Journal of Energy and Power Engineering 9 (2015) 326-335
doi: 10.17265/1934-8975/2015.04.002

Experimental Tests of the Operating Conditions of a Micro Gas Turbine Device

Roberto Capata
Department of Mechanical and Aerospace Engineering, University of Roma "Sapienza", Roma 00184, Italy

Received: November 26, 2014 / Accepted: January 05, 2015 / Published: April 30, 2015.

Abstract: The aim of this work is to analyze the performance of a commercial micro gas turbine, focusing on the analysis of the fuel consumption and the outlet compressor and turbine temperature at various rpm, and to evaluate and compare the efficiency of the device. A test bench has been assembled with the available equipment in the laboratory of the department of mechanical and aerospace engineering in Roma. By using the software supplied by the manufacturer, the evaluation of the operating performance of the device at different speeds has been performed, obtaining all the values of interest.

Key words: Gas turbine, test bench, experimental tests.

Nomenclature

AUX	Auxiliaries
CC	Combustion chamber
ECU	Electronic control unit
GT	Gas turbine
m	Mass flow rate (kg/s)
n	Rotational speed (rpm)
ORC	Organic rankine cycle
Q	Volumetric flow rate (m^3/s)
T	Temperature (K)
TIT	Turbine inlet temperature
TOT	Turbine outlet temperature

Greek symbol

β	Compression ratio

1. Introduction

The main problem of power conversion based on turbomachinery is the lack of suitable data of small-scale equipment. The main objective of this work is to supply a suitable data for integrating and optimizing the design of turbomachinery at small-scale. The described tests contribute to the advancement of compressor/turbine behavior at small scale, and supply a valid help to design a highly efficient and robust power conversion technology.

Already in 1980 tests on a small-scale, heat pump has been carried out [1]. The radial heat pump compressor, driven by a dynamic expander had a nominal speed of about at 160,000 rpm and achieved an isentropic efficiency of 60% at pressure ratios in excess of 6.6. In 1996 [2], a two stage radial compressor for automotive air-conditioners was studied and analyzed. The rotor processed R123 with tip diameters of 48 mm and 38 mm, characterized by pressure ratios of 3.2 and 1.9, respectively. The overall peak efficiency including the intercooler between the two stages was predicted to be around 77%. In 2000 [3], it has been published first results of a 4 mm diameter turbine with prismatic blades reaching 1,400,000 rpm and an output power of 5 W. In 2004 [4], some authors give an overview of the 10 W MIT (Massachusetts Institute of Technology) micro-gas turbine project was presented. In 2001[5], the feasibility study of gas turbine engine with an output of 100 W, using 3D radial impellers operating at a pressure ratio of 3, requiring rotational speeds in excess of 850,000 rpm, has been presented.

In 2003, measured data on a 12 mm radial compressor rotating at 420 krpm and reaching pressure

Corresponding author: Roberto Capata, adjoint professor, research fields: turbomachinery and energy systems. E-mail: roberto.capata@uniroma1.it.

ratios of 1.6 and efficiencies in excess of 70% [6] were presented. In 2006, Japanese researchers [7], demonstrated the feasibility of a radial compressor rotating at 720,000 rpm reaching a pressure ratios up to 2 and isentropic efficiencies in excess of 60%. In 2011 [8], the development of a 3 kW gas turbine engine based on automotive turbocharger technology has been presented. So far, literature reports no successful experimental run of a gas turbine engine with output powers lower than 10 kW. This collection of works clearly demonstrates:

(1) the interest of small-scale turbomachinery;

(2) its technical feasibility;

(3) but at same time, the lack of a detailed collection of experimental tests.

Before describing the tests campaign, it can be notice that the small-scale turbomachinery suffers mainly from:

(1) low Reynolds-number flows;

(2) large relative tip clearance;

(3) non-adiabatic flow condition.

The effect of low Reynolds-numbers compressor stages has been thoroughly captured in early work [9-11] where a correction based on the machine Re-number and surface roughness was proposed. It is suggested that friction losses within the impeller correlate well with the Moody-diagram. Hence, impellers operated at low Reynolds numbers and increased relative surface roughness are suggested to yield lower performance. Why experimental investigation to validate both the Casey Reynolds number correction and the effect of surface roughness at small-scale, however, is missing, the author try to provide it by a combination of CFD (computational fluid dynamics) analysis and experimental tests [10].

