# THE ULTRA-MICRO GAS TURBINE GENERATOR PROJECT AT UDR1: EXPERIMENTAL ASSESSMENT OF THE COMPRESSOR MAP AND OF THE PRESSURE LOSSES AT THE REGENERATIVE COMBUSTION CHAMBER

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***Abstract.*** *The paper reports the procedures and the results of a series of tests conducted at the Department of Mechanical & Aeronautical Engineering of UDR1 to obtain the map of an ultra-micro radial compressor ($D_{ext}$ = 0.038 m), as well as the head losses and the combustion efficiency of the corresponding ultra-micro combustion chamber. This work is a part of a research aimed at the conception, design and prototyping of an ultra-micro thermo-electrical device for portable power generation. The novelty of the research consists in the fact that the thermal engine is a (ultra-micro) gas turbine set. In previous papers the entire course of the project has been described, from the initial feasibility analysis to the optimization of the thermodynamic cycle and the selection of the most convenient materials. In a subsequent stage, several different configurations have been assessed to select the most proper geometry and structural characteristics of the most relevant components (radial compressor, radial turbine, combustion chamber, electrical motor and generator, bearings, regenerative heat exchanger). The final configuration of the prototype (for which a patent is pending) is described in the paper as well, together with some of the results of the final operational tests.*

***Keywords****: ultra-micro gas turbine, radial compressor, combustion chamber, compressor map, pressure losses*

## NOMENCLATURE

| | |
|---|---|
| A | Air fluxes |
| C | Compressor |
| CC | Combustion chamber |
| COT | Compressor outlet Temperature [K] |
| D | Diameter [m] |
| DOE | U.S. Department of Energy |
| EG | Electrical generator |
| EM | Electrical motor |
| k | Isentropic coefficient |
| F | Fuel |
| G | Gas fluxes |
| GT | Gas Turbine |
| m | Mass flow rate [g/s] |
| MEMS | Micro Electro Mechanical Systems |
| MST | Micro System Technology |
| MTBF | Mean time before failure |
| P | Power [W] |
| p | Pressure [bar] |
| Q | Flow rate |
| RE | Regenerative heat exchanger |
| SLPM | Standard liter per minute [l/min] |
| T | Turbine |
| T | Temperature [K] |
| TIT | Turbine Inlet Temperature [K] |
| TOT | Turbine Outlet Temperature [K] |
| U | Rotational speed [m/s] |
| UMGTG | Ultra Micro Gas Turbine Generator |
| UDR 1 | University of Roma 1 |
| | |
| $\beta$ | compression ratio |
| $\eta$ | efficiency |

## 1. INTRODUCTION

Although the quest for ultra-micro (and even nano-) portable power supply systems is over a decade old, and since 1997 several research teams all around the world have been actively involved in this rather niche-type field, today there are only few and incomplete prototypes of such devices, most of them failing to reach acceptable performances. The fundamental idea is that of powering a micro-scale electrical generator with a thermal device that can run on a generally available fuel (kerosene, propane/butane mixtures, methane, hydrogen, etc.), so that the user can avoid carrying a quite heavy battery package. Reciprocating engines being unfeasible at these small scales (below 500 W), the use of gas turbine micro-plants has been investigated. Major advances have been made in the US (Epstein et al. 1997), in Japan (Ishihama et al. 2003, Johnson et al. 2003), in Belgium (Leuven) (Peirs et al. 2003) and in Italy (Capata and Sciubba 2007, 2008).

All teams have until now adopted the same configuration used in "large scale" gas turbines. There are other configurational variations of this cycle, but for very small units, the simple cycle described above is the only one considered to date.

