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Development of a Surface Micromachined On-Chip Flat Disk Micropump

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Abstract: The paper presents research progress in the development of a surface micromachined flat disk micropump which employs the viscous and centrifugal effects acting on a layer of fluid sandwiched between a rotating flat disk and a stationary plate. The pump is fabricated monolithically on-chip using Sandia's Ultraplanar Multilevel MEMS Technology (SUMMiT™) where an electrostatic comb-drive Torsional Ratcheting Actuator (TRA) drives the flat disk through a geared transmission. The paper reviews available analytical models for flow geometries similar to that of the described pump, and presents a set of experiments which depict its performance and possible failure modes. Those experiments highlight future research directions in the development of electrostatically-actuated, CMOS-compatible, surface micromachined pumps. *Copyright © 2009 IFSA.*

Keywords: Flat disk micropump, viscous micropumps, surface micromachining, SUMMiT™.

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1. Introduction

Micropumps are key microfluidic components needed in many systems that handle fluids at the micro and nano scales. Their potential applications include implanted and transdermal drug delivery systems, micro Total Analysis Systems (µTAS) and electronic cooling [1], [2]. Review articles on micropump literature distinguish between mechanical and non-mechanical pumping principles [3]–[5]. Mechanical pumping is achieved by the mechanical movement of an element such as a diaphragm or a gear inside the pump chamber [6], [7], and they can be used with a wide variety of working fluids. Non-mechanical pumping relies on special effects such as electro-hydrodynamic (EHD), magnet-hydro-dynamic and electroosmotic effects [8]–[11]. These devices have no moving parts and are thus more reliable; however, they are generally limited by low flow rate and pressure rise capabilities, the need for high supply voltage, and the physical characteristics of the working fluids.

A number of recent investigations report on the utilization of the viscous drag effect in mechanical micropump design. This is motivated by the ability to generate a significant pressure heads in the viscosity dominated microscale flows by the simple rotation of a rigid element contiguous to the flow field. A number of realizations of viscous micropumps have been presented such as the eccentric disk viscous pump [12], the spiral channel micropump [13], and the single and double disks micropumps [14]. These micropumps are attractive because they are easy to fabricate, capable of handling a wide variety of fluids and can operate with no valves, which allows them to handle particle-laden fluids. The simplicity in design and operation of these pumps makes them good candidates for implementation in standard surface micromachining technologies where mechanical mechanisms with rotational freedom may be built on-chip [15], and where a number of verified actuators, couplers, and transmission mechanisms are available and can be readily incorporated into the driving system the rotating element [16], [17].

The paper presents a surface micromachined micropump consisting of a flat disk rotating continuously on top of a stationary plate. Figure 1 illustrates the operating concept of the pump. The rotating disk is maintained at a certain small distance above the stationary plate and pumped fluid is sandwiched in the space between the rotating disk and the fixed plate. When the disk rotates, the non-slip condition at its surface requires that the fluid rotate with the same velocity. Viscous effects diffuse away from the disk and induce a rotation in nearby fluid. Once particles between the fixed plate and the rotating disk have been accelerated by the plate, they are also repelled out in a radial flow. Continuity demands the outward moving fluid be replaced by an axial flow toward the disk from the quiescent fluid far from the disk. Fluid is thus pumped from the far stream toward the disk center, where the viscous forces induce a swirl; then the resulting centrifugal effect produces a radial flow.

Section 2 reviews available analytical solutions for the flow field in flow geometries similar to that of the pump. Section 3 presents an experimental verification for the validity of the proposed pumping principle on a scaled up miniaturized prototype. Section 4 presents the design of the pump in Sandia’s Ultraplanar Multilevel MEMS Technology (SUMMiT™) [18], where the driving, feeding, and sealing schemes are described. Finally section 5 presents a set of experiments that depict qualitatively pump performance and possible failure modes.

2. Mathematical Modeling

A number of early investigations have addressed the fluid motion between two co-rotating disks, e.g. Batchelor [19] and Stewartson [20]. Batchelor proposed that the core of fluid between the two plates away from the boundary layer on the disks rotates with constant angular velocity while Stewartson suggested that boundary layer forms only on the rotating disk and that the remaining fluid will not rotate if one disk is stationary or counter-rotating. A lubrication solution for the flow field between the two rotating disks has recently been presented by Aphale [21]. The problem considered by Aphale for the flow field between the plates of an open clutch is geometrically and dynamically similar to the flow between the plates of the pump considered in this work.

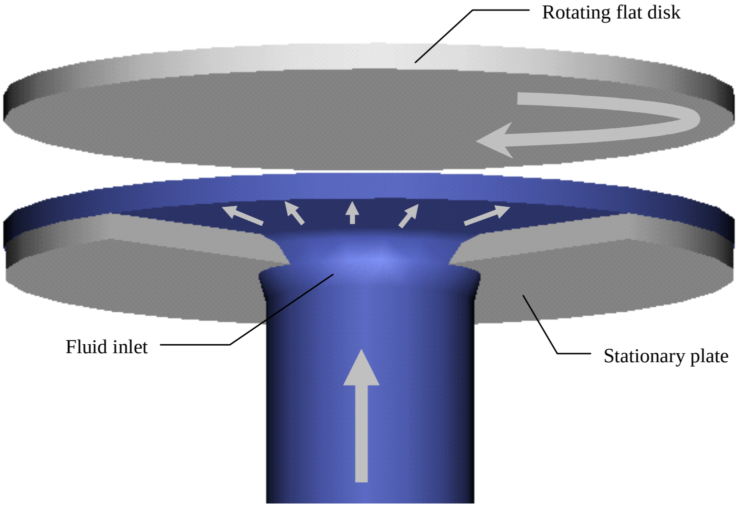

Fig. 1. The flat disk viscous pump concept.

