GT-2003-38866

MILLIMETER-SCALE, MEMS GAS TURBINE ENGINES

Alan H. Epstein

Gas Turbine Laboratory Massachusetts Institute of Technology Cambridge, MA 02139 USA epstein@mit.edu

ABSTRACT

The confluence of market demand for greatly improved compact power sources for portable electronics with the rapidly expanding capability of micromachining technology has made feasible the development of gas turbines in the millimeter-size range. With airfoil spans measured in 100's of microns rather than meters, these "microengines" have about 1 millionth the air flow of large gas turbines and thus should produce about 1 millionth the power, 10-100 W. Based on semiconductor industry-derived processing of materials such as silicon and silicon carbide to submicron accuracy, such devices are known as micro-electro-mechanical systems (MEMS). Current millimeter-scale designs use centrifugal turbomachinery with pressure ratios in the range of 2:1 to 4:1 and turbine inlet temperatures of 1200-1600 K. The projected performance of these engines are on a par with gas turbines of the 1940's. The thermodynamics of MEMS gas turbines are the same as those for large engines but the mechanics differ due to scaling considerations and manufacturing constraints. The principal challenge is to arrive at a design which meets the thermodynamic and component functional requirements while staying within the realm of realizable micromachining technology. This paper reviews the state-of-the-art of millimeter-size gas turbine engines, including system design and integration, manufacturing, materials, component design, accessories, applications, and economics. It discusses the underlying technical issues, reviews current design approaches, and discusses future development and applications.

INTRODUCTION

For most of the 60-year-plus history of the gas turbine, economic forces have directed the industry toward ever larger engines, currently exceeding 100,000 lbs of thrust for aircraft propulsion and 400 MW for electric power production applications. In the 1990's, interest in smaller-size engines increased, in the few hundred pound thrust range for small aircraft and missiles and in the 20-250 kW size for distributed power production (popularly known as "microturbines"). More recently, interest has developed in even smaller size machines, 1-10 kW,

several of which are marketed commercially [1, 2]. Gas turbines below a few hundred kilowatts in size generally use centrifugal turbomachinery (often derivative of automotive turbocharger technology in the smaller sizes), but are otherwise very similar to their larger brethren in that they are fabricated in much the same way (cast, forged, machined, and assembled) from the same materials (steel, titanium, nickel superalloys). Recently, manufacturing technologies developed by the semiconductor industry have opened a new and very different design space for gas turbine engines – one that enables gas turbines with diameters of millimeters rather than meters, with airfoil dimensions in microns rather than millimeters. These shirt-button-sized gas turbine engines are the focus of this review.

Interest in millimeter-scale gas turbines is fueled by both a technology push and a user pull. The technology push is the development of micromachining capability based on semiconductor manufacturing techniques. This enables the fabrication of complex small parts and assemblies - devices with dimensions in the 1-10,000 μ m size range with submicron precision. Such parts are produced with photolithographically-defined features and many can be made simultaneously, offering the promise of low production cost in large-scale production. Such assemblies are known in the US as micro-electrical-mechanical systems (MEMS) and have been the subject of thousands of publications over the last two decades [3]. In Japan and Europe, devices of this type are known as "microsystems", a term which may encompass a wider variety of fabrication approaches. Early work in MEMS focused on sensors and simple actuators, and many devices based on this technology are in large-scale production, such as pressure transducers and airbag accelerometers for automobiles. More recently, fluid handling is receiving attention. For example, MEMS valves are commercially available, and there are many emerging biomedical diagnostic applications. Also, chemical engineers are pursing MEMS chemical reactors (chemical plants) on a chip [4].

User pull is predominantly one of electric power. The proliferation of small, portable electronics – computers, digital assistants, cell phones, GPS receivers, etc. – require compact energy supplies. Increasingly, these electronics demand energy supplies whose energy and power density exceed that of the best batteries available today. Also, the continuing advance in microelectronics permits the shrinking of electronic subsystems of mobile devices such as ground robots and air vehicles. These small, and in some cases very small, mobile systems require increasingly compact power and propulsion. Hydrocarbon fuels burned in air have 20-30 times the energy density of the best current lithium chemistry-based batteries, so that fuelled systems need only be modestly efficient to compete well with batteries.

Given the need for high power density energy conversion in very small packages, a millimeter-scale gas turbine is an obvious candidate. The air flow through gas turbines of this size is about six orders of magnitude smaller than that of the largest engines and thus they should produce about a million times less power, 10-100 watts with equivalent cycles. Work first started on MEMS approaches in the mid 1990's [5-7]. Researchers rapidly discovered that gas turbines at these small sizes have no fewer engineering challenges than do conventional machines and that many of the solutions evolved over six decades of technology development do not apply in the new design space. This paper reviews work on MEMS gas turbine engines for propulsion and power production. It begins with a short discussion of scaling and preliminary design considerations, and then presents a concise overview of relevant MEMS manufacturing techniques. In more depth, it examines the microscale implications for cycle analysis, aerodynamic and structural design, materials, bearings and rotor dynamics, combustion, and controls and accessories. The gas turbine engine as a system is then considered. This review then discusses propulsion and power applications and briefly looks at derivative technologies such as combined cycles, cogeneration, turbopumps, and rocket engines. The paper concludes with thoughts on future developments.

THERMODYNAMIC AND SCALING CONSIDERATIONS

Thermal power systems encompass a multitude of technical disciplines. The architecture of the overall system is determined by thermodynamics while the design of the system's components is influenced by fluid and structural mechanics and by material, electrical and fabrication concerns. The physical constraints on the design of the mechanical and electrical components are often different at microscale than at more familiar sizes so that the optimal component and system designs are different as well. Conceptually, any of the thermodynamic systems in use today could be realized at microscale. Brayton (air) cycle and the Rankine (vapor) cycle machines are steady flow devices while the Otto [8], Diesel, and Stirling cycles are unsteady engines. The Brayton power cycle (gas turbine) is superior based on considerations of power density, simplicity of fabrication, ease of initial demonstration, ultimate efficiency, and thermal anisotropy.

A conventional, macroscopic gas turbine generator consists of a compressor, a combustion chamber, and a turbine driven by the combustion exhaust that powers the compressor. The residual enthalpy in the exhaust stream provides thrust or can power an electric generator. A macroscale gas turbine with a meter-diameter air intake area generates power on the order of 100 MW. Thus, tens of watts would be produced when such a device is scaled to millimeter size if the power per unit of air flow is maintained. When based on rotating machinery, such power density requires combustor exit temperatures of 1200-1600 K; rotor peripheral speeds of 300-600 m/s and thus rotating structures centrifugally stressed to several hundred MPa since the power density of both turbomachinery and electrical machines scale with the square of the speed, as does the rotor material centrifugal stress; low friction bearings; tight geometric tolerances and clearances between rotating and static parts to inhibit fluid leakage, the clearances in large engines are maintained at about one part in 2000 of the diameter; and thermal isolation of the hot and cold sections.

These thermodynamic considerations are no different at micro- than at macroscale. But the physics and mechanics influencing the design of the components do change with scale, so that the optimal detailed designs can be quite different. Examples of these differences include the viscous forces in the fluid (larger at microscale), usable strength of materials (larger at microscale), surface area-to-volume ratios (larger at microscale), chemical reaction times (invariant), realizable electric field strength (higher at microscale), and manufacturing constraints (limited mainly to two-dimensional, planar geometries given current microfabrication technology).

There are many thermodynamic and architectural design choices in a device as complex as a gas turbine engine. These involve tradeoffs among fabrication difficulty, structural design, heat transfer, and fluid mechanics. Given a primary goal of demonstrating that a high power density MEMS heat engine is physically realizable, MIT's research effort adopted the design philosophy that the first engine should be as simple as possible, with performance traded for simplicity. For example, a recuperated cycle, which requires the addition of a heat exchanger transferring heat from the turbine exhaust to the compressor discharge fluid, offers many benefits including reduced fuel consumption and relaxed turbomachinery performance requirements, but it introduces additional design and fabrication complexity. Thus, the first designs are simple cycle gas turbines.

How big should a "micro" engine be? A micron, a millimeter, a centimeter? Determination of the optimal size for such a device involves considerations of application requirements, fluid mechanics and combustion, manufacturing constraints, and economics. The requirements for many power production applications favor a larger engine size, 50-100 W. Viscous effects in the fluid and combustor residence time requirements also favor larger engine size. Current semiconductor manufacturing technology places both upper and lower limits on engine size. The upper size limit is set mainly by etching depth capability, a few hundred microns at this time. The lower limit is set by feature resolution and aspect ratio. Economic concerns include manufacturing yield and cost. A wafer of fixed size (say 200 mm diameter) would yield many more low power engines than high power engines at essentially the same manufacturing cost per



Figure 1: Simple cycle gas turbine performance with H₂ fuel.

wafer. (Note that the sum of the power produced by all of the engines on the wafer would remain constant at 1-10 kW.) When commercialized, applications and market forces may establish a strong preference here. For the first demonstrations of a concept, a minimum technical risk approach is attractive. Analysis suggested that fluid mechanics would be difficult at smaller scales, so the largest size near the edge of current microfabrication technology, about a centimeter in diameter, was chosen as a focus of MIT's efforts.

Performance calculations indicate that the power per unit air flow from the configuration discussed below is 50-150 W/(g/sec) of air flow (Figure 1). For a given rotor radius, the air flow rate is limited primarily by airfoil span as set by stress in the turbine blade roots. Calculations suggest that it might be possible to improve the specific work, fuel consumption, and air flow rate in later designs with recuperators to realize microengines with power outputs of as much as 50-100 W, power specific fuel consumption of 0.3-0.4 g/w-hr, and thrust-to-weight ratios of 100:1. This level of specific fuel consumption approaches that of current small gas turbine engines but the thrust-to-weight ratio is 5-10 times better than that of the best aircraft engine. The extremely high thrust-to-weight ratio is simply a result of the so-called "cube-square law". All else being the same as the engine is scaled down linearly, the air flow and thus the power decreases with the intake area (the square of the linear size) while the weight decreases with the volume of the engine (the cube of the linear size), so that the power-to-weight ratio increases linearly as the engine size is reduced. Detailed calculations show that the actual scaling is not quite this dramatic since the specific power is lower at the very small sizes [5]. A principal point is that a micro-heat engine is a different device than more familiar full-sized engines, with different weaknesses and different strengths.

Mechanics Scaling

While the thermodynamics are invariant down to this scale, the mechanics are not. The fluid mechanics, for example, are scale-dependent [9]. One aspect is that viscous forces are more

important at small scale. Pressure ratios of 2:1 to 4:1 per stage imply turbomachinery tip Mach numbers that are in the high subsonic or supersonic range. Airfoil chords on the order of a millimeter imply that a device with room temperature inflow, such as a compressor, will operate at Reynolds numbers in the tens of thousands. With higher gas temperatures, turbines of similar size will operate at a Reynolds number of a few thousand. These are small values compared to the 105-106 range of large-scale turbomachinery and viscous losses will be concomitantly larger. But viscous losses make up only about a third of the total fluid loss in a high speed turbomachine (3-D, tip leakage, and shock wave losses account for most of the rest) so that the decrease in machine efficiency with size is not so dramatic. The increased viscous forces also mean that fluid drag in small gaps and on rotating disks will be relatively higher. Unless gas flow passages are smaller than one micron, the fluid behavior can be represented as continuum flow so that molecular kinetics, Knudsen number considerations, are not important.

Heat transfer is another aspect of fluid mechanics in which microdevices operate in a different design space than large-scale machines. The fluid temperatures and velocities are the same but the viscous forces are larger, so the fluid film heat transfer coefficients are higher by a factor of about 3. Not only is there more heat transfer to or from the structure but thermal conductance within the structure is higher due to the short length scale. Thus, temperature gradients within the structure are reduced. This is helpful in reducing thermal stress but makes thermal isolation challenging.

For structural mechanics, it is the change in material properties with length scale that is most important. Very small length scale influences both material properties and material selection. In engines a few millimeters in diameter, design features such as blade tips, fillets, orifices, seals, etc may be only a few microns in size. Here, differences between mechanical design and material properties begin to blur. The scale is not so small (atomic lattice or dislocation core size) that continuum mechanics no longer applies. Thus, elastic, plastic, heat conduction, creep, and oxidation behaviors do not change, but fracture strength can differ. Material selection is influenced both by mechanical requirements and by fabrication constraints. For example, structure ceramics such as silicon carbide (SiC) and silicon nitride (Si_3N_4) have long been recognized as attractive candidates for gas turbine components due to their high strength, low density, and good oxidation resistance. Their use has been limited, however, by the lack of technology to manufacture flaw-free material in sizes large enough for conventional engines. Shrinking engine size by three orders of magnitude virtually eliminates this problem. Indeed, mass-produced, single-crystal semiconductor materials are essentially perfect down to the atomic level so that their usable strength is an order of magnitude better than conventional metals. This higher strength can be used to realize lighter structures, higher rotation speeds (and thus higher power densities) at constant geometry, or simplified geometry (and thus manufacturing) at constant peripheral speed. An additional material



Figure 2: Critical temperature change to cause fracture via thermal shock.

consideration is that thermal shock susceptibility decreases as part size shrinks. Thus, materials such as alumina (Al_2O_3) which have very high temperature capabilities but are not considered high temperature structural ceramics due to their susceptibility to thermal shock are viable at millimeter length scales (Figure 2). Since these have not been considered as MEMS materials in the past, there is currently little suitable manufacturing technology available [10].