For thermo-fluid dynamic reasons, the efficiency of a gas turbine set can be increased by either one of the following actions:

a) increase the maximum allowable combustion gas temperature, TIT
b) correspondingly increase the pressure ratio $\beta$ of the compressor C
c) maximize the degree of regeneration (i.e., the amount of the sensible heat of the hot turbine discharge gases recovered in the preheating of the air at compressor exit) in the heat exchanger RE

The increase of the TIT is limited solely by technological reasons: the blades of the turbine are usually made of special alloys that can sustain prolonged operation at TIT of about 1300 K. For higher gas temperatures, internally cooled blades have been employed, or alternatively ceramic materials have been used to manufacture the blades. In ultra-small GT, blade cooling is technically impossible, while ceramics have demonstrated to be unsuitable for the extremely high rotational speeds required by these devices.

The pressure ratio of the radial compressor is basically dictated by its rotational speed U, to whose square the specific work of the compressor (and hence the pressure ratio) is proportional.

Thus, higher $\beta$ require higher U, and this poses both fluid-dynamic (losses, leakage, efficiency) and structural (creep, vibrations) problems of difficult solution..

## 2. THE UMGTG DEVICE OF THE UDR1

The UMGTG-UDR1 device (figure 1) consists of the following fundamental parts: compressor C, turbine T, combustion chamber CC, regenerator RE, electrical motor EM, electrical generator EG, and case. External to the case, but of course properly connected to it, is the fuel tank assembly.

The operation of the device is as the same as that of a "large scale" GT (Capata and Sciubba 2005): in particular, one may safely assume that the representative thermodynamic process is a regenerated Brayton cycle.

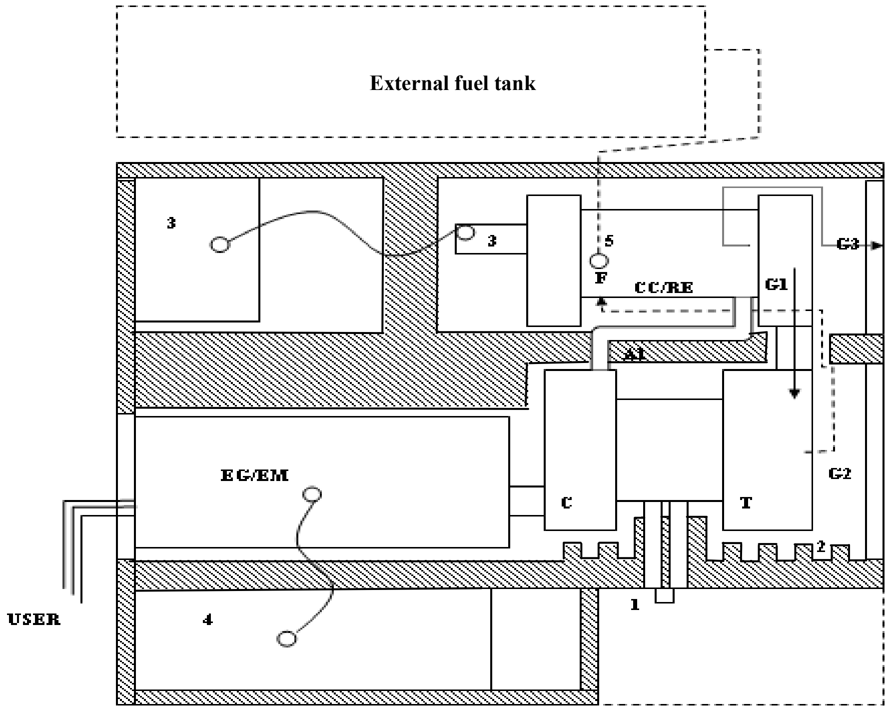
Here follow, the main components of the device and their use are briefly described.

The Electric motor leads the compressor. At the start, the outlet compressed air (flux A1) comes in to the combustion chamber/regenerative heat exchanger device.

Here, following a spiral path, the air gets in contact with the external wall of the chamber, realizing a first part of the regenerative process, by heat transfer. Once the combustion is stabilized, the outlet turbine exhausts (flux G2) evolve in the external fins of the regenerator, realizing the second part of the regeneration and in counter flow with the compressor air; at least the hot gases exit (G3). Thus the regeneration of the air is achieved, with undoubtedly saving fuel (F1). At same time, the electric motor is switched, by inverter and electronic control, and the device supply the required output.

