**STARTING CHARACTERISTIC ANALYSIS OF A RADIAL INFLOW TURBINE
FOR THE RE-GENERATIVE BRAYTON CYCLE****Susumu NAKANO*, Manabu YAGI*, Tadaharu KISHIBE*,
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ABSTRACT Microturbines have been developed as compact gas turbines to be applied in the re-generative Brayton cycle. A typical microturbine is composed of a centrifugal compressor and a radial inflow turbine. As such, the microturbine has a starting characteristic peculiar to radial inflow turbines. An idling state known as the windage point for mass flow rate can be formed because of improper inlet flow conditions for turbine expansion flow. The present study looked at the relationships between the radius ratio of the radial inflow turbine to the centrifugal compressor and the starting characteristic, and at the effects of turbine inlet flow conditions on the starting characteristic. Fundamental equations for the relationships between the radius ratio and the starting characteristic were obtained. Effectiveness of the equations was compared with experiment results obtained with a 150 kW class prototype microturbine.

Keywords: Micro Gas Turbine, Surging, Compressor, Re-Generative Brayton Cycle, Radial Inflow Turbine, Starting Pressure

1. INTRODUCTION

Microturbines are small capacity gas turbines to be applied in the re-generative Brayton cycle. Several types have been developed since the end of the 1990s in line with the expectation in the USA that distributed power generation would become a large market with deregulation [1], [2], [3]. A typical microturbine is composed of a centrifugal compressor and a radial inflow turbine [4]. Microturbines have a starting characteristic peculiar to radial inflow turbines. An idling state known as the windage point for mass flow rate can be formed because of improper inlet flow conditions for turbine expansion flow. Though characteristics of the windage point for turbo-chargers have been investigated [5], these results are not directly transferrable to microturbines because the latter have a pressure rise in the engine in addition to a compressor discharge pressure rise. In microturbines, starting pressure is strongly affected by the radius ratio between the turbine rotor to the compressor impeller because the pressure rise caused by centrifugal force acting on the turbine rotor depends on the radius of the turbine rotor and the inlet pressure of the turbine depends on the radius of the compressor impeller. As an idling state of the turbine rotor causes compressor surge and has a malign influence upon combustor ignition, the idling state is a very important matter for gas turbine systems applying radial inflow turbines. The starting characteristic of radial inflow turbines, especially the relationship for the radius ratio between the turbine rotor to the compressor impeller, has not been reported so far.

The present study looked at the relationships between the radius ratio of the radial inflow turbine to the centrifugal compressor for the starting characteristic, and at the effects of turbine inlet flow conditions on the starting

characteristic. Fundamental equations for starting pressure were obtained. Effectiveness of the equations was compared with experimental results obtained with a 150 kW class prototype microturbine [6].

2. STARTING PRESSURE**2.1 Outline of the Turbine System**

The cycle that is applied to the microturbine is the re-generative Brayton cycle. Figure 1 is a system diagram of the prototype microturbine. The main components are a single-stage centrifugal compressor, a single-stage radial inflow turbine, a low NO_x combustor, a recuperator, a permanent magnet generator, and an electrical power conversion system. Ambient air is taken from outside the system. It is compressed adiabatically by the single-stage centrifugal compressor and sent to the recuperator. The recuperator increases the air temperature by transferring heat of the turbine exhaust gas to the compressed air. The high temperature pressurized air enters the combustor and is mixed with injected fuel gas; the temperature of the air is high enough to provide fuel savings in the combustor. The combustion gas is expanded in the radial turbine and it drives the rotor system which is composed of a common shaft for the generator rotor, centrifugal compressor and radial inflow turbine. The common shaft is supported by two water lubricated bearings [7]. Figure 2 shows the turbine rotor of the prototype microturbine. After expansion in the turbine rotor, combustion gas is exhausted through the recuperator to the outside.