Tip clearance has been investigated by various teams [12, 13], all claiming that a 10% increase in tip clearance decreases performance by 3-4 points. Recent work, however, shows that this trend may be considerably smaller or even larger depending on the impeller geometry and suggests that so far no clear correlation was found [14]. Large-scale impellers are currently operating at relative tip clearances of approximately 0.01. For small-scale impeller, a tip clearance of about 10 m has to be considered, which is hardly achievable with standard manufacturing tolerances. This suggests that small-scale impellers need to operate at increased relative tip clearance and therefore their efficiency is reduced.

Decreasing scale increases the surface-to-flow ratio. The flow in small-scale turbomachinery may therefore not be treated as adiabatic and heat addition or drain significantly alters performance [15].

In turbomachinery mean flow prediction tools, the different loss mechanisms are generally considered in an isolated way and linearly added to be subtracted from the ideal Euler-work. Considering their mutual influence improve the accuracy of performance prediction. Today, a theoretical and experimental analysis on the cross-influence of the loss mechanism is missing.

All these considerations have suggested to carry out a campaign of tests on a small-scale gas turbine device, to fill those gaps, previously described in this paragraph. The tested device was a group for aero modeling [16]. The knowledge of the performance of the system has been dictated by project program, in which this campaign has been carried out. In the framework of this research (SEALAB—sea laboratory—project to build a high-performance marine vehicle), it needed to find the “quasi-optimal” propulsion device and therefore the analysis of selected group becomes of fundamental importance.

2. Preliminary Considerations on the Tests Campaign

The experimental tests reported in this paper are a direct consequence of two projects being developed at the department of mechanical and aerospace engineering of Roma 1. In the first research, the performance mapping Graupner/JetCat turbo-prop model (“SEALAB” project) is obtained, while, in the second case, the

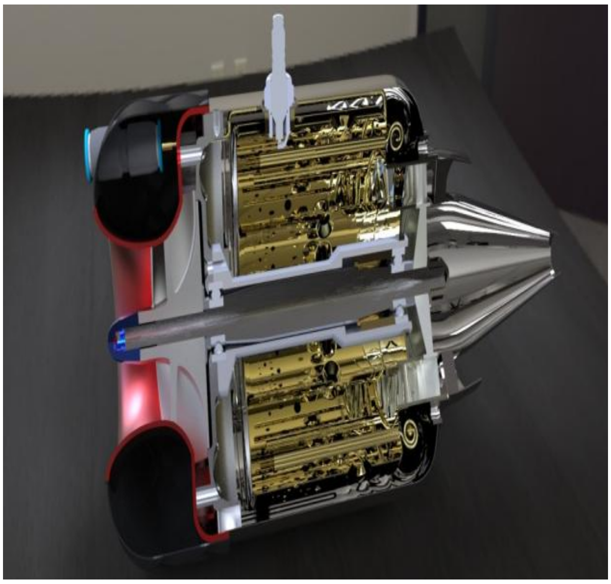

Fig. 1 Cut-away of the tested GT device.

thermal field mapping the combustion chamber and its specific consumption within the project PNR (national research plan) 714 with Italian Army Head Quarter for the realization of a portable device). Once a possible connection with an inverter has been realized, both devices, could be then used as a range-extender on hybrid vehicles, UAVs (unmanned aerial vehicle) or as simple portable generation tool.

2.1 The GT (Gas Turbine) Device

The Graupner/JetCat turbo-prop engine (Fig. 1) represents a successful combination of high power device and high-tech engineering. In the world of full-size aviation most types of propeller-driven machine have already been converted to turbojet power, but the engine relentless progresses have only just begun in the model aviation arena. As the name indicates, the turbo-prop engine, its full name is a turbo-jet propeller engine—comprises a gas turbine driving a compressor. The primary advantages of the aviation turbo-prop in full-size are in its compact shape and its economy and reliability at speeds below 700 km/h. Device characteristics are represented in Table 1.

Once the GT (gas turbine) device has been chosen, it was necessary to study all connections to build the test bench, and testing the group. The first step was to realize:

Table 1 GT published nameplate.

Technical characteristics	
Dimensions	$33 \times 130 \times 275$ mm^3
Rpm	30,000-112,000
Fuel consumption	130-700 mL/min
Fuel	Jet A1, kerosene, petroleum
Maintenance	Every 50 h
Exhaust temperature	480-620 ℃
Weight	2,359 g

(1) the hydraulic circuit;

(2) the electric circuit.

