THEORETICAL AND EXPERIMENTAL ANALYSIS OF WEARABLE CONTROL VALVE WITH SELF-HOLDING FUNCTION USING PERMANENT MAGNETS

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ABSTRACT

In wearable pneumatic driving system, the small-sized and lightweight valve is important. In addition, the bulky battery to drive valves will be required. The purpose of our study is to develop a valve with lower electric energy consumption that can be used for a long time with a small battery. In our previous study, the valve with self-holding function that consumes electric power only when the valve changes to open or close was proposed and tested. In this paper, the analytical model of the valve with self-holding function using permanent magnets is proposed and improved to estimate the performance of the valve theoretically. The behavior of magnets was observed by using high-speed camera to know the detail behavior of both magnets and to identify system parameter. As a result, we can predict the whole response of the valve using the proposed model.

KEY WORDS

Analysis, Valve with Self-holding Function, Low Energy Consumption, High-speed Camera

NOMENCLATURES

\( C_b, C_c \): Viscous damping coefficient of magnetic ball and cylindrical magnet [Ns/m]
\( D_b \): Outer diameter of the magnetic ball [m]
\( d_s, d_o \): Inner diameter of tube and orifice [m]
\( e_A, e_B \): Input voltage for solenoid A and B [V]
\( F_b, F_c, F_Q \): Force acted on magnetic ball generated by cylindrical magnet, solenoid and momentum of fluid flow [N]
\( F_{A_b}, F_{B_b} \): Force acted to cylindrical magnet by solenoid A and B [N]
\( i_A, i_B \): Current of solenoid A and B [A]
\( L_A, L_B \): Inductance of solenoid A and B [H]
\( l_s, l_c \): Stroke and length of cylindrical magnet [m]
\( m_b, m_c \): Mass of ball and cylindrical magnet [kg]

\( \rho \): Density of air [kg/m³]
\( \kappa \): Heat capacity ratio

\( \tau_{b}, \tau_{Fp}, \tau_{FQ} \): Torque generated by cylindrical magnet, differential pressure and momentum of fluid flow [Nm]

\( P_o, P_s \): Output and supply pressure [Pa]
\( Q_{a}, Q_{o} \): Exhaust and output mass flow rate [kg/s]
\( Q_{abs} \): Equivalent flow rate affected on the equivalent sectional area of the magnetic ball [kg/s]
\( R \): Gas constant [J/(kg K)]
\( R_A, R_B \): Resistance of solenoid A and B [Ω]
\( S_a, S_o \): Sectional area of valve and orifice [m²]
\( T \): Abstruse temperature [K]
\( t_b \): Thickness of rubber sheet [m]
\( \theta \): Angle from the initial angle \( \theta_0 \) [rad]
INTRODUCTION

Recently, wearable driving systems using pneumatic actuators for power assisted nursing care and rehabilitation have received much attention and active research[1-3]. In such a control system, it is necessary to mount not only pneumatic actuators[4] but also control valves on the body. However, in most of these research works[1-3], the control valves set on other place. To develop a wearable control valve that can drive pneumatic actuators, the size and weight of the valve become serious concerns[5-7]. In addition, the valve requires the electric power to be driven. The bulky battery to drive valves kills advantage of the wearable valve. The purpose of our study is to develop a valve with lower electric energy consumption that can be used for a long time with a small battery. In our previous study, the valve with self-holding function that consumes electric power only when the valve changes to open or close was proposed and tested[8,9]. In the next step, it is necessary to improve the performance of the proposed valve. In this paper, the analytical model of the valve with self-holding function using permanent magnets proposed in our previous study[10] is improved to estimate the performance of the valve theoretically. To identify the system parameters and know the detail behavior of both magnets in the valve, the behavior of magnets is observed by using high speed camera. We will discuss the performance of the valve by changing the design parameters theoretically and experimentally.

