dual-position pneumatic poppet valve, which is intended to be employed in liquid rocket engines, have been presented. Numerical analysis investigates the effects of key parameters, such as orifice size, stopper size, and valve inlet pressure, on the dynamic characteristics of valve opening and it is shown that the dual-position valve meets the expected requirements.

1. Introduction

The flow of cryogenic liquid oxygen to the combustor of a liquid rocket engine is controlled by a main oxidizer valve, which is installed between a turbopump and a combustor. It can be readily imagined that a faulty part of the valve failure could be catastrophic for any engine. Among the three valve types - poppet, butterfly, and ball - which are the most widely used as a shut-off valve, the poppet valve has proved to be the most reliable and versatile valve type for controlling the flow of cryogenic liquid oxygen in rocket engines.[1] The poppet valve is connected to an actuator which converts the energy of high pressure pilot gas to mechanical translation of the valve. The valve is pushed opened or closed by pressuring cavities and moving pistons by the actuator.

For extending the engine startup flexibility, a dual-position pneumatic poppet valve has been developed. The valve includes an intermediate position in addition to the fully opened and closed positions. The dual-position valve helps to reduce the abrupt impulse to the vehicle and engine by decreasing oxidizer flow rate during the engine startup. The engine startup using a relatively small amount of propellants at a sufficient high pressure insures stable engine ignition against hard start and prevents pump stall caused by sudden supply of large mass flow rate.[2] Partially opened position is required for controlling and optimizing the engine start transient. A ball valve had been developed for providing a partially opened position with the adjustable stop system for the Vinci engine.[3] The dual-position actuator of the ball valve required two independent pilot supplies. However, the poppet valve in the present study can provide an intermediate position with one pilot supply. The valve operating with one pilot supply is attractive as the simplification of the system configuration can lessen the risk of engine failure by reducing its components.

The operating mechanism of the dual-position valve is briefly described in the next section, and the numerical analysis are presented in the following sections to understand the effects of design parameters on the opening characteristics of the valve.

2. Operating mechanism

Figure 1 shows that two pilot cavities (Cavity 1 and Cavity 2), which are connected to each other, are used to achieve different pilot pressures during the pressurization and depressurization of the cavities. The operation of the dual-position valve is achieved by creating a difference in pilot pressure between the two cavities.[4]

Port 1 (P1) is used to supply and discharge high-pressure pilot gas. The pilot gas enters Cavity 1 through P1 and then flows into Cavity 2 via a tube connected between Ports 2 (P2) and 3 (P3). During the filling of the pilot gas inside the cavities, p_{a_1} remains higher than p_{a_2} due to the pilot gas flow passage (Cavity 1 to Cavity 2), i.e. $p_{a_1} > p_{a_2}$. The rate at which pilot pressure rises in two cavities is mainly determined by an orifice installed inside the tube. The pneumatic force induced by p_{a_1} pushes the stopper down to an intermediate position, preventing the valve moving part from reaching a full stroke. At this moment, the valve is partially opened due to pressurization inside the two cavities (shown as (c) in Fig. 1).

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After filling both cavities, the pilot gas is discharged through P1 to fully open the valve. During depressurization, the pilot gas flows from Cavity 2 to Cavity 1, causing p_{a_2} to remain higher than p_{a_1}, i.e. $p_{a_2} > p_{a_1}$. The pneumatic force due to p_{a_2} overcomes the total opposite force, including pneumatic force of p_{a_1}, and pushes up the valve moving part to a full stroke (as shown in (d) in Fig. 1). Once the valve is fully opened, the opening position is maintained solely by the hydraulic force of the passage flow without requiring pilot gas pressurization.

