# Optimal recuperator design for use in a micro gas turbine

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**Abstract**

In this paper the optimization of a recuperator for its use in a 2.5 kW micro gas turbine is presented. The challenge is to find a design with low pressure drops in the channels in combination with a high heat exchanger effectiveness, taking into account manufacturing limits and geometrical constraints of system integration. An inverse pressure drop model is used in combination with a heat transfer model that accounts for axial heat conduction in the walls. The influence of different geometrical constraints on the recuperator effectiveness is first studied. The recuperator is then integrated into the gas turbine. Two different recuperator design configurations are considered. An optimisation using cycle efficiency as a cost function is performed to determine the optimal heat exchanger configuration. It is found that the choice between the two configurations depends on the outer dimension constraints imposed on the recuperator.

*Keywords: Micro recuperator, Optimization, Inverse design, Heat exchanger, Micro gas turbine*

## 1 - INTRODUCTION

Heat recuperation is often used to improve the overall cycle efficiency of standard gas turbines. Small sized gas turbines rely even more on this improvement to compensate for the already low efficiency due to the significant lower achievable pressure ratios and turbine inlet temperatures when compared to conventionally sized gas turbines. Currently, no adequate micro recuperators have been tested in combination with a micro gas turbine in the range of 2 kW. MIT already proposed a micro recuperator design in 2001 [1] but this heat exchanger has not been further developed. Design requirements for micro recuperators were proposed in 2004 by K.U.Leuven [2], also in view of the development of a micro gas turbine [3]. In this research it was concluded that pressure drops are preferably located at the cold side of the recuperator. Therefore, the hot channels should be larger than the cold ones.

A main challenge in recuperator design is to find a compromise between low pressure drops and sufficiently high heat exchanger effectiveness. Pressure drop and heat exchanger effectiveness are strongly coupled by the geometry. In order to find optimal channel dimensions, constraints induced by system integration and limitations in production techniques should be taken into account. A convenient method to analyse the importance of geometrical constraints is an inverse pressure drop approach. Hereby, the inner dimensions of the heat exchanger are determined to meet both the given pressure drop and the outer geometry constraints.

Small heat exchangers generally suffer from performance deterioration due to streamwise heat conduction through the walls [4]. Earlier research in micro heat exchanger design shows that there is an optimal wall conductivity, with typical values slightly lower than 1 W/mK [5], which minimize the axial conduction effect while maintaining a sufficiently low thermal resistance path from the hot to the cold side. For higher conductivity materials axial conduction effects must be taken into account to evaluate the performance of a micro heat exchanger.

In this paper an inverse pressure drop analysis technique for heat exchangers is introduced and discussed. This model is then combined with a thermal model that takes into account streamwise conduction through the wall to analyse the influence of geometrical constraints, imposed on the micro channel core of a heat exchanger, on the effectiveness. Finally the effect of geometrical constraints on the optimal recuperator design for the current K.U.Leuven micro gas turbine design is shown.

## 2 – HEAT EXCHANGER MODEL

The recuperator under investigation is a counter-flow micro channel heat exchanger as depicted in Fig. 1. It consists of alternating hot and cold plate layers in which micro channels channels are produced. The heat exchanger has a total transverse area $A$, 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 ($c$) side and channels of lengths $L$.

*a. Pressure drop model*

The pressure drop over the heat exchanger is calculated [6], assuming that density changes over the heat exchanger only slightly affect the pressure drop:

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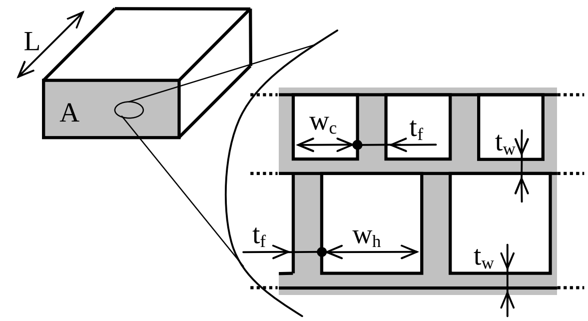


