EXPERIMENTAL RESULTS OF AN EJECTOR DRIVEN MICRO TURBINE GENERATOR

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Abstract: This paper reports on the design, thermodynamic cycle, and experimental results of an ejector-driven micro turbine generator. For this cycle, an ejector, as opposed to a compressor, creates a pressure gradient to drive a turbine. When connected to a micro-generator and an ethanol vapor boiler, a power output of 7.5 mW is delivered to a resistive load of 300 Ω at an angular velocity of 27,360 RPM.

Keywords: micro engine, micro turbine, ejector

INTRODUCTION

The realization of a self-sustaining micro-turbine engine has remained elusive due to the many difficulties associated with the Brayton cycle at the MEMS scale. For example, minimum component efficiencies for the compressor and the turbine must be realized for a Brayton cycle to close and produce net work [2]. However, at small scales the efficiencies of these components are diminished due to viscous effects at lower Reynolds numbers [3]. The realization of scale independent parameters such as tip speed is also problematic at the MEMS scale. For example, to reach sufficient tip speeds and to operate at high efficiency requires extremely high rotor speeds; this can only be achieved with gas bearings. Hydrostatic bearing operation requires an external pressure source, while hydrodynamic operation experiences difficulties during the startup period [4]. Lastly, manufacturing these turbomachinery components at the required tolerances poses technical challenges.

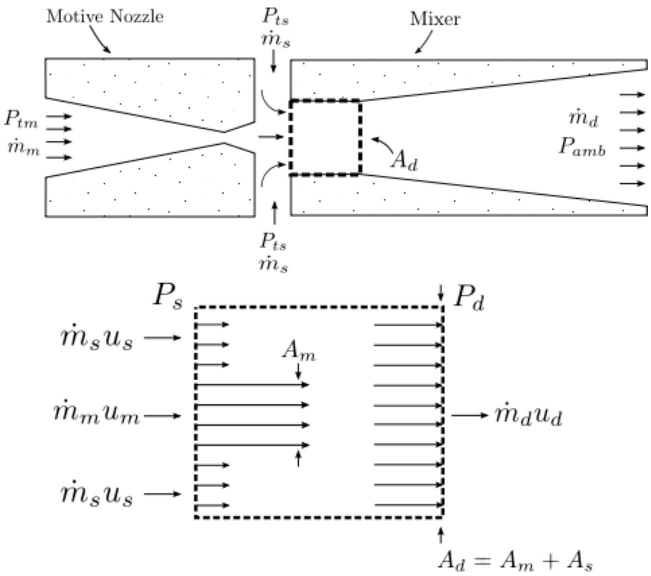

Figure 1: Control volume of the ejector mixing region.

This paper presents an alternative thermodynamic cycle that offers solutions to these challenges. This cycle is designed around static pumping devices, an injector and an ejector [1]. Both are based on the same fundamental principle: a high momentum motive fluid is mixed with a low momentum suction fluid, resulting in a discharge fluid with less overall momentum and thus a higher pressure (Fig. 1). The ejector is used primarily to create a pressure gradient across the turbine, while the injector pumps liquid into a high pressure boiler. The benefit of these components lies in their static and turbo-machine independent operation. Unlike a turbo-compressor, the ejector is uncoupled from the turbine and thus provides a pressure gradient across the turbine regardless of the turbine's rotor speed so long as heat is applied to the boiler.

This eliminates many problems associated with the startup phase and guarantees that the cycle will close at any operating efficiency. In addition, because there are no moving parts associated with these components, they can be more easily fabricated as their manufacturing tolerances are not as critical as those associated with rotating micro-turbomachinery components. Lastly, the suction provided by the ejector can be utilized to prime the hydrostatic gas bearings, removing the need for an external pressure source.

This paper will discuss both the thermodynamic model concerning this type of cycle and the preliminary experimental results of an ejector-driven micro-turbo-generator.

