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A Seat Microvalve Nozzle for Optimal Gas-Flow Capacity at Large-Controlled Pressure

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Abstract—Seat microvalves are the most common microvalve type for gas-flow control. This paper presents a general method for optimizing the flow capacity of a seat valve nozzle and diminishing the requirements on the valve actuator’s stroke length. Geometrical analysis and finite element (FE) simulations show that for controlling large gas flow at elevated pressure, the optimal nozzle design in terms of flow capacity for a given actuator performance is a multiple-orifice arrangement with miniaturized circular nozzles. Experimental results support the design introduced in this paper. [0881]

Index Terms—Flow optimization, flow simulation, microvalve, nozzle.

I. INTRODUCTION

A large number of microvalves have been developed during the last 20 yr for different fluidic applications. The majority of microelectromechanical systems (MEMS) valves have been designed for gas flow control and only a limited number are intended for liquid applications. For the majority of gas flow applications, leak tightness is required, i.e., the valve must allow a complete closure of the flow duct, and microvalves are, therefore, most often of the seat-valve type illustrated in Fig. 1. They contain a boss or a diaphragm structure, suspended above a substrate that contains an orifice. The substrate surrounding the orifice is the valve seat. The out-of-plane movement of the boss/diaphragm regulates the valve opening. The boss moves in the same direction as the pneumatic force on the boss, and thus requires an actuation energy linearly proportional to the controlled pressure, the orifice area and the actuation stroke.

When comparing micromachined gas microvalves with conventional valves in the same price class, one finds a poor flow capacity for the micromachined valves, typically a factor 10 below that of conventional valves. There are two main reasons for this. First, the small cross-sectional areas in micromachined fluid ducts limit the flow. Second, the high pneumatic forces occurring when controlling gas pressurized at elevated pressures (typical several bars) demand a high actuator performance. A certain type of actuator can deliver a certain maximum work per actuator volume. Therefore, either the force delivered by the actuator or the actuator’s stroke, i.e., the flow gap, are limited.

Fig. 1. Illustration of a seat microvalve nozzle, featuring a boss suspended above an orifice in a substrate. The boss is allowed to move in the direction of the pneumatic force, and closes the valve at a stroke $z = 0$.

Table I gives an overview of earlier presented seat microvalve designs [1]–[7]. The data presented were not always complete, and a direct relation between the pneumatic performance and valve dimensions was not always clear. However, the values indicate the dimensions and the performance of the devices. The actuator stroke of devices [1]–[3], [5] is an order of magnitude smaller than their orifice size and therefore the control gap forms the limiting factor for the flow handling capacity. Moreover, the actuator area of most devices [2]–[6] is an order of magnitude larger than their orifice size and therefore the actuator size rather than the flow path dimensions (orifice area and control gap) determines the device’s footprint area.

The novel design approach introduced in this paper minimizes the required device footprint area and the required actuator stroke for seat microvalves, and outlines a microvalve nozzle design for maximum mass flow per footprint area. The design and its theoretical background are discussed and finite element simulations support the theoretical assumptions. This paper shows that feature miniaturization can solve a typical microfabrication restriction.

II. CONCEPT AND DESIGN

A. Theoretical Background

The mass flow $\dot{m}$ of a nonviscous compressible gas through a small orifice with flow cross-sectional area A with respective upstream and downstream pressure P_{in} and P_{out} can be described, assuming isentropic flow, with the relation

$$\begin{aligned} \dot{m} &= \frac{P_{in}}{\sqrt{RT}} C_d A \left(-\frac{P_{out}}{P_{in}} \right)^{\frac{1}{\gamma}} \delta(\gamma) \sqrt{\left(\frac{P_{in}}{P_{out}} \right)^{\frac{\gamma-1}{\gamma}} - 1} \\ &= K_1(P_{in}, P_{out}) \cdot A \end{aligned} \quad (1)$$

