IMPROVED GAS FLOW MODEL FOR MICROVALVES

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ABSTRACT

Previously, a compact, pressure- and structure-based gas flow model for microvalves was developed [1]. It explained well the flow of gases ranging from 0.01 sccm to 2 slpm (limited only by availability of appropriate mass flow meters). Subsequently, several factors indicated the necessity for improvements to the model. In particular, the effect of valve seat periphery length was not accounted for.

As a consequence, this work presents a comprehensive yet compact compressible flow model for microvalves, which includes the effects of gas type, ambient temperature, pressure boundary conditions, and all important valve structures: valve seat periphery, inlet diameter, and membrane-to-valve-seat gap. The establishment of the model from measured data is described in detail. Projections for high flows in small area microvalves complete the description. The model covers accurately the full range of flow conditions, from seat-controlled flow through orifice-controlled flow. With these attributes, it is widely applicable to microvalves utilizing any form of actuation.

INTRODUCTION

Microvalves can be used in a wide variety of application areas. Control of gas flow makes them attractive for semiconductor process gas control and distribution, automotive manifold air pressure, mass air flow, and anti-lock brake control, and household natural gas control.

The previous model [1] described most of the behaviors required to design microvalves for these, and other, applications. Four factors have dictated substantial improvements in this model, however. First is a desire for a flow model to facilitate design of microvalves with flows in excess of 10 slpm. Second, the qualitative analogy drawn in a recent work [2] between electronic MOSFETs and pneumatic microvalves, begs a fully quantitative comparison, which only a comprehensive model can provide. Third, the effects of valve seat periphery length [3] were not included in the previous model. Finally, the tremendous, contemporary interest in arrays of microvalves and related microflow-control components also demands a comprehensive model.

Thus, this work presents a comprehensive yet compact compressible flow model for microvalves, which includes the effects of gas type, ambient temperature, pressure boundary conditions, and all important valve structures: valve seat periphery, inlet diameter, and membrane-to-valve-seat gap.

MODEL AND MEASUREMENTS

The model begins with the work of [1]. Compressible gas flow is treated assuming isentropic, adiabatic expansion of an ideal gas through an orifice [4]. The use of an exponential function was used to model appropriately the transition from flow controlled by the gap (seat-controlled flow), to flow controlled by the area of the valve inlet (orifice-controlled flow). This crucial function modeled the effective area of the valve. Formerly, this effective area was taken to be a function of only the valve inlet diameter and gap, without regard to the length of the periphery of the valve seat (see Figure 1). As such, the previous model worked well for valve inlets of square or circular dimension, but not for seats with an arbitrary area-to-periphery characteristic dimension.

As fully described in [1], the measurements involved a microfabricated silicon orifice, square in aspect, with a diameter (length of one side) of 540 μm, and a total periphery length of 2700 μm. A micrometer was used to act as the valve actuator, in order to control the gap precisely. Nitrogen at 298 K was used as the working fluid. Measurements were taken as a function of the gap, and as a function of the inlet pressure. The measurement results are seen in Figure 2. In order to show most conveniently the important linear nature of the flow vs. gap characteristic, the measurements for each inlet pressure shown on the legend are normalized to the asymptotic value of the flow, for large values of the gap parameter. The variation observed among the different curves is within the experimental error of the mass flow meter used in the measurements.

In [1], an exponential function described the effective area for seat-controlled flow, including the transition up to that for orifice-controlled flow (see Figure 2). This function was justified on the basis of the measured data of Figures 2 and 5. Subsequently, re-examination of this data revealed that the effective area in the seat-controlled regime was imperfectly modeled, since it predicted a quadratic dependence on valve gap, whereas the data (and simulation [3], and a previous, linear-only model [5]) supported better a linear dependence.

The resulting new flow model is shown in Equations (1). The left-hand box shows the full flow expressions, covering both sonic and subsonic flow. C_d is the coefficient of discharge for the valve, and principally for the orifice portion of the valve structure. C_v, also a model addition, accounts for departures of the flow from an ideal orifice, where such departures are due to the nature of the flow path, but not due to the key valve structural elements [6]. α and δ are gas-dependent function of the specific heat ratio γ. R is the universal gas constant, divided by the gas-dependent molecular

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weight. T is the stagnant temperature associated with the gas at the inlet to the microvalve. P_{in} and P_{out} are the inlet and outlet pressure boundary conditions, respectively.

