A STUDY OF MICROMACHINED DISPLACEMENT PUMPS FOR VACUUM GENERATION

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ABSTRACT

This paper explores the limits of micromachined displacement pumps for vacuum generation. We studied a single stage pump which has a theoretical vacuum level of 406 torr (limited by dead volume), but which experimentally achieves a base pressure of 494 torr. Careful characterization has enabled us to quantitatively understand and measure the various effects that limit the performance for this class of pumps. In particular, we have measured valve leakage under vacuum operation, and correlated this to valve design parameters such as seal ring width.

INTRODUCTION

Micro vacuum pumps are one of the key components in miniature systems for chemical and biological analysis, e.g. mass spectrometer. Due to their portability, miniature analyzing systems can be deployed in numerous ways, such as in space research and field applications.

Micromachined pumps have been well studied [1, 2], however, only a few have been used for vacuum generation. Examples include Knudsen pumps [3], vapor-jet pumps [4], and displacement pumps [5, 6]. Knudsen pumps have a relatively small flow rate, and vapor-jet pumps involve fluid heating that might lead to high power consumption. Thus, a displacement pump seems very promising in microsystems, provided reasonable vacuum levels can be achieved.

Our vacuum pump was designed to be pneumatically driven for purposes of characterization, but ultimately will be integrated with piezoelectrics, similar to our previous work on high pressure pumps [7]. Silicon-on-insulator (SOI) wafers were chosen to form the displacement membrane due to the high quality device silicon layer. One SOI wafer and one standard silicon wafer were fabricated by deep-reactive ion etching (DRIE) and other standard micromachining techniques. Two pyrex wafers bonded to both the top and bottom sides completed the pump.

MICROPUMP AND ITS OPERATION

Micropump Structure

The micromachined vacuum pump is shown in Fig. 1 schematically. The pump includes three pneumatic actuation ports along with inlet and outlet ports. Device layers, L2 and L3, integrate two sealing valves and one pumping piston. Pyrex layers, L1 and L4, provide protection and support.

Figure 1: Schematic cross-section view of the micromachined vacuum pump.

Figure 2: Micromachined vacuum pump (bottom view).

The actual pump is 30mm×50mm×2mm in dimension, as shown in Fig. 2. The two ports labeled as "input valve" and "output valve" are actuation ports for the two valves, one to seal the channel between inlet and the pump chamber, the other to seal the channel between pump chamber and the outlet. "Input port" in the figure connects to the inlet and "Output port" is open to outside. The pump chamber and valves are all designed to permit a 6 micron stroke. The pump chamber is 20 mm in diameter, and the valves are 4 mm in diameter with an approximate seal-ring diameter of 180 microns.

Fabrication

The critical process is the fabrication of the membranes of the two valves and the pump chamber. To ensure a durable movement, there must not be any stress intensive points on the membranes, and fillets at the corners of the membrane and substrate are the key, as illustrated in [8].

Figure 3: 6-step operation cycle (arrows indicate pushing-up actuation by applying positive pressures).

SOI wafers with a 10 μ m thick device layer were picked for the L3 layer. A DRIE process was fine tuned to etch the handle layer until the buried oxide layer was reached. A "U" shape etching was desired, with fillets on both ends and without any excess silicon in the middle. A double-side polished silicon wafer was used as the L2 layer, and was etched on both sides multiple times. The L2 and L3 layers were bonded, and then the L1 and L4 layers were bonded afterwards, all at the wafer level.

Pump Operation

The pump was operated in a reciprocal 6-step cycle (Fig. 3), among which Step 2 and 5 represent gas charging and discharging steps respectively. Gas inside the test volume flows into the pump chamber during Step 2 by pulling the piston down, while gas inside the pump chamber is pushed out through outlet port during Step 5. Arrows in Fig. 3 indicate pushing-up actuation by applying a positive pressure of ~ 2 atm, while pulling-down actuation is applied to the unmarked actuation ports by using a vacuum of ~ 100 torr.

RESULTS AND DISCUSSION

Testing Setup

The micromachined vacuum pump was mounted onto a customized testing jig with all ports facing down, as shown in Fig. 4. A vacuum pressure sensor, Parker MPS-V3N-PGA, measured the pressure change inside the test volume. The actuation pressures were connected to testing jig through gas switches, Parker Kuroda VA01PEP34-1U. A microcontroller was used to control the pump operation.

Figure 4: Testing setup for vacuum generation.

Flow Rate

By connecting the pump outlet to a flow sensor, the flow rate was measured. Below 2 Hz operating frequency, the pumping flow rate increases with frequency as expected. At frequencies higher than 2 Hz, the flow rate remains constant until it reaches 3 Hz, when the pumping speed starts to drop.

Figure 5: Pump speed at 760 torr.

