Internal Heat Mixing and External Heat Losses in an Ultra Micro Turbine

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ABSTRACT

The thermodynamic operation of an ultra micro turbine is developed and applied in different situations in this tiny engine where heat transfer is dominant. The first results obtained enlighten the energetic behavior and give some rules to enhance the performances.

INTRODUCTION

Since 1994, important studies are undertaken on micro engines for portable systems and micro drone propulsion. They were performed first in the U.S.A. (Epstein, 1997), (Cadou, 2002), mainly under the sponsorship of the D.A.R.P.A.. More recently, research on ultra micro turbine was also launched in Japan (Isomura, 2002), (Matsuo, 2002). At ONERA, in France, a watch over was conducted under the sponsorship of the D.G.A. (Armament General Delegation) to evaluate the potential performances of the different micro engine concepts.

Much literature is now available on this subject mainly offered by the M.I.T. (Spadaccini, 2002). It concerns the materials, the MEMS fabrication, the combustion, the internal aerodynamics, the gas bearings….in a micro turbine concept. Nevertheless none of these papers describes the overall thermodynamic behavior of these tiny engines. With the scaling down of the micro turbine, the ratio surface over volume increases, the Reynolds number decreases, the convective heat transfer coefficients increase and the mean Biot number, for materials such as SiC, is less than 0.1. All these tendencies reinforce the heat transfer which mainly constitute a thermodynamic irreversibility.

The main goal of this paper is to enlighten this particular thermodynamic operation where the hot and cold sources are so close. In particular the external heat losses produced by the free convection and the radiation towards outside are calibrated and the negative effect of inside heat mixing as mainly a result of the thermal short cuts in the stator material is described.

SEPARATION BETWEEN THE AERODYNAMIC INTERNAL LOSSES AND HEAT MIXING.

As the flow evolution in the compressor and in the turbine is no more adiabatic, one may apply the generalized relation between the pressure ratio and the total temperature ratio :

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$$\frac{T_{i2}}{T_{i1}} = \left(\frac{P_{i2}}{P_{i1}}\right)^{\left(\frac{\gamma_{air}-1}{\gamma_{air}\,K}\right)}$$

where $K = \eta_{pol\ comp}/(1+\lambda)$ for compressors

and $K = 1/(\eta_{pol\ turb} \times (1-\lambda))$ for turbines;

η_{pol} represents the polytropic efficiency for adiabatic flows and λ the fraction of heat transferred to the flow when compared to the mechanical power (compressor or turbine). This approach, which assumes that the aerodynamic losses do not change with heat transfer, has the virtue to separate the heat transfer and aerodynamic losses which both influence the total pressure rise.

MICRO TURBINE THERMODYNAMIC MODEL

Isotemperature of the micro turbine

As the mean Biot number in the engine is of the order of one percent, for a material like SiC, the hypothesis of constant temperature in the material is retained, with exception for the shaft. So we consider that there are three levels of temperature, one for the compressor rotor, the second for the turbine rotor and the third for the stator.

Heat transfer in the micro turbine

The fig. 1 and 2 present the scheme of the different convective and radiative heat transfer between the gas (or outside) and the material.

Heat transfer stations. The heat transfer considered here are:
1/ outside ⇒ stator, 2/ fluid ⇒ stator, mixing channel,
3/ fluid ⇒ stator, combustion chamber,
4/ fluid ⇒ stator, vaned diffusor,
5/ rotor/stator ⇒ rear compressor disc,
5'/ rotor/stator ⇒ rear turbine disc,
6/ fluid ⇒ stator, I.G.V.,
7/ rotor/stator ⇒ front compressor disc,
8/ fluid ⇒ rotor, compressor rotor,
9/ fluid ⇒ rotor, turbine rotor,
10/ fluid ⇒ stator, turbine stator (rotor shroud),
11/ fluid ⇒ stator, vaneless diffusor,
12/ fluid ⇒ stator, vaneless space, upstream I.G.V.,

13/ fluid ⇒ stator, vaneless space, upstream of the turbine rotor.

