MEASUREMENT OF COMPRESSOR AND TURBINE MAPS FOR AN ULTRA-MINIATURE GAS TURBINE

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Abstract: This paper presents a set-up for measuring compressor and turbine maps for an ultra-miniature gas turbine generator of 1 kW (impeller diameter of 20 mm). This set-up is a so-called turbo-shaft set-up containing compressor, turbine and air bearings, and allows testing and characterization of these components without the overhead and complexity of a complete gas turbine set-up. The device has been tested for speeds up to 75 000 rpm, producing a relative pressure up to 35 mbar. After improved balancing, tests will be conducted for speeds up to 500 000 rpm, with expected relative pressures up to 2 bar.

Key words: microturbine, gas turbine, turbo-shaft, characterization

1. INTRODUCTION

This paper presents a set-up to measure compressor and turbine maps for an ultra-miniature gas turbine. The application of this gas turbine is as a portable electricity generator of 1 kW. The system consists of a generator and regular gas turbine components: turbine impeller, compressor, combustion chamber, and recuperator. To cope with the high rotor speed of 500 000 rpm, air bearings are used.

Figure 1 shows the general layout of the gas turbine generator. To obtain a compact overall system (< 1 dm^3), an annular design is chosen for combustor and recuperator. Table 1 summarizes the most important parameters.

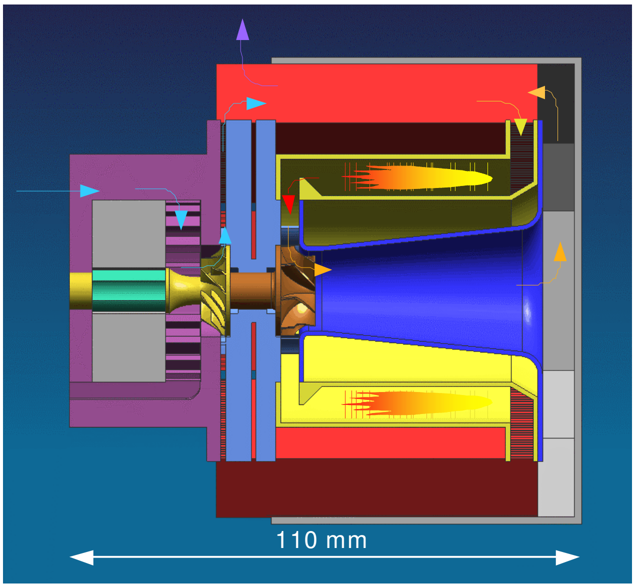

Fig. 1. Ultra-miniature gas turbine design.

Design and optimization of this turbine were presented at last year's PowerMEMS workshop [1]. Several papers discussed design and optimization of individual components [2-6]. The most critical and essential components are compressor, turbine and bearings: compressor and turbine because of their complex geometry and their impact on the cycle´s efficiency, the bearings because of the high required rotor speed.

Table 1: Theoretical gas turbine parameters

Nominal mass flow	20 g/s
Pressure ratio	3.0
Rotor speed	500 000 rpm
Impeller diameters	20 mm
Power	
Compressor	3800 W
Turbine	5083 W
Net mechanical output	1180 W
t-s polytropic efficiency	
Compressor	66 %
Turbine	78 %
Turbine inlet temperature	1200 K
Cycle efficiency	
Without recuperation	11 %
With recuperation	20 %

2. TEST SET-UP

2.1. Turbo-shaft

To test the performance of these critical components without the overhead of the complete system, a special set-up is built, containing only compressor, turbine, diffuser and bearings (see figures 2 and 3). This set-up is called a turbo-shaft set-up, due to its similarity with a turbo of a car engine. Compressed air is used to drive the turbine which in its turn drives the compressor. While the final gas turbine has an annular design, the turbo-shaft set-up uses supply and exhaust tubes connected to volutes collecting and distributing the compressed air. This approach is necessary to connect both compressor and turbine to valves and sensors.

