

# Inflow-Driven Valve System for Pulse Detonation Engines

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A new valve system for a pulse detonation engine that uses a simple inflow-driven piston-interrupting valve is proposed. This valve can generate an intermittent flow by using gas enthalpy, so no power source or control unit is necessary. The mass flow per valve unit mass is comparatively large, the thrust change with changing supply pressure is very responsive, and the inflow supply pressure range for stable operation is wide. In a mass flow rate measurement experiment using a single-piston inflow-driven valve, the operation frequency and mass flow rate were predictable. In a thrust measurement experiment with a pulse detonation rocket engine using a three-piston inflow-driven valve, the pulse detonation engine's stable operation over a wide range of supply pressure was confirmed and the time-averaged thrust was measured. The maximum time-averaged thrust of 22.6 N was achieved at a fuel (ethylene) supply pressure of 0.95 MPa and an oxygen supply pressure of 1.9 MPa. The maximum specific impulse of 279 s was achieved at an ethylene supply pressure of 0.8 MPa and an oxygen supply pressure of 1.6 MPa.

## Nomenclature

$A$	=	cross section
$e$	=	rebound coefficient
$F$	=	time-averaged thrust
$f$	=	operation frequency
$g$	=	gravitational acceleration
$I$	=	impulse
$I_{sp}$	=	specific impulse
$k$	=	spring constant
$M$	=	molar mass
$m$	=	mass
$\dot{m}$	=	mass flow rate
$N$	=	number of cycle
$p$	=	absolute pressure
$\mathfrak{R}$	=	universal gas constant
$S$	=	spark noise
$T$	=	absolute temperature
$t$	=	time
$u_{11}$	=	piston velocity at right wall ( $w1$ ) before collision
$u_{12}$	=	piston velocity at right wall ( $w1$ ) after collision
$u_{21}$	=	piston velocity at left wall ( $w2$ ) before collision

$u_{22}$	=	piston velocity at left wall ( $w2$ ) after collision
$V$	=	volume
$v$	=	specific volume
$w$	=	mechanical work
$x$	=	displacement
$\dot{x}$	=	velocity
$\ddot{x}$	=	acceleration
$\alpha$	=	pressure correction coefficient
$\Delta m$	=	mass difference
$\Delta t$	=	time interval
$\mu$	=	friction coefficient
$\phi$	=	equivalent ratio
$\psi$	=	propellant fill fraction

## Subscripts

$a$	=	ambient gas
ave	=	average
$b$	=	inside of a tank
cal	=	calculation
cr	=	critical
cycle	=	one cycle
$e$	=	exhaust
exp	=	experiment
$f$	=	fuel
$i$	=	purge gas
ign	=	ignition
in	=	inside of a cylinder
lc	=	load cell
$n$	=	natural length
$o$	=	oxidizer
ope	=	operation
out	=	outside of a cylinder
$p$	=	piston
$s$	=	supply
start	=	start point of total cycle
stop	=	stop point of total cycle
$t$	=	pulse detonation engine tube
$w1$	=	a right wall
$w2$	=	a left wall
0	=	before experiment
1	=	after experiment

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## I. Introduction

SINCE the detonation phenomenon was discovered in 1881, many fundamental studies [1,2] and application studies of detonation waves [3–11] have been carried out. A pulse detonation engine (PDE) [12–22] is a thermal engine that generates a detonation wave intermittently and performs its mechanical work or thrust in response to high-pressure gas generated by a detonation. A PDE can obtain nearly constant thrust and perform mechanical work by the generation of a high-frequency detonation wave. PDEs are classified based on whether they are airbreathing PDEs [23–29] or pulse detonation rocket engines (PDREs) [30–37] and based on whether they include a turbine, as pure PDEs, or not, as pulse detonation turbine engines [38,39]. According to Endo et al. [39], the thermal efficiency of the PDE cycle is higher than that of constant-volume combustion (Humphrey cycle) or constant-pressure combustion (Brayton cycle). With a stoichiometric hydrogen–air mixture, the thermal efficiency is 28% if the initial compression ratio is one. With a higher compression ratio, the thermal efficiency becomes higher. In a PDE, the propellant gas is compressed and combusted by a shock wave; thus, a PDE can generate high pressure without compression mechanisms, such as compressors and pistons. One goal in the development of supply valve systems is to determine how to supply the propellant intermittently while maintaining the simplified structure of a PDE. Valve technology and detonation initiation methods are core PDE technologies in the practical realization of a PDE. Many studies of PDE valves have been reported [32,40–45]. In particular, a valve system is needed that can achieve a large enough mass flow rate per valve unit to obtain a high thrust–weight ratio.

Figure 1 shows the schematic diagram of a valve system. The valve is composed of four main parts: an inflow blocker, a drive power that drives the blocker, an energy source that generates the drive power, and a control unit that controls the energy source (see Table 1). The valves for a PDE are classified into two broad groups: rotary valves [40] and reciprocal valves. The rotor plate of a rotary valve is driven by a motor (electromagnetic power), and the mass flow rate per valve unit mass is large because of the rotary movement. Such a valve can operate at high frequency by increasing the number of rotations of the valve. However, in the case of a large inertia moment around the axis, the response in thrust change is slow. Moreover, gas leaking occurs on the rotation sliding surface.

Reciprocal valves can be classified as piston-interrupting valves and gas-interrupting valves based on the kind of inflow blocker. If the piston mass of the piston-interrupting valve is light, the response in thrust change is quick, but the mass flow per valve unit mass is small because of the coil needed to drive the piston. A gas-interrupting valve (valveless) interrupts the inflow by means of the pressure gradient between the inflow and outflow generated by a detonation wave. Since this valve has no movable parts, the mass flow rate per valve unit mass is quite high, but the region of inflow for stable operation is narrow, because the inflow is restricted by the outflow. Each of these valve types has advantages and disadvantages, and there are technical problems when applying them to PDEs.

Recently, a PDE concept was proposed by Golub et al. [42] and Golovastov et al. [43] in which the piston can reciprocate in response to spring force and enthalpy generated by the detonation wave,

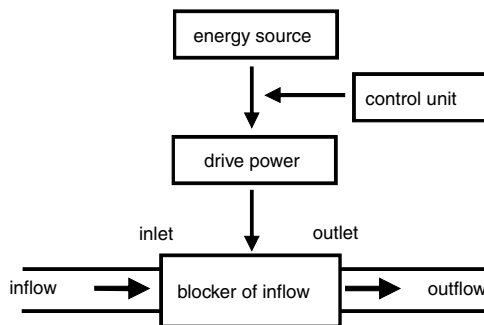


Fig. 1 Schematic diagram of a valve system.