During this operation, it has been necessary to include additional tools and instruments, to completely map the performance of the two turbo machines, i.e., the compressor and turbine. This has, inevitably, complicated the system, and it has been necessary to investigate how and where to insert these new tools. The tool/instrument features were considered, especially turbine side.

3. Test Bench Assembling

Test bench assembly has been created as follows: firstly, the hydraulic circuit for the fuel supply, then the electrical circuit for the regulation and control of the machine have been realized; third, once studied the available software (which only provides the turbine output temperature TOT (turbine outlet temperature) and fuel consumption as a function of the rotational speed), the points where to insert additional measuring instruments have been identified; finally, the whole measuring chain for these devices has been realized.

3.1 Hydraulic Circuit

The JetCat engine can use deodorized kerosene, 1-K kerosene or Jet-A1 as fuel. Fuel must be mixed with 5% synthetic turbine oil. JetCat itself recommends Aeroshell 500 turbine oil although any turbine oil that conforms to MS23699 standards will work. The input and output fuel piping must be connected to the electronic shut-off valve.

The manufacturer recommends checking and clean the fuel filters every ten flights/runs. When the engine

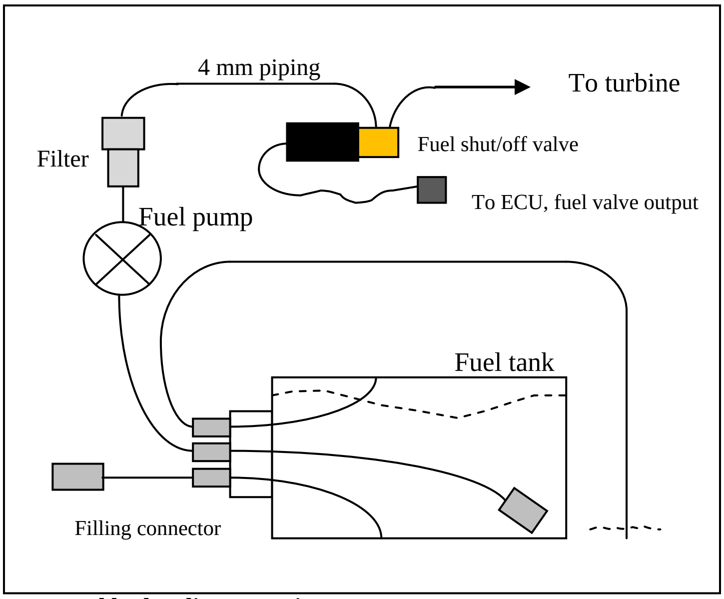

Fig. 2 Fuel supply system and hydraulic connections.

runs at full power, the control unit ECU checks the fuel line, from the pump to the engine.

The hydraulic system consists of the following components (Fig. 2):

- 5 L tank;
- fill fuel tank;
- 3 mm and 4 mm diameter tubes;
- electric pump (with its cables connected to the ECU);
- two solenoid valves (with its cables connected to the ECU);
- filter.

2.2 Electrical Circuit

Particular emphasis was placed on the implementation of the electric circuit. In fact, the whole system is controlled by the ECU. Thus, a wrong connection would have caused the failure of the tests. Electrical components are:

(1) the solenoid valves that control the fuel supply, and the cold-start device;

(2) the same ECU needs power supply;

(3) the fuel pump.

These components required a power supply, which is controlled and governed by the ECU. For these reasons, an electrical connection has been realized. The electrical circuit consists of the following components (Fig. 3):

(1) two 7.4 V and 6.4 A batteries;

(2) jet-tronic ECU P80 controller;

(3) control unit and external programming GSU (general service unit);

(4) two 6-wires phone cables;

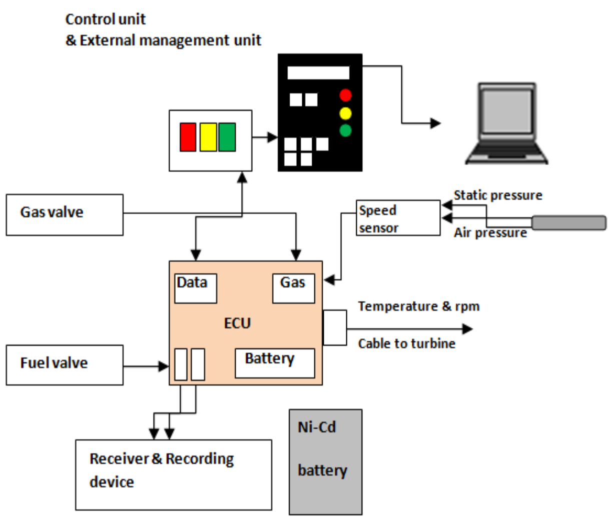

Fig. 3 Electric connections diagram.