TABLE I
OVERVIEW OF MICRO GAS VALVES WITH TYPICAL DIMENSIONS

Actuation principle	Orifice size	Actuator stroke-length	Actuator size	Pneumatic performance	Comments	Reference
electrostatic / pneumatic	Ø 3 mm	5 µm	3.6 mm	800 sccm @ 550 kPa	Pressure balanced	[1]
bimetal	□ 230 µm	27 µm	2.5 mm	100 sccm @ 170 kPa		[2]
bimetal	□ 245 µm	30 µm	4 mm x 4 mm	1 150 sccm @ 800 kPa	3-2 way valve	[3]
bimetal	□ 180 µm	100 µm	8.8 mm (chip size)	1 000 sccm @ 1.03 MPa	Hewlett Packard	[4]
SCE	Ø 1.5 mm	50 µm	6 mm x 6 mm	5 000 sccm @ 500 kPa	Fluistor™, Redwood Microsystems	[5]
SMA	□ 100-500 µm	150 µm	8 mm x 5 mm	1 000 sccm @ 200 kPa	TiNi Alloy Company	[6]

for subsonic flow and

$$\dot{m} = P_{in} C_d A \frac{\alpha(\gamma)}{\sqrt{RT}} = K_2(P_{in}) \cdot A \quad (2)$$

for sonic flow [8], [9]. $\delta = \sqrt{4\gamma/(\gamma+1)(\gamma-1)}$, $\alpha = \sqrt{\gamma(2/(1+\gamma))^{(\gamma+1)/(\gamma-1)}}$ with γ the ratio (C_p/C_v) of specific heats for the gas, R is the universal gas constant over molecular weight ratio, C_d is the gas discharge coefficient, and T the upstream temperature of the gas. K_1 and K_2 are the respective mass flow per unit of flow cross-sectional area ($\dot{m}/A$), depending only on the inlet (and outlet) pressure, the upstream gas temperature and the gas properties.

Typically one single flow restricting cross-sectional area A determines the flow capacity of a valve; other geometrical valve features are unimportant. In a seat microvalve, this minimum cross-sectional area is either the *seat area* A_{seat} between the valve seat and the boss, perpendicular to the substrate, or the *orifice area* $A_{orifice}$ in the substrate enclosed by the seat. These configurations are further called *seat-controlled*, respectively, *orifice-controlled*. The orifice area determines the pneumatic force exerted on the boss and thus the actuator performance requirements. Increasing a valve's flow capacity by enlarging the orifice area demands a more powerful (larger) actuator. Enlarging only the seat area, on the other hand, does not pose this drawback. Therefore, in an optimal design the seat area should be maximized while the orifice area should be kept minimal and form the one flow-restricting cross-sectional area. The orifice area is then fixed by the required flow performance $\dot{m}_{req}$ under certain work pressure conditions

$$A_{orifice} = \frac{\dot{m}_{req}}{K_{1,2}}. \quad (3)$$

Here, the dimensionless parameter

$$\eta = \frac{A_{seat}}{A_{orifice}} \quad (4)$$

will be used to discuss the nozzle design. As measure of nozzle performance, the mass-flow per orifice footprint area

$$\dot{m}_{fp} = \frac{\dot{m}}{A_{orifice}}. \quad (5)$$

will be used.

B. The Design Optimization

To optimize the nozzle design it is important to consider the geometrical shape of the orifice/nozzle. Equations (1) and (2) assume loss-less flow, but for all real flow there will be losses and the influence of this will increase with decreased dimensions. For an orifice, a hole in a thin plate, there will be losses due to eddies downstream the orifice causing the main stream to contract through a smaller effective diameter, the vena contracta. As the opening size is reduced, the orifice will resemble more to a short channel than a hole in a thin substrate and boundary layers starts developing downstream with viscous losses. All these losses, which are related to the perimeter, become increasingly more influential on the mass-flow per area for decreasing orifice dimensions.

In order to minimize these losses, the optimal orifice shape must be circular, that is the geometrical area with the minimum perimeter. We define $r = r_{min}$ as the minimum orifice radius for which the effect of the flow losses on the mass flow are negligible. For a circular orifice, the radius r and control gap/actuator stroke z are the only geometrical parameters that influence the