Table 1 Classification of general valve systems

Large classification	Small classification	Inflow blocker	Drive power	Energy source	Control part	Response in thrust change	Weight	Pressure range of inflow	Reference
Rotary valve	Piston- interrupting valve	Rotor plate	Electromagnetic power	Electrical energy	Control of rotary motor voltage	Slow	Light	Wide	[40]
	Gas- interrupting valve (valveless)	Piston	Electromagnetic power	Electrical energy	Control of on-off of electromagnet	Quick	Heavy	Wide	[32]
	Outflow-driven piston- interrupting valve	Outflow	Pressure	Enthalpy of outflow	Control of ignition timing	Quick	Very light	Narrow	[41]
	Inflow-driven piston- interrupting valve	Piston	Pressure	Enthalpy of outflow	Control of ignition timing	Quick	Light	Narrow	[42,43]
Reciprocal valve			Pressure	Enthalpy of inflow	Control of inflow flow rate	Quick	Light	Wide	[44,45]

thereby supplying propellant to a combustor intermittently. This system is very simple, because no power source or controller unit is necessary, but it is difficult to operate over a wide range of supply pressures because the supply pressure depends on the outflow conditions, such as the presence of a gas-interrupting valve.

Golovastov et al. [43] and Yamaguchi et al. [44] proposed an inflow-driven valve (IDV) in which inflow is used to interrupt the piston. This valve does not require an energy source inside the valve system, so the mass flow rate per valve unit mass is comparatively large and the region of inflow for stable operation is wide. Since this valve can change the thrust by simply changing the mass flow rate of the inflow, the response of the thrust change is very good. Baklanov et al. [41] carried out a combustion experiment with a two-piston IDV and found that the region of inflow for stable operation was restricted because their two-piston IDV did not have a piston to handle the purge gas. The authors did not measure the thrust in their experiment.

It is difficult to consider a supply valve system separately from the combustion chamber, because there are interaction problems of detonation initiation (purge, mixing, and ignition) and energy loss due to dissipation. Even if we can independently establish a supply valve system, we cannot confirm whether detonation can occur in the combustor until we carry out the thrust measurement by the total system. In the case of PDREs, we must concentrate on specific impulse performance and aim for a lightweight, simple design. With these goals in mind, we established a PDRE system with a three-piston IDV that included a piston for the purge gas. We performed thrust measurements using this PDRE system and evaluated the stable operation and performance of the thrust.

## II. Inflow-Driven Valve

### A. Inflow-Driven Valve Principle

An IDV [41] is composed of three main parts: a piston, a cylinder, and a spring. Figure 2 shows the operation conceptual diagram of an IDV, and Fig. 3 shows the ideal and actual operation cycles of an IDV as an  $x-p$  diagram.

When the piston is as position i in Fig. 2 (also position i in Fig. 3), propellant gas of supply pressure  $p_s$  is supplied from a tank into the cylinder and pushes the piston. The pressure difference  $p_s - p_a$  between the supply pressure  $p_s$  and the ambient pressure of the valve (generally atmospheric pressure)  $p_a$  is generated, and the piston begins to move in the positive direction of  $x_p$  in Fig. 2 (positions i to ii). In Fig. 3, the solid line shows the ideal operation cycle and the dashed line shows the actual operation cycle.

In the ideal operation cycle, the cylinder volume is significantly larger than the change of cylinder volume caused by moving the piston, and the supply pressure is constant while the piston displacement is in  $x_p \leq x_e$ . The exhaust process in  $x_p = x_e$  and supply process in  $x_p = x_s$  are carried out momentarily. When the

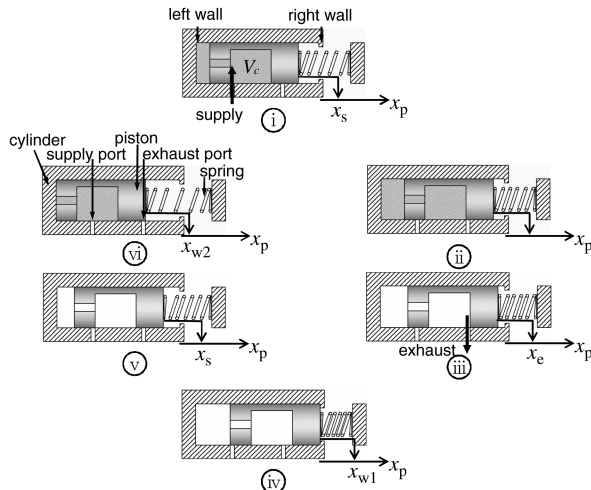


Fig. 2 Operation conceptual diagram of an IDV.

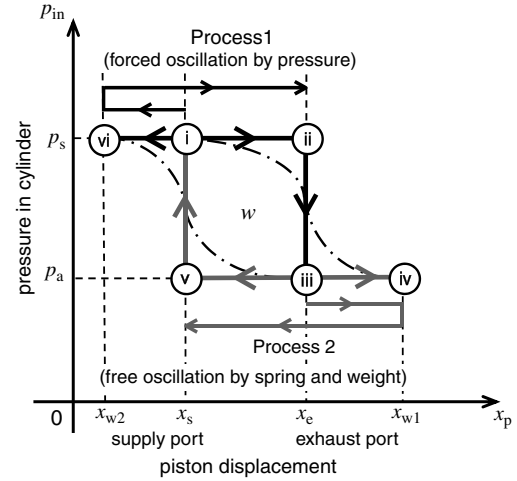


Fig. 3 Ideal and actual operation cycles of an IDV.

piston reaches exhaust position  $x_e$ , the propellant in the cylinder is exhausted and injected into the combustor (position iii).

After exhaustion, the piston is moved to position  $x_{w1}$  by inertia force. The piston rebounds at the right wall of the cylinder (position iv). Next, the spring pushes the piston in the negative direction of  $x_p$ . The piston passes through the supply position  $x_s$  (position v), and the propellant is supplied to the cylinder again (position i). The internal pressure of the cylinder rises, and the piston velocity decreases. After that, the piston reaches the left wall of the cylinder  $x_{w2}$  (position vi), and the piston returns to position i (position i). The piston repeats the following cycle: position i ( $\dot{x}_p > 0$ ) goes to position ii, which goes to position iii ( $\dot{x}_p > 0$ ), which goes to position iv, which goes to position iii ( $\dot{x}_p < 0$ ), which goes to position v, which goes to position i ( $\dot{x}_p < 0$ ), which goes to position vi, which goes to position i ( $\dot{x}_p > 0$ ). The operation principle of an IDV is composed of two processes. Process 1 is that the piston is vibrated by the supply pressure while position i ( $\dot{x}_p < 0$ ) goes to position vi, which goes to position i ( $\dot{x}_p > 0$ ), which goes to position ii. Process 2 is free oscillation dominated by the spring force and piston mass while position iii ( $\dot{x}_p > 0$ ) goes to position iv, which goes to position iii ( $\dot{x}_p < 0$ ), which goes to position v. By repeating this cycle, the IDV can perform the mechanical work based on gas enthalpy and generate an intermittent flow.

