International Journal of Fluid Machinery and Systems
Vol. 2, No. 4, October-December 2009

Original Paper

Design and Prototyping Micro Centrifugal Compressor for Ultra Micro Gas Turbine

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Abstract

In order to investigate the design method for a micro centrifugal compressor, which is the most important component of an ultra micro gas turbine, an impeller having the outer diameter of 20mm was designed, manufactured and tested. The designed rotational speed is 500,000 rpm and the impeller has a fully 3-dimensional shape. The impeller was rotated at 250,000 rpm in the present study. The experimental results of the tested compressor with the vaned and the vaneless diffusers were compared. It was found that the vaned diffuser attained the higher flow rate than the vaneless diffuser at the maximum pressure ratio. In addition the maximum pressure ratio was higher for the diffuser having a larger diffuser divergence angle at the high flow rate. These results were compared with those obtained by the prediction method used at the design stage.

Keywords: Centrifugal Compressor, Performance Characteristics, 3-dimensional Impeller, Prediction method

1. Introduction

Studies for an ultra micro gas turbine have been actively tried to use for very small mobile electrical power sources, ultra micro jet engines and so on, since the micro-electro-mechanical system (MEMS) and the micro-fabrication methods have been developed (Epstein et al., [1]-[9]). However, it is still unclear for the design methodology of ultra micro gas turbines, which is much smaller than the conventional micro gas turbines. In the conceptual design of an ultra micro gas turbine, the impeller diameter, D_2 of 4mm was considered for the feasibility of an ultra micro gas turbine including the manufacturing process (Epstein et al., [1]). For example, in the point of view of the surface roughness of an impeller, the minimum roughness is decided by the method of manufacturing. Thus, the friction loss due to the increase of relative roughness is very considerable as the decrease of an impeller diameter. Also in the Moody diagram, it is apparent that when the roughness increases, both the wall friction and the disk friction losses increase. In addition, for the miniaturization of a compressor, the rotational speed increases since the pressure ratio depends on the peripheral velocity. Thus, it is necessary to estimate the stress at the tip and the root of blades with the increase of rotational speed. Due to the eccentricity of the impeller and the shaft, it becomes difficult to decrease the tip clearance for a small impeller. The ratio of the clearance to the impeller outlet width increases as the decrease of the impeller outer diameter.

In the present study, in order to establish the design methodology of a micro centrifugal compressor, which is considered to be the most important component of an ultra micro gas turbine system, a 5 times model (D_2=20mm) of centrifugal impeller to the final target one was designed and tested by using a hot test rig, which enable to produce the test at the high rotational speed.

2. Design

Because the 10 times model to the final target impeller (D_2=40mm) was shown to be difficult to attain the high pressure ratio with the high efficiency (Hirano et al., [9]), a 5 times model (D_2=20mm) was designed and manufactured with the three dimensional configuration using a 5 axis numerical controlled machining. Moreover, three types of diffusers were designed and installed in the compression system in order to find out the matched one with the impeller.

Received May 9 2009; revised September 21 2009; accepted for publication September 28 2009: Review conducted by Prof. Seung Jin Song. (Paper number O09013)
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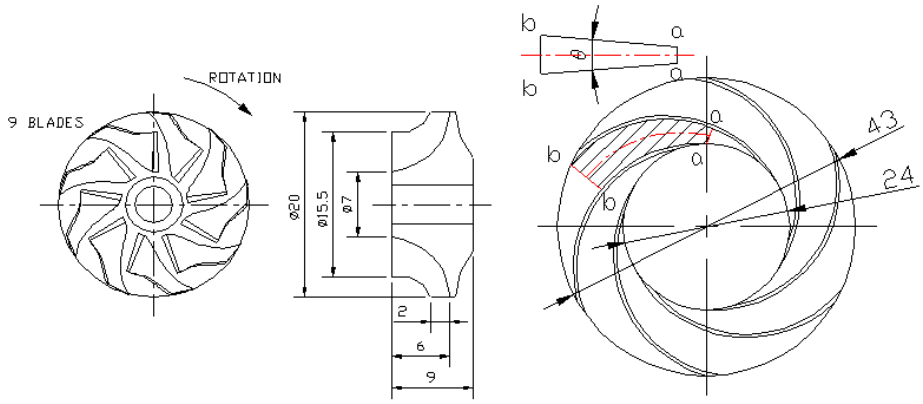
2.1 Design of compressor