(5) telephone cable—USB (universal serial bus) (for

connection to computer);

(6) auxiliary cables for the connection with the control radio;

(7) on-off extra security switch;

(8) thermocouple at the turbine's outlet.

3.3 Additional Instruments Implementation

To achieve a complete overview of the operating conditions, additional instruments are considered and inserted. Fig. 4 shows the additional measurement points used to map the compressor/turbine. The housing for the additional instruments has been created "ad hoc", on the case of the distributor and the return channel and in correspondence of the combustion chamber exit. The inlet conditions are the environmental ones. In this case, temperature and ambient pressure measuring, is repeated many times using commercial tools. However, the ambient temperature has fluctuated between 12 ℃ and 17 ℃, and the pressure between 100.67 kPa and 102.1 kPa. The operating speed and fuel consumption were derived from the power control software using turbine ECU. The additional devices inserted were a pressure sensor, flow-meter and two thermocouples [17, 18].

The insertion of a thermocouple, a pressure gauge and a flow-meter, compressor side, was necessary to map the working conditions of the compressor in any operating condition. This choice is derived from the analysis of the system and of the group. Indeed, using dedicated software, it is possible to only measure the turbine outlet temperature, the machine rotational speed and the fuel consumption. An analogous consideration for turbine, where the thermocouple probe is, covered by alumina and hardened with platinum thin coat, was inserted between stator and rotor of the turbine itself.

4. The Instruments Adopted

The instruments used in this second step of the tests are described below [18]:

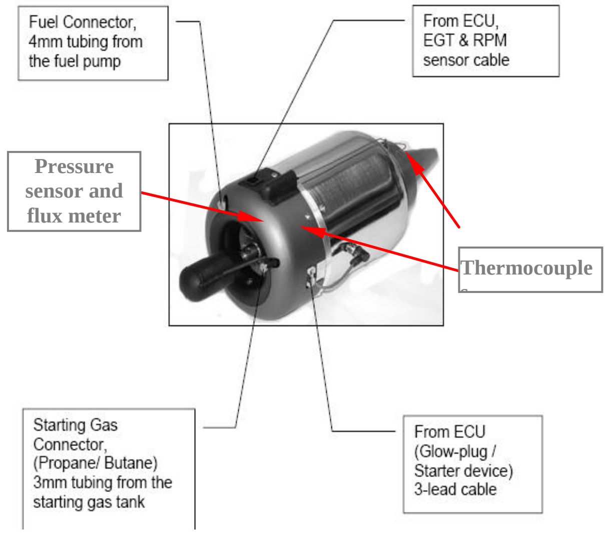

Fig. 4 Auxiliary instruments connection points.

(1) Thermocuople TC9M-A-1-K-C-L-K for high temperature. It is a sensor with special coating. Temperature measuring range of 100-1,600 ℃; protection sheath of recrystallized alumina KER 710(C-799) tube with hole at one end; special hardened platinum thimble;

(2) Thermocouple TC5-m-1-K-5-F-E-B-3; typical Fe-Co device; material AISI 16 stainless steel with measuring range of 0-600 ℃;

(3) Pressure gauge type TK-E-1-E-BO3U-M-V; measuring range of 0-3 bar;

(4) Flow meter MCR (main control room) D-G model.

It is a complex device that allows to measure:

• gas absolute pressure: this sensor references hard vacuum and reads incoming pressure both above and below local atmospheric pressure;

• gas temperature: MCR series flow controllers measures the incoming temperature of the gas flow. The temperature is displayed in degrees Celsius (℃);

• mass flow rate: the mass flow rate is the volumetric flow rate corrected to a standard temperature and pressure (typically 14,696 psia and 25 ℃).

