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Development of a hybrid ferrofluid seal technology for miniature pneumatic and hydraulic actuators

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ABSTRACT

Future microrobotic applications require actuators that can generate a high actuation force and stroke in a limited volume. Up to now, little research has been performed on the development of pneumatic and hydraulic microactuators, although they offer great prospects in achieving high force densities. One of the main technological barriers in the development of these actuators is the fabrication of powerful seals with low leakage. This paper presents a seal technology for linear fluidic microactuators based on ferrofluids. A design and simulation method for these seals has been developed and validated by measurements on miniaturized actuator prototypes. These actuators have an outside diameter of 2 mm, a length of 13 mm and have been tested using both pressurized air and water. Our current actuator prototypes are able to operate at pressures up to 1.6 MPa without leakage. At these pressures, forces up to 0.65 N have been achieved. The stroke of the actuators is 10 mm.

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1. Introduction

In order to improve the force density of microactuators, this research focuses on the development of actuators driven by pressurized gasses or liquids. Despite reports on the high force and power densities of these actuators [1–7], they remain uncommon in microsystems. In the past, a number of promising inflatable hydraulic balloon actuators have been reported [8–10], nevertheless these actuators are limited to short actuation strokes and have a non-linear actuation behaviour. In large scale applications, piston-cylinder pneumatic actuators are used because they can achieve relatively large strokes and their actuation force is a linear function of the applied pressure. Nevertheless, appropriate seal technologies need to be developed in order to apply these actuators in microsystems. Classic seal technologies such as rubber O-rings are unsuitable at microscale since small shape or size variations of the rubber O-ring, the seal house or the piston result in either high friction or leakage. In order to avoid these strict production requirements, this paper proposes to employ liquid seals based on a ferrofluid. The implementation of these seals is comparable to that of rubber O-rings, but the seal ring consists of a liquid (in this case a ferrofluid) instead of a rubber. Ferrofluids are a stable colloidal suspension of single domain superparamagnetic particles in a carrier fluid [11,12]. When a magnetic field is applied on a ferrofluid, it moves to the location with the highest magnetic flux density.

This property allows to confine the ferrofluid in for instance a ring shape, and hence to create a seal. The liquid nature of these seal rings allows them to adjust their shape to production errors on the piston and the seal house. This adaptation in shape does not affect the leakage or friction of the seals. Moreover, promising properties such as the restoration of the seal after a pressure overload have been observed.

Similar ferrofluid seals have already been proposed for rotating shafts [13–15]. This paper differs from former publications because it describes linear seals and because it focuses on the miniaturisation of these seals. Regarding miniaturisation, it is interesting to note that the particles suspended in ferrofluids are typically only 10 nm in diameter. This makes these fluids more attractive for microsystems than for instance magnetorheologic fluids [16]. In addition, a new hybrid seal configuration that can resist high pressures without leakage is developed and tested.

Fig. 1 depicts the actuator configuration that is investigated in this paper. It is a typical hydraulic double acting actuator, which means that the actuator is able to generate both pushing and pulling forces depending whether the pressure is applied to the pressure port 1 or 2, respectively (see Fig. 1). The goal of this research is to develop an actuator with a section of 1 mm² and a length of 15 mm that can generate forces of 1 N and a stroke of 10 mm. These specifications are based on requirements for a tool for minimally invasive surgery. However, these actuators can also find applications in a large number of other microrobotic applications such as inspection and assembly robots or tactile displays.

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Fig. 1. General layout of the fluidic actuators investigated in this paper.

2. Actuator design

2.1. Ferrofluid seals

As pointed out by Rosensweig [11], the classic Bernoulli equation can be extended to Eq. (1) in order to describe ferrofluids:

$$\rho \left(\frac{\partial v}{\partial t} + v \cdot \nabla v \right) = -\nabla p + \mu_0 M \nabla H + \eta \nabla^2 v + \rho v \quad (1)$$

with v the flow velocity, t the time, p the composite pressure, μ_0 the permittivity of free space, H the magnetic field, ρ the density of the fluid, g the gravity constant and M the magnetization (magnetic dipole moment per volume). From Eq. (1), it can be derived that the maximum seal pressure of a ferrofluid seal is ruled by Eq. (2) [11]:

$$\Delta P = M_s \cdot N \cdot (B_{\max} - B_{\min}) \quad (2)$$

with M_s the saturation magnetization, N the number of seal stages and $B_{\max}$ and $B_{\min}$ the maximal and minimal value of the magnetic flux density in the ferrofluid, respectively. Eq. (2) shows that the maximum seal pressure can be increased by optimizing following parameters:

- First, the material constant M_s should be as high as possible. In this research we investigated a ferrofluid from Ferrotec Corp. [12]. Measurements reported in an earlier publication [3] show that the saturation magnetization is probably around 40 kA/m.
- Second, the number of seals N that are placed after each other should be as high as possible. Nevertheless, N is limited because a high number of seals result in complicated structures. In addition, for a same seal length, a high N value results in higher values for $B_{\min}$ which reduces the gain in seal pressure per seal step (see Eq. (2)). Therefore, this paper only investigates single stage ($N=1$) and double stage ($N=2$) seals.
- The last design parameter that should be maximized is $(B_{\max} - B_{\min})$. This parameter is optimized using finite element simulations of the magnetic flux density for different seal geometries.

A study of different seal designs based on other arrangements of the permanent magnets within the seal has been discussed in an earlier publication [3]. This study showed that permanent magnets with a radial magnetization allow to generate higher seal pressure than magnets with an axial magnetization. Figs. 2 and 3 show simulations of the magnetic flux density along the piston wall (see dashed line) for magnets with a radial magnetization. Fig. 2 illustrates single stage seals while double stage seals are sketched in

Fig. 2. Top: Magnetic flux density as a function of the position in a single stage ferrofluid seal. Bottom: Sketch of a seal cross-section. The dashed line indicates the location where the magnetic flux is calculated.

Fig. 3. From the latter figure and Eq. (2), it can be derived that the maximum seal pressure of a double stage seal is approximately 112 kPa ($2 \cdot 40 \text{ kA/m} \cdot (1.6 \text{ T} - 0.2 \text{ T})$). The magnets used in these simulations all have an inside diameter of 0.3 mm, an outside diameter

Fig. 3. Top: Magnetic flux density as a function of the position in a double stage ferrofluid seal. Bottom: Sketch of the cross-section of the seal. The dashed line indicates where the magnetic flux is calculated.

Fig. 4. Schematic cross-section of a hybrid ferrofluid seal (left) and an equivalent hydraulic scheme of the seal (right).

of 1.2 mm and a remanent field B_r of 800 mT. The clearance between the seal and the piston is maximally 5 μm , and the spacer is made out of magnetically conductive steel.

2.2. Hybrid seals

As pointed out in the introduction of this paper, the aim of this research is to achieve actuation forces in the range of 1 N. If a 1 mm² piston is used, this means that pressures of typically 1 MPa need to be sealed. However, the above discussion shows that the maximum seal pressure of the ferrofluid seals investigated in this paper is limited to approximately 100 kPa. Therefore, a novel hybrid seal technology that combines a clearance seal [1] with the ferrofluid seals described above has been developed. The operation principle of these seals is illustrated in Fig. 4. In the case of hybrid ferrofluid seals, the pressure load that is applied on the seal is forced through a small clearance before it reaches the ferrofluid seal. While the driving fluid is flowing through the clearance, part of the pressure load is dissipated, and therefore the pressure that must be overcome by the ferrofluid seal is reduced. In order to dissipate part of the pressure load, a certain amount of flow through the clearance seal is necessary. This flow is gathered in the collector and channeled back through the reflux channel to the fluid reservoir (see Fig. 4). The flow collector ensures an even pressure distribution over the circumference of the seal. The right hand side of Fig. 4 shows a hydraulic scheme of the hybrid seal. In this figure, the ferrofluid seal is represented as an overpressure valve (i.e. a valve that opens at the

maximum seal pressure of the ferrofluid seal), the reflux channel and the clearance seal are represented by flow restrictions. As can be seen in this scheme, the clearance seal and the reflux channel are a hydraulic equivalent of an electric voltage divider. This analogy allows to dimension the seal such that the pressure acting on the ferrofluid seal stays below its threshold value. If the flow resistance of the clearance seal $\mathfrak{R}_{Clearance}$ is large in comparison with the flow resistance of the reflux channel $\mathfrak{R}_{Channel}$, the pressure P working on the ferrofluid seal can be reduced considerably. This gain in seal pressure G can be estimated by Eq. (3) in the case of laminar flow:

$$G = \frac{\mathfrak{R}_{Channel} + \mathfrak{R}_{Clearance}}{\mathfrak{R}_{Channel}} \quad (3)$$

with

$$\frac{1}{\mathfrak{R}_{Clearance}} = \frac{\pi \cdot R_o^4}{8 \cdot \eta \cdot l} (1 - k^2) \cdot \left[1 + k^2 - \frac{1 - k^2}{\ln(1/k)} \right] + \frac{\pi \cdot R_o^2 \cdot U}{P} \left(\frac{1 - k^2}{2 \cdot \ln(1/k)} - k^2 \right)$$

$$\frac{1}{\mathfrak{R}_{Channel}} = \frac{\pi \cdot R_{Tube}^4}{8 \cdot \eta \cdot l_{Tube}} \text{ and } k = \frac{R_o}{R_i}$$

R_i is the diameter of the piston rod, R_o is the diameter of the clearance seal, R_{Tube} is the diameter of the reflux channel, l is the length of the clearance seal, l_{Tube} is the length of the reflux tube, η is the viscosity of the driving fluid and U is the relative speed of the piston. Furthermore, the flow Q can be kept very low if the flow resistance of the clearance or the reflux channel is high enough. The flow rate can be estimated based on Eq. (4), in which ΔP is the total pressure drop over the clearance seal and the reflux channel:

$$Q = \frac{\Delta P}{\mathfrak{R}_{Channel} + \mathfrak{R}_{Clearance}} \quad (4)$$

Hybrid seals are particularly interesting for microdevices since they consist of relatively simple structures and allow to seal high pressures with relatively low friction forces. In addition, ferrofluid seals are relatively insensitive to production errors. In what follows we will investigate a seal with following dimensions: R_{Tube} 0.14 mm, l_{Tube} 100 mm, R_i 0.15 mm, R_o 0.158 mm, l 0.7 mm, η 0.001 Pa s and $U=0$ m/s. In this case, $G=11.1$ and $Q=13.6 \mu\text{l/s}$ at a pressure of 100 kPa. However, it should be noted that if the piston is placed completely eccentric in the seal orifice, the flow resistance $\mathfrak{R}_{Clearance}$ decreases with a factor 2.5 [1]. In that case $G=5$ and $Q=30 \mu\text{l/s}$ at a pressure of 100 kPa. This numerical example clearly shows that

Fig. 5. Illustration of a practical implementation of an actuator with two hybrid ferrofluid seals.

Fig. 6. Fabrication process of the pressure divider.

most of the pressure load is supported by the clearance seal. The ferrofluid seal is however still necessary in order to avoid any leakage of fluid outside the cylinder.

Fig. 5 sketches a practical implementation of the actuator shown in Fig. 4. As illustrated in this figure, it is also possible to implement a hybrid ferrofluid seal on the piston of the actuator. Many different seal configurations can be designed for this purpose, the seal shown in Fig. 5 works as follows: when the driving pressure P_h (Fig. 5, right) reaches a certain threshold value, the ferrofluid seal around the magnet on the left hand side will fail and allow a certain leakage Q to flow past the magnet into the reflux channel inside the piston. In this case, the magnet on the left hand side is acting as a restriction that reduces the pressure acting on the ferrofluid seal on the right hand side. Unless the actuator is driven by pressurized air, a tube needs to be connected to the end of the piston in order to collect the fluid that is flowing through the clearance seal. Since a small leakage between both pressure chambers is less detrimental than a leakage to the outside world, it is usually sufficient to use a clearance seal between the pressure chambers as illustrated in Figs. 1 and 4.

3. Actuator fabrication process

In what follows, the production of each of the components shown in this Fig. 5 will be discussed. The fabrication of the pressure divider is illustrated in Fig. 6. First, the outside diameter of the flange is determined by turning on a precision Hembrug lathe. The outside diameter of the flange is the same as that of the magnets in order to ensure good alignment of these components in the cylinder. Next, the outside diameter of the pressure divider is reduced over a certain length using a SARIX micro-electro discharge machine (μ EDM). The main reason for this reduction in diameter is to deal with misalignments between the pressure divider and the reflux channel inlet that is made in the cylinder wall (see Fig. 5). In step three, four grooves are milled to connect the clearance seal to the reflux channel. The orifice of the clearance seal is fabricated by μ EDM drilling in step four. Finally, the pressure divider is removed from the base material by wire- μ EDM.

