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Requirements for recuperators in micro gas turbines

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***Abstract*— Heat recuperation is a well known mean to improve the overall cycle efficiency of a standard gas turbine. Scaling a gas turbine to smaller dimensions has in general a negative influence on pressure ratio, turbine inlet temperature and pressure drops, and thus decreases the overall cycle efficiency. A thermodynamic analysis is performed to evaluate heat recuperation in micro gas turbines with respect to pressure drops and heat exchanger effectiveness as a function of gas turbine scaling. For a specific heat exchanger configuration the optimal recuperator volume and optimal channel dimensions are determined. Design requirements for the recuperator are established. This results in an assessment of suitable recuperator configurations for micro gas turbines.**

I. Introduction

Heat recuperation is often used to improve the overall cycle efficiency of standard gas turbines. In small sized gas turbines this improvement is however much more questionable. Indeed, both achievable pressure ratios and turbine inlet temperatures are significantly lower and pressure drops are much larger compared to conventionally sized gas turbines.

Thermodynamic analysis of micro gas turbines is already performed in literature to examine feasibility of target cycles in research projects [1], [2]. However, a detailed study about the influence of recuperator pressure drops and effectiveness on the overall cycle efficiency is not yet made and will be discussed in this paper.

In section II a thermodynamic analysis is performed to determine the effect of both cold and hot side pressure drops as well as the effect of heat exchanger effectiveness on the cycle efficiency of two different micro gas turbines. The MIT Brayton cycle corresponds to a 10-30 W micro gas turbine, with rotor diameter of 10 mm and a mass flow rate of 0.15 g/s [3]. The target Brayton cycle corresponds to a 100-200 W micro gas turbine presently being designed in our research project [4]. A rotor diameter of 20 mm and a rotational speed of 500,000 rpm are envisaged. The mass flow rate is 2.5 g/s. In order to assess the effects of gas turbine scaling on the overall cycle efficiency the study is extended in section III to a whole range of cycle parameters. Based on this research, first recuperator design requirements are determined.

MIT already proposed a recuperator design for use in a micro gas turbine [3]. This recuperator has an annular shape which fits well on the turbine and combustion chamber. This counterflow heat exchanger has rectangular channels placed in radial direction. The influence of this MIT recuperator on the overall cycle efficiency of both MIT and target Brayton cycle is discussed in section IV. The effect of dimensional scaling is studied in order to determine the optimal recuperator volume. Furthermore, a constant volume optimization is performed to establish the optimal channel dimensions for the 100-200 W gas turbine. This leads to alternative suitable recuperator configurations for micro gas turbines. Finally, the last section summarizes the conclusions.

II. Pressure drops and effectiveness

For the thermodynamic analysis following parameters are considered: the MIT Brayton cycle with compressor pressure ratio of 4 and turbine inlet temperature equal to 1600 K [5]; the target Brayton cycle with compressor pressure ratio of 3 and turbine inlet temperature equal to 1400 K. For both cycles the compressor and turbine isentropic efficiencies are assumed to be 61% and 65% respectively, and the relative pressure drops in combustion chamber and at the outlet of the gas turbine are chosen 3% and 0.75% respectively. The calculations are performed for a constant specific heat of 1088.8 J/kgK. The inlet conditions are 1 bar and 288.15 K.

The overall cycle efficiency of the Brayton cycle without recuperator (further denoted as the reference efficiency) amounts to 9.31% for the MIT Brayton cycle and 6.64% for the target Brayton cycle. An ideal recuperator (effectiveness of 1, no pressure drops) increases the overall cycle efficiency to 33.02% for the MIT and to 28.51% for the target Brayton cycle. The lower efficiencies of the target Brayton cycle are due to a lower pressure ratio and a lower turbine inlet temperature. A real heat exchanger, however, suffers from pressure drops and limitations in heat transfer, which decreases the overall cycle efficiency. To be useful, the recuperator should at least increase the cycle efficiency above the reference efficiency.