With reference to the solution domain illustrated in Figure 2, the non-dimensional form of the continuity and momentum equations in the *r, θ* and *z* directions are:

$$\frac{1}{r^*}\frac{\partial}{\partial r^*}\left[r^* u^*\right]+\frac{\partial w^*}{\partial z^*}=0 \tag{1}$$

$$\frac{\partial p^*}{\partial r^*}-\frac{v^{*2}}{r^*}=\frac{\partial^2 u^*}{\partial z^{*2}} \tag{2}$$

$$\frac{\partial^2 v^*}{\partial z^{*2}}=0 \tag{3}$$

$$\frac{\partial p^*}{\partial z^*}=0 \tag{4}$$

where the non-dimensional parameters are defined as follows:

$$r^*=r/r_m=r/\left[\left(r_i+r_o\right)/2\right],\ z^*=z/h,\ u^*=u/u_s=u/\left(\omega^2 r_m h^2 \rho/\mu\right),\ v^*=v/v_s=v/\left(\omega r_m\right),$$
$$w^*=w/w_s=w/\left(\omega^2 h^3 \rho/\mu\right),\ p^*=p/\rho v_s^2=p/\rho\left(\omega r_m\right)^2$$

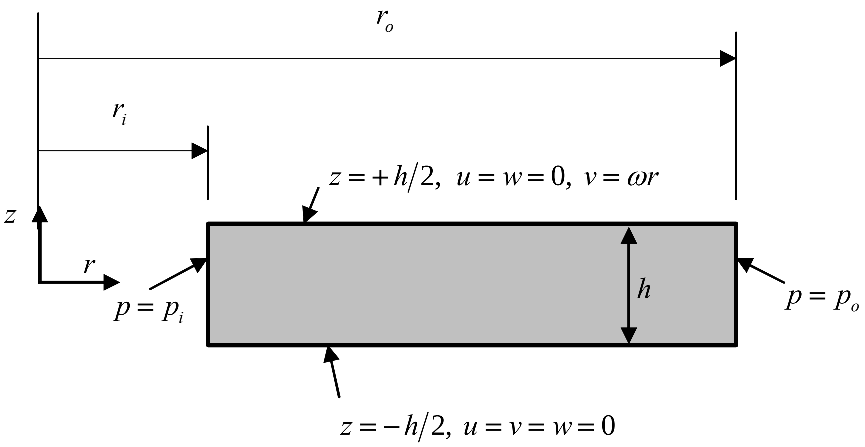

Fig. 2. Solution domain and boundary conditions.

The velocity conditions at the lower boundary ($z^*=-1/2$) are $u^*=0$, $v^*=0$ and $w^*=0$, and at the upper boundary ($z^*=+1/2$), they are $u^*=0$, $v^*=1$ and $w^*=0$. Pressure difference between the inlet ($r^*=r^*_i=r_i/r_m$) and the outlet ($r^*=r^*_o=r_o/r_m$) is assumed to be Δp^*. The solution of the above set of equations and boundary conditions for u^* is given by [21]:

$$u^*=\frac{\Delta p^*}{\Delta r^*}\left(\frac{z^{*2}}{2}-\frac{1}{8}\right)-\frac{z^{*2}r^*}{2}\left(\frac{z^{*2}}{6}-\frac{z^*}{3}+\frac{1}{4}\right)+\frac{4z^*r^*}{96}+\frac{7r^*}{192} \tag{5}$$

which when integrated between $z^*=-1/2$ and $z^*=1/2$ gives the flow rate as

$$Q^*=\frac{Q}{u_s h r_m}=\frac{\pi}{\ln\left(\dfrac{2+\Delta r/r_m}{2-\Delta r/r_m}\right)}\left(\frac{\Delta r/r_m}{20}-\frac{\Delta p}{6\rho v_s^2}\right) \tag{6}$$

Noting that $u_s=\omega^2 r_m h^2\rho/\mu$, and $v_s=\omega^2 r_m{}^2$, equation (6) may be written as:

$$Q=\frac{\pi h^3/\mu}{\ln\left(\dfrac{2+\Delta r/r_m}{2-\Delta r/r_m}\right)}\left[\frac{\Delta r/r_m}{20}\omega^2 r_m^2\rho-\frac{1}{6}\Delta p\right] \tag{7}$$

Equation (7) shows that the net flow in the pump results from a positive centrifugal effect, and a negative pressure effect. The presence of μ in the denominator of the expression in equation (7) may lead to the conclusion that Q increases with decreasing μ. However, there is a limit to this increase since the lubrication model becomes inapplicable when the viscous force is small in relation to the inertia or centrifugal force.

