Apex Seals for a MEMS Rotary Engine

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Abstract

This paper presents possible solutions to the problem of leakage in the MEMS rotary engine currently being developed at U.C. Berkeley. The project is motivated by the challenge to develop a system capable of a higher specific energy than primary batteries. Currently, leakage is the limiting factor in obtaining an engine with an acceptable efficiency. Fluidic analysis has been done to determine a necessary distance of 2 microns between the rotor and the housing of the engine, remembering that the compression ratio only needs to be maintained over the characteristic time, 2 msec for a shaft speed of 40,000 rpm. Mechanical analysis has also been completed to ensure that the apex seals will maintain this minimum distance by integrating a spring-loaded apex design. Test structures will also verify this analysis.

I. Introduction

Micro internal combustion rotary engines are a potential replacement for batteries in the near future. This comes from the energy density advantage hydrocarbons hold over the current alkaline batteries available, which happens to be 10:1 for an engine with 20% efficiency [1]. A limiting factor, in the development of an internal combustion engine with this efficiency is engine sealing. Without adequate engine sealing within the engine the compression ratio inside the rotary engine plummets causing the efficiency of the engine to do the same [2].

Otto Cycle: $$\eta_{th,Otto} = 1 - \frac{1}{r_v^{k-1}} \quad (1)$$

To ensure that the compression ratio is adequate, it is necessary to implement an engine sealing system capable of high compression ratios which minimizes leakage.

Engine sealing has been accomplished in the past. Mazda achieved sealing with an elaborate system, which consisted of approximately thirty parts, for their macro scale rotary engine [3]. However, it has been shown that the primary source of leakage in small-scale rotary engines is past the apex seal [2]. This is convenient, since it is necessary to keep systems as simple as possible due to the limits of MEMS fabrication.

The design of the apex seals for the micro engine will address the following issues: structural stiffness, engine fabrication tolerance, leakage rate, and assembly. Another concern is the tolerances of the rotor with respect to the housing (i.e. aspect ratios of both). A determination of the travel of the apex seal will need to be made to account for this tolerance. Fluidic analysis will be performed to determine the necessary gap distance needed to increase engine sealing. Establishing this distance will help prevent additional wear on the apex from over design. Finally, the design of the apex seal will need to be simple with no required assembly.

Fabrication

The process to be used for this design will be a single mask Deep Reactive Ion Etch (DRIE). The structural layer will be masked and then the rotor will be etched down 300 microns. The minimum line width is 1 µm. The minimum line spacing is >12 µm due to an aspect ratio of 50:1. The aspect ratio will be necessary to determine if the incline of the housing wall matches the incline of the rotor.

II. Design

The apex seal flexure designs being considered are shown in Figure 1. Of these, the single flexure (Figure 1a) has the simplest geometry, while the folded flexure (Figure 1b) sacrifices some simplicity for design flexibility. The third flexure type incorporates a bistable centrally-clamped parallel-beam adopted from the research of Qiu, Lang, and Slocum of Massachusetts Institute of Technology [4]. A bistable flexure is advantageous because it can potentially benefit the assembly process.

Because the geometry of the apex's tip is constant for the three proposed designs, the fluid analysis will also be constant. In contrast, the mechanical analysis of the apex's flexures will be unique for each design.

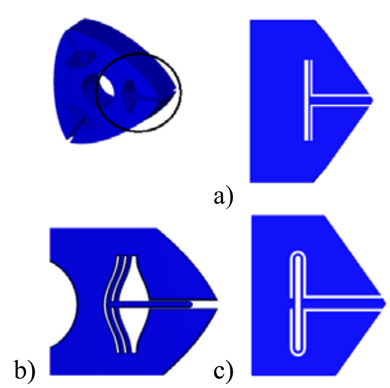

Figure 1: Rotary engine rotor with details for apex seal flexure designs: (a) single flexure, (b) bistable flexure, and (c) folded flexure.

