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Preliminary Considerations on the Thermodynamic Feasibility and Possible Design of Ultra-, Micro- and Nano-Gas Turbines

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Abstract

The paper describes a preliminary feasibility analysis of a nano-GT (50÷200 W) for portable power generation. The system is examined under both a thermodynamic and an operative point of view. The technical problems posed by a practical implementation of an operative system are assessed first via a preliminary calculation of the overall thermodynamic performance of the real cycle, and then via a first-order design of the main components. The extremely small size of the device makes a preliminary estimate of the components performance problematic (the available data are scant and unreliable): it is likely that a real advance in this field must go through a series of detailed fluid-dynamic simulations. The results of our preliminary sizing are compared with the existing technical literature in an attempt to further define the material selection, in view of a possible prototype production. Some considerations are also offered about the final utilisation of nano-GTs: in our opinion, they represent a realistic alternative to batteries for powering optical, GPS and other satellite devices, or in medical applications.

Keywords: Nano-GT, micro fabrication, power production

1. Introduction

The recent developments in the field of nano devices and micro fabrication techniques have opened the possibility of building a nano GT for power production[1] (Hepstein, 2003; Peirs et al., 2003). The manufacturing technologies of the semiconductor ceramic industry create the possibility of machining GT sets with diameters of millimetres, with airfoil dimensions spanning fractions of a millimetre. The growing capability of industry to build such small complex parts (dimensions in the 1-10,000 μm size range) and to assemble the whole device with submicron precision encourage the research in this particular field of energy systems. Such parts can be produced with photolithography[2] (Hepstein, 2003) and parallel manufacturing techniques, thus decreasing the product cost in large-scale production. The assemblies are known in the US as micro-electrical-mechanical-systems(MEMS), while in Japan and Europe they are referred to as "Microsystems". In this paper the assembled device will be referred to as a *nano-system* due to the dimensions considered here. A possible use for these devices is portable power generation; in fact, the proliferation of small, portable electronics-computers, digital assistants, cellular phones, GPS receivers, as well as the new emerging families of biomedical diagnostic sensors, require ultra-compact energy supplies.

[1] There is some confusion in the terminology referring to small-scale turbomachinery: the term "micro-gas turbines" is customarily used for units that deliver 25-100 kW. Therefore, we shall retain the name "nano-GT" for the devices discussed in this paper, in spite of the fact that in manufacturing this indicates sub-millimetre scales. We avoid the notation "ultra-micro GT" introduced in van der Braembusche (2002) because it may be misleading in the present context.

[2] Photolithography (also referred to as "microlithography" or "nanolithography") is a process used in the electronic industry to transfer a pattern from a photomask (also called reticle) to the surface of a substrate. Commonly, crystalline silicon in the form of a wafer is used as a choice of substrate, but there are several other options including, but not limited to, glass, sapphire, and metal.

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Also, the continuing advance in micro-electronics allows the shrinking of electronic subsystems of mobile devices such as ground robots and air vehicles. In particular, small mobile systems require increasingly compact power and propulsion units. Hydrocarbon fuels burned in air have 30 times the energy density of the best current lithium chemistry-based batteries. Given the need for high power density energy conversion in very small packages, a millimetre-scale gas turbine is a possible short term candidate. This paper studies the possibility of developing a nano-GT set for power production, and is essentially divided in two conceptual parts: a short discussion of microscale implications for cycle analysis and scaling effects on the main design parameters and some preliminary design considerations. A brief discussion about the need for specialised aerodynamic and structural design studies is also included, together with a preliminary analysis of some materials that look to be good candidates for industrial production. The paper closure provides the Authors' assessment of possible future developments.

2. Scaling Considerations

The overall system is determined by thermodynamics while the design of the system's components is influenced by fluid dynamic and structural considerations and by material selection. The physical constraints on the design of the micro-scale mechanical components are often different from the present large-scale standards, so that the optimal component and system designs are different as well. Conceptually, any of the thermodynamic cycles presently adopted for large-scale systems could be realised at micro-scale. The Brayton power cycle (gas turbine) is, though, superior for power density, simplicity of fabrication and ultimate efficiency.

A conventional GT set consists of a compressor, a combustion chamber and a turbine that powers the compressor. The residual enthalpy in the exhaust stream may provide thrust or feed a thermal recovery system. Under these considerations, tens of watts could be produced when such a device is scaled to millimetre size if the power per unit of air flow is maintained. In fact, dimensional analysis indicates that the power P generated by a gas turbine is proportional to the density of the gas, to the fifth power of the diameter and to the third power of the rotational speed:

$$P \propto \rho D^5 n^3 \tag{1}$$

The power per unit volume is thus:

$$P/V \propto \rho D^2 n^3 \tag{2}$$

For a given pressure ratio the velocity of the fluid at nozzle exit is - within certain limits - independent of the size of the nozzles. Thus, once the rotor head coefficient has been selected, the circumferential speed is constant and independent of the turbine size. This means:

$$D \cdot n = 2KU_{max} = \text{constant} \tag{3}$$

where K is a constant of order 1. The power density is thus inversely proportional to the size:

$$P/V \propto (8K^3U^3_{max})/D \tag{4}$$

and it is thus seen to <u>increase with miniaturisation</u>. The consequent mass reduction is advantageous for fluid-dynamic applications. Since the product $D \cdot n = 2 \cdot K \cdot U_{max}$ is limited by the admissible material stress, the rotational speed is inversely proportional to the diameter. For a turbine diameter of 10 mm, the rotational speed corresponding to sonic flow is already 650,000 rpm. For a turbine of 5 mm the rotational speed exceeds 1 million rpm. This is clearly beyond the limit of present ball bearings technology, and gas or liquid bearings are required.