2.2 The Radius Ratio of Turbine Rotor to Compressor Impeller and Starting Pressure Ratio

As mentioned in the introduction, an idling state of

the turbine rotor can occur when the inlet flow conditions of the turbine are improper for formations of expansion flow from the turbine inlet to the exit. The starting pressure ratio is defined as the pressure ratio between the turbine inlet pressure and exit pressure when the expansion flow just begins to start from the idling state, namely the point when the expansion flow just begins to start means the windage point for the mass flow rate. Turbine action depends on the enthalpy difference between the turbine inlet and the exit, and the inlet condition of the turbine before the combustion ignition depends on the exit condition of the centrifugal compressor. Though one of the effective ways to increase turbine power output is to increase the turbine radius, the turbine radius can not be increased by simply neglecting the compressor radius. This is because the expansion flow from the turbine inlet to the exit will be blocked by centrifugal force acting on the turbine rotor which increases with a turbine radius increase. So the radius ratio of the turbine rotor to the compressor impeller strongly affects the starting pressure ratio of the turbine. The radius ratio of the turbine rotor to the compressor impeller for existing turbine systems [4], [8] has rarely been beyond 1.2 except for some early developmental microturbines like in references [2] and [9], but neither have been reported to work as turbines yet.

In the present study, upstream pressure of the turbine is defined as the inlet total pressure of the turbine nozzle because the turbine nozzle is placed upstream from the turbine rotor.

Fig. 1 Microturbine system diagram

Fig.2 Photograph of the turbine rotor

3. CALCULATION METHODS

3.1 Starting Pressure Ratio for Radial Inflow Turbine

First, fundamental equations for the relationships between the radius ratio and the starting pressure are shown. Since the discussion for the starting pressure is focused on

states before combustion ignition, air is treated as the object fluid. Expansion work of the turbine for unit mass flow rate is expressed in Eq. (1), the consumption energy of pressurized air by centrifugal force is similarly given in Eq. (2).

$$L_T = \eta_T C_p (T_{0in,N} - T_{0ex,T}) \quad (1)$$

$$L_C = \eta_C \frac{\omega^2}{2} (r_{in,T}^2 - r_{ex,T}^2) \quad (2)$$

Starting pressure ratio between the total pressure of the turbine nozzle inlet to that of the turbine exit can be recognized as the pressure ratio when expansion work for the turbine is balanced by the consumption energy of pressurized air by centrifugal force. The relationship for the starting pressure ratio shown in Eq. (3) can be deduced from Eq. (1) = Eq. (2) when the flow in the turbine rotor can be assumed as adiabatic change.

$$\frac{P_{0in,N}}{P_{0ex,T}} = \left[1 - \frac{C_\eta \omega^2}{2C_p T_{0in,N}} (\alpha r_{ex,C})^2 \left(1 - \left(\frac{r_{ex,T}}{r_{in,T}} \right)^2 \right) \right]^{\frac{\kappa}{1-\kappa}} \quad (3)$$

When the pressure loss between the exit of the centrifugal compressor and the exhaust exit of the turbine system and the heat loss between the exit of the centrifugal compressor and the inlet of the turbine nozzle can be neglected, the total pressure ratio of the compressor exit to inlet can be assumed to equal that of the turbine nozzle inlet to turbine exit, and the total temperature of the turbine nozzle inlet in Eq. (3) can be transformed into the total temperature of the compressor inlet with an adiabatic relationship using the total pressure ratio of the turbine nozzle inlet to turbine exit. Eq. (3) is rewritten as Eq. (4)

$$\frac{P_{0in,N}}{P_{0ex,T}} = \left[1 + \frac{C_\eta \omega^2}{2C_p T_{0in,C}} (\alpha r_{ex,C})^2 \left(1 - \left(\frac{r_{ex,T}}{r_{in,T}} \right)^2 \right) \right]^{\frac{\kappa}{1-\kappa}} \quad (4)$$

Equation (4) gives the starting pressure ratio for the turbine using a radius ratio of the turbine inlet and exit, a radius ratio of the turbine rotor and compressor impeller, compressor inlet temperature, and turbine angular velocity.

On the other hand, when pressure loss and heat loss are considered, the pressure ratio of the total pressure of the turbine nozzle inlet to atmospheric pressure can be expressed as follows.

$$\begin{aligned} \pi &= \frac{P_{0in,N}}{P_a} \\ &= \left[(1 + \theta_1)^{\frac{\kappa-1}{\kappa}} + \frac{C_\eta \omega^2}{2C_p \gamma T_{0in,C}} (\alpha r_{ex,C})^2 \left(1 - \left(\frac{r_{ex,T}}{r_{in,T}} \right)^2 \right) \right]^{\frac{\kappa}{\kappa-1}} \end{aligned} \quad (5)$$

In Eq. (5), heat loss is considered as the temperature drop

between the compressor exit and the turbine nozzle inlet, and the coefficient γ expressed by Eq. (6) is adopted.