The electrical (3) ignition device (plug, electric circuits, etc), as well as the inverter (4) are into the case. They are localized on the both side of the "box", for easy substitution or maintenance. An oil lubrification piping (1) is considered, at the moment, to improve and increase the bearings MTBF. A future step is to substitute the hybrid bearings (GRW® ceramic balls and stainless AISI 316 support) with air ones.

Finally, to dissipate the exceed heat, many cooling fins (2) are shaping in the internal support of the case.




Legenda:

1. Bearings lubrification piping
2. Cooling fins
3. Starter/plug
4. Driver/inverter for EG/EM
5. Inlet fuel piping

Figure 1. Reference layout of the UMGTG (not to scale).

### 2.1 Compressor

Both rotors compressor and turbine are fitted on same shaft, with commercial ball hybrid bearings for elevated rotational speed (150000-200000 rpm [GRW ®]).
The compressor is a single-stage radial machine. The starter motor supplies the necessary power to launch the machine at start-up, raising the rotational speed of the compressor with a linear ramp until self-sustaining mode is attained.

Every effort was made in the design phase to properly insulate the compressor from the heat source, and as a result neither thermal stresses nor excessive bearing temperature represent a problem. Table 1 summarizes the rotor characteristics.

Table 1. Compressor data

| | |
|---|---|
| Material | ERGAL |
| External diameter [mm] | 38 |
| Internal diameter [mm] | 22 |
| Shaft diameter [mm] | 6 |
| Number of blades | 12 |

### 2.2 Combustion chamber & heat exchanger (patented)

The heat exchanger is an ultra-compact, annular type, gas-to-air device made of lamellar steel. Its operating temperatures range from 450 to 1300 K. Its efficiency is in the same range of the large-scale heat exchangers. The combustion chamber is a cylindrical chamber with prewhirl but no premixing. Air intake is tangential while fuel injection is radial. The hot gases exit either axially or tangentially: both solutions have been tested. Table 2 contains the main geometrical data of the combustion chamber.

The regenerative section is obtained by arranging the external wall of the combustor to be a three-layers cylindrical shell. The air from the compressor evolves in the internal cylinder, the gas turbine exhaust gas in the external one. This admittedly unusual arrangement is very compact, allows the regenerator to work in practice as a counter flow heat exchanger, and maximizes the heat input into the compressed air, “sandwiched” between two hot streams. Notice that the inner fins (compressed air side) are in direct contact with the external wall of the combustion chamber, and thus increase the degree of regeneration.

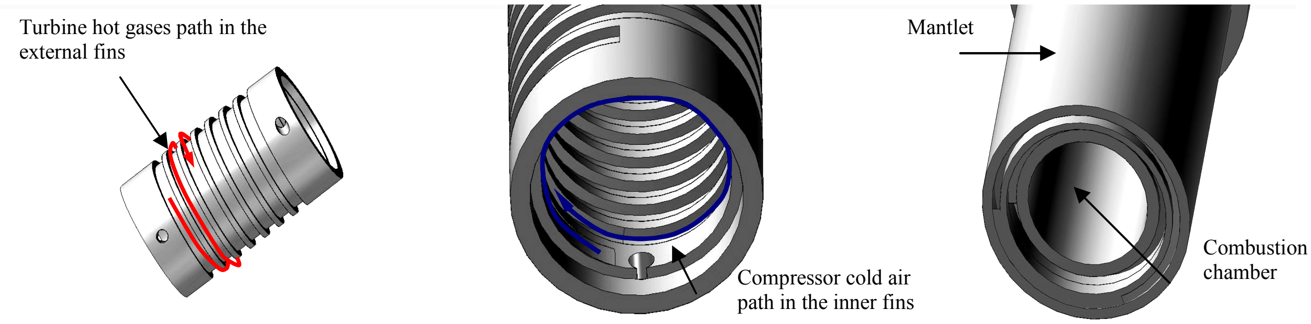



Figure 2. Sketch drawings of  complex of the combustion chamber/regenerative heat exchanger, and the final assembly

