An In-Pipe Mobile Micromachine Using Fluid Power*

(A Mechanism Adaptable to Pipe Diameters)

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To realize micro maintenance robots for small diameter pipes of nuclear reactors and so on, high power in-pipe mobile micromachines have been required. The authors have proposed the bellows microactuator using fluid power and have tried to apply the actuators to in-pipe mobile micromachines. In the previous papers, some inchworm mobile machine prototypes with 25 mm in diameter are fabricated and the traveling performances are experimentally investigated. In this paper, to miniaturize the in-pipe mobile machine and to make it adaptable to pipe diameters, firstly, a simple rubber-tube actuator constrained with a coil-spring is proposed and the static characteristics are investigated. Secondly, a supporting mechanism which utilizes a toggle mechanism and is adaptable to pipe diameters is proposed and the supporting forces are investigated. Finally, an in-pipe mobile micromachine for pipes with 4 - 5 mm in diameter is fabricated and the maximum traveling velocity of 7 mm/s in both ahead and astern movements is experimentally verified.

Key Words: Micromachine, Fluid Machinery, Moving Robot, Actuator, In-Pipe Mobile Machine, Pipe-Diameter Change, Toggle Mechanism

1. Introduction

Research and development of micromachines have been active in the field of engineering^{(1),(2)}. A micro maintenance system which inspects and repairs the inner wall of small diameter pipes of power plants and a micro system for medical treatment in human bodies are examples of targets⁽²⁾. In such micromachines, a maintenance micro robot for pipes of nuclear reactors with about 10 mm in diameter has not only much requirement to be realized but also feasible size in the near future.

To realize such practical micromachines, some of the authors have researched on mechanisms and con-

trol of in-pipe mobile micromachines using fluid power which features high output energy density⁽³⁾. In the previous papers, an inchworm traveling mechanism shown in Fig. 1 which consists of front and rear "supporting mechanisms" and a "propelling mechanism" driven by bellows microactuators⁽³⁾, a traveling mechanism controlled by one pressure supply tube⁽⁴⁾ and a traveling mechanism traversable branched pipes have been proposed, fabricated with 25 mm in

Propelling mechanism Supporting mechanism

Fig. 1 Traveling sequence of inchworm in-pipe mobile machine

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diameter and experimentally investigated.

In this paper, to realize in-pipe mobile micromachines adaptable to pipe diameters, a simple fluid microactuator using a rubber-tube and a compact supporting mechanism using a toggle mechanism are proposed and fabricated and the characteristics are experimentally investigated. Then, an in-pipe mobile micromachine adaptable to pipe diameters of 4–5 mm is designed and fabricated and the traveling performances in ahead and astern movements are experimentally investigated.

2. Proposition of a Rubber-Tube Actuator Constrained with a Coil-Spring

2.1 Structure of the proposed fluid actuator

Structure of the proposed actuator is shown in Fig. 2. The actuator consists of a rubber-tube covered with a coil-spring. The coil-spring eliminates enlargement of the rubber-tube in radial direction by the inside pressure and gives contraction force. The actuator is fabricated easily and inexpensively in millimeter size comparing to the bellows actuators. Also, the range of designed characteristics are wide with selecting spring constant and initial tension of the coil-spring.

2.2 Fabrication and basic experiments

Seven kinds of microactuators with 3 mm in diameter and 20 mm in length (working length: 10 mm) are fabricated with rubber-tubes (outside diameter: 2 mm, inside diameter: 1 mm) and coil-springs (outside diameter: 3 mm, inside diameter: 2 mm). Material of the tube is a silicon rubber with high endurance. Spring constants and initial tensions of the coil-springs are 0.069–1.7 N/mm and 0–1.6 N, respectively.

Static characteristics of the fabricated microactuators are measured. Flowrate is applied by a pump which has a bellows extended or contracted by a biased center shaft driven by a stepping motor controlled by a personal computer. The waveform is a rectangular wave with amplitude corresponding to the maximum inside pressure of about 1.2 MPa and 0.5 Hz in frequency. The working fluid is tap water with low viscosity of 1 mPa·s. Load force is applied in contraction direction with a coil spring. Inside pressure and output displacement of the microactuators are measured with a semiconductor pressure sensor and a magnetoresistance device,

Fig. 2 Structure of the rubber-tube actuator

respectively. The obtained analog signals are converted to digital signals via an A/D convertor with a sampling frequency of 1 kHz and fetched to a personal computer.

