Abstract
Micro-turbo chargers were manufactured and operated to test the compressor of a palmtop gas turbine generator at low temperature (< 100 °C). Impellers are 10 mm in diameter and have 3-dimensional blades machined using a 5-axis NC milling machine. The performance of the compressor were measured at 50 % (435,000 rpm) and 60 % (530,000 rpm) of the rated rotational speed (870,000 rpm). The measured pressure ratio is lower than predicted value, mainly because impeller tip clearance was larger than designed value. The measured adiabatic efficiency is unrealistically high due to heat dissipation from compressed air. The maximum rotational speed achieved using hydroinertia and hydrodynamic air bearings is 566,000 rpm and 165,000 rpm, respectively.

1 Introduction
In coming aging society with less working population, self-powered mobile machines like an autonomous robot are expected to work at construction sites, farms, hospices, homes etc. For the power source of such self-powered mobile machines, light weight, that is, high power density is highly important, and refueling is more convenient than time-consuming recharging especially at construction sites and disaster areas where an commercial electric power line is not available. From the social needs, 10-1000 W class miniaturized gas turbine generators are under development [1-6], because a gas turbine is potentially advantageous in power density compared to fuel cells and other heat engines.

We are developing a palmtop gas turbine generator with specifications shown in Table 1, because output power of 1 to several hundreds W/unit is often required by the self-powered mobile machines, and the limit of the miniaturization based on existing gas turbine configuration seems to be around 10 mm in the diameter of impellers [7]. To realize such a palmtop gas turbine, we must demonstrate that

1) a compressor and turbine achieve a required adiabatic efficiency of 68 %,
2) a rotor stably rotates up to a rated rotational speed of 870,000 rpm,
3) a combustor stably operates at a rated temperature of 1,050 °C, and
4) the compressor is thermally isolated from a hot combustor and turbine to achieve required adiabatic efficiency.

In this study, micro-turbo chargers to demonstrate (1) and (2) at low temperature (< 100 °C) were manufactured and tested.

2 Structure
2.1 Overall structure
Figure 1 shows the cross-sectional structure of the turbo charger. The turbo charger is mainly composed of a compressor, a turbine, radial bearings and a thrust bearing. We tested two types of turbo charger: one with hydroinertia air bearings [6] and one with hydrodynamic air bearings. The details of the bearings are described in the next subsection. For the turbo charger with the hydrodynamic air bearings, the journal bearing air ports shown in Fig. 1 are not opened. The parts composing the rotor are made of titanium alloy with high specific strength, and the other parts are made of stainless steel.

Figure 2 shows the impellers with a diameter of 10 mm. The compressor impeller is designed to achieve possibly highest efficiency by computational fluid dynamics (CFD) simulation, and to keep required strength by stress and natural frequency analysis at the operation point shown in Table 1. The turbine impeller is designed in the same method to produce power required by the compressor (about 350 W) using cold driving air (< 100 °C). The impellers are machined using a tapered ball endmill with a diameter of 0.5 mm installed at a 5-axis NC milling machine (Toshiba Machine, F-MACH with an additional rotating stage).

The impellers and the shaft are assembled with pre-loading by the tie bolt and tie nut, and the rotor is balanced within 0.2 g·µm using a special rotational balancing machine (Kokusai). Because the rotor is disassembled and reassembled to be inserted into the turbo
charger, the repeatability of the rotational balance is checked by repeating assembly and disassembly. The rotational imbalance deteriorates approximately to 1 g·µm after reassembly.

2.2 Bearings

At the rated rotational speed of 870,000 rpm, no existing bearing can be used. For the turbo charger, we designed and tested two types of air bearings: hydroinertia air bearings and grooved hydrodynamic air bearings.

**Hydroinertia air bearing**

The hydroinertia air bearing is a kind of externally-pressurized air bearing, which has a 2-3 times wider bearing gap than that of a conventional hydrostatic air bearing [6]. The hydroinertia bearing uses not only static pressure rise at the narrow side of the bearing gap but also static pressure drop due to the acceleration of supersonic air flow at the wide side to return a shaft to the bearing center. Air bearings make restoring force by static pressure difference between the wide and narrow bearing gap. In a conventional hydrostatic bearing, static pressure is positive at both wide and narrow bearing gap, but in the hydroinertia bearing, static pressure at the wide bearing gap becomes negative, so that larger restoring force, that is, higher load capacity is obtained.

