Performance modelling of a micro gas turbine at part-load operation

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Abstract:

The integration of absorption chillers in (micro trigeneration) systems based on micro gas turbines is a viable option to increase efficiency. As there are many equipment in a micro trigeneration system that are to be characterized with modelling for computational tools, it is not a simple procedure to optimize it for visibility study and predicted performance. This presentation is a study to analyse the accuracy of Malinowski's analytical model to predict the behaviour of a micro turbine operating at part load, at nominal and/or non-nominal conditions. The model generates turbine and compressor pressure ratios with performance equations, which are later with experimental data. It is developed as a computational tool to simulate the micro turbine at different operating conditions.

It consist an analytical model to simulate the micro gas turbine in a real cycle. It is used to compare model result with experimental data, obtained in CREVER Bench test, at URV. It is first verified that the model generated the same results developed by the author (not presented here). The model compared to different experimental data obtained in the CREVER bench test agree well. It gave good results and can be effectively used to compose the micro trigeneration model under development.

Keywords:

Micro gas turbine, Part-Load Operation, Performance Modelling, Analytical Model.

1 Introduction

Exponential increase in energy demand for industrial buildings worldwide has led to a similar demand for sustainable energy technologies [1], where the approach to the use of balance thermal heat energy resulting from industrial processes, is used to develop sustainable low temperature energy needs of the same building. An important and clear example of this type of technology is micro systems co / trigeneration, where the heat from exhaust gases after the generation of electricity is further used for comfort air conditioning to produce heat or cold. RD661/25 May2007 is the regulation in Spain for the activity of production of electricity and RD1699/2011 regulation covers the connecting conditions in 50kW range power micro trigeneration.

Micro trigeneration systems with gas turbines are considered thermally efficient due to its low thermal inertia[2], allowing reach of full power in a short time, making it ideal for applications to meet the sudden load requirements in industrial buildings. However, when processes have load variations, it decreases overall system efficiency caused by variations of compression, pressure ratio proportional to the load and electrical performance [3], as given in following literature using map performance components. Okelah [4] studied the performance of a fixed geometry turbo shaft engine co-using component maps. Al Hamdan and Ebaid [5]. carried out the modelling and simulation of a gas turbine engine ideal single axis. Aklilu et al. [6], developed a mathematical model to simulate a part-load operation of a single shaft gas compressor turbine with variable geometry, while Zhang and Cai [7] studied the performance of such generalized components. All these data are useful due to unavailable design information of part load characteristics from the manufacturers of the micro turbine. With the available literature parameters and models Malinowski and Lewandowska [8] developed an analytical model applicable to a typical micro turbine one axis, comprising of a compressor, turbine, combustion chamber and a heat recovery unit which responds to a regenerative Brayton cycle.

This article aims to verify the validity of the analytical model developed by these authors [8], comparing with experimental data taken at CREVER-URV group for a Capstone C30 micro turbine. The validated model is able to predict the behaviour of the micro turbine for the partial load energy operations, attaching great importance to the variation in shaft rotation speed of the turbine. The model at part-load performance heuristic proposal is based on reference [7].

It decreases the calculation error with the author's assumption that the processes in the turbine and compressor are isentropic and are calculated independent of specific heat variation in each component depending on the fluid temperature.

2 Micro turbine Thermodynamic Analytical Model

2.1 Turbine Characteristic Curves

This article uses an analytical model as proposed in Refs. [8, 9, 3], to calculate the performance of the main components of a micro gas turbine [10] when part load characteristics data are not available from the manufacturer for different ambient conditions. A dimensionless equation that calculates the pressure ratios and efficiencies of the compressor and turbine design parameters are functions of the relative turbine shaft rotational speed, and the relative mass flow, as presented in the equations (1), (2), and (3), where $\tilde{n}_t$ is the relative turbine shaft rotational speed, defined as:

$$\tilde{n}_t = \frac{\bar{n}_t}{\bar{n}_{t0}} = \left(\frac{n}{n_0}\right) . \sqrt{\frac{T_{03}}{T_3}} \quad (1)$$

$\tilde{m}_c$ and $\tilde{m}_t$ are the relative mass flow, defined as:

$$\tilde{m}_c = \frac{\bar{m}_c}{\bar{m}_{c0}} \quad (2)$$

$$\tilde{m}_t = \frac{\bar{m}_t}{\bar{m}_{t0}} \quad (3)$$

To model the compressor in dimensionless equations to predict the pressure ratio and efficiency, similar analogues equations (4) and (5) are used as calculated in the turbine performance.

$$\tilde{\sigma} = C_1 . \tilde{m}_c^2 + C_2 . \tilde{m}_C + C_3 \quad (4)$$

$$\tilde{\eta} = [1 - C_4(1 - \tilde{n}_c)^2] \left(\frac{\tilde{n}_c}{\tilde{m}_C}\right) \left[2 - \left(\frac{\tilde{n}_c}{\tilde{m}_C}\right)\right] \quad (5)$$

where C_1, C_2, C_3, C_4 and t_4, are parameters, as described in Refs. [8, 9, 3]; they are functions of the relative rotational speed, relatives mass flow, and of experimental data.