The Reynolds number, Re, characteristic of the flow is defined as:

$$Re = u \cdot L / \nu \tag{5}$$

with u a characteristic speed, L a characteristic dimension of the flow channels, and ν the kinematic viscosity. The flow velocity in turbomachinery passages is proportional to U_{max}. The Reynolds number is thus proportional to size and therefore decreases with miniaturisation. For small turbines the flow will be substantially less turbulent than in large-scale machines because the turbulent energy spectrum is "squeezed" between the large-eddy scales (whose size is of the order of the channel hydraulic diameter) and the Kolmogorov scales (whose size is proportional to $Re^{-1/4}$). This means that the viscous friction losses will be higher and that mixing of the fuel-air mixture will be somewhat slower. Current power densities are attained with combustor exit temperatures of 1200-1600 K, and current rotor peripheral speeds are in the range 300-400 m/s: this implies that the rotating structures are centrifugally stressed to several hundreds MPa, since the power density of both turbomachinery and electrical machines scales with the square of the speed. Particular attention must be given to the construction of low friction bearings; in fact, the tight geometric tolerances and clearances between rotating and stationary parts must be carefully studied to inhibit fluid leakage that at micro-scales cannot be neglected and introduces relevant losses. Also, heat exchange phenomena must be evaluated anew, due to the operational dimensions, possibly through a significant amount of CFD simulations, to improve and define the type and size of the combustion chamber and of the recovery heat exchanger. The physics and mechanics influencing the design of the components change significantly with sca1e, so that the final designs can differ substantially from the existing ones. Examples of these differences include the viscous forces in the fluid, usable strength of materials (larger at micro-scale), surface area-to-volume ratios (larger at micro-scale), and manufacturing constraints. There are many design choices for these systems. For example a regenerated cycle, which requires the addition of a heat exchanger, offers many benefits, including reduced fuel consumption and relaxed turbomachinery performance requirements, but it introduces additional design and fabrication complexity. In the following section, we shall therefore first analyse a configuration based on a simple Brayton cycle, and then two of its relatively minor modifications: intercooling and regeneration. More complex process configurations produce design problems that do not seem to admit a practical solution at the present state of the technology.

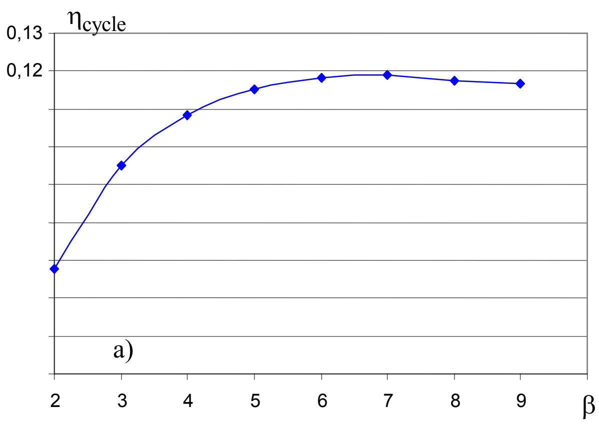
3.GT Cycle Calculations

3.1 The simple Brayton open cycle

We shall consider first a simple Brayton cycle, assuming polytropic expansion and compression and isobaric combustion. In particular:

- The TIT=1600K and the compressor inlet temperature T_1=T_{in}=288K (as suggested by [Hepstein 2003). Higher TITs are problematic, because at these scales it is very difficult to implement an efficient rotor cooling system. In fact, the device nano-dimensions would require an effective cooling system, due to the minimal thermal exchange surface: as a consequence of the practical impossibility of implementing such a system, the material behaviour strongly influences the choice of the configuration. For both ceramic and steel alloys, thermal stresses rapidly affect the overall life of the components:
- β=2 due to technological reasons: this is the optimal value of the pressure ratio that allows for a radial single-stage configuration for both compressor and turbine and does not require an excessive peripheral velocity;
- Isoentropic coefficients for compression and expansion are respectively equal to k_c = 1.4 for air and k_t = 1.32 for the hot combustion gases;
- The fuel is gaseous hydrogen. This choice is dictated by the need to limit the overall dimensions of the combustion chamber;
- We prescribe combustion, mechanical and electrical efficiencies equal to η_c = 0.9, η_m = η_e = 0.95 respectively;
- Pressure losses are fixed in $\Delta p_{exhaust}$ ≅1% and Δp_{cc} ≅ 3 % p_2;
- The polytropic efficiency was varied in the range 0.7 ≤ η_{pc} ≤ 0.8 and 0.72 ≤ η_{pt} ≤ 0.88, calculated adopting the following formula (Shepherd, 1956):

$$\frac{1-\eta_1}{1-\eta_2} = 0.5 + 0.5 \cdot \left(\frac{Re_1}{Re_2}\right)^{1/5} \tag{6}$$