OVERVIEW OF A MEMS GAS TURBINE ENGINE DESIGN

Efforts at MIT were initially directed at showing that a MEMS-based gas turbine is indeed possible, by demonstrating benchtop operation of such a device. This implies that, for a first demonstration, it would be expedient to trade engine performance for simplicity, especially fabrication simplicity. Most current, high precision, microfabrication technology applies mainly to silicon. Since Si rapidly loses strength above 950 K, this becomes an upper limit to the turbine rotor temperature. But 950 K is too low a combustor exit temperature to close the engine cycle (i.e. produce net power) with the component efficiencies available, so cooling is required for Si turbines. The simplest way to cool the turbine in a millimeter-sized machine is to eliminate the shaft, and thus conduct the turbine heat to the compressor, rejecting the heat to the compressor fluid. This has the great advantage of simplicity and the great disadvantage of lowering the pressure ratio of the now non-adiabatic compressor from about 4:1 to 2:1 with a concomitant decrease in cycle power output and efficiency. Hydrogen was chosen as the first fuel to simplify the combustor development. This expedient arrangement was referred to as the H₂ demo engine. It is a gas generator/turbojet designed with the objective of demonstrating the concept of a MEMS gas turbine. It does not contain electrical machinery or controls, all of which are external.

The MIT H_2 demo engine design is shown in Figure 3.



Figure 3: H₂ demo engine with conduction-cooled turbine constructed from six silicon wafers.

The centrifugal compressor and radial turbine rotor diameters are 8 mm and 6 mm respectively. The compressor discharge air wraps around the outside of the combustor to cool the combustor walls, capturing the waste heat and so increasing the combustor efficiency while reducing the external package temperature. The rotor radial loads are supported on a journal bearing on the periphery of the compressor. Thrust bearings on the centerline and a thrust balance piston behind the compressor disk support the axial loads. The balance piston is the air source for the hydrostatic journal bearing pressurization. The thrust bearings and balance piston are supplied from external air sources. The design peripheral speed of the compressor is 500 m/s so that the rotation rate is 1.2 Mrpm. External air is used to start the machine. With 400 μ m span airfoils, the unit is sized to pump about 0.36 grams/sec of air, producing 0.1 Newtons of thrust or 17 watts of shaft power. A cutaway engine chip is shown in Figure 4. In this particular engine build, the airfoil span is 225 μ m and the disks are 300 µm thick.

The following sections elaborate on the component technologies of this engine design. It starts with a primer on microfabrication and then goes on to turbomachinery aerodynamic design, structures and materials, combustion, bearings and rotor dynamics, and controls and accessories. A system integration discussion then expands on the high-level tradeoffs which define the design space of a MEMS gas turbine engine.

A PRIMER ON MICROMACHINING

Gas turbine engine design has always been constrained by the practicality of manufacturing parts in the desired shape and size and with the material properties needed. As with conven-



Figure 4: Cutaway H₂ demo gas turbine chip.

tional metal fabrication, the mechanical and electrical properties of MEMS materials can be strongly influenced by the fabrication process.

While an old-school designer may have admonished his team "Don't let the manufacturing people tell you what you can't do!", design for manufacturing is now an important concern in industry. Major decisions in engine architecture are often set by manufacturing constraints. Of course this was true in the design of Whittle's first jet engine, in which the prominent external, reverse flow combustors reflected the need to keep the turbine very close to the compressor to control rotor dynamics given that the forging technology of the day could only produce short, small diameter shafts integral with a disk [11].

Compared to manufacturing technologies familiar at large scale, current microfabrication technology is quite constrained in the geometries that can be produced and this severely limits engine design options. Indeed, the principal challenge is to arrive at a design which meets the thermodynamic and component functional requirements while staying within the realm of realizable micromachining technology. The following paragraphs present a simple overview of current micromachining technology important to this application and then discuss how it influences the design of very small rotating machinery. These technologies were derived from the semiconductor industry 15-20 years ago, but the business of micromachining has now progressed to the level that considerable process equipment (known as "tools") is developed specifically for these purposes [12].

The primary fabrication processes important in this application are etching of photolithographically-defined planar geometries and bonding of multiple wafers. The usual starting point is a flat wafer of the base material, most often single-crystal silicon. These wafers are typically 0.5 to 1.0 mm thick and 100 to 300 mm in diameter, the larger size representing the most modern technology. Since a single device of interest here is typically a centimeter or two square, dozens to hundreds fit on a single wafer (Figure 5). Ideally, the processing of all the devices on a wafer is carried out in parallel, leading to one of the great advantages of this micromachining approach, low unit cost. To greatly simplify a complex process with very many options, the devices of interest will serve as illustrative examples.

First, the wafers are coated with a light-sensitive photoresist. A high contrast black-and-white pattern defining the geometry is then optically transferred to the resist either by means of a contact exposure with a glass plate containing the pattern (a "mask"), or by direct optical or e-beam writing. The photoresist is then chemically developed as though it were photographic film, baked, and then the exposed areas are removed with a solvent. This leaves bare silicon in the areas to be etched and photoresist-protected silicon elsewhere. The etching process is based on the principle that the bare silicon is etched at a much higher rate, typically 50-100x, than the mask material. Many different options for making masks have been developed, including a wide variety of photoresists and various oxide or metal films. By using several layers of masking material, each sensitive to different solvents, multi-depth structures can be defined. Photoresist on top of SiO₂ is one example.

The exposed areas of the wafer can now be etched, either chemically or with a plasma. The devices we are concerned with here require structures which are 100's of microns high with very steep walls, thus a current technology of great interest is deep reactive ion etching (DRIE). In the DRIE machine, the wafer is etched by plasma-assisted fluorine chemistry for several tens of seconds, then the gas composition is changed and a micron or so of a teflon-like polymer is deposited which preferentially protects the vertical surfaces, and then the etch cycle is repeated [13]. The average depth of a feature is a function of the etch time and the local geometry. The etch anisotropy (steepness of the walls) can be changed by adjusting the plasma properties, gas composition, and pressure. In addition, these adjustments may alter the uniformity of the etch rate across the wafer by a few percent since no machine is perfect. One feature of current tools is that the etch rate is a function of local geometry such as the



Figure 5: Si wafer of radial inflow turbine stages.



Figure 6: A 4:1 pressure ratio, 4 mm rotor dia radial inflow turbine stage.

lateral extent of a feature. This means that, for example, different width trenches etch at different rates, presenting a challenge to the designer of a complex planar structure. A DRIE tool typically etches silicon at an average rate of $1-3 \mu m$ per minute, the precise rate being feature- and depth-dependent. Thus, structures that are many hundreds of microns deep require many hours of etching. Such a tool currently costs \$0.5-1.0M and etches one wafer at a time, so the etching operation is a dominant factor in the cost of producing such deep mechanical structures. Both sides of a wafer may be etched sequentially.

Figure 6 is an image of a 4 mm rotor diameter, radial inflow air turbine designed to produce 60 watts of mechanical power at a tip speed of 500 m/s [14, 15]. The airfoil span is 200 μ m. The cylindrical structure in the center is a thrust pad for an axial thrust air bearing. The circumferential gap between the rotor and stator blades is a 15 μ m wide air journal bearing required to support the radial loads. The trailing edge of the rotor blades is 25 μ m thick (uniform to within 0.5 μ m) and the blade roots have 10 μ m radius fillets for stress relief. While the airfoils appear planar in the figure, they are actually slightly tapered from hub to tip. Current technology can yield a taper uniformity of about 30:1 to 50:1 with either a positive or negative slope. At the current state-of-the-art, the airfoil length can be controlled to better than 1 μ m across the disk, which is sufficient to achieve high-speed operation without the need for dynamic balancing. Turbomachines of similar geometry have been produced with blade spans of over 400 μ m.

The processing of a 4-mm-diameter turbine stage is illustrated in Figure 7 as a somewhat simplified example. Note that the vertical scaling in the figure is vastly exaggerated for clarity since the thickness of the layers varies so much (about 1 μ m of silicon dioxide and 10 μ m of photoresist on 450 μ m of silicon). It is a 16-step process for wafer 1, requiring two photo masks. It includes multiple steps of oxide growth (to protect the surface for wafer bonding), patterning, wet etching (with a buffered hydrofluoric acid solution known as BOE), deep reactive ion etching (DRIE), and wafer bonding (of the rotor wafer, #1, to an adjoining wafer, #2, to prevent the rotor from falling out during processing). Note that wafer 2 in the figure was previously processed since it contains additional thrust bearing and plumbing features which are not shown here for clarity. In fact, it is more complex to fabricate than the rotor wafer illustrated.

The second basic fabrication technology of interest here is the bonding together of processed wafers in precision alignment



Figure 7: Simplified processing steps to produce the turbine in Figure 6 in a wafer stack.

so as to form multilayer structures. There are several classes of wafer bonding technologies. One uses an intermediate bonding layer such as a gold eutectic or SiO2. These approaches, however, result in structures which have limited temperature capabilities, a few hundred °C. It is also possible to directly bond silicon to silicon and realize the material's intrinsic strength through the entire usable temperature range of the material [16, 17]. Direct bonding requires very smooth (better than 10 nanometers) and very clean surfaces (a single 1- μ m-diameter particle can keep several square centimeters of surface from bonding). Thus, a very high standard of cleanliness and wafer handling must be maintained throughout the fabrication process. The wafers to be bonded are hydrated and then aligned using reference marks previously etched in the surface. The aligned wafers are brought into contact and held there by Van der Waals forces. The stack of wafers is then pressed and heated to a few hundred degrees for tens of minutes. Finally, the stack is annealed for about one hour at 1100°C in an inert gas furnace. (If a lower temperature is used, a much longer time will be needed for annealing.) Such a stack, well-processed from clean wafers, will not have any discernable bond lines, even under high magnification. Tests show the bonds to be as strong as the base material. The current state-of-the-art is stacks of 5-6 wafers aligned across a wafer to 0.5-1.0 μ m. More wafers can be bonded if alignment precision is less important. Note that the annealing temperature is generally higher than devices encounter in operation. This process step thus represents the limiting temperature for the selection of materials to be included within the device [18]. Figure 8 shows the turbine layer of Figure 6 bonded as the center of a stack of five wafers, the others contain the thrust bearings and fluid plumbing.

A third fabrication technology of interest for micro-rotating machinery is that which realizes a freely-spinning rotor within a wafer-bonded structure. We require completed micromachines which include freely-rotating assemblies with clearances measured in microns. While it is possible to separately fabricate rotors, insert them into a stationary structure, and then bond an overlaying static structure, this implies pick-and-place hand operations (rather than parallel processing of complete wafers) and increases the difficulty in maintaining surfaces sufficiently clean for bonding. A fundamentally different approach is to



(a) Conceptual Cross-Section

arrange a sequence of fabrication steps with all processing done at the wafer level so that a freely-rotating captured rotor is the end product. The process must be such that the rotor is not free at any time during which it can fall out, *i.e.* it must be mechanically constrained at all times. There are several ways that this can be accomplished. For example, the layer containing the rotor can be "glued" to adjoining wafers with an oxide during fabrication. This glue can then be dissolved away to free the rotor after the device is completely fabricated. In one version of the 4 mm turbine of Figure 6, an SiO₂ film bonds the rotor layer at the thrust bearing pad to the adjoining wafer, before the journal bearing is etched. Another approach employs "break-off tabs" or mechanical fuses, flimsy structures which retain the rotor in place during fabrication and are mechanically failed after fabrication is complete to release the rotor [19]. Both approaches have been proven successful.

The last MEMS technology we will mention is that for electronic circuitry, mainly for embedded sensors and electric machinery such as actuators, motors, and generators. The circuitry is generally constructed by laying down alternating insulating and conducting layers, typically by using vapor deposition or sputtering approaches, and patterning them as they are applied using the photoresist technology outlined above. While the microelectronics industry has developed an extremely wide set of such technologies, only a small subset are compatible with the relatively harsh environment of the processing needed to realize wafer-bonded mechanical structures hundreds of microns deep. Specifically, the high wafer annealing temperatures limit the conductor choices to polysilicon or high temperature metals such as platinum or tungsten. The energetic etching processes require relatively thick masking material which limits the smallest electrical feature size to the order of a micron, rather than the tens of nanometers used in state-of-the-art microelectronic devices.

Using the above technologies, shapes are restricted to mainly



⁽b) Electron Microscope Image of Cross-Section

Figure 8: Complete, 5-layer turbine "stack" including bearings and fluid plumbing.

prismatic or "extruded" geometries of constant height. Ongoing research with greyscale lithography suggests that smoothly variable etch depths (and thus airfoils of variable span) may be feasible in the near term [20]. Conceptually, more complex 3-D shapes can be constructed of multiple precision-aligned 2-D layers. But layering is expensive with current technology and 5-6 is considered a large number of precision-aligned layers for a microdevice. Since 3-D rotating machine geometries are difficult to realize, planar geometries are preferred. While 3-D shapes are difficult, in-plane 2-D geometric complexity is essentially free in manufacture since photolithography and etching process an entire wafer at one time. These are much different manufacturing constraints than are common in the large-scale world so it is not surprising the optimal machine design may also be different.

TURBOMACHINERY FLUID MECHANICS

The turbomachine designs considered to date for MEMS engine applications have all been centrifugal since this geometry is readily compatible with manufacturing techniques involving planar lithography. (It is also possible to manufacture single axial flow stages by using intrinsic stresses generated in the manufacturing process to warp what otherwise would be planar paddles into twisted blades, but such techniques have not been pursued for high-speed turbomachinery). In most ways, the fluid mechanics of microturbomachinery are similar to that of large-scale machines. For example, high tip speeds are needed to achieve high pressure ratios per stage. Micromachines are different in two significant ways: small Reynolds numbers (increased viscous forces in the fluid) and, currently, 2-D, prismatic geometry limitations. The low Reynolds numbers, 10^{3} - 10^{5} , are simply a reflection of the small size, and place the designs in the laminar or transitional range. These values are low enough that it is difficult to diffuse the flow, either in a rotor or a stator, without separation. This implies that either most of the stage work must come from the centrifugal pressure change or that some separation must be tolerated. The design challenges introduced by the low Reynolds numbers are exacerbated by geometric restrictions imposed by current microfabrication technology. In particular, the fabrication constraint of constant passage height is a problem in these high-speed designs. High work on the fluid means large fluid density changes. In conventional centrifugal turbomachinery, density change is accommodated in compressors by contracting or in turbines by expanding the height of the flow path. However, conventional microfabrication technology is not amenable to tapering passage heights, so all devices built to date have a constant span. How these design challenges manifest themselves are somewhat different in compressors and turbines.