Table 2. Combustion chamber data

| | |
|---|---|
| Material | AISI 314 |
| Diameter [mm] | 35 |
| Length [mm] | 60 |
| CC length [mm] | 55 |
| CC diameter [mm] | 25 |

## 3.TESTS MEASURE PROTOCOL

In this Section the testing procedures has been described. The campaign has been carried out, adopting the following procedures:

### *a. Definition of the acquisition time*

This parameter has been defined according to the type of the required measures. It has been focused on 3 operational mode:

- starte-up
- transitory
- steady condition

For each mode it has been chosen the "characteristic" time of acquisition, reported in table 3.

Table 3. Acquisition time

| Operational mode | Acquisition time |
|---|---|
| start | 3 minutes |
| transitory | 3÷5 minutes |
| steady | 10÷15 minutes |

### *b. Frequency of acquisition*

Such frequency has been varied within the sensitivity range of the instrument. The higher value is need to acquire the higher frequencies in the transitory. So we adopt 3 working frequency for each operational mode
.

- 10 Hz (steady)
- 100 Hz (start-up)
- 1000 Hz (transitory)

### *c. Environment parameters*

The external environment temperature has been measured, regularly, every 15 minutes during the tests. At the scope a bulb thermometer has been used (sensibility of 0.5 °C, error of ± 0.2 °C).

***d. Instruments nameplate***

- The pressure has been directly monitored by instrument BRONKHORST Series P-502C (400 bars) integrated (measure field: min. 2-100 mbar - max. 8-400 bar). The device supplies on display the value of increment of the pressure (measured in psi, conversion factor: 1 bar = 14,53 psi). The instrument is shown in figure 3a;
- The flow rate has been computed with the BRONKHORST device -series F-112AC (range field: min. 0,2-10 liters/min - max. 5-250 liters/min);
- The temperature of the compressor outlet has been directly measured by commercial thermocouple type K (acquisition range: -40° C, +1000°C – see figure 3b) directly connected to a digital device capable to supply the actual experimental temperature.

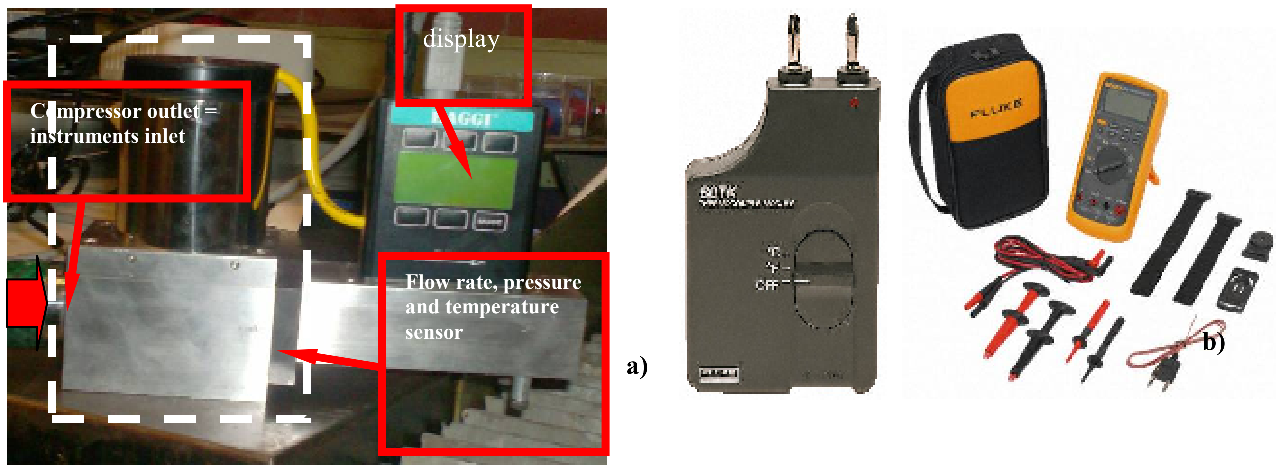



Figure 3. a) the BRONKHORST device, b) the K thermocouple and the digital device

**e. Pre-treatment and normalization of the data**

All data has been collected via PC and then has been so processed:
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;
2) Filtering: each data that exceeded 3$\sigma$ has been individually controlled, and neglected like spurious 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 mean value has been supplied ("rms").