The exhaust of the compressor is connected to an adjustable valve which serves as a variable load. Turbine pressure is controlled manually with a pressure regulator from Festo. By varying the load (through the valve) and speed (via the turbine pressure), and measuring flow and pressure in the exhaust tube, compressor maps can be obtained.

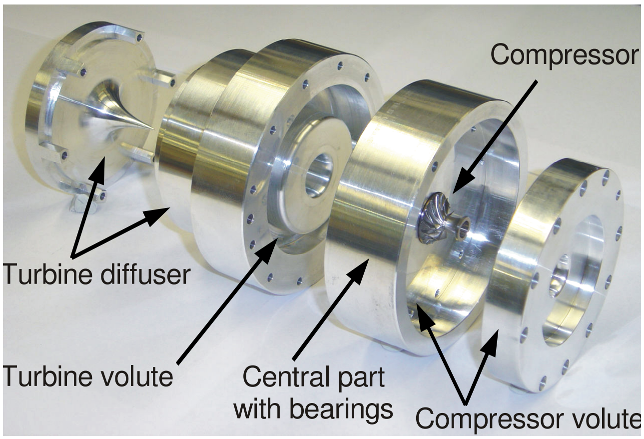

Fig. 2. Turbo-shaft setup.

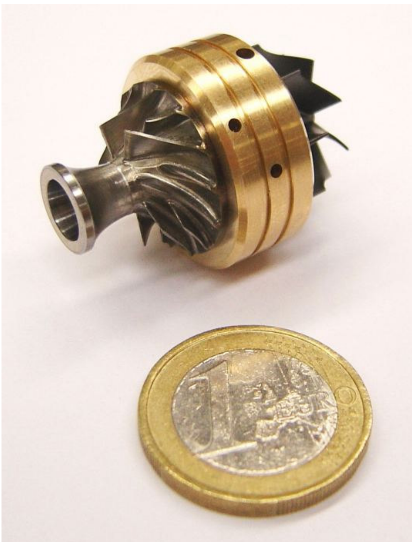

Fig. 3. Compressor (left), turbine (right) and air bearing insert. Rotor diameter: 20 mm. Size compared to 1 euro coin.

2.2. Instrumentation

Figure 4 shows the complete set-up including pressure regulator, variable load, tubing, and various sensors to measure flow, pressure, rotor speed, angular rotor reference and rotor vibration. Speed and phase reference are measured on the compressor nose using a KD310 Fotonic Sensor from Mechanical Technology Inc., by means of a black reference mark. Turbine pressure is measured in the supply tube using a PMP 1400 pressure transducer from Druck with a range of 16 bar (relative pressure). Compressor pressure is measured in the exhaust tube using a pressure sensor from Keller Druckmesstechnik, type PR-21S/2.5bar/80549.3, with a pressure range of 0-2.5 bar (relative pressure). Both compressor and turbine flow are measured with rotameters.

Rotor vibration is measured with two home-made single-fiber optical probes, based on reflected light intensity. The probes are positioned on the 20 mm rims of compressor and turbine. A data acquisition system from National Instruments (PXI-6123 with 500 kHz simultaneous sampling rate) is used to monitor rotor vibration.