A common fluid design challenge is turning the flow to angles orthogonal to the lithographically-defined etch plane, such as at the impeller eye or the outer periphery of the compressor diffuser. At conventional scale, these geometries would be carefully contoured and perhaps turning vanes would be added. Such geometry is currently difficult to produce with microfabrication, which most naturally produces sharp right angles that are deleterious to the fluid flow. For example, at the 2-mm-diameter inlet to a compressor impeller, 3-D CFD simulations show that a right-angle turn costs 5% in compressor efficiency and 15% in mass flow compared to a smooth turn [21]. Engineering approaches to this problem include lowering the Mach number at the turns (by increasing the flow area), smoothing the turns with steps or angles (which adds fabrication complexity), and adding externally-produced contoured parts when the turns are at the inlet or outlet to the chip.

Compressor Aerodynamic Design

The engine cycle demands pressure ratios of 2-4, the higher the better. This implies that transonic tip Mach numbers and therefore rotor tip peripheral speeds in the 400-500 m/s range are needed. This yields Reynolds numbers (Re) in the range of 10^4 for millimeter-chord blades. The sensitivity of 2-D blade performance to Re in this regime is illustrated in Figure 9, which presents the variations of efficiency with size for a radial flow compressor and turbine. While this analysis suggests that for low loss it is desirable to maximize chord, note that the span of the airfoils is less than the chord, implying that aero designs should include endwall considerations at this scale.

In conventional size machines the flow path contracts to control diffusion. Since this was not possible with established micromachining technology, the first approach taken was to control diffusion in blade and vane passages by tailoring the airfoil thickness rather than the passage height [21, 22]. This approach results in very thick blades, as can be seen in the 4:1 pressure ratio compressor shown in Figure 10. Compared to conventional blading, the trailing edges are relatively thick and the exit angle is quite high. The design trade is between thick trailing edges (which add loss to the rotor) or high rotor exit angles (which result in reduced work at constant wheel speed, increased diffuser loss, and reduced operating range).

Although the geometry is 2-D, the fluid flow is not. The relatively short blade spans, thick airfoil tips, and low Reynolds numbers result in large hub-to-tip flow variations, especially at



Figure 9: Calculated sensitivity of 2-D airfoil loss with Reynolds number [9].



Figure 10: A 500 m/s tip speed, 8 mm dia centrifugal engine compressor.

the impeller exit. This imposes a spanwise variation on stator inlet angle of 15-20 degrees for the geometries examined. This cannot be accommodated by twisting the airfoils, which is not permitted in current microfabrication. The limited ability to diffuse the flow without separation at these Reynolds numbers also precludes the use of vaneless diffusers if high efficiency is required, since the flow rapidly separates off parallel endwalls.

While extensive 2-D and 3-D numerical simulations have been used to help in the design and analysis of the micromachines, as in all high-speed turbomachinery development, test data is needed. Instrumentation suitable for fluid flow measurements in turbomachinery with blade spans of a few hundred microns and turning at over a million rpm is not readily available. While it is theoretically possible to microfabricate the required instrumentation into the turbomachine, this approach to instrumentation is at least as difficult as fabricating the microturbomachine in the first place. Instead, the standard technique of using a scaled turbomachine test rig was adopted [23]. In this case the test rig was a 75x linear scale-up of a 4-mm-diameter compressor (sufficiently large with a 300-mm-rotor diameter for conventional instrumentation) rather than the 2-4x scale-down common in industry. The geometry tested was a model of a 2:1 pressure ratio, 4-mm-diameter compressor with a design tip speed of 400 m/sec for use in a micromotor-driven air compressor [24]. This design used the thick-blade-to-control-diffusion philosophy discussed above. The rig was operated at reduced inlet pressure to match the full-scale design Reynolds number of about 20,000. A comparison of a steady, 3-D, viscous CFD (FLUENT) simulation to data is shown in Figure 11. The simulation domain included the blade tip gaps and right-angle turn at the inlet. It predicts the pressure rise and mass flow rate to 5% and 10%, respectively.

Tight clearances are considered highly desirable for compressor aerodynamics in general but are a two-edged sword for the thick-bladed designs discussed above. Small tip clearance reduces leakage flow and its associated losses, but increases drag for designs in which the blade tip is at least as wide as the passage. The full-scale blading dimensions of the microcompressor tested scaled-up was a blade chord of about 1000 μ m and a span of 225 μ m. Thus the design minimum tip clearance of 2 μ m (set to avoid blade tip rubs) represents 0.2% of chord and 1% of span. Figure 11 includes measurements of the sensitivity of this design to tip clearance.

Recent microfabrication advances using greyscale lithography approaches suggest that variable span turbomachinery may indeed be feasible [20]. This would facilitate designs with attached flow on thin blades. Compared to the thick blade approach, 3-D CFD simulations of thin blade compressors with a tip shroud show about twice the mass flow for the same maximum span and wheel speed, an increase in pressure ratio from 2.5 to 3.5, and an increase in adiabatic efficiency from 50% to 70% [25].

Isomura *et al.* have taken a different approach to centimeter-scale centrifugal compressors [26, 27]. They have chosen to scale a conventional 3-D aerodynamic machine with an inducer down to a 12 mm diameter for a design 2 g/s mass flow rate and 3:1 pressure ratio. The test article is machined from aluminum on a high-precision, five-axis miller. No test results have been reported to date.

Kang *et al.* [28] have built a 12-mm-diameter conventional geometry centrifugal compressor from silicon nitride using a rapid prototype technology known as mold shape deposition manufacture. It was designed to produce a pressure ratio of 3:1 at 500 m/s tip speed with a mass flow of 2.5 g/s and an efficiency of 65-70%. To date, they report testing up to 250 m/s and performance consistent with CFD analysis.

A major aerodynamic design issue peculiar to these very small machines is their sensitivity to heat addition. It is difficult to design a centimeter-scale gas turbine engine to be completely



Figure 11: Sensitivity of compressor pressure rise to tip clearance (% span).



Figure 12: The influence of heat addition on compressor performance (pressure ratio is π , the subscript "ad" refers to the adiabatic condition).

adiabatic, thus there will be some degree of heat addition through conduction. An isothermal compressor at fixed temperature exhibits behavior close to that of an adiabatic machine with the same amount of heat added at the inlet [29]. Thus, the influence of the heat addition shows up as reductions in mass flow, pressure rise, and adiabatic efficiency. The effect of heat addition on compressor efficiency and pressure ratio are shown in Figure 12. These effects can be quite dramatic at high levels of heat flow. The influence of this nonadiabatic behavior on the overall cycle performance will be discussed later.

The ultimate efficiency potential for compressors in this size range has yet to be determined. Figure 13 plots the polytropic efficiency of a number of aeroengines and ground-based gas turbine compressors using inlet-corrected mass flow as an indicator of size. The efficiency decreases with size but how much of this is intrinsic to the fluid physics and how much is due to the discrepancy in development effort (little engines have little budgets) is not clear. (Note that there is an inconsistency of about a percent in this data due to different definitions of efficiency, *i.e.* whether losses in the inlet guide vanes and the exit vanes or struts are included.)

Turbine Aerodynamic and Heat Transfer Design

While the aerodynamic design of a microfabricated, centimeter-diameter radial inflow turbine shares many of the design challenges of a similar scale compressor, such as a constant airfoil span manufacturing constraint, the emphasis is different. Diffusion within the blade passages is not the dominant issue it is with the compressor, so the thick blade shapes are not attractive. The Reynolds numbers are lower, however, given increased viscosity of the high temperature combustor exit fluid. The nozzle guide vanes (NGVs) operate at a Re of 1,000-2,000 for millimeter-chord airfoils.

One 6-mm-diameter, constant-span engine turbine is shown



Figure 13: Variation of engine compressor polytropic efficiency with size.

in Figure 14. With a 400 μ m span it is designed to produce 53 W of shaft power at a pressure ratio (T-S) of 2.1, tip speed of 370 m/s, and mass flow of 0.28 g/s. The reaction is 0.2 which means that the flow is accelerating through the turbine. Three-dimensional CFD simulations were used to explore the performance of this design using FLUENT. The calculational domain included the blade tip gap regions, the discharge of bearing air into the turbine, and the right-angle turn and duct downstream of the rotor. These calculations predict that this design has an adiabatic efficiency of about 60%. The remainder of the power goes to NGV losses (9%), rotor losses (11%), and exit losses (20%) [30]. These are very low aspect ratio airfoils (~ 0.25) and this is reflected in the shear on the endwalls being about twice that on the airfoil surfaces. The exit losses, the largest source of inefficiency, consist of residual swirl, losses in the right-angle turn, and lack of pressure recovery in the downstream duct. This implies that (a)



Figure 14: Silicon engine radial inflow turbine inside annular combustor; the flow passages in the NGV's are for bearing and balance air.

the rotor exit Mach number should be reduced if possible, and (b) that the turbine would benefit from an exit diffuser.

High engine-specific powers require turbine inlet temperatures (TIT) above the 950 K capability of uncooled single-crystal Si. The MIT demo engine was designed with a TIT of 1600 K and so requires turbine cooling. In the demo design the turbine is conductively cooled through the structure. The heat flow is on the same order as the shaft power, and the resultant entropy reduction is equivalent to 1-2% improvement in turbine efficiency. Advanced engine designs may use film cooling. A major issue in this case is the stability of a cold boundary layer on a rotating disk with radial inflow. While this is, in general, an unstable flow, Philippon has shown through analysis and CFD simulation that the region of interest for these millimeter-scale turbines lies in a stable regime (*e.g.* the boundary layers should stay attached) [30]. He then designed film-cooled turbines and analyzed these designs with CFD simulation.

Based upon the work to date, it should be possible to realize microfabricated single-stage compressors with adiabatic pressure ratios above 4:1 at 500 m/s tip speed with total-to-static efficiencies of 50-60%. Achievable turbine efficiencies may be 5-10% higher.

COMBUSTION

The primary design requirements for gas turbine combustors include large temperature rise, high efficiency, low pressure drop, structural integrity, ignition, stability, and low emissions. These requirements are no different for a microcombustor which may flow less than 1 g/s of air than for a 100 kg/sec large machine, but the implementation required to achieve them can be. A comparison between a modern aircraft engine combustor and a microengine is shown in Table 1 [31]. Scaling considerations result in the power density of a microcombustor exceeding that of a large engine. However, the combustor volume relative to the rest of the microengine is much larger, by a factor of 40, than that of a large engine. The reasons for this scaling can be understood in reference to the basics of combustion science [32].

Combustion requires the mixing of fuel and air followed by chemical reaction. The time required to complete these processes is generally referred to as the required combustion residence time and effectively sets the minimum volume of the combustor for a given mass flow. The mixing time can scale with device size but the chemical reaction times do not. In a large engine, mixing may account for more than 90% of combustor residence time. A useful metric is the homogeneous Damkohler number, which is the ratio of the actual fluid residence time in the combustor to the reaction time. Obviously a Damkohler of one or greater is needed for complete combustion and therefore high combustion efficiency. One difference between large and microscale machines is the increased surface area-to-volume ratio at small sizes. This offers more area for catalysts; it also implies that microcombustors have proportionately larger heat losses. While combustor heat loss is negligible for large-scale engines, it is a dominant design factor at microscale since it can

Table 1: A comparison of a microengine of	combustor w	ith a
large aeroengine combust	or	

	Conventional Combustor	Micro- Combustor
Length	0.2 m	0.001 m
Volume	0.073 m^3	6.6x10 ⁻⁸ m ³
Cross-sectional area	0.36 m^2	6x10 ⁻⁵ m ²
Inlet total pressure	37.5 atm	4 atm
Inlet total temperature	870 K	500 K
Mass flow rate	140 kg/s	1.8x10 ⁻⁴ kg/s
Residence time	~7 ms	~0.5 ms
Efficiency	>99%	>90%
Pressure ratio	>0.95	>0.95
Exit temperature	1800 K	1600 K
Power Density	1960 MW/m ³	3000 MW/m ³

(Note: residence times are calculated using inlet pressure and an average flow temperature of 1000 K.)

reduce the combustor efficiency and lower the reaction temperature. This narrows the flammability limits and decreases the kinetic rates, which drops the effective Damkohler number. As an example, Figure 15 [31] illustrates the viable design space for an H_2 -fuelled, 0.07 cc microcombustor as a function of the heat lost to the walls and as constrained by flame stability, structure limits, and cycle requirement considerations. The design space shown permits a trade between heat loss and stoichiometry, which is especially important when burning hydrocarbons with narrow stoichiometry bounds.

The design details are dependent on the fuel chosen. The design approach first taken was to separate the fuel-air mixing from the chemical reaction. This is accomplished by premixing the fuel with the compressor discharge air upstream of the combustor flame holders. This permits a reduction of the combustor residence time required by a factor of about 10 from the usual 5-10 msec. The disadvantage of this approach is a susceptibility to flashback from the combustor into the premix zone, which



Figure 15: Design space for Si H₂ microcombustor.

must be avoided. To expedite the demonstration of a micro-gas turbine engine, hydrogen was chosen as the initial fuel because of its wide flammability limits and fast reaction time. This is the same approach taken by von Ohain when developing the first jet engine in Germany in the 1930's. Hydrogen is particularly attractive because it will burn at equivalence ratios, ϕ , as low as 0.3 which yields adiabatic combustion temperatures below 1500 K, facilitating the realization of simple premixed designs.

Microcombustor technology has been developed in several full-sized (*i.e.* micro) test rigs which duplicate the geometry of an engine but with the rotating parts replaced with stationary swirl vanes [33]. In the Si micromachined geometry of Figures 3 and 4, to reduce heat losses through the walls and therefore to increase combustor efficiency, the inlet air wraps around the outside of the 0.2 cc combustor before entering through flame holders in a reversed flow configuration. This configuration is similar to the traditional reverse-flow engine combustor but scaled down to 0.1-0.3 g/sec air flow rate. The Si liner in this case is conduction- rather than film-cooled. In this premixed approach, fuel is injected near the inlet of the upstream duct to allow time for fuelair mixing without requiring additional combustor volume. This design takes advantage of microfabrication's ability to produce similar geometric features simultaneously, using 90 fuel injection ports, each 120 μ m in diameter, to promote uniform fuel-air mixing. A simple hot wire loop provides ignition [34].