THERMODYNAMIC CYCLE

The proposed cycle is derived from a steam locomotive cycle and an afterburning Brayton cycle (Fig. 2). Combustion takes place downstream of the turbine, and the generated heat is split between preheating the turbine inlet air with the use of a recuperator and vaporizing ethanol in the boiler. Note that for this cycle ethanol is used both as the fuel and as the motive vapor for both the injector and ejector. The power from the turbine can be estimated from the incompressible flow assumption, the ideal gas law, and by recalling that the total pressure at the turbine inlet is roughly equal to ambient pressure.

$$\frac{\dot{W}_t}{\dot{m}_s} = \eta_t \frac{\Delta P_t}{\rho} \approx \eta_t \frac{P_{amb}-P_{t,s}}{P_{amb}} R_{air} T_{t4} \qquad (1)$$

The pressure difference between the ambient and total suction pressures can be determined by the conservation of mass and momentum for the control volume shown in Fig. 1, and assuming isentropic expansion in the diffuser such that the exit dynamic head is approximately zero. The result is

$$\frac{P_{amb}-P_{t,s}}{\frac{1}{2}\rho\mu_m^2} = \sigma\left[2-\sigma-\frac{\alpha^2\sigma^3}{(1-\sigma)^2}+\frac{2\alpha\sigma^2}{(1-\sigma)}-\frac{2\alpha\sigma}{(1-\sigma)}\right] \qquad (2)$$

where σ is the area ratio and α is the ejector entrainment ratio

$$\sigma \equiv \frac{A_m}{A_d} \qquad (3)$$

$$\alpha \equiv \frac{\dot{m}_s}{\dot{m}_m} \qquad (4)$$

The system rejects heat in the ejector discharge flow and by the cooling and condensation of the ethanol vapor.

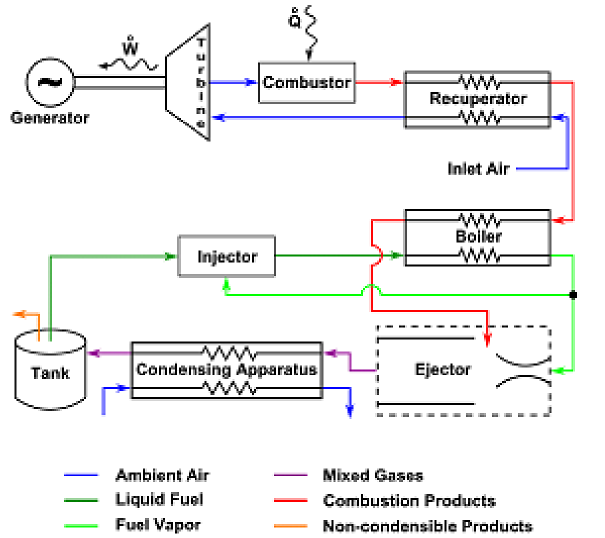

Figure 2: Schematic of an afterburning thermodynamic cycle driven by an injector.

However, the heat rejected in condensation typically far exceeds the heat rejected from the cooling of the exhaust gases. The overall thermal cycle efficiency can be approximated as

$$\eta \approx \frac{\dot{W}_t}{\dot{W}_t+\dot{m}_m h_{fg}} \qquad (5)$$

In addition, by further assuming the turbine power is significantly less than the rejected heat we can re-approximate the cycle efficiency as

$$\eta \approx \frac{\dot{W}_t}{\dot{m}_m h_{fg}} \qquad (6)$$

Combining equations (1) – (2) and (5) – (6) we obtain

$$\eta = \frac{\eta_t R_{air} T_{t4} \alpha\sigma\left[2-\sigma-\frac{\alpha^2\sigma^3}{(1-\sigma)^2}+\frac{2\alpha\sigma^2}{(1-\sigma)}-\frac{2\alpha\sigma}{(1-\sigma)}\right]\frac{1}{2}\rho\mu_m^2}{h_{fg}P_{amb}} \qquad (7)$$

where σ, μ_m, and T_{t4} are design parameters (μ_m is a function of boiler pressure). From the control volume analysis of the mixing region, it can be shown that the highest static pressure recovery occurs when $\sigma = 0.5$. For this geometric ratio, the highest cycle efficiencies will occur for an entrainment ratio of 0.53. Using these values and assuming a turbine efficiency of 50%, we can approximate the overall efficiencies for different parameters as shown in Table 1.

