A MICRO CHP SOLUTION BASED ON A MICRO GAS TURBINE

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Abstract: This paper presents early development of a μ-CHP solution based on a micro gas turbine (1-5kWe). The thermodynamic analysis performed shows the importance of reaching a high TIT and the sensitivity of the component's efficiencies on the global efficiency. The development of micro-turbo machinery is a challenging task. The option taken here is to use a multi-objectives and multi-disciplinary optimization chain including a highly flexible geometrical modeler. Finally a heat exchanger with very fine tubing is developed using software coding and CFD to increase the TIT and improve drastically the global efficiency.

Keywords: micro gas turbine, centrifugal compressor, heat exchanger

NOMENCLATURE

π	Pressure ratio
η	Efficiency
ε	Heat exchanger thermal efficiency
TIT	Turbine Inlet Temperature
β	Blade angle
θ	Angular coordinate

SUBSCRIPT

C	Compressor
T	Turbine
HEX	Heat exchanger
is	Isentropic
mec	Mechanical
hot	Hot side
cold	Cold side

INTRODUCTION

Decentralized electricity production in small buildings (home, small apartment buildings, office buildings, ...) with the requested combined heat production offers a solution where the electricity is free of charge for the user of the heat (or cold) production. This μ-CHP solution (Micro Combined Heat and Power) is thus very attractive if the technology delivering this solution is affordable from a cost point of view (1 k€ per kWe) and is answering to all operational requirements of a domestic use (low noise, low vibrations, small volume, low mass, quick start, safe, ...).

The current domestic μ-CHP solution on the market uses a Stirling motor. The solution is a reliable mature technology but it does not respect the buyer's requirements. The heat delivery is indeed too high for the requested electricity production (1-1.3 kWe) especially for passive houses but mainly the cost cannot be largely reduced due to the high mass and volume of the system. Two other heavy disadvantages of the Stirling motor are the slow start (4-5') and the obligation to work at a given rate at all times (no flexibility in the CHP production).

To solve these problems, we propose a solution based on a micro gas turbine (μ-GT) burning natural gas.

This very compact μ-CHP is also intended to deliver 1kWe but with an electrical efficiency of up to 30% and thus a heat production of about 2.2 kW corresponding to the requirements of an average house. In fact, the direct recycling of the heat generated by the power plant for heating purposes justifies this decentralized solution, together with other key necessities for in-house applications such as flexible electricity generation, high compactness, very low noise level, high reliability and potential low cost.

THERMODYNAMIC CYCLE ANALYSIS

The development of a new engine starts with the thermodynamic analysis. This work gives a prediction of the global performances as well as the requirements of each component. We perform the analysis with the commercial software GasTurb10. The engine configuration is a one spool turboshaft engine with a recuperator (Fig. 1).

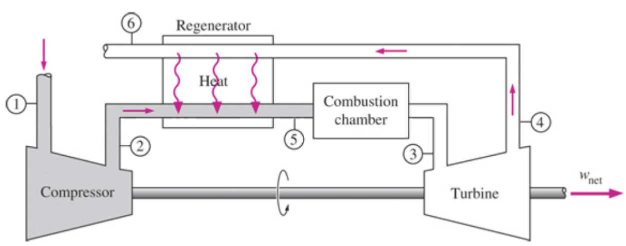

Fig. 1 Recuperated gas engine cycle

First, a state of the art study was performed in order to fix the performances of each component. Based on this database, the on-design analysis concludes that the use of advanced technologies would lead to 33% of global efficiency versus only 22% with current technologies. Secondly, the influence of the cycle parameters was tested. The baseline values are given in Table 1.

Parameters	Values
π_C	3
TIT (C)	1200
η_{isC}	0.75
η_{isC}	0.8
ε	0.9
$\pi_{HEXcold}$	0.95
π_{HEXhot}	0.95
η_{mec}	0.95

Table 1 Baseline value for the parametric study

Figure 2 shows the influence of the working point on the global efficiency. In practice for small centrifugal compressors, the pressure ratio is limited to 3 which nearly corresponds to the optimal value. The figure also explains the interest of authors [8][10] for high temperature materials like ceramics for the turbine wheel's manufacturing.

