

SHIRTBUTTON-SIZED GAS TURBINES: THE ENGINEERING CHALLENGES OF MICRO HIGH SPEED ROTATING MACHINERY

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ABSTRACT

MIT is developing micro-electro-mechanical systems (MEMS)-based gas turbine engines, turbogenerators, and rocket engines. Fabricated in large numbers in parallel using semiconductor manufacturing techniques, these engines-on-a-chip are based on micro-high speed rotating machinery with power densities approaching those of their more familiar, full-sized brethren. The micro-gas turbine is a 2 cm diameter by 3 mm thick Si or SiC heat engine designed to produce about 10 W of electric power or 0.1 N of thrust while consuming about 15 grams/hr of H₂. Later versions may produce up to 100 W using hydrocarbon fuels. This paper gives an overview of the project and discusses the challenges faced in the design and manufacture of high speed microrotating machinery. Fluid, structural, bearing, and rotor dynamics design issues are reviewed.

INTRODUCTION

High speed rotating machinery comes in many sizes. In recent years much emphasis has been placed on the large end of the business – 10 m diameter hydroelectric turbines, 300 ton ground-based gas turbine generators, 3 m diameter aircraft engines. These machines are engineered to produce hundreds of megawatts of power. The focus of this paper is the opposite end of the rotating machine size scale, devices a few millimeters in diameter and weighing a gram or two. These machines are about one thousandth the linear scale of their largest brethren and thus, since power level scales with fluid mass flow rate and flow rate scales with intake area, they should produce about one millionth the power level, a few tens of watts.

The interest in rotating machinery of this size range 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 micron size range with micron and even submicron precision. Such parts are produced using photolithography defined features and many can be made simultaneously, holding out the promise of low production cost. Such assemblies are known as micro-electrical-mechanical systems (MEMS) and have been the subject of thousands of publications over the last decade. Early work in MEMS focused on sensors and many devices based on this technology are in large scale production (such as pressure sensors and airbag accelerometers for automobiles). More recently, work has been done on actuators of various sorts. Fluid handling is receiving attention as well, for example MEMS valves are commercially available.

The user pull is predominately one of electric power. There is proliferation of small, portable electronics – computers, digital assistants, cell phones, GPS receivers, etc. – which require compact energy supplies. The demand is for 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 robots and air vehicles. These small, and in some cases very small, systems require increasing compact power and propulsion.

For compact power production, hydrocarbon fuels burned in air have 20-30 times the energy density of the best current lithium chemistry-based batteries. Thermal cycles and high speed rotating machinery offer high power density compared to other power production schemes and MEMS technology is advancing rapidly. Recognizing these trends, a group at MIT began research in the mid 1990's on a MEMS-based "micro-gas turbine generator" capable of producing tens of watts of electrical power

from a cubic centimeter-sized package (Epstein and Senturia, 1997; Epstein *et al.*, 1997). Since that time, related efforts have been started on a micromotor-driven air compressor and a bi-propellant, liquid rocket motor which utilize much of the same technology as the gas turbine. The attractiveness of these devices is predicated to a large part on their high power density. The power density is a function of the design of the rotating machinery. This paper reports on this work in progress, with emphasis on the rotating machinery development.

SYSTEM DESIGN CONSIDERATIONS

At length scales of a few millimeters, thermodynamic considerations are no different than for much larger devices. Thus, high power density for a simple Brayton cycle requires high combustor exit temperatures (1400-1800K) and pressure ratios above 2 and preferably above 4. This can be seen in the thermodynamic cycle calculation illustrated in Figure 1 which shows that several tens of watts can be expected from a machine with a 1 mm² intake area. The power density of rotating machinery, both fluid and electric, scale with the square of the peripheral speed, as does the stress in the rotor. Thus, high power density implies highly stressed rotating structures. Peripheral speeds of 300-600 m/s are needed to achieve pressure ratios in the 2:1-5:1 per stage range, assuming centrifugal turbomachinery. This implies rotor centrifugal stresses on the order of hundreds of megaPascals. These peripheral speeds in rotors a few millimeters in diameter require rotational rates on the order of 1-3 million rpm. Thus, low friction bearings are needed. Also, high speed rotating machinery generally requires high precision manufacturing to maintain tight clearance and good balance. For millimeter-sized machines to have the same fractional precision as meter-sized devices, the geometric precision requirements are on the order of a micron.

A Brayton cycle is not the only choice for power production from a MEMS-based thermodynamic cycle. Rankine, Stirling, and Otto cycles can all be considered candidates. The advantages and disadvantages of each differ. The principal advantages of the Brayton cycle are simplicity (only one moving part, a rotor, is needed), highest power density (due to the high throughflow Mach number and thus high mass flow per unit area), and the availability of compressed air for cooling and other uses. The primary disadvantage is that a minimum component efficiency (on the order of 40-50%) must be met for the cycle to be

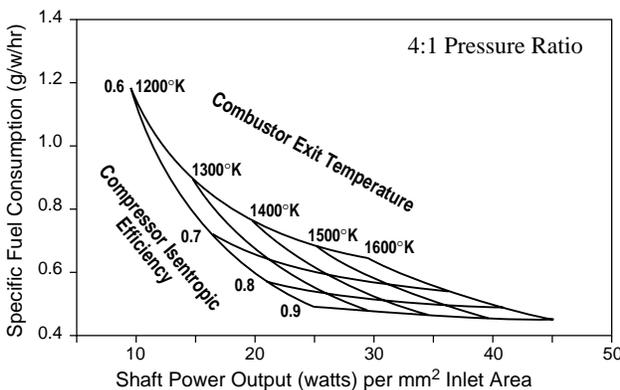


Figure 1: Simple cycle gas turbine performance with H₂ fuel.