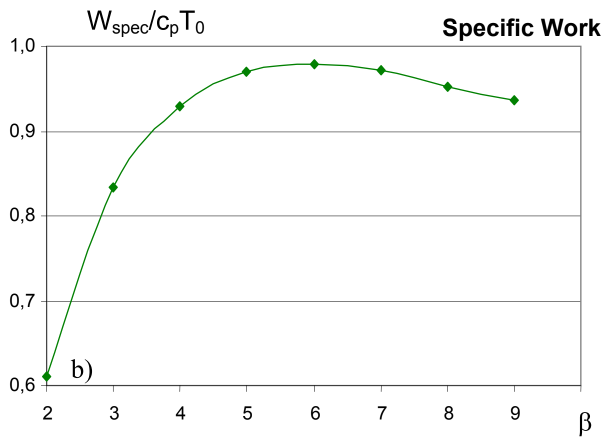


Figure 1. a) simple Brayton cycle efficiency; b) specific work in function of compression ratio for a simple Brayton cycle.

where Re_1 is the value calculated for the nano device (10^4) and Re_2 is the usual range for large-scale machines (10^7). Since the lower-than-usual polytropic efficiencies result in values of the cycle temperatures and of the specific work substantially different from those usually accepted for large-scale machines, we report in *Figures 1,* 2, 5 and *7* the behaviour of compressor outlet temperature, TOT and efficiency with respect to β_{cycle}, η_{pc} and η_{pt}. The simulations were performed using a modular simulator, CAMEL®, developed by our research group (Fiorini and Sciubba, 2005). For comparison, the values for optimal η are also reported here, though the corresponding configuration has not been studied.

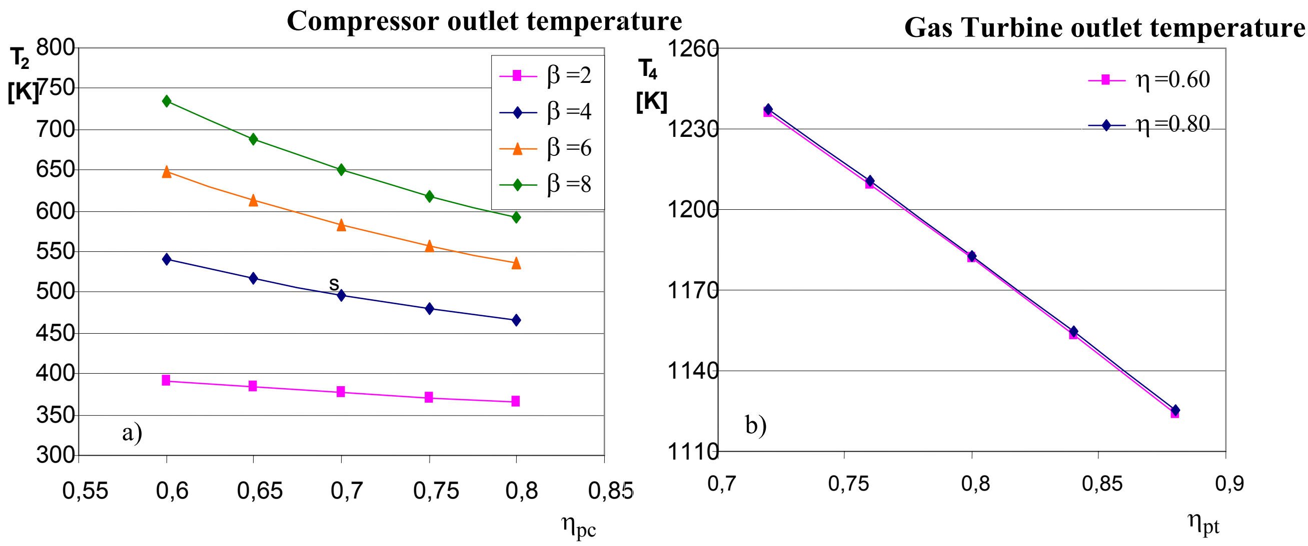


Figure 2. Dependence on the polytropic efficiency of: a) compressor outlet temperature and b) turbine outlet temperature.

TABLE I. SIMPLE BRAYTON CYCLE SIMULATION RESULTS.

β	η_{cycle}	m_{air} **(g/s)**	m_{fuel} **(g/s)**	W_t **(kJ/kg)**	W_c **(kJ/kg)**	W_{spec} **(kJ/kg)**
2	0.067	0.597	0.012	259.9	80.2	179.7
3	0.095	0.445	0.009	394.2	149.2	200
4	0.108	0.404	0.008	478.8	205.3	273.5
5	0.115	0.393	0.007	538.5	253.2	285.3
6	0.118	0.395	0.007	583.5	295.5	298
7	0.118	0.404	0.007	619.0	333.4	285.3
8	0.117	0.419	0.007	647.9	368.0	279.9
9	0.116	0.433	0.007	672.2	397.1	275.1

From a strictly thermodynamic point of view, the optimal compression ratio is seen to be in the neighbourhood of β=7. To attain a simple back-to-back radial compressor/radial turbine configuration, we imposed though β=2. TABLES II and III report a comparison between the corresponding cycle performances.