The plateau in Fig. 5 reflects the delay of pump chamber gas charging and discharging. In order to understand this effect, we can make an approximate estimate of the time constants for charging and discharging the pump chamber. By assuming Poiseuille flow, the fluidic resistances of charging and discharging (refer to Step 2 and 5 in Fig. 3) can be calculated approximately by integrating the resistances for rectangle channels [9] along the diameter of the chamber. For charging, we assume the pump piston is undeflected:

Figure 6: Resistance model for the pump chamber, for a flow along the chamber diameter, (a) for charging and (b) for discharging.

$$R_{charging} = \int_{-r}^r \frac{12\eta}{W(L)h^3} dL \quad (1-1)$$

where

$R_{charging}$: chamber fluidic resistance
 η : viscosity of air
 $W(L), L$: as labeled in Fig. 6
 h : chamber height, 6 μ m

Therefore, the fluidic resistance of pump chamber at charging step is

$$R_{charging} = 1.5 \times 10^{12} \text{ Pa}\cdot\text{s}\cdot\text{m}^{-3} \quad (1-2)$$

During discharging, we assume the central piston is fully pushed to the chamber ceiling with zero gap, and the gas exits from the annular periphery region. The flow path is simplified to two parallel trapezoid channels, and the chamber resistance is calculated as

$$R_{discharging} = 1.25 \times 10^{14} \text{ Pa} \cdot \text{s} \cdot \text{m}^{-3} \quad (1-3)$$

Analogous to electrical circuit, the fluidic capacitances of the pump chamber are derived from the relationship between flow rate and pressure drop

$$Q = C \frac{d\Delta P}{dt} \quad (2-1)$$

and mass conservation equation

$$Q = \frac{V}{\rho} \frac{d\rho}{dt} \quad (2-2)$$

where

Q : flow rate
 V : chamber volume
 ΔP : pressure difference
 ρ : density

From Eq. (2-1) and (2-2), we have

$$C = \frac{V}{\rho} \frac{d\rho}{d\Delta P} = \frac{V}{\rho} \frac{d\rho}{dP_{chamber}} = \frac{V}{P_{chamber}} \quad (2-3)$$

The fluidic capacitances for the pump chamber at 760 torr are

$$C_{charging} = 1.9 \times 10^{-14} \text{ Pa}^{-1} \cdot \text{m}^3 \quad (2-4)$$

$$C_{discharging} = 9.3 \times 10^{-16} \text{ Pa}^{-1} \cdot \text{m}^3 \quad (2-5)$$

Thus, the approximate time constants in the fluidic circuit are

$$\tau_{charging} = (R \cdot C)_{charging} = 0.029 \text{ s} \quad (3-1)$$

$$\tau_{discharging} = (R \cdot C)_{discharging} = 0.116 \text{ s} \quad (3-2)$$

The 500ms operating period, corresponding to 2 Hz frequency, is approximately twice the time constants of the charging and discharging steps plus 50ms each for the other four steps. Faster operation than 2 Hz could not improve the pump speed due to the inherent delay, and insufficient charging or discharging time might even degrade the speed, suggested from the flow rate testing.

Vacuum Generation

Based on the flow rate testing results, we measured the pump vacuum generation performance by leaving sufficient time for pump gas charging and discharging. With an operation frequency of 0.5 Hz, a base pressure of 494 torr was achieved (Fig. 7).

Vacuum performance is limited mostly by dead volume and valve leakage. Dead volume comes from design, e.g. the channel volume, other non-ideal factors such as wafer bow, and also insufficient charging or discharging. An efficiency factor α is introduced, which represents the fraction the pump chamber is evacuated each cycle during gas discharging ($0 < \alpha < 1$). Valve leakage comes from imperfect contact between the valve seat and valve boss. By taking all factors into account, we developed a model of displacement pumps for vacuum generation:

$$P_{vacuum} = \frac{P_0 \times (V_{channel} + (1 - \alpha) \cdot V_{chamber}) + P_0 \cdot Q \cdot T}{V_{channel} + V_{chamber}} \quad (4)$$

Figure 7: Vacuum generation performance. Speed is limited by the size of the test volume ($\sim 1 \text{ cm}^3$).

where

P_{vacuum} : ultimate vacuum generated
 P_0 : atmosphere pressure
 α : efficiency factor
 Q : leak rate
 T : period
 $V_{channel}$: pump channel volume
 $V_{chamber}$: pump chamber volume

Eq. 4 describes the vacuum generation performance of a single-stage pump. It shows clearly by increasing α , i.e. by reducing dead volume, the pump vacuum performance can be improved. Dead volume of the pump here includes four major parts: (1) channel volume; (2) residue chamber volume when chamber piston is fully pushed up; (3) dead volume caused by non-ideal structure factors like wafer bow; and (4) virtual dead volume caused by insufficient pump chamber gas charging and discharging. (1) & (2) are determined by design, (3) involves with fabrication process, and (4) is controlled by operation.

This model can help to understand better of the displacement pump for vacuum generation. However, before further discussion, another major limiter, leakage, needs to be studied.