The drawback of the hydroinertia bearing is large air consumption and relatively low rigidity at low eccentricity. Also, the wide bearing gap makes it difficult to tighten impeller tip clearance. In the turbo charger, the radial and thrust bearing gaps are designed at 30 µm and 40 µm, respectively, using the calculation method developed by Hikichi et al [8]. The measured radial and thrust bearing gaps are 25 µm and 44 µm, respectively.

**Hydrodynamic air bearing**

For the final gas turbine generator, a hydrodynamic air bearing is preferable, because it does not consume a part of air from the compressor, and has high whirl stability. Among several types of hydrodynamic air bearing, we first selected a herring-bone grooved air bearing and a spiral grooved air bearing for radial and axial support, respectively, because they are relatively easy to fabricate for a small shaft. The bearings are designed using a standard design method in Ref. [9]. The designed bearing gaps are 4.4 µm and 5.5 µm for the radial and thrust bearing, respectively. To keep such narrow bearing gaps, highly precise fabrication technology is required.

Figure 3 shows the shaft with bearing grooves fabricated by photolithography and wet etching. The herring-bone grooves on the cylindrical surface are fabricated using a special maskless exposure system with a rotating sample stage (Ball Semiconductor). Positive photoresist (Clariant, AZ 4400) is uniformly spread on the cylindrical surface by dipping the shaft into the photoresist and suspending it in solvent vapor. The image of the herring-bones is generated on a Digital Micromirror Device (Texas Instruments) in the maskless exposure system, and projected to the cylindrical surface by rotating the shaft step by step. After development and hard baking, the shaft is etched by the mixture of fluoric acid and hydrogen peroxide.

To fabricate the spiral groove on the thrust disc, the photoresist is first spread on the shaft by spinning the shaft. The spiral groove pattern is transferred using a
special photomask with a hole where the shaft passes, and then the shaft is wet-etched. By repeating these steps, the spiral grooves are fabricated on both sides of the thrust disc.

3 Experimental

3.1 Experimental setup

The turbo charger is set with the turbine facing up and the compressor facing down. Driving air is supplied from a scroll compressor to the turbine through an air heater (Sakaguchi E.H VOC). The air heater is used, when turbine power generated by cold air is not enough to drive the compressor, but it was not used in this study. Air for the hydroinertia bearings is supplied from another scroll compressor, and its pressure is set at 0.5 MPa using a pressure regulator.

Rotational speed is measured by detecting a mark on the shaft using an optical fiber displacement meter (Philtec, D20) and counting up the pulse signals using a tachometer. Also, using similar optical fiber displacement meters, the radial and axial vibration of the rotor are measured at the center of the shaft and the back side of the turbine impeller, respectively. Temperature and pressure are measured using 10 thermocouples and 10 diaphragm pressure sensors (Keyence, AP-43/44), respectively. Air flow rate is measured using hot wire flow meters (STEC, SEF-52) at the inlets of the turbine and compressor, and using tapered tube flow meters at the outlets.

3.2 Experimental results

Compressor performance

Figure 4 and 5 show the pressure ratio and adiabatic efficiency of the compressor (compressor diagrams), which were obtained using the turbo charger with the hydroinertia bearings. The compressor diagrams were obtained at about 50% (435,000 rpm) and 60% (520,000 rpm) of the rated rotational speed. The measured pressure ratio is lower than predicted value. This could be mainly because the impeller tip clearance was too large compared to a designed value of 25 μm. In this study, the impeller tip clearance was set large enough to prevent the impeller tip from crashing to the shroud, because the machining error of the impeller and shroud as well as assembly error were not within tolerances, and the bearing gaps were large.