TABLE II. SIMULATION RESULTS FOR β=2.

	m	**p**	**T**	**h**	**ex**	**O2**	**N2**	**CO2**	**H2O**
	(g/s)	**(kPa)**	**(K)**	**(kJ/kg)**	**(kJ/kg)**	**Weight fraction**			
point 1 (*Figure 3*)	0.597	101.3	288	30.36	0	0.23	0.757	0.0005	0.012
point 2	0.597	202.6	376	110.4	68.9	0.23	0.757	0.0005	0.012
point 3	0.609	196.5	1600	2277.5	1302.9	0.063	0.742	0.00049	0.193
point 4	0.609	101.3	1435	2017.6	1029.6	0.063	0.742	0.00049	0.193
$P_{turbine}$**(kW)**	0.150341								
$P_{compress}$**(kW)**	0.050341								
P_{out}**(kW)**	0.1								
Fuel(kg/s)	0.000012								

TABLE III. SIMULATION RESULTS FOR β=7 (OPTIMAL η).

	m	p	T	h	ex	O2	N2	CO2	H2O
	(g/s)	(kPa)	(K)	(kJ/kg)	(kJ/kg)	Weight fraction			
point 1	0.397	101.3	288	30.3	0	0.23	0.758	0.0005	0.012
point 2	0.397	709.1	617	363.7	275.9	0.23	0.758	0.0005	0.012
point 3	0.403	687.8	1600	2162.7	1370.0	0.089	0.744	0.000491	0.165
point 4	0.403	101.3	1178	1538.7	697.3	0.089	0.744	0.000491	0.165
$P_{turbine}$(kW)	0.239172								
$P_{compress}$(kW)	0.139172								
$P_{out(}$kW)	0.1								
Fuel(kg/s)	0.000007								

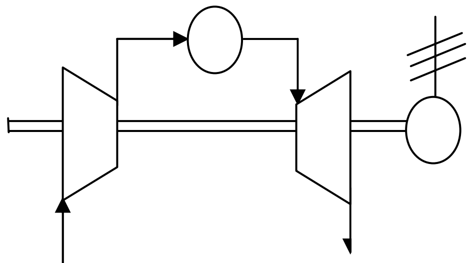


Figure 3. Layout of the simple Brayton cycle configuration.

3.2 The intercooled Brayton cycle

Our initial assumption is similar to the previous one, but in this case (N being the number of compressor stages)

$$\beta_{stage} = (\beta_{opt})^{1/N} \qquad (7);$$

$$\Delta p_{intercooler}=2\%\ p_2$$

$$T_3 > T_1 \text{ (see figure)}$$

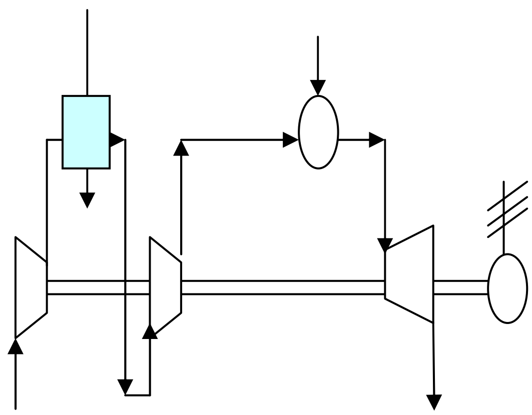


Figure 4. Layout of the intercooled system.

TABLE IV. INTERCOOLED BRAYTON CYCLE SIMULATION RESULTS.

β	**η_{cycle}**	**m_{air}**	**m_{fuel}**	**W_t**	**W_c**	**W_{spec}**
		(g/s)	**(g/s)**	**(kJ/kg)**	**(kJ/kg)**	**(kJ/kg)**
2	0.064	0.617	0.013	252.6	78.6	174
3	0.092	0.439	0.009	388.9	141.9	247
4	0.107	0.385	0.008	475.6	191.2	284.4
5	0.116	0.362	0.007	537.3	232.1	305.2
6	0.122	0.347	0.007	585.8	265.6	320.2
7	0.123	0.348	0.007	621.6	298.7	322.9
8	0.124	0.348	0.007	652.3	326.7	325.6
9	0.119	0.426	0.008	677.2	400.0	277.2
10	0.102	0.434	0.008	699.0	423.8	275.2