4.1 Pretreatment and Normalization of the Data

Once the test bench has been assembled (Fig. 5), the various tests were performed, which will be described in detail below. All data obtained were then processed in order to have a valid database. It is important to note that the procedure described immediately below, was created and designed in such a way to make repeatable tests, always under the same conditions, and, always, under the same start-up, running and turn-off operations. All data have been collected via PC and then has been processed as follows [17, 18]:

(1) Gauss analysis of the deviations: mean values, mode, median and standard deviation have been automatically calculated by the software of the used acquisition card; the value of σ is proportional to the uncertainty of the measurement;

(2) Filtering: each datum that exceeded 3σ has been individually controlled, and neglected like spur if possible acquisition errors were not found;

(3) Normalization: the values are standardized in order to control that their distribution followed a Gaussian-type;

(4) Presentation: for each measure, the relative standard deviation and the medium value has been supplied (rms).

4.2 Fuel Consumption and Turbine Outlet Temperature

Several data were provided by the software and reported on graphics (Fig. 6). The tests consist, once turned-on the machine (following the steps indicated by the manufacturer) and reach the self-sustainable point, in the fixing the rotational speed and measuring, for a determined time period, the operating characteristics of the machine. Table 2 shows the time intervals and the adopted rotational speed for the tests. The total amount of the tests time is about 20 h. It is lower than the maintenance fixed value (25 h).

It is important to note that, the tests # 1 and # 2 were conducted with an interval of 5 min between two consecutive ignitions (Tables 2-4). The test # 3 has been conducted differently (Table 5). Once the machine has been switched off, it was expected that the outlet temperature, turbine side, was less than 50 ℃.

As it can be seen, the software supplied by the manufacturer, does not provide enough data. Previously, we discussed changes in the measuring instruments with the addition of other devices. These instruments

Fig. 5 The assembled test bench experimental tests and results.

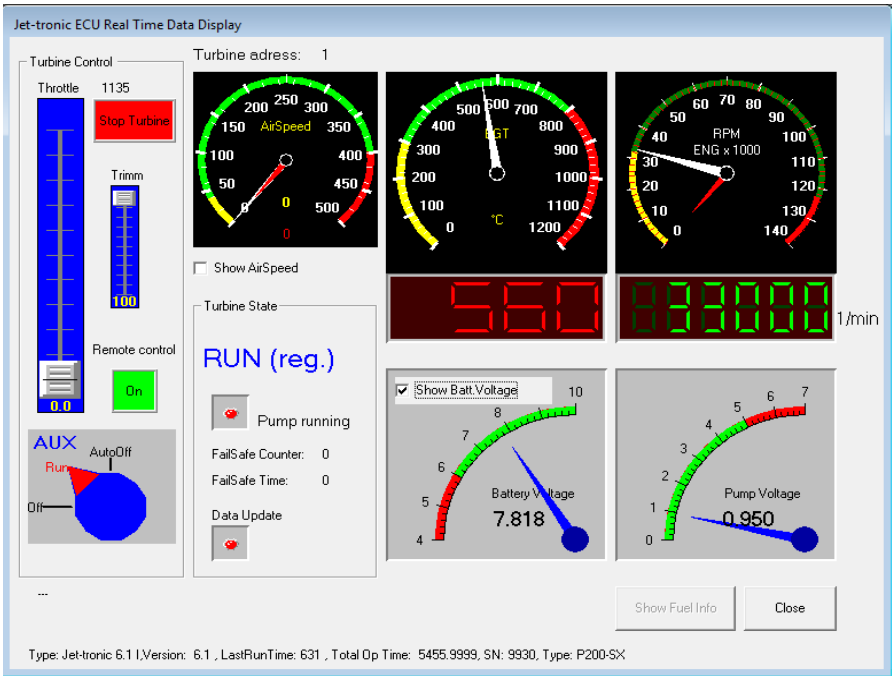

Fig. 6 The software interface with data at 33,000 rpm.

Table 2 Tests main data.

Test	rpm	Test time	# tests
Steady	33,000	5 m 50 s	10
Steady	45,000	5 m 30 s	10
Steady	60,000	5 m 40 s	10
Steady	75,000	5 m 30 s	10
Steady	90,000	5 m 42 s	10
Steady	110,000	5 m 20 s	10
Variable	33,000-108,000	5 m 33 s	4

Total tests time: = 377 m 22 s = 6 h 17 m 22 s.