WEARABLE CONTROL VALVE WITH SELF-HOLDING FUNCTION

Construction

Figures 1(a) and (b) show the schematic diagram and a photograph of the wearable valve with self-holding function, respectively. The valve consists of a flexible tube whose inner diameter is 2.5 mm and outer diameter is 4 mm, a brass valve seat (orifice) with an inner diameter of 0.5 mm, a permanent magnetic ball with an outer diameter of 2 mm, a cylindrical permanent magnet with an outer diameter of 5 mm and a length of 5 mm, a shin plastic cylinder (straw) with an inner diameter of 5 mm and two handmade solenoids. The valve seat and the magnetic ball are inserted into the flexible tube in order to make a check valve. Each handmade solenoid consists of a J-shaped iron center core with an outer diameter of 3 mm and 250 turns of coil made of enameled wire with an outer diameter of 0.35 mm and the internal resistance of 1.2 ohms. The end of each iron core has a shin rubber shock absorber with a thickness of about 1mm. The cylindrical magnet is inserted into the thin plastic cylinder so that the magnet can slide along the cylinder with a length of 14 mm and a stroke of 8 mm.

Operating principle

Figure 2 shows the operating principle of the tested valve. The operating principle is as follows: In the case of closing the valve as shown in Fig.2 (1), the iron core of solenoind B and the cylindrical permanent magnet are naturally attracted to each other. Then the magnetic ball makes the valve seat (orifice) closed by the generated force of momentum of the flow, as a check valve. In this condition, the valve holds the state of closing with no electrical input by the permanent magnetic force. When the electric input is applied to the solenoind B for about 0.01 seconds as shown in Fig.2 (2), the magnetic repulsive force is generated between the iron core and the cylindrical magnet. Then, the cylindrical magnet moves toward the iron core of solenoind A, and they are attracted to each other. After the cylindrical magnet moves by the repulsive force as shown in Fig.2 (4), the magnetic ball automatically moves toward the center of the valve seat.
Figure 2 Operating principle of the valve

(1) Holding to close without input
(2) Magnetic attraction
(3) Magnetic attraction
(4) Holding to open without input

Figure 3 Transient response of input current and output pressure of the valve

Figure 4 Analytical model of the tested valve

(1) Magnetic repulsion
(2) Magnetic attraction

Figure 5 Model of the magnetic ball in the pipe

(1) Characteristic of the valve

(2) Magnetic repulsion

(3) Magnetic attraction

(4) Holding to open without input

Holding to close without input

Characteristics of the valve

Figure 3 shows the transient response of the input current of solenoids A and B, and output pressure of the valve, respectively. In the experiment, the valve is connected to a tube with exhaust port with a constant sectional area. The input current was calculated using the internal resistance of 1.2 ohms and the impulse input voltage of 5 V. The electrical input voltage was applied for 0.008 seconds to the solenoid only when the valve began to open or close. Then, as a result of the experiment for static characteristics of the valve, the maximum output flow rate is 13.4 liters per minute for the supply pressure of 500 kPa. The value is almost the same as the output flow rate of the small-sized commercially available on/off valve. From the transient response of the output pressure in Fig.3, it can be seen that the valve can be held opened or closed after applying momentary electrical input. We also find that the dead time for opening the valve is 0.021 seconds, the dead time for closing is 0.016 seconds. The dead time of the valve is related to the moving speed of the cylindrical magnet. The difference of the dead time between the cases of opening and closing is caused by the time the cylindrical magnet moves and reaches point of attracting or repulsion to the magnetic ball. From the geometric arrangement between the magnetic ball and the cylindrical magnet as shown in Fig.1, it is easy to recognize that the time delay for closing is faster than that for opening. From Fig.3, it can be also seen that the induced current in the non-operating side solenoid occurs when the opposite solenoid is driven.

ANALYTICAL MODEL OF THE VALVE

Analytical model

In order to estimate performance of the valve, the analytical model of the valve is required. Figure 4 shows a model of the tested valve with self-holding function[10].

The model consists of an electric circuit of two solenoids, a motion of the cylindrical magnet and magnetic ball, and the flow in the pipe. The motion of the cylindrical magnet and the current in the electric circuit of the solenoid are given by following equations:

\[ m_s \frac{d^2 x_s}{dt^2} + C_s \frac{dx_s}{dt} = F_{CA} + F_{CB} \]  
\[ L_A \frac{di_A}{dt} + R_s i_A = e_A \]  
\[ L_B \frac{di_B}{dt} + R_s i_B = e_B \]

The subscript shows the location such as A or B side. Figure 5 shows the model of the magnetic ball in the pipe. In the model, the magnetic ball rotates at the fulcrum C as shown in Fig.5. The force \( F_{CA} \) and \( F_{CB} \) acted on the cylindrical magnet while solenoid A and B are given by following empirical formula based on experimental result when the input voltage of 5 V is applied.
The force $F_b$ acted on the magnetic ball is also given by following empirical formula.