The right-hand box shows the specific relationships for the effective valve area A_{eff}. The gap ratio parameter r defines the ratio between the gap z, and the hydraulic diameter D_h. (The hydraulic diameter is defined as $D=4A/W$, where W is the length of the periphery of the valve seat.) This ratio delineates the flow, as a function of r, into three regimes: seat-controlled flow; transition flow; and orifice-controlled flow.

For ratios of $r=z/D_h < r_0=0.15$ (seat-controlled region), the measured flow is found to be linear in r. (The previous model contained an inherent quadratic dependence, which the data did not support.) For large values of r (orifice-controlled region), the effective area is required to approach asymptotically the actual area A.

As with the previous model, an exponential function is chosen to characterize the transition region. This function is also shown in the right-hand box of Equations (1). The match point r_0 between the seat-controlled region and the transition region requires that the function $A_{eff}(r)$ and its first derivative be continuous. This requirement then results in a restriction on the exponential factor η in the equation. The first derivative, and the definition of the hydraulic diameter $D_h=4A/W$, dictate that $\eta = 1/4 - r_0$, which means the model has only a single free parameter. Using the flow data from [1], a value for η of 0.12 provides a best-fit of the model to the data, using linear regression techniques. Such a value results in $r_0=0.13$, which is quite close to the observed value of 0.15, and probably not different significantly in any statistical sense.

The results of the new model are evaluated against the data of [1] (see Figure 4). The results are actually somewhat improved (as determined by RMS error) compared to the previous model, especially at the low values of r.

DISCUSSION

As is evident from Figures 4 and 5, the model correctly predicts microvalve flow behavior. This behavior is reminiscent of a MOSFET [2], provided that the gap ratio r is taken to be analogous to the gate voltage in a MOSFET, and the inlet pressure is analogous to the drain voltage. Quantitative relation of the gap r to a pneumatic input actuation is easy using standard membrane theory [7]. Other quantitative analogies are drawn quickly by comparing the gas flow model equations to the related equations of a MOSFET. A comprehensive comparison of gas flow microvalve behavior, with electron flow MOSFET behavior, will be the subject of a subsequent work.

A surprising result of the modeling work is the very high flows which are available from very small structures. An example is shown in Figures 6 and 7. The valve seat design of Figure 7 packs a large amount of periphery length (approximately 13000 μm), into the same area as a conventional microvalve of our manufacture. For the conventional design, the valve inlet is of circular aspect, with a diameter of 1.5 mm, and a periphery length of approximately 4700 μm. Such a microvalve will flow approximately 9 slpm of N_2, under the conditions in the caption for Figure 6, and with a gap parameter of 50 μm. This flow is both predicted by the model, and measured in our manufactured devices. Using the high-periphery design, the microvalve will flow 10 slpm, but with a gap of only 5 μm. Cast in a different fashion, the microvalve will flow nearly 30 slpm, for a gap of 50 μm. Because of the highly polished nature of the silicon valve seat and silicon membrane used in these microvalves, they also have the ability to close to less than 0.01 sccm. As a consequence, they are appropriate for a wide range of industrial and instrumentation applications, demanding low leakage as well as wide dynamic range.

CONCLUSIONS

An improved microvalve gas flow model has been developed and characterized. It provides a powerful tool for microvalve design, for small and large flows, for microvalves of a wide range of dimensions. The model is appropriate for microvalves using any method of actuation. Because of the lumped element nature of the model -- and its relationship to parameters of interest to designers, and of control by device engineers -- it can be used to model individual devices, or arrays of devices.

Acknowledgement

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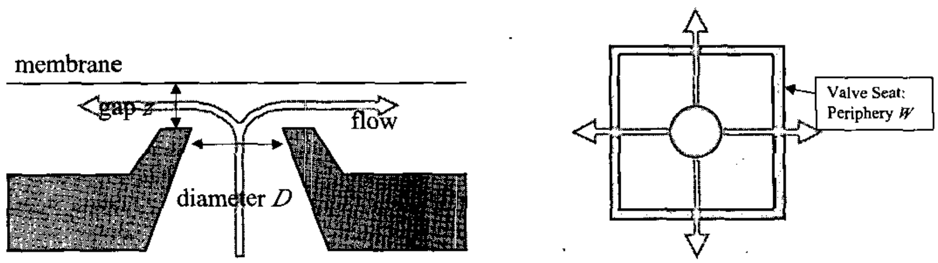

Figure 1: Important structural parameters in a microvalve. Left: Cross-section. Right: Plan view.

