Abstract—As part of a program to develop a micro gas turbine engine capable of producing 10–50 W of electrical power in a package less than one cubic centimeter in volume, we present the design, fabrication, packaging, and experimental test results for the 6-wafer combustion system for a silicon microengine. Comprising the main nonrotating functional components of the engine, the device described herein measures 2.1 cm × 2.1 cm × 0.38 cm and is largely fabricated by deep reactive ion etching through a total thickness of 3800 µm. Complete with a set of fuel plenums, pressure ports, fuel injectors, igniters, fluidic interconnects, and compressor and turbine static airfoils, this structure is the first demonstration of the complete hot flow path of a multilevel micro gas turbine engine. The 0.195 cm³ combustion chamber is shown to sustain a stable hydrogen flame over a range of operating mass flows and fuel–air mixture ratios and to produce exit gas temperatures in excess of 1600 K. It also serves as the first experimental demonstration of stable hydrocarbon microcombustion within the structural constraints of silicon. Combined with longevity tests at elevated temperatures for tens of hours, these results demonstrate the viability of a silicon-based combustion system for micro heat engine applications.

Index Terms—Microcombustion, micro-gas turbine engine, power MEMS.

I. INTRODUCTION

Recent advances in the field of silicon microfabrication technology have opened the potential for miniature combustion engines for portable power generation and micro air vehicle propulsion applications [1]. As part of a current Massachusetts Institute of Technology (MIT) program to develop such technologies, Epstein et al. [2] and Groshenry [3] have reported the design for a micro gas turbine generator capable of producing 10–50 W of electrical power while consuming 7 g of jet fuel per hour.

A discussion of relevant combustor scaling laws and preliminary assessments of several combustor concepts for these applications were presented by Waitz et al. [4], Stable hydrogen combustion in a microfabricated silicon combustor for this engine was first demonstrated by Mehra and Waitz [5], who showed that it was possible to attain exit gas temperatures in excess of 1800 K without compromising the structural integrity of the 0.066 cm³ combustion chamber. An SEM of this silicon microcombustor is shown in Fig. 1, along with a schematic of the baseline engine configuration.

The device shown in Fig. 1 served as the first demonstration of a high power density microcombustion system (~2000 MW/m³ [6]); efforts were subsequently undertaken to integrate it with the rest of the engine.

This paper reports the results of those integration efforts by presenting the design, fabrication, packaging, and experimental testing of the combustor in a typical engine configuration. Complete with fuel plenums and injection holes, pressure diagnostics, igniters, and compressor and turbine static airfoils, the re-
Fig. 2. Schematic and SEM cross-section of half of the axisymmetric 6-wafer static structure. (Note that the dies on the wafer had two types of combustor inlet holes—one type had slots as shown in Fig. 3, and the other had an annular opening as shown above.)

The design space is further complicated by the addition of a third factor—the material and fabrication constraints. While silicon microfabrication is instrumental to achieving the economy and high tolerances necessary to make a microengine viable, it is still largely limited to rudimentary 3-D geometries. Furthermore, while creep constraints in the rotating components limit wall temperatures to 900 K [7], chemical kinetics demand higher temperatures to achieve stable and efficient combustion. The walls of the combustor therefore have to be cooled below the operating gas temperatures inside the chamber. Since cooling the hot chamber walls can adversely impact the efficiency of the combustor, designing a device that is efficient, yet structurally durable, poses a significant design challenge.

The design of a microcombustor therefore mandates careful tradeoffs between power output, thermodynamic cycle parameters associated with the engine, physical dimensions, and material and manufacturing capabilities. These design considerations are described in the following sections.\(^1\)

The design and operation of such microsystems is also complicated by the difficulty in instrumenting the small experimental rigs. The inapplicability of conventional diagnostics mandates the development of “on-chip” sensors for temperature, pressure, etc.; efforts are currently underway to incorporate these transducers in future devices.