When considering the effect of scaling on the flow rate, we note that h is typically not scaled down with the same factor applied to r_m. For example, if r_m is scaled down by a factor of 1000, it may be sufficient to scale down h by a factor of 10. In this case, the zero pressure flow rate scales down by a factor of 1×10^9 assuming the same ω is used in the original and the scaled models. However, one has to consider the possibility of obtaining extremely high angular velocities in microscale models particularly when multiple gear reduction is used. In geared surface micromachined devices values of

ω above 1×10^6 rad/s are common [17]. Thus, a magnification scale factor of up to 1000 may be assumed for ω. Applying this to our example, the zero pressure flow rate drops only by a factor of 1×10^3.

3. Scaled Up Prototype Setup and Characterization

A scaled up prototype of the flat disk pump concept was prepared using precision machining. As shown in Figure 3, the pump prototype consists of a flat disk spinning with its shaft in housing and covered with a transparent flat plate. The disks are machined from an aluminum stock, with 12 mm flat disk diameter, 5 mm disk thickness, and 4 mm shaft diameter. The inlet to the pump is created at the center of the flat plate cover with 1.35 mm diameter and the outlet is located on the outer periphery of the pump housing. Pump housing is made from Teflon, where O-ring groove, bolts borings, and a cylindrical protrusion with the dimensions of the flat disk diameter and 5.4 mm height, at the upper face, and with the dimensions of the bearing, 8 mm diameter and 3 mm height, at the back face of the housing were machined. The inner diameter of the bearing is equal to the diameter of the flat disk shaft. The flat plate cover was prepared from Plexiglas to allow the visualization of the flow and ensures that there are no air bubbles during testing. The flat spinning disk is adjusted at the pump housing in close proximity with the flat plate cover creating a cylindrical fluid chamber. The height of the pump chamber is the distance between the flat plate cover and the spinning flat disk, where a 400 μm height is maintained by fixing the flat disk shaft to the motor through adjusting the coupling position.

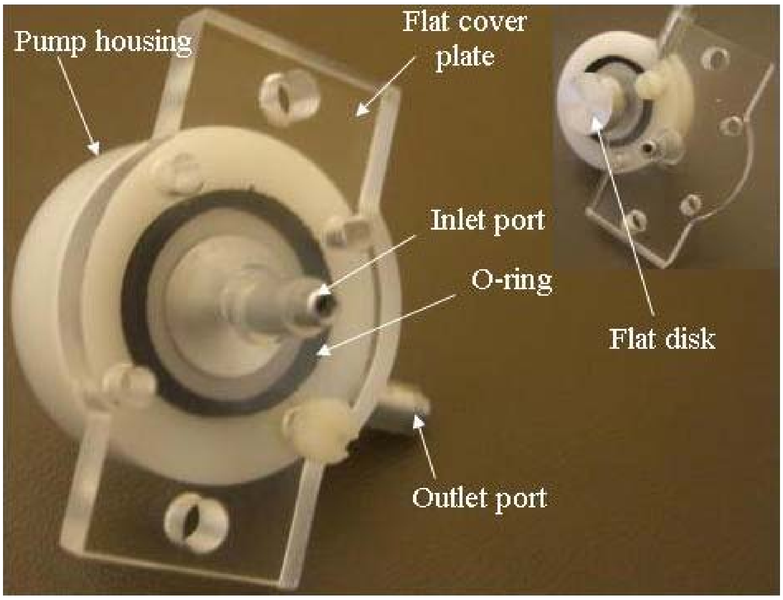

Fig. 3. The scaled up prototype of the flat disk pump and its components.

A schematic illustration for the prototype pump test setup is shown in Figure 4. Fluid enters at the center of the flat cover plate and leaves tangentially to the pump housing from the outlet port. The flat disk shaft is coupled with a Faulhaber 3257G 012CR DC micromotor, and guided in a Teflon sealed bearing, which also forms a seal to reduce the leakage. The pressure difference between the inlet and the outlet is measured using a SCX30 DN precision compensated pressure sensor, which was calibrated with the pressure of a water column.

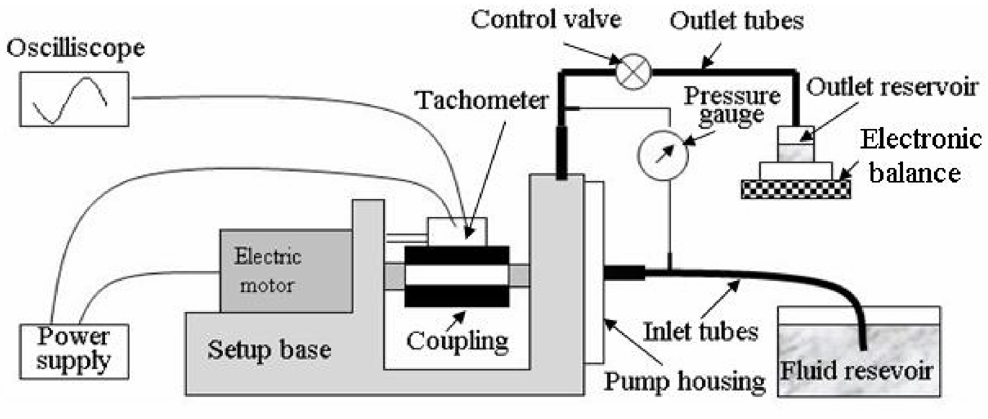

Fig. 4. The flat disk pump experimental setup.