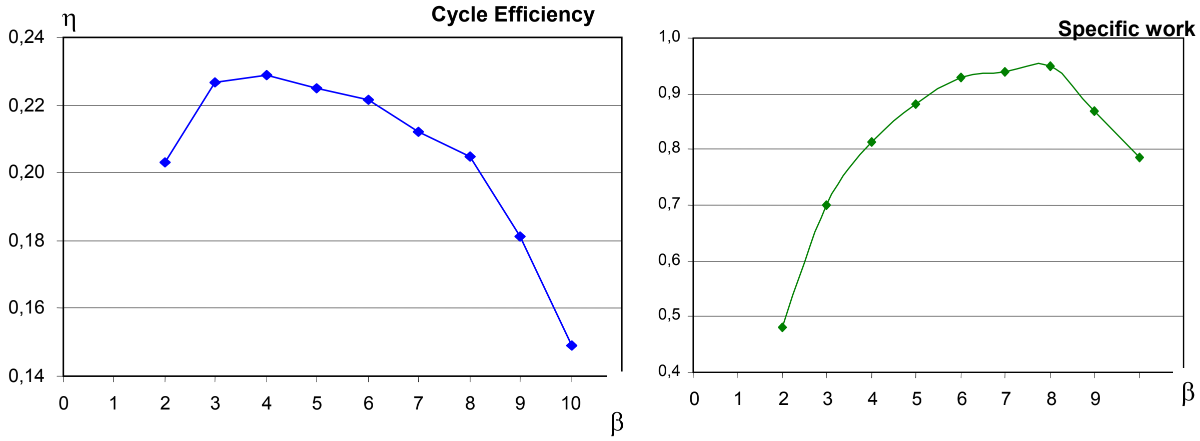


Figure 5. Dependence on the compression ratio of: a) intercooled cycle efficiency; b) specific work.

3.3 The intercooled/regenerated Brayton cycle

Here the assumptions are quite different (*Figure 6*), in particular:

$$T_7 - \Delta T_{preheater} = T_5$$
$$\Delta p_{reg} = 2\%\ p_4$$
$$p_7 = 1.02 p_{atm}$$

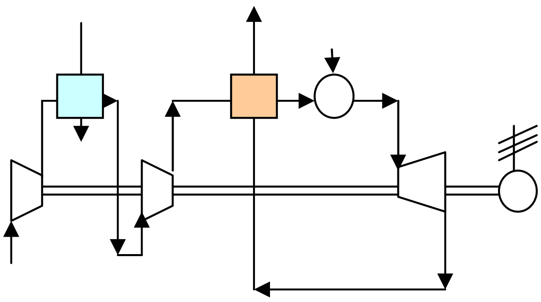


Figure 6. Layout with intercooling and regeneration.

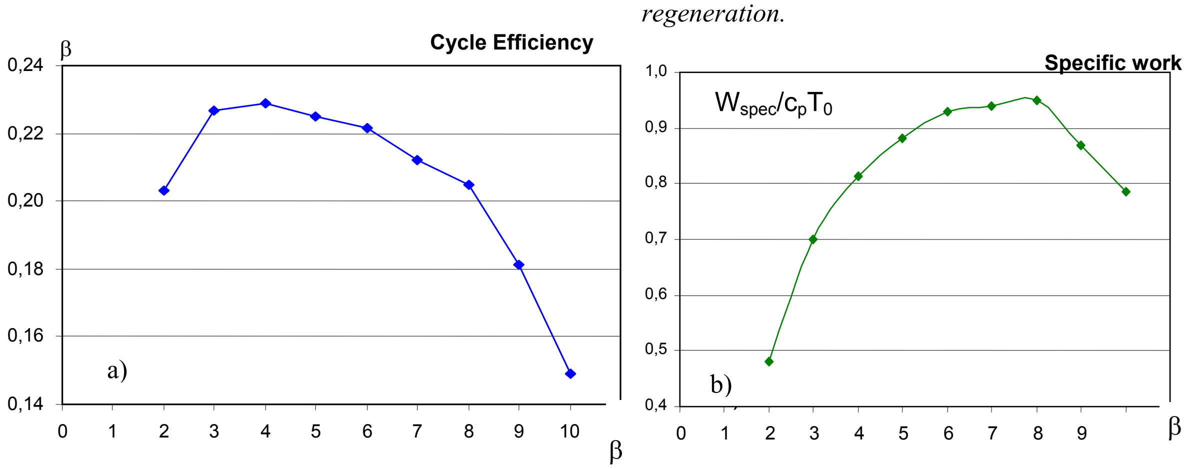


Figure 7. Dependence on the compression ratio of: a) intercooled/regenerated cycle efficiency; b) specific work.

TABLE V. INTERCOOLED/REGENERATED BRAYTON CYCLE RESULTS.

β	**η_{cycle}**	**m_{air} (g/s)**	**m_{fuel} (g/s)**	**W_t (kJ/kg)**	**W_c (kJ/kg)**	**W_{spec} (kJ/kg)**
2	0,202	0,784	0,004	220,2	78,6	141.6
3	0,226	0,546	0,004	347,7	141,9	205.8
4	0,228	0,474	0,004	430,8	191,2	239.6
5	0,224	0,442	0,004	491,0	232,1	258.9
6	0,221	0,422	0,004	539,0	265,6	273.4
7	0,211	0,422	0,004	574,9	298,7	276.2
8	0,204	0,421	0,004	605,9	326,7	279.2
9	0,181	0,539	0,005	632,2	400,0	232.2
10	0,149	0,548	0,006	655,0	423,8	231.2

TABLE VI. COMPARISON BETWEEN DIFFERENT SOLUTIONS.

Type of cycle	**SIMPLE**		**INTERCOOLED**		**INTERCOOLED & REGENERATED**	
β	**2**	**opt. $\beta = 7$**	**2**	**opt. $\beta = 8$**	**2**	**opt. $\beta = 8$**
η_{max}	0.06	0.11	0.06	0.12	0.20	0.228
m_{air} (g/s)	0.597	0.404	0.617	0.348	0.784	0.474
m_{fuel}(g/s)	0.012	0.007	0.013	0.007	0.004	0.004
w_{spec} (kJ/kg)	179.7	285.3	174	325.6	141.6	279.2

TABLE VI reports a synopsis of the results of the preliminary configuration study:

a) there is an advantage in increasing the β_c, in terms of fuel consumption and efficiency;

b) there is an advantage in increasing η_{pc} and η_{pt} in terms of thermo-mechanical stress of the components,

c) with the "optimal" settings, in each configuration studied, the mass flow rate for a 100 W device is about 0.0004 kg/s and therefore prefigures an overall external radial size of about 30 mm.