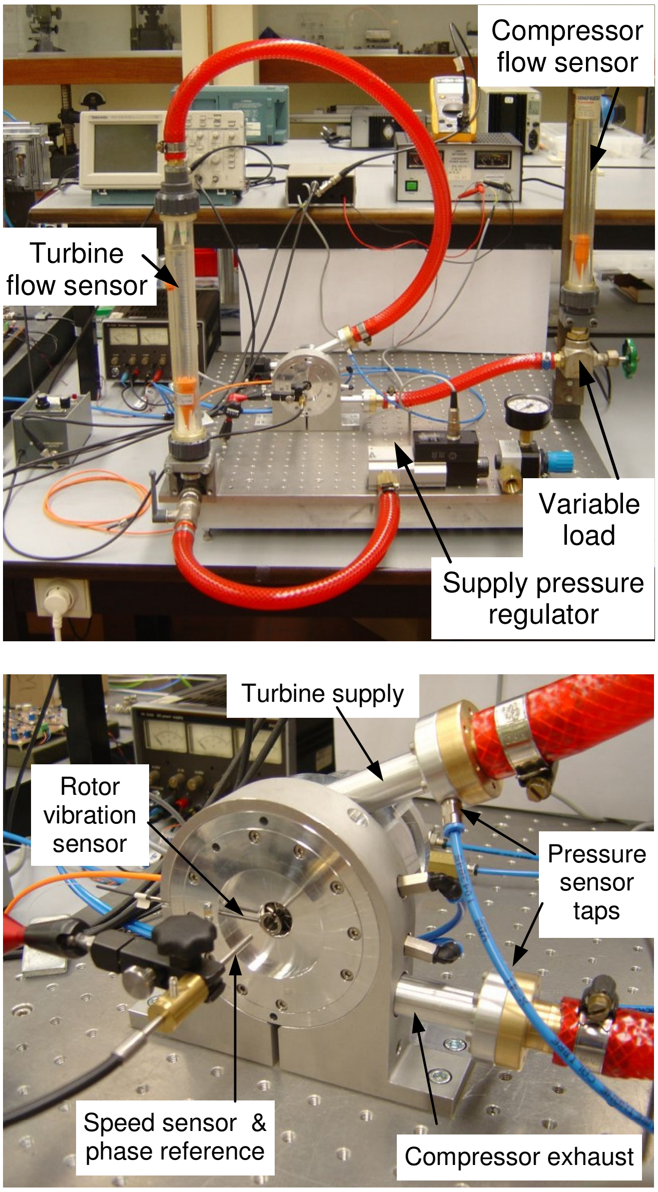

Fig. 4. Turbo-shaft set-up.

2.3. Balancing

The rotor has to be balanced in two planes to minimize both cylindrical and conical rotor vibrations. This balancing is very critical because the nominal rotor speed lies above the resonance frequencies of the suspension modes. Above these critical speeds, the shaft rotates around its center of gravity. Because radial bearing clearance is only 5 µm, the imbalance should be only a fraction of this value, thus maximally 1-2 µm. Also the conical imbalance has to be compensated very well as both thrust bearings have an axial clearance of only 8 µm.

Balancing is currently done in situ by adding small masses inside the cavity of the compressor nose (only one plane). Sufficiently fine balancing in two planes was not yet reached by the time this paper was written.

3. THEORETICAL CHARACTERISTICS

Figures 5 and 6 show theoretical compressor and turbine maps, expressing efficiency and pressure ratio as a function of speed and mass flow.