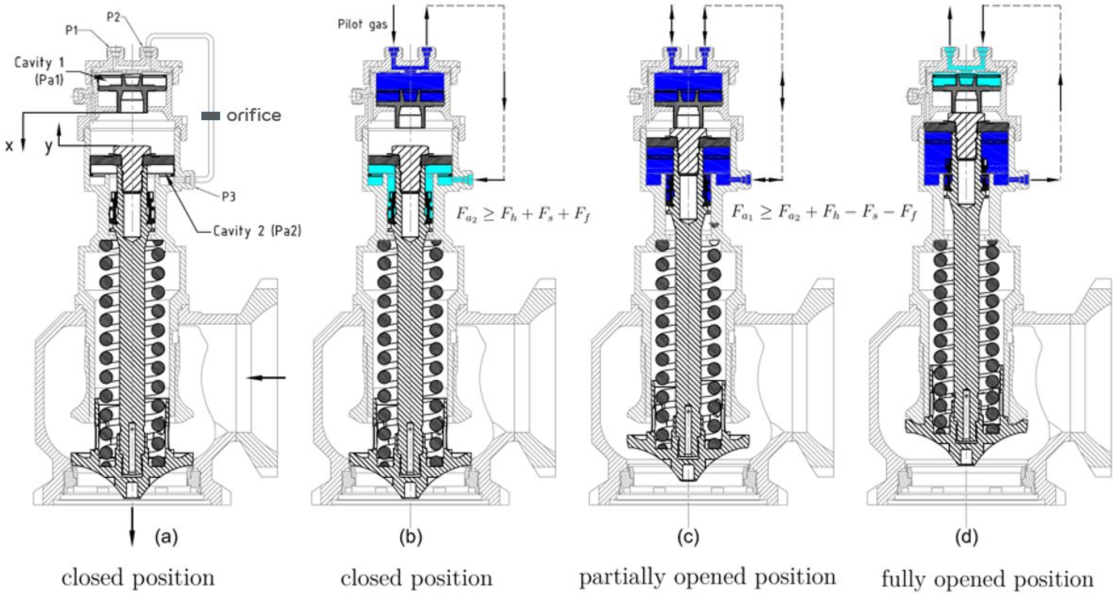

Figure 1: Operating mechanism of a dual-position poppet valve

3. Numerical analysis

The opening characteristics of the dual-position valve were evaluated by solving the equation of motion of the valve moving part ($F = ma$) through numerical analysis.

3.1 Pilot gas pressure & mass flow rate

Pneumatic forces, to be more precise, pilot pressures in the motion equation are obtained using the ideal gas equation with the calculated pilot gas mass inside the cavities. The pilot gas pressure and its differentiation with respect to time are derived from the idea gas equation as the following.

$$\dot{p}_a(t) = \frac{\dot{m}(t)RT}{V(t)} - \frac{p_a(t)\dot{V}(t)}{V(t)} \tag{1}$$

Due to the large diameter of the cavities, it is to consider the change in volume, $\dot{V}$ inside the cavities. The mass flow rate of the pilot gas, $\dot{m}$ in Eq. (1) is calculated based on the mass flow rate of the compressible gas flow.

3.2 Up to the intermediate position

The operation of opening the valve can be divided into two parts based on the position of valve. First, the valve begins to open and its movement is then restrained by a stopper at the intermediate position. The second part of the valve opening operation corresponds to a full opening by the movement of both the valve and the stopper as one unit.
The total force applied to the valve moving part at the initial closed position, without pilot gas, is as follows.

$$F = -F_h - F_s - F_f = -p_h A_{h_d} - F_s - F_f \tag{2}$$

Here, A_{h_d} refers to the area on which the downward hydraulic force is exerted. Once the pilot gas is supplied to Cavity 2 through Cavity 1, p_{a_2} increases and the valve begins to open the moment $F_{a_2} = F_h + F_s + F_f$ is reached. The retarding

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force caused by the passage flow is not considered here because of its negligible magnitude. The equation of motion for the valve moving part up to the intermediate position can be expressed simply as follows.

$$F = F_{a_2} + F_h - F_s - F_f = p_{a_2}(t)A_{a_2} + p_h A_{h_u} - k\left(l + y(t)\right) - F_f \tag{3}$$

The direction of the hydraulic force immediately become opposite when the valve opens. The acceleration of the moving part during the valve opening can then be expressed as follows.

$$\ddot{y} = \frac{1}{m_2}\left(p_{a_2}(t)A_{a_2} + p_h A_{h_u} - k\left(l + y(t)\right) - F_f\right) \tag{4}$$

The valve travel, $y(t)$ can be determined by the equation of motion during the valve opening. To perform numerical calculation, a first-order ordinary differential equation system is formulated using the following variables.

$$\mathbf{X} = \begin{bmatrix} x_1(t) \\ x_2(t) \\ x_3(t) \\ x_4(t) \\ x_5(t) \\ x_6(t) \end{bmatrix} = \begin{bmatrix} x(t) \\ y(t) \\ \dot{x}(t) \\ \dot{y}(t) \\ p_{a_1}(t) \\ p_{a_2}(t) \end{bmatrix} \tag{5}$$