$$F_b = 0 \ (l_e - x_c > 0.001), \quad F_b = 0.1 \ (l_e - x_c \leq 0.001) \quad (5)$$

In the initial state of the ball (dashed line as shown in Fig.5), the ball has plugged up the orifice. The initial angle $\theta_0$ is given by the following equation from geometric relations.

$$\theta_0 = \sin^{-1}(d_c/D_d) \quad (6)$$

The torque $\tau_{FB}$ (a clockwise torque is positive) acted on the magnetic ball by the cylindrical magnetic force $F_b$ is given by the following equation.

$$\tau_{FB} = \frac{D_b}{2}F_b \cos(\theta_0 - \theta) \quad (7)$$

In the model, $F_b$ is given by the empirical relation (5). The torque $\tau_{FB}$ generated by the force caused by the differential pressure is assumed to be by

$$\tau_{FB} = \frac{\pi}{4}d_o^2(P_e - P_o)\frac{d_c}{2} \quad (8)$$

The force $F_Q$ generated by the momentum of the fluid flow $Q_{ob}$ is given by

$$F_Q = 4\frac{Q_{ob}^2}{\rho d_o^{1.5}} \quad (9)$$

The mass flow rate $Q_{ob}$ is equivalent flow rate affected on the equivalent sectional area of the magnetic ball, and it is given by the following equation.

$$Q_{os} = \frac{D_b^2}{d_o^{1.5}}Q_o \quad (10)$$

From the relations mentioned above, the torque $\tau_{FQ}$ generated by the momentum of the fluid flow is given as follows.

$$\tau_{FQ} = 4\frac{D_b^2}{\rho d_o^{1.5}}Q_o \frac{d_c}{2} \quad (11)$$

The maximum angle $\theta_{max}$ is the angle in the case that the magnetic ball touches tube wall as shown in Fig.3, from geometric relationship, is given by the next equation.

$$\theta_{max} = \theta_0 + \sin^{-1}\left(\frac{d_c - D_b - d_c}{D_c}\right) \quad (12)$$

From these equations mentioned above, the total torque affected to the magnetic ball $\tau = \tau_{FB} + \tau_{FQ}$ is given by the following equation.

$$\tau = \frac{D_b}{2}F_b \cos(\theta_0 - \theta) - \frac{\pi}{4}d_o^2\frac{d_c}{2}(P_e - P_o) - \frac{4}{\rho \pi d_o^{1.5}}Q_o^2 d_c \frac{d_o}{2} \quad (13)$$

The equation of rotational motion of the ball is given by

$$\frac{7}{5}m_b \left(\frac{D_b}{2}\right)^2 \frac{d^2\theta}{dt^2} + C_b \frac{d\theta}{dt} = \tau \quad (14)$$

The opening area $S_o$ of the valve changes according to the angle $\theta$ of the ball. To express this sectional area simply, the sectional area is approximately given by the difference between the sectional area of the inner bore of the tube and the ball, and is expressed as the following equation.

$$S_o = \frac{\pi}{4}(d_c^2 - D_b^2 \sin^2(\theta_0 - \theta)) \quad (15)$$

The mass flow rate $Q_o$ which flows through the orifice while the valve opens is given by

$$Q_o = S_o \sqrt{\frac{2}{RT} \left(\frac{P_e}{P_o}\right)} \quad (16)$$

Function $f(z)$ that expresses the state of flow state is given as follows.
When a constant orifice for exhaust is connected to the valve with a constant volume $V_o$, as shown in Fig.6, the pressure change according to difference between the supply mass flow rate $Q_o$ and the exhaust flow rate $Q_a$ is given as follows.

$$dP/dt = \frac{kRT}{V_o}(Q_o - Q_a)$$  \hspace{1cm} (18)$$

The exhaust mass flow rate $Q_a$ is given as follows.

$$Q_a = S_aP_a R T \left( \frac{P_a}{P_o} \right)$$  \hspace{1cm} (19)$$

where $S_a$ is a sectional area of the exhaust orifice, and $P_a$ is the downstream pressure (atmospheric pressure).

By using these equations from (1) to (19), we can calculate the dynamic characteristics of the valve such as the pressure response.