The 5 times model was designed based on the design concept of a small turbocharger. The mass flow rate, G of 20g/s, the pressure ratio, π of 3 and the efficiency, η of 68% were estimated at the rotational speed, N of 500,000rpm. The stress and the vibration analyses for the impeller proved to be enough to rotate at 500,000rpm. Not only the vaneless diffuser but also two types of vaned diffusers with different divergence angle were designed in order to investigate the effectiveness of vaned diffuser for the ultra micro compressor. The diffuser inlet vane angle was matched to the flow angle at the diffuser inlet estimated for 250,000rpm. The two types of vaned diffuser, possessed the divergence angles of θ=10 deg. and θ=16 deg., respectively. The designed impeller and diffuser are shown in Fig. 1. The major dimensions of the impeller and the diffusers are given in Table. 1. Figure 2 shows the results by the FEM analysis for the stress and the deformation.

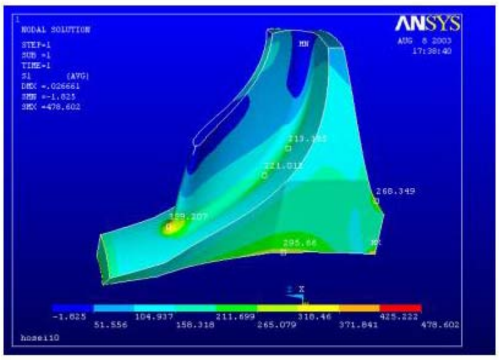


Unit:[mm]

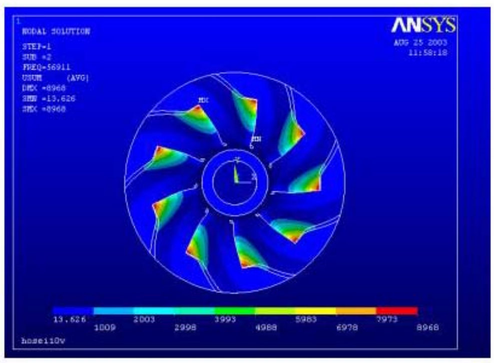
Fig. 1 Impeller and diffuser

Table 1 Main dimensions of impeller and diffuser

Impeller		Diffuser (vaned)	
Inlet diameter [mm]	15.5	Inlet diameter [mm]	24
Exit diameter [mm]	20	Exit diameter [mm]	43
Number of blades	9	Number of blades	6
Thickness of blades [mm]	0.34	Thickness of blades [mm]	0.5
Inlet blade height [mm]	4.25	Inlet blade height [mm]	2.15
Exit blade height [mm]	2.5	Exit blade height [mm]	2.15
Inlet blade angle [deg.]	51	Divergence angle [deg.]	10, 16
Exit blade angle [deg.]	30		



(a) Distribution of stress result



(b) Distribution of deformation result

Fig. 2 Stress contour and deformation mode due to centrifugal force (N=500,000rpm)

2.2 Prototyping of compressor

The pictures for the components of the ultra micro compressor are shown in Fig. 3. The compressor was driven by a turbine of a turbocharger of a small automobile engine. The compressor section consists of the impeller, the diffuser and the casing with the suction nozzle. The impeller was made of aluminum alloy (A2617). The suction nozzle was made of aluminum alloy and the diffuser and casing were made of brass. The diffusers with and without vanes were installed on the casing wall downstream of the impeller. The clearance between the shroud tip of the impeller and the casing wall was set to 0.25mm.