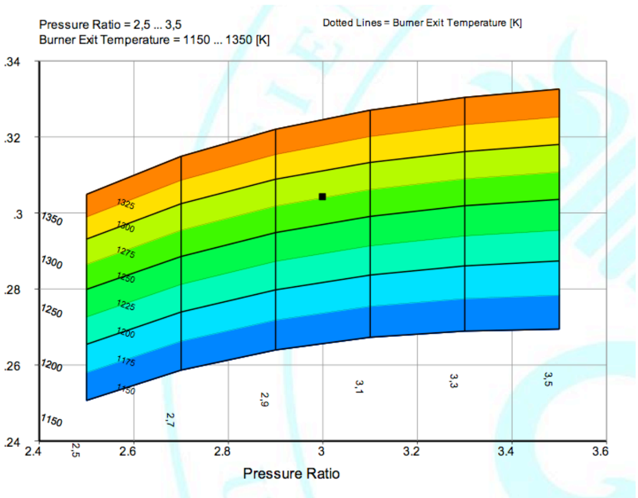

Fig. 2 Influence of the GT working point

The efficiency of the compressor is a key parameter because it deeply influences the global efficiency. For example, a 5% drop in the compressor efficiency results in a 6% drop in the global efficiency. This represents a challenge since a good efficiency is hard to obtain because of the small size of the compressor. The turbine efficiency has a similar effect on the global efficiency.

The addition of a recuperator creates a performance peak that does not exist on the simple cycle (Fig.3). The increase in the recuperator thermal efficiency creates a sharper peak and pressure losses move the peak to lower pressure ratios. In the design of a heat exchanger, there is no net advantage in optimizing either the pressure losses or the thermal efficiency since they both have roughly the same effect on the global efficiency. T. Stevens [5] has isolated that the optimal cold to hot pressure losses ratio is the square root of the compressor pressure ratio.

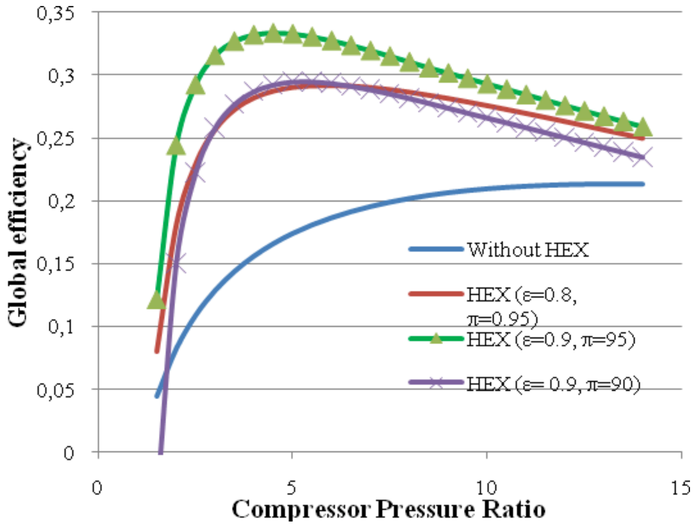

Fig. 3 Global efficiency versus pressure ratio for different HEX

CENTRIFUGAL COMPRESSOR MODELER

A highly flexible geometrical modeler has been developed to be included into an optimization chain. It includes modern features like splitter blades and main blades with lean and rake angles. For this application, the parameterization must be as flexible as possible to allow for the generation of innovative geometries and sufficiently bullet-proof to generate feasible geometries.

First, the meridional contours are defined by 4^{th} order Bezier curves, in red on Figure 4. The displacement of the second and the fourth points are restricted to lines joining the extreme points. Incidentally the curve is tangent to both lines at the tips. The middle point's position is restricted by the concavity of the curve. Secondly, the camber line's circumferential position θ is defined by equation 1 and allows for the transformation from the β distribution on Figure 5 to the 3D camber line. Finally, the thickness is added to fix the blade geometry.