![Figure 6 Model of experimental setup using the valve](image)

**Table 1 Identified system parameters**

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<th>Parameters</th>
<th>Value</th>
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<th>Value</th>
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**Identified system parameters**

Table 1 shows the identified system parameter of the valve. The most parameters were measured. Some parameter such as $C_c$ and $C_b$ were identified so that the calculated results using the model could agree with the experimental results.

**Calculated results**

Figure 7 shows the calculated result using the proposed model and the identified system parameters when input voltage of 5 V was applied to solenoid B for 0.015 seconds at the beginning and also applied to solenoid A for 0.015 seconds after 0.1 seconds. In Fig.7, the circles show the experimental results using the tested valve. The line shows the calculated result. From Fig.7, it can be seen that the model can predict the behavior of the output pressure change such as a dead time when the valve opens. However, there are some differences between the calculated and experimental results when the valve closes. We think that it is caused that the model could not predict the behaviors of the cylindrical magnet and the magnetic ball exactly. Therefore, it is necessary to observe the behavior of both magnets.

**IMPROVED MODEL BASED ON RESULTS USING HIGH-SPEED CAMERA**

**Behavior observation of the magnets**

In order to observe the motion of the cylindrical magnet and magnetic ball, the high speed camera (Photron Co. Ltd., FASTCAM SA-X) was used. Figure 8 shows the experimental setup for observation of magnet’s behavior. The experimental setup consists of the tested valve, the high speed camera, a compressor, regulator to adjust the supplied pressure of the valve, a pressure transducer (Keyence Co. Ltd., AP-32), a personal computer as a recorder and a micro-computer (Renesas Co. Ltd., H8/3664) that can drive two solenoids in the valve. In the experiment, the same driving signal of the valve in the case of the pressure response as shown in Fig.7 was applied. It means that two solenoids were driven alternately for 0.015 seconds in a period of 0.2 seconds with a duty ratio of 50 %. The supply pressure of 500 kPa was applied. The input voltage is 5 V. Figure 9 shows the transient view of motion of the cylindrical magnet and magnetic ball in the valve. From the video, the displacement of the cylindrical magnet and magnetic ball were measured.
Figure 10 and 11 shows the transient responses of the displacement $x_c$ of the cylindrical magnet and the vertical displacement of the magnetic ball in the tube, respectively. These displacements were measured by images of the video. In both figures, the circles show the measured displacement. From Fig. 9 and 10, the behavior of the cylindrical magnet has a dead time of 0.015 seconds from when the solenoid B is activated and a dead time of 0.022 seconds from when the solenoid A is activated. We found that these delays such as dead time were caused by the static friction between the cylindrical magnet and the plastic cylinder. The dead time means the time until the generated force of solenoids reaches at the static frictional force. We also found that the difference of dead time when the solenoid A and B are activated is caused by the force affected from the cylindrical magnet to the magnetic ball. Figure 12 shows the spinning rotational angle when the magnetic ball turns on the spot. From the analysis of the video, we found that the magnetic ball appears the spinning rotation when the cylindrical magnet gets closer to the magnetic ball even if the distance between the cylindrical magnet and the ball is not less than 1 mm. However, in the reverse case when the cylindrical magnet starts to remove from the ball, the ball is affected the cylindrical magnet and stays on the wall of the tube. After removing the magnetic force from the cylindrical magnet, the ball starts to move toward the orifice. As a result, we can assume this delay of the magnetic ball when the valve closes as a dead time of 0.005 seconds. The dead time for the magnetic ball is caused by the ball has N and S poles.

Improved model and calculated results

As consideration of these delays caused by friction of the cylindrical magnet and the inference the cylindrical magnet into the model, the motions of the cylindrical magnet and the magnetic ball when the valves closes are approximated as follows.

$$m_c \frac{d^2 x_c}{dt^2} + C_c \frac{dx_c}{dt} = F_{CA}(t - 0.015) + F_{CB}(t - 0.022)$$  \hspace{1cm} (20)

$$\frac{7}{5} m_b \left( \frac{D_b}{2} \right) \frac{d^2 \theta}{dt^2} + C_b \frac{d\theta}{dt} = r(t - 0.005)$$  \hspace{1cm} (21)

Figures 10 and 11 also show the calculated results of the displacement of the cylindrical magnet and the vertical displacement of the magnetic ball in the tube using the
previous and the improved models, respectively. In both

![Figure 13](image1.png)

**Figure 13** Calculated response of output pressure using improved model

figures, the broken lines show the results using the previous model. The solid lines show the results using the improved model. In the calculation using the improved model, the values of viscos damping coefficient $C_c = 0.8$ kg/s and $C_b = 0.2 \times 10^{-6}$ kg m²/s were identified again. It can be seen that the improved model can predict the behaviors of both magnets exactly. Figure 13 shows the transient response of the output pressure of the valve using same experimental conditions as shown in Fig.7. In Fig.13, the circles show the experimental results and the solid line shows the calculated result using the improved model. We found that the calculated result using the improved model agree well with the experimental result. As a result, we can confirm that the improved model is valid to estimate the performance of the valve.