Mechanical Analysis
In a rotary engine, the apex seals maintain contact with the walls of the combustion chamber at all times. To determine this dynamic displacement of the apex tips, the housing is measured to determine the fabrication tolerance. Inaccurate fabrication will cause a variance between the designed housing profile and the fabricated profile. Figure 2 demonstrates a comparison of the mask layout and

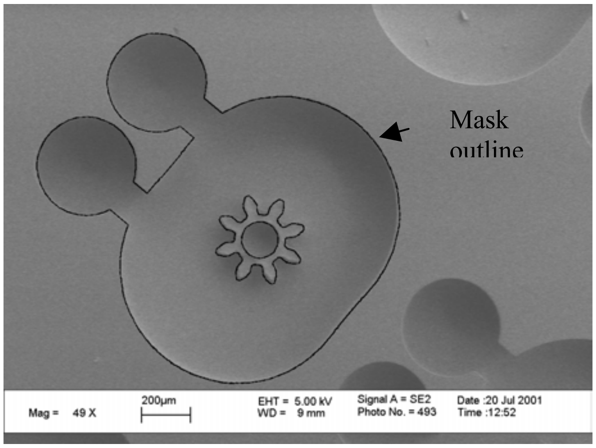

Figure 2: Analysis of fabrication tolerances for the engine housing.

fabricated engine parts. From this analysis, the required travel of the apex is measured to be less than 10 µm.

Friction due to the apex seals will be approximately proportional to the force normal to the engine housing. To estimate the normal force, numerical calculations and finite element models will be made to characterize the proposed designs. The force of the apex seal against the housing wall is defined as

$$F = kx \tag{2}$$

where F is the force, k is the stiffness of the flexure, and x is the deflection.

The spring stiffness for the single flexure and folded flexure designs are calculated using unified beam theory [5]. For the single flexure design:

$$k = -\frac{192EI}{l^3} \tag{3}$$

For the folded flexure design:

$$k = EI\left[\frac{l_1^{\,3} + l_3^{\,3}}{6} - \frac{\left(l_1^{\,2} + l_3^{\,2}\right)^2}{4\left(l_1 + l_2 + l_3\right)}\right]^{-1} \tag{4}$$

where E is the Young's modulus, l_1 & l_3 are the long flexures, l_2 is the short, connecting flexure (Figure 3), and I is the moment of inertia of the flexure.

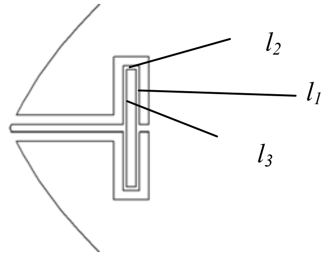

Figure 3: Folded flexure design showing correlating lengths, l_1, l_2, *and* l_3.

The moment of inertia is calculated by

$$I = \frac{b}{12}\left[h^3 + \frac{3h^2b}{s} + \frac{4hb^2}{s^2} + \frac{2b^3}{s^3}\right] \tag{5}$$

where b is the width of the flexure, h is the height, and s is the aspect ratio of the fabrication process.

The stiffness of the bistable flexure will be calculated through finite element modeling. The single and folded flexure stiffnesses will be verified by the same method of finite element analysis.

Fluidic Analysis
The fluidic analysis of the engine is complex. The Reynolds number, the ratio between inertia and surface effects, for the system is small (i.e. ~1), thus surface effects will play a larger role than it does in macro-scale applications. This turns out to actually be a helpful phenomenon since the increased effect of viscous forces will tend to decrease the leakage rate. In addition, due to a small distance (1-10 microns depending on the design) between the rotor and the housing, the analysis will lie within the boundary layer.

