

DESIGN AND EXPERIMENT OF 400,000 RPM HIGH SPEED ROTOR AND BEARINGS FOR 500W CLASS MICRO GAS TURBINE GENERATOR

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Abstract: 500W class Micro gas turbine generator (MTG) is under development. It is designed to operate with the self-sustaining speed of 200,000rpm and the generating speed of 400,000rpm. It is composed of motor/generator, shaft, bearings, turbine, compressor impeller, combustor and recuperator and so on. The rotor is composed of inconel 718 shaft and $\text{Sm}_2\text{Co}_{17}$ magnet, and the small-sized air foil bearings are fabricated to support the high speed rotor. To evaluate the vibration of rotor and bearings at 400,000 rpm, high speed motor and motoring test rig are designed and fabricated. So far, we achieved the rotating speed of 280,000 rpm using the motoring test rig and small retrofit is ongoing. In this study, we present the design procedure of rotor and bearings for the operating speed of 400,000 rpm and its experimental results such as rotor vibrations.

Keywords: Micro gas turbine generator, Air foil bearing, Ultra high speed motor, Rotor dynamics

INTRODUCTION

Recently, unmanned mobile robots such as UGV (Unmanned Ground Vehicle), UMS (Unmanned Maritime System), UAV (Unmanned Air Vehicle) and humanoid robot to perform the dangerous missions for the soldiers are actively under development all around world, especially in USA, Israel and France. One of the biggest barriers for the unmanned mobile systems is to secure the small-sized high-power energy source which can provide the long-term energy with high energy density. While the battery has high power density, it has drawbacks that its energy density is low, the continuous runtime is short and the recharging time is long. The fuel cell, the other type of mobile energy source, has high energy density, whereas it has low power density and it is not easy to apply to the electric motor. Based on these reasons, MTG (Micro Gas Turbine Generator) is focused as a new energy source for the unmanned mobile robots because it can meet the requirement of both of high energy and power density, and Japan, Belgium, Switzerland and USA are continuing the research to develop it [1-3]. Currently, 500W class MTG is under development in Korea. Self-sustaining speed and generating speed of 500W class MTG is 200,000rpm and 400,000rpm respectively, which are determined from the cycle analysis. Rotor and bearings are very important components for the stable operation at the ultra high speed of 400,000rpm and should be designed to have the structural strength and avoid the critical speeds. MTG rotor is designed to have the diameter of 10mm and the first critical speed of 448,000 rpm, and the motoring test rig composed of rotor, air foil bearings and high speed motor is designed and fabricated to evaluate the motoring stability of rotor and bearings. This study presents the design procedure of rotor and bearings for MTG and the test results using the motoring test rig up to 280,000rpm.

CONFIGURATION OF MTG

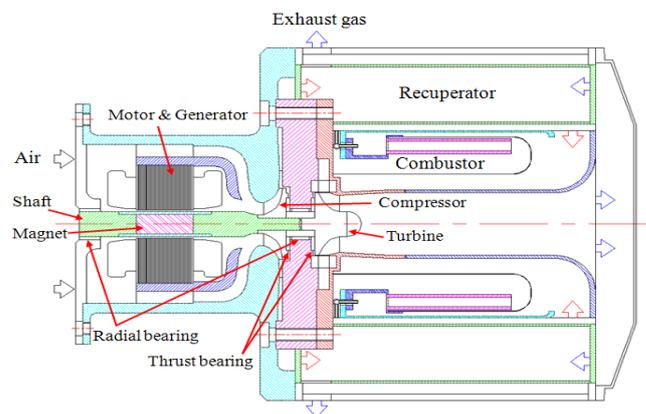


Fig. 1: Preliminary layout of 500W class MTG.

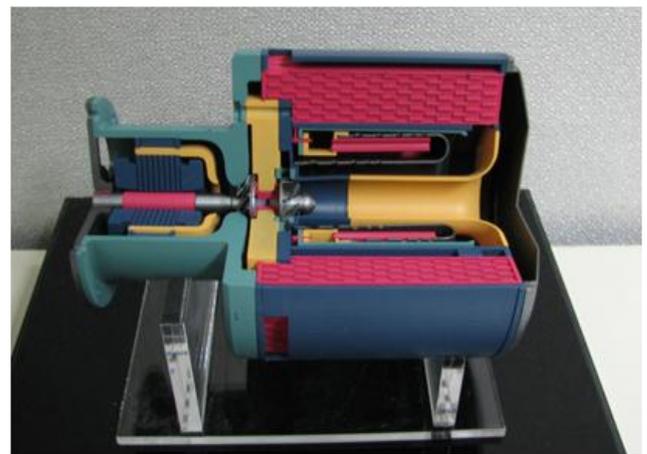


Fig. 2: Mockup of 500W class MTG.