The range of speed investigated in this study is between 1000 and 4500 rpm. The rotational speed is measured by attaching a tachometer to the coupling between the motor and the disk shaft, and the readings were reported through the Oscilloscope signals. Glycerin, the high viscous fluid with a viscosity of 1412 mPa.s, and density of 1261 kg/m3 was used as the working fluid.

The inlet and outlet tubes are 240 mm long and have an inner diameter of 4.5 mm. The inlet tube is connected to the fluid reservoir, which is large enough to avoid level changes during operation, and the outlet tube is connected to the collection reservoir, which is set above digital electronic balance used to read the mass of the pumped fluid during fixed intervals of time. The tube diameter is larger than that of the inlet and outlet ports, because an expansion of the fluid in the tubes will occur, which yields reduction in the flow velocity and the major losses through the test loop.

The assembly of the experimental setup begins by fixing the bearing to the back face of the housing, and setting the spiral disk with it is shaft into the housing. The external motor is fixed to the base, and the coupling is connected freely to the motor shaft. The pump is then fixed to the coupling and adjusted according to the required fluid gap to the motor, fixed to the base, and the inlet and outlet tubes are connected. Before turning on the power supply and adjusting the tachometer, the air is bled from the test loop, and then the motor is activated to the desired speed. The readings are taken after ensuring steady state flow conditions, where the pumped fluid is collected for fixed interval of times and the flow rate is estimated. While, for the maximum pressure difference values, the digital pressure sensor is mounted such as the pressure difference values at steady state conditions across the pump are measured directly after closing a ball valve completely at the far end of the outlet tube, and recording the maximum reachable value. Fluid leakage is noticed to increase from the pump while reaching the maximum pressure values.

The performance of the flat disk pump was investigated experimentally by plotting the flow rates against the disk rotational speed at zero pressure head. Figure 5 shows the analytical and experimental results. The fluid viscosity used in the analytical plot is estimated at the average temperature, $T_{av} = (T_i + T_o)/2$, where T_i, T_o are temperatures measured at the inlet and outlet of the pump respectively. Experimental flow rates exceed the analytical estimation in most of the angular speed range. This can be related to the effect of the side walls of the rotating cylinder that produce Couette similar flow pumping effect, and to the increase on the pump temperature due to friction and viscous heating effects at higher speeds, where lower viscosities and higher flow rates are yield. These factors were neglected at the analytical model.

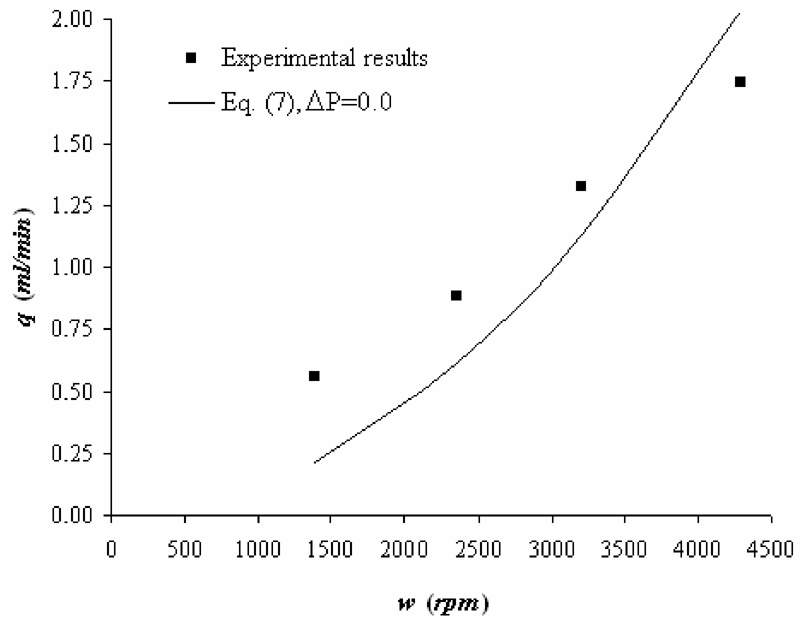

Fig. 5. Flow rate vs. rotational speeds.

The maximum pressure rises for the flat disk pump are also plotted at different rotational speeds ranging from 1000 to 4500 rpm as shown in Figure 6. The results were obtained by fully closing the control valve at the outlet tube. A maximum pressure rise of 196 mbar is achieved at a rotational speed of 4285 rpm.

