EXPERIMENTAL VERIFICATION OF COMPRESSOR PERFORMANCE FOR AN ULTRA-MICROGASTURBINE

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Abstract: This paper presents the experimental verification of a 20 mm compressor prototype developed for a 1 kWe ultra-microgasturbine. The gasturbine has a nominal speed of 500 krpm, a nominal air flow of 20 g/s and a nominal pressure ratio of 3. The compressor is tested in a turbo-shaft setup containing compressor, turbine and aerostatic bearings. The test speed is currently limited to 261 krpm due to bearing instability. At this reduced speed, the generated relative static pressure reaches 0.41 bar. The close agreement between numerical and measured compressor maps suggests that the target pressure ratio will be attained at nominal speed. To achieve the target speed of 500 krpm, externally damped bearings are under development.

Keywords: microgasturbine, performance, compressor map, air bearing, turbo

INTRODUCTION

This paper presents the experimental verification of a 20 mm compressor prototype developed for a 1 kWe (electric) ultra-microgasturbine. The gasturbine has a nominal speed of 500 krpm, a nominal air flow of 20 g/s and a nominal pressure ratio of 3. Design and optimization of this turbine were presented at the PowerMEMS 2007 workshop [1]. First low-speed results were presented at PowerMEMS 2008 workshop [2]. The current paper presents high-speed results in close agreement with the numerical simulations.

TEST SETUP

Turbo-shaft setup

The compressor is tested in a so-called turbo-shaft setup containing compressor, turbine and aerostatic bearings (see figure 1). Compressor and turbine have the same geometry as in the complete gasturbine. While the final gas turbine has an annular design, the turbo-shaft setup 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. Compressed air is used to drive the turbine which in its turn drives the compressor. The exhaust of the compressor is connected to an adjustable valve which serves as a variable load. Turbine pressure is controlled with a manual pressure regulator. By varying the load (through the valve) and speed (via the turbine pressure), and by measuring flow and pressure in the exhaust tube, compressor maps are obtained.

Fig 1: Exploded view of turbo-shaft setup.

Rotor and bearings

Figure 2 offers a more detailed view on the rotor and bearing construction. The rotor consists of a titanium compressor and ceramic turbine (Si₃N₄-TiN composite) [3], both 20 mm in diameter, connected by a steel bayonet coupling. The rotor is supported by a bronze aerostatic bearing unit [4], featuring one radial and two thrust bearing surfaces. The radial bearing has a diameter of 8 mm, a clearance of 5 µm and 2 rows of 8 feedholes, 150 µm in diameter. Both thrust bearings have a clearance of 8 µm and a single row of 8 feedholes, also 150 µm in diameter. The radial and thrust bearings are fed in a separate way such that different pressures can be applied. In the tests below, the radial bearing is fed at 10 bar, the thrust bearings at 7 bar (relative pressure). Labyrinth seals separate the thrust bearings from the pressure at the
The setup provides the possibility of in-situ balancing by means of small balancing weights which are mounted in the compressor inlet nose and at the turbine end. The two-plane influence coefficient method is used to determine the rotor imbalance. This two-plane method is adequate since the target speed stays far below the first bending mode of the rotor.

Figure 3 shows the real rotor assembly as used in the tests described below.

Instrumentation

Figure 4 shows the complete instrumented setup 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 spark eroded 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 in-house developed single-fiber optical probes, based on reflected light intensity. The probes are positioned on the 20 mm rim of the turbine and the nose of the compressor. A data acquisition system from National Instruments (PXI-6123 with 500 kHz simultaneous sampling rate) is used to monitor rotor vibration.

Fig. 2: Rotor assembly and air bearings.

Fig. 3: Assembled rotor mounted in the air bearing unit.