Table 3 Run #1.

n (rpm)	TOT (K)	Consumption (L/min)
30,000	782	0.123
45,000	755	0.172
60,000	756	0.240
75,000	734	0.327
90,000	765	0.444
110,000	891	0.795

Table 4 Run #2.

n (rpm)	TOT (K)	Consumption (L/min)
30,000	833	0.124
45,000	807	0.156
60,000	781	0.210
75,000	791	0.277
90,000	803	0.371
110,000	935	0.731

Table 5 Run #3.

n (rpm)	TOT (K)	Consumption (L/min)
30,000	768	0.123
45,000	753	0.168
60,000	750	0.245
75,000	745	0.323
90,000	772	0.443
110,000	904	0.795

have allowed to record via PC, the values needed for the mapping of the compressor and turbine, which was the aim of the tests.

4.3 Compressor Tests

To be able to have a detailed analysis of the compressor, the outlet pressure and the temperature have been measured [19, 20]. Moreover, the devices, that measure the pressure, are also capable to measure the evolving mass flow rate. The initial conditions—the temperature and the pressure of the test chamber were detected [21]. In Table 6, all results are reporting. The Figs. 7 and 8 show the various operational conditions and trends. Note that some data were obtained by interpolation (at high). This is because the GT device has a limit on rotational speed (in this case limited

Table 6 Compressor main data.

n (rpm)	T_{inlet} (K)	T_{outlet} (K)
30,000	285-290	309.82
45,000	285-290	320.51
60,000	285-290	328.80
75,000	285-290	342.31
90,000	285-290	369.79
110,000	285-290	405.24
P_{inlet} (kPa)	Q_{outlet} (m^3/s)	β
100.6-102.1	0.16	2.25
100.6-102.1	0.20	2.32
100.6-102.1	0.21	2.40
100.6-102.1	0.23	2.54
100.6-102.1	0.26	2.70
100.6-102.1	0.31	2.94
m_{inlet} (kg/s)	η^*	
0.15	0.63	
0.18	0.64	
0.20	0.66	
0.27	0.67	
0.32	0.69	
0.38	0.72	

*Calculated.

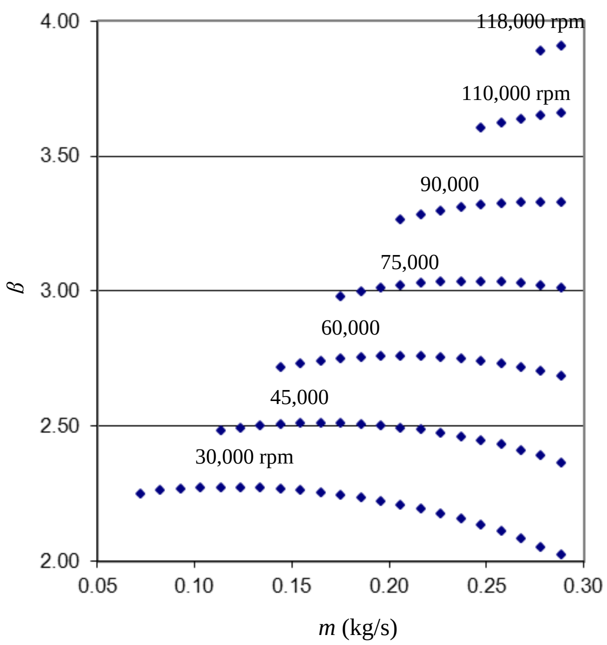

Fig. 7 Variation of compression ratio with mass flow rate.

to 118,000 rpm), while the manufacturer indicates higher speed can be reached. This restriction is due to, solely, safety reasons.

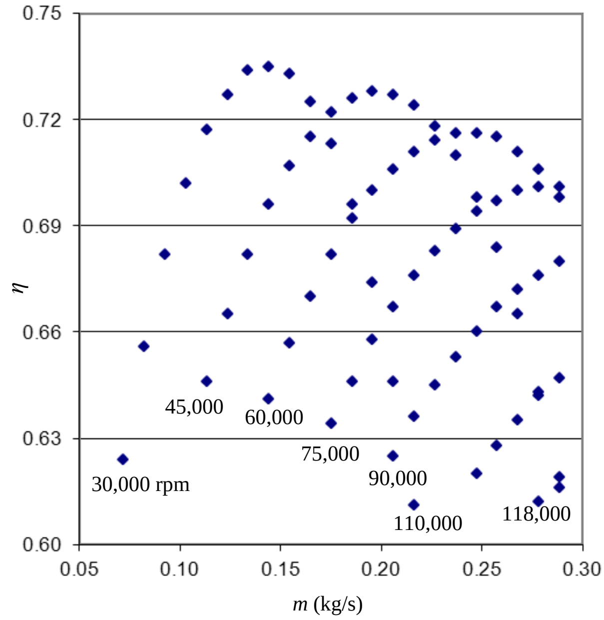

Fig. 8 Variation of efficiency with mass flow rate.