Subscripts 1 and 2 correspond to the parameters of Cavity 1 and Cavity 2, respectively. The time differentials of the variables are as follows.

$$\mathbf{X}' = \begin{bmatrix} \dot{x}_1(t) \\ \dot{x}_2(t) \\ \dot{x}_3(t) \\ \dot{x}_4(t) \\ \dot{x}_5(t) \\ \dot{x}_6(t) \end{bmatrix} = \begin{bmatrix} \dot{x}(t) \\ \dot{y}(t) \\ \frac{1}{m_1}\left(p_{a_1}(t)A_{a_1} - F_{f_1}\right) \\ \frac{1}{m_2}\left(p_{a_2}(t)A_{a_2} + p_h A_{h_u} - k(l + y(t)) - F_{f_1}\right) \\ \begin{cases} \left(\frac{A_1 p_o}{\sqrt{RT/\gamma}}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} - \dot{m}_{out}\right) \times \frac{RT}{V_{0_1} + \Delta V_1 x(t)/H_1} \\ \quad - \frac{p_{a_1}(t)}{V_{0_1} + \Delta V_1 x(t)/H_1} \times \frac{\Delta V_1 \dot{x}(t)}{H_1}, \quad \text{for } \frac{p_{a_1}}{p_o} < \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} \\ \left(\frac{A_1 p_o/R}{\sqrt{T/(2c_p)}}\left(\frac{p_{a_1}}{p_o}\right)^{1/\gamma}\sqrt{1 - \left(\frac{p_{a_1}}{p_o}\right)^{(\gamma-1)/\gamma}} - \dot{m}_{out}\right) \times \frac{RT}{V_{0_1} + \Delta V_1 x(t)/H_1} \\ \quad - \frac{p_{a_1}(t)}{V_{0_1} + \Delta V_1 x(t)/H_1} \times \frac{\Delta V_1 \dot{x}(t)}{H_1}, \quad \text{for } \frac{p_{a_1}}{p_o} \geq \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} \end{cases} \\ \begin{cases} \frac{A_2 p_{a_1}(t)}{\sqrt{RT/\gamma}}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \times \frac{RT}{V_{0_2} + \Delta V_2 y(t)/H_2} \\ \quad - \frac{p_{a_2}(t)}{V_{0_2} + \Delta V_2 y(t)/H_2} \times \frac{\Delta V_2 \dot{y}(t)}{H_2}, \quad \text{for } \frac{p_{a_2}}{p_{a_1}} < \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} \\ \frac{A_2 p_{a_1}/R}{\sqrt{T/(2c_p)}}\left(\frac{p_{a_2}}{p_{a_1}}\right)^{1/\gamma}\sqrt{1 - \left(\frac{p_{a_2}}{p_{a_1}}\right)^{(\gamma-1)/\gamma}} \times \frac{RT}{V_{0_2} + \Delta V_2 y(t)/H_2} \\ \quad - \frac{p_{a_2}(t)}{V_{0_2} + \Delta V_2 y(t)/H_2} \times \frac{\Delta V_2 \dot{x}(t)}{H_2}, \quad \text{for } \frac{p_{a_2}}{p_{a_1}} \geq \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} \end{cases} \end{bmatrix} \tag{6}$$

The numerical simulation was performed using the explicit Runge-Kutta 4th-order method.

3.3 Up to full opening position

Once the pilot gas is discharged through P1, p_{a_2} becomes higher than p_{a_1}. The valve moving part with the stopper starts opening further if the pneumatic force inside Cavity 2 (F_{a_2}) gets higher than the sum of other forces that push the moving part to the closing direction. The sum of the forces applied to the moving part at the intermediate position is as follows.

$$F = F_{a_2} + F_h - F_s - F_f - F_{a_1} = p_{a_2}(t)A_{a_2} + p_h A_{h_u} - k\left(l + y(t)\right) - F_f - p_{a_1}(t)A_{a_1} \tag{7}$$

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The acceleration of the moving part can be expressed as the following.

$$\ddot{y} = \frac{1}{m_t}\left(p_{a_2}(t)A_{a_2} + p_h A_{hu} - k\left(l + y(t)\right) - F_f - p_{a_1}(t)A_{a_1}\right) \tag{8}$$

