FEASIBILITY STUDY OF 100WATTS CLASS MICRO TURBOCHARGER FOR MICRO GAS TURBINE ENGINE

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Abstract: A 100 Watts class Micro turbocharger for Micro Gas Turbine Engine is designed, fabricated and tested. The rotor is supported by Micro Foil Bearings. The compliant foil journal bearing and thrust bearing are designed to withstand high load of vibrations at the operational speed 870,000 rpm. The compressor of impeller diameter 10mm is designed to have a pressure ratio 3 at the operational speed. Test is executed in room temperature. The rotor is stably driven at 380,000rpm (43% of target). Pressure ratios are well matched with designed values.

Key Words: Micro Gas Turbine Engine, Foil Bearing, Turbocharger, Compressor Performance

1. INTRODUCTION

Micro Gas Turbine Engine is a new portable power source based on 100watts Class Micro Gas Turbine. It is required extremely high rotating speed to get enough compressor pressure ratios due to its small size. In addition, a temperature of turbine inlet part is very high (about 1,000 ℃). Existing rolling element bearings and conventional lubricating bearings have performance limits in these extreme environments (high operating speeds and temperatures). To overcome these problems, foil bearings are selected. Foil bearings have inherently low power loss as compared to rolling element bearings. It is about 5.5 times low losses at the operation speed 1 million rpm [1]. Also, foil bearings are available in high temperature with special coating on the shaft.

In other groups, the externally pressurized gas bearings are used and operated at 1.7 million rpm [2]. Compressor pressure ratio of 3 has been achieved by using hydro-inertia bearings [3]. However, these bearings need to be supplied compressed air for the operation. Therefore, hydrodynamic bearings such as foil bearings are advantageous for self-operation.

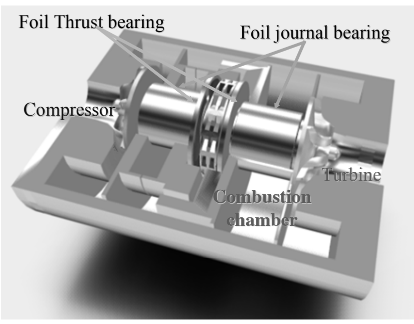


Fig. 1: 100watts class micro gas turbine engine

2. MICRO GAS TURBINE ENGINE

2.1 Concept of Micro Gas Turbine Engine

The main concept of Micro Gas Turbine Engine supported by foil bearings is shown in figure 1. The system consists of a compressor, a turbine, a generator and a combustion chamber. The rotor is like a small turbocharger that has an additional impulse turbine at the edge of a thrust collar. The compressed air from the compressor gives a drives to the impulse turbine. The generator magnet is positioned inside of a compressor part of rotor which is surrounded by a foil journal bearing and generator coils at the housing. The compressed air is also served as a cooling material by passing through a channel around the generator. It shields the compressor from the heat conduction and keeps the generator magnet temperature into the Curie temperature.

The system is based on the Brayton cycle. The preliminary cycle analysis results are shown in figure 2. The required operational speed to achieve pressure ratio 3 by an impeller diameter 10mm is 870,000 rpm [4].

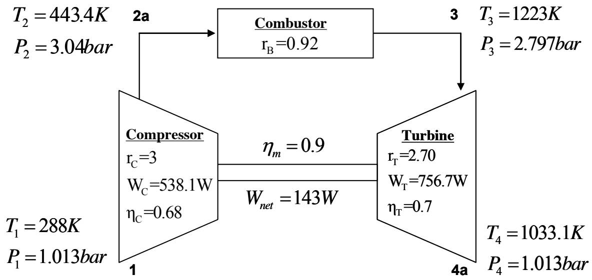


Fig. 2: Micro gas turbine cycle analysis

2.2 Micro Turbocharger

The most critical requirement for the gas turbine system is assumed to achieve the designed pressure ratio. A rotor speed affects mostly the compressor performance. For the first step of a development, we focus into the reliability of Micro compressor of the impeller diameter 10mm. Following figure shows the cross section of the micro turbocharger. We separate the combustion chamber out of the system in order to neglect thermal effect considerations. The rotor is supported by two compliant foil journal bearings in radial direction, two compliant foil thrust bearing in axial direction.