$$\gamma = \left(1 - \frac{\Delta T_2}{T_{0ex,C}}\right) \left(1 + \frac{\Delta P_2}{P_{0in,N}}\right)^{\frac{\kappa-1}{\kappa}} \quad (6)$$

When the summation of the starting pressure and the pressure loss between the compressor exit and turbine exhaust exit is beyond the surge pressure of the compressor, the turbine system can not be realized. So the condition written by Eq. (7) needs to be obtained.

$$\sigma \geq \pi + \theta_2 \quad (7)$$

3.2 Numerical Analysis Method

To observe the flow pattern in the turbine rotor around the conditions of turbine starting point, 3D viscous flow analyses were performed. The numerical analysis method was a steady flow analysis using Dawes code [10]. The turbulence model was the standard k- ϵ model [11]. The computational region is shown in Fig. 3. It is composed of one pitch region for both the turbine nozzle and turbine rotor and it includes each blade profile. Inlet total pressure, inlet total temperature, and exit static pressure are given as boundary conditions. Numbers of mesh division are as follows: in the blade height direction, 22; in the circumference direction, 22; and in the flow direction, 90.

Fig. 3 Calculation region

4. EXPERIMENTAL APPARATUS AND METHOD

4.1 Isolated Turbine Test

An isolated turbine test was done to confirm turbine characteristics when disconnecting the turbine inlet air line from the compressor exit air line. The pressurized air from the compressor was discharged into the atmosphere, and turbine inlet air was supplied from another air supplier. Inlet air conditions were regulated as inlet pressure from atmospheric pressure to 0.5MPa and temperature from ambient temperature to 523 K.

First, the generator rotor was started and held at the rotation speed of the test. Regulated air was sent from the air supplier to the turbine inlet. Air flow rate was measured by an orifice located at the mid-point between the air supplier and the turbine inlet. Air temperature and pressure were measured at the inlet of the turbine nozzle.

To avoid supplying excessive air into the turbine, the maximum inlet pressure of the turbine nozzle was set as 1.01 times the surge pressure of the compressor.

4.2 Prototype Microturbine Test

A test using the packaged prototype turbine system was done to confirm turbine characteristics when connected to the compressor: in particular, flow rate change after firing was evaluated. Flow rate measurements of the prototype microturbine test were based on the differential pressure of an inlet air filter which had been calibrated in a standard wind tunnel.

5. CALCULATION AND EXPERIMENTAL RESULTS

5.1 Starting Pressure Ratio

Starting pressure ratio was calculated with reference to the dimensions for the centrifugal compressor and the radial inflow turbine of the prototype microturbine. The turbine inlet radius was 0.113 m, the turbine exit radius (i.e., the root mean square of the tip radius and hub radius of the turbine exit) was 0.042 m, and the compressor exit radius was 0.097 m. The compressor performance characteristics of the prototype microturbine are shown in Fig. 4. The surge line was calculated using the method of reference [12].

Fig. 4 Compressor performance characteristics

Fig. 5 Calculation results of starting pressure ratio

Fig. 6 Temperature effects on starting pressure ratio

Fig. 7 Margins of the starting pressure

Figure 5 plots results calculated by Eqs. (3) and (4) using $C_\eta=0.85$ which was obtained from the comparison with the following experimental results. Calculations were done to change α as a parameter. When the compressor inlet temperature is 288 K, the starting pressure lines for the radius ratio $\alpha > 1.3$ are higher than the surge line. This means that the turbine rotor cannot start. For the conditions of the radius ratio $\alpha < 1.2$, the turbine rotor can start throughout the whole range from the rotation start to the rated rotation speed of 51,000 rpm. On the other hand, for the turbine inlet temperature of 523 K, all lines are under the surge line. So air temperature rise caused by combustion ignition can induce easy starting of the turbine rotor. Though the effects of pressure loss and heat loss are not considered, the present results indicate the limitation for the radius ratio of the turbine rotor and the compressor impeller.

Effects of turbine inlet air temperature on starting pressure at the rotation speeds of 5,000 rpm and 10,000 rpm are shown in Fig. 6. For both rotation speeds, when the inlet air temperatures of the turbine increase, starting pressures decrease. Though significant effects for starting pressure reduction can be observed from room temperature to 500 K, few effects appear above 500 K.