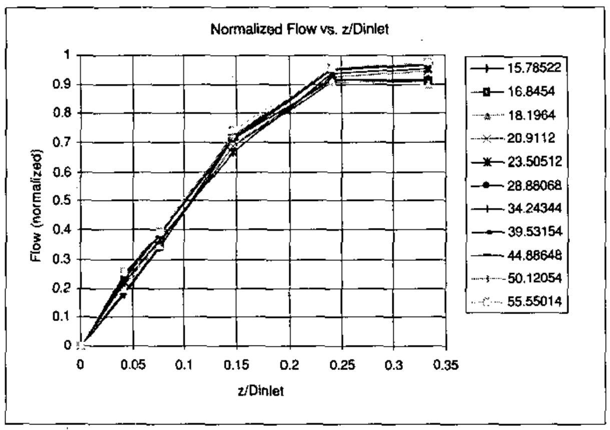

Figure 2: Raw Data, Normalized to Maximum Flow Value.

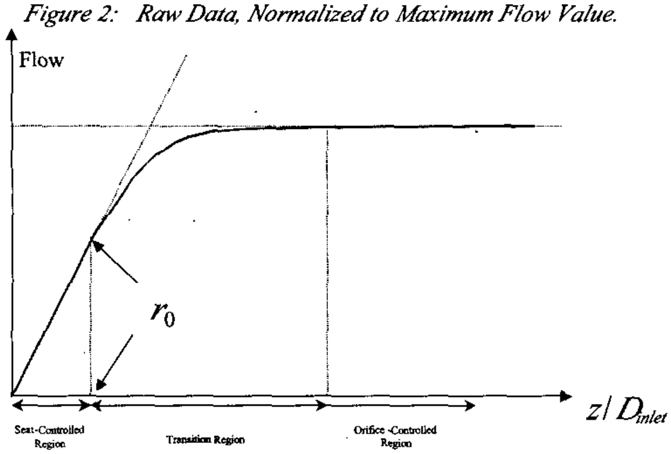

Figure 3: General Microvalve Flow Behavior

$$\dot{m}_{sonic} = C_d C_v A_{eff} \alpha(\gamma) \frac{P_{in}}{\sqrt{RT}}$$

$$\dot{m}_{subsonic} = C_d C_v A_{eff} \delta(\gamma) \frac{P_{in}}{\sqrt{RT}} \left(\frac{P_{out}}{P_{in}}\right)^{\frac{\gamma+1}{2\gamma}} \sqrt{\left(\frac{P_{in}}{P_{out}}\right)^{\frac{\gamma-1}{\gamma}} - 1}$$

$$A_{eff} = Wz = WD_h r \qquad \left(0 < \frac{z}{D_h} < r_0\right)$$

$$A_{eff} = WD_h r_0 + (A - WD_h r_0)\left[1 - \exp\left(-\frac{r - r_0}{\eta}\right)\right] \qquad \left(r_0 < \frac{z}{D_h}\right)$$

$$A_{eff} = A \qquad \left(\frac{z}{D_h} >> 1\right)$$

Equations 1: Compact Flow Model for Compressible Flows in Microvalves.