**f. Tests elapsed time**

The tests have been subdivided in several sessions, with effective actual time for every session between 5÷7 hours. The total testing hours has been, therefore, equal to 42 h.

**4. COMPRESSOR MAPS**

A dedicated test bench was built (figure 4) to reproduce the structural stress on the device in "cold" conditions (the turbine being driven by a jet of compressed air at room temperature) and derive the compressor map.

A volumetric high-precision flow meter, a pressure gage and a thermocouple are inserted on the turbine inlet channel to evaluate the inlet conditions and to ensure steady operation. The same type of devices are inserted on the compressor outlet channel, to compute the pressure ratio and the compressor efficiency. A thermocouple and a pressure gauge are inserted on the turbine exhaust. The results are summerized below.

Table 4. Compressor experimental data

| Q(SLPM) | p(bar)* | T(°C) | β | η |
|---|---|---|---|---|
| | | | | |
| 21,03 | 1.0138 | 26,44 | 2,00 | 0,691429 |
| 23,50 | 1.0145 | 26,45 | 2,02 | 0,699505 |
| 37,20 | 1.0262 | 26,48 | 2,03 | 0,702632 |
| 38,10 | 1.0275 | 26,50 | 2,03 | 0,701215 |
| 46,90 | 1.0372 | 26,55 | 2,04 | 0,701026 |

| | | | | |
|---|---|---|---|---|
| 48,00 | 1.0392 | 26,57 | 2,04 | 0,699993 |
| 50,00 | 1.0420 | 27,10 | 2,04 | 0,649097 |
| 67,00 | 1.0475 | 27,63 | 2,05 | 0,606531 |
| 70,00 | 1.0516 | 28,16 | 2,05 | 0,568902 |
| 87,00 | 1.0606 | 28,69 | 2,06 | 0,537791 |
| 96,00 | 1.1012 | 29,30 | 2,10 | 0,522079 |
| 101,00 | 1.1149 | 29,91 | 2,12 | 0,498528 |