As mentioned above, though, the implementation of a multi-stage compressor and expander at a nanoscale is not immediately feasible; in the following, therefore, we shall maintain the b=2 constraint, and size the machine accordingly.

3.4 A brief summary of Camel® processing software

CAMEL (Cycle Analysis by Modular ELements) is a modular process simulator conceived and implemented in the last decade by the Mechanical and Aeronautical Department of the University of Roma1, "La Sapienza".

The code is devoted to the simulation of all energy conversion processes.

CAMEL is equipped with a user-friendly graphical interface and some post-processing utilities to allow the user to perform exergy, thermo-economic and extended analysis on any process configuration for which a mass and energy balance has led to a converged process solution.

The code is based on a completely object-oriented approach, and it is entirely written in C++. It includes specific libraries containing functions and subroutines for evaluating physical, chemical and thermodynamic properties of all working media involved in the process (water, steam, brine, technical gas or different kinds of fuels).

The code modularity allows to "construct" any desired plant configuration by "assembling" on the screen the components included in the code library; the only requirement is that the user-constructed configuration does not violate elementary physical principles (in which case the code will respond with a proper error message). If a particular new process requires additional components, they can be easily implemented making use of the specific utilities attached to the code.

On the basis of the user-assembled configuration the code automatically constructs an interconnection matrix and solves the proper equation system.

This global system of equations contains both balance equations for mass and energy and the proper thermodynamic relations, and its solution describes the thermodynamic changes that each component operates on each stream. In mathematical terms, this equation system is not a closed one, and it therefore needs boundary - and initial conditions in terms of known flow parameters. The solver is based on an optimized Newton-Raphson algorithm .

4. Mechanic and Structural Scaling

While the thermodynamic phenomenological aspects are invariant down to this scale, the mechanical ones are not. The fluid mechanics, for example, are scale-dependent (Capata and Sciubba, 2004; Hepstein, 2003; Hepstein, 1997), and in particular it is well known that viscous forces are more important at small scales. Pressure ratios of 2:1 to 4:1 per stage imply turbomachinery tip Mach numbers that are in the high subsonic or supersonic range both in radial compressors and in radial and axial turbines. Airfoil chords of the order of a millimetere force, for instance, the compressor to operate at Reynolds numbers in the order of the tens of thousands. With higher gas temperatures, turbines of similar size will operate at a Reynolds number of a few thousands.

These are very small values compared to the 10^5-10^7 range of large-scale turbomachinery, and viscous losses will be correspondingly larger. But viscous losses make up only about a third of the total fluid loss in a high speed machine (tip leakage, shock wave losses etc. add to them) so that the decrease in machine efficiency with size is not necessarily as relevant as Equation 6 above would seem to indicate. Higher viscous forces also mean that fluid drag in small gaps and on rotating disks will be higher. For future reference, it ought to be noted that unless gas flow passages are smaller than one micron, the fluid behaves as a continuum .

Heat transfer is another aspect in which the phenomenology may be different than in large-scale machines. The fluid temperatures and velocities are the same but the viscous forces are larger, so the fluid film heat transfer coefficients are higher by a factor of about 3 (Hepstein, 2003; Peirs et al., 2003). Thus, the temperature gradients within the structure are reduced. This is helpful in reducing thermal stress but makes thermal insulation critical. For what the structural design is concerned, the dependence of material properties on the length scales becomes an important parameter. Here, differences between mechanical design and material properties begin to blur. Thus, elastic, plastic, heat conduction, creep and oxidation behaviours do not change, but fracture strength may show a substantial increase. Material selection is influenced both by mechanical requirements and by fabrication constraints. For example, structured ceramics such as silicon carbide (SiC) and silicon nitride (Si_3N_4) have long been recognised as attractive candidates for gas turbine components due to their high strength, low density and good oxidation resistance. However, their use has been limited so far by the lack of technology to manufacture flaw-free material in sizes large enough for conventional engines. Some research teams (Peirs et al., 2003) are considering the adoption of “normal” steel alloys but this design choice implies an increase in the overall dimensions of about one order of magnitude, from millimetres to centimetres. An additional material consideration is that thermal shock susceptibility decreases as component sizes shrink. Thus, materials which have very high temperature resistance but are not considered “high temperature candidates” because of their susceptibility to thermal shock (for example structural ceramics) may become viable at millimetre length scales.

5. Overview of Some Possible Nano Gas Turbine Engine Design

The main goal of this project is to show that a nano-GT set is feasible by demonstrating a prototype operation of such a device.

This implies a complete set of specifications and a certain fabrication simplicity. Most current, high precision, micro-fabrication technology applies to silicon. Since Si is highly sensitive to creep above 950 K, this becomes an upper limit for the turbine rotor entry temperature; with an impulse configuration, this means a TIT of about 1200K, which is too low a combustor exit temperature to close the engine cycle with the component efficiencies presently available; this implies the necessity of structural cooling for Si turbines.