Table 1: Micro vs. Macro Material Properties

	Ni-based Super Alloys	Titanium Alloys	Macro (Micro) Ceramics	Micro Silicon
Centrifugal Stress [$\sqrt{\sigma_f/\rho}$] (m/s)	330	420	420 (670)	1000
Thermal Stress [$\alpha E/\sigma_{fy}$]	2.7×10^{-3}	1.2×10^{-3}	2.0×10^{-3} (1.1×10^{-3})	0.9×10^{-3}
Stiffness [E/ρ] (MPa/Kgm ⁻³)	~26	~25	~95	~70
Max Temp (°C) limiting factor	~1000 (creep)	~300 (strength)	~1500 (oxidation)	~600 (creep)

self-sustaining; only then can net power be produced. As this is a first-of-its-kind effort that challenges the capabilities of several disciplines, especially microfabrication, simplicity is a very desirable virtue. The Brayton cycle seems most attractive in this regard.

Mechanics Scaling

Thermodynamic considerations do not change as machines become smaller, but mechanics considerations do. Structural mechanics, fluid mechanics, and electromechanics all change in a manner important to machine performance and design as length scale is decreased by a factor of order 1000.

For structural mechanics, it is the change in material properties with length scale that is most important. A relatively small set of materials are accessible to current microfabrication technology. The most commonly used by far is Silicon (Si), while Silicon Carbide (SiC) and Gallium Arsenide (GaAs) are used to date mainly in niche applications. For many of the metrics important to high speed rotating machinery, Si and SiC are superior to most commonly used metals such as steel, titanium, and nickel-based superalloys (Spearing and Chen, 1997). This can be seen in Table 1 which compares materials in terms of properties important for centrifugal stress, thermal stress, vibrations, and hot strength (Figure 2) (Mehra, 2000). Materials such as Si and SiC are not used in conventional-sized rotating machinery because they are brittle. Their usable strength is dominated by flaws introduced in manufacturing and flaw population generally scales with part volume. However, these materials are available in the form required for semiconductor manufacture (thin wafers) with an essentially perfect, single crystal structure. As such, they have high usable strength, values after micromachining above 4 GPa have been reported (Chen *et al.*, 1998), several times higher than that of rotating machinery metallic alloys. This higher strength can be used either to realize higher rotation speeds (and thus higher power densities) at constant geometry, or to simplify the geometry (and thus the manufacturing) at constant peripheral speed. To date, we have adopted the later approach for expediency. An additional materials consideration is that thermal shock increases with length scale. Thus, materials which have very high temperature capabilities but are not considered high temperature structural ceramics (such as alumina or sapphire) due to their susceptibility to thermal shock, are viable at the millimeter and below length scales. Since these have not

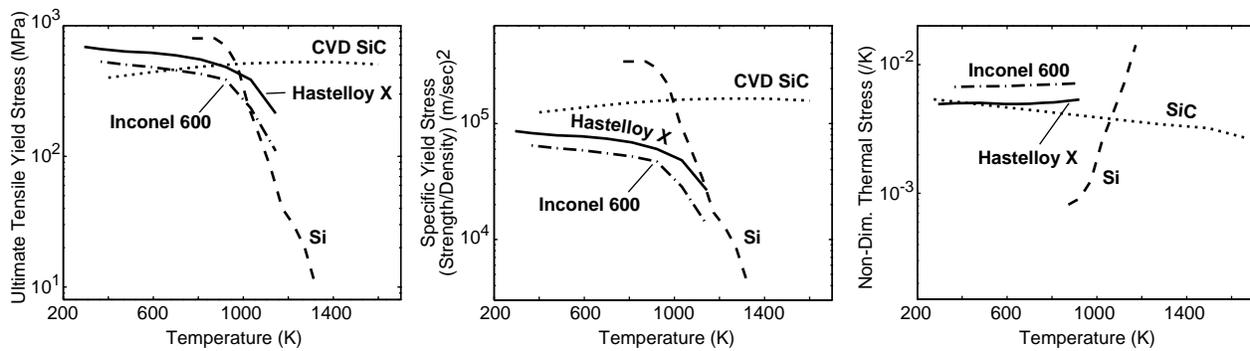


Figure 2: Material properties relevant to rotating machinery.

been considered as MEMS materials in the past, there is currently little suitable manufacturing technology available for these materials (Spearing and Chen, 1997).

Fluid Mechanics Scaling

Fluid mechanics are also scale dependent (Jacobson, 1998). One aspect is that viscous forces are more important at small scale. Pressure ratios of 2:1-4:1 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 Reynolds number of a few thousand. These are small values compared to the 10^5 - 10^6 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 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 Knudsen number considerations are not important.

Heat transfer is another aspect of fluid mechanics in which micro devices 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 that the 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 conductivity 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.

Fabrication Considerations

Compared to manufacturing technologies familiar at large scale, current microfabrication technology is quite constrained in the geometries that can be produced. The primary fabrication tool is etching of photolithographically-defined planar geometries. The resultant shapes are mainly prismatic or "extruded" (Ayon *et al.*, 1998). Conceptually, 3-D shapes can be constructed of multiple precision-aligned 2-D layers. To this end, Si wafers

can be stacked and diffusion-bonded with bond strength approaching that of the native material (Mirza and Ayon, 1999). But layering is expensive with current technology and 10 is considered a large number of precision-aligned layers for a micro device (wafers can currently be aligned to about 1 micron). Thus 3-D rotating machine geometries are difficult to realize so that 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 (wafers range from 100 to 300 mm in diameter and may contain dozens or hundreds of devices). These are much different manufacturing constraints than common to the large-scale world so it is not surprising the optimal machine design may also be different.