(a) Impeller (b) Diffuser (θ=10 deg.) (c) Diffuser (θ=16 deg.) (d) Casing

Fig. 3 Tested impeller, diffuser and casing by machining

3. Experimental apparatus and method

In this study, the hot test rig with a combustor was used to attain the higher rotational speed. The burned air-coal oil mixture from the combustor drove the turbine to rotate the coaxial impeller. The gas temperature at the combustor exit was approximately 873 K and the rotational speed was controlled by air flow rate and the fuel supply rate. The test rig is shown in Fig. 4.

The performance tests were carried out under the conditions of JIS B 8340. The flow rate was measured by the flow meter. The total pressure of the compressor at 32 mm downstream of the compressor outlet was measured by a micro pressure sensor inserted into the delivery duct. Also, the performance characteristics were obtained by measuring the inlet and outlet static pressures of the impeller, the static and the total pressures at the diffuser outlet and the temperature of air at the compressor outlet. The rotational speed was measured by a photo-electric revolution counter. The tests were carried out at 220,000rpm, 240,000rpm and 250,000rpm, respectively. The sites for measurement are shown in Fig. 5.

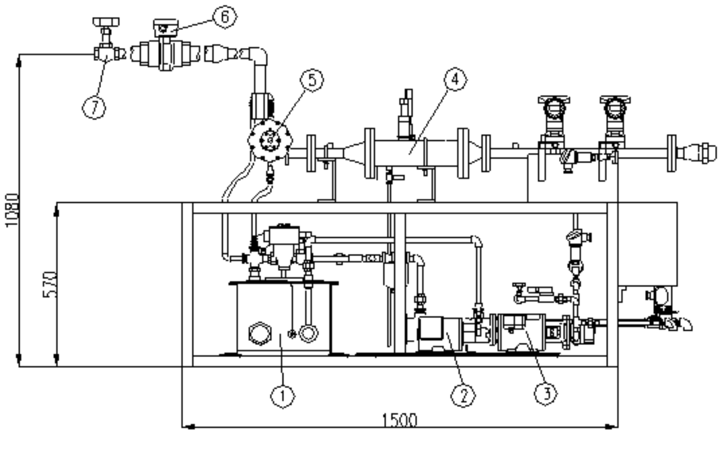
The pressure ratio π_t, static pressure ratio π_s, flow ratio ϕ are calculated by using the following equations.

$$\pi_t = \frac{P_t}{P_a} \tag{1}$$

$$\pi_s = \frac{P_s}{P_a} \tag{2}$$

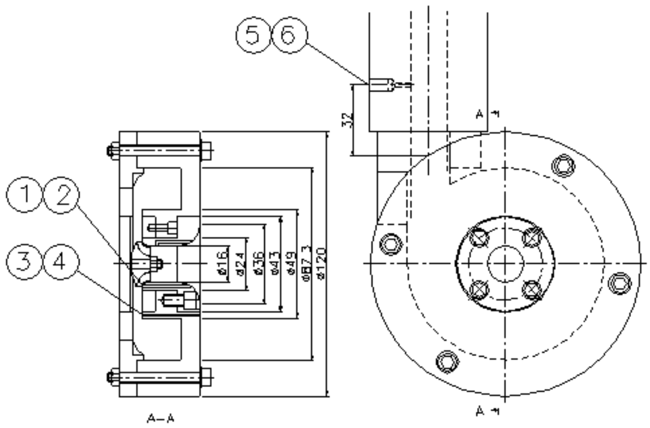
$$\phi = \frac{Q/60}{2\pi \cdot u_2 \cdot D_2 \cdot b_2} \tag{3}$$

Here the P_a is the atmospheric pressure, P_t is the total gage pressure, P_s is the static gage pressure, Q is the volume flow rate, u_2 is the tip speed, D_2 is the impeller outlet diameter, b_2 is the impeller outlet blade height.



