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EXPERIMENTAL VERIFICATION OF THE FEASIBILITY OF A 100W CLASS MICRO-SCALE GAS TURBINE AT IMPELLER DIAMETER 10MM

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ABSTRACT

Feasibility of a 100W class micro-scale gas turbine with centrifugal impeller of diameter 10mm has been studied experimentally. The rotor that is required to rotate at 870,000 rpm to generate the compressor pressure ratio 3 has successfully been achieved the stable operation by using hydroinertia gas bearings. The compressor efficiency higher than that required by the target cycle has been measured. The combustor has achieved stable self-sustained combustion at the combustion efficiency higher than 99.9%. The heat conduction analysis based on measured data showed that it is possible to keep the compressor under the compressor exit temperature 170degC when the turbine inlet temperature is 1050degC. Based on these results, the feasibility of the micro-scale gas turbine at impeller diameter 10mm has successfully been proven at component level.

INTRODUCTION

Since after the MIT group started the micro-engine project [1] to develop a shirt-button size gas turbine fabricated by MEMS (Microelectromechanical Systems) technology, researches on gas turbines at micro-scale are taking place at variety of place over the world due to its strong technological impact. Meanwhile, recent rapid advancement in autonomous robots and mobile electric equipments are requiring better mobile power source than batteries. Although, battery is a good power source for applications that require large current to power many servomotors, it has a weakness of low energy density that the operation time before recharging is short. These day's humanoid robots can operate only about 30 minutes after half a day of battery recharging. This will largely limit the applications of humanoid robots and mobile electric devices, and the improvement of the battery performance is desired. However, a lot of technology improvements have already been achieved over decades, and there is not much room left for improving the energy density of batteries. Hence, increase of the operable time of the autonomous robots by battery will require proportional increase of battery weight, which in turn, requires more power. This will cause a snowball effect that makes the system large and heavy.

To increase the energy density to achieve long operable time, some people are expecting the development of fuel cells. Fuel cell is known for its high energy density and high thermal efficiency. However, it should be noted that the power density of the fuel cell is low. It can be used to power applications that require high voltage with small current, but it does not fit to power electric motors. Large current density causes significant drop of the efficiency in fuel cells. Recent intense effort of developing the fuel cells for automobiles may someday enable high power density fuel cell, but it will not be in the near future.

Power sources that have both high power density and high energy density are internal combustion engines. Among variety of internal combustion engines, the type that fits for micro-size power sources for autonomous robots would be the continuous rotation type without any friction seals and with continuous combustion, because the surface roughness, and therefore the friction loss, and the heat dissipation will become relatively large at micro-scale. One of such types of internal combustion engines is gas turbine. Gas turbine at micro-scale can be a good candidate for the power source for mobile machines.

To realize a small power source for autonomous robots, development of the micromachine gas turbine generator is underway at a group under Tohoku University [2]. Currently, the project is trying to prove the feasibility of the micromachine gas turbine by component tests.

OBJECTIVE OF THE RESEARCH

The objective of this research is to clarify the feasibility of a 100W class micro-scale gas turbine by experimentally proving if each component can achieve the performance required to realize the gas turbine engine cycle.

REQUIREMENTS

The target cycle of the 100W class gas turbine generator is shown in Fig.1. The selected compressor pressure ratio is

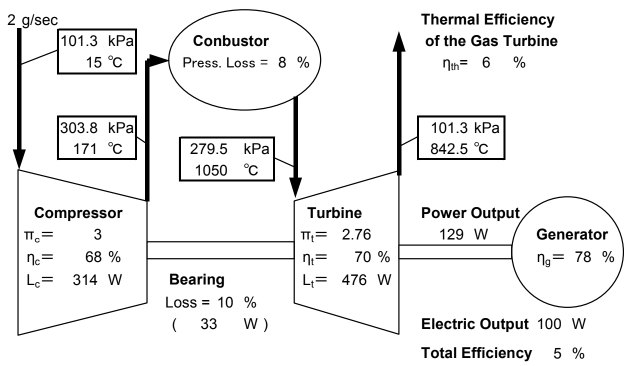


Fig.1 Target cycle of the 100W class micro-scale GT

3 and the flow rate is 2g/sec. By separate analysis, the diameter of such a compressor was found to be about 10mm. Since the pressure ratio of the centrifugal compressor is a function of the rotor tip speed, the rotational speed increases to achieve the same pressure ratio by smaller diameter, and for an impeller of diameter 10mm, the rotor is required to rotate at 870,000rpm to produce the pressure ratio 3. Based on this cycle, the major critical requirements for each component to realize a 100W class gas turbine are found to be as follows.