Figure 1 to 4 present isolines of constant cycle efficiency η as a function of heat exchanger effectiveness ϵ and cold and hot side pressure drops dp_c and dp_h. In Figure 1 and 3 all pressure drops are located at the cold side of the recuperator ($dp_h = 0$). In Figure 2 and 4 all pressure drops are located at the hot side ($dp_c = 0$). From Figure 1 it can be seen that the maximum allowable pressure drop at the cold side of the heat exchanger dp_c (dp_h=0, ϵ=1) is 130.95 kPa. The allowable pressure drop in the hot side of the heat exchanger dp_h (dp_c=0, ϵ=1) amounts to 50.40 kPa (see Figure 2). Thus, for the MIT Brayton cycle it can be concluded that the pressure drop at the hot side of the heat exchanger is more critical

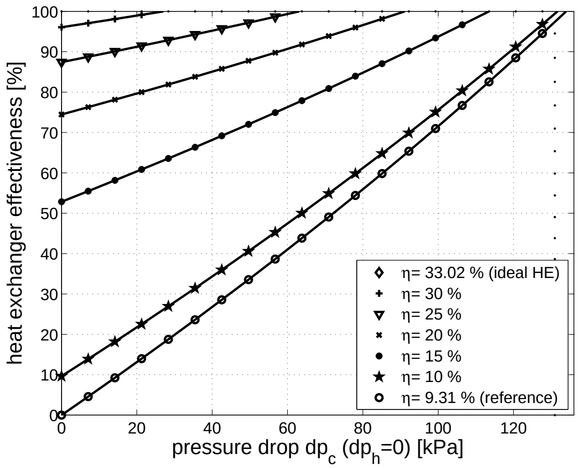

Fig. 1. Results of thermodynamic analysis for pressure drop in cold side of recuperator for MIT gas turbine (PR=4, TIT=1600 K)

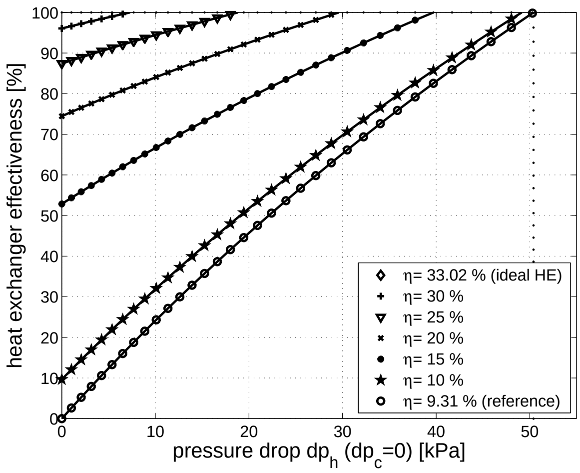

Fig. 2. Results of thermodynamic analysis for pressure drop in hot side of recuperator for MIT gas turbine (PR=4, TIT=1600 K)

than the one at the cold side. For real heat exchangers with a heat exchanger effectiveness lower than one, the allowable pressure drop to keep the same overall cycle efficiency is even lower. Indeed, a real recuperator should be located in the area left from the isoline of the reference efficiency without recuperator.

Similar conclusions can be shown for the target Brayton cycle: the maximum allowable pressure drop to hold the reference efficiency is 74.51 kPa for the cold side of the heat exchanger (dp_h=0, ϵ=1) (see Figure 3) and 32.50 kPa for the hot side (dp_c=0, ϵ=1) (see Figure 4).

Comparing the results of the MIT and the target Brayton cycle, it can be seen that pressure drops become more important for lower pressure ratio and turbine inlet temperature, because the allowable pressure drops are lower. As such, high pressure ratio and turbine inlet temperature in the development of micro gas turbines facilitates the use of a recuperator.

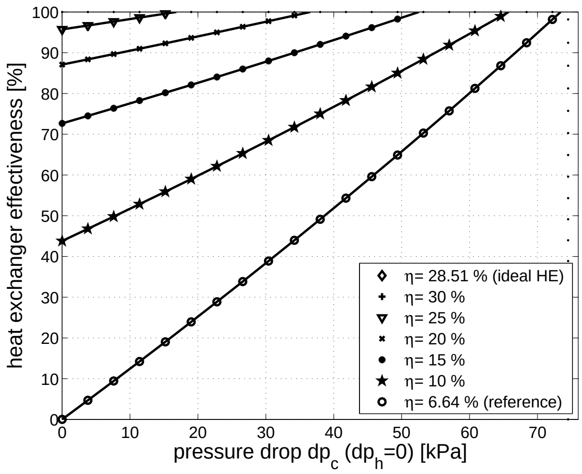

Fig. 3. Results of thermodynamic analysis for pressure drop in cold side of recuperator for the target Brayton cycle (PR=3, TIT=1400 K)

