Design of an Electrodynamically Actuated Microvalve Using COMSOL Multiphysics and MATLAB

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Abstract: This paper describes the design of a normally closed, electrodynamic microvalve. Magnetic forces between a permanent magnet in the valve cover and a soft magnet in the valve seat hold the valve closed. The combination of electrodynamic actuation and a mechanical restoring spring are used to open the valve. A device model and a design optimization strategy using COMSOL Multiphysics and the MATLAB Optimization Toolbox are presented. The theoretical performance of the optimized valve is calculated and shows that a maximum static pressure approaching 1 MPa is achievable.

Keywords: Microvalve, electrodynamic, microfluidics, MEMS.

1. Introduction

Microfluidic systems are an active topic of research in the MEMS field, with applications ranging from chemical analysis to fuel supplies for micro power systems [1]. Many of these microfluidic systems require a valve for flow regulation. Electrodynamic actuation, which involves the interaction between a coil and a permanent magnet, is of interest for microfluidic valves because of its large actuation stroke and the possibility of employing magnetic latching to achieve normally-open, normally-closed, or bistable devices. While this transduction method is known to possess a scale-independent force per unit volume [2], it has received little attention for use in microvalves due to the difficulties associated with fabricating microscale permanent (hard) magnets. Recent advances in microfabrication of permanent magnets with dimensions of tens to hundreds of microns [3, 4], however, make improved microvalve designs possible. In particular, electrodynamic actuation shows promise for high pressure microvalve applications. Previous electrodynamic microvalve design efforts [2, 5-9] have reported closing pressures in the tens of kilopascals. The goal of this design effort is to develop a high pressure microvalve capable of a closing pressure of 1 MPa to regulate the flow of gaseous fuels (e.g. propane) for micro heat engines or fuel cells.

2. Design

The electrodynamic microvalve geometry is pictured in Figure 1. The valve consists of a fluid inlet and a valve cover suspended by four serpentine springs as shown. Attractive forces between the soft and permanent magnets, which exceed inlet fluid forces, hold the valve locked in the closed position. Passing current through the coils on the surface of the valve seat changes the magnetic field and allows opening of the valve with the help of the serpentine spring. Figure 2 illustrates the opened and closed configurations of the microvalve.

The ultimate objective for the design of a high pressure electrodynamic microvalve is maximization of the closing force while ensuring the ability to open the valve. For the present design, passing current through the coils results in a reduced closing force, but not an opening force. In other words, the electrodynamic actuation force by itself is insufficient to overcome the magnetic attraction between the soft magnet and permanent magnets. Thus, the purpose of the coils is to reduce the closing force enough that the supporting springs may open the valve. The z-direction magnetic force acting on a magnet is given from Kelvin's formula [10, 11] as

$$F_m = \mu_0 \int_{V} M_z \frac{\partial H_z}{dz} dV , \qquad (1)$$

where the integral is over the permanent magnet volume, μ_0 is the permeability of free space, M_z is the z-component of the magnetization, and H_z is the z-component of the magnetic field strength. The latter two quantities are not easily found using analytical methods, so COMSOL Multiphysics is used to calculate them.