Two-dimensionality is not a crippling constraint on the design of high speed rotating machinery. Figure 3 is an image of a 4 mm rotor diameter, radial inflow turbine designed to produce 60 watts of mechanical power at a tip speed of 500 m/s (Lin *et al.*, 1999). The airfoil height is 200 microns. The cylindrical structure in the center is a thrust pad for an axial air bearing. The circumferential gap between the rotor and stator blades is a 15 micron wide air journal bearing required to support the radial loads. The trailing edge of the rotor blades are 25 microns thick (uniform to within 0.5 microns) and the blade roots have 10 micron radius fillets for stress relief. This Si structure was produced by deep reactive ion etching (DRIE). While the airfoils appear planar in the figure, they are actually slightly tapered

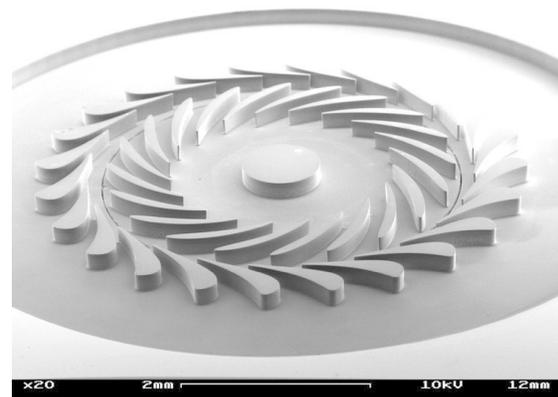


Figure 3: 4 mm rotor diameter radial inflow turbine.

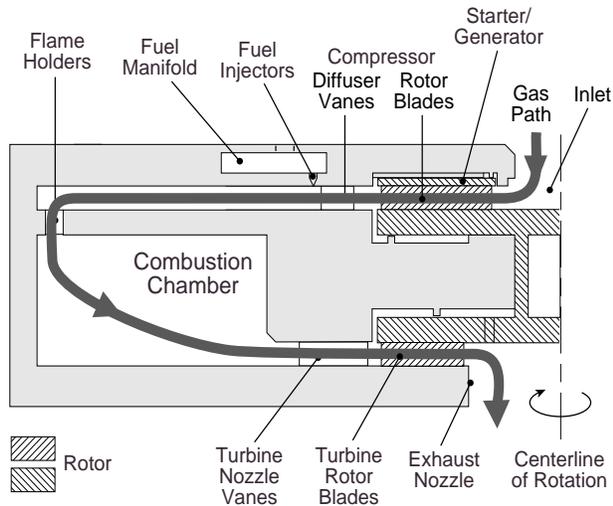


Figure 4: Baseline design microengine cross-section.

from hub to tip. Current technology can yield a taper uniformity of about 30-50:1 with either a positive or negative slope. The constraints on airfoil heights are the etch rate (about 3 microns per minute) and centrifugal bending stress at the blade root. Turbomachines of similar geometry have been produced with blade heights of over 400 microns.

The effort described herein has been focussed on micromachinery which are produced with semiconductor fabrication technology (MEMS). Other manufacturing techniques may be feasible as well, especially as the device size grows into the centimeter range. The MEMS approach was chosen here because it is intrinsically high precision and parallel production, offering the promise of very low cost in large quantity production. Initial estimates suggest that the cost per unit power might ultimately approach that of large gas turbine engines.

GAS TURBINE ENGINE

Considerations such as those discussed above led in 1996 to the preliminary or "baseline" gas turbine engine design illustrated in Figure 4. The 1 cm diameter engine is a single-shaft arrangement with a centrifugal compressor and radial inflow turbine, separated by a hollow shaft for thermal isolation, and supported on air bearings. At a tip speed of 500 m/s, the adiabatic pressure ratio is about 4:1. The compressor is shrouded and an electrostatic starter generator is mounted on the tip shroud. The combustor premixes hydrogen fuel and air upstream of flame holders and burns lean (equivalence ratio 0.3-0.4) so that the combustor exit temperature is 1600 K, within the temperature capabilities of an uncooled SiC turbine. The design philosophy was to use a high turbine inlet temperature to achieve acceptable work per unit air flow, recognizing that component efficiencies would be relatively low and parasitic losses high. With a 4 mm rotor diameter, the unit was sized to pump 0.15 gram/sec of air and produce 10-20 watts of power at 2.4 million rpm. The engine is constructed from 8 wafers, diffusion-bonded together. The turbine wafer was assumed to be SiC. This design served as a baseline for the research in component technologies described in later sections.

One primary goal of the project is to show 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. By 1998, the requisite technologies were judged sufficiently advanced to begin building such an engine with the exception of fabrication technology for SiC. 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 turbine cooling is required. 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 4:1 to 2:1 with a concomitant decrease in cycle power output and efficiency. This expedient arrangement is 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.

The H₂ demo engine design is shown in Figure 5. The compressor and turbine rotor diameters are 8 mm and 6 mm respectively (since the turbine does not extract power to drive a generator, its size and thus its cooling load could be reduced). 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 and reducing the external package temperature. The rotor is supported on a journal bearing on the periphery of the turbine and by thrust bearings on the rotor centerline. The peripheral speed of the compressor is 500 m/s so that the rotation rate is 1.2 M rpm. External air is used to start the machine. With 400 micron tall airfoils, the unit is sized to pump 0.36 grams/sec of air, producing 11 grams of thrust or 17 watts of shaft power. First tests of this engine are scheduled for 2000.

COMPONENT TECHNOLOGIES

Given the overview of the system design requirements outlined above, the following sections discuss technical consideration of the component technologies. For each component, the overriding design objective is to devise a geometry which yields the performance required by the cycle while being consistent with near-term realizable microfabrication technology.

A problem common to all of the component technologies is that of instrumentation and testing. At device sizes of mi-

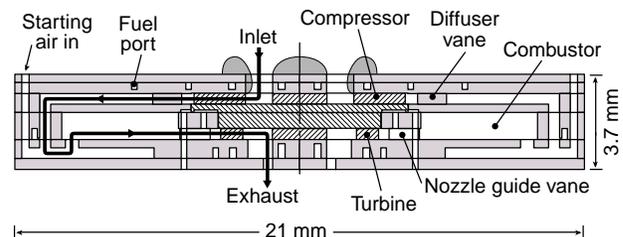


Figure 5: H₂ demo engine with silicon, cooled turbine.

rons to hundreds of microns, instrumentation cannot be purchased and then installed, rather it must be fabricated into the device from the start. While technically possible, this approach can easily double the complexity of the microfabrication, and these devices are already on the edge of the state-of-the-art. To expedite the development process therefore, whenever possible development was done in superscale rigs, rigs large enough for conventional instrumentation.