Fig. 1 and Fig. 2 show the configuration and mockup of 500W class MTG. It is composed of motor/generator, centrifugal compressor, radial turbine, rotor/bearings, combustor and recuperator. The target volume of MTG is 1,000 cm³ which is the suitable small size to be the portable energy source, and the

rotor is designed to be connected with the compressor and turbine and drive them directly, which helps to minimize the volume. The rotor is supported by the air foil bearings which help to make MTG lighter and smaller. The motor/generator is a PMSM (Permanent Magnet Synchronous Motor) with 3 phases, 2 poles and 6 slots. Because it is also important to insulate the compressor/motor/generator part under the relatively low temperature from the combustor/turbine part under the high temperature, Zirconia with the low thermal conductivity of 3 W/m-K is selected as a thermal insulator.

DESIGN OF MTG ROTOR

Configuration of MTG Rotor

Fig. 3 shows the configuration of MTG rotor. The integral-type compressor-turbine shaft, $\text{Sm}_2\text{Co}_{17}$ permanent magnet for the motor/generator and shaft stud are fitted into Inconel 718 sleeve with the interference of 40 μm . In order to secure the rotating stability of MTG rotor-bearing system, the first bending critical speed should be more than 440,000 rpm, 110 % of the rated speed. To achieve the high critical speed, the shorter length and bigger diameter of the rotor is desirable. On the other hand, to achieve the structural strength for the ultra high speed of 400,000 rpm, the smaller diameter of the rotor is desirable. After the rotordynamic analysis and structural analysis as shown in Fig. 4 and 5, the diameter and the length of the rotor are determined to be 10 mm and 79.3 mm, respectively. The integral-type compressor-turbine shaft helps to increase the structural strength compared to the assembled-type compressor-turbine shaft. Based on the rotordynamic analysis, it is predicted that free-free mode first bending frequency would be 5,952 Hz and the first bending critical speed would be 448,000 rpm with about 11 % margin.

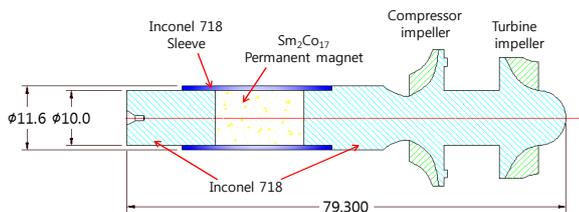


Fig. 3: Configuration of rotor.

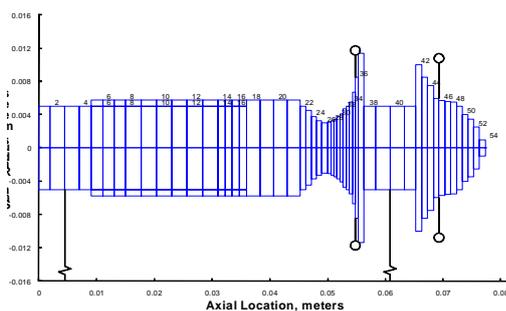


Fig. 4: Rotordynamic analysis model for rotor.

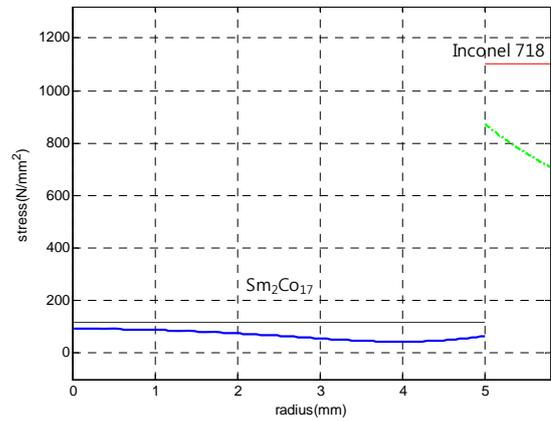


Fig. 5: Von Mises stress in $\text{Sm}_2\text{Co}_{17}$ permanent magnet and Inconel 718 sleeve for 400,000 rpm and 30 °C.