The simplest way to cool the turbine in a millimeter-sized machine is to eliminate the shaft, and thus channel the turbine sensible heat to the compressor by conduction, then recovering a sizeable portion of it -by convection- to the fluid being compressed.

This brought some researchers (Groshenry, 1995) to consider silicon as a material of choice but new ceramics or steel alloys cannot be discounted, and this is one of the fundamental design choices, requiring extensive experimental tests on the materials at the foreseeable scales. About the fuel there are basically two possible choices: natural gas or hydrogen. Particular attention will be concentrated at this stage on the combustor, to realise a simple prototype model.

Some additional design considerations are listed here below:

a) The first problem is to select a proper machine configuration, axial or radial. Our choice is the radial one on account of the constraint of compact overall dimensions and simplicity in fabrication of the compressor and turbine. We are, though, investigating also an axial turbine configuration;

b) The second problem is the number of stages. The single stage has lower energy density and efficiency, on the other hand it is the best solution to limit the overall dimensions. The multi-stage option has a higher energy density and is more efficient, but leads to serious fabrication-related problems. At the current state of technology, for example, the construction of a three-stage nano-turbine is very difficult (Peirs et al., 2003);

c) The adoption of intercooling and regeneration is obviously the optimal solution. But the possibility of installing an intercooler has to be verified in practice, especially for its feasibility from a micro-fabrication point of view.

The first order design procedure hinted at in (Hepstein, 2003) and discussed in detail in (Capata and Sciubba, 2004) indicates that the centrifugal compressor and radial turbine rotor diameters are about 5 mm and 4 mm respectively.

The compressor discharge air wraps around the outside of the combustor to cool the combustor walls, capturing the waste heat and increasing the combustor efficiency while reducing the external package temperature. The whole architecture of the systems is similar to the "MEMS" proposed in (Hepstein, 2003). The rotor radial loads are supported by a journal bearing on the periphery of the compressor. A thrust balance piston behind the compressor disk supports the axial loads (Capata and Sciubba, 2004).

We are considering the use of the hydrostatic journal bearings as the US research group has done. Similar considerations suggest that the design peripheral speed of the compressor is 400 m/s so that the rotation rate is only slightly lower than 160000 rad/s (Capata and Sciubba, 2004; Hepstein, 1997).

A possible cutaway of an "engine chip" is shown in *Figure 8*.

As an indication of how far from technological maturity the whole field is, consider though that an entirely different system configuration can be proposed that is absolutely realistic and just as feasible as the one shown in *Figure 9*. Suppose that a thermodynamic analysis of the cycle has indicated β=8 as an optimum. Such a pressure ratio can be attained only with a multistage radial compressor: if 3 stages are used, and account is made for the unavoidable pressure losses, we can predict that the expansion ratio across the turbine is about 7.5, which is clearly too high for a single stage radial expander. But such an expansion ratio can be easily accommodated by a 3-stage axial turbine.

This configuration has the additional advantage that, if the similarity laws valid for large-scale machines approximately hold at nano-scales, for a given mass flow rate the axial turbine stages can be designed with lower tip diameter than the radial ones, thus reducing both the radial dimensions and the material stress at high temperature.

By this example, we do not want to imply that β_{opt}=8, nor that 3-stage configuration is an "optimal" one in any sense; we just want to stress that the design of nano-GT devices must be approached with an open mind, and that experience gained on "established" large-scale configuration cannot be directly translated into the design of nano-scale machines.