Bearings

As in all high speed rotating machinery, the rotor must be supported for all radial and axial loads seen in service. In normal operation this load is simply the weight of the rotor times the accelerations imposed (9 g's for aircraft engines). If a small device is dropped on a hard floor from two meters, several thousand g's are impulsively applied. An additional requirement for portable equipment is that the support be independent of device orientation. The bearings and any associated equipment must also be compatible with the micro device's environment, high temperature in the case of the gas turbine engine. Previous MEMS rotating machines have been mainly micromotors turning at significantly lower speeds than of interest here and so could make do with dry friction bearings operating for limited periods. The higher speeds needed and longer lives desired for micro-heat engines require low friction bearings. 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. Magnetic bearings have two disadvantages for 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. Since these temperature are below those encountered in the micro-gas turbine, cooling would be required. For these reasons, effort was first concentrated on designs employing electric fields. These designs examined did not appear promising in that the forces produced were marginal

compared to the bearing loads expected (Miranda, 1997). Also, since electromagnetic bearings are unstable, feedback stabilization is needed, adding to system complexity.

Air bearings support their load on thin layers of pressurized air. If the air pressure is supplied from an external source, the bearing is known as *hydrostatic*. If the air pressure is derived from the motion of the rotor, then the design is *hydrodynamic*. Hybrid implementations combining aspects of both approaches are also possible. Since the micromachines in question include air compressors, both designs are applicable. Either approach can readily support the loads of machines in this size range and can be used on high temperature devices. All else being the same, the relative load-bearing capability of an air bearing improves as size decreases since the surface area-to-volume ratio (and thus the inertial load) scales inversely with size. Rotor and bearing dynamics scaling is more complex, however.

The simplest journal bearing is a cylindrical rotor within a close-fitting circular journal (Figure 6). This geometry was adopted first as the easiest to microfabricate. Other, more complex variations might include wave bearings and foil bearings. The relevant physical parameters determining the bearing behavior are the length-to-diameter ratio (L/D); the gap between the rotor and journal ratioed to the rotor radius (c/R); 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 $(c/R)^5$ which tends to dominate the design considerations. Load bearing also scales with L/D (Piekos *et al.*, 1997).

The design space available for the microrotating machinery is constrained by manufacturing capability. We have chosen to fabricate the rotor and journal structure at the same time to facilitate low cost, volume manufacturing. The most important constraint is the etching of vertical side walls. By pushing the limitations of published etching technology, we have been able to achieve taper ratios of about 30:1-50:1 on narrow etched vertical channels for channel depths of 300-500 microns as shown in Figure 7 (Lin *et al.*, 1999). This capability defines the bearing length while the taper ratio delimits the bearing gap, c . To

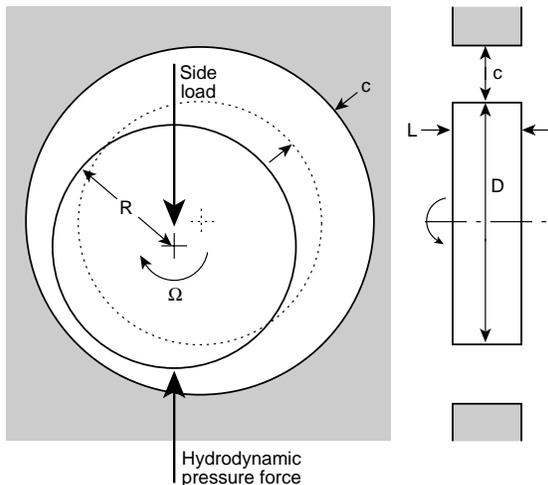


Figure 6: Gas bearing geometry and nomenclature. The gap, c , is greatly exaggerated in this figure.

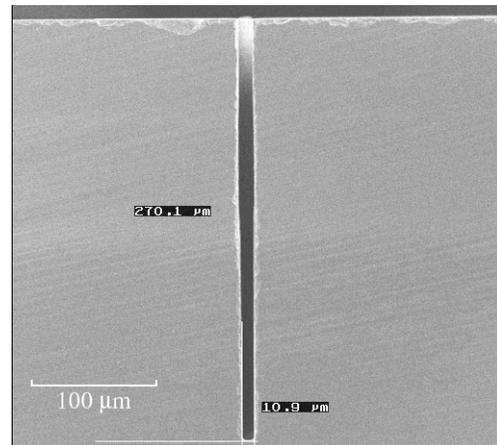


Figure 7: Narrow trenches can be etched to serve as journal bearings.

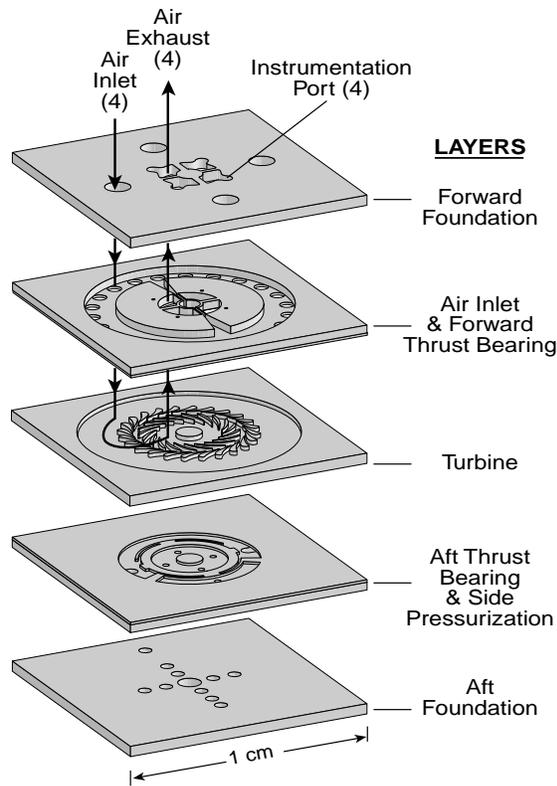


Figure 8a: Exploded view of five layers comprising the turbine bearing rig.

minimize gap/radius, 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 bearing drag) and low L/D (therefore reduced stability). In the radial turbine shown in Figure 3, the journal bearing is 300 microns long with an L/D of 0.075, c/R of 0.01, 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.