The combustor was tested in several configurations including variations of flame holder and dilution hole geometry. Combustion efficiencies approaching 100% have been reported with pressure ratios of about 0.95-0.98. The H_2 data in Figure 16 shows the variation of combustor efficiency versus mass flow rate for two configurations, one purely premixed (no dilution holes) and one



Figure 16: Measured performance of 0.2 cc, Si microcombustors using H₂ fuel.

in which dilution holes have been added to the liner creating a dual-zone combustor [31]. The missing data is due to instrumentation burnout. The dual-zone configuration, in which the dilution jets set up recirculation zones within the combustor, extends the operating range by about a factor of two at a cost of 10-20% in combustor efficiency. These combustors have been operated at exit temperatures above 1800 K.

Hydrocarbon fuels such as methane and propane have reaction rates only about 20% of those of H₂, requiring larger combustor volumes for the same heat release. They also must react closer to stoichiometric and therefore at higher temperatures, above 2000 K. For gas phase (homogeneous) combustion designs this requires a multizone burner (stoichiometric zone followed by a dilution region) as used on most large gas turbines. Alternatively, heterogeneous reactions on the surface of a catalyst can widen the flammability limits and so reduce the combustion temperature. Both approaches have been demonstrated at microscale. Ethylene (which has a high reaction rate) and propane have been burned in the H₂ combustors described above. The combustion efficiency with ethylene approached 90% while that for propane was closer to 60%. These fuels need larger combustor volumes compared to hydrogen for the same heat release. Data for a variety of geometries and fuels is reduced in terms of Damkohler number in Figure 17, which shows that values of greater than 2 are needed for high chemical efficiency [31].

Catalytic microcombustors have been produced by filling the combustor volume of the above geometries with a platinumcoated nickel foam. For propane, the catalyst increased the heat release in the same volume by a factor of 4-5 compared to the propane-air results discussed above. Pressure drops through the foam are only 1-2% [35]. Presumably catalytic combustor performance can be improved by a better choice of catalyst (platinum was selected for H₂) and a geometry optimized for catalytic rather than gas-phase combustion.

1.0 Chemical efficiency 0.8 0.6 0.4 Six-wafer (annular) Six-wafer (slotted) 95% confidence 0 0.2 Dual-zone Three-stack 01 0 2 4 6 8 10 12 14 Damkohler No.

Figure 17: Measured microcombustor performance as a function of Damkohler number.

Takahashi et al. [36] are developing combustors designed

for somewhat larger gas turbines, with flow rates of about 2 g/s. Designed for methane, these are a miniature version of can-type industrial combustion chambers with a convection-cooled liner and dilution holes. These are conventionally machined with volumes of 2-4 cc. The combustion efficiencies of these units have been demonstrated as above 99% at equivalence ratios of about 0.37 with a design combustor exit temperature of 1323 K. The design residence time is about 6.5 ms. Matsuo *et al.* [37] constructed a still larger (20 cc volume, 16 g/s flow rate) conventionally-machined combustor burning liquefied natural gas. They report a combustor exit temperature of about 1200 K.

Overall, experiments and calculations to date indicate that high efficiency combustion systems can be engineered at microscale and achieve the heat release rate and efficiency needed for very small gas turbine engines.

BEARINGS AND ROTOR DYNAMICS

The mechanical design of gas turbine engines is dominated by the bearings and rotor dynamics considerations of highspeed rotating machinery. Micromachines are no different in this regard. As in all high-speed rotating machinery, the basic mechanical architecture of the device must be laid out so as to avoid rotor dynamic problems. The high peripheral speeds required by the fluid and electromechanics lead to designs which are supercritical (operate above the natural resonant frequency of the rotor system), just as they often are in large gas turbines.

Key design requirements imposed by the rotor dynamics are that mechanical critical (resonant) frequencies lie outside the steady-state operating envelope, and that any critical frequencies that must be traversed during acceleration are of sufficiently low amplitude to avoid rubs or unacceptable vibrations. The bearings play an important role in the rotor dynamics since their location and dynamical properties (stiffness and damping) are a major determinant of the rotor dynamics. The bearings in turn must support the rotor against all radial and axial loads seen in service. In addition to the rotor dynamic forces, the bearing loads under normal operation include all the net pressure and electrical forces acting on the rotor as well as the weight of the rotor times the external accelerations imposed on the device. For aircraft engines this is usually chosen as 9 g's, but a small device dropped on a hard floor from two meters experiences considerably larger peak accelerations. An additional requirement for portable equipment is that the rotor support be independent of device orientation. The bearing technology chosen must be compatible with the high temperatures in a gas turbine engine (or be protected within cooled compartments) and be compatible with the fabrication processes.

Early MEMS rotating machines have been mainly microelectric motors or gear trains turning at significantly lower speeds and for shorter times than are of interest here, so these made do with dry friction bearings operating for limited periods. The higher speeds and longer lives desired for micro-heat engines require low friction bearings. The very small size of these devices precludes the incorporation of commercially available rolling contact bearings. A microfabricated bearing solution is needed. Both electromagnetic and air bearings have been considered for this application.

Electromagnetic bearings can be implemented with either magnetic or electric fields providing the rotor support force. Although extensive work has been done on the application of magnetic bearings to large rotating machinery, work is just beginning on magnetic bearings for micromachines. In addition to their complexity, magnetic bearings have two major challenges in this application. First, magnetic materials are not compatible with most microfabrication technologies, limiting device fabrication options. Second, Curie point considerations limit the temperatures at which magnetic designs can operate to below those encountered in the micro-gas turbine, so considerable cooling may be needed. For these reasons, the first efforts concentrated on designs employing electric fields. The designs examined did not appear promising in that the forces produced were marginal compared to the bearing loads expected [38]. Also, since electromagnetic bearings are unstable, feedback stabilization is needed, adding to system complexity.

Gas bearings support their load on thin layers of pressurized gas. For micromachines such as turbines they have intrinsic advantages over electromagnetic approaches, including no temperature limits, high load bearing capability, and relative manufacturing simplicity. At large scale, gas bearings are used in many high-speed turbomachinery applications, including aircraft environmental control units, auxiliary power units, 30-70 kW "microturbines", and turbochargers [39]. At smaller scale, gas bearings have been used in gyroscopic instruments for many years. All else being the same, the relative load-bearing capability of a gas bearing improves as size decreases since the volume-to-surface area ratio (and thus the inertial load) scales inversely with size. Rotor and bearing dynamics scaling is more complex [40]. However, rotor dynamics in this application are somewhat simplified compared to large engines since the structure is very stiff, so only rigid body modes need be considered. In the following paragraphs we will first discuss journal bearings which support radial loads and then consider thrust bearings needed for axial loads.

The simplest journal bearing is a cylindrical rotor within a close-fitting circular journal. Other, more complex, variations used in large size machines include foil bearings and wave bearings. These can offer several advantages but are more difficult to manufacture at very small size. Thus, the plane cylindrical geometry was the first approach adopted since it seemed the easiest to microfabricate. Gas bearings of this type can be categorized into two general classes which have differing load capacities and dynamical characteristics. When the gas pressure is supplied from an external source and the bearing support forces are not a first order function of speed, the bearing is termed *hydrostatic*. When the bearing support forces are derived from the motion of the rotor, then the design is *hydrodynamic*. Hybrid implementations combining aspects of both are also possible. Since the MEMS gas turbines include air compressors, both approaches are



Figure 18: Gas bearing radial unit load capacity variation with speed.

applicable. Both can readily support the loads of machines in this size range and can be used at very high temperatures. The two types of bearings have differing load and dynamic characteristics. In hydrodynamic bearings, the load capacity increases with the speed since the film pressure supporting the rotor is generated by the rotor motion. This can be true for a hydrostatic bearing as well if the film pressure is increased with increasing rotor speed, for example if the pressure is derived from an engine compressor. However, when the supporting film pressure in a hydrostatic bearing is kept constant, the load capacity decreases slightly with increasing speed. The calculated unit load capacity (support force per unit area of bearing) of plane journal microbearings is compared with the measured capacity of conventional air foil bearings in Figure 18. The hydrostatic bearing is at a constant pressure. For hydrodynamic bearings the load capacity is a function both of rotational speed and of bearing length (L) to diameter (D) ratio. Microbearings currently have low L/D's due to manufacturing constraints, so their load capacity is less.

The relevant physical parameters determining the bearing behavior are the length-to-diameter ratio (L/D); the journal gapto-length ratio (g/L); and nondimensional forms of the peripheral Mach number of the rotor (a measure of compressibility), the Reynolds number, and the mass of the rotor. For a bearing supported on a hydrodynamic film, the load bearing capability scales inversely with (g/D)⁵ which tends to dominate the design considerations [41].

The design space available for the micro-journal bearing is greatly constrained by manufacturing capability, especially if the rotor and journal structure are fabricated at the same time (which avoids the need for assembly and so facilitates low cost, waferlevel manufacturing). The most important constraint is the etching of vertical side walls. Recent advances of etching technology yields taper ratios of about 30:1 to 50:1 on narrow (10-20 μ m) etched vertical channels 300-500 μ m deep [15]. This capability defines the bearing length while the taper ratio delimits the bearing gap, g. For hydrodynamic bearings we wish to maximize the footprint and minimize gap/diameter to maximize load capacity, so the bearing should be on the largest diameter available, the periphery of the rotor. The penalty for the high diameter is relatively high area and surface speed, thus high bearing drag, and low L/D and therefore reduced stability. In the radial 4000- μ m-diameter turbine shown in Figure 6, the journal bearing is 300 μ m long and about 15 μ m wide, so it has an L/D of 0.075, g/D of 0.038, and peripheral Mach number of 1. This relatively short, wide-gapped, high-speed bearing is well outside the range of analytical and experimental results reported in the gas bearing literature. It is much closer to an air seal in aspect ratio.

The dynamical behavior of the rotor is of first order concern because the high rotational speeds needed for high power density by the turbo and electrical machinery require the rotor to operate at rotational frequencies several times the lowest radial resonant frequency of the bearing/rotor system. The dynamics of gas bearings on a stiff rotor can be simply represented by the rotor mounted on a set of springs and dampers, as illustrated in Figure 19. The fluid in the bearing acts as both the springs and the principal source of damping. It also generates the destabilizing cross-stiffness forces which cause instability at high speeds. As in many conventional engines, the rotor must traverse the critical frequency and avoid instabilities at higher speeds. For example, Figure 20 illustrates the whirl radius versus speed for a 4-mm-diameter turbine with a $12-\mu$ m-wide bearing. Plotted on the figure are experimental data and a fit of an analytical fluid mechanic spring-mass-damper model of the system to that data. The resonant peak amplitude is reached as the rotor crosses a "rotor critical" (resonant) frequency. If the peak excursion exceeds the bearing clearance, then the rotor hits the wall, i.e. "crashes". A well-known characteristic of a spring-mounted rotor system (a so-called "Jeffcott rotor") is that at speeds below the critical frequency the rotor revolves around its geometric center, while well above the critical frequency the rotor revolves around its center of mass. Thus the dotted line in the figure, the



Figure 19: Gas journal bearing model.



Figure 20: Transcritical response of the micro-journal gas bearing in Figure 6.

asymptote of the curve fit, is a measurement of the rotor imbalance expressed in terms of radial displacement of the rotor center of mass from the geometric center. The measured imbalance shown in the figure is ~2 μ m, compared to the 12 μ m bearing clearance (*i.e.* at 12 μ m imbalance the rotor would strike the wall on every revolution).

We thus have two rotor dynamic design considerations, traversing the critical frequency and ensuring that the frequency for the onset of instability is above the operating range. For a hydrostatic bearing the critical frequency simply scales with the pressure in the bearing. The damping ratio (mainly viscous damping) decreases with increasing speed. Thus, the maximum amplitude the rotor experiences while crossing the critical frequency increases with bearing pressure, *i.e.* the peak in Figure 20 moves up and to the right with increasing pressure [42]. This suggests the strategy of crossing the critical frequency at low pressure and low speed and then increasing the pressure to stiffen the bearing as the rotor accelerates to increase the speed at onset of instability [43].

The rotor imbalance is another factor which influences both the peak amplitude crossing the critical frequency and the onset of instability. Large rotating machines are usually dynamically balanced by measuring the imbalance and then adding or subtracting mass to reduce it. In many micromachines it is possible to avoid the need for dynamic balancing because the base material used to date (single-crystal silicon) is extremely uniform and, with sufficient care, the etching uniformity is sufficient to produce adequately balanced rotors. Typically, the center of mass is within 1-5 μ m of the geometric center. For the turbine in Figure 6, the blades must be etched to about $\pm 1 \ \mu m$ span uniformity across the 4000-µm-diameter disk. For rotors made up of several wafers, the alignment between wafers must also be considered (also about 1 μ m is needed) [44]. Using the balance measurement capability evident in Figure 20 and laser etching, it is also possible to dynamically balance a microrotor. It is unclear at this time whether dynamic balancing or manufacturing uniformity is a superior approach.

Hydrostatic bearings are stable from zero speed up to the stability boundary. However, centered hydrodynamic bearings are unstable at low rotational speed but stable at high speeds. Commonly, such bearings are stabilized by the application of a unidirectional force which pushes the rotor toward the journal wall, as measured by the eccentricity, the minimum approach distance of the rotor to the wall as a fraction of the average gap (0 = centered, 1 = wall strike). At conventional scale, the rotor weight is often the source of this side force. At microscale, (1) the rotor weight is negligible, and (2) insensitivity to orientation is desirable, so a scheme has been adopted which uses differential gas pressure to force the rotor eccentric. Extensive numerical modeling of these microbearing flows has shown that such a rotor will be stable at eccentricities above 0.8-0.9 [45]. For the geometry of the turbine in Figure 6, the rotor must thus operate between 1-2 μ m from the journal wall. This implies that deviations from circularity of the journal and rotor must be small compared to 1 μ m, an additional manufacturing requirement.