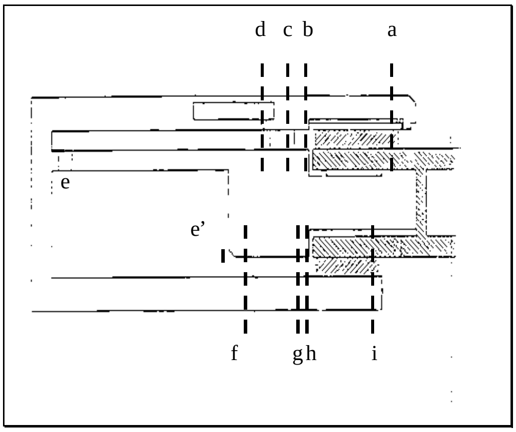

Fig.1 Scheme of the microturbine of the MIT type with useful stations for heat transfer

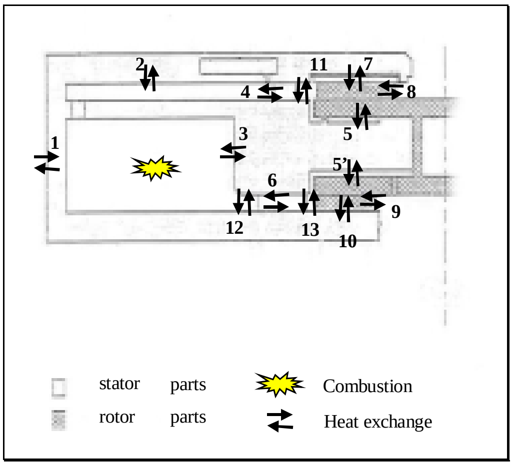

Fig.2 Main heat transfer numbers

Thermodynamic flow model and calculation chart

At first the air mass flow, the pulsation and the geometry are given. Then the heat exchange surfaces are evaluated. At a first step, the aerothermal conditions of the flow are calculated in the sections listed in fig. 1, assuming an adiabatic evolution in the compressor and in the turbine. This allows the heat transfer coefficients initialisation.

A set of equations describing the thermodynamic balance, for the stator, the compressor rotor, the turbine rotor, the shaft, the main flow volumes between adjacent sections, the rotor/stator cavity flows, is then laid.

The unknowns of the system are here the temperatures T_{stator}, $T_{rotor\ comp}$, $T_{rotor\ turb}$, the total temperatures of the flow in the considered sections and the fuel mass flow rate. This equation system is then linearised by initialization of an assumed value of T_{stator} in the radiative heat transfer formula. The system is then inverted and the linearisation loop is on until convergence. The heat transfer rates are then performed and the compressor pressure ratio and the turbine expansion ratio are compared. If necessary the tangential flow velocity at the turbine rotor inlet is updated. After convergence the aerothermal conditions (ρ, p, T, V) of the fluid and the heat transfer coefficients are updated. The convergence test on the external loop is operated on the heat transfer coefficients. After overall convergence, the aerothermal conditions, the heat transfer rates and the power (compressor power, turbine power, output power) are obtained. A corresponding calculation software based on this chart was developed (*hot button* software).

SOME APPLICATIONS OF THE "HOT BUTTON" SOFTWARE

First results

The compressor and turbine polytropic efficiencies were taken equal to 0.7 and 0.6. The comparison between the adiabatic operation of the micro turbine and the non adiabatic one shows us that the heat transfer lower the useful power from 13.4 to 4.7 Watts. This result reinforces the interest of the present study. On the other hand the influence of the maximum gas temperature at the turbine inlet appears to be moderate (between 1500 and 1700K).

Doubling the size of the micro turbine

The size of the micro turbine was multiplied by two in order to recover a correct power. The design air mass flow was multiplied by 4 (m=0.4g/s) and the rotation speed was divided by 2 ($N = 1.2510^6$ rpm). With this new size, the useful power reaches 21.7 Watts.

Disc friction losses

When doubling the size of the micro turbine, the disc friction power (Frechette, 2001a) is only 18% of the net power instead of 66% for the original configuration.