Here, $m_t = m_1 + m_2$ and $F_f = F_{f_1} + F_{f_2}$. The first-order ordinary differential equation system for describing the fully opening is as follows.

$$\mathbf{X}' = \begin{bmatrix} \dot{x}_1(t) \\ \dot{x}_2(t) \\ \dot{x}_3(t) \\ \dot{x}_4(t) \\ \dot{x}_5(t) \\ \dot{x}_6(t) \end{bmatrix} = \begin{bmatrix} \dot{x}(t) \\ \dot{y}(t) \\ \frac{-1}{m_1+m_2}\left(p_{a_2}(t)A_{a_2} + p_h A_{h_u} - k(l + H_2 - x(t)) - F_{f_1} - F_{f_2} - p_{a_1}(t)A_{a_1}\right) \\ \frac{1}{m_1+m_2}\left(p_{a_2}(t)A_{a_2} + p_h A_{h_u} - k(l + y(t)) - F_{f_1} - F_{f_2} - p_{a_1}(t)A_{a_1}\right) \\ \begin{cases} \left(\dot{m}_{in} - \frac{A_{e_1} p_{a_1}(t)}{\sqrt{RT/\gamma}}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}}\right) \times \frac{RT}{V_{0_1} + \Delta V_1 x(t)/H_1} \\ \quad - \frac{p_{a_1}(t)}{V_{0_1} + \Delta V_1 x(t)/H_1} \times \frac{\Delta V_1 \dot{x}(t)}{H_1}, \quad \text{for } \frac{p_e}{p_{a_1}} < \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} \\ \left(\dot{m}_{in} - \frac{A_{e_1} p_{a_1}/R}{\sqrt{T/(2c_p)}}\left(\frac{p_e}{p_{a_1}}\right)^{1/\gamma} \sqrt{1 - \left(\frac{p_e}{p_{a_1}}\right)^{(\gamma-1)/\gamma}}\right) \times \frac{RT}{V_{0_1} + \Delta V_1 x(t)/H_a} \\ \quad - \frac{p_{a_1}(t)}{V_{0_1} + \Delta V_{0_1} x(t)/H_1} \times \frac{\Delta V_1 \dot{x}(t)}{H_1}, \quad \text{for } \frac{p_e}{p_{a_1}} \geq \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} \end{cases} \\ \begin{cases} -\frac{A_{e_2} p_{a_2}(t)}{\sqrt{RT/\gamma}}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \times \frac{RT}{V_{0_2} + \Delta V_2 y(t)/H_2} \\ \quad - \frac{p_p(t)}{V_{0_2} + \Delta V_2 y(t)/H_2} \times \frac{\Delta V_2 \dot{y}(t)}{H_2}, \quad \text{for } \frac{p_{a_1}}{p_{a_2}} < \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} \\ -\frac{A_{e_2} p_{a_2}/R}{\sqrt{T/(2c_p)}}\left(\frac{p_{a_1}}{p_{a_2}}\right)^{1/\gamma} \sqrt{1 - \left(\frac{p_{a_1}}{p_{a_2}}\right)^{(\gamma-1)/\gamma}} \times \frac{RT}{V_{0_2} + \Delta V_2 y(t)/H_2} \\ \quad - \frac{p_p(t)}{V_{0_2} + \Delta V_2 y(t)/H_2} \times \frac{\Delta V_2 \dot{y}(t)}{H_2}, \quad \text{for } \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} < \frac{p_{a_1}}{p_{a_2}} \leq 1 \\ \frac{A_{e_2} p_{a_1}/R}{\sqrt{T/(2c_p)}}\left(\frac{p_{a_2}}{p_{a_1}}\right)^{1/\gamma} \sqrt{1 - \left(\frac{p_{a_2}}{p_{a_1}}\right)^{(\gamma-1)/\gamma}} \times \frac{RT}{V_{0_2} + \Delta V_2 y(t)/H_2} \\ \quad - \frac{p_p(t)}{V_{0_2} + \Delta V_2 y(t)/H_2} \times \frac{\Delta V_2 \dot{y}(t)}{H_2}, \quad \text{for } \frac{p_{a_1}}{p_{a_2}} > 1 \end{cases} \end{bmatrix} \tag{9}$$

Here, $\dot{m}_{in}$ represents the mass flow rate of the pilot gas into Cavity 1 from Cavity 2. As the valve moving part and the stopper move together, $x(t) = H_2 - y(t)$, and $\dot{x}(t) = -\dot{y}(t)$, and $\ddot{x}(t) = -\ddot{y}(t)$.