Table 1. Efficiency Approximations.

	Case 1	Case 2
$\frac{P_{boiler}}{P_s}$	3	30
T_{t4}	500 K	1600 K
η	1.13%	5.36%

EXPERIMENT AND RESULTS

An experiment was conducted to demonstrate that an ejector powered by ethanol vapor could create a pressure gradient across a turbine and drive its attached micro-generator to deliver electrical power.

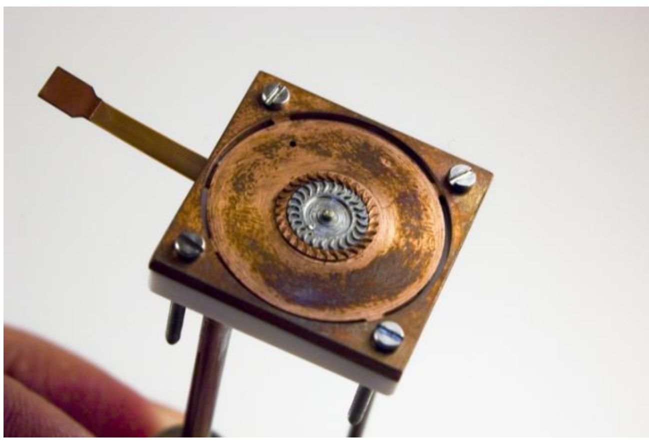

Figure 3. The original micro-turbine design bonded to the rotor of a permanent magnet generator with protruding leads.

The turbine was originally designed and micro-machined as a radical flow impulse turbine with an NGV outer-diameter of10 mm and blade heights of 250 μm (Fig. 3). However, due to the configuration and choice of materials, eddy current losses prevented the turbine from operating on design.

A new turbine with a rotor outer-diameter of 11 mm and blade heights of 750 μm was 3D printed out of ABS plastic for the final experiments. The ejector was micro-machined with a throat diameter of 719 μm, an area ratio of 1:8, and was driven by ethanol vapor from a conventionally-sized boiler. The turbine rotor was bonded to a 3-phase Faulhaber 1202-H-006-BH permanent magnet DC motor with the outputs being rectified to DC with the use of 1N5817 Schottky diodes (Fig. 4).

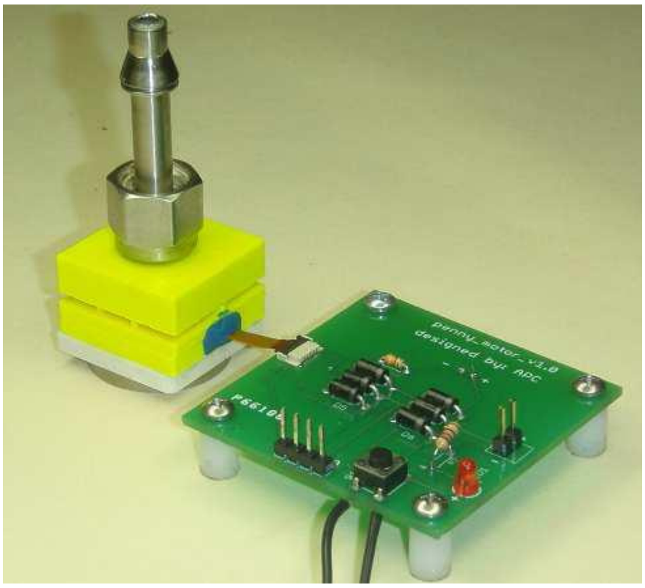

Figure 4. 3D printed turbo-generator connected to power electronics (boiler and ejector not shown).