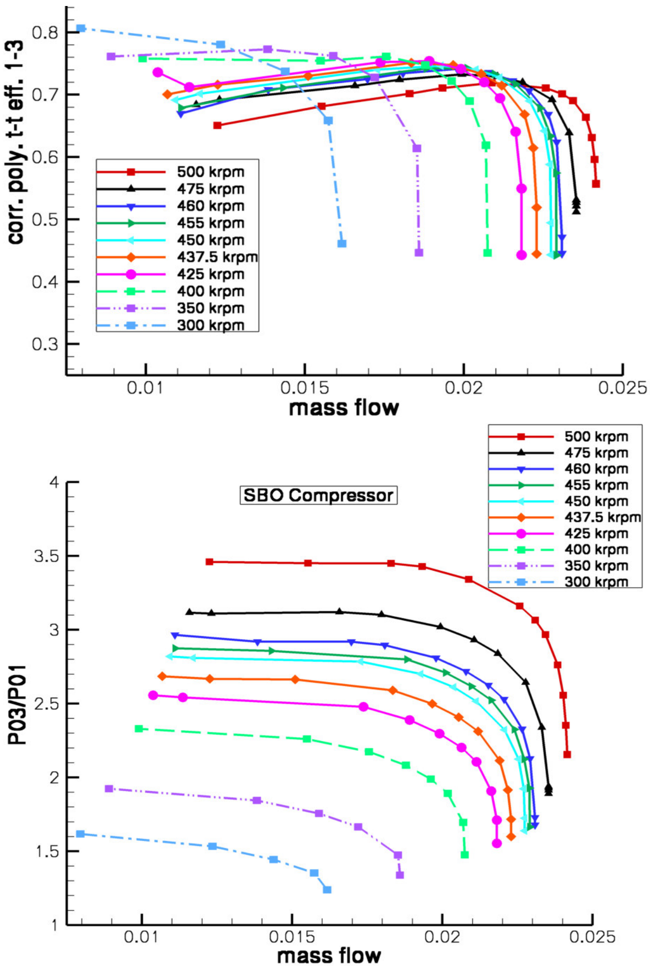

Fig. 5. Theoretical efficiency and pressure ratio of the compressor as a function of speed and mass flow.

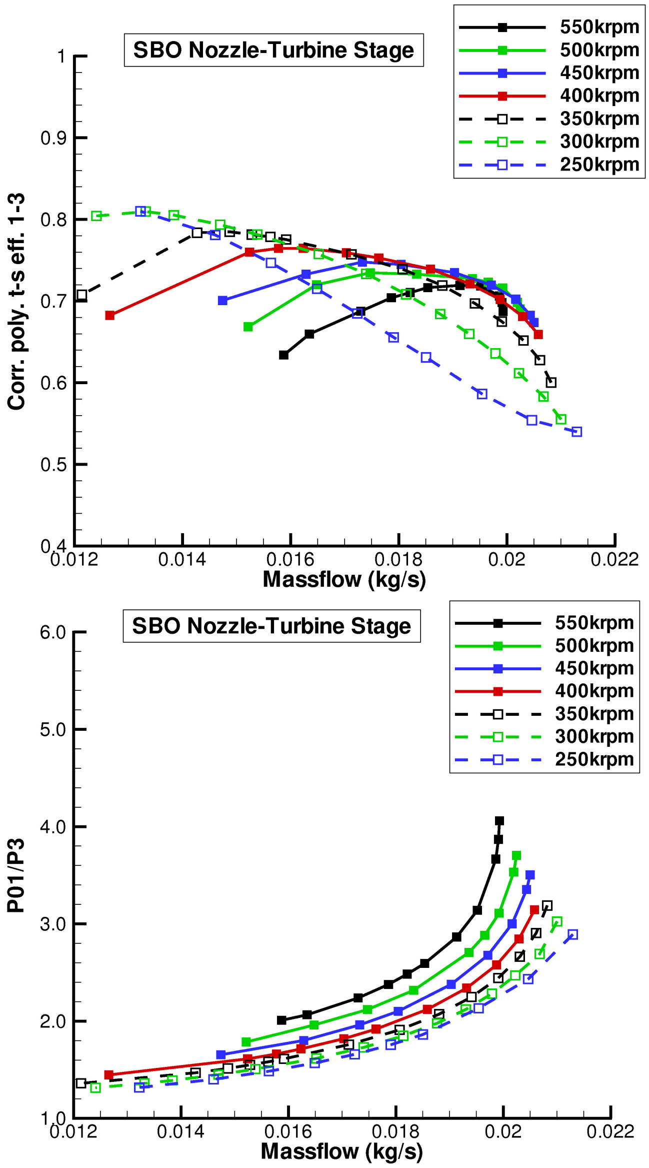

Fig. 6. Theoretical efficiency and pressure ratio of the turbine as a function of speed and mass flow.

4. MEASURED CHARACTERISTICS

Figures 7 and 8 show measured pressure and flow characteristics for turbine and compressor for two extreme situations: with the load valve completely open or completely closed. Speed was limited to 1250 Hz (75 000 rpm) because of residual imbalance. Radial impeller vibrations were kept within an amplitude of 2.3 µm, to stay safely within the limits of bearing clearance. As balancing is currently being improved, high-speed results are expected soon.