A rotor must be supported against axial as well as radial loads and so requires thrust bearings in addition to the radial bearings discussed above. Both hydrostatic and hydrodynamic approaches have been demonstrated. In either approach, the bearing must support the axial loads and remain stable. The devices built to date have been designed for sub-critical thrust bearing operation so that bearing behavior traversing the critical frequency is not an issue.

Hydrostatic thrust bearings meter external air through supply orifices onto the bearing surface. The 400- μ m-diameter thrust pad at the rotor center of the 4-mm-diameter turbine in Figure 6 rotates relative to a stationary thrust bearing surface of similar diameter. The stationary bearing surface is perforated with a circular array of 12- μ m-diameter nozzle orifices fed from a plenum which supplies the gas lubricating film between the bearing surfaces. A cross-section is shown in Figure 21. At a rotor-stator gap of 1.2 μ m, a flow of 10 sccm at 2-5 atm is needed to provide sufficient load capacity (0.5 N) and axial stiffness



Figure 21: Geometry of (a) hydrodynamic and (b) hydrostatic thrust bearings (not to scale).



Figure 22: Hybrid hydrodynamic (spiral grooves) and hydrostatic (orifices) 0.7 mm dia thrust bearing.

 $(2 \times 10^5 \text{ N/m})$. Stiffness is maximized when the pressure drop through the supply orifices equals that of the radial outflow from the orifice discharge to the bearing edge.

Hydrodynamic thrust bearings use viscous drag, often enhanced with shallow spiral grooves, to generate a pressure gradient in the bearing which increases toward the rotor center. The pressurized gas film provides the bearing load capacity and stiffness. This self-pumping eliminates the need for an external air supply and simplifies the manufacture since bearing air supply plumbing is not required, reducing the number of wafers needed. It also adds an additional design consideration - rotor liftoff, i.e. the minimum rotational speed needed to develop sufficient pressure to eliminate rubbing between the stationary and rotating parts. Figure 22 is a 700- μ m-diameter hydrodynamic thrust bearing with 1.5 μ m deep spiral grooves that was tested on the microturbine of Figure 6. It lifts off at about 80,000 rpm. Such a bearing at 10⁶ rpm will dissipate about 0.2 watts, about the same as that of a hydrostatic bearing of equal load capacity [46].

	Ni-based Super Alloys	Titanium Alloys	Micro SiC	Micro Silicon
$\begin{array}{c} Centrifugal \ Stress \\ [\sqrt{\sigma_f / \rho}] \ (m/s) \end{array}$	330	420	670	1000
Thermal Stress $[\alpha E/\sigma_{f/y}]$	2.7 x 10 ⁻³	1.2 x 10 ⁻³	1.1 x 10 ⁻³	0.9 x 10 ⁻³
Stiffness [E/ρ] (MPa/Kgm ⁻³)	~26	~25	~95	~70
Max Temp (°C) (life limit)	~1000 (creep)	~300 (strength)	~1500 (oxidation)	~600 (creep)

Table 2: Design Considerations and Material Properties of Interest for Gas Turbines

STRUCTURES AND MATERIALS

Structural considerations for the design of a MEMS gas turbine are in many ways similar to those of conventional engines. The design space is defined by the requirements of the thermodynamics (which require high stress and high temperatures), the properties of the materials, and the manufacturing capabilities. The material properties, in turn, are very much dependent on their processing. This section reviews materials selection, structural design features, high temperature structures, analysis of such microstructures, and packaging (installation) technology.

Materials

Materials for gas turbine engines must exhibit high specific strength (strength/density) at high temperatures. High temperature operation also requires creep and oxidation resistance. Other properties of interest include fracture toughness, modulus, and resistance to thermal shock (Table 2). MEMS processing technologies are much more mature for silicon than for other materials so it is the first material a MEMS engineer considers (not so for a gas turbine designer). In terms of strength at temperature, single-crystal Si is the equal of common nickel alloys and, because it has only 1/3 the density, its specific strength is much higher, as illustrated in Figure 23. It is quite oxidation-resistant



Figure 23: Material properties relevant to high speed, high temperature rotating machinery.

and has thermal conductivity approaching that of copper, so it is resistant to thermal shock. On these grounds, it is not a bad material for gas turbine engines. However, at temperatures below about 900 K, Si is brittle, so usable strength is very much a function of the details of the processing. Structural life must be assessed with statistical methods.

Chen *et al.* [47] have reported room temperature strengths up to 4 GPa for micromachined Si specimens. Moon *et al.* [48] have measured the strength and creep rate of Si at temperatures up to 1000 K. From these measurements and a detailed model of the creep behavior of the material, it appears that long-lived structures can be designed for stress levels up to about 500 MPa at 850 K. Oxidation is another concern for high temperature structures. Conductively-cooled Si combustor tests were run at exit temperatures up to 1800 K [33]. The thickness of the oxide layers grown were in agreement with standard models of Si oxidation. These imply that uncoated Si airfoils can have a life of a few hundred hours. Longer lives may require coatings. Si nozzle guide vanes run for 5 hrs at 1600 K in a microcombustor exhaust show little degradation, Figure 24.

Silicon carbide has about 600 K more temperature capability than Si, but the SiC microfabrication technology is much less mature. SiC is available in single-crystal wafers and can be precision-etched but SiC wafers cost 100x more than Si at the moment and etch rates are about 10x slower. An alternative to direct SiC etching is to etch a female mold in Si and then fill the mold (for example by chemical vapor deposition, CVD), and dissolve away the Si, leaving an SiC precision structure. The challenges here are realizing SiC with the needed mechanical properties and dealing with the intrinsic stresses induced by the combination of the high temperatures of the CVD process and the difference in coefficient of thermal expansion between the two materials. A variation on this approach is to use CVD to fill cavities in Si wafers with SiC, bond another Si wafer over the filled cavity, and then process the pair as though it were a standard Si wafer. This yields SiC-reinforced silicon structures which have more temperature capability than Si but are easier to manufacture than SiC [49]. The increased temperature capability of a turbine like that in Figure 6 increases with the thickness of the SiC insert, Figure 25. Another approach being pursued is



Figure 24: 200 μm high, Si turbine blades new and after 5 hrs at 1600 K gas temperature in a microcombustor exhaust.



Figure 25: Usable strength of Si/SiC/Si hybrid structure in tension.

reaction sintering of powdered SiC to form parts such as turbine rotors [50].

Additional structural materials of interest for MEMS gas turbines include glasses for thermal and electrical isolation, and very high temperature materials such as sapphire. There is considerable microfabrication experience with glass but very little with the refractory ceramics because these have not been considered as MEMS materials in the past.

Structural Design Considerations

Structural design of a MEMS gas turbine has many of the same considerations as the design of large machines: basic engine layout is set by rotor dynamic considerations, centrifugal stress is the primary rotor load, stress concentrations must be avoided, and hot section life is creep- and oxidation-limited. Some large engine concerns do not exist at micron scale. For example, the microrotors are very stiff so that backbone bending is not a concern; thermal stress from temperature gradients is not important at these sizes; maintenance is not a design issue; and fasteners do not exist here so the engineering details involved with bolting, static sealing, etc. do not exist [51].

Although many of the design considerations are independent of size, the engineering values are not, of course. Airfoils need fillets at the roots to avoid stress concentrations with radii of 10-30 μ m. Surface finish is important with roughness measured in nanometers. Forced response excitation of blade rows must be avoided with blade-bending frequencies on the order of megahertz rather than kilohertz, the rotor once-per-rev frequency is 20 KHz rather than 200 Hz. For the turbine of Figure 6, the lowest blade mode is 2.5 MHz while the blade passing frequency is 0.9 MHz. Below 850-900 K, silicon is a brittle material so that probabilistic analysis is a preferred method for failure analysis. Such techniques applied to the turbine rotor geometry of Figure 6 at peripheral speeds of 500 m/s predict failure probabilities of 10^{-10} to 10^{-8} , depending upon the flaw population assumed [52]. In a rotor constructed from single-crystal Si, all of the flaws are likely to be surface flaws. In a rotor of CVD SiC, volumetric flaws may also exist. In either case, the flaw population and thus the usable strength of the material is a strong function of the manufacture, as it is at any scale. A variation of a factor of four in strength has been reported for deep-etched Si depending upon post-etch surface treatment [53].

Large engines use standard tubing fittings and electrical connectors to pass fluids and electrical signals to the outside world. These do not exist at microscale. Most computer chips and MEMS devices do not require fluid connections and those that do operate not much above room temperature. For these applications, there are a variety of adhesives and polymer systems. A micro-gas turbine can have a surface temperature above 700 K and require fluid connection at pressures of 10 atm or more, so that high strength, high temperature packaging approaches are needed. One approach which has proven successful is an adaptation of the hermetic package technology used for military electronics. This joins Kovar (a nickel alloy) tubing to silicon using glass as the bonding agent. The joining is done in a furnace above 200 atm [54].

ENGINE CONTROLS AND ACCESSORIES

All gas turbine engines require control systems to insure safe operation. Typically, the control system adjusts the fuel flow to deliver the requested power, and monitors engine operation to avoid unsafe conditions such as over-speed, over-temperature, or surge. Such a control system consists of sensors (speed, pressure, temperature, etc.), a feedback controller with a suitable set of control laws (now implemented in a digital computer), actuators such as a fuel control valve (often called a fuel management unit or FMU), and compressor stability devices as needed (bleed valves, fences, variable stators). Engine accessories include an ignition system, fuel pump, lubrication system, and starter. All of this functionality is needed for a millimeter-scale MEMS gas turbine and all must fit within a micro chip if the accessories are not to dwarf the engine. Following in the tradition of large engine development, the controls and accessories have received less attention to date than the major engine subsystems such as the compressor, turbine, and combustor but they are no less important to the ultimate success of the concept.

Engine Controls

The simplest engine control would consist of a single sensor feeding a digital controller which commands the fuel flow rate valves. The functional requirements for the valving and the sensors stem from the engine dynamics as represented in the control laws and from the engine environment. There is a very rich literature on MEMS sensors and valves, literally thousands of papers, and some units are commercially available. As for large engines, however, the combination of harsh environment, high frequency response, high accuracy, and high reliability means that sensors and actuators for MEMS-scale gas turbines must be specifically engineered for that environment.

Engine control laws are generally based on reduced order models of the engine dynamics. The dynamics of millimeterscale engines are, of course, much faster than larger engines and can also include phenomena not seen in large engines. The additional dynamics arise if there is significant heat transfer from the hot section into the compressor [55]. In this case the heat transfer degrades the compressor performance so that the pressure ratio and mass flow are a function of hot section temperature as well as shaft speed. Since the heat transfer has a time constant not much faster than the rotor acceleration, it alters the dynamics of the gas turbine from that of a first order system (as large engines are) to a second order system, requiring additional sophistication in the control law design. The best way of avoiding this complexity is to thermally isolate the hot and cold sections of the engine, which is, of course, desirable for improved thermodynamic performance.

Sensors

Large-scale gas turbines use compressor pressure ratio and/ or rpm as the primary input to the fuel control system. Sensor selection for a MEMS engine is a trade among observability of the state (dynamical information represented by the sensed quantity), response time, difficulty of fabrication, and environmental compatibility. Liu [55] used a dynamic model of a MEMS engine to evaluate the suitability of various sensor options including rpm and compressor discharge pressure or temperature. Of these, rpm is the most sensitive and temperature the least. Sensing is complicated by the high rotational frequencies (1,000,000 rpm) and high temperatures (600 K at the compressor discharge) in a very small engine. In principle, sensors can be fabricated integrally with an engine or located remotely, with each approach presenting challenges. A sensor remote from the gas path suffers reduced frequency response, which is already a challenge at the MEMS scale. In addition, it is difficult to be very remote from the gas path in an 0.5-mm-long engine. Integral sensors must withstand the high temperature of the gas path. Even for sensors fabricated in the cold sections, they must be capable of withstanding a wafer-processing environment that exceeds 1000-1400 K. This effectively precludes the use of low temperature materials such as polymers and most metals in the device design. Integral sensors have several advantages, however. They can be very small and thus have high frequency response, and many can be fabricated in parallel for low cost redundancy.

One integral solution was developed by Tang [56] who adapted a hot-film-type sensor to this application (Figure 26). Designed for placement on the wall above a rotor blade tip, this 50 μ m square sensor is a heated, serpentine, polysilicon resistor positioned over a trench for thermal isolation. The sensor and its



Figure 26: A 50 μ m sq hot film RPM and temperature sensor.

leadouts are all polysilicon which is selectively doped to adjust its resistivity (high for the sensor, low for the leads). Polysilicon has the advantage that it is compatible with most semiconductor fabrication techniques and can withstand high temperatures. Simulations confirmed by shock tube testing showed this approach to have sufficient sensitivity and frequency response to respond to the flow perturbations above a compressor blade tip as predicted by a 3-D CFD simulation. With a total thickness of less than 1 μ m, such sensors could be fabricated on the casing above the compressor blade tips. This type of resistor has also been shown to be usable as an igniter.

Fuel Control Valves

Very small engines are the topic of this discussion so the fuel control valves should be equivalently small. If integrated within the engine, the valve design must then be fully compatible with the fabrication and operating conditions of the gas turbine. This choice strongly constrains the valve design space. For example, the high processing and operating temperatures prohibit the use of polymers, so a hard valve seat must be used. The principal design requirements are flow rate, pressure, frequency response, very low power consumption and leakage, and high temperature capability.