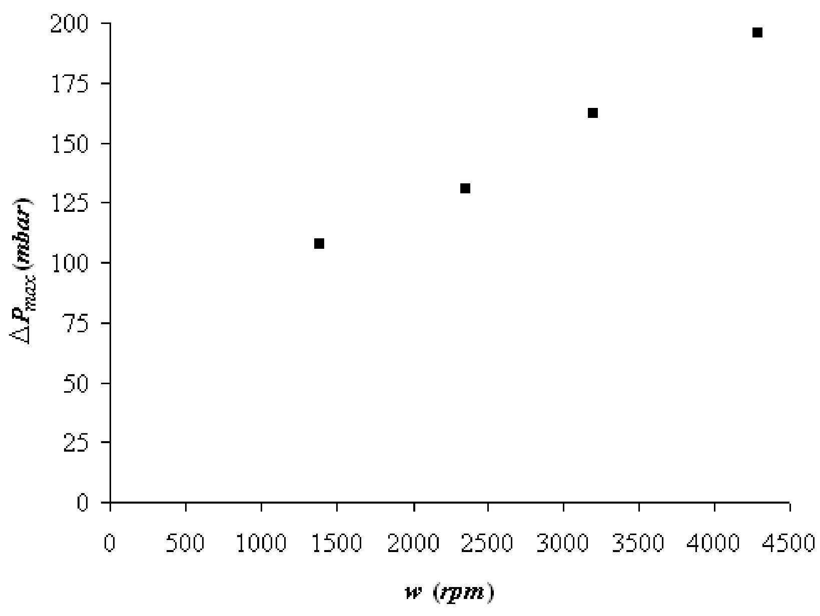

Fig. 6. Maximum pressure rises vs. rotational speeds.

4. Micropump Implementation in SUMMiT

The flat disk micropump was implemented in five levels of polysilicon using Sandia's Ultraplanar Multilevel MEMS Technology SUMMiT. Figure 7 shows a schematic of the pump concept and illustrated the adapted sealing and driving schemes. A 700 µm flat disk is housed in a closed chamber above a fixed plate with a 5 µm gap between the disk and the plate. The disk is driven mechanically by an electrostatic torsional ratcheting actuator (TRA) [22] through a torque amplification gear train. The flat disk has gear tooth on its periphery, which mesh with matching teeth on the output gear of the gear train. Four rollers around the periphery of the disk constrain its in-plane motion and provide the necessary rotational freedom. Figure 8 shows one fabricated micropump and Figure 9 shows a cross sectional view through the center of the flat disk illustrating the interconnection between the different polysilicon levels in the pumping chamber. The bulk of the flat disk is defined in the third polysilicon level (poly 3) and fluid inlet is created below the flat disk by a 100-micron inlet hole etched through the backside of the wafer using a Bosch etching process [23].

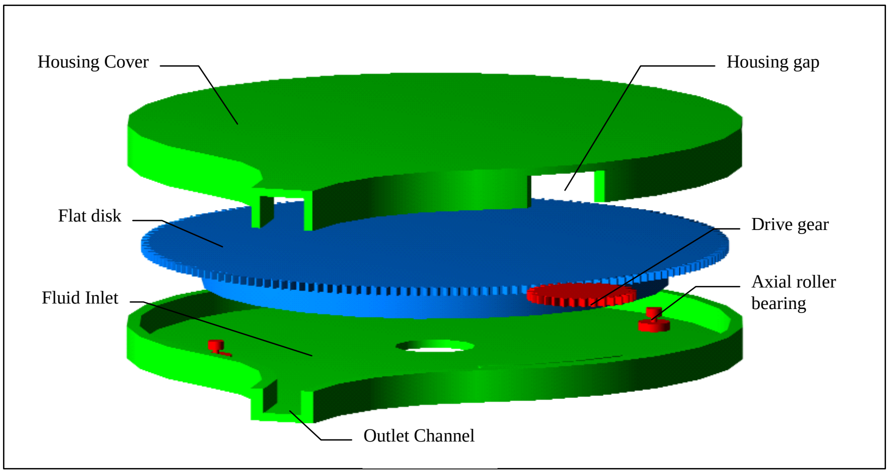

Fig. 7. A schematic of the flat disk micropump showing driving and sealing schemes.

As seen in Figure 7, sealing of the micropump is achieved by enclosing the flat disk with a housing shell, penetrated only by the driving gear through a small gap in the housing wall, allowing the flat disk to mesh with the driving gear through the teeth on its periphery. The pumping chamber is connected to an outlet channel with fluid outlet created at its end using the same process.

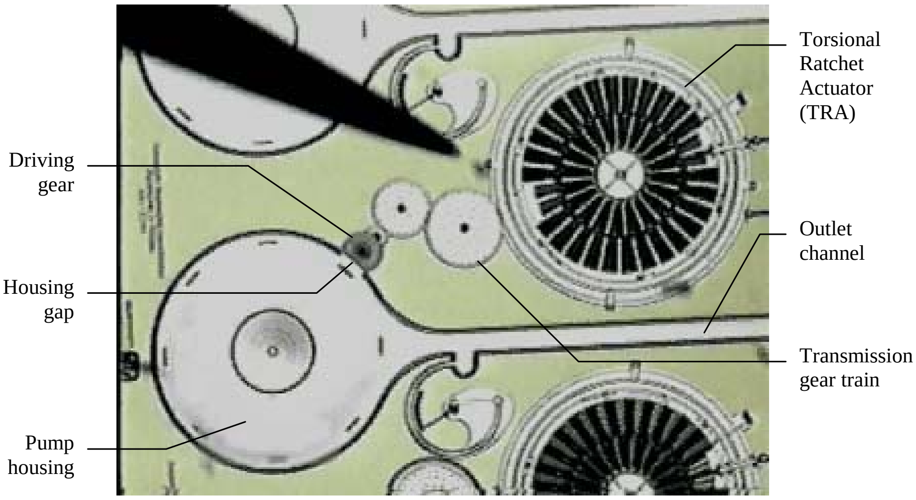

Fig. 8. An example flat disk micropump fabricated in SUMMiT.

