Experimental Investigation of A Twin Shaft Micro Gas-Turbine System

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Abstract. Due to the fast depletion of fossil fuels and its negative impact on the environment, more attention has been concentrated to find new resources, policies and technologies, which meet the global needs with regard to fuel sustainability and emissions. In this paper, as a step to study the effect of burning low calorific value fuels on gas-turbine performance; a 50 kW slightly pressurized non-premixed tubular combustor along with turbocharger based twin shaft micro gas-turbine was designed and fabricated. A series of tests were conducted to characterize the system using LPG fuel. The tests include the analysis of the temperature profile, pressure and combustor efficiency as well as air fuel ratio and speed of the second turbine. The tests showed a stable operation with acceptable efficiency, air fuel ratio, and temperature gradient for the single and twin shaft turbines.

1. Introduction

Gas turbines, which used in power generation, are designed mainly to run on natural gas and kerosene. An attractive feature of gas turbines that it is capable to run with various types of fuels [1]. The main differences in micro gas turbine as compared to industrial size gas turbines, include the way that wasted heat is recuperated, efficiency and capability to run in combined cycles [2]. Due to the challenges which face using fossil fuels, a special attention is required on biomass combined gas turbines.

A pressurized premixed cyclone combustor drive two shaft micro gas turbine have been investigated by Al-ttab and Zainal [3]. CFD modeling was carried out, but it was limited to the design and optimization of a cyclonic combustor. Jaafar et al. [4] designed and fabricated a gas-turbine liquid fuel burner with different radial air swirler angles; the investigation was concerned with the effect of the swirler size on exhaust gas emission. The thermo chemical power group in University of Genoa developed a new gas-turbine laboratory; a general purpose experimental rig was installed based on a modified 100 kW micro gas turbine [5].

This paper discusses the preliminary experimental results for a low-cost twin shaft gas turbine developed for use with syngas. The tests were conducted using LPG fuel as a benchmark for studying the effect of burning low calorific value gaseous fuel similar to syngas on the gas-turbine performance.

2. Design Approach

The gas-turbines consist of a compressor, combustor, and turbine. The heat input for the gas-turbine is provided by combustors. Gas turbine combustor design falls into three categories; i.e. tubular, annular, and tubo-annular. The amount of heat added to the combustor can be quantified by [6]:

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where \( \dot{m}_{\text{fuel}} \) and \( CV_{\text{fuel}} \) are the mass flow rate and the calorific value of the combusted fuel respectively. By knowing the inlet condition, the combustor could be sized. The parameter which facilitate combustion chamber sizing is the reference velocity, \( U_{\text{ref}} \), which is the mean velocity at the maximum cross sectional area of the casing in the absence of the liner\(^7\):

\[
U_{\text{ref}} = \frac{\dot{m}}{\rho A_{\text{ref}}}
\]

where \( \dot{m} \) is the air mass flow rate, \( \rho \) is the air density and \( A_{\text{ref}} \) is the cross sectional area of the casing. The relationship between the size and pressure loss in the combustion chamber for the optimal cross-sectional area of the casing is given by \([4, 7]\):

\[
A_{\text{ref}} = \left[ \frac{\Delta P}{\dot{m}^3 \dot{q}_{\text{ref}}} \right]^{\frac{1}{3}}
\]

where \( \Delta P \) is the pressure loss factor, and \( \frac{\Delta P}{\dot{m}^3 \dot{q}_{\text{ref}}} \) is the total pressure drop across the combustor. Equation (3) can be used to calculate the reference area for the straight combustors. Table 1 shows some typical values of pressure loss in practical chambers.

| Type of chamber | \( \frac{\Delta P}{P} \) | \( \frac{\Delta P}{\dot{m}^3 \dot{q}_{\text{ref}}} \) | \( \frac{\dot{m}^3 \dot{q}_{\text{ref}}}{A_{\text{ref}} P} \) |
|-----------------|-----------------|-----------------|-----------------|
| Tubular         | 0.07            | 37              | 0.0036          |
| Tubo-annular    | 0.06            | 28              | 0.0039          |
| Annular         | 0.06            | 20              | 0.0046          |

For tubular combustors, the ratio of length to the diameter should be small to reduce the formation of NOx emissions and reduce the weight of the combustor. Furthermore, the increase of liner area results in lower velocities and higher residence time which are good for combustion characteristics.

3. System Descriptions

The small size gas turbine system consists of the following parts:

3.1. Combustion Chamber

The combustion chamber is a tubular combustor with thermal capacity of 50 kW. The consumption rates of LPG fuel and air were calculated using Equation (1). For simplicity and ease of fabrication, a combustor length to diameter ratio of 2 was chosen. In order to reduce the air pressure and increase the flue gas velocity at the combustor exit, the combustor was joined to the turbine by a nozzle. The dimension of the combustor is shown in Table. 2.

| Material  | Wall thickness (mm) | Diameter (mm) | Liner/Annulus Ratio | Length/Diameter Ratio | Liner Holes (mm) | Nozzle Height (mm) | LPG Injection Diameter (mm) |
|-----------|--------------------|---------------|---------------------|----------------------|-----------------|-------------------|-----------------------------|
| S. Steel  | 4                  | 260           | 0.7                 | 2                    | 10              | 100               | 2                           |

3.2. Turbochargers and Air fuel system

Two turbochargers were selected for this work. A Nissan H 25 turbocharger was selected and mounted as the high pressure turbine based on its air flow rate and pressure ranges required in this work. For the second turbine where a high torque is needed, the Jasma T70 turbocharger was used. Since both turbochargers rotate with a high rotational speed, a vehicular oil pump driven by one-horse power electric motor was used to provide the turbocharger with the needed lubrication.