The measured adiabatic efficiency shows unrealistically high values. This is mainly due to heat dissipation from compressed air. Adiabatic efficiency, \( \eta_c \), is the ratio of required adiabatic work calculated from measured pressure ratio to actual work calculated from measured temperature ratio, and is given by

\[
\eta_c = \left( \frac{T_{in}}{T_{out}} \right)^{(\gamma-1)/\gamma} - 1
\]

where \( \pi \) is pressure ratio, \( T_{in} \) and \( T_{out} \) are inlet and outlet temperature, respectively, and \( \gamma \) is the ratio of specific heats. If \( T_{in} \) drops due to the heat dissipation from compressed air, the actual compressing work is underestimated, so that adiabatic efficiency is overestimated from the above equation. Heat isolation is necessary for compressed air, because heat dissipation relatively increases in such a small machine.

Characteristics of the hydroinertia air bearing

Using the hydroinertia air bearings, a rotational speed of 566,000 rpm, which is 65% of the rated rotational speed, was achieved. Figure 6 shows the vibration of the rotor 240 ms before the crash at 566,000 rpm. In the radial vibration, an oscillation with low frequency (envelope) represents the whirl of the rotor, and that with high frequency synchronizes with rotation. The latter was influenced by uneven reflection rate on the shaft where the optical fiber displacement sensor approached, and does not represent just the rotation-synchronous mechanical vibration.

The rotor crashed to the radial bearing due to the whirl. The whirl ratio (rotational speed/whirl speed) observed in Fig. 6 is 8. The whirl is the precession of a rotor which is excited by the component of shaft load normal to the direction of eccentricity due to
Fig. 6  Rotor vibration 240 ms before the crash at 566,000 rpm

hydrodynamic effect. The whirl speed is identical with the resonant speed of the bearing system. Thus, the product of the resonant speed and whirl ratio is the maximum achievable rotational speed. The resonant speed is determined by rotor weight and bearing stiffness, but the mechanism to determine the whirl ratio is not figured out. Simulation study based on CFD will help us to design the bearing with higher whirl stability.

Characteristics of the hydrodynamic air bearing

We tested the turbo charger with the hydrodynamic air bearing several times, but mostly the rotor crashed to the bearing below 100,000 rpm. The measured radial bearing gap is about 10 µm, which is approximately twice the designed value. From theoretical calculation shown in Fig. 7, the critical speed at this radial bearing gap is found to be several tens thousands rpm, and it is thought that the crash occurred due to the resonance of the bearing system. Using a better balanced rotor, the resonant peak becomes narrower, so that it becomes easier to pass the critical speed in acceleration. In this study, the rotor balance could not be good enough to pass the critical speed, because the rotor was disassembled and reassembled to be inserted into the turbo charger after balanced.

The maximum rotational speed achieved using the hydrodynamic air bearings is 165,000 rpm, which is in supercritical range. From the measured vibration of the rotor, we cannot find the reason of the crash. Further studies including fluidic simulation in the bearing gaps and the improvement of fabrication technology are necessary to approach the rated rotational speed.

4 Conclusion

In this study, a micro-turbo charger was manufactured and operated to test the compressor of a palmtop gas turbine generator at low temperature (< 100 °C). Impellers are 10 mm in diameter and have 3-dimensional blades machined using a 5-axis NC milling machine.

Fig. 7  Critical speed of the hydrodynamic air bearing

The performance of the compressor were measured at 50 % (435,000 rpm) and 60 % (530,000 rpm) of the rated rotational speed (870,000 rpm) by driving a turbine using compressed air at room temperature. The measured pressure ratio is lower than predicted value. This could be mainly because impeller tip clearance was larger than designed value. The measured adiabatic efficiency is unrealistically high due to heat dissipation from compressed air. The maximum rotational speed achieved using hydroinertia and hydrodynamic air bearings is 566,000 rpm and 165,000 rpm, respectively.

The obtained results are not satisfactory, but critical issues to be solved in future works were determined. This seems an important step toward the realization of the palmtop gas turbine generator.

Acknowledgements

This study was supported partly by the New Energy and Industrial Technology Development Organization of Japan, and partly by the Asian Office of Aerospace Research and Development.

References