(1) Bearing to stably operate at 870,000rpm,
(2) Compressor to achieve adiabatic efficiency higher than 68%,
(3) Combustor to achieve self sustained combustion, and
(4) Compressor to be sufficiently isolated from the heat of the combustor and the turbine.

BEARING PERFORMANCE

The required rotor speed results in DN value of 3,480,000 at shaft diameter 4mm which is determined from structural requirement. This DN value is not practically realizable by today's ball bearing technology. Hence, air bearing has been selected. However, gas bearings have weakness of whirl instability at high speed. The hydrostatic gas bearings, which are the most popular type of gas bearings, are known to have the whirl instability speed at half the rotor speed, and that the whirl speed cannot go beyond the first flexure resonance speed. Hence, it is not possible to realize 870,000 rpm by hydrostatic gas bearings. Another popular type of gas bearings is hydrodynamic gas bearings. Previous study on whirl instability of various hydrodynamic gas bearings by Isomura et al. [3] showed that some small stably operable conditions exist for herringbone type and lobe type hydrodynamic gas bearings, but bearing clearance has to be as small as 3 to 4μm. This bearing clearance seems to be too small to be practical, because heat generated by the viscous shear flow in the small bearing clearance will cause heat expansion of the shaft and the bearings, and therefore, it would be difficult to maintain the range of the bearing clearance required for the stable operation.

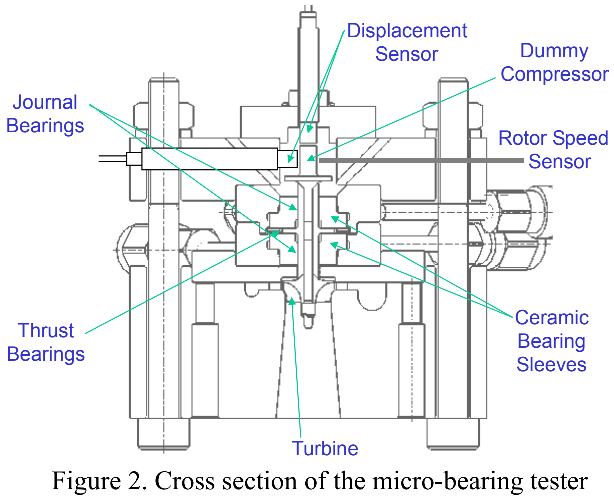


Figure 2. Cross section of the micro-bearing tester

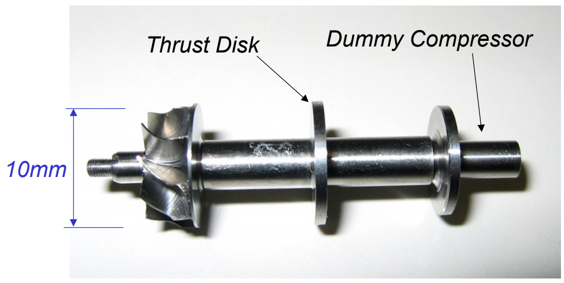


Figure 3. Rotor used for the micro-bearing test

Hence, the candidates of gas bearings to use for the micro-turbo machines are narrowed down to either foil bearings that has a compliancy, or hydroinertia gas bearings that has large bearing clearance and large anti-whirl stability. Hydroinertia gas bearings are a type of hydrostatic gas bearings with large bearing clearances. The whirl frequency of the hydroinertia gas bearings is known to be higher than that of hydrostatic gas bearings. These characters of the hydroinertia gas bearings are preferable at micro-scale. Hence, hydroinertia gas bearings have been studied and tested.