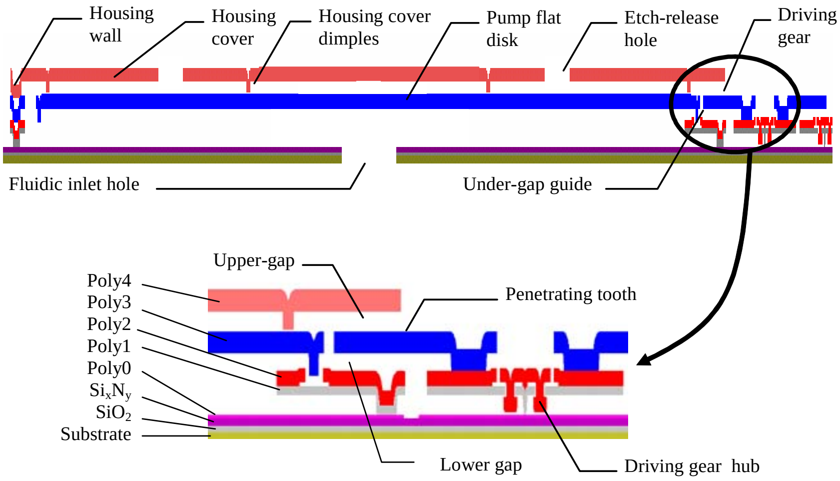

Fig. 9. A cross section through the flat disk pump centerline.

Dimples below the first, second and third polysilicon levels prevent the rotating disk from wobbling and limit its in-use adhesion to the polysilicon levels above. The top cover of the housing is defined in the upper silicon level (poly 4), and is anchored down to the ground poly0 level through a wall in the in – between levels. A number of etch-release holes on the top cover serve as outlets for the sacrificial oxide during the final release process of the device. The close-up view in Figure 9 shows the area where the output gear penetrates the housing shell and an additional sealing arrangement is constructed below the housing gap, where an under-gap guide is created in Poly1/Poly2. The guide is anchored to Poly0 and is penetrated by a Poly3 dimple, and provides a mirror of the top surface seal below the bottom face of the poly3 drive gear.

5. Micropump Testing

5.1. Filling and Pumping

Pump and channel filling was accomplished using capillary forces and moderate pressure applied to a syringe feeding the pump inlet hole through capillary tubes. The pumping chamber was filled with de-ionized water by exerting pressure through the capillaries connected to the pump inlet hole using a syringe. As the pressure was increased, a water meniscus appeared near the housing gap above the drive gear as seen in Figure 10. No leakage took place out of this gap at the applied pressure as surface tension forces are effectively sealing the gap. The effect of motion of the drive gear on the liquid in the gap and any leak that might be developed at this gap as a result were investigated for very slow actuation speed to observe the location of the liquid meniscus on top of the drive gear. The undisturbed location of the meniscus shown in Fig. 10 is approximately 5 microns from the edge of the cover. The drive gear was then turned manually using the probe tip. The meniscus was pulled out from under the cover along the top of the gear as it rotated. As the liquid is pulled out its contact angle drops below the equilibrium contact angle and a negative pressure develops in the liquid (vacuum) as a result. This vacuum tends to pull the liquid back under the cover.

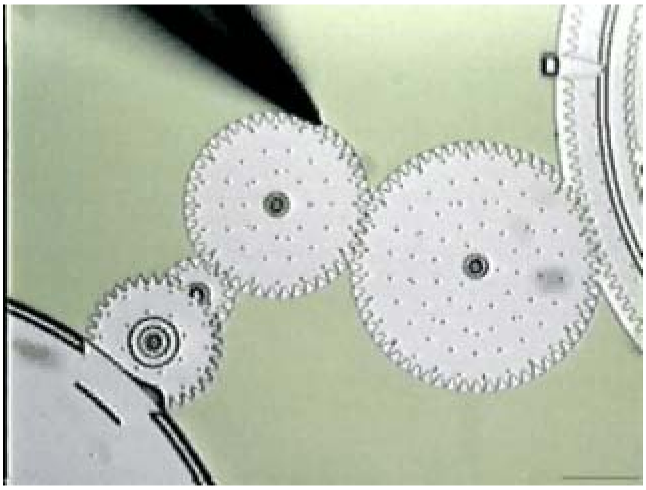

Fig. 10. Water meniscus appearing above the driving gear near the housing gap.

We then began experimenting with various drive signals. At 2 Hz we were able to actuate the pump using a 50 V quarter sine wave signal. The frequency was then increased to 30 Hz, and the device operated smoothly with a 60 V drive signal and we did not notice any leakage, indicating that the liquid was contained under the cover by surface tension at these rotation rates. We observed the device under the microscope the entire time, and a water drop started to develop at the output capillary after approximately 3 minutes of pump operation at 30 Hz. The size of the drop increases as the pump continues to operate. If the pump is stopped, the drop would retract, but would still be visible as seen in Figure 11. While observing the device under the microscope, we also noticed tooth skipping during pump operation. In tooth skipping, the tooth of the driving gear would deflect elastically, go over the tooth of the driven gear and then return to its original position. Tooth skipping occurred frequently and is due to the large torque required to rotate the disk of the pump, which is caused by surface tension and viscous effects. Due to tooth skipping it was not possible to accurately relate the angular frequency of the driving signal to the actual rotational speed of the flat disk on the pump.

