

INTEGRATED DESIGN OF A MICRO RECUPERATOR IN A GAS TURBINE CYCLE

T. Stevens, F. Rogiers, M. Baelmans

Department of Mechanical Engineering, Katholieke Universiteit Leuven, Belgium

Abstract: In this paper, a micro recuperator design for use in a 1.5 kW micro gas turbine is presented. The design is determined by a multi-dimensional optimization in which cold and hot side recuperator pressure drops are used as optimization parameter. The optimal design leads to an expected heat exchanger effectiveness of 79.82 % for relative pressure drops at cold and hot side of 2.03 % and 3.90 % respectively. It is observed that the cold and hot side pressure drops are uniquely correlated.

Key Words: micro recuperator, micro gas turbine, pressure drops

1. INTRODUCTION

Heat recuperation is often used to improve the overall cycle efficiency of standard gas turbines. For small sized gas turbines this improvement is even more important to compensate for the already low cycle efficiency due to significant lower achievable compressor pressure ratios and turbine inlet temperatures. A high heat exchanger effectiveness together with low pressure drops are favourable to achieve maximal cycle efficiency. Finding a compromise between these conflicting requirements is the main challenge in recuperator design.

Several micro recuperators for use in micro gas turbines with output powers smaller than 3 kW are already in development. Sullivan et al. [1] proposed an annular recuperator with cold and hot flow channels arranged in a counterflow configuration. First experiments showed a relative pressure drop of 5 % in the cold channels and 12 % in the hot channels and an effectiveness of 50 %. Nagasaki et al. [2] and Matsuo et al. [3] have presented a micro recuperator for palm-top size applications of nearly 3 kW output power. The conceptual design has an offset-strip-fin configuration and consists of 8 annularly arranged box-type recuperators that are placed around the turbine. The expected effectiveness is 80 %, with total relative pressure drop of 6.3 %. The same research groups have also developed a micro recuperator for use in a button-sized gas turbine with nearly 15 W output power [3]. The latter is based on the use of square microchannels in a three-path counterflow configuration. An heat

exchanger effectiveness of about 80 % and a total relative pressure drop of 7.8 % is expected.

Although several designs for micro recuperators were proposed, less attention is paid to the generalization of these concepts in specific design rules. Only Stevens et al. [4] mentioned first general design requirements for recuperators in micro gas turbines. These requirements primarily concentrated on the relation between cold and hot side pressure drops. It was concluded that pressure drops are preferably located at the cold side of the recuperator.

In this paper, optimal micro recuperator design is further explored. In section 2 the micro recuperator configuration is proposed as a component of a 1.5 kW micro gas turbine in development in the PowerMEMS project [5], [6]. A description of the optimization method is then given in section 3. Based on this optimization an optimal micro recuperator design is determined in section 4. In addition, it is shown that there is a fixed ratio between the cold and hot side recuperator pressure drops when it is integrated in the gas turbine. Based on these findings, a simplified optimization with parameter reduction is elaborated and compared with the multi-dimensional one.

2. RECUPERATOR CONFIGURATION

The micro recuperator under investigation is a counterflow microchannel heat exchanger. It consists of 6 recuperator blocks positioned around the gas turbine (see Fig. 1). One block has a total cross-sectional area A and length L . The inner geometry consists of alternating hot and cold plate

layers in which microchannels are fabricated (see Fig. 2). There is a wall thickness t_w between the layers, a fin thickness t_f between the channels, a number of n square channels of width w for the hot (h) and cold (c) plates of the recuperator.

A picture of one recuperator block with stacked cold and hot plates, together with the details of both plates is shown in Fig. 3.