If one can optimize the rotor/stator gaps of the main discs, the axial gap of the electric generator must be as small as possible in order to obtain a good electric efficiency. The fig.3 gives the evolution of the net power as function of the axial electric generator gap. For the non insulated configuration the net power is very low for a gap less than 10 µm. This means that the use of an electrostatic generator (Frechette, 2001b), (Köser, 2001) leads to a severe penalty in performances.

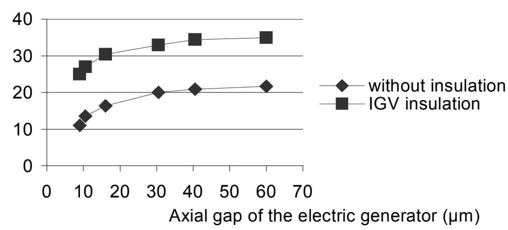

Fig. 3 Net power (Watts) versus axial gap

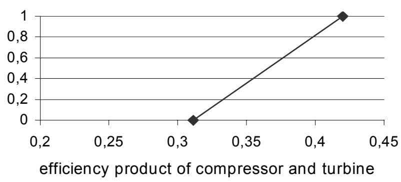

Fig.4 Net non dimensional power

Aerodynamic efficiency influence

The product $\eta^2 = \eta_{pol\ comp} * \eta_{pol\ turb}$ of the polytropic efficiencies has a strong influence on the net power even in a non adiabatic evolution. The fig.4 shows the evolution of the net power versus this aerodynamic loss parameter, the reference non dimensional power being obtained for $\eta^2 = 0.42$. For $\eta^2 = 0.31$ the micro turbine gives no more power.

HEAT TRANSFER PARTIAL INSULATION

In the hot button software one can insulate a component of the micro turbine by putting the corresponding heat transfer coefficients equal to zero. This methodology is interesting to understand the heat transfer organization and its influence on the performances. It is also useful to know which parts of the micro turbine must be insulated in priority in order to recover a part of the thermal efficiency. Some results of this parametric study are presented on fig. 5.

The calculations are performed in two situations. In the first one, $T_{rotor\ comp} = T_{rotor\ turb}$, configuration where the main disc is common to the compressor and the turbine or when the conduction in the shaft is dominant. In the second one, $T_{rotor\ comp} \neq T_{rotor\ turb}$, the heat transfer in the shaft is governed by its heat resistance.

The insulation of the burner alone is non operant, this is because the heat transfer is transferred in the turbine and in a large part in the I.G.V.. On the contrary the insulation of the I.G.V. appears to be much efficient. In this case the net power increases from 21.7 to 35 Watts ($T_{rotor\ comp} = T_{rotor\ turb}$) that is a 61% rise. The insulation of the compressor is also efficient but it seems practically difficult to manage it both on the rotor and on the stator.

When the heat transfer between the turbine rotor and the compressor rotor is driven by the thermal resistance of the shaft, a moderate gain in net power is obtained. But this apparent gain is cancelled by the additional disc friction losses occurring when three discs operate instead of two (two for the compressor rotor and one for the turbine rotor).

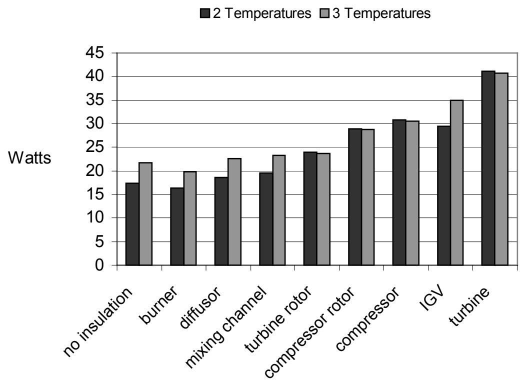

Fig.5 Net power for partial thermal insulation

EXTERNAL HEAT LOSSES

In order to try to enhance the performances of the micro turbine we have studied the benefit effect of a heat shield with a thickness of 2 mm (fig. 6).