Fig. 2. Overview of the $r - z$ parameter optimization space (top right) for nozzle design, and a closeup (middle left) of this plot. The analytically predicted air flow per orifice area $\dot{m}_{fp}(r, z)$ (7) and (8) for a 100-kPa pressure drop is color plotted as background. Results from FE analysis are displayed as round, color-coded dots. The parameter values for which $\eta = 1$ divide the parameter space in a seat-controlled region with nonoptimal flow capacity, and an orifice-controlled region with a constant maximum flow capacity. The optimal nozzle design parameters are at the junction of the $r = r_{\min}$ line and the $\eta = 1$ line. The white squares indicate earlier reported microvalve nozzle designs [2]–[4], [6]. The two bottom plots show cross-sections of the $r - z - \dot{m}_{fp}(r, z)$ space at $r = 20 \mu\text{m}$ (bottom left) and $z = 20 \mu\text{m}$ (bottom right). Again, FE results are indicated with color-coded dots, and the analytical model is indicated with solid curves. Note the overall agreement between the numerical FE results and the analytic model, except where at very small values of r frictional losses start playing a dominating role. The value $r_{\min} \approx 10 \mu\text{m}$ can be estimated from the bottom right plot. The top-left and middle-right 3-D illustrations show the FE-calculated pressure distribution in an orifice under, respectively, orifice-controlled and seat-controlled conditions.

Fig. 3. Illustration of the effect of the flow performance of a nozzle under different inlet pressure conditions. Analytic (lines) and numeric (dots) predictions of the mass flow per orifice area for air through nozzles with radius $r = 10 \mu\text{m}$ are plotted versus the actuator stroke z for inlet pressures of 33.3 kPa (dotted/triangles), 66.7 kPa (dash-dotted/squares), and 100 kPa (dashed/circles) relative to the atmospheric pressure. As expected, the numeric mass flow predictions are lower than the analytic predictions since the latter do not account for wall-induced frictional losses, which are not negligible at this small orifice radius.

flow handling capacity of the nozzle. We can now rewrite (4) and (5), respectively, as

$$\eta = \frac{2z}{r} \quad (6)$$

$$\dot{m}_{fp}(r, z) = K_{1,2} \frac{2z}{r} = \eta K_{1,2} \text{ for } \eta < 1 \text{ (seat - controlled)} \quad (7)$$

$$\dot{m}_{fp}(r, z) = K_{1,2} \text{ for } \eta > 1 \text{ (orifice - controlled)}. \quad (8)$$

Figs. 2 and 3 overview the $r - z$ parameter space.

The flow capacity in orifice-controlled designs is independent from r and z , allowing reduction of the actuator stroke z without limiting the flow capacity. The optimal design is determined by the minimum required actuator stroke z that still allows for the maximum mass-flow per orifice area $K_{1,2}$ and occurs for the parameters of r and z where $\eta = 1$ (orifice-controlled flow) and $r = r_{min}$ (negligible flow losses), i.e., where

$$z_{opt} = \frac{r_{min}}{2}. \quad (9)$$

The control of any required overall mass flow $\dot{m}_{req}$ can be accomplished with a multinozzle design consisting of n parallel nozzles, where

$$n \approx \frac{\dot{m}_{req}}{\pi K_{1,2} r_{min}^2} \quad (10)$$

with actuator stroke z_{opt} (9). The optimization procedure is illustrated in Fig. 4. The new design is the replacement of a single large orifice and boss with a multiorifice nozzle that consists of n orifices and bosses with a $\sqrt{n}$ times linear size reduc-

Fig. 4. Schematic visualization of the constant total orifice area ($A_{orifice} = n \cdot (A_0/n) = A_0$) and the increasing total seat area ($S = n \cdot (S_0/\sqrt{n}) = \sqrt{n} \cdot S_0$) for an increasing miniaturization factor n . The left column is a top view of the orifice shape in the substrate, the middle column illustrates the orifice area, and the right column illustrates the total length S of the seat.

tion. For any given value of n , the total (summed) orifice area ($A_{mo,orifice}$) remains constant, while the seat length S , and thus the total seat area ($A_{mo,seat}$), increases $\sqrt{n}$ times:

$$S \sim \sqrt{n}. \quad (11)$$

Two implementations of multinozzle designs are illustrated in Fig. 5. They differ in the manner in which the flow is guided toward the small orifices. The first design, illustrated in Fig. 5(a), features an out-of-plane gas flow through the valve. In the second design, shown in Fig. 5(b), an in-plane flow feeds the orifices from the side of the valve. It is important that the cross-sectional area of the flow path toward the innermost orifices does not limit the flow, which can be ensured by using a low density of orifices or a high boss height-over-thickness ratio.

The reduced control gap stroke z_{opt} makes stroke-limited actuation principles, such as electrostatic actuation [1], [10], [11] and stacked piezoelectric actuation, feasible for control of large flows.