### B. Inflow-Driven Valve Mechanical Model

Figure 4 shows a dynamic model of an IDV. The piston mass is  $m_p$ , the natural length of the spring is  $x_n$ , and the spring constant is  $k$ . We considered the three forces acting on the piston: the spring force,  $-k(x_p - x_n)$ , the frictional force in proportion to the piston velocity  $\mu \dot{x}_p$  ( $\mu$  is the coefficient of friction), and the force generated from the difference between the inside and outside pressure of the valve,  $A_p(p_{in} - p_{out})$ , where  $A_p$  is the piston cross-sectional area,  $p_{in}$  is the inside pressure of the cylinder, and  $p_{out}$  is the outside pressure of the

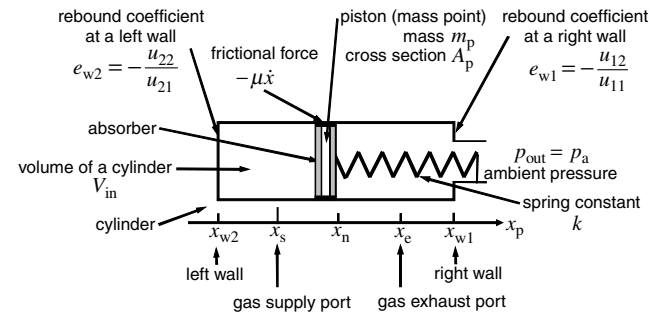


Fig. 4 Dynamic model of an IDV.

cylinder (the outside pressure  $p_{out}$  is generally the atmospheric pressure  $p_a$ ).

The piston begins to move from the initial state  $(x_n, p_s)$ . When it reaches  $x_p = x_e$ , the inside pressure becomes  $p_{in} = p_a$ , and then inertia causes the piston to continue to move in the positive direction of  $x_p$ . In position  $x_p > x_e$ , the piston velocity decreases because the inside pressure is  $p_{in} = p_a$ . If the piston reaches the right wall  $x_{w1}$ , it rebounds. The piston velocity at the right wall  $w1$  before collision is  $u_{11}$ , and after collision, it is  $u_{12}$ . The rebound coefficient is expressed as  $e_{w1} = -u_{12}/u_{11}$ .

While inside pressure is maintained at  $p_{in} = p_a$ , the piston displacement decreases to  $x_p = x_s$ . The inside pressure rises to supply pressure  $p_{in} = p_s$  again at  $x_p = x_s$ . Next, the piston moves in the negative direction of  $x_p$ , while the inside pressure is maintained at  $p_{in} = p_s$ . In some situations, the piston reaches the left wall  $x_{w2}$  and the piston rebounds. In this case, the piston velocity at the left wall  $w2$  before collision is  $u_{21}$ , and after collision, it is  $u_{22}$ . The rebound coefficient is expressed as  $e_{w2} = -u_{22}/u_{21}$ . Finally, the piston displacement increases and the following cycle starts. The motion equation of the piston is given as follows:

$$m_p \ddot{x} = -k(x_p - x_n) - \mu \dot{x}_p + A_p(p_{in} - p_{out}) \quad (1)$$

This mechanical model assumes the ideal operation cycle. The natural frequency is  $\sqrt{(k/m_p)}/2\pi$ , and the resonance frequency of forced oscillation of this spring-damper-valve system is  $\sqrt{(k/m_p)(1 - 2\zeta^2)}/2\pi$ , where  $\zeta(=\mu/2\sqrt{m_p k})$  is the damping coefficient. Under the conditions of the calculation, the resonance frequency was nearly equal to the natural frequency because the

square of the damping coefficient  $\zeta^2$  was negligibly small. Moreover, the piston collides with the sidewalls, and the operation frequency is much higher than the natural frequency. The operation frequency increased in proportion to the supply pressure, and the maximum operation frequency was about twice the natural operation frequency.

In this case, the inside pressure of cylinder  $p_{in}$  in the first process is given as follows:

$$p_{in} = \alpha p_s$$

In the experiment, a finite time was necessary to supply and exhaust. With an increase in the operation frequency, open time for the supply and exhaust ports decreases. However, in this model, the supply and exhaust processes are carried out discontinuously. Decrease of the operation frequency due to these actual processes is not considered in this model. Therefore, we assumed that the calculation frequency  $f_{cal}$  was identical to the experimental frequency  $f_{exp}$  by using a pressure correction coefficient  $\alpha$ . The inside pressure of cylinder  $p_{in}$  in the second process is given as follows:

$$p_{in} = p_{out}$$

We can obtain the piston movement by solving this equation.

### C. Inflow-Driven Valve Mass Flow Rate

The mass flow per one cycle of an IDV was obtained as the mass of the gas put into the cylinder. When the cylinder volume is  $V_c$ , the supply pressure is  $p_s$ , and the temperature of the gas is  $T_{ave}$ , which is constant. The mass flow rate per one cycle is obtained from the following ideal gas equation:

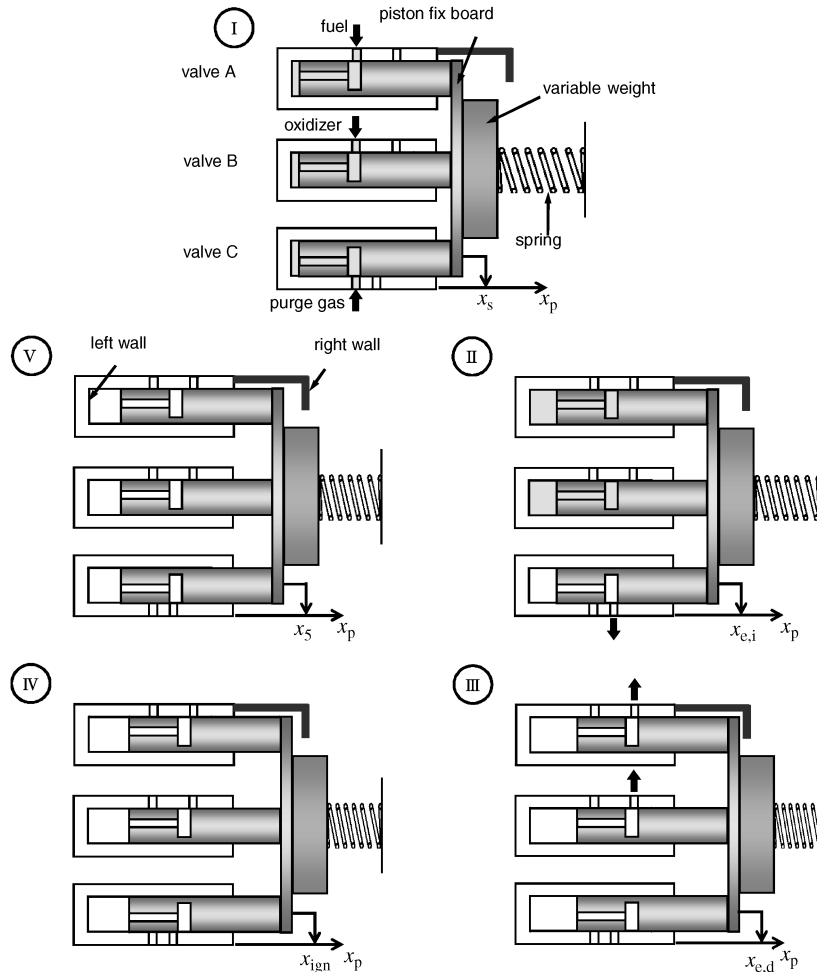


Fig. 5 Operation conceptual diagram of a three-piston IDV.

$$m_{\text{cal,cycle}} = \frac{MV_{\text{in}}(p_s - p_a)}{\mathfrak{R}T_{\text{ave}}} \quad (2)$$

where  $M$  is the molar mass,  $\mathfrak{R}$  is the universal gas constant, and  $T_{\text{ave}}$  is the average gas temperature in the tank between before and after the operation. The mass flow rate is given as follows:

$$\dot{m}_{\text{cal}} = m_{\text{cal,cycle}} f_{\text{cal}} \quad (3)$$

where  $f_{\text{cal}}$  is the operation frequency, which was obtained from the trajectory of the piston displacement, which was obtained by solving the Eq. (1).

#### D. Three-Piston Inflow-Driven Valve System for Pulse Detonation Rocket Engine

A three-piston IDV has a piston for each type of gas; i.e., oxidizer, fuel, and purge gas (see Fig. 5). The propellant (oxidizer and fuel) and purge gas must be exhausted to the PDE tube on different schedules. Figure 6 shows the open–close operation of each port and the on–off operation of the igniter. Since each piston is fixed by a fixed board, the three pistons can be synchronized. As shown in Fig. 5, each cylinder has a supply port and an exhaust port. Although all supply port positions in direction  $x_p$  are the same, the exhaust port positions in direction  $x_p$  for propellant or purge gas are different.

Initially, all gas is supplied to the IDV at the same time in state I of Fig. 5 ( $x_p = x_s$ , state I of Fig. 6). Then, the piston begins to move in the positive direction of  $x_p$  due to the generated pressure difference. The purge gas is first exhausted from the IDV and injected into the combustor in state II of Fig. 5 ( $x_p = x_{e,i}$ , state II of Fig. 6). Since the oxidizer and the fuel are inside the cylinder, the piston moves in the positive direction of  $x_p$ . In state III ( $x_p = x_{e,d}$ , state II of Fig. 6), the oxygen and the fuel are exhausted from the IDV and injected into the combustor. When all gases are exhausted completely, the piston moves in the negative direction of  $x_p$  because of spring force after the piston rebounds from the wall. While the piston moves in the negative direction of  $x_p$ , the ignition is carried out in state IV, where all exhaust ports are closed ( $x_p = x_{\text{ign}}$ , state IV of Fig. 6). Neither the detonation wave nor the burned gas can propagate into the cylinder from the tube attached to the PDE tube. Next, the piston position reaches state V ( $x_p = x_5$ , state V of Fig. 6), where the exhaust port for the purge gas is open. If the purge gas was not exhausted completely before this, exhaustion is completed at this stage. Finally, the piston returns to the initial position I and each type of gas is supplied. Compared with a two-piston IDV [44,45], the three-piston IDV can operate stably over a wider range of supply pressure, because purge gas is used in the driver gas.

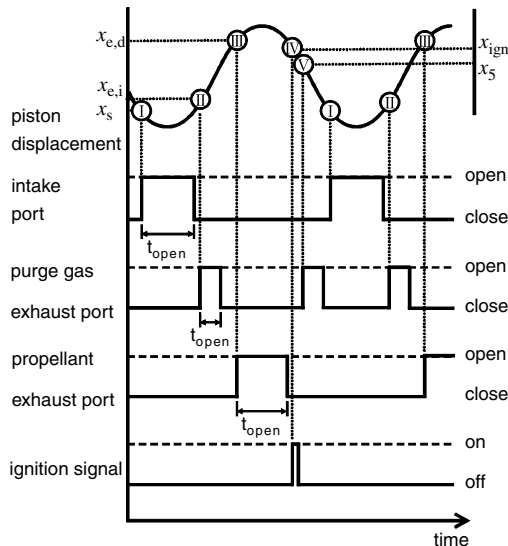


Fig. 6 Piston displacement and operation sequence.

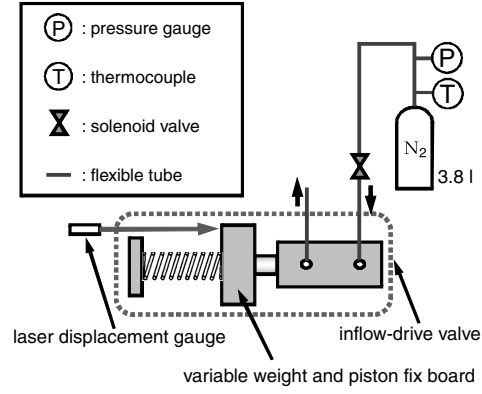


Fig. 7 Schematic diagram of mass flow rate measurement.

### III. Inflow-Driven Valve Mass Flow Rate and Pulse Detonation Rocket Engine Thrust Measurements

#### A. Inflow-Driven Valve Mass Flow Rate Measurement

Figure 7 shows a schematic diagram of mass flow rate measurement, and Fig. 8 shows a photograph of a single-piston IDV used for mass flow rate measurement. The piston was fixed by a metallic plate. The metallic plate moved on the rail guide, and the piston mass could be varied by putting a mass on the metallic plate. The piston had four O rings [Misumi, nitrile rubber (NPB)] to prevent gas leaks. Table 2 shows the experimental conditions. As shown in Fig. 7, the tank (Swagelok, 304-HDF4-1GAL, 3.8 l) was attached to the supply port of the IDV. A pressure gauge (Keller, PAA-23) and thermocouple (Keyence, TF-C11) were installed in the tank. The tank was filled with nitrogen gas, and the IDV operated while the nitrogen gas was exhausted from the IDV. We measured the mass flow per one cycle  $m_{\text{exp,cycle}}$  passing through the IDV based on the changed state of the tank. The gas was assumed to be perfect gas. The mass flow per one cycle was computed as follows:

$$m_{\text{exp,cycle}} = \frac{MV_b \Delta p}{\mathfrak{R}T_{\text{ave}} N} \quad (4)$$

where  $\Delta p$  is the pressure difference in the tank,  $V_b$  is the inner volume of the tank, and  $N$  is the number of the cycle. We obtained the average mass flow as  $N = 10$ . The mass flow rate  $\dot{m}_{\text{exp}}$  is expressed as follows:

$$\dot{m}_{\text{exp}} = m_{\text{exp,cycle}} f_{\text{exp}} \quad (5)$$

where  $f_{\text{exp}}$  is the operation frequency of the IDV, which was obtained from piston displacement.