* increment of pressure

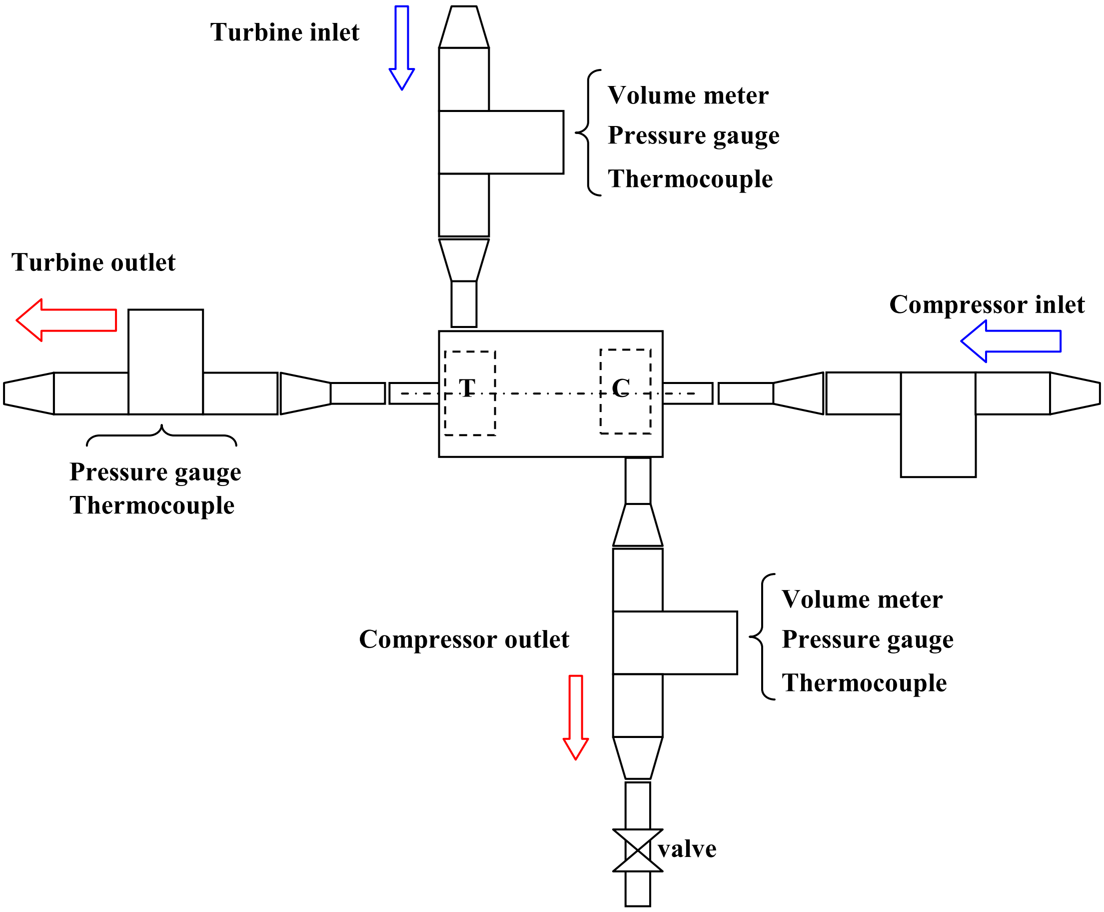



Figure 4. The dedicated test bench

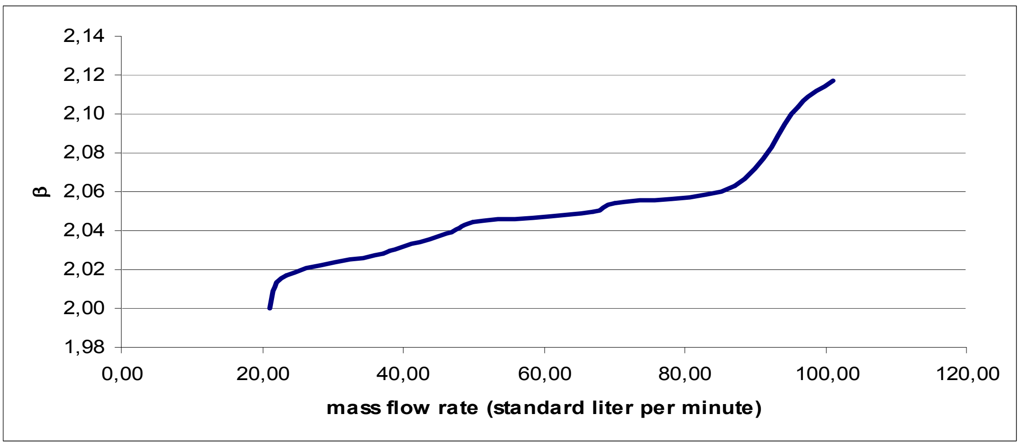



Figure 5. The experimental map $\dot{m}/\beta$ of the compressor

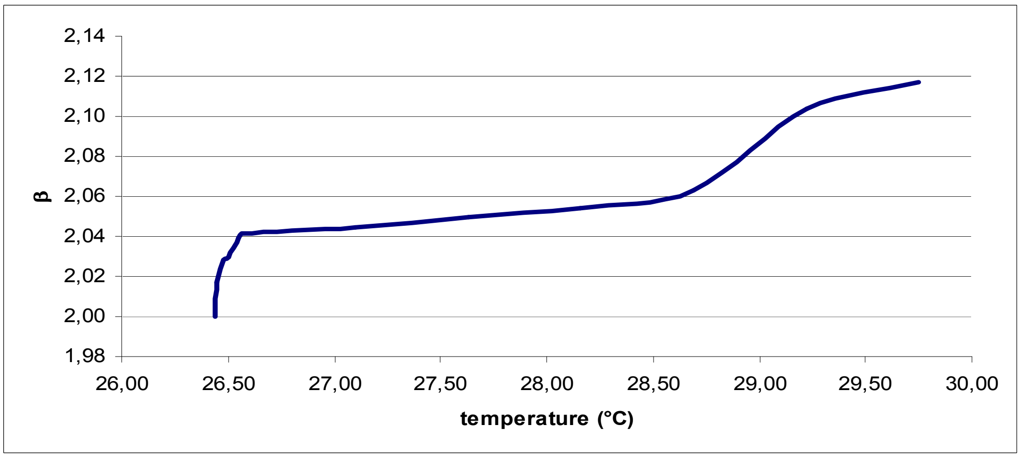



Figure 6. The experimental map T/β of the compressor

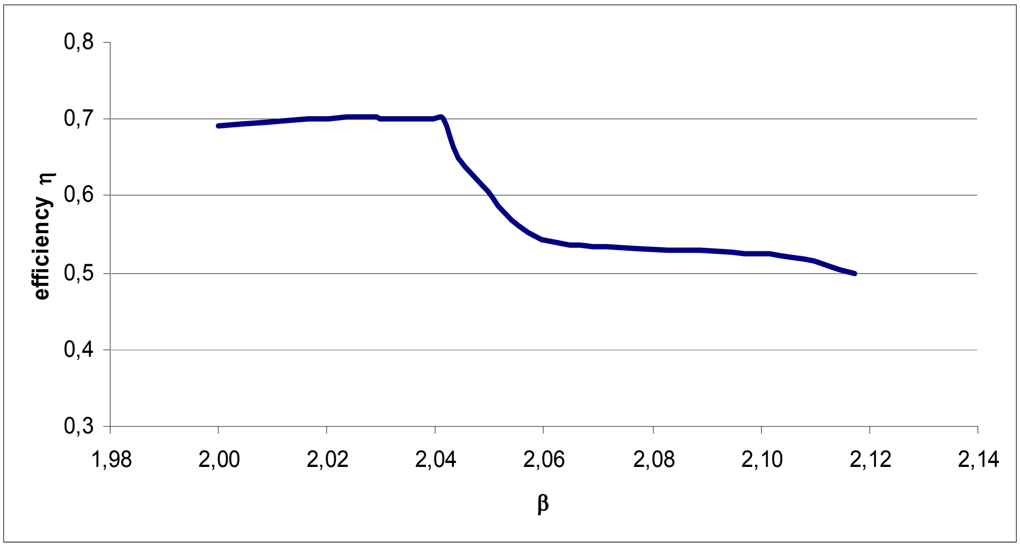



Figure 7. The experimental map β/η of the compressor

## 5. COMBUSTION CHAMBER PRESSURE LOSSES EVALUATION

The losses in the combustion chamber have been measured by the experimental setup represented in figure 8. A jet of compressed air (2 bar) is sent into the CC/RE device to analyze the actual pressure losses in the inner fins.