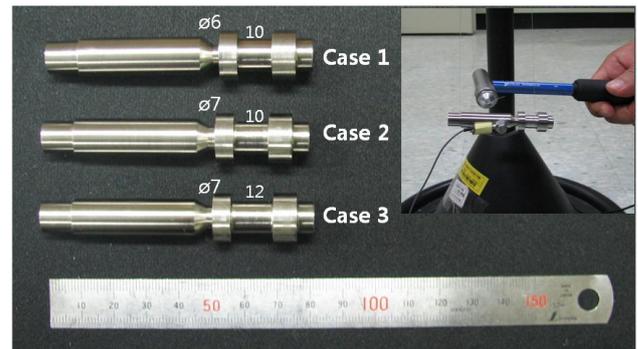


Fig. 6: 3 cases of test rotors for impact test.

Design Verification of MTG Rotor

In order to verify the bending frequency of the designed MTG rotor, three cases of test rotors are fabricated to perform the impact test as shown in Fig. 6. The differences between 3 cases exist on the diameter of the front neck of compressor inlet area and the length of the radial bearing area between compressor and turbine. The purpose of this experiment is just to find out how the predicted free-free mode bending frequency based on the rotordynamic analysis is close to the measured value and the construct the reliable rotordynamic analysis model. The configuration and materials of the dummy rotors is same as Fig. 3 except compressor and turbine impeller are dummy disk. It should be noted that the accelerometer is attached to the rotor for the impact test and it has a significant effect on the bending frequencies because the size of accelerometer is relatively big compared with the test rotor. Therefore, the accelerometer should be modeled as an added mass in the rotordynamic analysis. Table 1 shows the first bending frequencies of 3 test rotors from the impact test and the rotordynamic analysis including accelerometer effect and they yield the pretty close results. After constructing the reliable rotordynamic analysis model as aforementioned, this model is applied to the MTG rotor in Fig. 3 and the free-free first bending frequencies and the first bending critical speeds are predicted as shown in Table 2. The analysis is performed for the cases of 'with and without accelerometer' for the future reference of the

impact test on the fabricated MTG rotor. Based on Table 2, the diameter of the neck and the length of radial bearing area is determined to be 7 mm and 10 mm, respectively.

Table 1: Comparisons of free-free first bending frequency of test rotors between impact test and analysis.

	Impact test	Analysis
Case 1	5,107 Hz	5,109 Hz
Case 2	5,778 Hz	5,780 Hz
Case 3	5,493 Hz	5,495 Hz

Table 2: Predicted free-free first bending frequencies and first critical speeds of MTG rotors.

	Free-free first bending		First critical speed [rpm]
	With accelerometer	Without accelerometer	
Case 1	5,424 Hz	5,952 Hz	403,000
Case 2	6,088 Hz	6,668 Hz	448,000
Case 3	5,745 Hz	6,277 Hz	418,000

MOTORING TEST OF MTG BEARING

Design and Fabrication of Motoring Test Rig

Motoring test rig as shown in Fig. 7 is design and fabricated to evaluate the performance of the radial air foil bearings. The 400,000 rpm class ultra high speed motor with the rated output power of 2 kW is equipped to the test rig to rotate the test rotor. The simple bar-shaped test rotor with the length of 81mm and the outer diameter of 10mm is composed of Inconel 718 shaft stud and $\text{Sm}_2\text{Co}_{17}$ magnet fitted to Inconel 718 sleeve. The free-free first bending frequency and the first bending critical speed are predicted to be 6949 Hz and 440,000 rpm, respectively. 4 eddy current type gap sensors are equipped at the front and rear part to measure the rotor vibration.

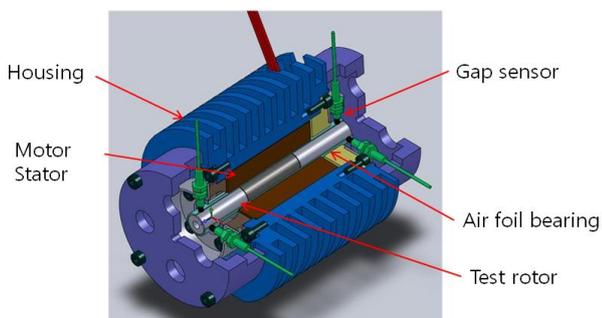


Fig. 7: Cross section view of designed motoring test rig.