\(^1\)The design and operation of such microsystems is also complicated by the difficulty in instrumenting the small experimental rigs. The inapplicability of conventional diagnostics mandates the development of “on-chip” sensors for temperature, pressure, etc.; efforts are currently underway to incorporate these transducers in future devices.
The volume of the combustion chamber in the static structure was determined by rescaling the volume of the 3-stack microcombustor, which operated at atmospheric pressure, in order to obtain the same residence time at the design operating conditions of the 6-stack [5]. The maximum die size was also limited to 2.1 cm to accommodate at least ten dies on a 4–in wafer. Table I shows the design operating parameters for the static structure and compares them with those for the previously demonstrated 3-stack microcombustor.

### A. Design of the Recirculation Jacket

Previous experimental testing of the 3-stack microcombustor had shown that although ambient heat loss from the structure reduced the combustor efficiency to approximately 70%, it was instrumental to the survival of the silicon; the walls of the combustion chamber operated at hundreds of Kelvin below the combustor gas temperature [5]. In an attempt to improve combustor efficiency without violating the structural integrity of the device, the static structure incorporated a combustor recirculation jacket as shown in Fig. 2. The recirculation jacket was designed to 1) recover the lost energy of the combustion chamber to preheat the incoming reactants, while 2) allowing the compressor discharge air to cool the hot walls of the chamber, enabling the silicon to survive at the high gas temperatures needed for stable combustion. Using computational fluid dynamics (CFD) solutions, the size of the recirculation jacket was set at a maximum possible width of 400 μm to minimize the pressure loss within the duct. In order to support the interior chamber, eight 100-μm-wide bridges were used to connect it with the outer walls. These bridges were also intended to minimize the heat conduction between the combustion chamber and the outer walls of the device. Pictures of these features are shown in Fig. 4.

### B. Fuel Injector Design

As shown in Fig. 2, the static structure was designed with three sets of fuel injectors located at different points along the flow path to evaluate the tradeoff between mixing effectiveness and potential upstream burning in the recirculation jacket. The size and spacing of the injector holes was determined by using semi-empirical models to optimize the penetration and lateral spreading of the jets in order to minimize the streamwise length needed for complete fuel–air mixing [9]. Four of the six dies on the wafer were optimized for hydrogen injection; the remaining six were intended for propane. The final design resulted in two sets of axial arrays located downstream of the compressor vanes at a radius of 4.8 mm and 8 mm, respectively, and one set of ra-

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**Table I**

Operating Parameters for the 3-Stack Microcombustor and the Static Structure (Note that the Flow Residence Times Are Based on an Average Flow Temperature of 1000 K)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>3-stack</th>
<th>Static structure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air flow rate</td>
<td>0.18 gm/sec</td>
<td>0.36 gm/sec</td>
</tr>
<tr>
<td>Operating pressure</td>
<td>4 atm.</td>
<td>3 atm.</td>
</tr>
<tr>
<td>Overall die size</td>
<td>$15 \times 15 \times 1.8$</td>
<td>$21 \times 21 \times 3.8$ mm</td>
</tr>
<tr>
<td>Combustor o.d.</td>
<td>10 mm</td>
<td>18.4 mm</td>
</tr>
<tr>
<td>Combustor l.d.</td>
<td>5 mm</td>
<td>9.6 mm</td>
</tr>
<tr>
<td>Combustor height</td>
<td>1 mm</td>
<td>1 mm</td>
</tr>
<tr>
<td>Combustor volume</td>
<td>66 mm$^3$</td>
<td>190 mm$^3$</td>
</tr>
<tr>
<td>Residence time</td>
<td>0.5 msec</td>
<td>0.5 msec</td>
</tr>
</tbody>
</table>

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1 These calculations were performed using Fluent ver. 4.2. Details of the application of this code can be found in [8].