When optimizing the nozzle design in terms of gas flow switching speed, rather than mass-flow per orifice area, values of $\eta < 1$ must be considered. In seat controlled flow, i.e.,

$$\frac{d\dot{m}}{dz} \propto 2\pi r K_{1,2} \text{ where } \eta < 1 \quad (12)$$

while in orifice-controlled flow

$$\frac{d\dot{m}}{dz} \rightarrow 0 \text{ where } \eta > 1. \quad (13)$$

Seat-controlled valves thus allow a faster valve switching time.

C. Size Reduction and Its Limitations

The increasing influence of frictional losses for reduced orifice diameter limits downscaling of the orifice size. Frictional

Fig. 5. Two implementations of the novel multiple-nozzle design. In a), the gas flows through the device from top to bottom. The actuator is positioned on the side of the boss structures. In b), an in-plane flow to feed the orifices. The actuator is positioned on top of the boss structures.

Fig. 6. A continued decrease of the orifice radii requires a decrease of the substrate thickness to prevent frictional flow losses in the orifice. (a) Original nozzle shape. (b) Continuously miniaturizing the orifice radii increases frictional flow losses. (c) Thinning the substrate alleviates this, but might lead to diminished mechanical strength of the substrate.

losses in turbulent duct flow mainly occur in the turbulent boundary layer and depend on first hand on the length of the flow duct (and can for macroscopic pipe flow be estimated using the Moody Chart [9]). Thinning the nozzle substrate, as illustrated in Fig. 6, can reduce the orifice length, but decreases the mechanical strength of the substrate.

Disregarding viscous losses in the above analysis is valid for compressible, turbulent gas flow as long as the turbulent boundary layers are thin compared with dimensions. Increased miniaturization will increase the influence of the boundary layers and consequently increased frictional losses and ultimately lead to laminar gas flow at high Mach number. To the authors' knowledge, such conditions are not quantified with analytical formulae. Moreover, the low substrate thickness-to-orifice radius ratio in the particular orifice geometries discussed in this paper leads to not fully developed flow conditions. A deeper investigation of the edge-induced frictional losses is recommendable and would complete parts of the Moody chart, but seen the scope of such study this is not included in this paper. The mere length-scale of the frictional boundary layer becomes comparable to the mean free path of the gas molecules. Therefore, gas rarification effects may need to be taken into account [12].

Another issue that needs to be taken into account is that in many industrial applications, the cleanliness of the handled gas puts a practical limit to the minimum feature size due to the risk for clogging. Particle sensitivity of the increased seat length needs to be considered as one can assume that the particle sensitivity is in linear relation with the seat length, the latter increasing with the square root of the miniaturization (11).

Depending on the fabrication technology, either the manufacturability of the ultraminiaturized orifices, frictional flow losses or gas cleanness requirements will form the practical limit to

the optimization procedure and determine the value of $r_{\min}$ and z_{opt} .

D. Other Nozzle Optimizing Solutions

A different approach for increasing the seat area was described elsewhere [8], [11], [13] and involves the shaping of a seat structure in a meander. For the same total orifice area, such design will always require a larger actuator stroke. Fig. 7 illustrates that the seat length of the meander shape can always be further increased, and thus the required actuator stroke decreased, by cutting of the meander arms and arranging them as separate orifices. Repeating this procedure results eventually in an improved design, similar to the multiorifice design described earlier. Meander-shaped structures are therefore suboptimal either in terms of flow performance or in terms of required actuator stroke.

III. FE SIMULATIONS

A. The FE Model

To further investigate the nozzle design, finite element simulations were performed for different nozzle geometries. The model is based on a multinozzle design of the type shown in Fig. 5(a) with the gas flow direction from the top of the structure to the bottom. The orifices are modeled using an axis-symmetric model with the radial flow boundary conditions set to zero (i.e., as a symmetry surface) due to symmetry. An overview of the model and its parameters is shown in Fig. 8. It consists of a cylindrical boss suspended over a circular orifice in a substrate. This geometry could be the typical result of a DRIE based fabrication process. Further details are omitted in the model for simplicity, because its purpose is to validate the general flow and actuator optimization procedure, rather than to evaluate a

Fig. 7. (a) The original valve orifice. Illustration of a meander design (b) improves the seat-length while keeping the orifice area constant. The dashed line indicates the original seat design. The total orifice area in this design remains unchanged while the seat area increases. (c) “Cutting of” the arms of the meanders increases the seat-length even further. (d) Repeating this procedure results eventually in an improved multinozzle design.