The micromagnets are fabricated by laser micromachining at the company Audemars. The spacer is made from a steel caliper pen. This magnetically conductive pen is cut to length using wire- μ EDM, and the central bore is fabricated by μ EDM drilling. The fabrication processes used for the piston and the cylinder have been described in a former publication on clearance seals [1]. The cylinder is made by milling and reaming a brass cylinder on a KERN MMP micromilling machine. The piston rod is also made out of brass in order to avoid the piston to be attracted by the magnets of the seal. The piston rod has an outside diameter of 0.3 mm and an internal bore of 0.12 mm.

The most challenging aspect of the actuator fabrication is the assembly of all the components. It is important to note that all the components of the seal are glued to the cylinder, and therefore, leakage is only possible through the gap between the piston and the seal. This gluing is however particularly difficult because very small amounts of glue must be dispensed in a controlled way, relatively deep into the cylinder. For this purpose, a glue nozzle was developed (see Fig. 7). The nozzle has an outside diameter of 120 μ m and

Fig. 7. Glue nozzle used for the assembly of the actuator.

an inside diameter of 60 μ m. The nozzle is about 10 mm long and is used successfully with Loctite Hysol 9483 glue. Fig. 8 shows an actuator during the assembly process.

The actuator shown in Fig. 8 is using a plain clearance seal to isolate the two pressure chambers; alternatively, an actuator using an internal hybrid ferrofluid has been fabricated. This actuator is shown in Fig. 9, and is in accordance with the geometry presented in Fig. 5. In this case, the piston rod cannot be connected directly to the external system, since the reflux channel inside the piston rod must be connected to a tube. In order to overcome this problem, a clamping rod has been mounted on the piston as illustrated in Fig. 9.

Special attention is needed for the fluidic interconnections between the supply tubes and the cylinder wall. Glue based interconnections have been proposed in earlier publications [1,2]. Here, a modified version of the heat-shrink interconnections developed by Pan et al. [17] has been incorporated in the cylinder wall. These interconnections are fabricated in the following way (see Fig. 10): a channel with a flange is machined in the cylinder wall by μ EDM with Wire-Electro-Discharge-Grinding (WEDG) formed electrodes [18]. Next a micro-heat-shrink tube [19] is placed over the flange, which is then heated until the heat-shrink tube is enveloping it. Finally, an ordinary tube is connected to the other side of the heat-shrink tube (see Figs. 8–10). Fig. 10a shows the connector before attaching the tube, Fig. 10b, after attaching the tube, and Fig. 10c shows a cross-section of the interconnection. The heat-shrink tube

Fig. 8. Fluidic actuator with an external hybrid seal during assembly.

Fig. 9. Fluidic actuator with an external and internal hybrid seal before (top) and after assembly (bottom).

Fig. 10. Overview of the heat-shrink based fluidic interconnections implemented in this research. (a) Interconnection before attaching the tube, (b) interconnection after attaching the tube, and (c) cross-section of the fluidic interconnection.

used in this interconnection is the Iridium X2-025-CLR from the company *Cobalt Polymers* [19]. Measurements showed that these connections can resist normal forces of about 2 N and resist pressures of 1 MPa.

Fig. 11. Layout of the test setup used for the characterization of the fluidic microactuators.

Table 1

Measurement of the average seal pressure and its standard deviation.

Fluid	Port 1	Port 2	Reflux	Seal pressure (kPa)
Air or water	Closed	Pressurized	Closed	130 ± 40
Air	Open	Pressurized	Open	260 ± 50
Air	Pressurized	Open	Open	500 ± 40
Water	Open	Pressurized	Open	540 ± 50
Water	Pressurized	Open	Open	>1600

4. Measurements

The actuators discussed above have been characterized in the test setup shown in Fig. 11. The hydraulic part of the setup consists of a membrane pump (Prominent γ) that can generate pressures up to 1.6 MPa and two controllable restrictions that allow to adjust the pressure applied to the actuator. This pressure is measured by a MEMS based pressure sensor (Druck PMP 1400) while the actuation force is measured by a sensitive strain gauge bridge. Both variables are monitored by a data acquisition system. The system shown in Fig. 11 is used to drive the actuator with pressurized water. A similar setup is implemented to investigate the properties of the actuator when using pressurized air. All measurements are performed on the actuator shown in Fig. 8.