Fig. 4: Instrumented turbo-shaft setup. Top: complete setup. Bottom: detail.
MEASUREMENT

Rotor dynamics

Bearing instability currently limits the speed to 261 krpm (4350 Hz). Figure 5 shows the spectra for radial vibrations measured at this speed on both the compressor and turbine discs. The synchronous peaks at 4350 Hz are caused by the residual imbalance after balancing. Figure 6 shows the rotor has passed the conical critical speed but not yet the cylindrical one.

Fig. 5: Radial rotor vibration spectrum measured at a rotor speed of 4350 Hz (261 krpm).

The subsynchronous peaks at 2.2 kHz in figure 5 appear suddenly and rise very fast when approaching the maximal speed, and indicate that a crash is near. They correspond to a conical mode that becomes self-excited at a rotational frequency of about 4.3 kHz. As this type of whirl typically occurs at about half or a fraction of the rotational frequency, it is often called half-speed or fractional speed whirl. However, this terminology is misleading as there is no direct relation between the frequency of the instability and the rotational speed.

The other two peaks in the spectrum are mainly prominent on the turbine side. Their origin is not clear to the authors at this moment.

There are actually 4 vibration modes or whirls: 2 conical and 2 cylindrical ones, for both types a forward and a backward one. These modes can be excited at synchronous speed or become self-excited at any speed. Thus the same mode can be excited by both mechanisms as illustrated by the waterfall diagram in figure 7: The same conical mode is excited synchronously by rotor imbalance when rotating at 2.2 kHz, and by self-excitation when rotating at 4.35 kHz. The eigenfrequency of the whirl is nearly identical in both situations. However, the consequence of both situations is different. In case of synchronous excitation, the rotor can pass the resonance by quickly running through the ‘danger zone’, while in the self-excited situation the whirl cannot be passed and forms a real upper limit for speed.

New externally damped aerodynamic bearings (discussed in a separate paper) have been developed that suppress this instability and will allow the turbine to be tested at its design speed.

Conical instability (subsynchronous)

Conical critical speed

Fig. 6: Synchronous rotor vibrations.

A compressor map is constructed as shown in Figure 8, plotting pressure build-up against mass flow for different rotor speeds. The data is measured and plotted along constant speed lines. A speed is selected by controlling the turbine supply. The different data points along the lines of constant speed are obtained by varying the compressor load. As power...
consumption of the compressor changes with load, the turbine supply has to be slightly adjusted to keep speed constant.

Both numerical and measured characteristics are plotted on the same graph. Numerical results are given for speeds up to 500 krpm which is the nominal speed of the system. Measured results are given for speeds up to 240 krpm (4 kHz). Measurements are not at exactly the same speed as the theoretical curves, and therefore 2 measured curves are slightly extrapolated to the speed of the neighboring theoretical curve: 240 krpm extrapolated to 250 krpm, and 210 krpm extrapolated to 200 krpm. For extrapolation, pressure is assumed to vary quadratically with speed, while mass flow is assumed to vary linearly with speed.

**DISCUSSION**

Theoretical and measured results coincide very well, considering the uncertainty on the CFD calculations at low Reynolds numbers [5] (especially regarding the simple Baldwin-Lomax turbulence model) and the use of a tubular diffuser in the experimental setup as opposed to the annular diffuser in the numerical model. As the compressor is now operated at approximately half the nominal speed, the generated pressure is still low (0.41 bar relative static pressure) due to the (more than) quadratic relationship with speed. Nevertheless the good agreement between numerical and measured results suggests that the target pressure ratio will be attained at nominal speed.

**CONCLUSION**

Bearing instability is currently limiting the rotational speed to 261 krpm, compared to the target speed of 500 krpm. As a result the relative static pressure generated by the compressor reaches only 0.41 bar. Nevertheless the good agreement between numerical and measured results in the lower speed range suggests that the target pressure ratio will be attained at nominal speed. New externally damped aerodynamic bearings have been developed that eliminate this instability and will allow the turbine to be tested at its design speed.

**REFERENCES**