The compressor generates a pressure up to 35 mbar at zero flow rate. The pressure built up for maximal flow is evidently lower, but follows the zero flow curve for speeds below 900 Hz, which is explained by the fact that mass flow or pressure below

900 Hz is not sufficient to lift the float inside the rotameter, in this way blocking the flow.

Turbine pressure and flow are lower for zero compressor flow, in agreement with the fact that the compressor generates no work, except for the ventilation losses.

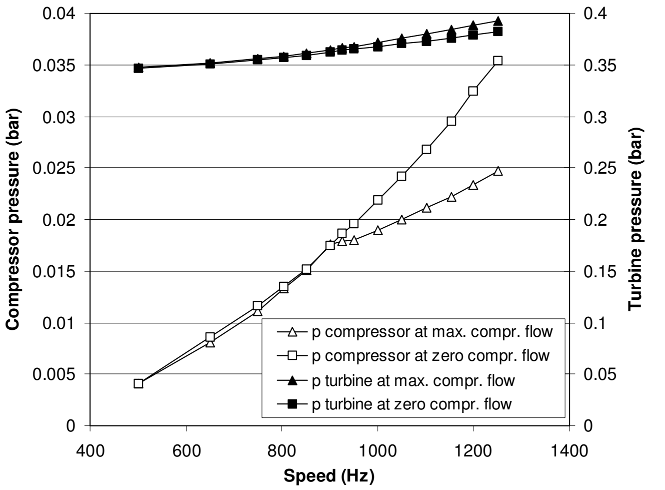

Fig. 7. Compressor and turbine pressure versus speed (relative pressures).

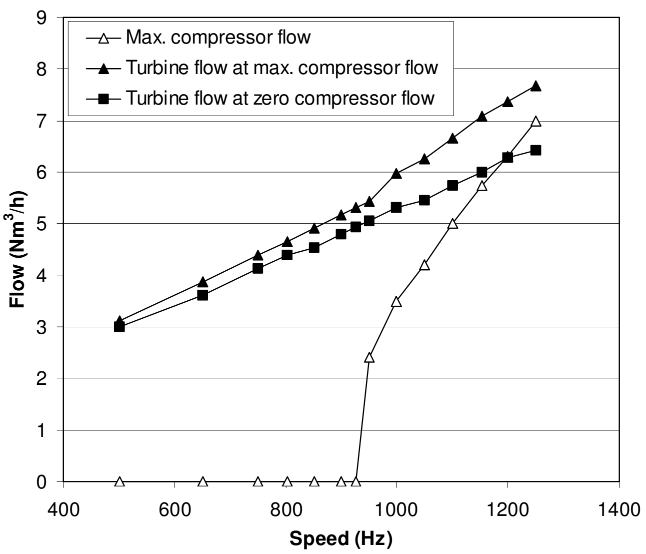

Fig. 8. Flow versus speed for turbine and compressor.

5. DISCUSSION & FUTURE WORK

Due to the preliminary nature of the test results and the limited rotor speed, it is difficult to compare these results with the theoretical data. Nevertheless, the tests prove that the set-up operates reliably and that the trends are correct. With some reservations, extrapolation of the compressor pressure to the nominal speed (pressure assumed to vary with the square of rotor speed) predicts a compressor pressure of 1.6 bar (relative pressure) at zero flow, compared to the design value of 2 (pressure ratio 3).

First job is to improve balancing of the rotor mass to within 1-2 μm of the bearing centers, such that the above graphs can be extended to the full operational speed of the gas turbine (500 000 rpm) and beyond. Also turbine torque and power will be measured, by using acceleration tests [7], while the compressor's exit is closed or its shroud removed (no power taken up by the compressor, except for ventilation and bearing losses). Deceleration tests determine bearing and ventilation losses. By observing flow and pressure drop across the turbine, power consumption is derived. Combining this with the net power determined by the acceleration tests, gives turbine efficiency. Net power combined with the previously determined compressor data, also results in compressor efficiency.

6. CONCLUSION

A turbo-shaft set-up was built to measure the characteristics of turbine and compressor. The set-up works well, but imbalance problems have so far limited the test speed. After fine balancing, high-speed characterization will be carried out.

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