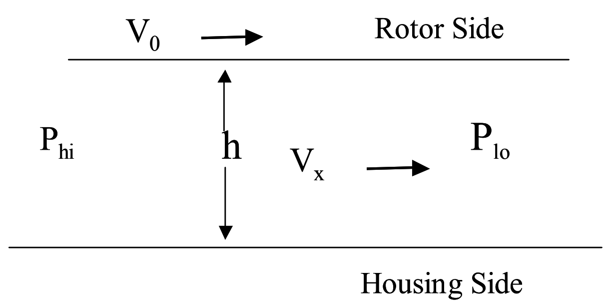

Figure 4: Poiseuille-Couette model used for fluidic analysis of the flow past the apexes.

As a first order approximation, a combined Poiseuille-Couette flow analysis was completed. The schematic for the model is shown in Figure 4. The assumptions made for such analysis included two-dimensional flow, steady state, incompressibility, and fully developed, viscous flow. Error will most likely come from assuming the flow is steady but the results should provide a worst-case scenario. The momentum equation for such a flow is given by

$$\frac{1}{\mu}\frac{dP}{dx} = \frac{d^2V_x}{dy^2} \qquad (6)$$

assuming $V_x(0) = 0$ and $V_x(h) = V_0$ [6]. Solving the momentum equation for the velocity profile, V_x, in terms of the gap distance, h, the viscosity, μ, and pressure difference, dP, the width of the apex, dx, and the speed of the apex, V_0,

$$V_x = \frac{1}{2\mu}\frac{\partial P}{\partial x}(y^2 - yh) + V_0\frac{y}{h} \qquad (7)$$

The volumetric flow rate, Q, for system is given by

$$Q = \int_0^h V_x \cdot l dy = l\left(\frac{V_0 h}{2} - \frac{h^3}{12\mu}\frac{\partial P}{\partial x}\right) \qquad (8)$$

where l is the thickness of the rotor.

The first term in equation (8) gives the volumetric flow rate due the motion of the apex seal and the second term defines the leakage that would occur it there was no motion within the system.

Another significant value to consider is the compression ratio of the engine after 2 ms, which is the time required to complete a third of the cycle for an engine rotating at 40,000 rpm. To determine the compression ratio after 2 msec, the fluid is assumed to be an ideal gas, which is warranted since the fluid is mostly air. Thus, the ideal gas law was implemented as

$$P_{i+1} = \frac{P_i V_{i+1}}{V_i} = \frac{P_i\left(V_i - \frac{h^3}{12\mu}\frac{\Delta P}{\Delta x}t\right)}{V_i} \qquad (9)$$

where P_i and V_i are the pressure and time at time interval i respectively.

This characteristic time of 2 ms is important since the higher the compression ratio is after this allotted time the higher the engine's efficiency will be. As can be seen from the above equation, the pressure will depend heavily on the gap size. Thus, apex seal designs capable of decreasing this gap between the rotor and the housing will be necessary.

There is one final phenomenon to analyze when modeling this system, apex liftoff. Apex liftoff is due to the same situation which occurs in a computer's hard drive: the drive begins to rotate so fast that the reader actually begins to liftoff due to the boundary layer flowing over the hard drive. An apex seal may do the same thing since it is rotating at high speeds as well; however, the length scale is different. Initial calculations suggest the engine speeds are too low for such liftoff to occur. Further reference to liftoff may be found in [7].

III. Test structures

A set of test structures has been conceived to analyze the quality and performance of the fabrication process and proposed design.

Inconsistencies are expected between fabrication process runs, as well as across the surface of the wafer due to a variety of factors: changing machine performance, cross-wafer variation in Si etches, etc. To characterize each process run, structures of varying width and line spacing are used to determine the minimum line width and spacing (Figure 5). From a cleaved section of these test structures, the aspect ratio can also be measured. These process measurements are made on a scanning electron microscope.

A layout of an N x M matrix of rotors will be used to test N flexure stiffnesses and M initial displacements (preloading) of the apexes. These variables will be used to characterize the individual designs. The goal is to find the optimal design for assembly and performance.

To characterize the drag (fluidic and frictional) of the system on the rotors a test structure is devised. This can be done by placing the rotor in a cylinder which has ports which air can be blown through as in Figure 5. The drag can then be characterized by obtaining measurements of inlet pressure, flow rates, and rotor speed.