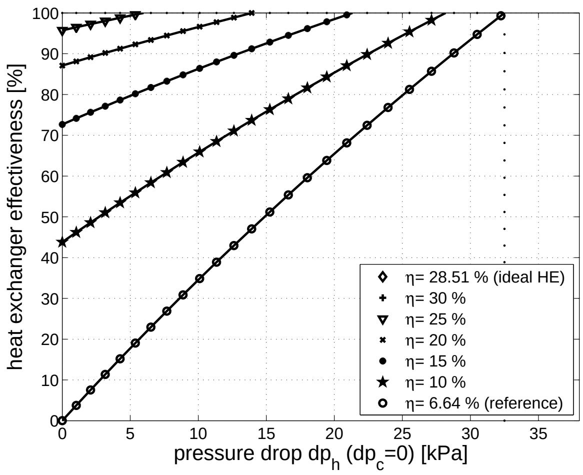

Fig. 4. Results of thermodynamic analysis for pressure drop in hot side of recuperator for the target Brayton cycle (PR=3, TIT=1400 K)

Figure 5 presents isolines of constant cycle efficiency as a function of heat exchanger effectiveness and pressure drop ratio dp_c/dp_h for a given total pressure drop of 20 kPa. From this figure it is clear that ϵ should be as high as possible in order to increase the cycle efficiency. Further more, it can be seen that the pressure drop should preferably be located in de cold channels. It should be noted that the higher ϵ, the less important becomes the difference between the pressure drops at cold and hot side of the heat exchanger.

III. General effect of gas turbine scaling

In order to assess effects of gas turbine scaling, a more general study is performed. It is assumed that pressure ratio and turbine inlet temperature decrease with decreasing gas turbine dimensions because of technical limitations. Figure 6 and Figure 7 present the maximum allowable pressure drop for a heat exchanger with ϵ=1 as a function of pressure ratio, for different turbine inlet temperatures. In Figure 6 the pressure drop is located at the cold side of the heat exchanger ($dp_h = 0$) while in Figure 7 it is at the hot side ($dp_c = 0$).

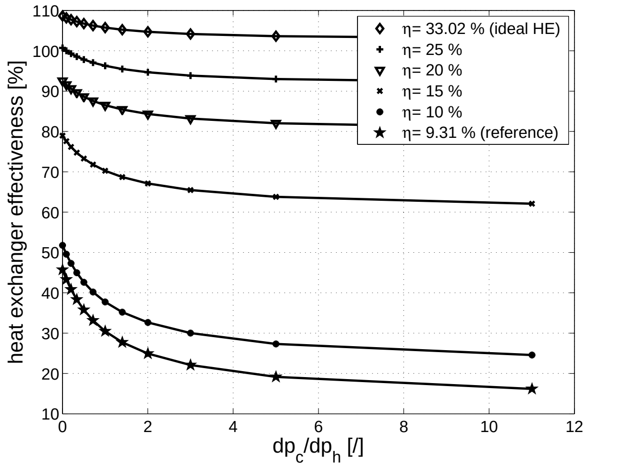

Fig. 5. Results of thermodynamic analysis for constant total pressure drop dp=20 kPa (PR=4, TIT=1600 K)

In both figures it can be seen that the smaller the compressor pressure ratio the lower the maximum allowable pressure drop for the same heat exchanger effectiveness, provided that the turbine inlet temperature is high enough. Compressor pressure ratio and turbine inlet temperature should be as high as possible. This again emphasizes the importance of high pressure ratio and turbine inlet temperature both with respect to cycle efficiency and power output. Moreover, these results are independent of recuperator volume and only based on a thermodynamical cycle analysis. Taking account of downscaling the recuperator volume, the difficulty of designing recuperators for small sized gas turbines is even more challenging. Indeed, for thermodynamical reasons the pressure drops should be smaller, while simultaneous reduction of recuperator volume will increase the pressure drop.