The cross section of the micro-bearing tester and the rotor developed for the test are shown in Figure 2 and 3, respectively. A turbine is located at the end of the shaft to power the rotor made of Ti-6Al-4V, and a dummy compressor is located at the other end to enable high precision displacement measurements by ϕ3mm eddy current sensor. The thrust disk is located at the center. The diameter and the length of the journal bearings are 4mm and 2.7mm, respectively. The outer and inner diameters of the thrust bearings are 10mm and 5mm, respectively. The bearing supply gas pressure can be changed for each thrust bearing and 2 each adjacent supply gas pressure holes of journal bearings. Each journal bearing has 8 supply gas holes. The bearing clearances are 31μm for the journal bearings, and 17 μm for the thrust bearings at their center position.

After controlling several design parameters, the bearings has run stably up to 891,000 rpm (Figure 4), at the supply gas pressure 700kPa for the thrust bearings, and 600kPa and 150kPa for cross diagonal directions of the journal bearings. Stable operation of the bearings at a speed higher than the required speed has successfully been achieved.

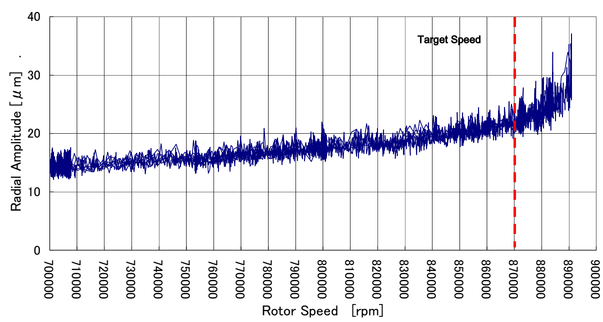


Fig.4 Radial amplitude of the rotor vibration

COMPRESSOR PERFORMANCE

Using the rotor and the bearings proven for the stable operation at the design speed by the micro-bearing tester, the compressor performance test has been conducted up to 83% of the design speed. The same bearing sleeves, rotor shaft and the drive turbine impeller are transplanted from the micro-bearing tester to the micro-compressor tester. A conventional three-dimensional rotor shape has been selected for the compressor rotor to achieve the highest possible performance. The cross section of the compressor test rig and the tested compressor are shown in Figure 5 and 6, respectively. Both the compressor and the drive turbine have 10mm of the diameters and 0.1mm of the minimum blade thickness. All the rotating parts are made of Ti-6Al-4V and the impellers are machined by micro-5-axis NC milling machine (Toshiba Machine Co., Ltd, F-mach with an additional rotating stage), using a tapered ball end mille of diameter 0.5mm and taper angle 3 degrees. The passages around the compressor and the exit scroll are made of Poly-Ether Ether Keton (PEEK) to reduce the heat leakage. The heat conductivity of PEEK is 2.4W/m-K, while that of stainless steel is 17W/m-K. The heat leakage is proportional to the surface area (-L2) and the heat generation is proportional to the volume (-L3). Thus the relative heat leakage is proportional to the inverse of the length scale, and becomes non-negligibly large at micro-scale. Therefore, it is important to obtain sufficient heat shielding for accurate efficiency measurement.