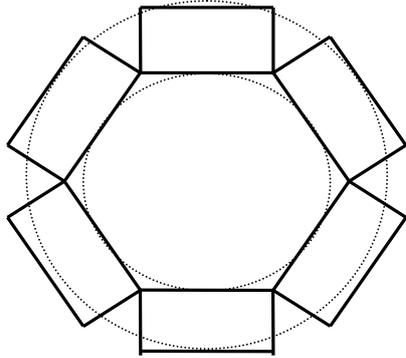


Fig. 1: Recuperator configuration

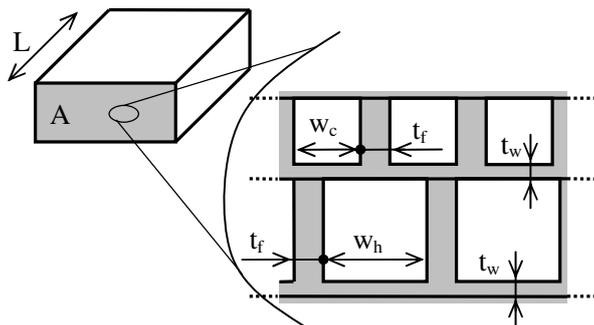


Fig. 2: Micro recuperator channel distribution

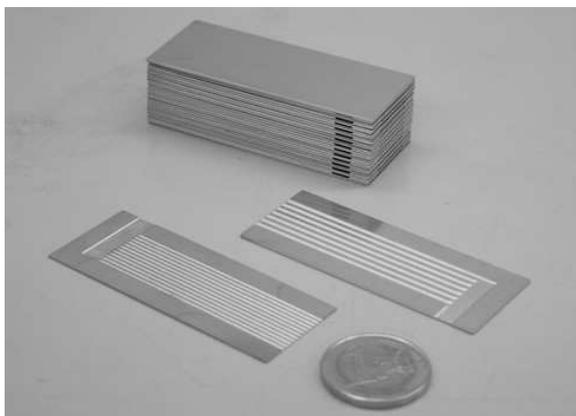


Fig. 3: One recuperator block with channels

3. OPTIMIZATION METHOD

In order to optimize the inner recuperator geometry, the dimensions of hot and cold channels ($D_{h,c}$ and $D_{h,h}$) as well as the number of channels (n_h and n_c) are computed for a set of cold and hot side pressure drops Δp_c and Δp_h . Mass flow rate, fluid properties, wall thickness, fin thickness, channel length and total recuperator cross-sectional area are fixed. The channels are assumed to be square and the wall thickness between the channels is chosen 100 μm in order to compensate for manufacturing limitations.

Based on the calculated geometrical dimensions the heat exchanger effectiveness ε is determined with a heat transfer model [7]. As small heat exchangers generally suffer from performance deterioration due to streamwise heat conduction through the walls [8, 9], axial conduction effects are taken into account.

The cycle efficiency η can be calculated based on the pressure drops Δp_c and Δp_h and the heat exchanger effectiveness ε [10]. The Brayton cycle in which the recuperator is incorporated has the following component characteristics: compressor pressure ratio $PR = 3.3$, turbine inlet temperature $TIT = 1200$ K, compressor efficiency $\eta_c = 0.61$, turbine efficiency $\eta_t = 0.80$, gas turbine outlet pressure drop $\Delta p_o = 2$ kPa, combustor pressure drop $\Delta p_{cc} = 2.5$ kPa, ambient inlet pressure and temperature and a mass flow of 20 g/s. Without recuperation, the cycle efficiency equals to 11 %.

4. CALCULATIONS AND RESULTS

The Brayton cycle efficiency η and the heat exchanger effectiveness ε are calculated for a recuperator with fixed volume (channel length $L = 53$ mm and cross-sectional area $A = 0.0032$ m²).

Isolines of heat exchanger effectiveness ε are shown in Fig. 4. The isolines are monotonously decreasing with higher values for larger pressure drops. This shows clearly that heat exchanger effectiveness and pressure drops are conflicting requirements in recuperator design as cycle efficiency increases for smaller pressure drops. It should be noted that each point in this figure is uniquely related to a recuperator geometry with the optimal channel dimensions.

The isolines of cycle efficiency η are shown in Fig. 5. The optimal recuperator has a cold side pressure drop Δp_c of 6,777 Pa (2.03 %) and a hot side pressure drop Δp_h of 3,954 Pa (3.90 %). The corresponding heat exchanger effectiveness ε and cycle efficiency η are respectively 79.82 % and 20.11 %.

For a wall thickness of 100 μm between the channels, the optimal recuperator inner geometry consists of 5,925 cold channels with hydraulic diameter of 355 μm and 3,784 channels with hydraulic diameter of 613 μm .

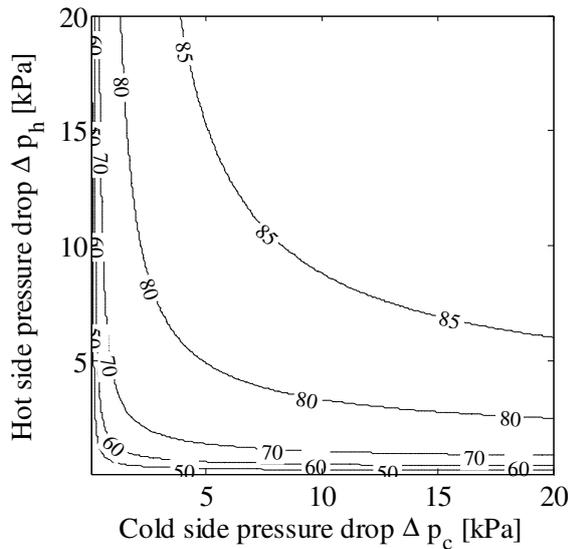


Fig. 4: Iso-effectiveness lines for recuperator

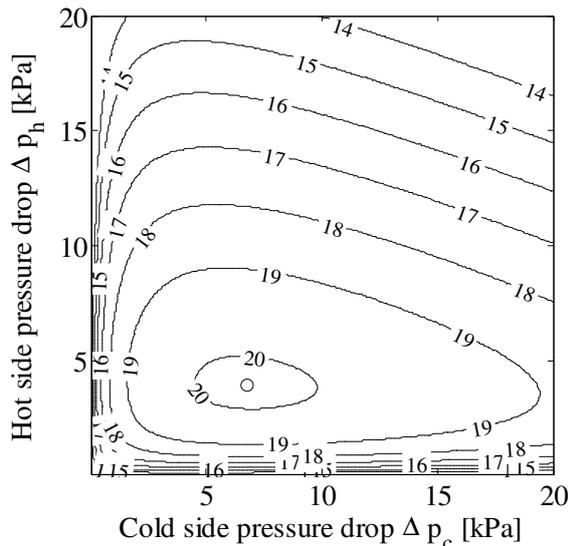


Fig. 5: Iso-efficiency lines for Brayton cycle

Another representation of the influence of the recuperator on Brayton cycle efficiency is given