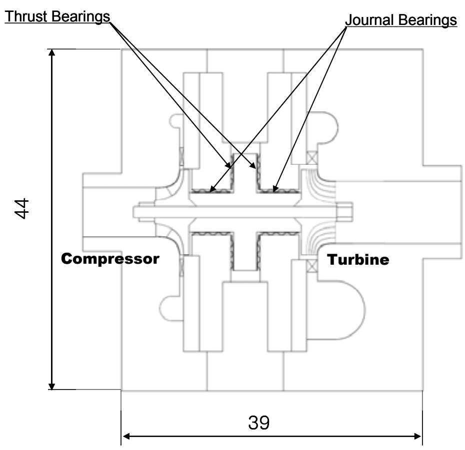


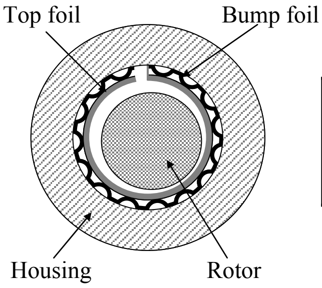
Fig. 3: Cross section of the micro turbocharger

3. SYSTEM DESIGN

3.1 Bearing Design

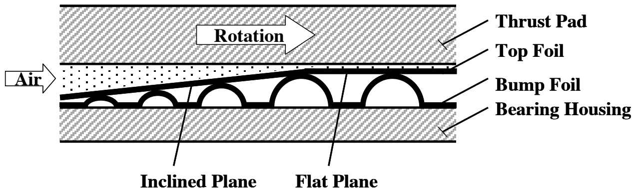
It is required to operate at 870,000 rpm to realize the gas turbine cycle. It means that bearings have more than 4 million DN values. Journal bearings should be sustainable in the harsh environments of high loads caused by synchronous vibrations with a large residual unbalance. Axial thrust forces of the impellers are considered for designing thrust bearings. Hence, a high-enough load capacity is the main keyword of the bearing design. A compliant foil bearing is designed via iterative procedures. It is designed, analyzed and scaled to the current size to establish the dynamic characteristics.

5mm diameter compliant foil journal bearings and 15mm out-diameter compliant foil thrust bearings are designed and fabricated. Their properties are shown in the following figure.



Foil Material	Inconel718
Top Foil Thickness	80μm
Bump Foil Height	280μm
Bearing Clearance	40μm

(a) Compliant foil journal bearing



Inner Diameter	5.5mm
Outer Diameter	15mm
Angle of Inclined Plane	13.5°
Number of Total Pad	4

Foil Material	Inconel718
Top Foil Thickness	80μm
Bump Foil Height	280μm
Bearing Clearance	40μm

(b) Compliant foil thrust bearing

Fig. 4: Micro foil bearings

3.2 Rotordynamic Analysis

The rotordynamic characteristics of a micro power system have previously been investigated by Lee *et al*. [5]. The rotordynamic model is established using a finite element program. The operating speed of the micro engine is beyond the first and second critical speed which is the rigid mode, and much lower than that of the third mode which is the first bending mode. 1st bending critical speed lies around 1.2million rpm. About 20% of margin, it is assumed that the operational speed 870,000 rpm has no critical mode vibration. For the assurance of stability of rotor-bearing system, the logarithmic decrements have been analyzed. The logarithmic decrements have all positive values, so that rotor is expected to be stable over the entire operating speed range.

3.3 Aerodynamic Design

The aerodynamic design has been executed along the designed Brayton cycle. Pressure ratio 3 with a mass flow rate 3.429 g/s is the target point at the operational speed (870,000 rpm). The compressor performance is predicted by 3-dimension numerical calculation. Figure 5 shows the design-ed compressor performance map. Pressure ratios are predicted along the rotational speed ratio 0.2 to 1.1. It is assumed that the pressure ratio increases exponentially with higher rotational speed.

In the case of designing rotor blades, a conventional 3D rotor shape are designed and fabricated for the highest possible performances. The compressor of 10mm impeller diameter has 6 blades and 6 splitters. The turbine with same size of compressor has 11 blades. Considering the fabrication of the impellers, the turbine is made of Inconel718 for the use in a high temperature. The compressor is made of SUS301 for the ease of manufacturing and a weight balance with turbine. Both impellers are machined by 5 axis NC milling machine.

4. EXPERIMENTAL TEST

4.1 Experimental Setup

The rotor and the turbine are made of Inconel 718 for a high temperature operation.

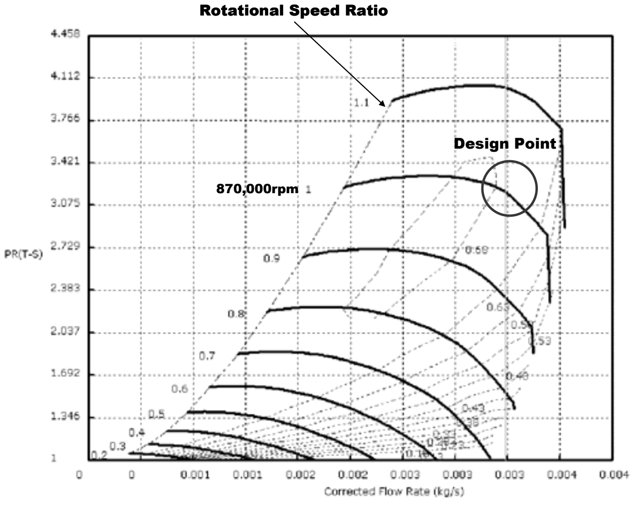


Fig. 5: Compressor performance map

Fig. 6: 10mm diameter compressor impeller

However, Tests are examined at room temperature (25℃) in this case. The main purpose of the test is to estimate the compressor performance of micro turbocharger by measuring of rotational speeds and compressor exit pressures.