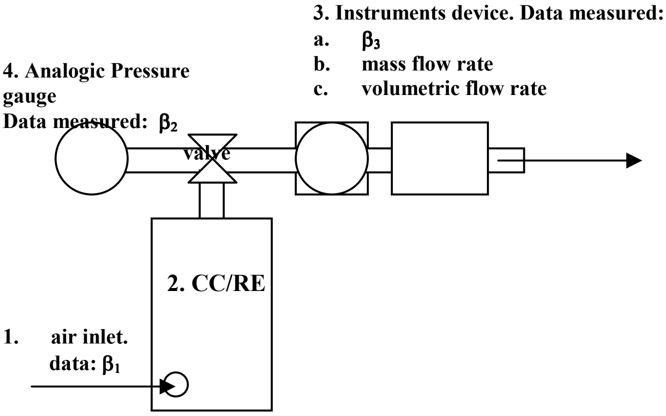



Figure 8. Test bench layout

Table 5. Reference data to calculate the head losses

| $\beta_3$ | $\Delta\beta_3$ | $V_3$ | $m_3$ | e% |
|---|---|---|---|---|
| kPa | kPa | $cm^3/s$ | $cm^3/s$ | % |
| 159 | 0 | 1040,0 | 1065,0 | 0,2 |
| 223 | 1 | 1463,3 | 1508,3 | 0,7 |
| 270 | 2 | 1688,3 | 1766,7 | 0,9 |
| 308 | 4 | 1991,7 | 2113,3 | 1,3 |
| 359 | 5 | - | - | 1,5 |
| 404 | 7 | - | - | 1,7 |

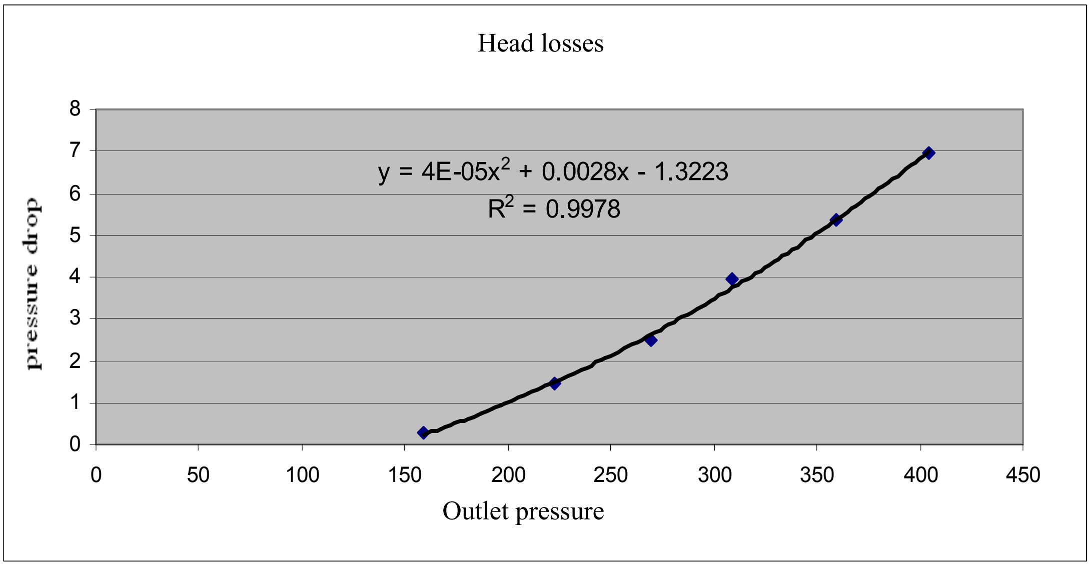



Figure 9. Evaluation of the pressure drop in the combustion chamber

The data could be fitted by a second order polynomial (see Figure 9) with an accuracy equal to 0.9978. The head losses vary with the second power of the outlet pressure and are of the order of 2% at steady conditions (see column " e%" of table 5).

## 6. FUTURE DEVELOPMENTS AND CONCLUSIONS

All components having been tested separately (Capata and Sciubba 2007, 2008), they have then undergone final. An additional “final test” campaign is underway at the time of this writing, its goal being both that of obtaining prototype acceptance by the customer and of indicating future possible improvements to introduce in the device before its pre-commercialization.

Meanwhile we want to underline the approach to realizing devices at these scales utilizes industry-derived -and rather well demonstrated- micromachining technology.

The economic impact of UMGTG devices will depend both on their performance levels and the manufacturing costs, both of which have yet to be proven. It is certainly possible, however, that ultra micro GT sets may one day be competitive with conventional machines for what the installed kW cost is concerned.

Even at much higher costs, they would have already no competitors as compact power sources for portable electronics equipment and ultra-small vehicles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