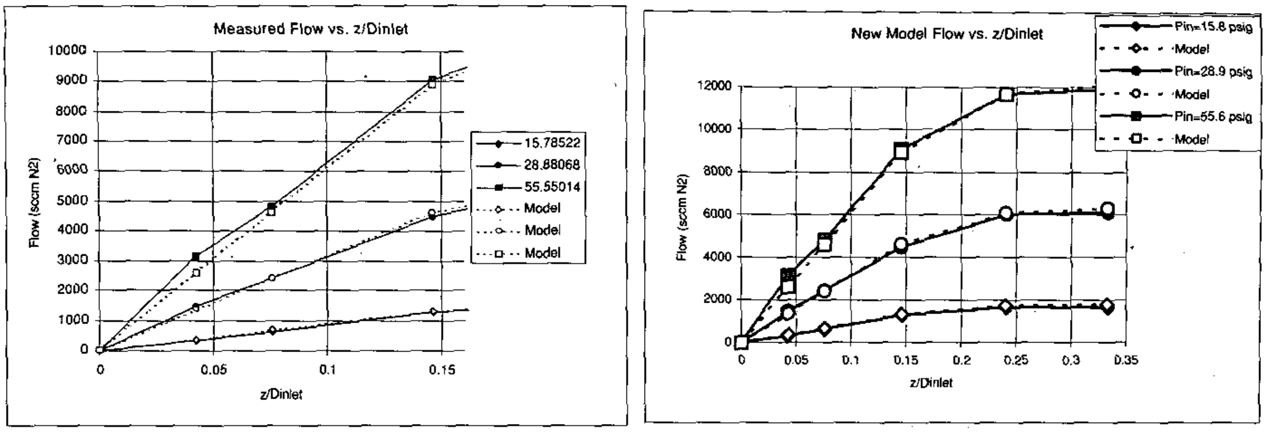

Figure 4: Flow vs. Normalized Gap r, Measured and Modeled

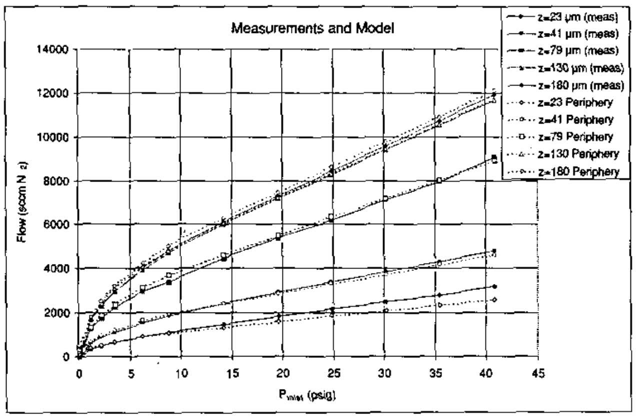

Figure 5: Full Data Set, Compared to Model.

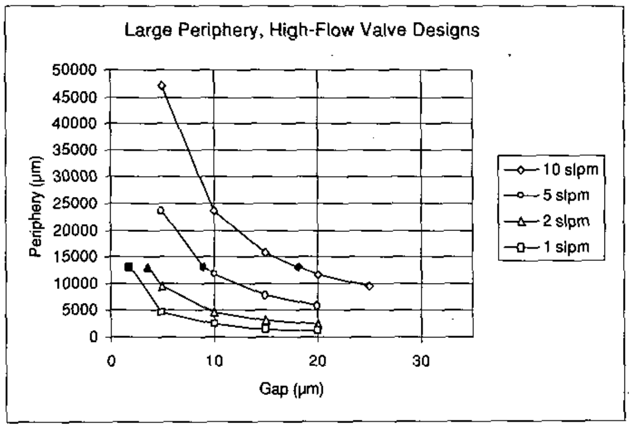

Figure 6: Modeled Iso-Flow Curves for High Flow Microvalve Designs. Flow Conditions: N_2; P_{in}=35 psia; P_{out}=15 psia; T=298K; C_d=0.84. The dark symbols represents designs which fit within the size constraints of a present high-flow design, yet delivers three times the flow for similar boundary and ambient conditions.

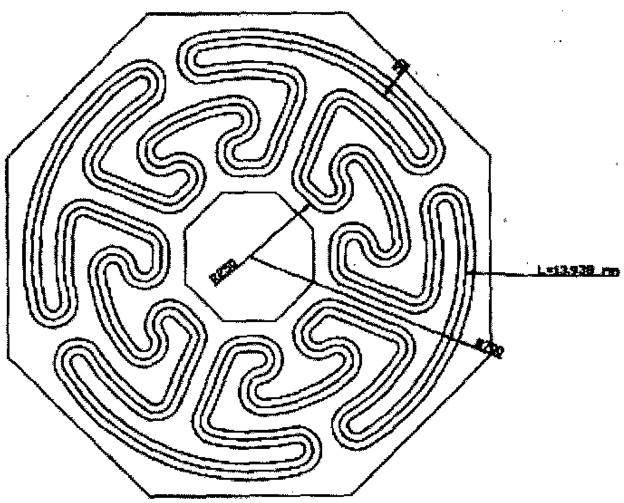

Figure 7: High-Flow Microvalve Seat Design with Large Periphery