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Fig. 3. Composite of 35 SEMs of the individual wafers prior to bonding along with an exploded schematic showing the detailed fluid flow path. (Note that this die has slotted openings into the combustion chamber.)
interfacial exchange and mixing as well as heat loss at the wall. Calculation zones is a function of their size and shape, being set by the combustor efficiency. Further, the temperature of the recirculation zones are reservoirs for hot products and serve as the ignition source for the reactants flowing into the combustor. The size and shape of the recirculation zones within the combustor are set in part by the combustor inlet geometry. In designing the recirculation zones, one wants them to be large enough to serve as effective ignition and flame stabilization zones but not so large as to take up most of the combustor volume, thereby reducing the residence time for the reactants and negatively impacting the combustor efficiency. Further, the temperature of the recirculation zones is a function of their size and shape, being set by interfacial exchange and mixing as well as heat loss at the wall.

Two different designs were tested.

1) The first configuration was composed of 60 slots that were 2.2 mm long, had an inner and outer radius of 7 mm and 9.2 mm, respectively, and produced many small recirculation zones at the entrance of the combustor. These would tend to increase combustor efficiency at the expense of stable operating range.

2) The second configuration was a single annular opening that was 1.2 mm wide, producing a single large recirculation zone within the combustor. This configuration would be expected to allow operation over a wide range of conditions but with poorer combustor efficiency.

SEMs of both these configurations are shown in Fig. 6.

IV. Fabrication Process

The fabrication of the static structure required ten deep dry anisotropic etches and two shallow etches. A total of 13 masks were required for the process, including one global alignment mask. The steps are illustrated in Fig. 7, and the process is listed as follows.

A. Photolithography and Deep Reactive Ion Etching

Wafer 1 required a single 400-μm-deep dry anisotropic etch to define the inlet holes to the three fuel plenums, the pressure plenum, and the compressor.

Wafer 2 was first 5 μm shallow etched from the bottom to define the tip clearance for the compressor. (Even though there were no rotating blades in the static structure, the clearance was required to maintain the thermal insulation between the combustion chamber and the outer walls of the device.) Following the shallow etch, the wafer was flipped, and 200-μm-deep fuel and pressure plenums were anisotropically dry etched from the top. Finally, 200-μm-deep axial fuel injector holes were anisotropically dry etched from the bottom.

Wafer 3 comprised the inflow swirl vanes and involved two 400-μm-deep anisotropic etches. The 400-μm compressor blades were first etched from the top side, the wafer was then flipped, and finally, the 400-μm recirculation jacket was etched from the bottom.

Wafer 4 comprised the 1000-μm-deep combustion chamber and the turbine NGVs. First, 5-μm-deep radial fuel injectors were etched on the bottom surface of the wafer. The 400-μm deep NGV blades were then etched from the bottom. Finally, the wafer was flipped, and the remaining 600 μm of the combustion chamber was etched from the top.

Wafer 5 contained the combustor inlet holes, and like wafer 3, also involved two 400-μm-deep anisotropic etches. The combustor inlet slots were first etched on the top side of the wafer. The remaining 400 μm of the recirculation jacket was then etched from the bottom.

Wafer 6 required a single 400-μm etch for the exit hole, the igniter ports, and the aft pressure port.
For each of the patterning steps, 10 μm of positive photoresist was used as a mask (Hoechst AZ 4620); the wafers were patterned using double-sided infra-red alignment. The deep etches utilized a time-multiplexed inductively coupled plasma of SF₆ etchant and C₆F₅ passivating polymer. The performance of the etcher was optimized by characterizing the recipes over a wide parameter space [10], thereby allowing the fabrication of high tolerance and high aspect ratio structures such as the 400-μm-high compressor and turbine airfoils and a 1000-μm-deep combustion chamber supported by 100-μm-wide bridges.

Fig. 8 shows pictures of the device along different axial planes.

B. Aligned Fusion Bonding

Following the completion of individual processing, the wafers were aligned fusion bonded. The wafers were RCA-cleaned first [11], then aligned bonded, and finally post annealed in an 1100°C furnace for 1 hr.