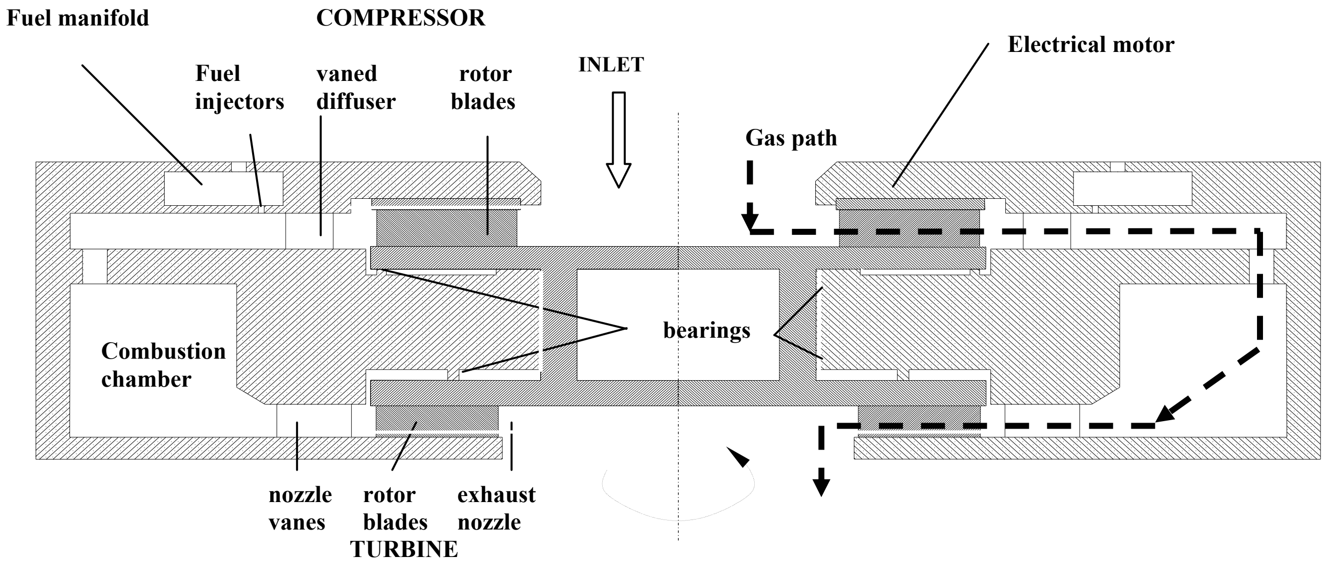


Figure 8. A possible radial/radial, single stage back-to-back GT set configuration.

6. Conclusions

One approach to realising devices at these scales utilizes industry-derived and rather well demonstrated micromachining technology. The economic impact of these devices will strongly depend on the attainable performance levels and on the manufacturing costs, both of which have yet to be proven. It is certainly possible, however, that nano-GT sets may one day be competitive with conventional machines when 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[3]. Competing with a 50-60% efficient central power plant clearly requires much higher

[3] Consider that the average conversion efficiency (in terms of the primary fuel consumed to generate the electricity that recharges the battery) of a battery system for a 200W electronic device is lower than 0.25. And nano-fuel cells, whose practical feasibility must still be demonstrated, have an overall calculated efficiency of about 0.3 (US Environmental Protection Agency-EPA 2005).

performance levels. Even excluding cogenerative use of the waste heat, a local nano-gas turbine must have an efficiency of 20%, a level which looks futuristic at this time. Should however these levels be approached, then the possible redundancy, the extreme quietness and high power density may make an array of millimetre-scale machines an attractive solution. Emergency power installations, which do not require as high a performance as base power ones, can be an attractive application for these small machines if their capital costs are sufficiently low. One advantage that millimetre-scale gas turbines can offer for many applications is that they are in fact very quiet. This stems from their high frequencies and short length scales. The high-pitch sound that they produce is relatively easy to muffle and is quickly attenuated by the surrounding environment. Another obvious application is that of using stacks of nano-GTs as a thermal engine in a hybrid vehicle. The drawback is that nano-GT sets are yet extremely expensive to develop. Though these costs may be amortised over many units, manufacturing cost is still a major issue for mass-produced devices, even after (and if!) a satisfactory performance of a prototype application is demonstrated.

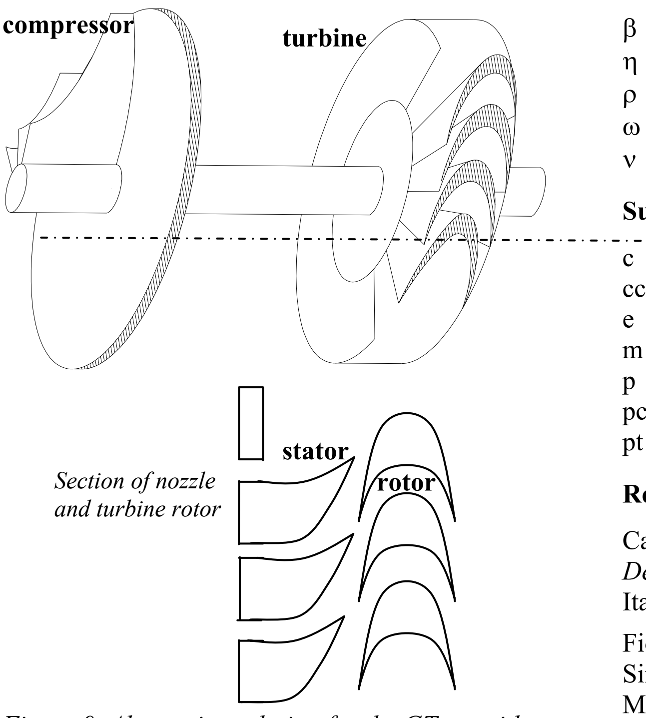


Figure 9. Alternative solution for the GT set with radial compressor and axial turbine stage.

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Nomenclature

CC	combustion chamber
D	diameter [m]
ex	exergy [J/kg]
GT	gas turbine
h	entalpy [J/kg]
k	isentropic coefficient
L	equivalent lenght [m]
m	mass flow rate [g/s]
MEMS	Micro Electro Mechanical Systems
N	number of compressor stages
n	rpm
P	power [W]
p	pressure [Pa]
RE	regenerator
Re	Reynolds number
T	temperature [K]
TIT	turbine inlet temperature [K]
TOT	turbine outlet temperature [K]
U	rotor tip velocity [m/s]
V	volume [m^3]
W	specific work [J/kg]

Greek letters

β	pressure ratio
η	efficiency
ρ	density [kg/m^3]
ω	rotational speed [rad/s]
ν	kinematic viscosity

Subscripts

c	compressor
cc	combustion chamber
e	expansion
m	mechanical
p	polytropic
pc	polytropic compression
pt	polytropic expansion

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