Next, effects of pressure loss and heat loss are discussed from results of Eq. (5). Some values like γ , θ are needed for the calculations. Those were obtained from the prototype microturbine test. Pressure loss was approximated by quadratic functions of rotation speed. Figure 7 shows the margins for starting pressure. If the

margin is a negative number, the turbine rotor cannot start. For $\gamma=1.0$ there was no consideration of temperature drop as heat loss, while $\gamma \neq 1.0$ considered the temperature drop. For $\alpha=1.2$, results without consideration of temperature drop are positive at rotation speeds above 13,000 rpm, and they are around zero from 0 to 13,000 rpm. But results with consideration of temperature drop are negative in the whole region to 20,000 rpm. Then $\alpha=1.16$ is the value which is adopted in the prototype microturbine. Results with temperature drop are around zero at speeds under 10,000 rpm, and they become positive above that. This seems to be the limitation for the radius ratio of the turbine rotor to the compressor impeller of the present machine and moreover, results for $\alpha=1.3$ are clearly negative from the rotation speed of zero for both cases with and without consideration of temperature drop.

From those results, it is concluded that temperature drop from the compressor exit to the turbine nozzle inlet significantly affects starting pressure of the radial inflow turbine.

5.2 Results of Isolated Turbine Tests

Figure 8 shows the measured ratios of total pressure of the turbine nozzle inlet to atmospheric pressure while rotation speed of the generator rotor was held at 5,000 rpm and 10,000 rpm. Measurements were done for two inlet air temperatures of the turbine nozzle of 297 K and 523 K. Pressure ratios for the inlet air temperature of 297 K are higher than those of 523 K for both rotation speeds. It is clear that the inlet air temperature of the turbine nozzle decreases the pressure ratio and that agrees with predictions by Eq. (3). When the mass flow rate is constant, raising the inlet air temperature of the turbine nozzle can lower the pressure ratio. When the pressure ratio is constant, raising the inlet air temperature can increase the mass flow rate.

Intersecting points of the pressure ratio characteristic lines and the vertical axis indicate each starting pressure. The efficiency ratio C_η which is shown in Eqs. (3), (4) and (5) is unknown, and it is difficult to specify the value concretely. So it was specified from comparing the calculation results which were obtained by substituting some C_η s into Eq. (3) with the experimental values. Results calculated by Eq. (3) with $C_\eta=0.85$ were plotted in the figure. These points show good correspondence with the points extrapolated by the quadratic function from the experimental data for all four cases of experiments. Fundamental equations described before with $C_\eta=0.85$ can

Fig. 8 Experimental results of the isolated turbine test

be used as prediction formulas for starting pressure of radial inflow turbine.

The compressor performance characteristic lines for the rotation speeds of 5,000 rpm and 10,000 rpm are plotted in Fig. 8. In both cases for the inlet air temperature of 297 K, the crossing points of the compressor performance characteristic lines and pressure ratio lines are close to the surge point of the compressor. Raising the inlet air temperature of the turbine nozzle, will shift the matching point of the compressor and turbine into the large mass flow rate region which means a stable region for compressor operation without surge.

5.3 Results of Numerical Analysis

Flow conditions for the objects of numerical analysis were small flow rates. The flow rate of 5,000 rpm is about 5 % of the rated mass flow rate, and that of 10,000 rpm is 15 %. As it is difficult to discuss the results of the numerical analysis quantitatively, only the difference in flow patterns with inlet flow conditions was observed. Numerical analysis was done at 10,000 rpm for two inlet air temperatures of the turbine nozzle, 297 K and 523 K. Boundary conditions for pressure of both cases were as follows: total pressure of the turbine nozzle inlet, 108 k Pa; static pressure of the turbine exit, 100 k Pa.

Figures 9 (a) and (b) show relative velocity streamlines of the numerical analysis results. Though these figures are focused on the turbine rotor, the exit areas of the turbine nozzle are included. Three kinds of cross sections are shown; the hub surface, the blade tip surface, and the middle surface between them. The two figures have two definite features. One is the exit velocity of the turbine nozzle and the other is the flow pattern in the turbine inlet region. The exit velocity of the turbine nozzle at inlet air temperature of 523 K is larger than that of 297 K. The exit velocity of the turbine nozzle at the inlet air temperature of 523 K is about 90m/s, and that is about three times larger than of 297 K. Though the inlet pressures are the same in both cases, the heat drop at the turbine in the case of 523 K is 1.8 times larger than that of 297 K because of the enthalpy increase due to the higher inlet temperature. That accelerates flow at the exit of the turbine nozzle, and it is easy for the flow to get into the turbine rotor. In Fig. 9 (a), a large vortex flow region can be seen whose rotational direction is counter to the rotational direction of the turbine rotor, and which almost fills the inlet region of the turbine rotor. This vortex flow prevents turbine nozzle exit flow from entering the turbine rotor. In Fig. 9 (b), the definite vortex flow pattern in the inlet region of the turbine rotor disappears. Increase of the exit velocity of the turbine nozzle due to the higher inlet air temperature weakens the vortex generation.