Valve Leak Rate

For a single-ring valve seat, the valve flow is generally limited by the gap between valve seat and valve boss, and at low pressure difference, the fluidic resistance R_{valve} is as in [10]:

$$R_{valve} = \frac{6\eta}{\pi h^3} \ln\left(\frac{r_1}{r_2}\right) \quad (5)$$

where

R_{valve} : valve fluidic resistance
 r_1 : outer radius of valve seat
 r_2 : inner radius of valve seat
 h : gap between valve seat and valve boss

With inner radius of the valve seat as $50 \mu\text{m}$, a variety of valves were fabricated and their leak rates were measured. All the valves went through the same fabrication process and the same testing environment, which ensures they have the same gap h . As can be seen in Fig. 8, the fluidic resistance is proportionally to the

logarithm term and the leak rate results agree with that.

Figure 8: Valve leakage rate versus seal ring width (reproduced from [11]). Solid line is a best fit using Eq. 5.

Virtual Dead Volume

To explore the effect of virtual dead volume on vacuum performance, we conducted a two-step experiment. The performance is expected to be improved with larger discharging time which reduces the virtual dead volume based on the analysis above. The pumping started with a discharging time of 300ms per cycle. When the vacuum generation reached an equilibrium with a vacuum of ~530 torr, we increased the discharging time to 600ms and the pressure was reduced further to below 500 torr, as shown in Fig. 9.

Figure 9: Vacuum generation by tuning discharging time.

Analysis of Vacuum Generation Performance

With the leak rate, Q in Eq. 4, we have a better understanding of the vacuum generation performance of this displacement pump. Operation was fine tuned to minimize the virtual dead volume and an efficiency factor of 0.6 was obtained theoretically by neglecting part (3) of the dead volume. Without consideration of the leak term, we get $P_{vacuum} = 406$ torr from Eq. 4. By incorporating leak rate into the equation, we can have $P_{vacuum} = 425$ torr. The difference between tested result of 494 torr and calculated 425 torr would be mainly contributed by the imperfect assumption to neglect part (3) in dead volume estimation. Wafer bow may account for as much as 15% of the total pump chamber volume and the difference is right within that range.

CONCLUSION

We have developed a single stage micromachined

displacement vacuum pump. Its vacuum performance was measured to achieve a base pressure of 494 torr experimentally, and the factors limiting its performance were systematically studied. With all the knowledge we have gained, we built a model that explains the testing results and improves our understanding of displacement pump for vacuum generation. Dead volume and leakage are the two major limiters on vacuum performance. Our immediate next step is to reduce the dead volume with a more compact design and optimized fabrication process, and also to minimize leakage with valve seat redesign.

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REFERENCES

- [1] P. Woias, "Micropumps—Past, Progress and Future Prospects", *Sensors and Actuators B*, 105, 28 (2005).
- [2] D.J. Laser, and J.G. Santiago, "A Review of Micropumps", *Journal of Micromechanics and Microengineering*, 14, R35 (2004).
- [3] S. McNamara, and Y.B. Gianchandani, "On-chip Vacuum Generated by a Micromachined Knudsen Pump", *Journal of Microelectromechanical Systems*, 14, 741 (2005).
- [4] M. Doms, and J. Muller, "A Micromachined Vapor-jet Vacuum Pump", *Transducers & Eurosensors '07, The 14th International Conference on Solid-State Sensors, Actuators and Microsystems*, Lyon, France, 4C1.1 (2007).
- [5] K.P. Kamper, J. Dopfer, W. Ehrfeld, and S. Oberbeck, "A Self-filling Low-cost Membrane Micropump", *Proc. 11th Annual Int. Workshop on Micro Electro Mechanical Systems*, Heidelberg, Germany, 432 (1998).
- [6] A.A. Astle, H.S. Kim, L.P. Bernal, K. Najafi, and P.D. Washabaugh, "Theoretical and Experimental Performance of a High Frequency Gas Micropump", *Sensors and Actuators A*, 134, 245 (2007).
- [7] D.C. Roberts, H. Li, J.L. Steyn, O. Yaglioglu, S.M. Spearing, M.A. Schmidt, and N.W. Hagood, "A Piezoelectric Microvalves for Compact High-frequency, High Differential Pressure Hydraulic Micropumping Systems", *Journal of Microelectromechanical Systems*, 12, 81 (2003).
- [8] H.Q. Li, D.C. Roberts, J.L. Steyn, K.T. Turner, J. A. Carretero, O. Yaglioglu, Y.-H. Su, L. Saggere, N.W. Hagood, S.M. Spearing, M.A. Schmidt, R. Micak, and K.S. Breuer, "A High Frequency High Flow Rate Piezoelectrically Driven MEMS Micropump", *Technical Digest of the 2000 IEEE Solid State Sensor and Actuator Workshop*, Hilton Head Isl., SC, 6/4-8/00, Transducer Research Foundation, Cleveland (2000), pp. 69-72.
- [9] S.D. Senturia, *Microsystem Design*, Kluwer Academic Publishers, Boston, 2000.
- [10] P. Galambos, C.D. James, J. Lantz, R.C. Givler, J. J. McClain, and R.J. Simonson, "Passive MEMS Valves with Preset Operating Pressures for Microgas Analyzer", *Journal of Microelectromechanical Systems*, 18, 14 (2009).
- [11] V. Sharma, "MEMS Micropump for a Micro Gas Analyzer", Ph.D. Thesis, Massachusetts Institute of Technology, 2009.

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