Table 2. Experimental Results

Property	Units	Value
Rotor Speed	RPM	27,360
Turbine Pressure Ratio	-	1.05
Boiler Pressure	atm	15
V_{DC}	V	1.49
Turbine inlet temp.	K	293
Power	mW	7.5

The rectified DC source was then connected across a variable resistor that was adjusted until the maximum power output was obtained. The conditions at the optimal operating point are shown in Table 2.

Power from the engine was also used to light a row of LEDs (Fig. 5). Note that the maximum power does not take place by setting the load resistance to the equivalent stator resistance as prescribed in the maximum power transform theorem.

The reason for this is that the maximum power transfer theorem assumes a fixed voltage source, whereas for a turbo-generator the voltage is coupled to the rotor speed. Therefore, the load resistance must be set such that the turbine can reach its design speed (Fig. 6).

Figure 5. LEDs powered by the micro-turbine generator.

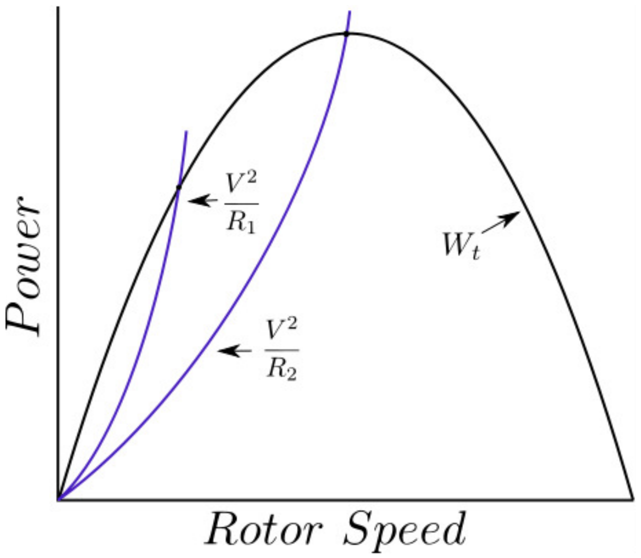

Figure 6. Turbine shaft power and load power as a function of rotor speed.

CONCLUSION

This paper has demonstrated that, with no moving parts, an ejector can produce a pressure gradient to drive a micro-turbine and generate power. An advantage of this operating mode is that, unlike a standard Brayton cycle, there is no minimum required efficiency for the cycle to close and the engine to function. In addition, the ejector provides a means of creating the pressure gradient required by hydrostatic gas journal bearings. This will allow the bearings to operate hydrostatically at low speeds until the RPM increases, thereby allowing hydrodynamic bearing operation. The manufacturing tolerances of these static pumping devices can also be much less stringent than those of micro-turbomachinery.

The thermodynamic cycle was analyzed with both compressible and incompressible flow assumptions, and a basic method of estimating the cycle thermal efficiency was presented. Experiments were conducted to demonstrate the viability of a power cycle designed around an ejector-driven micro-turbine.

For future work, a fully functional and integrated engine will be designed, manufactured, and tested. A turbo-compressor will be added to the cycle upstream of the recuperator with the intention of improving efficiency. The ejector would provide a means of creating the necessary conditions in the transient phase before the turbo-compressor reaches sufficient tip speeds for self-sustaining operation.

REFERENCES

[1] W.G. Gardner, I. Wang, J.W. Jaworski, N.A. Brikner, and J.M. Protz. Experimental investigation and modeling of scale effects in jet ejectors. Journal of Micromechanics and Microengineering, 20(8), 2010.

[2] J. L. Kerrebrock. Aircraft Engines and Gas Turbines. The MIT Press, Cambridge, Massachusetts, 2nd edition, 1992.

[3] C. Lee, S. Arslan, and L.G. Fréchette. Design principles and measured performance of multistage radial flow microturbinemachinery at low Reynolds numbers. Journal of Fluids Engineering, 130(11), 2008.

[4] L. X. Liu, C. J. Teo, A. H. Epstein, and Z. S. Spakovszky. Hydrostatic gas journal bearings for micro-turbomachinery. Journal of Vibration and Acoustics, 127(2):157-164, 2005.