In addition to the parameters mentioned above, there are two other parameters, namely p and m, that are used to fit the equations, for predicting the full load performance, manufacturer's standard performance based on experimental data and other operating conditions like partial load turbine operation and effect of ambient varying conditions.

Using the equations of the analytic model, with the modified Flügel formula, and second law of thermodynamics, the parameters of the characteristics curves of the compressor and turbine, namely, the pressure ratios and efficiencies were calculated. This paper used the Malinowski's model to compare performance of a single shaft micro turbine Capstone C30 of the CREVER laboratory; The design point parameters are used to parametrize the characteristics curves as inlet and outlet temperatures, isentropic efficiencies, etc. [8].

Pressure drop calculation:

The model calculates the pressure drop along the cycle, using equation (6):

$$P_j = (1 - k).P_i \tag{6}$$

Where $\mathbf{P_i}$ is the pressure at the inlet to that element, and $\mathbf{P_j}$ is that of the outlet of this element and k is the relative pressure loss that can be calculated using equation (7):

$$k = k_0 \left(\frac{m}{m_0}\right)^2 . \left(\frac{P_{i0}}{P_i}\right)^2 . \left(\frac{T_i}{T_{i_0}}\right) \tag{7}$$

The measure of the overall system pressure loss is the coefficient:

$$\phi = (1 - k_{in}).(1 - k_{out}).(1 - k_{cc}).(1 - k_{rl}).(1 - k_{rh}) \tag{8}$$

Where the following values of k, were used: kin0=0,01; kout0=0,01; kcc0=0,02; krl0=0,02; krh0=0,02.

2.2 Micro gas turbine Modelling

The gas turbine used in this work, has a heat exchanger regenerator inside, as shown in Figure (1) schematic model). It also shows a control volume (cv) involving the turbine. The combustion air is first filtered and used to cool the electric generator before it enters the compressor (1). In the compressor, it is pressurised and flows into the regenerator to preheat, with heat recovery from the exhausted gases passing through the other side of the heat exchanger (2). After this, the preheated air (2r) is mixed with fuel (3) in the combustion chamber and is burnt. The hot pressurized exhaust gas converts its energy in an expansion process through the turbine (4) that drives the compressor and the electric generator; then the exhaust gases enter the recuperator (4r) and are finally vented out with potential waste heat. The modelling procedure is based on some isentropic efficiencies for compressor and turbine.

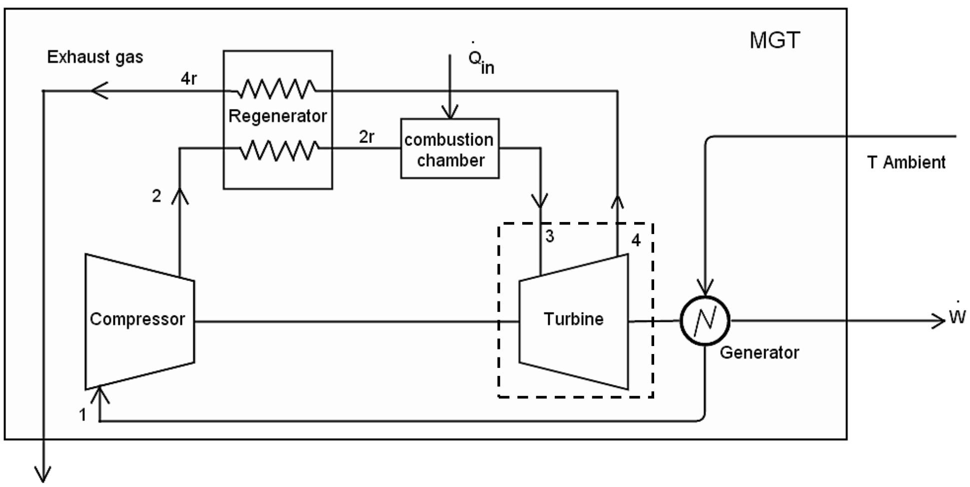

Figure 1. Schematic model of micro gas turbine.