A final test structure is made to verify the effect of gap distance on the compression ratio. This structure is implemented by placing a rotor in a housing equipped with pressure transducers in the wall of the housing. Pressurizing the combustion chamber and measuring the pressure at the two locations allows one to obtain the compression in the engine as a function of time.

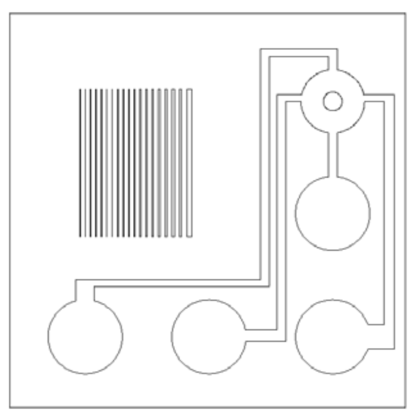

Figure 5: Test structures to characterize aspect ratio (left) and drag (right)

IV. Expected Results

Once the apex seal design is implemented, it is expected that increased compression ratio will depend on how well the apex seal can decrease the gap between the apex and the housing. Using the equations set forth in the previous section concerning fluidic analysis, the time required to reach a compression ratio of 4:1 is listed in Table 1. The time depends on the fluid used which is due to the difference of the fluid viscosity for oil and air. These two fluids will be part of the fuel mixture used and should also give the range for the fluid viscosity of the system. Assuming a characteristic time of 2 msec, the maximum allowable distance to achieve adequate sealing with air is 2 μm. If the fluid is closer to oil the necessary gap is between 5 and 10 μm. This is due to the high viscosity of oil, decreasing the likelihood of the fluid escaping.

h (μm)	For Air (sec.)	For Oil (sec.)
10	0.0000219	0.00121
5	0.000175	0.0097
2	0.00272	0.152
1	0.0219	1.21
0.5	0.175	9.7

Table 1: Expected time for system using oil or air to reach a compression ratio of 4:1

Calculations have been made to determine typical spring stiffnesses of the single and folded flexure designs (Table 2). The values for each individual rotor would be calculated to characterize the matrix of parts.

Young's Modulus, E	160 GPa
h	300 μm
b	5 μm
s	50:1
l for single flexure	200 μm
k_{single}	4.3 10^7N/m
$l_1 = l_3$ for folded flexure	82 μm
l_2 for folded flexure	13 μm
$k_{folded\ flexure}$	2.5 10^7N/m

Table 2: Typical stiffness calculations for single and folded flexure designs

V. Conclusion

The limits of conventional silicon fabrication techniques will be tested to determine the maximum potential of a MEMS rotary engine. Tolerances are critical to engine design due to the need to maintain the high pressures necessary for the operation of the internal combustion engine. Efficiency suffers without gas tight sealing of the combustion chambers.

Sealing in a rotary engine has been a critical issue on the macro [3] and the mini [2] scales. The same challenge is expected on the micro scale. MEMS fabrication capabilities will limit the use of conventional techniques for minimizing engine leakage, but scaling effects may aid in sealing. On the micro-scale, the effects of fluid viscosity dominate and could therefore reduce leakage.

Spring-loaded seals at the apexes of the rotor have been determined to significantly improve engine performance on the mini scale [2]. If apex sealing has the same impact on a MEMS rotary engine, a simpler design requiring little or no assembly will be used. The integrated flexure designs introduced here require no assembly, while the bistable design makes assembly of the rotor and housing easier.

If sealing is not improved by the sealing system implemented it may be caused by one of two things. The first is the apex seals may need to be stiffer than originally thought due to deviations not realized in the theoretical analysis presented. The second issue which may be that a system utilizing a more viscous fluid may be necessary if meeting the necessary gap distances are not possible due to process deficiencies. However, if sealing is improved then the ability to create a rotary engine on the MEMS scale is greatly increased and MEMS rotary engines may possibly be an alternative to current batteries.

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