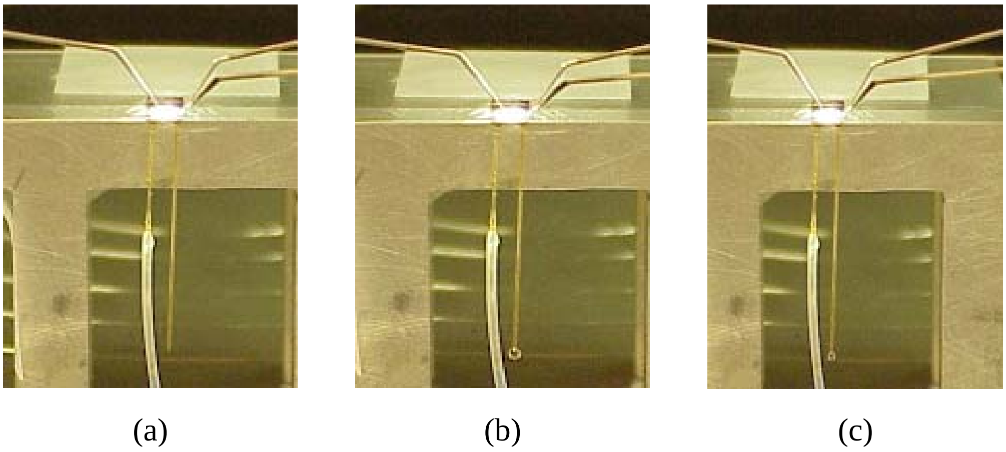

Fig. 11. Outlet capillary during 30Hz pumping. (a) before actuation (b) five minutes after actuation, pump running (c) five minutes after actuation, pump stopped.

5.2. Pump Leakage

To investigate leakage from the pump, we purposely exerted pressure on the syringe connected to pump inlet to force fluid out of the pump chamber. As the pressure is increased beyond the 2 Atm water is pushed further out on the top of the drive gear until the contact angle allows water to extend from the top of the housing cover (rather than the bottom of the pump cover where the gap is) to the drive gear. Additional pressure drives water above the top surface of the cover and it spreads over the highly wetting top cover surface as illustrated in Figure 12. The pool of water on the top cover then gradually evaporates and liquid is drawn back into the pump if the pressure is removed. The water does not spread over the surface of the chip unless significantly more liquid is pushed though the syringe. Water did not leak onto other surfaces nor did it leak through the 1 micron square etch release holes on the housing cover. It also did not leak out of the gap below the drive gear as the Poly1/Poly2 guide under the gap effectively cuts off this avenue for water to escape by providing a mirror of the top surface tension seal. There is no leak out of this gap at low applied pressure as surface tension forces are effectively sealing the gap. The water is confined to the hydrophilic polysilicon cover with native oxide even after leaking. In summary, the pump has a surface tension seal at the drive gear and no significant leaks at any other locations. This seal holds water under the pump cover and allows the pump gear to be actuated at pressures up to approximately 2 Atm gage pressure.

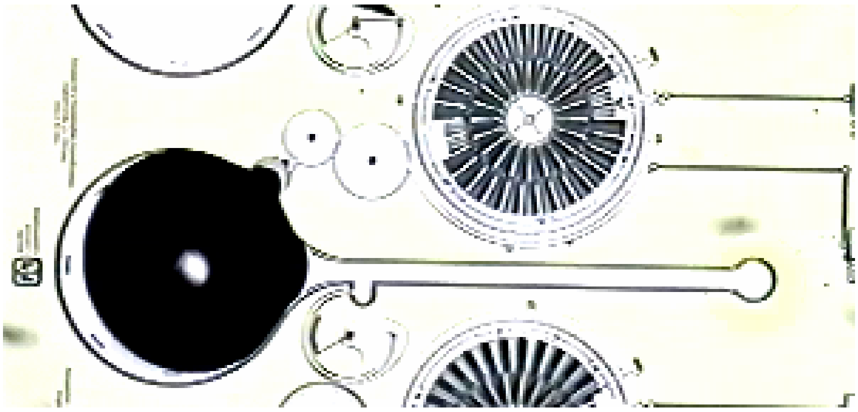

Fig. 12. Water pool forming on the top surface of the housing cover at a pressure beyond 2 Atm.

5.3 Failure mechanisms – rocking, rotating comb stiction, pawl fracture

A quarter sine wave, 50 % duty cycle drive signal was found to be most effective in actuating the pump. A 50 V peak amplitude quarter sine signal was required to overcome static friction and actuate the pump in air. At 50 V a force of 10-20 μN is delivered by the TRA, and this accounts for the static friction force in the gear transmission and pump. Locations of surfaces that might be touching and therefore contribute to static friction are (1) dimple 4 just above the rotating poly 3 disk – contact possible even though the surfaces are no longer attached, (2) dimple3 just below the rotating poly3 disk that rides just above the poly2 guides at the location where the gear transmission meshes with the pump gear, and (3) the pump gear hub.

We were able to actuate the flat disk at frequencies of up to approximately 500 Hz using the quarter sine wave 50 V signal applied to the TRA. At higher frequencies we were not able to actuate the loaded TRA for more than a few ratchet steps. Higher voltage signals resulted in higher force smoother actuation (fewer missed ratchet steps). However at voltages of 80 to 100 V there was significant bending of the pawl leading to pawl failure (fracture). Therefore the dry operating range of the pump driven by the TRA was from approximately 50 to 100 V and from 1 to 300 Hz.