(1)Tank (2)Oil Pump (3)Fuel Pump (4)Combustor
(5)Compressor (6)Flowmeter (7)Flow Control Valve

Fig. 4 Test rig with combustor



(1)Impeller outlet pressure (2)Impeller outlet temperature
(3)Diffuser outlet pressure (4)Diffuser outlet temperature
(5)Duct outlet pressure (6)Duct outlet temperature

Fig. 5 Measurement setup

4. Experimental results and discussion

Figure 6 shows the experimental results with the vaneless diffuser at 200,000rpm, 240,000rpm and 250,000rpm. The coordinate indicates the mass flow rate and the ordinate indicates the total pressure ratio to the atmospheric condition. At each rotational speed, the flow rates did not changed as the flow control valve was gradually closed from the full valve opening due to the choked flow rate. At the surge flow rate, the time-dependent pressure started to fluctuate. The maximum pressure ratios and efficiencies at the rotational speeds of 220,000rpm, 240,000rpm and 250,000rpm were π=1.25, η=58.5%, π=1.28, η=57.9% and π=1.32, η=57.8%, respectively. Figure 7 shows the experimental results of the tested compressor with the vaned and the vaneless diffusers at 250,000 rpm. The operating ranges of the vaned diffusers were narrower than that of the vaneless diffuser as expected at the design stage. The attained maximum total pressure ratios were higher for the vaned diffusers as also expected. Concerning the surge, the vaned diffusers exhibited much higher flow rates of the inceptions and their flow rates are nearly the same. When the effect of the divergence angle of the vaned diffuser was examined, the wider divergence angle exhibited the superior characteristics. Generally speaking, the divergence angle around θ=10 deg. has been considered to be appropriate for this type of vaned diffuser. However, in the present cases of the very small compressor, the wider divergence angle of θ=16 deg. exhibited the superior characteristics. The characteristics of the vaneless diffuser showed the cubic curves. The problem of matching between the impeller and the diffuser should be investigated in more detail.

In addition, these results thus obtained were compared with those obtained by the prediction method used for the design. In the prediction method, the inducer incidence, the wall friction, the secondary flow, the leakage flow, the mixing and the disk friction losses were taken into account. The relationship between the blockage at the diffuser throat and the pressure recovery coefficient was also estimated. The Wiesner's formula was used for the slip coefficient (Galvas, [10]). Figure 8 and Figure 9 show the performance characteristics estimated by the prediction method for the compressor performance at the impeller outlet and diffuser outlet, respectively. The predicted results showed the good agreements with the experimental ones. However, the accurate prediction of the performance characteristics was difficult, especially near the surge flow rate, because the parameters for the design of a compressor with larger diameters than that of the present study were used conventionally.

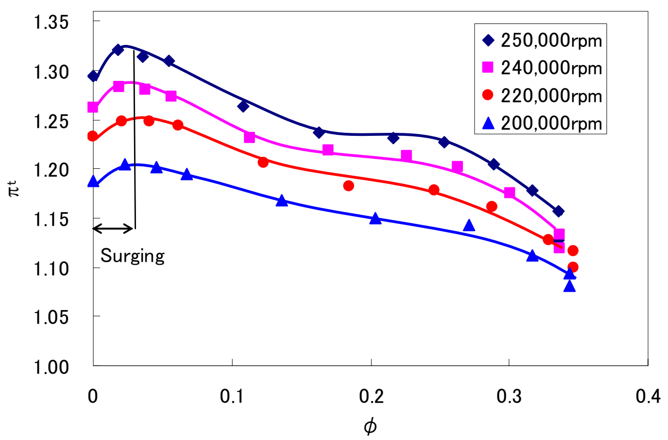


Fig. 6 Non-dimensional total pressure ratio (vaneless)