#### B. Pulse Detonation Rocket Engine Thrust Measurement

We evaluated the thrust performance of the PDRE system with a three-piston IDV and recorded stable operation of the IDV. Figure 9

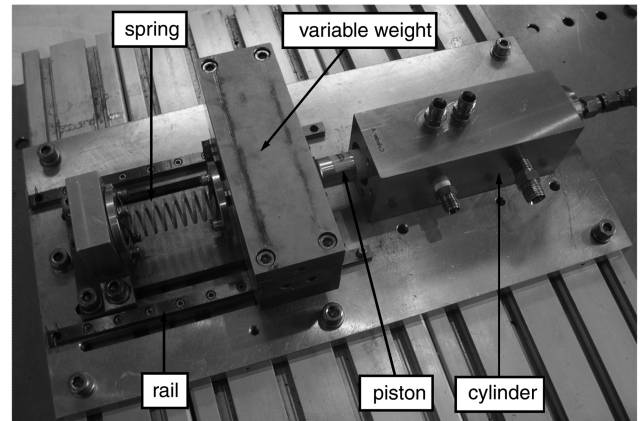


Fig. 8 Photograph of a single-piston IDV.

**Table 2** Condition of mass flow rate measurement

Shot number	Spring constant $k$ , N/m	Piston mass $m_p$ , kg
1	9800	3.88
2	4900	3.88
3	2940	3.88
4	9800	12.80
5	4900	12.80
6	2940	12.80

shows a schematic diagram of the thrust measurement experiment apparatus. The three pistons were fixed by a metallic plate, the metallic plate moved on the rail guide, and the piston mass could be varied by putting a mass on the metallic plate in common with the single-piston IDV. Each piston had four O rings (Misumi, NPB) to prevent gas leaks. A proximity sensor was used for ignition (Keyence, EV-118M). The proximity sensor sent an ignition signal, as seen in state IV of Fig. 5. Table 3 shows the experimental conditions. We used an ethylene–oxygen mixture as the propellant, and we used helium gas as the purge gas. The supply pressure of oxygen was changed from 0.6 to 1.9 MPa.

If the PDE tube was filled with purge gas without diffusion, the thickness of the purge gas per PDE tube was 13% (approximately 150 mm) at a helium supply pressure of 2.6 MPa. This supply pressure was decided empirically so that the mechanism could operate with stability. In shot numbers 1, 4, 7, 8, 11, and 14, the thrust measurement was carried out three times to ensure repeatability of the experiment, and the average values were plotted. As shown in Fig. 9, a pressure gauge (Keller, PAA-23) and a thermocouple (Keyence, TF-C11) were installed between the tank and the pressure regulator to measure the gas mass in the tank. The mass flow rate was obtained from the change in state in the tank. The mass differences  $\Delta m_{\text{exp}}$  between, before, and after the operation were computed as follows:

$$\Delta m_{\text{exp}} = m_{\text{exp},0} - m_{\text{exp},1} = \left( \frac{V_b}{v_{b,0}} - \frac{V_b}{v_{b,1}} \right) \quad (6)$$

where  $V_b$  is the inner volume of the tank, and the inner volume for oxygen, ethylene, and helium was 47.4, 47.1, and 47.5 liters, respectively. The variable  $v_b$  is the specific volume of each gas. The pressure of each gas inside the tank was 2.0 MPa or more. The real gas effect was considered, because the density was high and interaction between the molecules could not be neglected. Based on the state equation of van der Waals [46], the specific volume in Eq. (6) was obtained by solving the following equation:

$$\left( p_b + \frac{a}{v_b^2} \right) (v_b - b) = RT_b \quad (7)$$

where  $p_b$  is the internal pressure of the tank,  $T_b$  is the internal temperature of the tank, and  $a$  and  $b$  were given by the following equations:

$$a = \frac{27R^2T_{\text{cr}}^2}{64p_{\text{cr}}} \quad b = \frac{RT_{\text{cr}}}{8p_{\text{cr}}}$$

where the critical pressure  $p_{\text{cr},o}$  and critical temperature  $T_{\text{cr},o}$  of oxygen are 5.04 MPa and 154.6 K, the critical pressure  $p_{\text{cr},f}$  and critical temperature  $T_{\text{cr},f}$  of ethylene are 5.03 MPa and 282.4 K, and the critical pressure  $p_{\text{cr},i}$  and critical temperature  $T_{\text{cr},i}$  of helium are 0.23 MPa and 5.2 K, respectively. The mass flow rate was evaluated from the mass difference and operating time interval as follows:

$$\dot{m}_{\text{exp}} = \frac{\Delta m_{\text{exp}}}{\Delta t_{\text{ope}}} \quad (8)$$

where  $\Delta t_{\text{ope}}$  is the operating time interval determined by solenoid valves (CKD, AB41 series). The solenoid valve was controlled by a programmable logic controller (Keyence, KV-700). The mass flow was small in the condition of low supply pressure (shots 1, 2, 8, and 9), so the operating time interval was extended to improve measurement accuracy.

We measured the thrust by using a PDRE and a load cell. In our PDRE, a PDE tube was fixed to a rail guide. A spring ( $k = 9800$  N/m) was attached between the PDE tube and the load cell to smooth the impulse of the detonation wave.