Given the complexity of bonding six patterned wafers with thicknesses varying between 400 μm and 1000 μm and a total stack thickness of 3800 μm, this step posed a significant challenge. In fact, the first two builds of the static structure suffered from poor bonding yield and produced dies that leaked along the bondlines upon pressurization.

This low bonding yield was traced to surface contamination resulting from passivating fluorocarbon deposits on the bottom surface of the wafers during through-wafer DRIE. Subsequently, sacrificial oxide coatings were used to protect the back side of all wafers during through-wafer etches. The bonding protocol was also improved to minimize wafer handling and to allow multiple wafer bonding of all six wafers in one single step.

As a result of these measures, wafer bonding on the final build produced 100% yield at diesaw; an infra-red image of this bonded stack is shown in Fig. 9.

V. PACKAGING

Packaging the static structure in an experimental test rig required

1) five front side fluidic connections for main air, three fuel lines, and one upstream pressure port;
2) one back side fluidic connection for the aft pressure port in the combustion chamber;
3) one backside electrical connection for the igniter.

The requirement for pressure-sealed electrical and mechanical interconnects was complicated by the high-temperature operating environment of the device. (The walls were expected to
operate at temperatures of approximately 1000K; the device was intended to be tested at pressures as high as 5 atm.)

After initial attempts with high temperature ceramic adhesives, fluidic interconnects for the static structure were made via a glass bead interconnect scheme developed by London, Harrison, and Spearing for similar applications in a micro rocket engine [12]. As illustrated in Fig. 10, this interconnect was made by heating an annular-shaped glass-preform in a 1300 K furnace to directly bond a kovar tube to the silicon surface.

An electrical igniter was also similarly packaged by potting a kovar wire in glass and pressure sealing the fixture through the back side of the static structure. The wire was then resistively heated to ignite the fuel–air mixture and initiate combustion inside the chamber.

Fig. 11 shows pictures of the static structure in its final configuration. (The braze plate at the other end of the kovar tubes was directly attached to a macrofabricated fixture with conventional o-ring fittings.)

VI. EXPERIMENTAL TEST RESULTS

Following satisfactory packaging of the device, the static structure was experimentally tested with the following instrumentation:

1) mass flow controllers to monitor the fuel and air mass flow rates;
2) two type K, 0.010–in sheathed thermocouples to measure the temperature of the outer walls;
3) one type K, 0.010–in sheathed thermocouple to measure exhaust gas temperature at the exit;
4) digital pressure transducers to measure the static pressure in the upstream recirculation jacket and inside the combustion chamber.

A. Effectiveness of the Recirculation Jacket

In order to compare the performance of the static structure with that of the previously demonstrated 3-stack microcombustor [5], premixed hydrogen-air combustion tests were carried out in the static structure at the atmospheric design mass flow rate of the 3-stack microcombustor (\(\dot{m}_{\text{f}} = 0.045 \text{ g/s}\)). These tests were intended to compare the two devices under back-to-back test conditions and allow quantification of any performance increase that might result due to the presence of the recirculation jacket or from improved packaging schemes.

Fig. 12 compares the experimentally measured exit and wall temperatures for the two devices under similar operating conditions. The plot shows an increase in static structure exit gas temperature and a corresponding decrease in outer wall temperature. It also suggests that the static structure is capable of attaining the desired 1600 K turbine inlet temperature at a lower equivalence ratio; this results in a lower fuel consumption that is reflected in an increased efficiency as shown in Fig. 13.

The presence of the recirculation jacket and a larger combustion chamber therefore improves the efficiency of the combustor by 15–50% over the previously described 3-stack. The outer wall temperatures are also lower by approximately 100 K, reducing the overall heat loss and making packaging of the device somewhat easier.
Overall, these tests demonstrate the feasibility of high-temperature, high-efficiency microcombustion within the structural constraints of a silicon combustor and with low overall heat loss from the device.