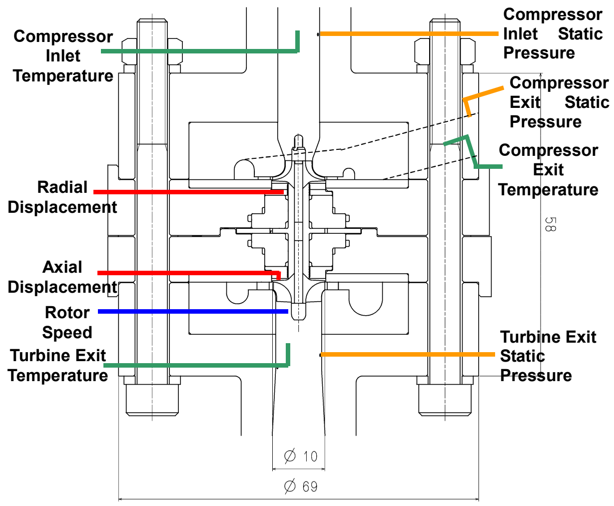


Figure 5. Cross section and the instrumentation of the micro-compressor test rig

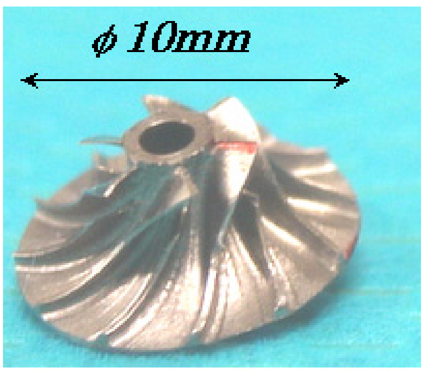


Fig 6. Close-up view of the micro-compressor impeller

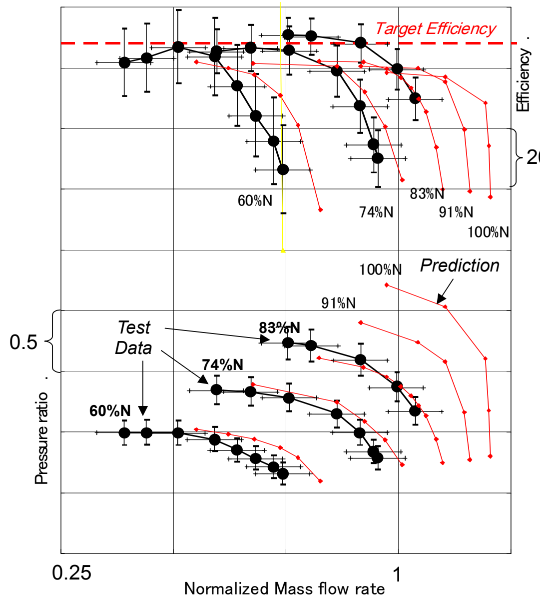


Fig 7. Micro-compressor test results

The rotor vibrations in axial and radial directions are monitored by laser reflection type displacement sensors of diameter 0.8mm. The static pressures and the total temperatures are measured at both the inlet and the outlet of the compressor. The measurements at the outlet of the compressor are made at the exit of the scroll, which is not shown in the cross sectional view of Figure 5. K-type thermo-couples of diameter 0.5mm without the sheath at the measurement point are used for temperature measurement. The total pressures are calculated using the cross sectional area of the measurement points.

The compressor is tested at the tip clearance 75µm. The test results showed reasonable match of the pressure ratio and the efficiency to the design prediction (Figure 7) by 3-dimensional numerical simulation. The maximum adiabatic efficiency larger than that required by the target cycle has been achieved, and the feasibility of the micro-compressor has been proven.

COMBUSTOR PERFORMANCE

Micro-combustors for both Hydrogen fuel and Methane fuel have developed and tested. The hydrogen combustor has a capacity of 2cc, and the Methane combustor has a capacity of 17cc. Can type combustors are selected for both combustors after a preliminary double scale test showed an order larger heat loss for an annular type combustor. The micro-combustor for hydrogen fuel is shown in Figure 8, and the test results for both of the fuel are shown in Figure 9. The combustion efficiency as high as 99.9% have been achieved with a self sustained combustion, and the NOx emission has been less than 3 ppm. The feasibility of the micro-combustors has successfully been proven.

HEAT ISOLATION

Shielding of the heat between the hot parts, such as the combustor and the turbine, and the compressor is very important because the compressor efficiency drops drastically by the wall temperature increase, as Isomura et al. [3] showed. The capability of the heat shielding has been assessed by using the data that measured in micro-combustor tests. The micro-combustor consists of combustion region inside the liner, and the low temperature air flowing between the liner and the outer housing. The temperature drop due to this low temperature air flow is measured during the micro-combustor test, and the measured data has been used