The microvalve represents a coupled magnetic-mechanical-fluidic system. Because

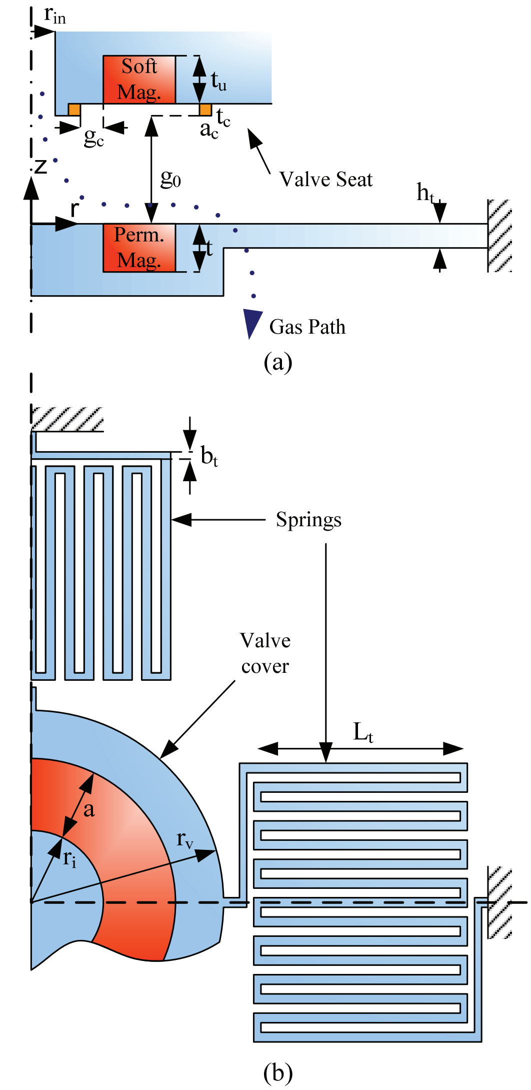

Figure 1. Electrodynamic microvalve geometry: (a) partial cross-sectional view of mating components; (b) partial plan view of valve cover.

the valve is designed for gaseous fuel at high pressures, the flow is compressible and likely turbulent as well. Thus, the analysis in the fluid domain is difficult. For the first-order design presented here, the valve is designed such that it can open without the aid of fluid forces and can stay closed against a maximum inlet pressure. Thus, the approach taken here is to treat the magnetic and mechanical domains as those of primary interest. In addition, the designs of magnetic and mechanical components are considered separately. Valve opening and closing are treated as quasi-static processes.

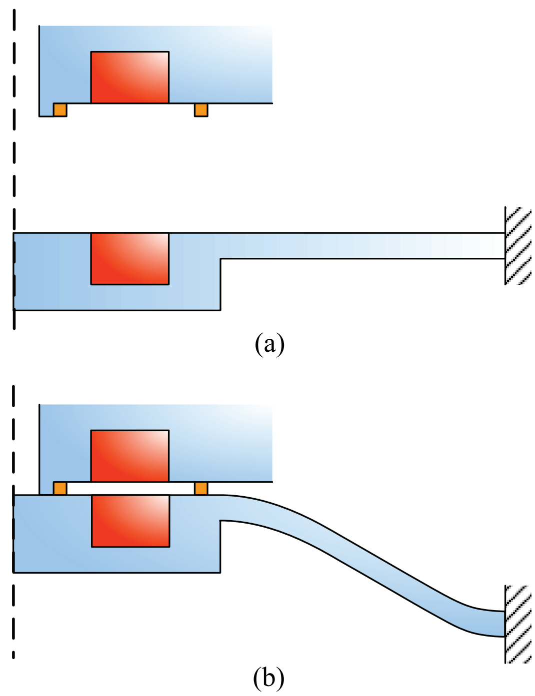

Figure 2. Cross-sectional view of mating components of the electrodynamic microvalve in the (a) open and (b) closed configurations.

Using this design philosophy, the goal in the magnetic domain is to maximize the closing force. However, the closing force cannot be considered alone because extremely large closing forces may result in a valve that cannot be opened. Thus, the true goal is to maximize the differential in closing force between when the coils are inactive ($J = 0$) and active ($J = J_{\max}$) while achieving a target closing force. The maximum current density $J_{\max}$ is based on a maximum power constraint and is given as

$$J_{\max} = \sqrt{\frac{P_{\max}}{a_c t_c L_{coil} \rho_e}}, \tag{2}$$

where $P_{\max}$ is the maximum power dissipation (set at 100 mW), a_c and t_c are the coil dimensions shown in Figure 1, and ρ_e is the resistivity of the coil material. The coil length is approximated as

$$L_{coil} \approx 2\pi(r_i - g_c - a_c/2) + 2\pi(r_i + a + g_c + a_c/2) + 2(a + 2g_c + a_c) = 2\pi(2r_i + a) + 2(a + 2g_c + a_c). \quad (3)$$

This expression for coil length accounts for the inner and outer turns and two connections between them.

After finding the geometric parameters that yield the optimal magnetic force, the mechanical domain is considered. The goal in this domain is to "tune" the spring to provide sufficient force for opening when the coils are active but not sufficient force for opening when the coils are inactive.

Throughout the design process, material properties are necessary to perform calculations. The materials used in the design of the microvalve and their relevant properties are summarized in Table 1. Material compatibility, particularly compatibility with gaseous propane, was considered in the selection of each material.