The total impulse  $I_{\text{exp}}$  was given as follows:

$$I_{\text{exp}} = \int_{t_{\text{start}}}^{t_{\text{stop}}} F_{\text{lc}} dt \quad (9)$$

where  $t_{\text{start}}$  is the start time of the operation, and  $t_{\text{stop}}$  is a point at the intersection of the baseline (0 N) with the last negative gradient, because the load cell output did not become 0 N after the operation and the load cell output was different in each experiment. The time-averaged thrust  $F_{\text{exp}}$  was expressed as follows:

$$F_{\text{exp}} = \frac{I_{\text{exp}}}{t_{\text{stop}} - t_{\text{start}}} \quad (10)$$

The experimental specific impulse  $I_{\text{sp,exp}}$  was obtained using the following equation:

$$I_{\text{sp,exp}} = \frac{F_{\text{exp}}}{g(\dot{m}_{\text{exp},o} + \dot{m}_{\text{exp},f})} \quad (11)$$

From the measured mass flow rate  $\dot{m}_{\text{exp}}$ , the operation frequency  $f_{\text{exp}}$ , the volume of PDE tube  $V_t$ , and the gaseous species, we calculated the specific impulse  $I_{\text{sp,cal}}$  and the calculated thrust  $F_{\text{cal}}$ . The specific impulse can be a function of the number of cycles  $N$ , the equivalent ratio  $\phi$ , and the propellant fill fraction in combustor  $\psi$  as follows:

**Table 3** Condition of thrust measurement

Shot number	Times	Spring constant $k$ , N/m	Piston mass $m_p$ , kg	Oxygen supply pressure $p_o$ , MPa, gauge	Ethylene supply pressure $p_f$ , MPa, gauge	Helium supply pressure $p_i$ , MPa, gauge	Operating duration $\Delta t_{\text{ope}}$ , ms
1	3	6650	6.81	0.6	0.3	2.6	5000
2	1	6650	6.81	0.8	0.4	2.6	5000
3	1	6650	6.81	1.0	0.5	2.6	4000
4	3	6650	6.81	1.2	0.6	2.6	3000
5	1	6650	6.81	1.4	0.7	2.6	3000
6	1	6650	6.81	1.6	0.8	2.6	3000
7	3	6650	6.81	1.9	0.95	2.6	3000
8	3	9010	7.02	0.6	0.3	2.6	5000
9	1	9010	7.02	0.8	0.4	2.6	5000
10	1	9010	7.02	1.0	0.5	2.6	3000
11	3	9010	7.02	1.2	0.6	2.6	3000
12	1	9010	7.02	1.4	0.7	2.6	3000
13	1	9010	7.02	1.6	0.8	2.6	3000
14	3	9010	7.02	1.9	0.95	2.6	3000



pressure, because the IDV used inflow enthalpy. With increasing supply pressure, the force that dominated the piston motion was changed from inertia force of the piston and restorative force of the spring to surface force by supply pressure, and the operation frequency increased. In this experiment, the mass flow rate was also increased in proportion to the operation frequency. The biggest benefit of an IDV is that operation autonomously reaches a different limit cycle when the supply mass flow is changed (the size of the orifice is changed). In fact, the mass of the outflow and the operation frequency autonomously increase while maintaining the sinusoidal operation and the valve mass when the mass flow of the inflow (inflow pressure) increases. Even if the mass flow of the inflow is small, an IDV can maintain the mass flow of the outflow, because the operation frequency can be increased by decreasing the piston mass. In our experiment, as shown in Fig. 11, the mass flow rate of 5.4 g/s (operation frequency of 16.7 Hz) was achieved at a supply pressure ratio ( $p_s/p_a$ ) of 8.7, a piston mass of 3.88 kg, and a spring constant of 9800 N/m. By using the present mechanical model of the IDV, we achieved a mass flow rate of 5.6 g/s (operation frequency of 37.8 Hz) at a supply pressure ratio of 4.3, a piston mass of 0.3 kg, and a spring constant of 9800 N/m.

The maximum mass flow rate of 6.1 g/s (maximum operation frequency of 17.3 Hz) was achieved under the conditions of  $k = 9800$  N/m,  $m_p = 3.88$  kg, and  $p_s = 1.0$  MPa. The initial conditions for the model calculation are shown in Table 4. A comparison of the calculated frequency and the experimental frequency shows that the experimental value corresponded with the

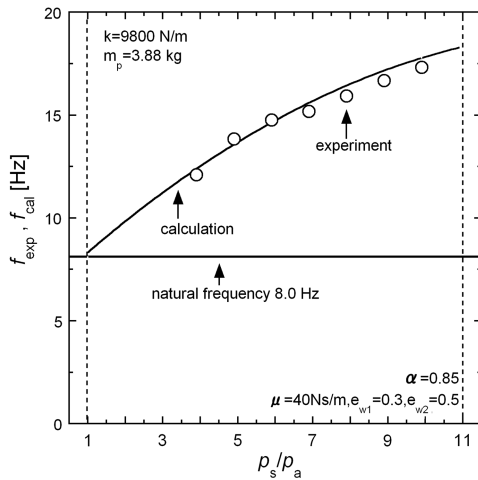


Fig. 10 Operation frequency versus supply pressure ( $k = 9800$  N/m, and  $m_p = 3.88$  kg).

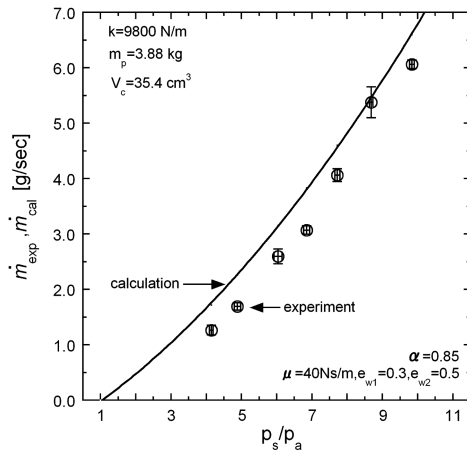


Fig. 11 Mass flow rate versus supply pressure ( $k = 9800$  N/m and  $m_p = 3.88$  kg).

Table 4 Initial condition of the mechanical model

Parameter	Variable	Value
Initial position	$x_n$	0.060 m
Position at left wall	$x_{w2}$	0.060 m
Supply port position	$x_s$	0.076 m
Exhaust port position	$x_e$	0.076 m
Position at right wall	$x_{w1}$	0.090 m
Spring natural length	$x_n$	0.060 m
Friction coefficient	$\mu$	40 Ns/m
Piston cross section	$A$	0.00071 m <sup>2</sup>
Rebound coefficient at right wall	$e_{w1}$	0.3
Rebound coefficient at left wall	$e_{w2}$	0.5
Pressure correction coefficient	$\alpha$	0.6 ~ 1.0

calculated value within 2 Hz. However, the rebound coefficient and the frictional coefficient were set to  $e_{w1} = 0.3$ ,  $e_{w2} = 0.5$ , and  $\mu = 40$  Ns/m, respectively, to conform to the experimental mass flow rate, and the pressure correction coefficient  $\alpha$  was between 0.6 and 1.0. In this model calculation, the internal pressure of the cylinder rises discontinuously, and the internal pressure is maintained as the supply pressure  $p_s$  at the same time as when the supply port opens. On the other hand, opening and closing of the valve ports was carried out for a finite length of time in the experiment. In the case of increasing the spring constant, the opening and closing time of the valve ports was shorter in proportion to the increase in operation frequency, and the experimental mass flow decreased in comparison with the model mass flow. For this reason, as shown in Figs. 12 and 13, if the spring constant increased (from  $k = 2940$  N/m to  $k = 9800$  N/m), the pressure correction coefficient  $\alpha$  was used to conform to the flow rate ( $\dot{m}_{exp}/\dot{m}_{cal}$ ). In addition, the flow rate increased in proportion to the increasing supply pressure, and the flow rate was higher than one in the case of a piston mass of 12.8 kg (see Fig. 13). This was due to changing the opening and closing times of the valve ports and altering the mass flow by changing the supply pressure. For simplification, we did not consider the dependence property of the supply pressure in this model.