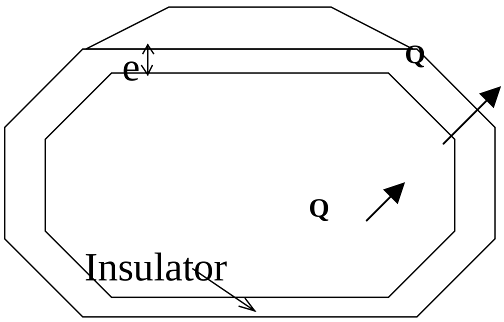

Fig. 6 Insulator arrangement around the microturbine

We performed a simple calculation assuming that, as a first approximation, the temperature of the stator is not changed. This is not really true because some insulation rises the temperature of the stator but this gives a good order of the magnitude of the temperature difference between the stator (SiC or Si) and the outside skin. This simple model, independent of the “hot button software”, satisfies steady heat flux conditions at the boundaries of the coating, with a plane or a spherical conduction scheme.

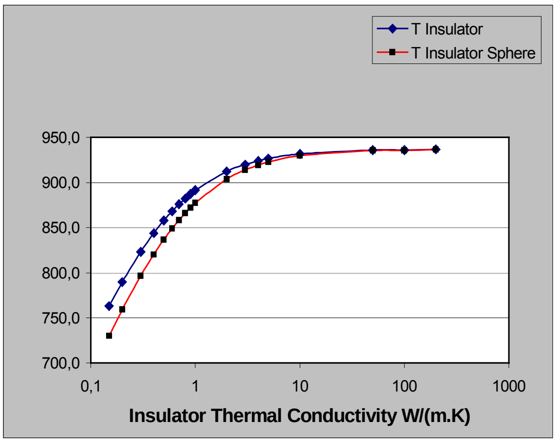

Fig. 7 Insulator temperature function of the thermal conductivity (µturbine ϕ = 1 cm, emissivity = 0.87)

For a 2 mm insulator, the material is efficient if its thermal conductivity is less than 1 (fig.7). In particular the conductivity of SiO_2 which is about 2 at high temperature seems too important except if we decrease the emissivity by an appropriate surface deposit.

A second method to know the evolution of the performances of the micro turbine, is to use “the hot button” software by studying the influence of the emissivity parameter of the outside skin (fig.8). Even if the results show some oscillations in convergence, we can estimate that the maximum gain in net power is about ten percent.

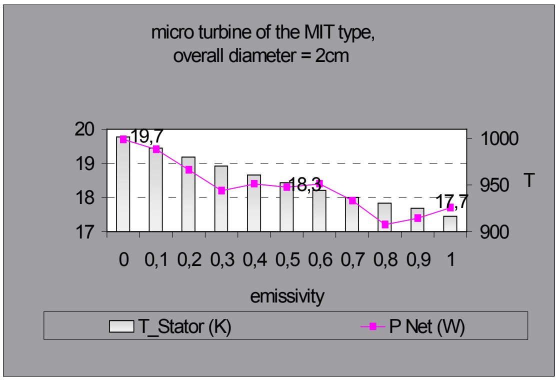

Fig.8 Emissivity influence on the performances

CONCLUSION

A thermodynamic model, taking into account strong heat transfer in an ultra micro turbine, was developed and applied in different situations.

The results show that the non adiabatic operation of the turbomachines leads to a strong penalty in performances. Doubling the size of the micro turbine allows to recover a suitable power.

The heat insulation of the I.G.V. allows to recover a large part of the lost power.

The insulation of the combustion chamber alone does not work. This means that the good efficiency to maintain in this component is the chemical efficiency.

The dramatic influence of the compressor and turbine efficiency is confirmed in a non adiabatic operation.

The use of a low conductivity material for the stator would improve by a large amount the performances.

The external radiative heat losses are estimated to ten percent of the nominal power for the 2 cm diameter microturbine. A thermal shield of 2 mm thickness is a good way to enhance the performances and is certainly a necessity for integration.

A better knowledge of the heat transfer coefficients at low Reynolds numbers in the different components of the micro turbine has to be obtained to refine the results of this study.

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