4.4 Turbine Tests

For turbine tests, the same procedure has been used. It has been monitored the inlet and outlet turbine temperatures, by using specific thermocouples. The results are shown in Table 7. The turbine outlet mass flow rate is calculated by the value of fuel and the air mass flow rate in the combustion chamber. The value of pressure drop, in CC (combustion chamber), is suggested by the manufacturer. And finally, it was possible to determine, preliminarily, the specific work. The knowledge of all operational conditions allows to compile Table 8. In Fig. 9, the variation of pressure ratio, TOT and work with mass flow rate are shown, respectively.

Table 7 Turbine main data.

n (rpm)	TIT (K)	TOT (K)	B (±3%)*	m_{inlet} (kg/s)**
30,000	786	773	2.18	0.16
45,000	782	753	2.25	0.19
60,000	785	743	2.38	0.22
75,000	794	733	2.46	0.29
90,000	869	763	2.62	0.34
110,000	1,078	893	2.85	0.41

*Estimated (manufacturer suggest to consider a pressure drop in combustion chamber within 3%); **Calculated ($m_{air,inlet}$ + $m_{fuel,inlet}$).

Table 8 Summarized GT device data.

n (rpm)	Consumption (L/min)	Consumption (kg/s)
30,000	0.123	0.00165
45,000	0.172	0.00229
60,000	0.240	0.00320
75,000	0.327	0.00436
90,000	0.444	0.00592
110,000	0.795	0.01061
Specific consumption (kg/J)		Specific work (J/kg)
5.6954×10^{-6}		2,987
1.6960×10^{-6}		10,230
1.3240×10^{-6}		15,363
9.6366×10^{-7}		23,253
4.7896×10^{-7}		44,203
2.6974×10^{-7}		95,526

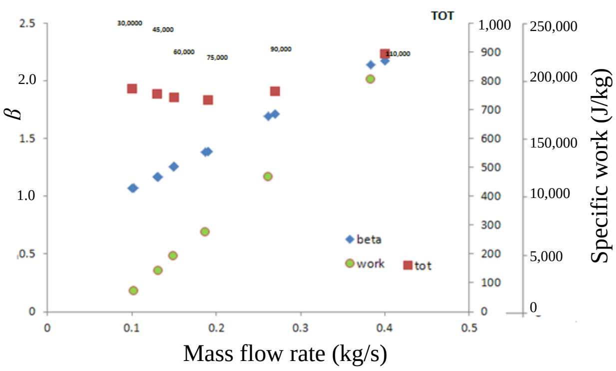

Fig. 9 Variation of the pressure ratio, TOT and specific work with mass flow rate.

5. Conclusions

The series of tests were conducted to highlight some very important aspects.

First of all, the "repetitiveness" of the tests made it possible to have a "confidence" in the treatment and interpretation of data, allowing to recognize and eliminate all errors, during measurements.

Compressor side: the tests performed made it possible to draw the operational maps; both the range of "compression ratio" and "mass flow rate" have been measured. This information is very useful for any other applications than those pre-defined by the manufacturer.

Turbine side: the efficiency of the turbine increases the rotation speed increases, and the highest value (approximately 10%) is reached at 110,000 rpm (lower if it is compared to standard values for gas). It is due to the final application required to the turbine, which operates in propulsion mode for airplanes models.

Important information is the fuel consumption. It varies from 123.5 mL/min (at 33,000 rpm) to 796 mL/min (at 110,000 rpm).

Finally, a consideration on the possibility of recovering the thermal waste energy. The outlet temperature is always higher than 450 ℃. This gives us the option to use the exhaust gases as a low heat source for another cycle (for example, an organic rankine cycle), to increase the total efficiency, in particular vehicular application. The net work obtained from the entire system could also be used for the electric power generation that can be required by small utilities.

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