Stability is an important consideration for all high speed rotating machines. When centered, hydrodynamic bearings are unstable, especially at low rotational speed. 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 micro scale, (1) the rotor weight is negligible, and (2) insensitivity to orientation is desirable, so we have adopted a scheme which uses differential gas pressure to force the rotor eccentric. Extensive numerical modeling of these microbearing flows have shown that the rotor will be stable at eccentricities above 0.8-0.9 (Piekos and Breuer, 1998). For the geometry of the turbine in Figure 3, the rotor must thus operate between 1-2 microns from the journal wall (Piekos *et al.*, 1997). This implies that deviations from circularity of the journal and rotor must be small compared to 1 micron.

To test these ideas, two geometrically similar turbine-bearing

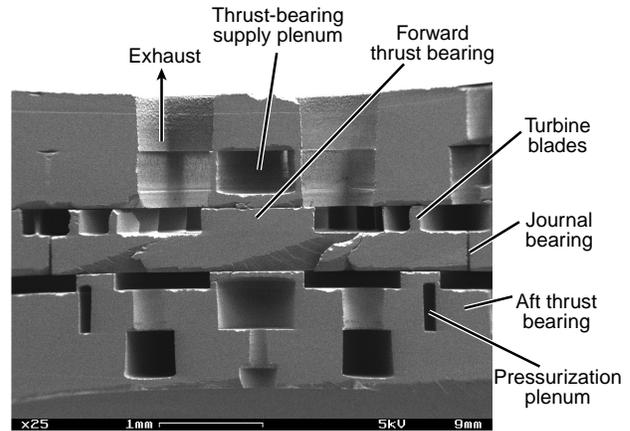


Figure 8b: Five-layer microturbine bearing rig with 4 mm dia rotor.

ing test rigs have been built and tested using the same bearing geometry, one at micro scale with a 4 mm diameter rotor and the other a macro scale unit 26 times larger. The macro version was extensively instrumented for pressure and rotor motion measurements (Orr, 1999). The microturbine bearing test rig, shown in Figure 8, consists of five stacked layers, each fabricated from a single Si wafer (Lin *et al.*, 1999). The center wafer is the radial inflow turbine of Figure 3, with a 4,200 micron diameter, 300 micron thick rotor. The turbine rotor is a parallel-sided disk with blades cantilevered from one side. While such a simple design is viable in silicon above 500 m/s, the centrifugal stresses are too high for metals without tapering of the disk (so the macro version is limited to 400 m/s). The wafers on either side contain the thrust bearings and plumbing for the side pressurization needed to operate the rotor eccentrically. The outside wafers contain the intake, exhaust, and vent holes. In this test device the thrust bearings are hydrostatic, pressurized by external air, and the journal bearing can operate in either hydrodynamic or hydrostatic mode. Figure 9 is data taken from an optical speed sensor during hydrostatic bearing operation.

Turbomachinery Fluid Mechanics

In many ways the fluid mechanics of microturbomachinery are similar to that of large scale machines, for example, high

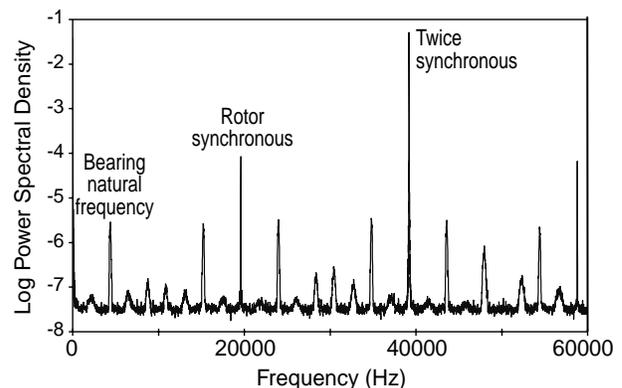


Figure 9: Speed data from microturbine at 1.2 M rpm.

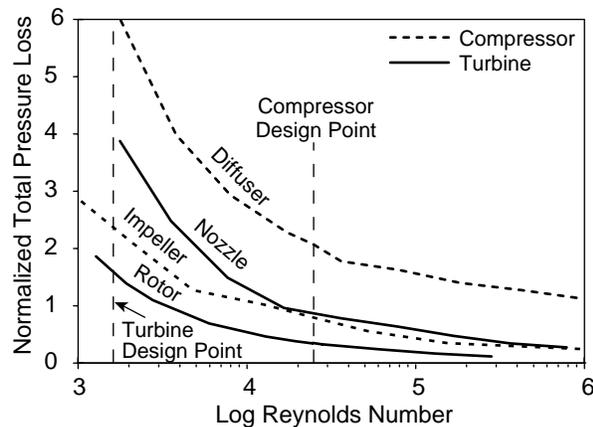


Figure 10: 2-D calculation of compressor and turbine losses vs. Reynolds number (size).

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 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, however, that it is difficult to diffuse the flow, both in the rotor and the stator, without separation. This implies that most of the stage work must come from the centrifugal pressure change. It also precludes the use of vaneless diffusers (if high efficiency is required) since the flow rapidly separates off parallel endwalls. Figure 10 illustrates the variations of efficiency with size for a radial flow compressor and turbine as estimated with a 2-D CFD code (Jacobson, 1998).