Basic assumptions in turbine modelling summary:

- The turbine operates at a constant controlled exit temperature. .
- The air and combustion gases are ideal gases.
- The specific heat is variable and is used as in [18].

Energy Balance

Observing the vc involving the turbine in figure (1), and considering the process adiabatic, the first law becomes:

$$W_t = m_{gas}(h_3 - h_4) \tag{9}$$

Entropy Balance

Considering an isentropic process, and Cp variable, the second law becomes:

$$\int_{T_3}^{T_4} \frac{C_p(T)}{T}.dT - R.\ln\left(\frac{1}{\sigma_t}\right) = 0 \tag{10}$$

Isentropic Efficiency

In addition, as the real expansion process deviates from the isentropic it is necessary to introduce isentropic efficiency equation. The value of isentropic efficiency is calculated in the analytical model.

$$\eta = \frac{h_2 - h_1}{h_2 - h_1} \tag{11}$$

Pressure Drop

In order to calculate the pressure condition outside the turbine it is necessary to have the turbine pressure relation σ_t, calculated in the analytical model. The parameter K_t represents the relative percentage of pressure drop in the turbine. In addition, the calculation of the pressure outside the compressor also uses such pressure relation value that comes from similar analytical model.

$$P_4 = P_3\,\sigma_t(1 - K_t) \tag{12}$$

The equations generated constitute a linear system and when it is solved gives the values of the energy power generated in fuel burner, the work of the compressor and of the turbine, the electricity generated, and the heat rejected in the exhausted gases.

Mechanic work, Electric Energy and the Rejected Heat

$$W_m = (W_t\,.\eta_{tm} - W_c/\eta_{cm}) \tag{13}$$

$$E_e = \eta_g.\,\eta_e.W_m \tag{14}$$

$$Q_L = Q_H - Q_R \tag{15}$$

This model was programmed in EES - Engineering Equation Solver and the results obtained for the micro turbine C30 agreed with those presented in Ref [8].

3 Testing micro turbine in a test bench

The performance of the micro gas turbine was studied in the existing multifunctional test bench of CREVER-URV, as described in [10]. Several trials were conducted to study the performance at a fixed power rate and at part load conditions. In the experiments, the temperatures at the outlet of the compressor and the turbine together with the shaft rotational speed were measured and registered by a data logger.

Figure 2 shows the variation of the temperatures at the exit of the compressor and turbine with the inlet ambient air temperature when the micro turbine was operated at 23 kW. As shown in Figure 2, the temperature at the compressor outlet increases with the ambient air temperature. In addition the temperature at turbine outlet decreases while the shaft rotational speed increases with 2 degree ambient rise (from 1561 rps at 290.2 K up to 1598 rps at 292.2 K).

The micro turbine was also tested at reduced partial load from 25 kW to 4 kW at intervals of 2 kW, varying the inlet air temperature from 290.2 K to 294.2. The results have been represented graphically in Figure 3, with the values obtained for the shaft rotating speed against the power load.