Yang *et al.* [57] developed MEMS fuel-metering valves for gaseous hydrocarbon fuels such as propane. The design is a simple silicon spring-mounted plunger opened by electrostatic forces and closed by a combination of the spring and fluid pressure forces. The electrostatic approach has the advantages that very little power is needed to open a valve (40 nW) and the electrical materials (polysilicon) are compatible with high temperature semiconductor fabrication technology. Such a valve is shown in Figure 27. The 2-mm-square valve has a 1000- μ mdiameter plunger which rises 3 μ m off the seat when actuated. An 8-inch wafer of valves would contain about 5000 individual units. The valve opens against 10 atm pressure and, when open, flows 35 sccm of N₂ at a pressure drop of 0.5 atm. Frequency response is several hundred hertz. Cyclic testing of the valve has demonstrated a 50,000+ cycle life for the units tested. This design is on-off. The actuation-pressure scaling laws favor small valves so the intent is to use a parallel array of 20 on-off valves, each with a capacity of 5% of the maximum fuel flow, to meter the fuel. All 20 valves would consume less than 1 mW total power and operate at temperatures approaching 1000 K so they can be embedded in an engine chip. Many other arrangements are possible, such as logarithmic spacing of the valve orifice sizes to give finer fuel flow control.

Starter-Generator

Microelectrical machinery is required for power generation and electric starting, if desired. There is an extensive literature on microelectric motors, which is not reviewed here, but little work on generators. The requirements for the devices of interest here differ from previous work in that the power densities needed are at least two orders of magnitude greater than that of conventional-size and previous micromachines. Also, the thermal environment is much harsher. Integrating the electric machine within the engine offers the advantage of mechanical simplicity in that no additional bearings or structures are required over that needed for the fluid machinery. There is also a supply of cooling air available.

Both electric and magnetic machine designs can be considered and, to first order, both approaches can yield about equivalent power densities. Since the magnetic machines are material property-limited at high temperature and because of the challenges of microfabricating magnetic materials (which are not compatible with standard semiconductor manufacturing techniques), electric designs were first explored. Power density scales with electric field strength squared, frequency, and rota-



Figure 27: A 1 mm dia fuel control valve on Si beam springs.



Figure 28: Fields and charges in a microscale electric induction motor-generator.

tional speed. The micromachinery of interest here operates at peripheral velocities 1-2 orders of magnitude higher than previously reported micromotors, and so yields concomitantly more power. Electric machines may be configured in many ways. Here an induction design was chosen since it requires neither electrical contact with the rotor nor knowledge of the rotor position.

The operation of an electric induction machine can be understood with reference to Figure 28 [58]. The machine consists of two components, a rotor and a stator. The rotor is comprised of a 5-20 μ m thick good insulator covered with a few microns of a poor conductor (200 M Ω sheet resistivity). The stator consists of a set of conductive radial electrodes supported by an insulator. A traveling electric potential is imposed on the stator electrodes with the aid of external electronics. The resulting rotating electric field then induces an image charge on the rotor. Depending on the relative phase between the motion of rotor charges (set by the rotor mechanical speed) and that of the stator field (set by the external electronics), the machine will operate as a motor, generator, or brake. Torque increases with the square of the electric field strength and frequency. The maximum electric field strength that an air gap can maintain without breakdown is a function of the gap dimension. In air, the breakdown field is a maximum at a gap of a few microns so that micromachines can potentially realize higher power density than large machines of the same design. Frequency is constrained by external electronics design and by fabrication constraints on the stator electrode geometry. Current technology is limited to about 300 volts and 1-2 MHz. This is consistent with a 6-mm-diameter machine producing about 10 watts at a 3 μ m air gap. A 4-mm-diameter, six-phase, 131-pole (786 electrodes, each 4 μ m wide) stator for such a machine is shown in Figure 29 [59]. Note that such an electric motor-generator occupies less than 20 μ m thickness at the surface of the rotor and stator (mainly the insulator thickness). Thus the power density of this machine (excluding the external electronics) is many times that of a conventional magnetic motor-generator, on the order of 100 MW/m³. Fréchette et al. [60] have reported a similar design run as an electric motor. The torque produced by these devices has agreed with theoretical predictions but high power operation has yet to be reported.



Figure 29: A 131-pole, 6-phase, 4 mm dia electric induction stator.

To maximize power output, induction machines such as these require the spacing between the rotor and stator to be on the order of the stator pitch. The electrical torque produced scales with the square of the rotor-stator spacing, a few microns in this case. However, in these high-speed machines, the rotor periphery is at sonic velocity so the viscous drag in a gap of only 2-3 microns is extremely large. Indeed, this drag is the major loss mechanism for such an electrical machine. Thus, there is a basic design tradeoff for the electric motor-generator between power density and efficiency. While it may be possible to alter the local geometry to reduce the drag somewhat [24], the drag still makes up about half the total loss and limits the efficiency (shaft to net electrical) of these designs to 40-50%.

A magnetic induction machine has many fewer poles so that the optimum rotor-stator spacing is much larger (30-50 μ m) and the drag concomitantly lower. Koser and Lang [61] designed such a machine based on the microplating technology developed by Park *et al.* [62]. A four-pole stator for a 4-mm-diameter, induction motor-generator, designed to be functionally equivalent to the electric machine in Figure 29, is shown in Figure 30. More recent versions of this stator includes a laminated magnetic



(Courtesy of M. Allen)

Figure 30: A 4-pole stator for a 4 mm dia magnetic motorgenerator.

return path to reduce eddy current losses. The efficiency of this magnetic generator is calculated to approach 60%. It has the additional advantage that its external low frequency, low voltage electronics are easier to engineer and more compact than the high frequency, high voltage electronics of electric machines. Current stator materials are limited to only 500 K, however. Advanced materials may increase the operating temperature to 800 K. The rotor requires several hundred microns of iron for the magnetic path, which presents structural design challenges at 1,000,000 rpm. Magnetic machines will require careful thermal management when embedded within a MEMS gas turbine engine.

ENGINE DESIGN TRADES, COMPONENT INTEGRATION, AND DESIGN EVOLUTION

Gas turbine engines are more than a set of components bolted together. Rather, a successful gas turbine is a highly integrated system engineered to meet specific requirements, often with artful compromises between conflicting demands of the fluid, thermo, structural, and manufacturing engineering. A MEMS gas turbine is no different in this respect. The two dominant design considerations are the fabrication complexity and the thermodynamic cycle requirements. The principal challenge is to arrive at a design which meets the thermodynamic and component functional requirements while staying within the realm of realizable micromachining technology.

For any gas turbine, maximizing net engine output power includes maximizing both power per unit mass flow (specific power) and mass flow. Specific power is sensitive to component efficiency and pressure ratio, especially at the low pressure ratios under consideration here, 2:1 to 4:1. High pressure ratio in a single-stage centrifugal machine implies high wheel speeds; 500 m/s peripheral speed was chosen as the maximum compatible with geometrically simple Si construction. This also requires bearings and rotor dynamics capable of such high-speed operation. High efficiency implies optimal airfoil design and tight clearances. Tight clearances, in turn, imply high manufacturing precision, careful design for centrifugal and thermal growth, and robustness to the occasional high-speed rub. The other key to high specific power is high turbine inlet temperature (TIT). This requires high temperature materials for the turbine or cooling or both. High mass flows require high through-flow Mach numbers and large flow areas. With these requirements in mind, let's examine how these trades influence the design of a MEMS gas turbine. Specifically, the next section discusses why the demo engine design in Figures 3 and 4 is so configured. The principal geometric constraint on the design is that imposed by the current state-of-the-art in precision (micron-level accuracy) etch-depth capability of 300-500 μ m. We will start with the turbomachinery.

The engine design of Figure 3 requires six wafers: two form the rotor (and the annular combustor volume), and two are needed on each side of the rotor for the hydrostatic thrust bearings and their associated plumbing. Replacing the hydrostatic bearings with hydrodynamic ones would eliminate one wafer. Adding a generator mounted on a rotating compressor tip shroud would add one wafer. Adding control valves would add an additional wafer above the compressor. Such a complete, self-contained (other than electronics) gas turbine generator could thus be built with seven or eight wafers, not including external electronics.

In a centrifugal compressor and turbine of geometry similar to those in Figures 10 and 14, turning at 400-500 m/s tip speed, turbomachine design is constrained to a blade span of 400-500 μ m by fabrication technology and bending stress at the blade roots. Given the blade height, air flow increases with rotor diameter. Rotor diameter is constrained by the number of engines required per wafer and by etching uniformity (if a design goal is to avoid the necessity of dynamically balancing the rotor). Etch uniformity scales inversely with rotor area and blade height (i.e. mass flow is constrained by etch uniformity technology). Another factor influencing uniformity is rotor/stator airfoil count. Etch rate is a function of the local geometry, so for best circumferential uniformity there should be an even multiple of rotor and stator airfoils. This, of course, is deleterious to longterm vibrational life for the airfoils. If the rotor is fabricated from more then one wafer and etched with multiple masks, then mask and wafer alignment is an issue which favors the fewest possible wafers and masks. Rotor diameters up to 10 mm with 400-500 μ m blade spans, fabricated from 1-3 wafers, are consistent with current capabilities.

Bearing placement is another first-order concern. The bearings must support the load and remain in their stable operating regime. The load capability of air bearings scales with the bearing area. The bearing length is constrained by current etching technology to 300-500 μ m (and widths of 10-25 μ m), so load capacity scales with bearing diameter. For hydrostatic journal bearings of constant length, the optimum width of the bearing for maximum stable operating speed scales inversely with diameter. So as the bearing diameter is reduced, the air required to operate the bearing goes up (since the flow per unit area grows faster than the area shrinks) and the load capacity goes down. This implies that bearing diameter should be maximized, *i.e.* on the rotor diameter. Placing the bearing here also eliminates the need for a seal on the rotor, since in this case the leakage air is the air bearing fluid, but this approach increases bearing drag.

The cycle needs turbine inlet temperatures of at least 1400-1600 K to meet the application goals. These relatively high temperatures are needed due to the relatively high losses in the components and the secondary systems such as bearings. For example, the shaft mechanical losses are 4-8% of the shaft work for the design in Figures 3 and 4, ten times that of engines in the 5,000-30,000 lb thrust range. These temperatures are above the 900-950 K maximum operating temperature of a single-crystal Si rotor. In the engine design of Figure 3, the turbine rotor is conductively cooled through the compressor disk to the compressor air, which has a quite deleterious effect on the cycle performance requires thermally isolating the turbine from the compressor which can be accomplished by hollowing the shaft between them (100-200 μ m of the turbine disk in Figure 14). When this is done,

the temperature capability of the turbine rotor must be increased. As for large engines, some combination of increased material capability and improved cooling is needed. Improved material capability can be provided by reinforcing the Si turbine disk with SiC, with the temperature capability increasing with fraction of SiC. So-called "film cooling" (it is really a film insulation) is the most effective cooling technique here since, in this case, the heat is prevented from entering the solid.

As with large engines, improving performance at the MEMS scale will require some combination of more complex designs, better materials, and improved manufacturing technology. For example, cycle pressure ratios much over 4:1-5:1 will require multiple spools (probably not concentric). Higher turbine inlet temperatures will require improved cooling and fabrication technology for such materials as silicon carbide, silicon nitride, and sapphire. Improved propulsive efficiency can be realized with bypass engine designs. Better compression system performance will require more attention to thermal isolation. These are all physically possible but will require considerable investment in both the disciplinary and microfabrication technologies.

ONGOING TECHNICAL DEVELOPMENTS

As of this writing, no one has yet reported an operating gas turbine engine at this scale. Many of the turbomachinery efforts appear to be currently wrestling with rotor dynamics issues. At MIT, the requisite component technologies for the H_2 demo engine of Figures 3 and 4 have been demonstrated at the component level – turbine, bearings, combustor, etc – but a complete device fabricated to the design specification has yet to be tested. The challenge appears to be one mainly of stringent process control in a long, complex fabrication sequence, which is different in detail but in many ways as challenging as that for a microprocessor. Meanwhile, work is ongoing on improving component aerodynamic performance, hydrocarbon combustors, thermal isolation, high temperature materials (SiC), bearing system robustness, and electric and magnetic generators.

MEMS gas turbines differ not only in design and manufacture from conventional engines, they differ in development process as well. Specifically, since the engines are monolithic



Figure 31: Specific core power vs turbine rotor inlet temperature (after Koff).

blocks of silicon, they cannot be disassembled, reworked, and then reassembled during development. Instead, a new engine must be built from scratch. This is a process which takes about three months since even a seemingly minor change, for example increasing bearing clearance, may require some process development. Conceptually, the parallelism inherent in the MEMS manufacture could be an advantage in development. For example, if the optimum clearance was unknown, a wafer could be built with engines of various clearances. To date, this approach has not proven productive at MIT since the majority of the problems encountered have largely been unanticipated or attributable to inadequate quality control.

Future developments can be considered in three broad categories: evolution of MEMS gas turbines in both technology and application, other approaches to gas turbine engines in the centimeter-size range, and additional applications for millimeter-tocentimeter-sized devices based on the technologies discussed above.

The current MEMS engine design has a projected performance level comparable with gas turbines of the 1940's (Figure 31). At these low cycle pressure ratios, small improvements in pressure ratio and component efficiency have disproportionate returns of efficiency and output power (Figure 32). So it is likely that the performance of these devices can be increased, perhaps up to that of the 1950's. Some of the improvements will come from component evolution, others from new configurations or more complex cycles. The approaches of recuperated engines and combined cycles have been explored.

Combined Cycles

Cycles with heat exchangers are attractive at microscale because heat exchangers scale favorably as size is reduced.



Figure 32: Simple cycle performance variation for low pressure ratios, π (η = adiabatic efficiency, T_{t4} = turbine inlet temperature, T_{t2} = compressor inlet temperature).



Figure 33: Micron-scale counterflow heat exchanger.

MEMS heat exchangers have the advantage of high effectiveness and their repetitive structure is readily producible by microfabrication techniques. A micrograph of a high temperature heat exchanger is shown in Figure 33 [63]. There is now a large literature on MEMS heat exchangers, including both single- and multiple-phase devices, and their behavior [64].

Given the high temperatures and poor efficiencies of these very small gas turbines, their exhaust contains relatively large amounts of high quality heat. Thus, performance is greatly improved when this waste heat is productively utilized. Calculations suggest that both MEMS-combined cycles and co-generation cycles are feasible. One such steam-combined cycle system increases the net system output by about 50% over that of the gas turbine alone [65]. Cogen systems providing cooling or condensing water are obvious applications when the gas turbine is used for portable power generation.