Examples of the experimental results are shown with solid lines in Fig. 3. Figures 3(a) and (b) are for spring constant of 0.56 kN/m and initial tension of 1.6 N. Figures 3(c) and (d) are for spring constant of 0.23 kN/m and initial tension of 0 N. Buckling is occurred with a spring used in Figs. 3(c) and (d) and load force of 1.2 N.

Based on the experiments, the following results are obtained:

(1) Stable amplitude (in Figs. 3(a) and (b), the amount is 2.5 mm) is realized with high spring constant without buckling.

(2) The actuator has a hysteresis (in Figs. 3(a) and (b), the value is about 20%) caused by elastic characteristics of the rubber. With smaller spring constant, the hysteresis becomes large.

(3) With small spring constant and large load force, buckling is occurred.

For initial tension of the coil-springs, no distinct result has been obtained because of the scattered residual stress. For an in-pipe mobile micromachine stated after, a coil-spring used in Figs. 3(a) and (b)

Fig. 3 Examples of static characteristics of the rubber-tube actuators

is used.

2.3 Mathematical model

As a first step to establish an optimal design method, a mathematical model is derived. As shown in Fig. 4, the actuator is modelled as a cylinder and a piston which is applied spring force of the coil-spring, nonlinear elastic force of the rubber and load force reacted against the output force. The elastic characteristics of rubber is modelled as a quadric in pressure increase. The curve in pressure decrease is obtained by shifting the curve in pressure increase by the hysteresis H_p . Each parameter is identified by using 10 data with 5 actuators having different coil-springs. The data are only for the load force of 0.39 N. The estimated static characteristics for different load force are compared to the experimental results and evaluated. The quadric curve of the elastic force is obtained as an approximation of 10 curves which are obtained by subtracting spring and load forces from quadrics passing through the minimum, maximum and mid points. The hysteresis H_p is assumed to be linear with tensional stress change applied to the rubber.

As a result, the following equations are derived :

$$x = \begin{cases} \frac{2A_1K_c(P - A_1F - A_2) - A_2\sqrt{4A_1A_3K_c(P - A_1F - A_2) + A_2^2A_3^2 + 4A_1^2K_c^2} + A_2^2A_3}{2A_1^2K_c^2}, & (P \geq A_1F + 2A_2) \\ 0 & (P \leq A_1F + 2A_2) \end{cases} \quad (1)$$

$$H_p = A_4\Delta\sigma_r + A_5, \quad (2)$$

where, x : output displacement of the actuator, P : inside pressure of the actuator in pressure increase, F : output force of the actuator, K_c : spring constant of the coil-spring, $\Delta\sigma_r$: tensional stress change of the rubber, $A_1=0.20 \text{ mm}^{-2}$, $A_2=83 \text{ kPa}$, $A_3=48 \text{ mm}^{-1}$, $A_4=0.12$, $A_5=7.6 \text{ kPa}$.

Dotted lines in Fig. 3 show the results of the derived mathematical model. It is verified that the obtained results have good agreements in Figs. 3(a) and (b). However, in Figs. 3(c) and (d), as the spring constant is small and the effect of nonlinearity of elastic characteristics of rubber is large, the error is large. As a result, a reliable mathematical model for large spring constant is derived.

3. Proposition of Toggle-Type Supporting Mechanism Adaptable to Pipe Diameters

3.1 Principle of the toggle-type supporting mechanism

The previous supporting mechanisms are C-shaped rings whose diameters are changed by the bellows microactuators⁽³⁾⁻⁽⁵⁾. To adapt to large pipe diameter change, magnification of the bellows displacement is indispensable and it is difficult to be realized in compact size.

In this paper, a new supporting mechanism shown in Fig. 5(a) is proposed. The proposed supporting mechanism is a toggle mechanism constructed with

Fig. 4 Model of the rubber-tube actuator

link type supporting parts which are called the "supporting legs", hereafter. Torque T_0 is set to be small. When external force f is applied in positive direction, the following equation is required to be in hold mode with smaller friction than the maximum static friction :

$$f \begin{cases} \leq \frac{2T_0}{L \cos \phi (1/\mu - \tan \phi)} & (\tan \phi < 1/\mu) \\ > 0 & (\tan \phi \geq 1/\mu) \end{cases} \quad (3)$$

When $\tan \phi \geq 1/\mu$, as the mechanism is always in hold mode, the supporting force is infinity. On the other

(a) Structure

(b) Analyzed supporting forces

Fig. 5 Toggle-type supporting mechanism

Fig. 6 Proposed shape of supporting leg adaptable to pipe diameters

hand, for $f < 0$, the following equation is derived:

$$|f| \leq \frac{2T_0}{L \cos \phi (1/\mu + \tan \phi)} \quad (4)$$

With small torque T_0 , the supporting force is small.