**B. Hydrogen Combustion Tests**

Additional hydrogen-air tests were subsequently conducted inside the static structure in order to characterize the operating space for hydrogen-air combustion. Figs. 14 and 15 plot the exit gas temperatures and combustor efficiency as a function of mass flow rate for different fuel–air ratios. The corresponding outer wall temperature under these conditions is plotted in Fig. 16. The figures show exit gas temperatures in excess of 1600 K and overall efficiencies as high as 95%. The operating temperature of the outer wall is also seen to be limited below 750 K due to the insulating properties of the combustor recirculation jacket.

Stable combustion could only be sustained up to approximately one third of the design mass flow rate of the device. The device was designed to operate with choked turbine vanes and, thus, elevated pressures in the combustion chamber. However, it was found during testing that the minimum ratio of residence time to chemical kinetic time did not occur at the design point but rather in the unchoked portion of the operating regime (less than approximately 0.2 g/s). This led to blow out prior to the attainment of the high operating pressures needed to sustain stable combustion inside the device at design mass flow rates. For $\phi = 0.6$ at a mass flow rate of 0.16 g/s, the residence time in the combustor was approximately 0.65 ms. The minimum residence time at which a flame was stabilized in the device was approximately 0.42 ms.

A closer examination of the heat transfer through the device also showed that at low mass flow rates, the heat loss to the ambient constituted a significant fraction of total heat generated inside the chamber. Along the left-hand side of the curves in Figs. 14 and 15, the performance of the combustor was therefore limited by the low thermal efficiency of the device.
As the mass flow rate of the device was increased along a constant equivalence ratio curve, the heat loss became a small fraction of the total heat generated (approximately 5 W out of 100). The thermal efficiency of the device therefore approached unity at high mass flow rates; however, decreasing residence time limited the chemical conversion efficiency in this regime. Along the right-hand side of the curves in Figs. 14 and 15, the performance of the combustor was therefore limited by the low chemical efficiency of the device. Further discussion of the analysis that supports these assertions is contained in Mehra [8].

Efforts are currently underway to extend the blow out boundaries of the device by independently controlling the pressure-mass flow characteristic of the device in order to increase the operating pressure inside the chamber.

Among the two different inlet geometries, the slotted inlet configuration was found to exhibit significantly higher efficiency near (95% for 0.044 g/s < \( \dot{m} < 0.12 \text{ g/s} \)) for the kinetically limited regime of the operating space. This was attributed to the presence of multiple, small recirculation zones between the slots; these were more effective in facilitating uniform and rapid ignition of the incoming reactants, thereby increasing the chemical conversion efficiency of the device at high mass flow rates. As expected, the smaller recirculation zones were also less stable, resulting in quenching of the reaction at lower mass flow rates. The maximum stable operating range of the device was 10% lower than for the annular inlet geometry.

C. Fuel Injection Tests

In addition to premixed hydrogen–air combustion tests, non-premixed tests were also conducted to evaluate the performance of the three fuel injection schemes. Fig. 17 plots the exit gas temperature and efficiency measurements for the three different fuel injection schemes and compares them with results from the premixed tests. The figure shows a five-point drop in efficiency due to fuel–air unmixedness upon injection through the first set of axial injectors; injection further downstream has up to a 50-point impact on efficiency.

Injection through the first set of radially located holes is therefore considered to be acceptable; the mixing is expected to improve further as the injectors are operated closer to design conditions. Efforts are currently under way to do additional testing in order to evaluate the performance of the injectors at higher mass flow rates and pressures.\(^3\)

D. Materials and Structural Tests

The static structure was tested at high temperatures and pressures for several hours. Fig. 18 plots the cold flow curves for one of the devices after up to 38 hours of combustion testing. (The gas temperatures during these tests exceeded 1800K.) The curves show minimal change, suggesting that the device continues to pressure seal after high temperature operation. The ultimate failure resulted from silicon fracture due to initiation of the combustion in the recirculation jacket.