2.1 Optimization in Magnetic Domain

The first step in the design process is to determine the magnet and coil configuration that

Table 1: Material properties of interest for design process

Material	Use	Properties of Interest
Silicon [12]	Structural components	Young's Modulus: $E = 150$ GPa Density: $\rho_s = 2300\ \text{kg/m}^3$ Ultimate strength: $\sigma_u = 7$ GPa
Samarium Cobalt [3]	Permanent Magnet	Remnant Magnetization: $B_r = 0.6$ T Density: $\rho_m = 8300\ \text{kg/m}^3$
Permalloy [13]	Soft Magnet	Relative permeability: $\mu_r = 8000$
Copper [14]	Coils	Resistivity: $\rho_e = 1.724\times10^{-8}\Omega\cdot\text{m}$

results in the optimum force reduction when the coils are active. The objective function for minimization (note the negative sign) can be written as

$$f_{obj}\left(\overline{X}\right) = -\left[F_m\big|_{\substack{g=0\\J=0}} - F_m\big|_{\substack{g=0\\J=J_{\max}}} \right], \quad (4)$$

where F_m is the electromagnetic force and g is the distance the valve is opened relative to the bottom edge of the coils. Thus, the objective is to *maximize* the difference between the closing force with inactive coil and the closing force with active coil. The variables of interest for this optimization are:

- Permanent and soft magnet width, a
- Permanent magnet height, t
- Soft magnet height, t_u
- Magnet inner radius, r_i
- Coil width and height, a_c and t_c
- Distance between magnet edge and coil edge, g_c

The result is 7 design variables, all of which correspond to geometric dimensions, that make up the design vector $\overline{X}$. A formal statement of the optimization problem is below.

Objective

Minimize: $f_{obj}\left(\overline{X}\right) = -\left[F_m\big|_{\substack{g=0\\J=0}} - F_m\big|_{\substack{g=0\\J=J_{\max}}} \right]$

Design Variables

$$\overline{X} = \left(a, t, t_u, r_i, a_c, t_c, g_c\right)$$

Constraints

1. *Inlet:* $r_{in} \geq a_{flow,\min}/2$. This constraint accounts for the fact that the minimum inlet channel and one turn of the coils must be able to fit in the center region of the annular magnet. For the optimization, $r_{in} \approx r_i - a_c - g_c$ to avoid introducing r_{in} as an optimization variable and $a_{flow,\min} = 10\mu m$.
2. *Closing:* $F_m\left(g = 0, J = 0\right) \geq F_{f,in,\max}$, where $F_{f,in,\max} = p_{\max} A_{in}$. The electromagnetic closing force must exceed the maximum inlet fluid pressure force. The inlet area is $A_{in} \approx \pi r_{in}^2$. This means that the inlet is

assumed to encompass the entire region inside of the inner coil turn.

3. *Bounds:* $\overline{LB} \le \overline{X} \le \overline{UB}$. To ensure optimization results are able to be fabricated and are realistic, each design variable is subjected to upper and lower bounds. The minimum achievable feature size (in this case $0.5\ \mu$m was used) provides a good lower bound for most geometric parameters, but the upper bounds are not so clear. In general, a value of 1 mm was chosen for upper bounds. Coils are an exception; the maximum typical coil thickness t_c achievable via conventional electroplating methods is around $25\ \mu$m. The lower and upper bounds for each parameter are found in Table 2.

The optimization was performed in MATLAB using *fmincon*, a function for nonlinear constrained optimization. It makes use of sequential quadratic programming (SQP), which is a local optimization technique.

All magnetic calculations were obtained using an axisymmetric azimuthal induction currents model (AC/DC module) in COMSOL. Forces were calculated via a subdomain integration in COMSOL over the permanent magnet using equation (1). The model file was saved in MATLAB format and converted to a function with parameterized inputs. This MATLAB functional form of the COMSOL model was used to evaluate the objective function during optimization using *fmincon*. Figure 3 shows an example of the magnetic domain model with a typical mesh. Figure 4 is a graphical depiction of the interaction between MATLAB's *fmincon* and COMSOL.

The optimized geometric dimensions (rounded to the nearest μm) and objective function value are found in Table 3. The final value of the objective function indicates that for the optimized geometry, energizing the coils results in a 150 μN reduction in closing force. No constraints or bounds were reported as active at the conclusion of the optimization, though the unrounded values of a_c and t_u were very close to their bounds. Based on the optimization results, Figure 5 shows the electromagnetic closing force acting on the valve cover over a range of open gap distances for the cases when the coils are active $(J = J_{\max})$ and inactive $(J = 0)$.