4. Opening characteristics

To ensure that the engine reaches full thrust level within a specified time, critical specifications related to valve opening characteristics must be defined for engine startup operation. For instance, the time it takes for the valve to fully open from the opening command should be less than 1.0 second. For a stable intermediate stage of the engine, the time during which the pilot pressures inside two cavities (C1 and C2) maintain the pilot source pressure (= 22 MPaA) should be more than 0.2 seconds. This is one of the principal considerations on the valve opening characteristics. In order to effectively control engine start, p_{a_2} should increase as quickly as possible during the first step opening (for partial opening), provided that the increasing rate of p_{a_2} is much higher than that of the valve inlet pressure p_h. During the second step opening (for full opening), p_{a_2} should decrease slowly enough for reliable valve opening.

4.1 Analysis results

Figure 2 shows the evolution of pneumatic pressures in the Cavities 1 and 2 with valve dynamics calculated by numerical analysis. As shown in Fig. 2(a), the pneumatic pressures in two cavities change successively by the orifice with a diameter of 1.8 mm installed between two cavities. p_{a_1} increases and decreases faster than p_{a_2} during the pressurization

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and the depressurization of the cavities. The effective diameters of the pilot gas supply system including the friction forces obtained from actual operating results are used in the numerical calculation.[5,6]
Once p_{a_2} increases up to about 7 MPaA, the valve starts to open and is eventually stopped by the stopper that is pressured by the pilot gas. Points A and B represent the moments when the valve starts to open and arrives at the partially opening position, respectively. During the partially opening, p_{a_2} drops abruptly due to the fast increment of the volume of Cavity 2. After maintaining the pilot source pressure inside the cavities for a period of time ($\simeq$ 0.2 sec.), when the pilot gas is discharged, the valve starts to further open at p_{a_1} = 19.2 MPaA and p_{a_2} = 21.1 MPaA (at Point C in Fig. 2). The valve continues to open up to the full opening position (at Point D in Fig. 2) due to the pressure difference between Cavities 1 and 2. The volume changes in Cavities 1 and 2 during the valve opening result in an increment of the pressure decrease rates ratio, dp_{a_2}/dp_{a_1}, as shown in pressure evolutions between Points C and D.