\(^3\)A potential negative effect of injecting fuel through the first fuel injector configuration lies in the possibility of initiating combustion in the upstream recirculation jacket. However, since upstream combustion was only observed at very low mass flow rates, it is not expected to be a concern in the high mass flow, design-operating regime of the microengine.

These results therefore demonstrate the structural integrity of the device at high temperatures and pressures up to 1.3 atm, as
long as the mixture does not burn in the upstream recirculation jacket. Efforts are currently underway to evaluate the structural integrity of the device at higher pressures and mass flow rates.

E. Hydrocarbon Test Results

Although the fast chemical reaction rates of hydrogen allow it to serve as an ideal fuel for a residence time-limited microengine, storage and increased energy density requirements ultimately dictate the use of a hydrocarbon fuel for micro air vehicle propulsion and portable power applications. Tests were therefore also carried out to identify the stable operating regime for hydrocarbon combustion in the static structure.

Based on the ratio of the laminar flame speed for hydrogen–air and hydrocarbon–air combustion, the reaction rate for hydrocarbon fuels can be expected to be slower than that for hydrogen by a factor of 5–50 [13], [14]. This suggests the need for a larger combustion chamber for design point hydrocarbon operation in the microengine. Alternatively, for a given volume, hydrocarbon combustion may be stabilized at lower mass flow rates (and, hence, power densities).

Figs. 19 and 20 plot the experimentally measured exit gas temperatures for ethylene–air and propane–air combustion in the static structure. To date, these tests have demonstrated stable ethylene–air combustion with exit gas temperatures in excess of 1600 K, combustor efficiencies between 60–80%, and a maximum power density of approximately 500 MW/m³. Since the reaction rates for propane are much slower than those of ethylene, propane–air combustion could only be stabilized with power density levels of approximately 140 MW/m³. For ethylene–air, the minimum residence time for which stable combustion was obtained was ≈0.72 ms, whereas for propane–air, it was ≈2.5 ms.

These results serve as the first experimental demonstration of stable hydrocarbon combustion within the structural constraints of a silicon chamber. Efforts are currently under way to further examine the use of ethylene for a propulsion engine from a systems perspective and to develop alternative catalytic strategies for design point hydrocarbon combustion in the microengine.

Fig. 19. Exit gas temperature measurements for ethylene-air combustion in the static structure. (Again, the break in the curve results from the inability to measure gas temperatures in excess of 1600–1800 K.)

Fig. 20. Exit gas temperature measurements for propane-air combustion in the static structure (ϕ = 0.8).

VII. CONCLUSIONS AND FUTURE WORK

This paper presented the development of a 6-wafer combustion system for a silicon micro gas turbine engine. Fabricated by deep reactive ion etching through a total thickness of 3800 μm, this structure serves as the first demonstration of the hot flow path of a multilevel microengine. The device was packaged with electrical and fluidic interconnects and tested under hot-flow conditions in order to characterize its performance for hydrogen–air and hydrocarbon–air combustion. It survived high-temperature operation for several tens of hours, thereby demonstrating the viability of silicon-based combustion systems for micro heat engine applications.

Future research work will primarily focus on additional design point testing to establish combustion stability boundaries for the current design of the microengine and to further develop hydrocarbon combustion strategies for the engine. The hot static components will also be integrated with the rotating compressor and turbine; first builds of the entire engine are expected in 2000.

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4The reaction rates for ethylene are approximately twice as high as those for propane [14]; hence, it was considered more favorable from a chemical kinetics perspective.
REFERENCES


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His current research involves the development of a hydrocarbon-fueled microscale combustion system for a micro gas turbine engine. His research interests in the MEMS field include microfluidics, combustion at the micro-scale, and high-speed micro-turbomachinery. Other areas of interest are aero-propulsion, fluid mechanics, reacting flows, thermodynamics and heat transfer, and short-duration turbomachinery testing.

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