Table 2: Upper and Lower Bounds of Optimization Variables

Parameter	LB, μm	UB, μm
a	20	1000
t	20	1000
t_u	20	1000
r_i	5	1000
a_c	0.5	50
t_c	0.5	25
g_c	0.5	100

Table 3: Results of Optimization in the Magnetic Domain

Optimal solution $\overline{X}_{opt}$	Optimum, μm
a	92
t	116
t_u	20
r_i	65
a_c	49
t_c	10
g_c	2
$f_{obj}\left(\overline{X}_{opt}\right)$	-150 μN

2.2 Spring Design

For the valve to open, a spring force must be applied that is greater in magnitude than $F_m\left(J = J_{\max}\right)$ but acts in the opposite direction. However, as illustrated in the free body diagram of Figure 6, fluid forces also act to open the valve and add to the required spring force. To ensure the valve can stay closed against a maximum fluid force, the spring force magnitude

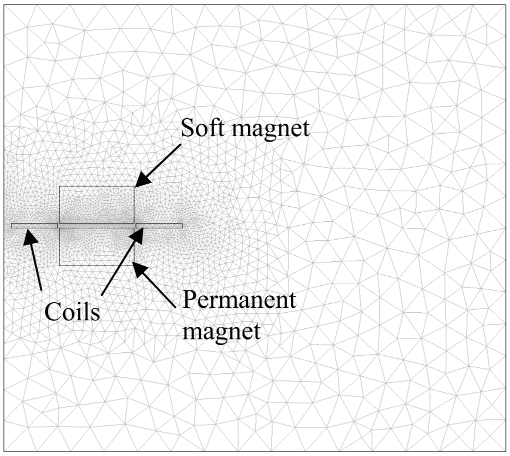

Figure 3. Typical mesh of magnetic COMSOL model.

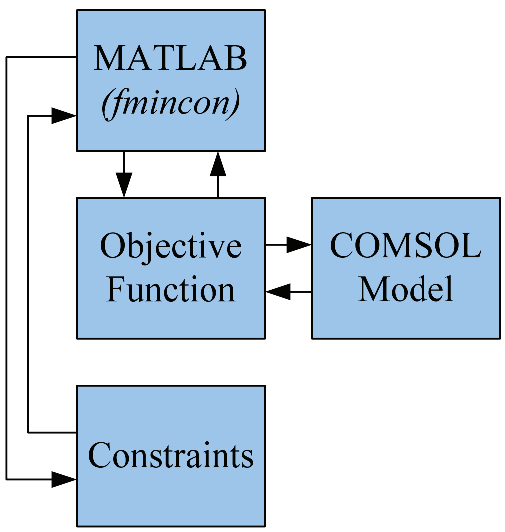

Figure 4. Program flow of the optimization process.

should be only slightly greater than $F_m(J = J_{\max})$. These considerations are reflected in a "target" spring force/displacement line that is also included in Figure 5. For a spring with these characteristics, the figure shows that the maximum distance the valve can open without significant assistance from fluid forces is nearly 15 μm.

The difference between the $F_m(J = 0)$ and $F_{springs}$ curves in Figure 5 represents the maximum fluid force that can be applied to the

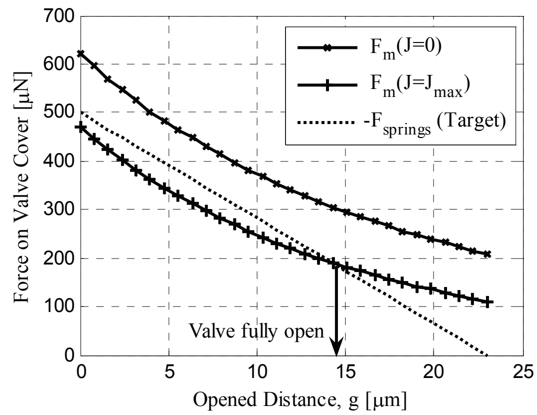

Figure 5. Electromagnetic and spring forces acting on the valve cover.