First, the maximum seal pressure of the actuator is determined for both pressurized air and water as driving fluid. The first row of Table 1 shows the maximum seal pressure when the pressure divider is switched off (reflux channel closed). In that case, the maximum seal pressure is 130 kPa, which is in good agreement

Fig. 12. Actuation force of the actuator as a function of the applied pressure in the case of pressurized water (top) and air (bottom).

Fig. 13. Repeated actuation cycles for both the pushing force (top) and pulling force (bottom) of the actuator using pressurized water.

with the 112 kPa estimated based on Eq. (1) earlier on. When using a pressure divider and water as driving fluid, the seal pressure is increased 4.2 times. This is in agreement with the factor 5 derived for G earlier on. The maximum seal pressure is always higher if the pressure is applied on port 1 because in that case the piston acts as an additional seal. The relatively high standard deviation on the measurements shown in Table 1 is due to the fact that is sometimes difficult to detect the exact moment of failure of the seal. Also other factors that are difficult to control, such as the amount of ferrofluid in the seal and the eccentricity of the piston, influence the seal pressure [3].

Fig. 12 shows the actuation force of the actuator as a function of the applied pressure in the case of pressurized water (top) and air (bottom). These graphs also show the theoretical actuation force expected theoretically based on the applied pressure times the piston area. As can be seen in these figures, the actuator has a relatively high hysteresis. This hysteresis is not inherent to the seal technology, but is probably the result of friction between the piston and the cylinder due to misalignments in the assembly process. The hysteresis can be reduced slightly by using pressurized water instead of air as shown in Fig. 12. However, in order to reduce it significantly, the alignment of the piston needs to be improved. This is important in order to improve the linearity and controllability of the actuator [7]. Fig. 13 shows that the measured hysteresis is repeatable and that pulling force is lower than the pushing force, as expected from the smaller active piston area.

The flow through the reflux channel was 42 $\mu\text{l/s}$ at a driving pressure of 200 kPa. This is in agreement with the 27–60 $\mu\text{l/s}$ estimated earlier on. The maximum actuation force measured was 0.65 N at

1.4 MPa. The actuation force is lower than the desired 1 N force but it can still be improved by increasing the driving pressure. However, the fluidic interconnections may need to be redesigned since leakage of the interconnections has been experienced at 1.6 MPa. The maximum seal pressure can also be increased by reducing the tolerances on the clearance seal.

The stroke of the actuator is 10 mm, which is 77% of the actuator length. In addition, promising properties such as the restoration of the seal after a pressure overload have been observed. Finally, it should be noted that during these experiments the actuator is driven by large scale commercially available valves. Nevertheless, similar experiments could also be performed using for instance the microvalves presented in Ref. [20].

5. Conclusions

Hydraulic microactuators show promising properties for the generation of high force densities at microscale. However, appropriate microseals need to be developed in order to seal these actuators. This paper presents a new seal technology that combines a clearance seal with a ferrofluid seal. These seals are able to seal high pressures without leakage and with low friction. Therefore, they are particularly interesting for microapplications. These seals have been designed and integrated in a miniaturized fluidic actuator. Forces up to 0.65 N at a driving pressure of 1.4 MPa have been observed. The actuation stroke is 77% of the total actuator length.

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Biographies

Michael De Volder was born in Antwerp, Belgium, in 1978. He received his master degree in mechanical engineering from the University of Leuven (Belgium) in 2002.

He performed his Ph.D. research at the same university on hydraulic microactuators and microvalves. In 2005, he stayed as a visiting researcher at the Precision & Intelligence Laboratory of the Tokyo Institute of Technology. In 2007, he obtained his Ph.D. degree and started his postdoctoral research in collaboration with IMEC, Belgium. In 2008, he stayed as a visiting postdoctoral researcher in nanotechnology at the Massachusetts Institute of Technology, and the University of Michigan. His current interests include micro- and nanotechnology.

Dominiek Reynaerts was born in Tienen, Belgium, in 1963. He received his mechanical engineering degree from the Katholieke Universiteit Leuven, Belgium, in 1986. He has been working at the Mechanical Engineering Department of that same university, as a research assistant, starting from 1986. He obtained his Ph.D. degree in mechanical engineering in 1995, and became assistant professor at the Katholieke Universiteit Leuven in 1997. He currently is a professor in mechanical engineering with research and teaching activities in precision engineering, micro-mechanical systems, advanced actuators and design methodology. He is a member of IEEE and euspen.