Alternate Approaches

Several teams are working to build gas turbines in the few grams/sec mass flow range using a wider set of manufacturing technologies than MEMS, including conventional metal-forming techniques [66, 67, 37] and mold shape deposition manufacture [28]. Such an approach has its own set of advantages and disadvantages. One major advantage is the lack of an upper size constraint imposed by microfabrication technology, so that engines in the 100's of watts may be feasible. Its principal challenge compared to the MEMS approach may be one of manufacturing cost. However, there are many low-cost precision manufacturing technologies for small parts and a need such as this may foster new ones.

Additional Devices

The technology needed for a MEMS gas turbine is based on microscale high-speed rotating machinery including disci-



Figure 34: A 15 N (3.3 lb) thrust bipropellant liquid rocket engine.

plinary foundations in aerodynamics, combustion, materials, electromechanics, controls, and bearings. The same technologies can be applied to other microscale systems. One example is a motor-driven air blower or compressor [60]. Another example is a micro-bipropellant liquid rocket motor which is under development [68]. This propulsion system includes a regeneratively-cooled silicon thrust chamber and nozzle (Figure 34), turbopumps (Figure 35), and liquid control valves [69, 70]. The propellant flow rate is about 5 g/s. The silicon thrust chamber is designed to operate at chamber pressures of 125 atm and temperatures of 3000 K. The pressure in the liquid cooling jacket around the combustion chamber is above 200 atm. These devices operate at much higher power densities than those of the current gas turbine components and demonstrate that very high pressure, high power density silicon structures and devices are feasible. Preliminary tests of the cooled thrust chamber and turbopump are promising. In support of the engineering design of these devices, fundamental studies have been conducted on cavitation in micropumps [71], and the cooling behavior of a variety of rocket propellants in micro-geometries [72].



Figure 35: A 2.5 g/sec turbopump rotor (the pump is the inner blade row, the turbine the outer).

Pacing Technology

Microfabrication is the pacing technology in the development of MEMS gas turbine engines. Fortunately, the rate of progress and innovation in MEMS fabrication dwarfs those of the more mature technologies familiar to gas turbine developers. Much of this progress stems from the wide promise that MEMS offers for a variety of applications, justifying large research investments across a broad set of approaches [73]. Indeed, several fabrication limitations that existed at the start of this effort in the mid 1990's have since disappeared. Thus, many of the engineering design constraints now attributable to the microfabrication process limitations are likely be relaxed as new technology is developed.

ECONOMICS AND THE FUTURE OF MILLIMETER-SCALE GAS TURBINES

Whittle and Von Ohain were successful in their early jet engine developments because they set engineering goals based on the requirement of going fast, significantly faster than the piston-powered aircraft of the day [74]. In contrast, the gas turbine developers who based their requirements on competing on range with piston-powered propeller aircraft were unsuccessful at that time because their goal required much higher levels of performance, performance that was many years away. In this sense, we can ask what is required to make millimeter-scale gas turbines real - real in that machines are in production and making a discernable impact on society. There are two answers, one technical, the other economic. Many of the detailed technical issues were discussed above, but some must be considered in concert with the economics of power production. In the broadest sense, to be more than a curiosity, these very small gas turbine engines must fulfill societal needs. What are the possible applications and what levels of performance are needed for each? The answers must consider the alternative engineering solutions to each potential application.

The two major applications of these very small gas turbines mirror those of large engines, power production and aeronautical propulsion. Power production in the short term is aimed at portable applications where the very much larger energy density of a

Energy Source	Potential* Whr/kg	Practical Whr/kg** Based on Conversion Device Efficiency
Lithium battery	1,400	175 (LiSO ₂)
		300 (LiSOCl ₂)
TNT	1,400	NA
Methanol	6,200	1,500-3,100
Diesel fuel	13,200	1,320-5,000
Hydrogen	33,000	1,150-23,000
* Based on enthalpy		(After R. Paur

* Based on enthalpy

** (System conversion efficiency) x (Energy available in fuel)



Figure 36: Concept of a MEMS gas turbine engine packaged as a standard military battery.

hydrocarbon fuel compared to a lithium battery chemistry means that even a very inefficient gas turbine can be attractive (Table 3). In the short term, a 5-10% overall system efficiency (chemical to net electric power output) is sufficient to make a gas turbine engine solution an attractive alternative to a battery. Figure 36 is a concept of an engine packaged in the form factor of a high performance 50 W military battery (which costs about \$100). Most of the volume is fuel which implies that, in this application, specific fuel consumption is more important than specific power (since multiple engines could be packaged together).

A somewhat different compact power application is auxiliary power for flight vehicles, especially ones with high temperature environments. Highly redundant, distributed power for actuation, sensors, etc. may be attractive in many applications.

Competing with a 50-60% efficient central power plant requires much higher performance. Even including cogen use of the waste heat, a local gas turbine must have an efficiency of 20-30%, a level which looks many years away at this time. Should these levels be approached, then the redundancy and extreme quietness and compactness may make an array of millimeterscale machines an attractive solution. Emergency power applications would not require as high performance as base power and can be an attractive application for these small machines if the capital costs are sufficiently low.

In addition to compactness and redundancy, one advantage that millimeter-scale gas turbines do offer for many applications is that they are very, very quiet, even in large arrays. This stems from their high frequencies (blade passing is several hundred kilohertz, beyond the audible range) and short length scales (millimeter-diameter exhaust jets, which mix rapidly). The high frequency sound that is produced is relatively easy to muffle and quick to attenuate naturally.

Propulsion is an obvious application for very small engines since cubed-square scaling means that they can have very high thrust-to-weight ratios and be extremely compact. The US Defense Department is investing in reconnaissance airplanes with gross takeoff weights as low as 50-100 g. These aircraft have lift-to-drag ratios on the order of 5 so that an 0.1 N class thrust MEMS gas turbine such as that in Figure 3 is an attractive power plant. It is much better than a battery-powered electric solution, and IC engines in production are ten times too large, very noisy, and have poor fuel consumption. For subsonic flight applications, thrust and propulsive efficiency can be increased by adopting bypass engine configurations. High-speed flight may be another attractive application. Here the intrinsic high temperature capability (900 K) of even a silicon engine cold section means that the propulsion system can operate at high ambient temperatures. Configurations other than a gas turbine may also be realized, such as a ramjet or pulse detonation engine. Independent of the speed regime, clearly several engines can be used to realize increased thrust levels for larger vehicles. How large a thrust level is a function of how many engines can be practically assembled together (large phased array radars may use 10⁵ modules) and the performance of the millimeter-scale engines relative to that of large engines. Since it is unlikely that engines of this size will approach the fuel economy of large engines, their use may be restricted to short duration applications use such as lift engines, where very high thrust-to-weight ratio, compactness, and redundancy may command a premium.

Both gas turbine engines and complex semiconductor devices are very expensive to develop and there is no reason to believe that a marriage of the two will prove substantially less so. Development costs can be amortized over many units. Manufacturing cost, however, is always a major issue for mass-produced devices. There is vast experience on the production cost of semiconductor wafers in large-scale production, much less on MEMS. For example, a standard 200-mm-diameter silicon CMOS wafer from a foundry may cost \$600-1000 in large-scale production quantities (the industry is currently moving to 300mm-diameter wafers to reduce costs). A specialized wafer produced in small lots (such as GaA's used for radar and RF) may be ten times more expensive. The cost per chip is also a function of the number of chips per wafer (i.e. the size of the chip). Thus, a geometrically large, state-of-the-art microprocessor chip may cost \$200 to manufacture, while a small watch chip is only tens of cents. From this we infer that a wafer of MEMS gas turbines might produce several kilowatts of power and cost several thousand dollar to manufacture, putting the specific cost in the \$0.5-5 per watt range. On the low end this cost approaches that of large power plants.

If the more optimistic projections of their price and performance are achieved, then it is possible that these millimeter-scale machines and the technologies on which they are based may begin to transform the mechanical infrastructure of society in a manner similar to the changes in information handling wrought by the microprocessor. They will augment and even displace large central facilities. Distributed power, water treatment, and chemical processing may be realized, providing a more robust society. Many of the more significant changes this technology will engender are likely unrecognized at this time.

CONCLUDING REMARKS

Over the past eight years, research has started on very small gas turbine engines and related technologies. It is quite clear from the work to date that millimeter-scale gas turbine engines and similar high-speed rotating machinery, combustion, and energy conversion systems are technically feasible. One approach to realizing devices at these scales utilizes the semiconductor industry-derived micromachining technology known as MEMS. The economic impact of these devices will be dependent on the performance levels and the manufacturing costs, both of which have yet to be proven. It is certainly possible, however, that MEMS gas turbines may one day be competitive with conventional machines in a cost per installed kilowatt. Even at much higher costs, they will be very useful as compact power sources for portable electronics, equipment, and small vehicles.

ACKNOWLEDGEMENTS

Much of the work summarized herein is the intellectual accomplishment of a team at MIT including current and former research engineers and faculty – A. Ayon, K. Breuer, J. Brisson, F. Ehrich, R. Ghodssi, Y. Gong, S. Jacobson, R. Khanna, J. Lang, H. Li, C. Livermore, Y. Peles, M. Schmidt, S. Senturia, Z. Spakovszky, M. Spearing, C. Tan, S. Umans, I. Waitz, X. Zhang – and a large number of extraordinarily talented and hardworking graduate students, post-docs, and technicians. D. Park prepared the manuscript (and many, many others). It has been a great pleasure working with them all.

Special thanks goes to Drs. R. Paur and T. Doligalski of the US Army Research Office who were among the first to recognized the potential value of this technology and have nurtured it over the years.

The author would also like to thank Dr. J. L. Kerrebrock for 35 years as a mentor, colleague, and friend.

The research at MIT cited herein has been supported by the US Army and DARPA.

REFERENCES

[1] Gerendas, M., Pfister, R., 2000, "Development of a Very Small Aero-Engine," ASME Paper 2000-GT-0536, ASME Turbo Expo, Munich, Germany.

[2] Nakajima, T., Fukikawa, Y., Goto, T., Iio, M., 1995, "The Development of the Micro Gas Turbine Generator," 1995 Yokohama International Gas Turbine Congress, Yokohama, Japan.

[3] Senturia, S., 2001, *Microsystem Design*, Kluwer Academic Publishers, Boston, MA.

[4] Jensen, K.F., 2001, "Microreaction Engineering - Is Small

Better?" Chemical Engineering Science, 56, pp. 293-303.

[5] Epstein, A.H., and Senturia, S.D., 1997, "Macro Power from Micro Machinery", *Science*, **276**, p. 1211.

[6] Epstein, Senturia, Al-Midani, Anathasuresh, Ayón, Breuer, Chen, Ehrich, Esteve, Fréchette, Gauba, Ghodssi, Groshenry, Jacobson, Kerrebrock, Lang, Lin, London, Lopata, Mehra, Mur Miranda, Nagle, Orr, Piekos, Schmidt, Shirley, Spearing, Tan, Tzeng, and Waitz, 1997, "Micro-Heat Engines, Gas Turbines, and Rocket Engines", AIAA 97-1773, presented at 28th AIAA Fluid Dynamics Conference, 4th AIAA Shear Flow Control Conference, Snowmass Village, CO.

[7] Groshenry, C., 1995, "Preliminary Design Study of a Micro-Gas Turbine Engine," M.S. Thesis, MIT Department of Aeronautics and Astronautics.

[8] Park, D-E, Lee, D-H, Yoon, J-B, Kwon, S., and Yoon, E., 2002, "Design and Fabrication of Micromachined Internal Combustion Engine as a Power Source for Microsystems," *Technical Digest of MEMS 2002*, the 15th IEEE International Conference on Micro Electro Mechanical Systems, Las Vegas, NV, pp. 272-275.

[9] Jacobson, S.A., 1998, "Aerothermal Challenges in the Design of a Microfabricated Gas Turbine Engine", AIAA 98-2545, 29th AIAA Fluid Dynamics Conference, Albuquerque, NM.

[10] Spearing, M.S., 2000, "Materials Issues in Microelectromechanical Systems (MEMS)," *Acta Mater.*, **48**, pp. 179-196.

[11] Hawthorne, W.R., 1994, "Reflections on United Kingdom Aircraft Gas Turbine History," *J. of Eng. for Gas Turbines and Power*, **116**, pp. 495-510.

[12] Madou, M., 1997, *Fundamentals of Microfabrication*, CRC Press, Boca Raton, FL.

[13] Ayón, A.A., Lin, C.C., Braff, R., Bayt, R., Sawin, H.H. and Schmidt, M., 1998, "Etching Characteristics and Profile Control in a Time Multiplexed Inductively Coupled Plasma Etcher," 1998 Solid State Sensors and Actuator Workshop, Hilton Head, SC.

[14] Lin, C.C., 1999, "Development of a Microfabricated Turbine-Driven Air Bearing Rig," Ph.D. Thesis, MIT Department of Mechanical Engineering.

[15] Lin, C.C., Ghodssi, R., Ayón, A.A., Chen, D.Z., Jacobson, S., Breuer, K.S., Epstein, A.H., and Schmidt, M.A., 1999, "Fabrication and Characterization of a Micro Turbine/Bearing Rig", presented at MEMS '99, Orlando, FL.

[16] Mirza, A.R., and Ayón, A.A., 1999, "Silicon Wafer Bonding for MEMS Manufacturing," *Solid State Technology*, **42**, pp. 73-78.

[17] Miki, N., Zhang, X., Khanna, R., Ayón, A. A., Ward, D., and Spearing, S.M., 2002, "A Study of Multi-stack Silicon-direct

Wafer Bonding for MEMS Manufacturing" The Fifteenth IEEE International Conference on Micro Electro Mechanical Systems (MEMS2002), Las Vegas, NV, pp. 407-410.

[18] Mehra, A., Ayón, A.A., Waitz, I.A., and Schmidt, M.A., 1999, "Microfabrication of High Temperature Silicon Devices Using Wafer Bonding and Deep Reactive Ion Etching", *IEEE/ ASME J. of Microelectromechanical Systems*, **8**, pp. 152-160.