The design challenges introduced by the low Reynolds numbers are exacerbated by geometric restrictions imposed by current microfabrication technology. In particular, constant passage height is a problem in these high speed designs. High work on the fluid means large density changes. In conventional centrifugal turbomachinery, fluid density change is accommodated by contracting (in compressors) or expanding (in turbines) the height of the flow path. However, current microfabrication technology is not amenable to tapering passage heights; even a step change is a major effort. One approach is to control the diffusion in blade and vane passages by tailoring the airfoil thickness rather than the passage height. This approach can result in very thick blades as can be seen in the 4:1 pressure ratio compressor shown in Figure 11. Compared to conventional blading the trailing edges are relatively thick. The design choice is either thick trailing edges (which add loss to the rotor) or high rotor exit angles (which result in increased diffuser loss and reduced operating range). Several approaches are being pursued in parallel here. Note that, 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 the impeller exit. This imposes a spanwise variation on stator inlet angle (15-20 degrees for the geometries examined) which cannot be accommodated by twisting the airfoils (which is not permitted by microfabrication) (Mehra *et al.*, 1998).

Another 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 extremely 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 cost 5% in compressor efficiency and 15% in mass flow compared to a smooth turn (Mehra *et al.*, 1998). 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 significant fabrication complexity), and adding externally-produced contoured parts when the turns are at the inlet or outlet to the chip.

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. As an aid to detailed flow measurements, a 75 times linear scale model of a 400 m/s tip speed, nominally 2:1 pressure ratio, 4 mm diameter compressor was built and operated at reduced inlet pressure to match the design Reynolds number of about 20,000 (Shirley, 1998). A comparison of CFD simulation to data is shown in Figure 12. The simulation correctly predicts the pressure ratio but not the mass flow rate.

Based upon our work to date, it should be possible to microfabricate 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 to 10% higher.

Combustion

The primary design requirements for combustors include temperature rise, efficiency, low pressure drop, structural integrity, ignition, and stability. These requirements are no different for a microcombustor but the implementation required to achieve them can be. 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 *combustion resi-*

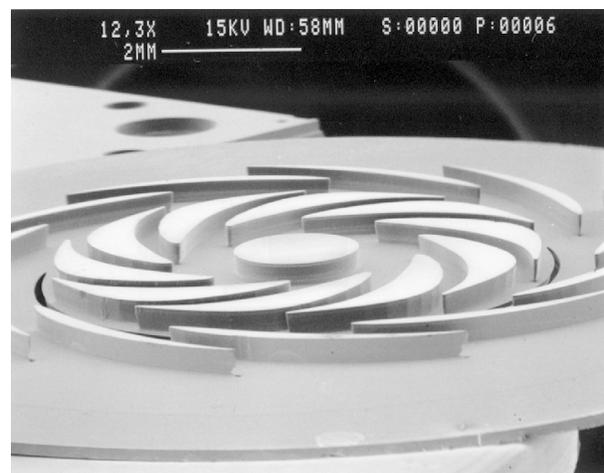


Figure 11: Microcompressor with 8 mm dia rotor.

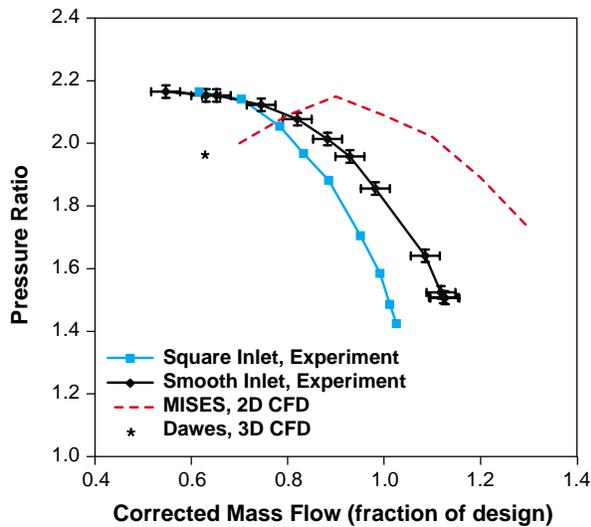


Figure 12: Data and simulations on a 400 m/s tip speed compressor at $Re \approx 20,000$.

dence 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 (typically mixing accounts for more than 90% of combustor residence time). Thus, the combustor volume is a greater fraction of a microengine than a large engine, by a factor of about 40 for the devices designed to date. Another difference between large and micro scale machines is the increased surface area-to-volume ratio at small sizes. This implies increased heat loss from microcombustors but offers more area for catalysts.

The design details are dependent on the fuel chosen. One design approach taken has been 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 by a factor of about 10 from the usual 5-10 msec. The disadvantage to this approach is a susceptibility to flashback from the combustor into the premix zone, which 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 1930s). Specifically, hydrogen will burn at equivalence ratios as low as 0.3 which yields adiabatic combustion temperatures below 1500 K. Hydrocarbon fuels must be operated closer to stoichiometric and therefore at higher temperatures, above 2000 K. The reduced heat load from the low temperature combustion, combined with the high thermal conductivity of silicon, means that silicon (which melts at 1600 K) is a viable structural material for a H_2 combustor (Waitz *et al.*, 1998; Mehra *et al.*, 1999a).

Silicon combustors have been built which duplicate the geometry of the engines in Figures 4 and 5, with the rotating parts replaced with stationary swirl vanes (Mehra and Waitz, 1998). The combustor volumes are 66 and 190 mm^3 , respectively (Figures 13 and 14). The designs take advantage of microfabrication's

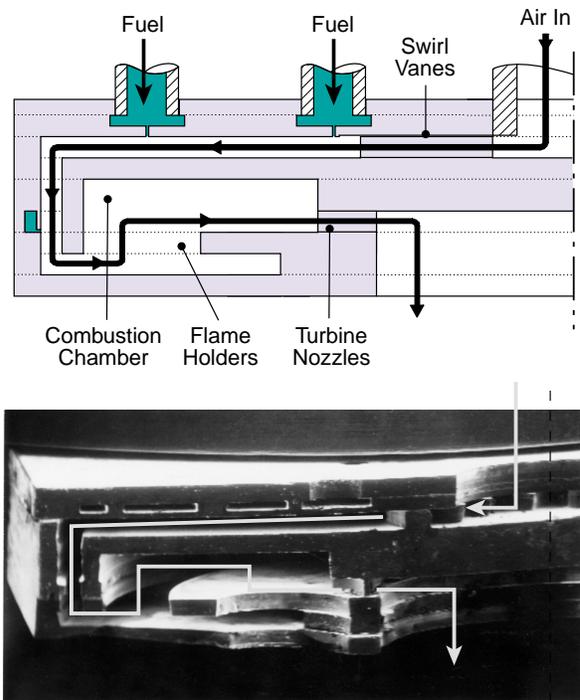


Figure 13: Microcombustor which comprises the static structure of the demo engine of Figure 5.