fabrication method. The ratio between the total model radius R and the orifice radius r is kept at a constant factor 3, ensuring that the ratio between orifice area and nozzle footprint area is constant for any degree of miniaturization. Unless indicated, the seat overlap $r_{\text{boss}} - r$ is $3 \mu\text{m}$. Finally, the inlet length L_{in} and the outlet length L_{out} were chosen as a compromise that ensures that the simulated part of the orifice is large enough, i.e., no critical changes in the gas flow are left out from the model, while keeping the amount of elements in the simulation manageable for every simulation run. In the simulations, the outlet is set at atmospheric pressure and the gas properties of air as a compressible gas are used (the relative inlet pressure was differed between different simulations). The choice of turbulence model in the FEA was found important for getting the solution to converge, and the Re-Normalized Group (RNG) Turbulence Model, a modified $k - \varepsilon$ turbulence model suitable when there are strong curvatures, was found to give good result. The turbulence model is not valid immediately adjacent to the walls and ANSYS uses the usual logarithmic-overlap layer (also called the “Log-Law of the Wall”) to approximate the velocity profile in the boundary layers. ANSYS default values for constants were used in both the RNG turbulence model and the log-law [14]. For the later that corresponds to smooth walls for macroscopic tube flow [9].

B. FE Simulation Results

The FE simulation results are plotted in Fig. 2 and 3. The average mass flow per unit of orifice area (5) is chosen as the measure for the flow capacity of a specific nozzle geometry.

Fig. 2 (top-left and middle-right) shows typical FE simulation results of, respectively, seat-controlled and orifice-controlled flow. It can be seen that the pressure drop is located at the seat and in the orifice, respectively.

The mass flow per unit of orifice area was calculated for a series of geometrical parameters with FE simulations and compared with the analytically predicted mass flow. A first observation is the agreement between the analytical flow predictions and the FEA. Second, as expected, the mass flow per unit of orifice area is independent of both the orifice radius and the stroke-length when the flow is orifice-controlled. For decreasing orifice radius, the flow becomes orifice-controlled.

Fig. 8. Schematic overview of parameters and boundary conditions used in the finite element analysis.

Note also the flow performance drop at small orifice radii where the flow suffers from frictional losses. This is not accounted for in the analytic predictions. The frictional losses in turbulent flow are mainly occurring in the boundary layers along the walls and it is important to notice that the used ANSYS default values for the log-law of the wall is for macroscopic smooth walls. It should be possible to simulate impact of wall roughness by variation of these constants. The model may need adaptation to include gas rarification effects and the scope of such study is too large to include in this paper and should be subject to a general completion of the Moody chart. The ANSYS flow predictions in the frictional range must, therefore, be treated with caution.

A number of FE simulations were also performed to investigate the influence of seat overlap $r_{\text{boss}} - r$ on the flow capacity. Large seat overlap is can be an important feature when leak tightness is required. Structures with an orifice radius of $10 \mu\text{m}$, a substrate thickness of $10 \mu\text{m}$ and a stroke-length of $10 \mu\text{m}$ were simulated, while the boss radius was varied between 13, 20, and $30 \mu\text{m}$ (i.e., a seat overlap of 3, 10, and $20 \mu\text{m}$, respectively). The change in mass flow per unit of orifice area was below 1.7% and all three designs were clearly orifice-controlled. This indicates that a large seat overlap does not significantly change the flow capacity of the design.

IV. EXPERIMENTAL

During the publishing process of this paper, and inspired by our preparatory work [15], Henning constructed and tested multiple-orifice flow test structures which behavior is in accordance with the predicted nozzle performance [8]. He also suggests a curve-fitting analytical formula to describe the transition between orifice-controlled and seat controlled flow.

V. CONCLUSION

The above study indicates that by optimizing the nozzle design of a seat microvalve, the flowthrough the valve can be opti-

mized and the valve actuator requirements can be reduced compared with today's microvalves. This results in an improved flow performance and a valve footprint area (cost) reduction. A novel orifice design is presented, containing multiple orifices and a limited stroke-length. Design limits caused by manufacturability, frictional losses, reliability, and leak tightness are discussed. Analytic predictions and FE simulations confirm the improved flow capacity of the design.

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