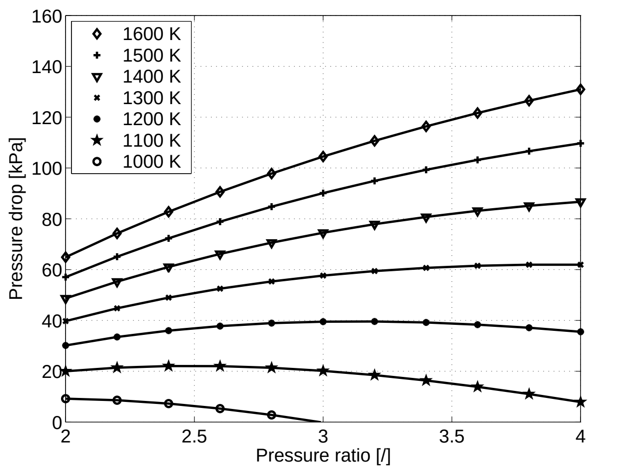

Fig. 6. Maximum allowable pressure drop dp_c for different turbine inlet temperatures ($\epsilon = 1$ and $dp_h = 0$)

These thermodynamic analysis leads to first recuperator design directives. In order to minimize the pressure drop at the hot side of the heat exchanger, the hot channels should be taken wider than the cold ones. Both cold and hot channels must be designed in a way to minimize the pressure drops.

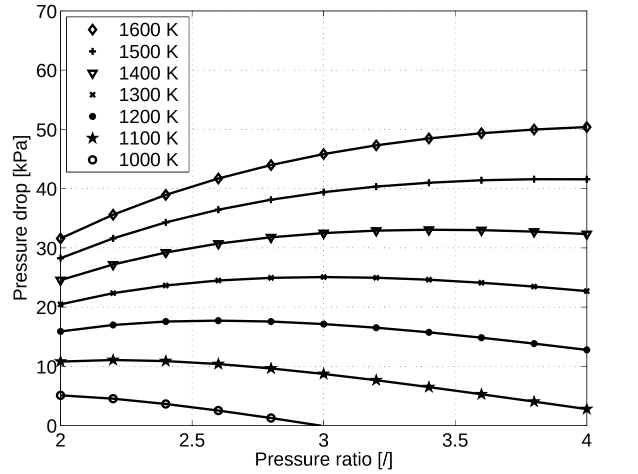

Fig. 7. Maximum allowable pressure drop dp_h for different turbine inlet temperatures ($\epsilon = 1$ and $dp_c = 0$)

IV. Heat exchanger configuration

In the previous sections, the influence of the recuperator on the cycle efficiency is assessed thermodynamically without any assumption about the recuperator design. In this section, the influence of the MIT recuperator on the cycle efficiency is calculated. Conductive and convective heat transfer and standard pressure drop correlations are used [6]. Entrance effects are considered. Loss coefficients of 1 and 0.6 are assumed for the inlet and outlet respectively.

For the MIT Brayton cycle with a mass flow rate of 0.15 g/s through the micro gas turbine, a heat exchanger effectiveness of 43.36%, a cold side pressure drop of 20.73 kPa and a hot side pressure drop of 1.73 kPa are obtained. This results in an overall cycle efficiency of 11.84%. This value is higher than the reference cycle efficiency of 9.31%. When the MIT recuperator, with a volume of 0.1 cm^3, is integrated in the target Brayton cycle with the same mass flow rate 0.15 g/s, the overall cycle efficiency decreases to 7.4%, which is higher than the reference cycle efficiency of 6.64%, but lower than the overall cycle efficiency of the MIT Brayton cycle with the same recuperator. The penalty of having lower pressure ratio and turbine inlet temperature can be counteracted by increasing the recuperator volume. Indeed, scaling the recuperator volume to 0.9 cm^3 results in the same overall cycle efficiency 11.84%.

When the MIT recuperator is integrated in the target Brayton cycle with a higher mass flow rate of 2.5 g/s, the recuperator volume should be increased to an unacceptable volume of 3593.7 cm^3 in order to reach the same overall cycle efficiency of 11.84%. Thus, it can be concluded that other heat exchanger configurations must be used in order to reach high overall cycle efficiency with lower recuperator volume. Indeed, for each pressure ratio, turbine inlet temperature and mass flow rate, an optimal channel configuration has to be deduced given a certain available volume.