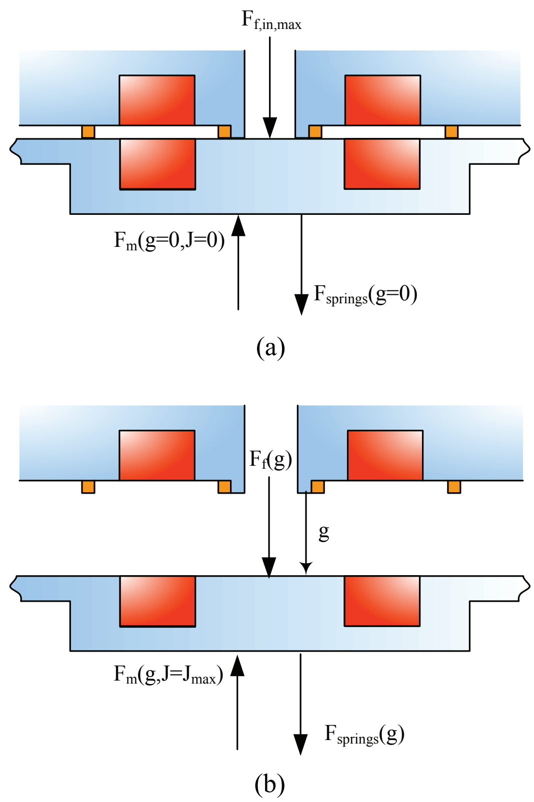

Figure 6. Forces acting on the valve cover in the (a) closed and (b) open positions.

valve cover and it still be able to snap closed. This fluid force is shown in Figure 7 as a function of opened distance. When the valve is closed ($g = 0$), the maximum inlet fluid force is

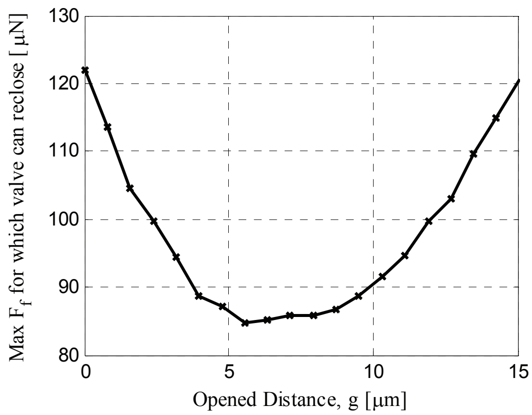

Figure 7. Maximum fluid force that the valve can overcome to snap closed.

$F_{f,in,\max} = 122\ \mu N$, which corresponds to an inlet pressure of 198 kPa (for $r_{in} \approx r_i - a_c - g_c$). The target spring of Figure 5 has been shown to provide acceptable opening and closing performance. The goal, then, is to design a physical spring with the characteristics of the target spring: a stiffness of $K_{springs} = 21.7$ μN/μm. Investigation of straight beam springs showed that such a low stiffness could not be achieved with a reasonable length, so a serpentine spring was adopted. Analytical approximations for the stiffness of serpentine springs are available in the literature [15] and details of the calculations will not be discussed here. A "rotated serpentine spring" (pictured in Figure 1) was selected because of its high compliance. The analytical stiffness estimates were used to determine the appropriate spring dimensions (b_t, h_t, L_t) and number of sections (N) to achieve the desired stiffness using four rotated serpentine springs attached at 90° intervals around the valve cover circumference.

The primary constraint in finding the beam dimensions was fitting the beam in limited die space; in this work, the die size was taken to be no greater than 10 mm × 10 mm. In finding the dimensions of the serpentine spring, spacing between beam elements of $s = 5\ \mu m$ was used to simplify later fabrication. The analytical solution was then checked using a more accurate nonlinear finite element analysis to ensure exact tuning of the spring. An appropriate valve cover radius r_v was also chosen at this stage. All of these dimensions are collected in Table 4.

Table 4: Remaining Microvalve Design Parameters

Description	Symbol	Value
Beam width	b_t	8 μm
Beam height	h_t	16 μm
Beam length	L_t	225 μm
Number of serpentine sections	N	6
Spring stiffness	$K_{springs}$	21.6 μN/μm
Initial gap	g_0	23 μm
Valve cover radius	r_v	190 μm
Inlet radius	r_{in}	6 μm

The inlet area also demanded revisiting because the maximum input pressure could be improved through further reduction of inlet area, A_{in}. The inlet radius was thus chosen as the lowest whole number in μm that guaranteed closing under the maximum fluid pressure of 1 MPa. This operation does not invalidate any previous results and simply requires a sealing surface for the valve cover between the inlet and inner coil turn. The new value of r_{in} is found in Table 4.

3. Conclusions

This paper contains discussion of the modeling and optimization of a normally closed, electrodynamically actuated microvalve theoretically capable of closing against a high pressure of 1 MPa. Electrodynamic actuation has been shown in this work to hold considerable promise for use in high pressure microvalves and thus merits further investigation.

A thorough understanding of fluid forces in the microvalve is necessary to refine estimates of attainable sealing pressures. In addition to fluidic modeling, other future work should include a thermal analysis of heating in the coils and estimation of the valve's dynamic characteristics. Physical realization of the device requires further microfabrication process development as well.

4. References

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