Thus, the proposed supporting mechanism is a simple and compact mechanism which has small supporting force for $f < 0$ and large supporting force for $f > 0$, as the supporting leg is opened and the friction becomes large. Moreover, as the supporting force is different with the external force directions, an inchworm in-pipe mobile micromachine is able to travel without sequential hold/release control of the supporting mechanisms.

3.2 Improvement to adapt the pipe diameters

Figure 5(b) shows the calculated results of Eqs. (3) and (4). For pipe diameter $D=5$ mm, large supporting force is realized. However, for $D=3$ mm, the supporting force is small, because the opening torque of the supporting leg is small as the angle is small. It is the problem that the supporting force is dependent on the pipe diameters.

To overcome the problem, a new supporting leg shown in Fig. 6 is proposed. In the proposed supporting leg, the angle ϕ of a line which is extended between a touching point of the supporting leg to pipe wall and the rotation center is constant independent of the pipe diameters.

The theoretical curve is derived by using variable p as a parameter as follows:

$$\begin{cases} |x| = \frac{C}{\sqrt{1+p^2}} \exp\left(-\frac{1}{\tan \phi} \tan^{-1} p\right), \\ y = px \end{cases} \quad (5)$$

where, C : constant. The calculated curve with $\tan \phi = 2.5$ is shown by a solid line in Fig. 7. The dotted line in Fig. 7 shows a circle with biased center. It is found that a circle with biased center is able to approximate the proposed curve. As a circle is far easier to fabricate than the proposed curve, in this paper, the proposed curve is approximated by a circle with biased center.

3.3 Design of the proposed supporting leg

Coordinate systems for analysis are defined as

Fig. 7 Approximation of theoretical curve with circle

shown in Fig. 8(a). A coordinate system $o-xy$ is fixed at the supporting leg. A coordinate system $o-x_\theta y_\theta$, which is rotated by angle θ , is fixed at the in-pipe mobile micromachine.

Assume that the radius and the center on the coordinate system $o-xy$ of the circle are R and $(a, 0)$, respectively. Then, it is found through geometrical analysis that the trajectory of touching points on the coordinate system $o-x_\theta y_\theta$ becomes a circle with a center at $(0, R)$ and a radius of a . In this design, to reduce the deviation of angle ϕ , angle ϕ for the maximum pipe diameter $D_{\max}$ is set to be the value for the minimum pipe diameter $D_{\min}$ as shown in Fig. 8(b). Moreover, to eliminate an interference of the supporting leg to pipe wall as shown in Fig. 8(c), a condition that the y coordinate of tip of supporting leg should larger than the value y_0 as follows:

$$y_0 = \beta D_{\min} / 2 \quad (6)$$

is added. As a result, the following equations are obtained:

$$\alpha = \tan \phi_{\min} \leq \frac{\sqrt{(3+\beta)D_{\max}D_{\min} - D_{\max}^2 + \beta D_{\min}^2}}{D_{\max} - D_{\min}}, \quad (7)$$

$$R = \frac{(\alpha^2 + 1)D_{\max}D_{\min}}{\alpha^2(D_{\max} + D_{\min})}, \quad (8)$$

$$a = \frac{\sqrt{\alpha^2 + 1}D_{\max}D_{\min}}{\alpha^2(D_{\max} + D_{\min})}, \quad (9)$$

$$\begin{aligned} \tan(\theta_{\max} - \theta_{\min}) \\ = \frac{\alpha(D_{\max} - D_{\min})\sqrt{4D_{\max}D_{\min} - \alpha^2(D_{\max} - D_{\min})^2}}{2D_{\max}D_{\min} - \alpha^2(D_{\max} - D_{\min})^2} \end{aligned} \quad (10)$$

Equation (10) shows the rotation angle of the supporting leg when the pipe diameter is changed from $D_{\min}$ to $D_{\max}$.