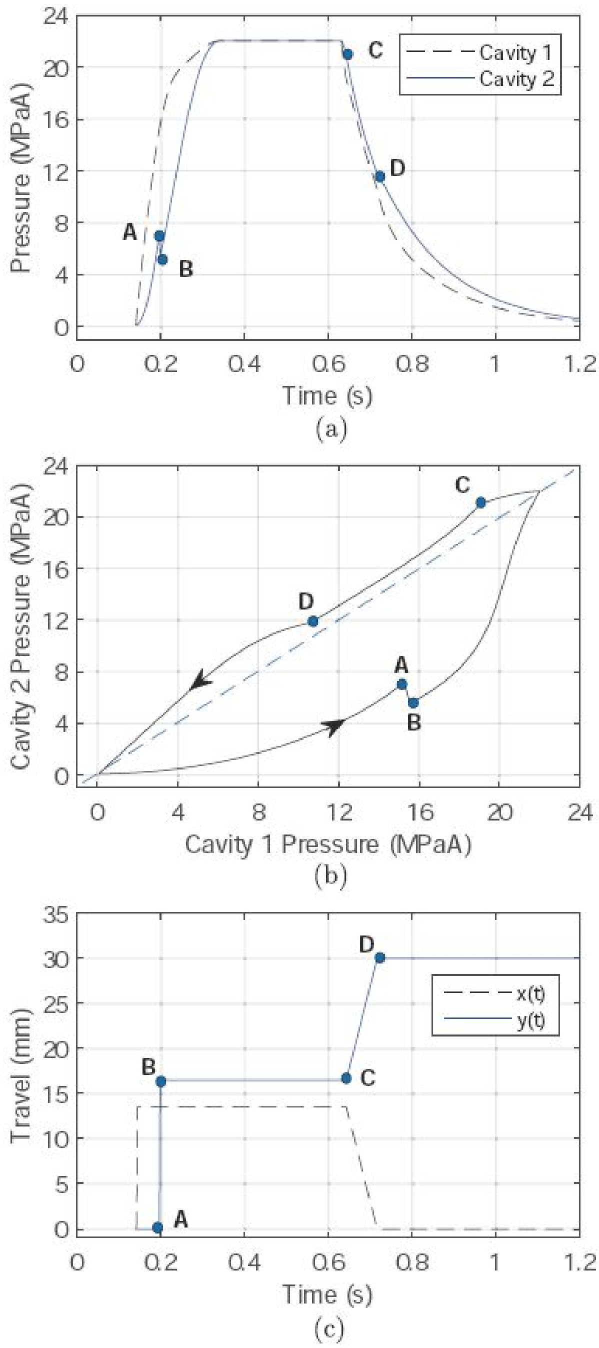

Figure 2: Numerical analysis on cavity pressure evolutions and valve travel (d_1 = 2.4 mm, d_2 = 1.8 mm, d_{a3} = 62 mm, p_{h_0} = 2 MPaG).

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4.2 Effects of the orifice size, the stopper size, the stopper size, and the valve inlet pressure

The valve opening characteristics were analyzed numerically to investigate the impact of the orifice size, stopper size, and valve inlet pressure. Initially, the impact of orifice size d_2 on valve opening dynamics is examined. The rate at which p_{a_2} rises is crucial in effectively controlling valve opening time. To open the valve against the hydraulic force of passage flow at a design moment, p_{a_2} must increase much faster than valve inlet pressure p_h. It is evident that p_{a_2} with a larger orifice increases more quickly, but the valve traveling velocity required for full opening position decreases as orifice size increases. Therefore, the orifice size d_2 must be determined while considering the rate at which p_{a_2} increases and the valve traveling time

The opening characteristics up to the partial opening position do not seem to be affected by the stopper size. However, p_{a_2}, at which the valve begins to open for full opening position, decreases and valve travel time increases as the stopper size becomes larger. The appropriate p_{a_2}, at which the valve begins to open up to full opening position, must be determined while considering the valve travel time, the difference in pilot gas source pressure, and the valve envelope influenced by stopper size. In particular, a very small difference in pilot gas pressure could result in unstable operation in partial opening due to insufficient pneumatic force inside Cavity 1, which keeps pressing the valve moving part.

To optimize the engine start operation, a specific range of valve inlet pressures at the valve opening moment are considered. Therefore, it is important to understand the valve opening characteristics according to the valve inlet pressure. As valve inlet pressure increases, the time to start opening for partial opening is delayed, and the valve opens faster to the full opening position. The valve inlet pressure at the valve opening moment must be determined while considering its impact on valve opening dynamics related to engine startup operation.

5. Conclusions

A dual-position pneumatic poppet valve has been developed, which provides a partially opened position. Numerical analysis, along with actual measurement data, confirms that the valve meets the expected requirements. The numerical analysis also investigates the effects of key parameters, such as orifice size, stopper size, and valve inlet pressure, on the dynamic characteristics of valve opening. Engine tests using the dual-position pneumatic poppet valve are planned in the near future to confirm its functionality during engine startup operations.

Nomenclature

A_a = area where F_a is applied, [N]
A_{h_d} = area on which the downward hydraulic force is applied at valve opening, [N]
A_{h_u} = area on which the upward hydraulic force is applied at valve opening, [N]
F_a = pneumatic force due to p_a, [N]
F_f = friction force, [N]
F_h = hydraulic force, [N]
F_s = spring force, [N]
H_1 = max travel of stopper, [mm]
H_2 = max valve travel, [mm]
V = volume filled with pilot gas, [m^3]
d = effective diameter for entering pilot gas [mm]
d_{a3} = stopper diameter, [mm]
k = stiffness of spring, [N/mm]
l = initial compressed length of spring, [mm]
m = mass of pilot gas, [kg]
m_1 = mass of stopper, [kg]
m_2 = mass of valve moving part, [kg]
p_a = pilot gas pressure inside the cavity, [MPa]
p_o = source pressure of pilot gas, [MPa]
x = travel of stopper, [mm]
y = valve travel, [mm]

subscript

1 = Cavity 1 (C1)
2 = Cavity 2 (C2)

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