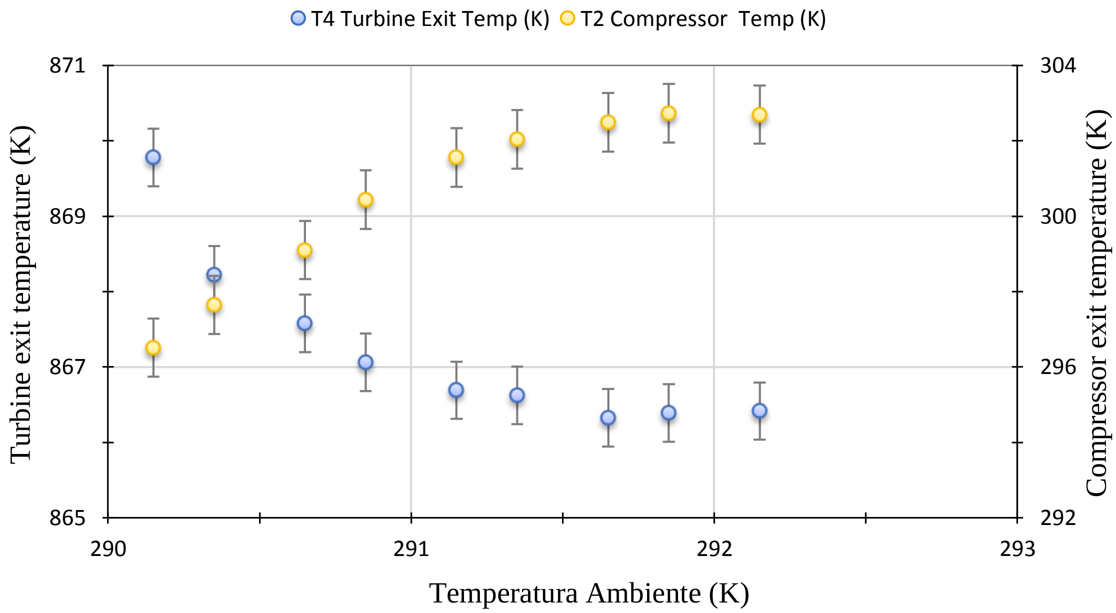

Figure 2, Effect of inlet air temperature on the outlet temperature of compressor (T2) and turbine (T4) at power load of 23 kW.

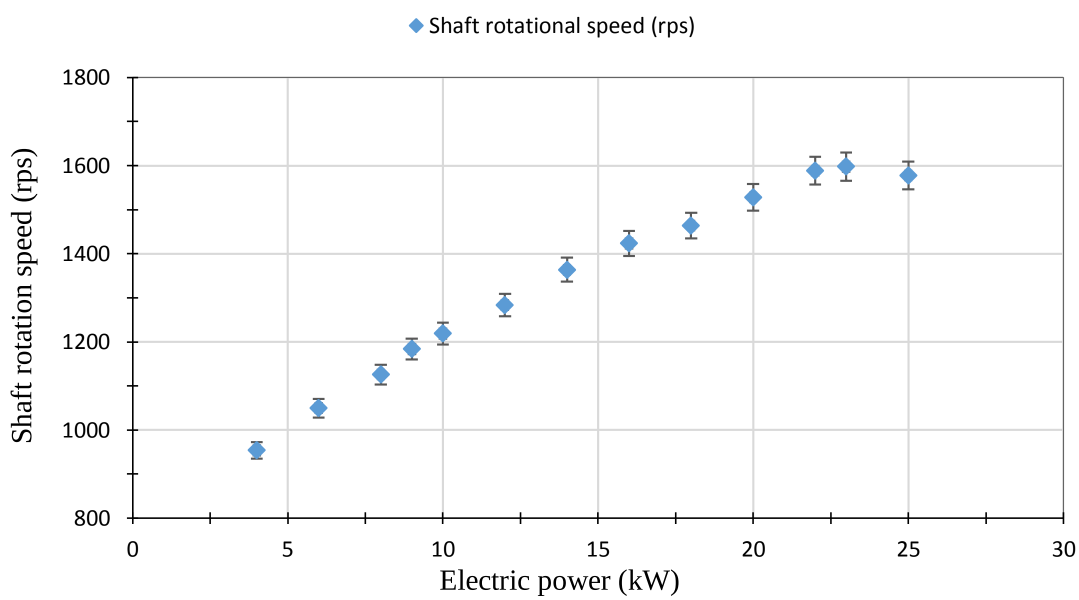

Figure 3. Effect of the variation of air temperature and load on the shaft rotational speed in inlet air temperature rate (290 K- 294K)

The experimental data obtained in the tests described in the previous section are compared to verify the accuracy of the proposed model. Figure 4 presents a comparison between the experimental and calculated data of the shaft rotation speed, at different inlet air temperatures, to a fixed electrical load of 23 kW. As predicted, the experimental and calculated data are in good agreement, with deviation of temperatures of compressor and turbine at exit within the range of 0.5 to 1.5 K.

Compared part load conditions expressed as shaft rotational speed data (for both calculated and experimental alternatives) show the deviation is lower than 0.15% (see Figure 3), while the ambient air temperature is changed within a range of (290 K- 294 K), and their corresponding values for the exit temperature of the compressor and turbine, present deviation lower than 2 K.