The test rig view of micro turbocharger is shown in figure 7. The rotor vibrations are measured by Fiber Optic Displacement Sensor (*Philtec RC60*). The radial displacements are measured at the edge of a thrust collar. The axial vibration is measured at the center of a nut of the compressor impeller. The rotational speed is calculated by FFT analysis from the measured data. Pressure transducer (*Entran, EPI-L21-7B*) is installed at the exit of the compressor scroll. The compressor exit is connected with a mass flow meter (*Bronkhorst, F-113AC-aAHD-55-V*). A precise gauge valve is installed at the end of the mass flow meter, controls the compressor exit mass flow rate.

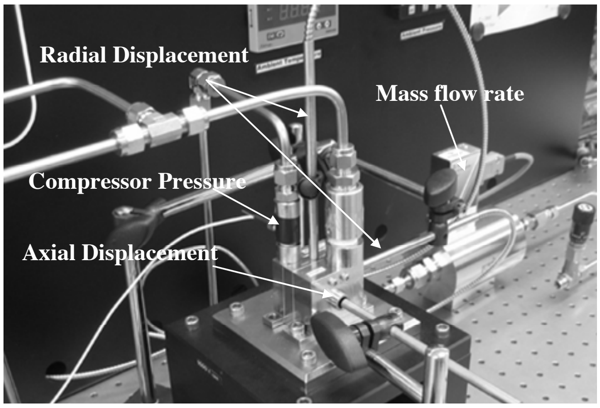


Fig. 7: Test rig setup

4.2 Experimental Result

The test of micro turbocharger of 5 mm shaft diameter has been performed. The test time was almost 8 minutes, and the rotor has stably been operated at 380,000 rpm for all test time. Figure 8 is the waterfall frequency spectrum which shows the operation of micro turbocharger at a startup.

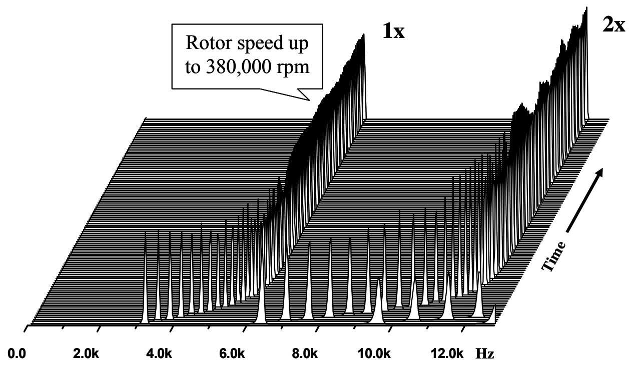


Fig. 8: Waterfall frequency spectrum at startup

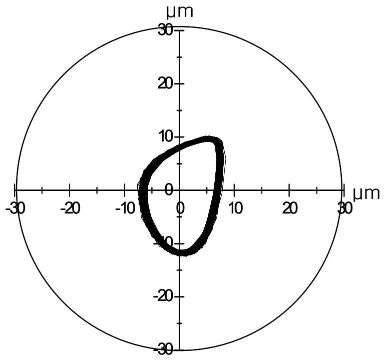


Fig. 9: Orbit at the rotational speed 380,000 rpm

For the operation at 380,000 rpm, the rotor synchronous and subsynchronous responses are both well controlled. The main portion of the rotor response is synchronous and the amplitude of synchronous vibration in all directions is about 20μm. No subsynchronous or other responses appeared. These clear frequency spectrums indicated a well damped rotor-bearing system.

The compressor is tested at the tip clearance 0.050 mm. Mass flow rate. The test results show a reasonable match of the experimental values to the designed values. It is expected that the pressure ratio would be increased if the impeller rotates faster.

5. CONCLUSION

In this paper, 100 watts class Micro turbo charger supported by Micro foil bearings is designed, fabricated and tested. The rotor is stably driven at 380,000 rpm and gained proper pressure ratios. However, the operational speed was only 43 % of the designed speed. The higher rotor speed will improve the general performance of Micro Gas Turbine Engine.

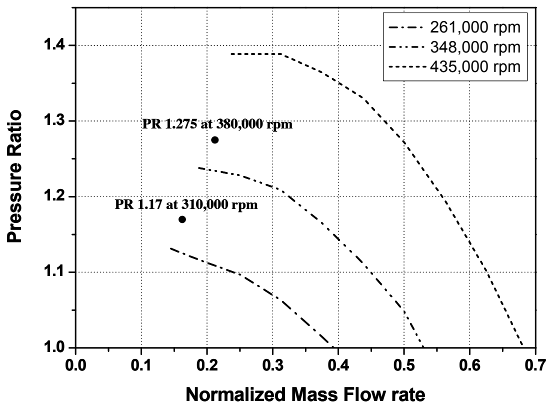


Fig. 10: Compressor test results vs. Design values

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