The most common failure mechanism observed in wet-running was tooth skipping which caused a rocking motion in which the ratchet pawl pushed the TRA ring gear during voltage application and then the ring gear rocked back as the pawl returned to start another ratchet stroke. As a result the pawl pushed the same ring gear tooth when it actuated again and there was no ratchet precession. The entire mechanism rocked back and forth without turning the pump. Redesign of the TRA so that the backlash stop engages sooner should also reduce this problem. Backlash was evident in the TRA ring gear and in the gears immediately attached to it. During backlash the pump ring gear did not appear to move either forward or backward implying that the backlash was caused by the inability of the TRA to overcome static friction in the pump and move the pump gear. The energy of the forward stroke was therefore stored as twist in the gear transmission rather than being transmitted to the pump ring gear and the acted to return the TRA ring gear to its original position when the voltage dropped.

These failure modes point to one overriding cause. The TRA actuator used is underpowered for this application. An actuator that delivers more torque and/or a pump that has less friction are required for efficient and reliable pump operation. In addition the alignment of the entire gear train during operation of the TRA should be examined to determine if a redesign will reduce the amount of radial force applied to any of the gears in the gear train, and therefore the possibility of tooth skipping.

6. Conclusions

The paper presented the development of a micropump concept which works by the continuous rotation of a flat disk above a stationary plate, with pumped fluid sandwiched between the rotating disk and a stationary plate. The pump utilizes the centrifugal and viscous effects and it was fabricated in five-level of silicon using SUMMiT, where a comb-drive torsional ratcheting actuator provides the power needed to drive the flat disk of the pump in a sealed housing through a geared transmission. The paper reviewed available analytical models for flows in similar geometries and showed that the concept is feasible at the microscale if high rotational speeds are provided. Results from a macroscale prototype verified the analytical predictions. A SUMMiT implementation of the pump illustrating the driving, feeding, sealing schemes devised for this type of pump were presented and a set of experiments were used to characterize its performance and possible failure modes.

The flat disk pumping principle is a simple pumping approach which employs both the viscous and the centrifugal effects generated on a fluid layer sandwiched between a rotating disk and a stationary plate. The resulting planar device is simple and does not require close contact between mating elements as in other viscous pump designs. Sealing of the housing gap is achieved through surface tension and appears satisfactory for low to intermediate pressure applications. Experimental results demonstrate that the pump could be filled and it could pump liquid through its outlet channel.

The flat disk micropump represents an elementary first step in the effort toward achieving a reliable CMOS compatible surface micromachined pumps. Future work on this concept may address utilizing an actuator that delivers more torque and/or a pump that has less torque to operate. In addition the alignment of the entire gear train during operation of the TRA should be examined to determine if a redesign will reduce the possibility of tooth skipping. On the analytical front research efforts should investigate the influence of pump geometry (disk diameter, inlet diameter, and fluid layer height) and operating conditions (disk speed and working fluid) on pump performance (flow rate and pressure head).

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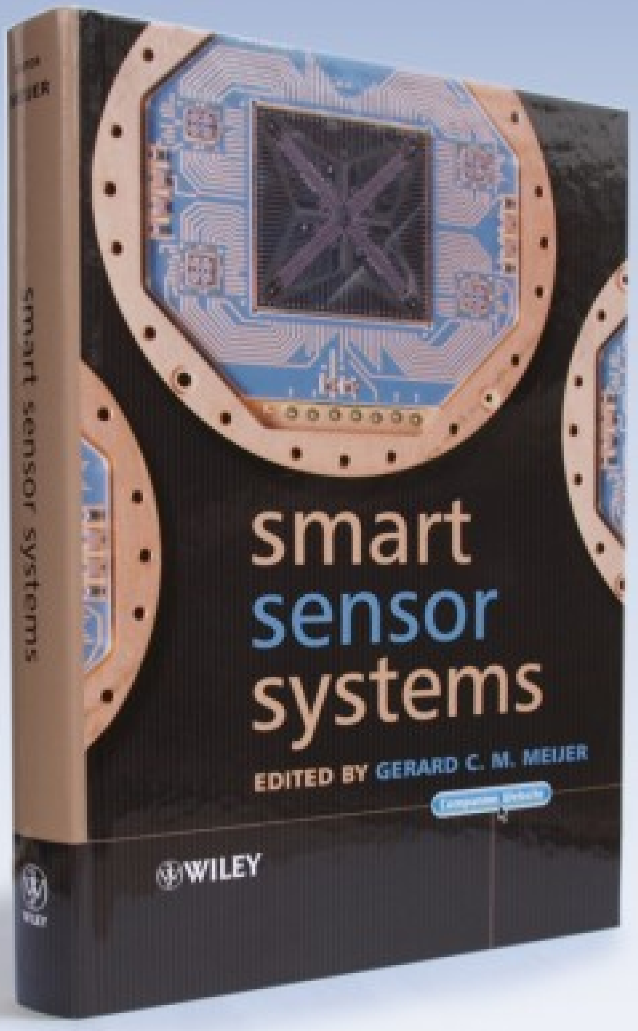
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