**CALCULATED AND EXPERIMENTAL ESTIMATION OF VALVE PERFORMANCE**

**Inference of stroke of the cylindrical magnet**

In order to estimate the performance of the valve for changing a design parameter, the stroke of the cylindrical magnet $l_c$ was selected as a design parameter. Figure 14 shows the calculated results of output pressure with various strokes of the cylindrical magnet. In the calculation, the impossible conditions ($l_c < 4$ mm) using the real valve were also included. We can see that the dead time of the valve when the solenoid B is activated can be decreased according to the stroke being shorter. It can be also found that the dead time of the reverse motion in the case when the solenoid A is activated is not changed because of the inference of the cylindrical magnet mentioned above.

Figure 15 shows the experimental results of output pressure by using the real valve with the stroke $l_c$ of 4, 6 and 8 mm. Each color line means the difference of the stroke $l_c$. From Fig. 14, the experimental result also shows same tendency of decreasing dead time of the valve according to using shorter stroke, and the dead time when the valve closes is not also changed. As a result, the dead time of the valve when the valve opens decreases from 25 ms to 20 ms by using shorter stroke $l_c$ of 4 mm.

![Figure 14](image2.png)

**Figure 14** Calculated results of output pressure using various stroke of the cylindrical magnet

![Figure 15](image3.png)

**Figure 15** Experimental results of output pressure using various stroke of the cylindrical magnet

![Figure 16](image4.png)

**Figure 16** Virtual simulation in the case when the frictional force of the cylindrical magnet becomes half

**Inference of virtual change of friction**

As a virtual estimation of the performance of the valve, the simulation that the static friction between the cylindrical magnet and the plastic cylinder (straw) becomes half, that is the dead time of 7.5 ms when the valve opens and the dead time of 11 ms when the valve closes, was executed. Figure 16 shows the result mentioned above. In Fig.16, the broken line and circles show the calculated and the experimental results using the real valve with the stroke $l_c$ of 4 mm. The solid line
shows the calculated result using virtual setting mentioned above. From Fig. 16, if it is possible to decrease the static friction between the cylindrical magnet and plastic cylinder, it seems that the dynamic characteristics of the valve will be drastically improved.

As our future work, we are going to apply a less frictional material such as a teflon tube to the valve to improve the dynamics of the valve. In addition, to prevent the spinning rotation of the magnetic ball because of the magnetic pole for decreasing the time while the ball rotating, an iron ball is going to apply to the valve instead of the magnetic ball.

CONCLUSIONS

The study aiming at theoretical and experimental analysis of the valve with self-holding function using permanent magnets was summarized as follows.

1) The output response and behavior of the inner magnets of the valve with self-holding function were calculated by using the proposed analytical model of the valve which includes the motion of the cylindrical magnet and the magnet ball, the electric circuit of the solenoid and the behavior of fluid flow and identified system parameters.

2) The behavior of the cylindrical magnet and the magnetic ball in the valve was observed by using high speed camera to know the detail behavior of both magnets. The improved analytical model of the valve was also proposed. The system parameters were identified again. As a result, we can confirm that the calculated results using the improved model agree well with the experimental result in the behaviors of both magnets and output pressure of the valve.

3) In order to estimation the performance of the valve, the transient response of the output pressure of the valve with various strokes of cylindrical magnet was investigated theoretically and experimentally. As a result, by using the shorter stroke \( l_c = 4 \text{ mm} \), the dead time of the valve improved from 25 ms to 20 ms when the valve opens.

4) As a virtual estimation of the performance of the valve, the dynamics of the valve in the condition of less friction between the cylindrical magnet and the plastic cylinder was simulated. As a result, we can confirm that the dynamics of the valve will be improved by decreasing the sliding friction of the cylindrical magnet.

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REFERENCES