Both optimal recuperator volume and channel dimensions are determined, based on the MIT recuperator set up. The overall dimensions of the MIT recuperator are scaled. For each volume, the optimal channel dimensions and numbers of channels are determined for hot and cold channels. This optimization occurs for constant overall recuperator dimensions, which means that the ratio of recuperator height and both inner and outer dimensions of the annular cross-sectional shape from the recuperator is kept constant. Figure 8 presents the cycle efficiency as a function of the recuperator volume with optimized channel dimensions and number of channels. The full line represents the target Brayton cycle with a mass flow rate of 0.15 g/s, whereas the dotted line represents the target Brayton cycle with a mass flow rate of 2.5 g/s. For the cycle with a mass flow rate of 0.15 g/s, the heat exchanger effectiveness and cold and hot side pressure drops are also plotted in full line.

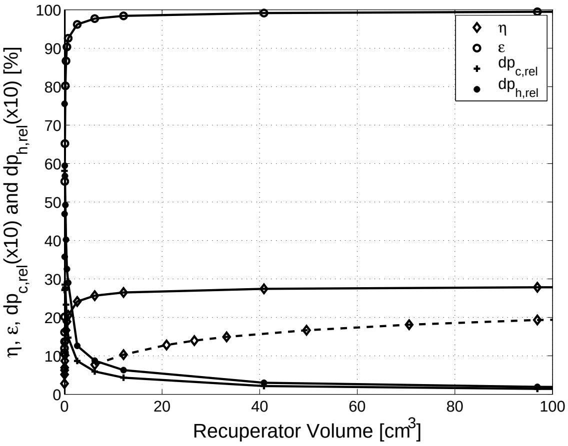

Fig. 8. Cycle efficiency as a function of recuperator volume for mass flow of 0.15 g/s (full line) and mass flow of 2.5 g/s (dotted line) in target cycle

The curve of the overall cycle efficiency η is monotonously rising, while the slope is decreasing with the recuperator volume. It can be seen that the higher the volume, the higher the heat exchanger effectiveness and the lower the pressure drops. The optimal recuperator volume can be defined based on a cost function that increases with volume or material cost. The cost function can be defined with respect to cycle efficiency and volume, f.e. for A and B positive constants, this could be expressed as $Cost = A\eta(V) - BV$. An optimal volume is than obtained for $\frac{dCost}{dV} = 0$ resulting in $\frac{d\eta(V)}{dV} = \frac{B}{A}$ as a prescription for an ideal recuperator. As such, the slope of the cycle efficiency curve in Figure 8 determines the optimal design. As an example, $3A = B$ is assumed. Thus, an optimal volume of 26.61 cm^3 is achieved for the target Brayton cycle with mass flow of 2.5 g/s, corresponding to an overall cycle efficiency of 13.95%. For the target Brayton cycle with a mass flow of 0.15 g/s, the optimum volume is 0.19 cm^3, corresponding to a cycle efficiency of 13.76 %. It can be concluded that optimizing the channel dimensions and numbers of channels drastically influences the overall cycle efficiency, even within the limitation of constant overall recuperator dimensions.

Furthermore, when the overall recuperator dimensions are changed as well, further improvement on cycle efficiency is expected. A constant volume optimization with varying ratio of overall recuperator dimensions is performed in order to define more recuperator design requirements. The optimization determines optimal number of channels, optimal channel cross-sectional dimensions as well as optimal channel lengths. From this optimization, performed on the MIT recuperator configuration, it can be concluded that all channels should be as short as possible and the number of channels as large as possible. It can be seen that same conclusions are valid for other geometric configurations.

From the discussed recuperator design requirements, two alternative heat exchanger configurations can be suggested. The first one has an annular shape with axially oriented channels. This recuperator should be placed between the compressor and the combination of turbine and combustion chamber. The second configuration has radially oriented channels, where the recuperator surrounds all other components of the micro gas turbine. Both configurations satisfy the discussed recuperator design requirements and will be investigated in future research.

V. Conclusion

A thermodynamic analysis has been performed to evaluate heat recuperation in micro gas turbines. The influence of downscaling is investigated. It can be concluded that the heat exchanger effectiveness should be as high as possible and the pressure drop should be preferably located at the cold side of the recuperator. For this reason, the hot channels should be larger than the cold ones. From the optimization for constant volume, it can be concluded that the channel lengths should be short and the cross-sectional area as large as possible, both for cold and hot channels. An important improvement is expected in two alternative heat exchanger configurations, which will be investigated in more detail in future research.

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