[19] Fréchette, L.G, Jacobson, S.A., Breuer, K.S., Ehrich, F.F., Ghodssi, R., Khanna, R., Wong, C.W., Zhang, X., Schmidt, M.A., and Epstein, A.H., 2000, "Demonstration of a Microfabricated High-Speed Turbine Supported on Gas Bearings," Hilton Head Solid-State Sensor & Actuator Workshop, Hilton Head Island, SC, pp. 43-47.

[20] Ghodssi, R., 2003, private communication.

[21] Mehra, A., Jacobson, S.A., Tan, C.S., and Epstein, A.H., 1998, "Aerodynamic Design Considerations for the Turbomachinery of a Micro Gas Turbine Engine", presented at the 25th National and 1st International Conference on Fluid Mechanics and Power, New Delhi, India.

[22] Mehra, A., 1997, "Computational Investigation and Design of Low Reynolds Number Micro-Turbomachinery," M.S. Thesis, MIT Department of Aeronautics and Astronautics.

[23] Shirley, G., 1998, "An Experimental Investigation of a Low Reynolds Number, High Mach Number Centrifugal Compressor," M.S. Thesis, MIT Department of Aeronautics and Astronautics.

[24] Fréchette, L.G., 2000, "Development of a Microfabricated Silicon Motor-Driven Compression System," Ph.D. Thesis, MIT Department of Aeronautics and Astronautics.

[25] Sirakov, B., 2003, private communication.

[26] Isomura, K., Murayama, M., Yamaguchi, H., Ijichi, N., Asakura, H., Saji, N., Shiga, O., Takahashi, K., Kawakubo, T., Watanabe, T., Yamagata, A., Tange, H., Tanaka, S., Genda, T., and Esashi, M., 2002, "Design Study of a Micromachined Gas Turbine with Three-Dimensional Impeller," presented at the 9th International Symposium on Transport Phenomena and Dynamics of Rotating Machinery, Honolulu, HI.

[27] Isomura, K., Murayama, M., Yamaguchi, H., Ijichi, N., Asakura, H., Saji, N., Shiga, O., Takahashi, K., Tanaka, S., Genda, T., and Esashi, M., 2002, "Development of Microturbocharger and Microcombustor for a Three-Dimensional Gas Turbine at Microscale," ASME Paper GT-2002-30580, presented at ASME Turbo Expo, Amsterdam.

[28] Kang, S., Johnston, J.P., Arima, T., Matsunaga, M., Tsuru, H., and Prinz, F.B., 2003, "Micro-Scale Radial-Flow Compressor Impeller Made of Silicon Nitride, Manufacturing and Performance," ASME Paper GT2003-38933, *Proc. ASME* Turbo Expo 2003, Atlanta, GA.

[29] Gong, Y., 2002, private communication.

[30] Philippon, B., 2001, "Design of a Film Cooled MEMS Micro Turbine," M.S. Thesis, MIT Department of Aeronautics and Astronautics.

[31] Spadaccini, C.M., Mehra, A., Lee, J., Lukachko, S., Zhang, X., and Waitz, I.A., 2002, "High Power Density Silicon Combustion Systems for Micro Gas Turbine Engines," GT-2002-30082, ASME International Gas Turbine Institute TURBO EXPO '02, Amsterdam, The Netherlands.

[32] Waitz, I.A., Gautam, G., and Tzeng, Y.-S., 1998, "Combustors for Micro-Gas Turbine Engines," *ASME Journal of Fluids Engineering*, **120**.

[33] Mehra, A., Zhang, X., Ayón, A.A., Waitz, I.A., Schmidt, M.A., and Spadaccini, C.M., 2000, "A 6-Wafer Combuston System for a Silicon Micro Gas Turbine Engine," *Journal of MicroElectro-Mechanical Systems*, **9**, pp. 517-527.

[34] Zhang, X., Mehra, A., Ayón, A.A., and Waitz, I.A., 2002, "Development of Polysilicon Igniters and Temperature Sensors for a Micro Gas Turbine Engine," *Technical Digest of MEMS* 2002, the 15th IEEE International Conference on Micro Electro Mechanical Systems, Las Vegas, NV, pp. 280-283.

[35] Spadaccini, C.M., Zhang, X., Cadou, C.P., Miki, N., and Waitz, I.A., 2002, "Development of a Catalytic Silicon Micro-Combustor for Hydrocarbon-Fueled Power MEMS," *Technical Digest of MEMS 2002*, The 15th IEEE International Conference on Micro Electro Mechanical Systems, Las Vegas, NV, pp. 228-231.

[36] Takahashi, K., Murayama, M., Isomura, K., and Tanaka, S., 2002, "Development of a Methane Fueled Combustor for Micro-Scaled Gas Turbine," *Technical Digest of Power MEMS 2002*, Tsukuba, Japan, pp. 44-46.

[37] Matsuo, E., Yoshiki, H., Nagashima, T., and Kato, C., 2002, "Development of Ultra Micro Gas Turbines," *Technical Digest* of Power MEMS 2002, Tsukuba, Japan, pp. 36-39

[38] Mur Miranda, J.O., 1997, "Feasibility of Electrostatic Bearings for Micro Turbo Machinery," M.Eng. Thesis, MIT Department of Electrical Engineering and Computer Science.

[39] Walton, J.F., and Hesmat, H., 2002, "Application of Foil Bearings to Turbomachinery Including Vertical Operation," *Trans. ASME*, **124**, pp. 1032-1039.

[40] Spakovszky, Z.S., 2003, "Scaling Laws for Ultra-Short Hydrostatic Gas Journal Bearings", to be presented at the 19th Biennial Conference on Mechanical Vibration and Noise.

[41] Breuer, K., Ehrich, F., Fréchette, L., Jacobson, S., Lin, C-C, Orr, D.J., Piekos, E., Savoulides, N., and Wong, C-W,

2003, "Challenges for Lubrication in High Speed MEMS," in *Nanotribology*, Hsu and Ying (eds), Kluwer Academic Press.

[42] Liu, L., Teo, C.J., Miki, N., Epstein, A.H., and Spakovszky, Z.S., 2003, "Hydrostatic Gas Journal Bearings for Micro-Turbomachinery", to be presented at the 19th Biennial Conference on Mechanical Vibration and Noise.

[43] Ehrich, F.F., and Jacobson, S.A., 2003, "Development of High Speed Gas Bearings for High-Power-Density Micro-Devices", *J. of Eng. for Gas Turbines and Power*, **125**, pp. 141-148.

[44] Miki, N., Teo, C.J., Ho, L., and Zhang, X., 2002, "Precision Fabrication of High-Speed Micro-Rotors Using Deep Reactive Ion Etching (DRIE)," Solid-State Sensor, Actuator and Microsystems Workshop, Hilton Head Island, SC.

[45] Piekos, E.S. and Breuer, K.S. 1998, "Pseudospectral Orbit Simulation of Non-Ideal Gas-Lubricated Journal Bearings for Microfabricated Turbomachines," Paper No. 98-Trib-48, presented at the Joint ASME/STLE Tribology Conference, Toronto, Canada; also to appear in *Journal of Tribology*.

[46] Wong, C.W., Zhang, X., Jacobson, S.A., and Epstein, A.H., 2002, "A Self-Acting Thrust Bearing for High Speed Micro-Rotors," *Technical Digest of MEMS 2002*, the 15th IEEE International Conference on Micro Electro Mechanical Systems, Las Vegas, NV, pp. 276-279.

[47] Chen, K-S, Ayón, A., and Spearing, S.M., 1999, "Controlling and Testing the Fracture Strength of Silicon at the Mesoscale", to be published in the *Journal of the American Ceramic Society*.

[48] Moon, H.-S., Anand, L., and Spearing, S.M., 2002, "A Constitutive Model for the Mechanical Behavior of Single Crystal Silicon at Elevated Temperature", *Mat. Res. Soc. Symp. Proc.*, **687**, B9.6.1.

[49] Choi, D., Shinavski, R.J., Steffier, W.S., Hoyt, S., and Spearing, S.M., 2001, "Process Development of Silicon-Silicon Carbide Hybrid Micro-Engine Structures", *Materials Research Society Symposium Proceedings*, **687**.

[50] Sugimoto, S., Tanaka, S., Li, J-F, Watanabe, R., and Esashi, M., 2000, "Silicon Carbnide Micro-Reaction-Sintering Using a Multilayer Silicon Mold," *IEEE Transducers*, pp. 775-780.

[51] Spearing, S.M., and Chen, K.S., 1997, "Micro-Gas Turbine Engine Materials and Structures", presented at 21st Annual Cocoa Beach Conference and Exposition on Composite, Advanced Ceramics, Materials and Structures.

[52] Chen, K-S, Spearing, S.M., and Nemeth, N.N., 2001, "Structural Design of a Silicon Micro-Turbo-Generator," *AIAA Journal*, **39**, pp. 720-728.

[53] Chen, K-S, Ayón, A.A., Lohner, K.A., Kepets, M.A., Melconian, T.K., and Spearing, S.M., 1998, "Dependence of

Silicon Fracture Strength and Surface Morphology on Deep Reactive Ion Etching Parameters", presented at the MRS fall Meeting, Boston, MA.

[54] Harrison, T.S., London, A.P., Spearing, S.M., 2001, "High Temperature, High Pressure Fluid Connections for Power Micro-Systems," Paper EE6.5, *Mat. Res. Soc. Symp. Proc.*, p. 654.

[55] Liu, C., 2000, "Dynamical System Modeling of a Micro Gas Turbine Engine," MS Thesis, MIT Department of Aeronautics and Astronautics.

[56] Tang, D., 2001, "Rotor Speed Microsensor for the MIT Microengine," M.S. Thesis, MIT Department of Mechanical Engineering.

[57] Yang, X., Holke, A., and Schmidt, M.A., 2002, "An Electrostatic, On/Off MEMS Valve for Gas Fuel Delivery of a Microengine," Solid-State Sensor, Actuator and Microsystems Workshop, Hilton Head Island, SC.

[58] Nagle, S.F., and Lang, J.H., 1999, "A Micro-Scale Electric-Induction Machine for a Micro Gas Turbine Generator," 27th Annual Meeting of the Electrostatics Society of America.

[59] Ghodssi, R., Fréchette, L.G., Nagle S.F., Zhang X., Ayón A.A., Senturia S.D., and Schmidt M.A., 1999, "Thick Buried Oxide in Silicon (TBOS): An Integrated Fabrication Technology for Multi-Stack Wafer-Bonded MEMS Processes," *Proceedings of the 1999 International Conference on Solid-State Sensors and Actuators*, Sendai, Japan, pp. 1456-1459.

[60] Fréchette, L.G., Nagle, S.F., Ghodssi, R., Umans, S.D., Schmidt, M.A., and Lang, J.H., 2001, "An Electrostatic Induction Micromotor Supported on Gas-Lubricated Bearings," IEEE 14th International Micro Electro Mechanical Systems Conference, MEMS 2001, Interlaken, Switzerland.

[61] Koser, H., and Lang, J. H., 2000, "Modeling a High Power Density MEMS Magnetic Induction Machine," *Proc. Fourth International Conference On Modeling and Simulation of Microsystems*, Hilton Head, SC, pp. 286-289.

[62] Park, J.W., Cros, F., Allen, M.G., 2002, "A Sacrificial Layer Approach to Highly Laminated Magnetic Cores," MEMS 2002 IEEE International Conference, Las Vegas, NV, pp. 380-383.

[63] Sullivan, S., Zhang, X., Ayón, A.A., Brisson, J.G., 2001, "Demonstration of a Microscale Heat Exchanger for a Silicon Micro Gas Turbine Engine," *11th International Conference on Solid-State Sensors and Actuators, Transducers 01*, Technical Digest, Munich, Germany, pp. 1606-1609.

[64] Harris, C., Kelly, K., Wang, T., McCandless, A., and Motakef, S., 2002, "Fabrication, Modeling, and Testing of Micro-Cross-Flow Heat Exchangers," *J. of Microelectromechanical Systems*, **11**, pp. 726-735.

[65] Cui, L., 2003, "Design and System Analysis of Micro-Scale Rankine Cycle Power Systems," M.S. Thesis, MIT Department of Aeronautics and Astronautics.

[66] Isomura, K., Murayama, M., and Kawakubo, T., 2001, "Feasibility Study of a Gas Turbine at Micro Scale," ASME Paper 2001-GT-101, presented at ASME Turbo Expo.

[67] Isomura, K., Murayama, M., Yamaguchi, H., Ijichi, N., Saji, N., Shiga, O., Tanaka, S., Genda, T., Hara, M., and Esashi, M., 2002, "Component Development of Micromachined Gas Turbine Generators," *Technical Digest of Power MEMS 2002*, Tsukuba, Japan, pp. 32-35.

[68] London, A.P., Epstein, A.H., and Kerrebrock, J.L., 2001, "A High Pressure Bipropellant Microrocket Engine," *AIAA J. of Propulsion and Power*, **17**, pp. 780-787.

[69] Deux, A., 2001, "Design of a Silicon Microfabricated Rocket Engine Turbopump," M.S. Thesis, MIT Department of Aeronautics and Astronautics.

[70] Jamonet, L., 2002, "Testing of a Microrocket Engine Turbopump," M.S. Thesis, MIT Department of Aeronautics and Astronautics.

[71] Pennathur, S., Peles, Y., and Epstein, A.H., 2002, "Cavitation at Micro-Scale in MEMS Fluid Machinery," Paper IMECE 2002-33328, presented at ASME Int. M.E. Conference & Expo, New Orleans, LA.

[72] Joppin, C., 2002, "Cooling Performance of Storable Propellants for a Micro Rocket Engine," M.S. Thesis, MIT Department of Aeronautics and Astronautics.

[73] National Research Council, 2002, *Implications of Emerging Micro- and Nanotechnologies*, The National Academies Press, Washington, DC.

[74] Golley, J., *Genesis of the Jet*, 1996, B. Gunston, Tech. Ed., Shrewsbury, England.