$$R d\theta = dm \tan(\beta) \qquad (1)$$

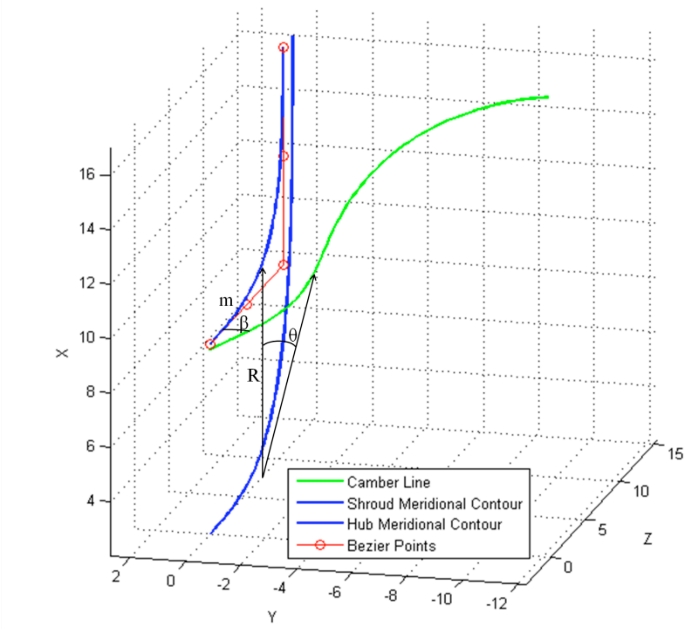

Fig. 4 Blade camber line construction

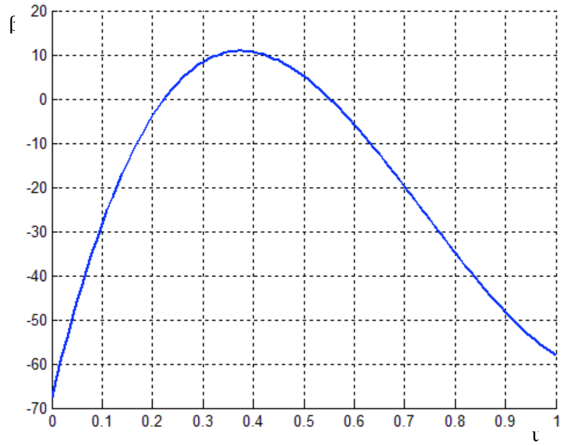

Fig. 5 Evolution of the flow angle beta

HEAT EXCHANGER

Incorporating a heat exchanger (Fig. 6) that recuperates heat from the exhaust gases to increase the TIT into the cycle triples the efficiency of the otherwise uncompetitive micro CHP solution and is therefore essential.

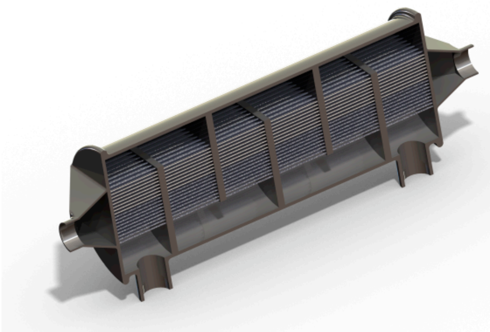

Fig. 6 Shell and tube heat exchanger

The shell and tube layout is a good compromise between both main design considerations, i.e. heat transfer efficiency and loss in air pressure. It is impossible to optimize both since they oppositely affect each other. The heat transferred by the exchanger increases the efficiency of the cycle but the loss in pressure decreases the useful power. The loss in power is not as important since the power/weight ratio is not a crucial issue as the machine remains static while in use. The best compromise is found by trying all possible geometries using a Matlab code. The calculations are done by means of the ε−NTU method; it takes into considerations all factors such as conduction, convection, laminar and turbulent flow, fouling and pressure variations. The output is an acceptable design shown in Figure 7 with the dimensions listed in Table 2.

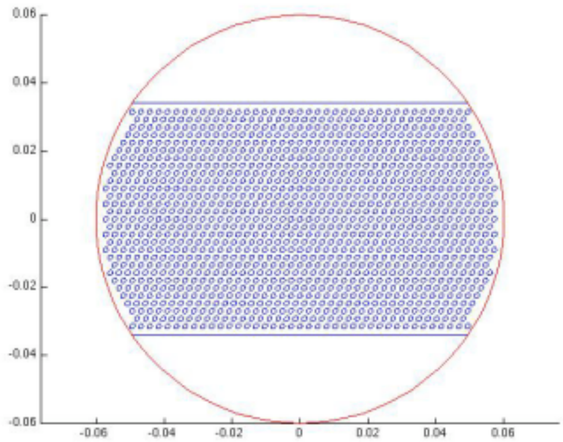

Fig. 7 HEX tube arrangement

Parameters	Values
Tube diameter	0.8 mm
Tube thickness	30 µm
Space between tubes	1.68 mm
Shell diameter	16 cm
Tube length	55 cm
Number of baffles	6
Number of tubes	3665
Velocity in the tubes	10 m/s
Velocity in the shell	8.7 m/s