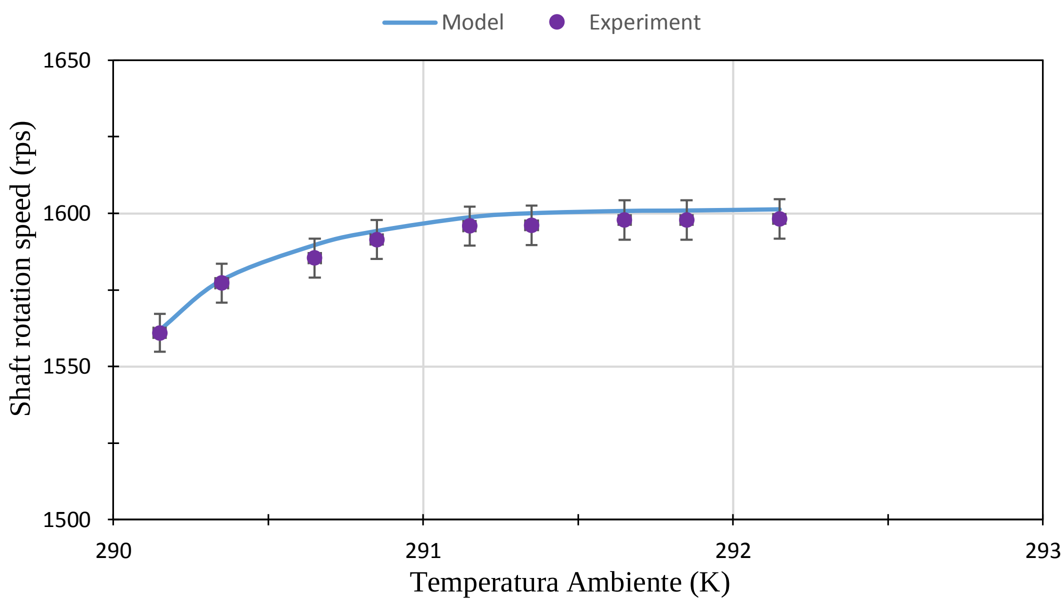

Figure 4. Model comparison test 1- effect of the variation of ambient temperature and load on the shaft rotation speed.

4 Performance modelling

The model can describe the full expected behavior of micro turbine for full rated load, part load, at nominal rated and/or outside nominal conditions. Table 2, shows the performance of the model at rate load (100%), 75% and 50% of load; electrical power generated, the flow of exhaust gas, and its outlet temperature. These values were calculated at inlet air temperature of micro turbine 298.15 K.

Table 2. Thermal performance of the turbine outside nominal conditions

Ambient Pressure (kPa)			110.13
Ambient Temperature (K)			298. 15
Part Load	100%	75%	50%
Electric Power (kW)	27.16	20.55	13.7
Mass flow rate (m/s)	0.2864	0.2798	0.2394
Exhaust gases temperature (K)	479.1	438.5	397.1

5 Conclusions

The model described, has been applied to calculate the Capstone C30 micro turbine performance characteristics. The results of the tests performed on our test bench to study the influence of inlet air temperature and load, and the results obtained with the model have good agreement. It is seen that the outlet temperatures of the turbine and the compressor have a maximum deviation of 2 degrees, while the rotational frequency does not deviate more than 0.15% when comparing the data with the experimental model.

Finally, after the verification of the model with experimental data it is concluded, that it can predict the micro turbine pressure ratios, efficiencies, and thermodynamic properties, of the points of the Brayton cycle, the heat and work flow rates through the process equipment during the operation of the micro turbine.

The model is able to predict, analyze, and make inferences about the turbine behavior at different points of operation, for varying parameters of the cycle like pressure ratio, efficiency, temperature of the inlet air variation and the energy produced. So, as it gave good results it can be effectively used to compose the micro trigeneration model under development.

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Nomenclature

T	temperature, K
P	pressure, kPa
Q	heat rate, W
R	gas constant, kJ/(kg K)
w	specific work, kJ/kg
W	power, W
Cp	constant-pressure specific heat, kJ/(kg K)
m	mass flow rate through the compressor and turbine, kg/s
h	specific enthalpy, kJ/kg
LHV	lower heating value, kJ/kg
n	shaft rotational speed, rps
k	relative pressure loss

Greek symbols

s	specific entropy, kJ/kgk
ε	heat recovery efficiency

η	Efficiency
ϕ	total relative pressure loss
σ	pressure ratio

Subscripts and superscripts

atm	Atmospheric
c	Compressor
cc	combustion chamber
e	Electric
f	Fuel
g	generator
in	Inlet
out	Outlet
r	heat recovery
rh	heat recovery, high pressure side
rl	heat recovery, low pressure side
s	Isentropic
t	Turbine
0	design value
~	divided by design value, $\tilde{\mathrm{m}}_{\mathrm{c}} = \bar{\mathrm{m}}_{c}/\bar{\mathrm{m}}_{c_0}$

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