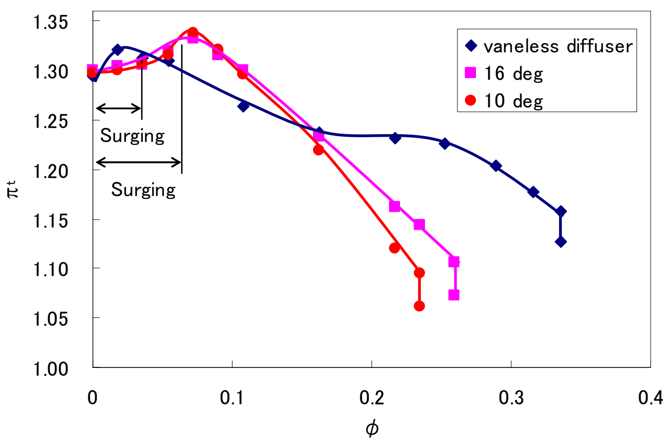


Fig. 7 Non dimensional total pressure ratio (250,000rpm)

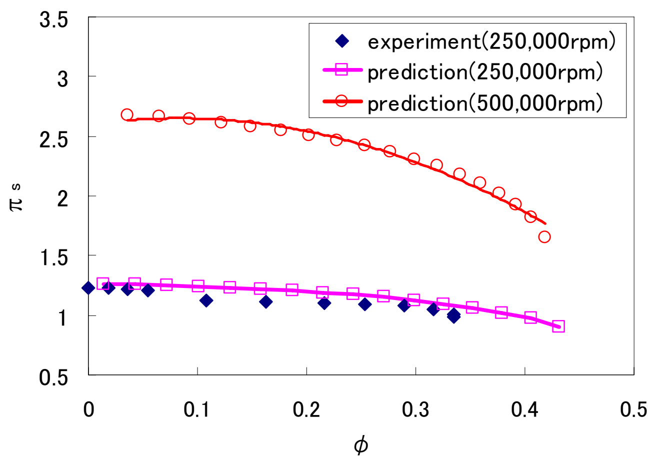


Fig. 8 Non-dimensional static pressure ratio at the impeller outlet (vaneless)

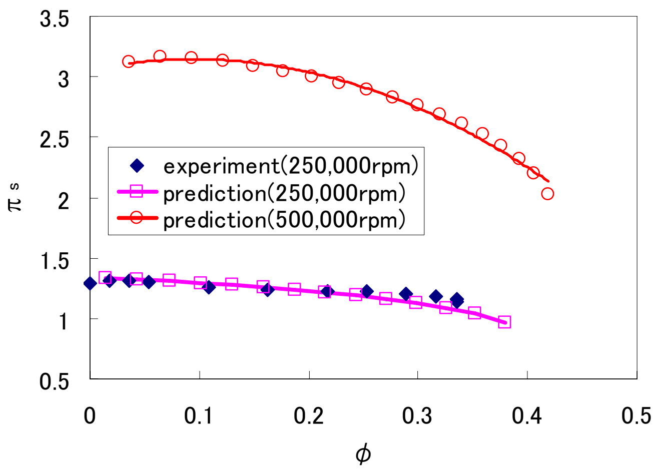


Fig. 9 Non dimensional static pressure ratio at the diffuser outlet (vaneless)

5. Conclusion

In order to establish the design methodology of an ultra-micro compressor, a 5 times size model with the 3-dimentional shape was designed and tested. The hot test rig with a combustor was employed instead of the cold air rig in order to attain the higher rotational speed. The vaned diffuser with the divergence angle of 16 deg., which was wider than the conventional one, showed the superior characteristic than that of 10 deg. for the present ultra micro compressor. In the next step of the present research, the test at rotational speed of 500,000rpm will be tried.

Acknowledgments

The authors would like to wish to express the sincere gratitude to Professor Shimpei Mizuki. We would also like to acknowledge the support of manufacturing the test rig with a combustor by Maruwa Electric Incorporated.

Nomenclature

D_2	Impeller outlet diameter [mm]	π_t	Total pressure ratio [-]
b_2	Impeller outlet blade height [mm]	π_s	Static pressure ratio [-]
n	Rotational speed [rpm]	η	Adiabatic efficiency [%]
ϕ	Flow ratio [-]	θ	Diffuser divergence angle [degree]

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