ability to produce similar geometric features simultaneously. For example, the larger combustor has 90 fuel injection ports, each 120 microns in diameter, to promote uniform fuel-air mixing. The smaller combustor operating at a power level of 200 watts is shown in Figure 14 along with a CFD simulation of the flowfield. These tests demonstrated that conductively-cooled silicon turbine vanes can survive at 1800 K gas temperatures for 5 hrs with little degradation (Figure 15). Measurements have shown that these microcombustors can achieve efficiencies in the mid 90% range (Mehra *et al.*, 1999b). Data at various equivalence ratios (ϕ) are shown in Figure 16. (The gaps in the lines are due to thermocouples failing at high temperature.) Ignition has not proven a problem; a simple hot wire ignitor has proven sufficient, even at room pressure (Mehra, 2000).

Hydrocarbons are more difficult to burn. Initial tests show that the existing combustors can burn ethylene and propane, but at reduced efficiencies since the residence times are too short

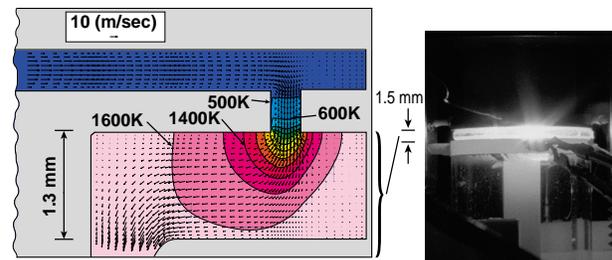


Figure 14: 200 watt microcombustor.

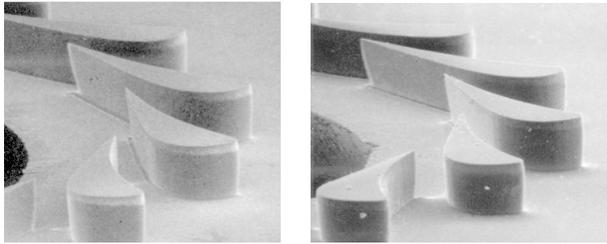


Figure 15: Silicon turbine vanes as built and after 5 hours in 1800 K combustor outflow.

for complete combustion. Other approaches being pursued here include a stoichiometric zone-dilution scheme similar to conventional gas turbine combustors and catalytic combustion.

Electrical Machinery

Microelectrical machinery is required for power generation and as a prime mover for a starter or various pumps and compressors. There is an extensive literature on microelectric motors, which will not be 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 above those reported in the literature to date. Also, the thermal environment is generally harsher. Integrating the electric machine within the device (gas turbine, compressor, etc.) offers the advantage of mechanical simplicity in that no additional bearings or structures are required over that needed for the fluid machine. Also, there is a supply of cooling fluid available.

As in the case of electric bearings, both electrostatic 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, electrostatic (commonly referred to as *electric*) designs were first examined. Power den-

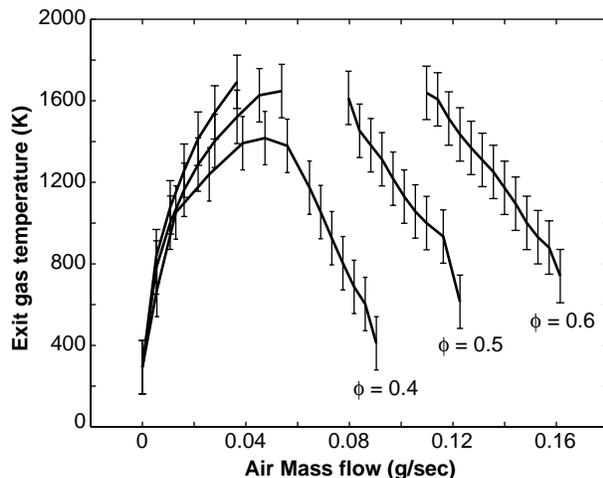


Figure 16: Microcombustor performance with hydrogen fuel as a function of equivalence ratio (ϕ).

sity scales with electric field strength (torque), frequency, and rotational 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 (Bart and Lang, 1989).

The operation of an electric induction machine can be understood with reference to Figure 17 (Nagle and Lang, 1999). The machine consists of two components, a rotor and a stator. The rotor is comprised of a 5-20 μm 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 a good insulator. A moving electric field is imposed on the stator electrodes with the aid of external electronics. The stator field then imposes a charge distribution 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, the machine can operate as a motor, generator, or brake. Torque increases with 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, it is a maximum at a few microns so that micromachines can potentially realize higher power density than large machines

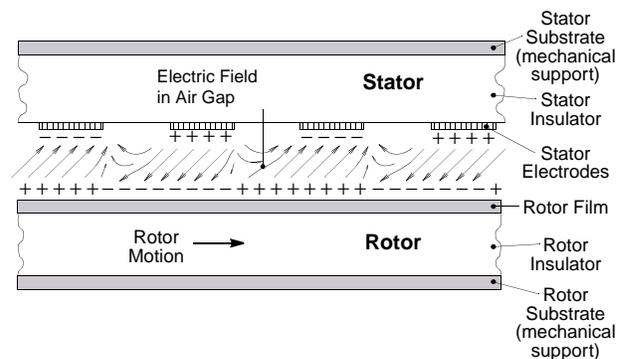


Figure 17a: Electric charge and fields within an electric induction motor.