in Fig. 6. For a given heat exchanger effectiveness ε , isolines of cycle efficiency are drawn. These isolines are straight lines and the slope of these lines seems to be independent of the value of ε . It can be seen that the tangent of the iso- η -lines with the iso- ε -lines gives the local optimal efficiency η for each heat exchanger effectiveness ε . It is also clear that the optima along these iso- ε -lines, shown as dots in Fig. 6, seem to be on a straight line. This means that the cold and hot side pressure drops are uniquely correlated and that the number of parameters in this optimization problem can be reduced by one as soon as the pressure drop ratio $\Delta p_c/\Delta p_h$ is determined. $\Delta p_c/\Delta p_h$ in this case is about 1.71. If this pressure drop ratio would be known in advance the optimization could be performed at a lower computational cost. Indeed, in this case the cycle efficiency η can be computed as a function of Δp_c only, as shown in Fig. 7. It should be noted that this confirms and extends the findings of Stevens et al. [4] where it was concluded that the hot side pressure drop should be smaller than the cold side one.

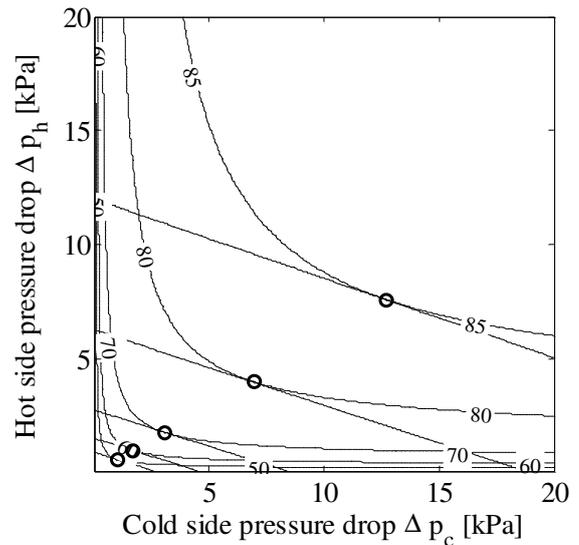


Fig. 6: Iso-efficiency η lines for Brayton cycle

With this simplified procedure the optimal recuperator inner geometry consists of 5,780 cold channels with hydraulic diameter of 360 μm and 3,686 channels with hydraulic diameter of 622 μm . This results in a predicted heat exchanger effectiveness of 79.80 % and a cycle efficiency of 20.11 %. The cold side pressure drop is now 6,884 Pa (2.06 %) and the hot side pressure drop is 4,017 Pa (3.96 %).

Compared with the results of the multi-dimensional optimization an excellent agreement is achieved. Indeed, a relative error of less than 2 % for the geometrical parameters, and a relative error of less than 0.02 % on heat exchanger effectiveness and cycle efficiency are observed. It should be noted that an a priori estimation of the pressure drop ratio further improves the optimization procedure and is therefore a challenge for further research.

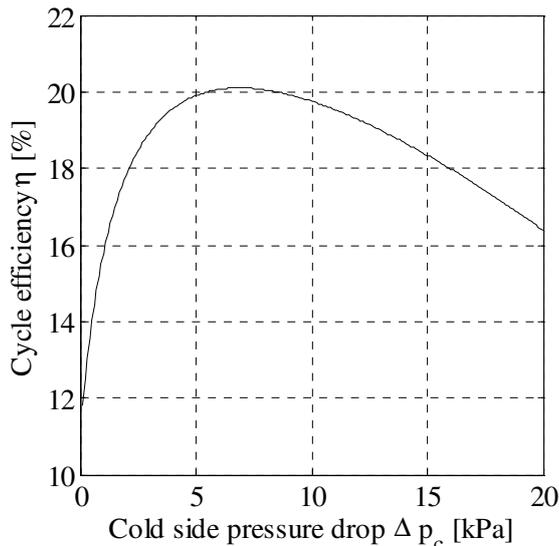


Fig. 7: Optimum for simplified optimization

5. CONCLUSION

In this paper, a multi-dimensional optimization is performed for the design of a micro recuperator for use in a 1.5 kW gas turbine. It is proven that there is a fixed ratio between cold and hot side pressure drops Δp_c and Δp_h . As a consequence, optimization can be simplified by parameter reduction. The pressure drop ratio $\Delta p_c/\Delta p_h$ is 1.71 in the gas turbine under consideration. The results of the simplified optimization are compared with the multi-dimensional ones and show an excellent agreement.

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