Motoring Test Results of MTG Rotor

Fig. 8 shows the test rotors, the radial air foil bearing and the experimental setup for motoring test. The rotor vibrations are measured with 25kHz sampling rate. Unfortunately, the test rotors are

roughly balanced because it was impossible to arrange the balancing machine for the small size rotors such as the ones at this experiment in the near future. The radial vibration started to increase noticeably from about 200,000 rpm and the magnitude of x directional synchronous vibration of the front and rear part reached up to 20 μm and 25 μm at 280,000 rpm, respectively, and the motoring test stopped to prevent the damage of the bearings. Waterfall plot of the rotor vibrations for the rotational speed of 0 – 280,000 rpm is described in Fig. 9. Because the rigid body mode critical speed is less than 50,000 rpm, the vibration is mostly caused by the unbalance response.

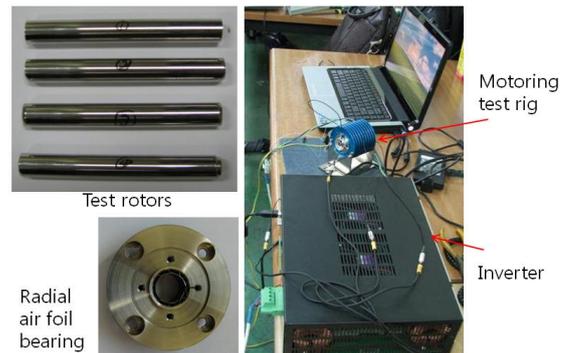
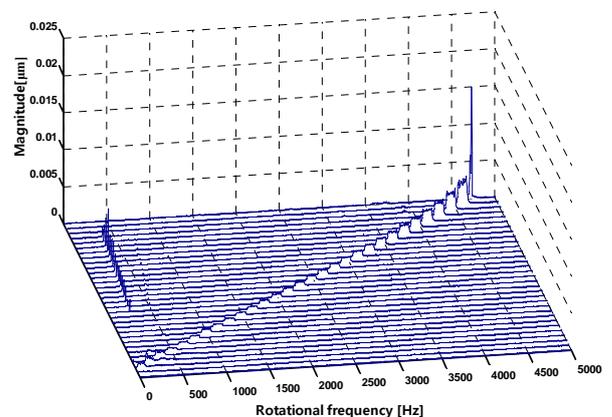
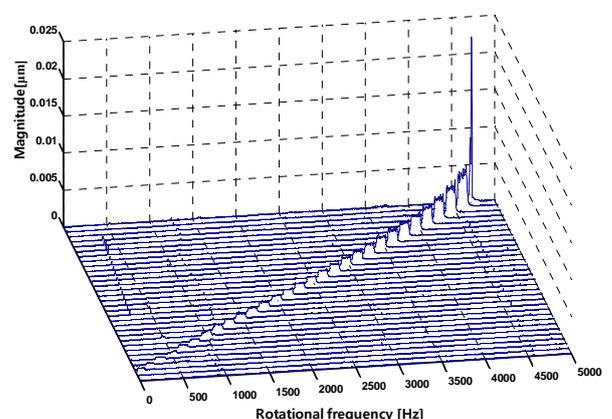


Fig. 8: Test rotors, radial air foil bearing and experimental setup for motoring test.



(a) x directional vibration of front part



(b) x directional vibration of rear part

Fig. 9: Waterfall plots of rotor vibrations for the rotational speed of 0 - 280,000 rpm

CONCLUSION

In this study, we presented the design procedure and experimental results of the rotor and bearings for 500 W class MTG with the rated speed of 400,000 rpm. The rotordynamic analysis model for MTG rotor is reliably constructed by using impact test for the test rotors. The motoring test rig to evaluate the radial air foil bearings is fabricated, and the motoring test was performed up to 280,000rpm. Although the motoring test couldn't be continued up to the rated speed, 400,000 rpm due to the big synchronous vibration caused by the unbalance response, there was no noticeable problem in the radial air foil bearing point of view. Currently, we arranged the balancing machine for the small size rotors and the motor test will be continued with the balanced test rotor. Furthermore, we are fabricating the high temperature motoring test rig to evaluate the reliability of the radial air foil bearing under the high temperature, and it will be continued after this motoring test under the room temperature finishes.

ACKNOLEGEMENT

This work was supported by the Next Generation Military Battery Research Center program of Defense Acquisition Program Administration and Agency for Defense Development

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