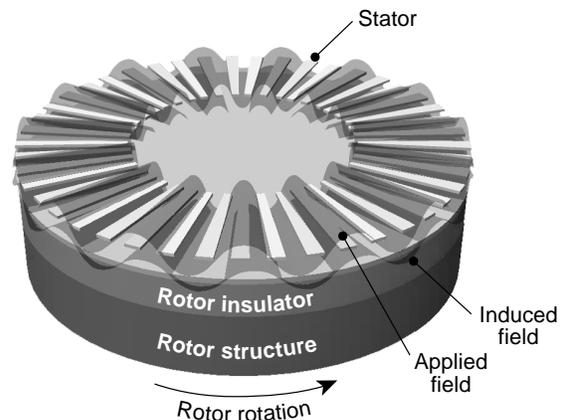


Figure 17b: Electric-induction conceptual layout of motor at expanded vertical scale.

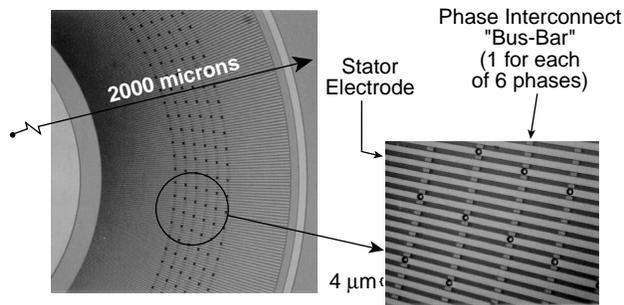


Figure 18: High power density electric stator structure.

of the same design. Frequency is constrained by external electronics design and by fabrication constraints on the stator electrode geometry. We are exploring the design space centered about 100-300 volts and 1-3 MHz. This is consistent with a machine power of about 10 watts with a 4 mm diameter and a 3 μm air gap. A six-phase, 131-pole (786 electrodes) stator for such a machine is shown in Figure 18 (Ghodssi *et al.*, 1999).

OTHER APPLICATIONS

While the focus of this paper has been microfabricated micro-gas turbine engines, other MEMS devices which use the same fundamental high speed rotating machinery technology and design approach are also under development. One such machine is a micro-air turbine generator. This is basically the turbine-bearing rig of Figures 3 and 8 with a generator on the side of the disk opposite the turbine blades. It was designed as a test bed for component development but could serve as an electric generator running from a supply of compressed gas (with inlet temperatures up to about 900 K). Another variant is a motor-driven air compressor/blower (Figure 19). In this case, the turbine blading is replaced by compressor blading and the electric machine on the opposite side of the rotor runs as a motor. It is designed to pressurize small fuel cells and aspirate scientific instruments. It could also serve as the compressor in a refrigeration cycle for the cooling of electronics, sensors, or people.

A microbipropellant liquid rocket motor is also under development. This is a regeneratively-cooled silicon structure de-

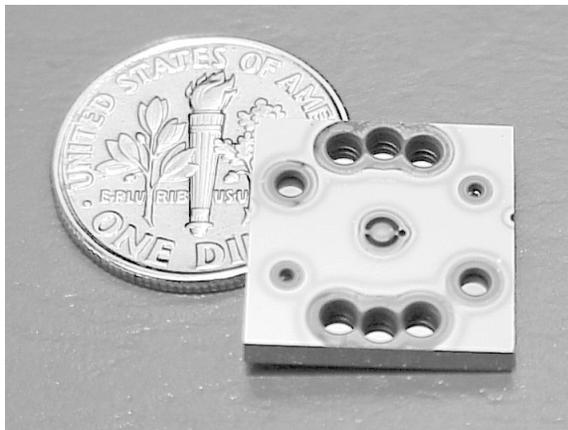


Figure 19: Micromotor-compressor test rig.

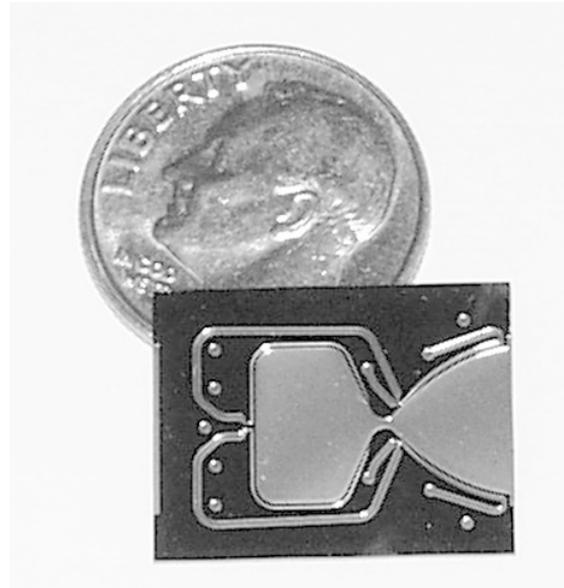


Figure 20: Microfabricated, 15 Nt thrust, bipropellant, cooled rocket engine.

signed to produce 15 N of thrust at 125 atm chamber pressure (Figure 20). Preliminary tests of the cooled thrust chamber are promising. A complete engine system would include integrated turbopumps and controls.

CLOSING REMARKS

It now appears that microfabricated high speed rotating machinery is feasible. Such micromachines are the enabling technology for micro-heat engines such as gas turbines, Rankine cycles, and rocket engines. These devices will find a host of applications in energy conversion and power production, coolers and heat pumps, and propulsion for both micro and macro vehicles. Furthermore, the components – pumps, compressors, turbines, motors, generators – are themselves useful for a host of fluid handling, transducer, and energy system applications.

One lesson learned to date is that conventional engineering wisdom does not necessarily apply to micromachines. Different physical regimes demand different design solutions. Different manufacturing constraints lead to different configurations. Not necessarily inferior, just different. The first devices made are crude and have low performance compared to their more familiar large-sized brethren, just as gas turbines did when they were first introduced. It seems likely, however, that the economic potential of high power density micromachinery will spur the investment in research and development needed to greatly evolve the performance of the devices.

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