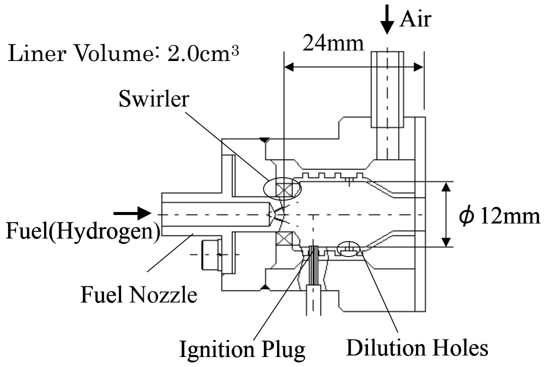
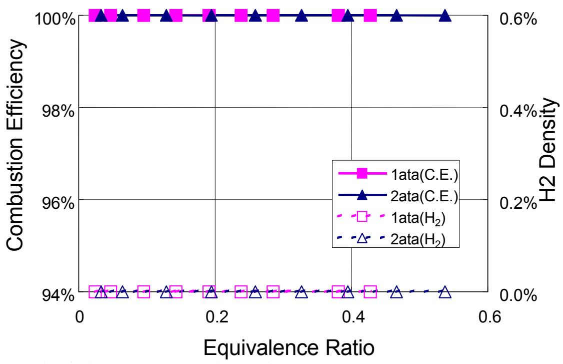
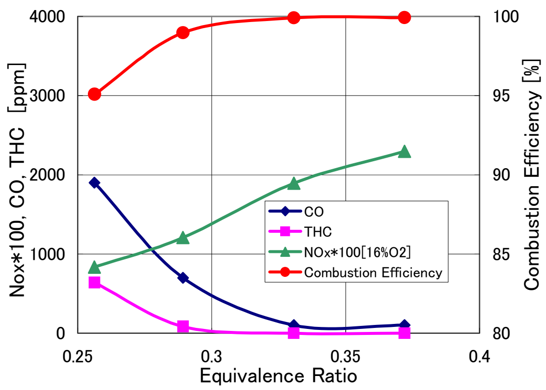


Fig 8. Cross section of the 2cc micro-combustor for Hydrogen fuel



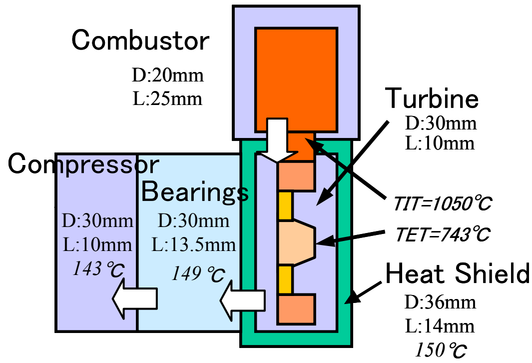
(a) Hydrogen micro-combustor



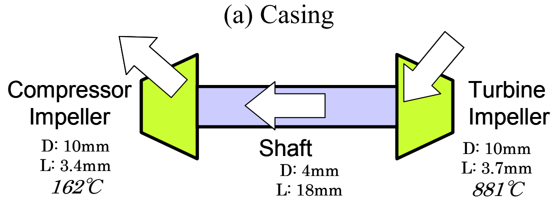
(b) Methane micro-combustor

Fig 9. Micro-combustor test results

to calculate the capability of the heat shielding. The calculation results in Figure 10 show that both the compressor impeller and the wall temperature can be



(a) Casing



(b) rotor

Fig 10. Heat conduction calculation results

reduced to the temperature under the compressor exit temperature when the turbine inlet temperature is 1050degC. Hence the feasibility of the heat isolation has been proven.

CONCLUSION

From the micro-combustor test, micro-bearing test, and the micro-compressor test, the four major critical requirements for each component to realize such a gas turbine have been proven.

(1) Bearing has stably operated at above 870,000rpm.
(2) Compressor has achieved adiabatic efficiency higher than 68%.
(3) Combustor has achieved self sustained combustion.
(4) Compressor has been shown to be sufficiently isolated from the heat of the combustor and the turbine.

Hence, the 100W class micro-scale gas turbine with impeller diameter 10mm has been proven to be feasible.

ACKNOWLEDGMENT

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