Table 2 HEX preliminary design parameters

The theoretical nature of the correlations used by the Matlab parametrical study renders the results imprecise. The data obtained is in fact an estimation that provides a useful insight on the magnitude the various values should have. To achieve a precise design, a CFD simulation is done on the software Ansys Fluent. First a single tube is modeled to calculate the precise pressure loss within it under the specified dimensions and working conditions, the result is a drop of 1.7% in pressure, which is better than the Matlab prediction. Secondly a matrix of 66 tubes is modeled to analyze the heat flow and the pressure variation. A sequence of simulations is ran using the known working conditions and the results as new input values for subsequent steps in order to model the entire exchanger with the simplified version. Figure 8 shows the temperature distribution that is used to confirm the values obtained from the Matlab estimate and to determine the accurate coefficient of convection. Figure 9 represents the pressure distribution and is in agreement with the theoretical values. The new convection coefficient is used to re-run the optimization software to achieve the final design which is shortened from 55 to 40 cm and uses 5 baffles instead of 6 as shown in Figure 10. The final performances are listed in Table 3.

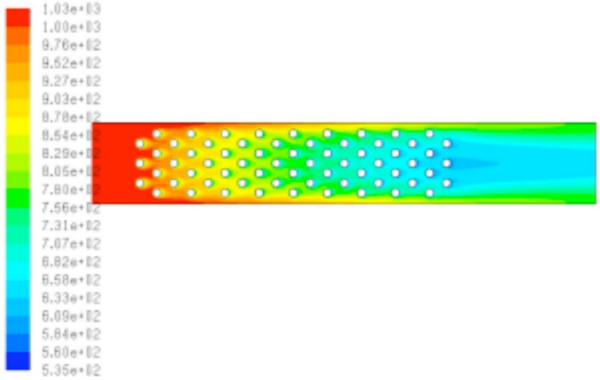

Fig. 8 HEX temperature distribution

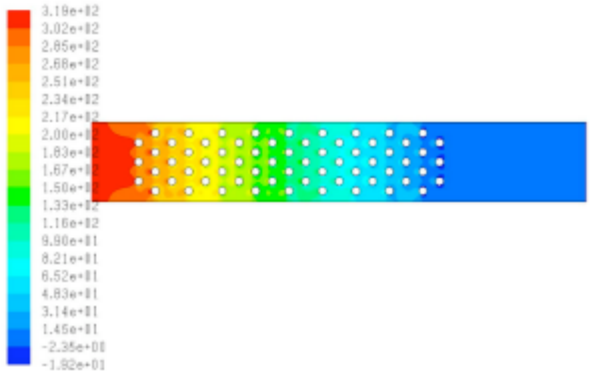

Fig. 9 HEX pressure distribution

Parameters	Values
NTU	11.3
Thermal efficiency	95.5%
Tube pressure loss	3.0%
Shell pressure loss	5.7%

Table 3 Final design performance

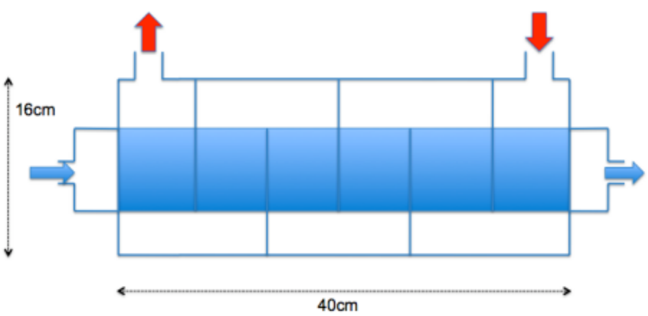

Fig. 10 Final heat exchanger layout

CONCLUSIONS

Micro gas turbine engine offers solution to reduce largely the cost and reliability of µ-CHP. Ongoing CFD simulations on the heat exchanger and optimization calculations on the compressor suggest that the micro gas turbine will be a commercially viable CHP solution for future European alternative decentralized energy production.

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