Calculated mass flow rates are 0.135 kg/s for 297 K, and 0.172 kg/s for 523 K. Those values show good agreement with the experimental results.

5.4 Results of the Prototype Microturbine Test

Rotation speed of the prototype machine test was 4,000 rpm because there is a critical speed around 5,000 rpm where rotor vibration of the lower mode occurs. To pass through the critical speed smoothly, the ignition point for the combustor was chosen as 4,000 rpm. Flow condition changes after the ignition were observed in the prototype

Fig. 9 (a) Calculation results for turbine nozzle inlet air temperature of 297 K

Fig. 9 (b) Calculation results for turbine nozzle inlet air temperature of 523 K

Fig. 10 Experimental results of the prototype microturbine

microturbine test. Target inlet gas temperatures of the turbine nozzles were 523 K and 773 K. In Fig. 10, gas means combustion gas after the ignition and air before the ignition.

Figure 10 shows the changes of mass flow rate and

inlet air temperature of the turbine nozzle. Changes of mass flow rate are described as mass flow rate ratio based on the rate just before ignition at 4,000 rpm. Both results for inlet air temperatures of 523 K and 773 K show an increase of 1.5 times for the mass flow rate after the ignition. Results of the prototype test have similar trends to the results of the isolated turbine test. But the effect of air temperature after the ignition for mass flow rate change is small. From the calculation results shown in Fig. 6, the effect of inlet air temperature for the pressure ratio becomes small for above temperatures above 500 K.

6. CONCLUSIONS

The main findings of the starting characteristic analysis of a radial inflow turbine are listed below.

1. Fundamental equations for starting pressure of radial inflow turbine were deduced. They described the relationships between the radius ratio of the turbine and compressor, the radius ratio of the inlet and exit of the turbine rotor, and the inlet air temperature of the turbine nozzle or compressor.
2. The efficiency ratio, denoted as $C\eta$ in the fundamental equations, was chosen as 0.85 and there was good agreement with the experimental results when this value was used.
3. There was a limitation for the radius ratio of the turbine rotor to the compressor where stable operation without compressor surge could be obtained. For the prototype microturbine studied here, the radius ratio could not be larger than 1.3.
4. Pressure loss and heat loss between the compressor outlet and turbine inlet where a recuperator is placed significantly affected the turbine starting characteristic. In particular, the heat loss had a considerable effect to increase starting pressure.
5. In the small mass flow rate condition near the idling state, a large vortex flow pattern almost reaching from the hub to the tip in the inlet region of the turbine rotor was found from the numerical analysis. Though this vortex flow prevented the turbine nozzle exit flow from entering the turbine rotor, by raising the inlet air temperature of the turbine nozzle, the definite vortex flow pattern in the inlet region of the turbine rotor disappeared and the mass flow rate increased.
6. Regarding stable operation near the limitation for the radius ratio of the turbine rotor to the compressor, operation with ignition at a low rotation speed was proposed and the effectiveness was verified.

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NOMENCLATURE

C_p : specific heat at constant pressure (J/(kg K))
 C_η : efficiency ratio $C_\eta = \eta_c / \eta_T$ (-)
 N : rotation speed (rpm)
 P : pressure (Pa)
 ΔP : pressure loss (Pa)
 r : radius (m)
 T : temperature (K)

ΔT : temperature drop (K)

α : radius ratio between turbine rotor and compressor impeller. (-)

γ : coefficient of temperature drop in Eq. (6)

η : efficiency (-)

θ : ratio of pressure loss and atmospheric pressure (-)

κ : specific heat ratio (-)

π : pressure ratio of turbine inlet total pressure and atmospheric pressure (-)

σ : pressure ratio of surge pressure and atmospheric pressure (-)

ω : angular velocity (rad/s)

Subscripts

a: atmosphere

C: compressor

ex: exit

in: inlet

N: nozzle

T: turbine

0: total

1: region from turbine exit to exhaust

2: region from compressor exit to turbine nozzle inlet

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