The fuel system allows for non-premixing LPG charge, the fuel supplied from LPG cylinders through a pressure regulator and control valve and then injected into the combustor via eight holes distributed in 10 mm diameter stainless steel circular pipe acts as a flame holder. The starter system
consisted of a V shape tube connected directly with the compressor inlet; one of the two ends connected to the 1½ hp blower which provided the system with air at the starting stage, the other end was opened to atmosphere.

**Figure 1.** Schematic of the experimental rig
(1) Combustor (2) High pressure turbine (3) Low pressure turbine (4) Starting blower (5) Oil pump motor (6) Starting air entrance (7) Pressure gauge (8) Pitot tube (9) LPG rotameter (10) LPG cylinder (11) LPG regulator (12) Entrance air valve (13) Oil pump (14-19) thermocouples (20) Cooling water line (21) Flame holder (22) Ignition plug

4. Test Method

Figure 1 shows a simplified over view of the experimental setup used. Six type K thermocouples distributed in equally axial distance were attached in the combustor to monitor the temperature inside the combustor; the thermocouples were connected with Pico USB TC-08 thermocouple data logger with 0.5°C accuracy to reflect the system temperature. The flow rate of the fuel and the air was measured using 150 mm Concoa flow meter and calibrated Pitot tube respectively. Lutron DT-2259 optical tachometer with an accuracy of 0.1 rpm was used to measure the turbine rotational speed.

In the starter system, LPG and air were injected into the combustor; then the ignition was initiated using a spark plug. For warming up purpose, the system was allowed to run for about 5-10 minutes before the fuel flow rate increasing.

5. Results and Discussions

The single shaft turbine was successfully run with LPG fuel. The temperature and turbine speed increased gradually with increase fuel flow rate. When the temperature in the primary combustion zone reached 1050°C the system started to run, and the compressor could provide the combustor with the needed air. At this stage, the starting blower was stopped.

Shown in Figure 2 is the variation of combustor efficiency at different heat input. The use of the flame tube to control the amount of the combustion and dilution air, affected in improving the combustor efficiency and lead to an excellent temperature gradient inside the combustor. The main problem in this case was the high operation temperature. On the other hand, as shown in Figure 5, removal of the flame tube increase of the amount of admitted air and hence lowered the operation temperature and affected the unsteady combustor temperature gradient, especially in the primary combustion zone which is exposed to the cold air coming from the compressor.

Shown in Figure 3 is the variation of the turbine entrance temperature with heat input. Due to the low amounts of admitted air into the combustor for the flame tube tests, the turbine entrance temperature was relatively higher compared to that one with no flame tube.

**Figure 2.** Single turbine combustor efficiency for different heat input

**Figure 3.** Single turbine inlet gas temperature for different heat input

Generally, as the amount of the burned fuel increased the rotational speeds increased and hence increase the air intake rate into the combustor. The combustor efficiency calculated based on the ratio of the flue gases' thermal power out from the combustor to the combustor thermal power input. As shown in Figures 4 and 5, the tests showed an acceptable air fuel ratio. The excess air range was 120% - 221% which is acceptable for gas turbines.
For the twin shaft turbine, the compressor could suck the air from atmosphere and inject it into the combustor. For safety reasons, the air was allowed to be sucked through a half inch hole located at the suction line to prevent the high air admission which may affect in flame quenching.

Shown in Figure 6 is the variation of the temperature and excess air with the total flue gases. At the start, the fuel-burning process inside the combustion chamber was at a low air fuel ratio. The first turbine speed was strongly affected by the back pressure of the second turbine. The air-fuel ratio increased gradually by the increase of the first turbine speed, the excess air range was 45%-217%. As shown in Figure 7, the second turbine speed and combustor efficiency was increased gradually with the fuel increasing. The twin shafts turbine tests showed lower combustor efficiency compared to that one of the single turbines, this is because of the high energy lost by the combuster surface.

6. Conclusions
A 260 mm diameter tubular combustor along with low cost twin shaft micro gas turbine system was designed and fabricated. The system has been characterized with LPG fuel. The test showed a successfully running with an acceptable air fuel ratio, temperature gradient, and efficiency; the combustor efficiency range was 34%-87% while the turbine entrance temperature was 550°C–975°C. Due to high heat loss by the combustor surface, the two shaft turbine tests showed a lower combustor efficiency compared with the single turbine.

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