If the drive gas was assumed to be a perfect gas, the IDV used 10% (30 kJ/kg) of the kinetic energy of a gas molecule to maintain the piston motion. This energy was sufficiently small compared with the chemical energy of a combustible gas (as an example, the chemical energy of a stoichiometric ethylene–oxygen mixture is approximately 50 MJ/kg). Velocity and temperature gradients were generated in the IDV system, and the energy dissipation was generated. The energy used to drive a piston and the dissipation are sufficiently small compared with the energy needed for thrust. In an airbreathing engine system, if the air passes through the IDV, the total pressure drop by the IDV is serious. However, in a rocket engine system, the total pressure drop in the fuel-oxidizer supply system does not much affect the thrust performance.

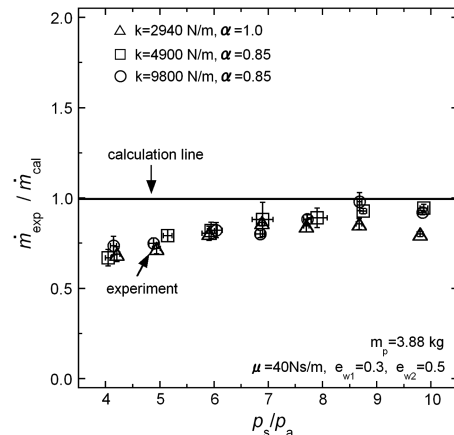


Fig. 12 Mass flow ratio versus supply pressure ( $m_p = 3.88$  kg).



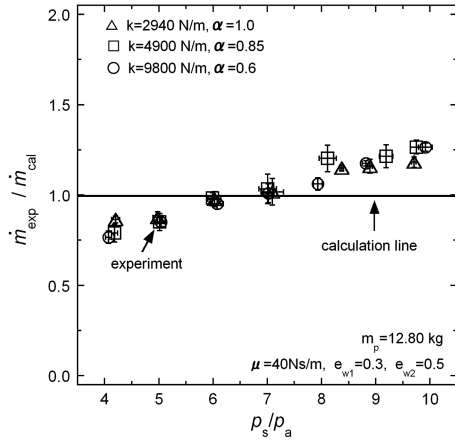


Fig. 13 Mass flow ratio versus supply pressure ( $m_p = 12.80$  kg).

### B. Pulse Detonation Rocket Engine Thrust Measurement

Figure 14 shows the time history of piston displacement  $x_p$ , spark noise  $S$ , and load cell output  $F_{lc}$  in shot 13. Figure 15 shows one cycle of displacement  $x_p$ , the spark noise  $S$ , and the exhaust pressure of oxygen and helium in shot 13. We used piston displacement to determine the valve opening time and ignition time. Using this method, we confirmed that the PDRE system with the IDV performed stably, and a time-averaged thrust was generated.

Figure 16 shows the mass flow rate of the oxygen and ethylene with changing supply pressure under the experimental condition of  $k = 9010$  N/m and  $m_p = 7.02$  kg. The error bars of the vertical axis are the square error in three experiments. The error bars of the horizontal axis are the square error of the accidental error and the instrumental error of the pressure regulator for oxygen. The lines in Fig. 16 show the mass flow rates  $\beta\dot{m}_{cal}$ , which were obtained by using the calculated mass flow rate  $\dot{m}_{cal}$  of Eq. (3) and the correction coefficient  $\beta$ ; however, the calculated operation frequency in this equation  $f_{cal}$  was replaced with the experimental operation frequency  $f_{exp}$ . These mass flow rates were identical to the experiment mass flow rates when the correction coefficients  $\beta$  were 0.25 and 0.35 for ethylene and oxygen. In the thrust measurement experiment, the pressure regulators were used to supply a constant pressure. In contrast, the pressure regulators were not used in the mass flow measurement experiment. Moreover, the operation frequencies under all of the conditions of the thrust measurement experiment were comparable to the operation frequency of the mass flow measurement experiment. Therefore, this decrease of mass flow rate occurred due to restriction by the pressure regulators.

Figure 17 shows the time-averaged thrust  $F_{exp}$  and the calculated thrust  $F_{cal}$  under the experimental condition of  $k = 9010$  N/m and  $m_p = 7.02$  kg, and Fig. 18 shows the experimental specific impulse  $I_{sp,exp}$  and the calculated specific impulse  $I_{sp,cal}$ . The symbols  $\bullet$  and  $\blacktriangle$  show the experimental values, and  $\circ$  and  $\triangle$  show the calculated

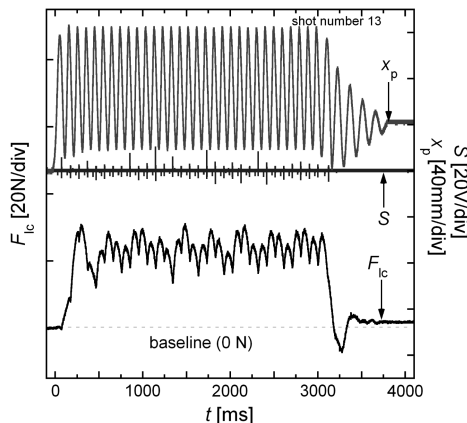


Fig. 14 Time history of thrust measurement experiment (shot 13).

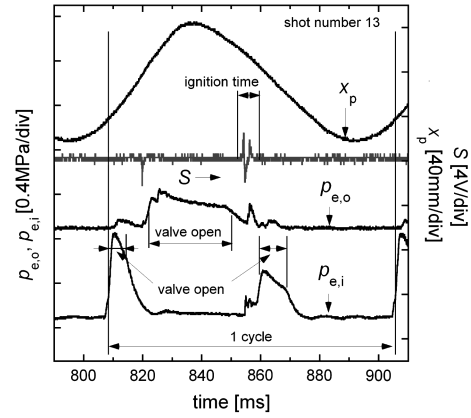


Fig. 15 One cycle of the thrust measurement experiment (shot 13).