3.4 Fabrication of the proposed supporting leg

Based on the Eqs. (7)-(10), a supporting leg is designed for $D_{\max}=5.25$ mm, $D_{\min}=3.00$ mm and $\beta=0.8$. In this paper, the maximum pipe diameter is set

Fig. 8 Geometrical analysis of shape of supporting leg

to be 5 mm, however, as the in-pipe mobile micromachine is supported at the tip of supporting leg when $D = D_{\max}$, the value of $D_{\max}$ mentioned above is selected.

The rotation angle of the supporting leg calculated by Eqs. (10) and (7) is shown in Fig. 9. When α is large, the supporting force is large as the angle ϕ is large, however, it is found that the rotation angle of the supporting leg is also large. The rotation angle has limitation because the supporting leg is controlled by a shape memory alloy wire as stated after. In this paper, considering the characteristics of the shape memory alloy wire, $\alpha = 2.00$ is selected.

Fig. 9 Relation between parameter α and rotation angle of supporting leg

Fig. 10 Designed supporting leg

Fig. 11 Analyzed supporting forces of the proposed supporting mechanism adaptable to pipe diameters

The designed shape of the supporting leg is shown in Fig. 10. The small hole on the upper right side is to fix the shape memory alloy wire. Figure 11 shows the analyzed supporting force of the designed supporting leg. It is shown that the designed supporting leg

Fig. 12 Calculated ratio R_{sf} of supporting forces in positive and negative directions

Fig. 13 Fabricated in-pipe mobile micromachine

produces large supporting force independent of pipe diameters. The designed supporting leg is fabricated and controlled by a shape memory alloy wire and the supporting force is measured in an acrylic pipe. As a result, for pipe diameter $D=4$ mm, supporting force is 0.71 N for hold mode and 0.073 N for release mode. For $D=5$ mm, the value is 0.49 N for hold mode and 0.060 N for release mode.

Next, ratio R_{sf} of supporting forces in positive and negative directions are calculated for fixed $D_{max}=5.25$ mm and variable D_{min} . For simplification, equality is assumed in Eq.(7). The results are shown in Fig. 12. For $\mu=0.3$ as an example, when $D_{min} \leq 3$ mm, the ratio R_{sf} becomes small. The adaptable range of pipe diameters of the proposed supporting mechanism is determined by the characteristics.

4. Fabrication of an In-Pipe Mobile Micromachine

4.1 Structure of the fabricated in-pipe mobile micromachine

The fabricated in-pipe mobile micromachine is shown in Fig.13, which uses the fabricated microactuator stated in chapter 2 and the fabricated supporting mechanisms stated in chapter 3. In each of front and rear supporting mechanisms, two pairs of

Fig. 14 Traveling distance of the fabricated machine

supporting legs are placed face to face. By changing the pair of legs, ahead and astern movements are controlled. To control the legs, a shape memory alloy wire is used, which produces large force although the response is not high. In the actual design, V-shaped shape memory alloy (Ti-50Ni (at%)) wires with 0.27 mm in diameter is used and the angle is controlled by Joule's heat by supply current and natural air cooling. Because of this mechanism, the minimum pipe diameter of the in-pipe mobile micromachine becomes 3.4 mm.

4.2 Traveling experiments

In a horizontally located acrylic pipe, traveling experiments are performed. Traveling distance is measured by a magnetoresistance device. The measured results are shown in Fig. 14. It is verified that the traveling direction is controlled by a shape memory alloy and that the traveling velocity of 7 mm/s is realized for pipe diameters 4 and 5 mm under no-load condition.

5. Conclusions

To realize an inchworm in-pipe mobile micromachine adaptable to pipe diameters, newly-devised fluid microactuator and supporting mechanism are proposed and fabricated and they are applied to an in-pipe mobile micromachine. The obtained results are as follows:

(1) A simple and inexpensive rubber-tube actuator constrained with a coil-spring is proposed and fabricated and the static characteristics are

experimentally investigated. Also, a mathematical model is derived.

(2) A compact toggle-type supporting mechanism adaptable to pipe diameters is proposed and fabricated and the supporting force is investigated experimentally and theoretically. A design method is established.

(3) An in-pipe mobile micromachine for pipes with 4 - 5 mm in diameter is designed and fabricated. Through traveling experiments, it is verified that the traveling direction is controlled and the maximum traveling velocity of 7 mm/s under no-load is realized independent of pipe diameters.

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