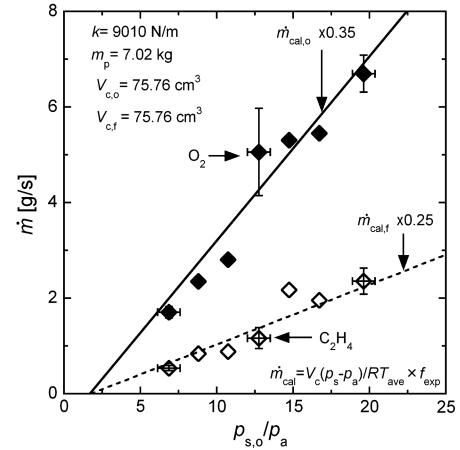


Fig. 16 Mass flow rate versus oxygen supply pressure ( $k = 9010$  N/m and  $m_p = 7.02$  kg).

values. The lines in Figs. 17 and 18 show the fitted lines obtained by using the least-squares method, and  $\phi$  shows the equivalent ratios calculated from the experimental mass flow rate. The error bar of the vertical axis is the square error in three experiments. The error bar of the horizontal axis is the square error of the accidental error and the instrumental error of the pressure regulator for oxygen.

Figure 19 shows the thrust ratio ( $F_{exp}/F_{cal} = I_{sp,exp}/I_{sp,cal}$ ) under all experimental conditions. The variable  $\psi_{plug}$  in Fig. 19 was the propellant fill fraction in the case when propellant filled the tube to the position where the spark plug was installed (61 mm from the

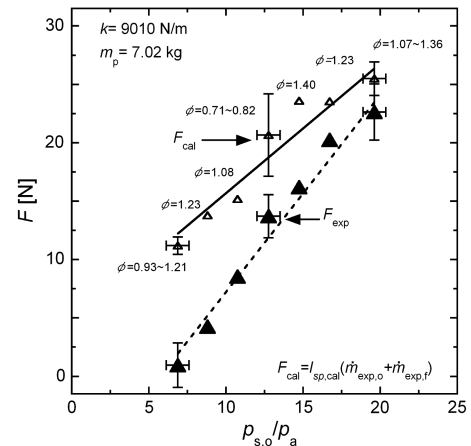


Fig. 17 Thrust versus oxygen supply pressure ( $k = 9010$  N/m and  $m_p = 7.02$  kg).

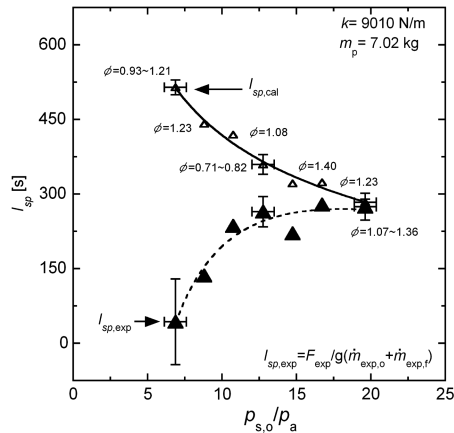


Fig. 18 Specific impulse versus oxygen supply pressure ( $k = 9010$  N/m and  $m_p = 7.02$  kg).

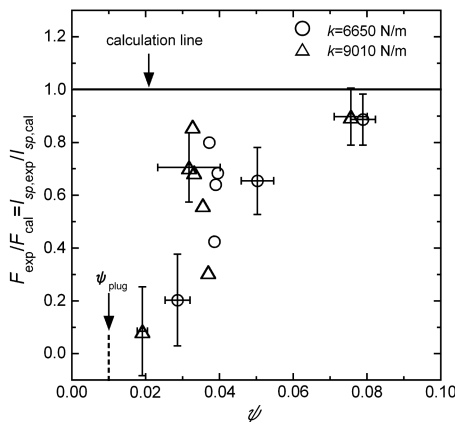


Fig. 19 Thrust ratio versus propellant fill fraction.

closed end of the PDE tube). As shown in Fig. 19, when the oxygen supply pressure was at maximum, the propellant fill fraction was approximately 0.08 and the thrust ratio was 0.8–0.9. In this case, the experimental value was lower than the calculated value. This was probably due to the lack of mixing of the propellant. In a multicycle experiment, the distribution of the equivalent ratio in a PDE tube is not uniform. Hence, the deflagration to detonation transition (DDT) length was longer, and the plateau pressure was lower than the ideal plateau pressure. In comparison with the experiment, a detonation wave was generated from the closed end in the calculation, and the PDRE cycle was operated in this region. When the propellant fill fraction fell to 0.05, the thrust ratio was maintained at near 0.7–0.8. In this region, not only the lack of mixing of the propellant but also the lack of filling with the propellant must be considered. Furthermore, at a lower propellant fill fraction, the thrust ratio decreased rapidly. In the case of this PDRE, a detonation was not generated in a less than propellant fill fraction of 0.04 due to the insufficient distance for DDT to occur.

In addition, the maximum time-averaged thrust of 22.6 N was achieved under the condition of a fuel (ethylene) supply gauge pressure of 0.95 MPa, an oxygen supply gauge pressure of 1.9 MPa, and a spring constant of 9010 N/m. The maximum specific impulse of 279 s was achieved under the condition of an ethylene supply pressure of 0.8 MPa, an oxygen supply pressure of 1.6 MPa, and a spring constant of 9010 N/m.

## V. Conclusions

An IDV for a PDE system was proposed, and the operation principal and the mechanical model for this system were presented.

The IDV can generate an intermittent flow by a piston driven by gas enthalpy.

A flow measurement experiment with a single-piston IDV was carried out, and a maximum mass flow rate of 6.3 g/s (piston mass of 3.8 kg, spring constant of 9800 N/m, supply pressure of 1.0 MPa, and operation frequency of 17.3 Hz) was confirmed. The experimental mass flow rate was 70–120% in comparison with the calculated mass flow rate. It was shown that the operation frequency and mass flow rate could be predicted. Moreover, a thrust measurement experiment of the PDRE using a three-piston IDV was carried out. Within a wide range of supply pressures, the stable operation of this valve was confirmed, and its time-averaged thrust was measured. A maximum time-averaged thrust of 22.6 N [piston mass of 7.02 kg, spring constant of 9010 N/m, fuel (ethylene) supply gauge pressure of 0.95 MPa, oxygen supply gauge pressure of 1.9 MPa, and operation frequency of 10.2 Hz] and a maximum specific impulse of 279 s [piston mass of 7.02 kg, spring constant of 9010 N/m, fuel (ethylene) supply gauge pressure of 0.8 MPa, oxygen supply gauge pressure of 1.6 MPa, and operation frequency of 10.2 Hz] were confirmed.

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