**Figure 1 –** Schematic of the micro channel heat exchanger

$$\Delta p = \frac{1}{2}\rho_m V_{ch,m}^2 \left( K_c + f_m \frac{4L}{D} + K_e \right). \qquad (1)$$

where $\rho_m$ is the mean density, $V_{ch,m}$ the channel mean bulk velocity and $K_c$ and $K_e$ are the contraction and expansion loss coefficients (respectively set to 0.5 and 1 corresponding to sharp-edged in- and outflow and large collectors [7]). $f_m = 14.32/Re_{D,m}$ is the mean Fanning friction factor for square channels and laminar developed flow [8] with $Re_{D,m}$ the channel Reynolds number based on the hydraulic diameter $D=w$. Additional collector losses due to friction and turns are neglected.

*b. Heat transfer model*

The heat exchanger effectiveness is calculated as follows:

$$\varepsilon = \frac{C_c}{C_{\min}} \theta_{c,out} ,$$

where the dimensionless outlet temperature $\theta_{c,out}$ is obtained from solving the following system of differential equations expressing thermal energy conservation in the hot stream ($h$), cold stream ($c$) and the wall ($w$) along the flow direction:

$$\begin{cases} \dfrac{d\theta_h}{d\tilde{x}} = NTU_h \left( \theta_w - \theta_h \right) \\ \dfrac{d\theta_c}{d\tilde{x}} = -NTU_c \left( \theta_w - \theta_c \right) \\ \dfrac{d^2\theta_w}{d\tilde{x}^2} = \dfrac{NTU_h}{M_h} \left( \theta_w - \theta_h \right) + \dfrac{NTU_c}{M_c} \left( \theta_w - \theta_c \right) \end{cases}$$

Hereby $\tilde{x} = x/L$ is the dimensionless length in the flow direction, $\theta = (T - T_{c,in})/(T_{h,in} - T_{c,in})$ the dimensionless bulk temperature, $NTU = UA/C$ the number of transfer units with $C = \dot{m}c_p$ the heat capacity of the total (hot or cold) fluid flow through the heat exchanger and $M = k_w A_w / LC$ the conduction parameter which is a measure for the importance of streamwise conduction in the wall. The system is subject to boundary conditions expressing preset hot and cold inlet temperatures and isolated wall ends. A solution to the system of differential equations is obtained by finite difference discretization and subsequent solving the obtained linear system of algebraic equations for the temperature. The *UA*-values are determined from calculation of an equivalent thermal resistance between the bulk fluid temperature and the bulk wall temperature. The assumption of an isothermal wall temperature line located in the middle of the wall between hot and cold channels induces independent fluid to wall thermal resistances for the hot and cold side. Consequently the calculation of the equivalent thermal resistance between the fluid side and the wall can be performed based on the unit cell shown in Fig. 2. The *UA*-value of this unit cell is given by:

$$\frac{UA}{2n} = \left( \frac{1}{w \cdot h} + \frac{0.5t_w}{k_w \cdot w} \right)^{-1} + 2\left( \frac{1}{0.5w \cdot \eta_f \cdot h} + \frac{0.5t_w}{k_w \cdot 0.5t_f} \right)^{-1}$$

with $k_w$ the thermal conductivity of the heat exchanger material, $\eta_f = \tanh\left(mw/2\right)/\left(mw/2\right)$ the fin efficiency with $m = \sqrt{2h/k_w t_f}$ and $h = Nu.k_{fl}/D$ the mean convective heat transfer coefficient with $k_{fl}$ the fluid conductivity and $Nu$ = 3.61 corresponding to a square channel, developed laminar flow and uniform heat flux conditions [9].

## 3 – INVERSE SOLUTION PROCEDURE

Given the large amount of internal and external configuration parameters, the importance of geometrical constraints is first investigated by fixing the pressured drop over the channels. Therefore, an inverse pressure drop design is applied to determine channel dimensions ($w_h$ and $w_c$) and the number of channels ($n_h$ and $n_c$) for a given mass flow rate, fluid properties, pressure drop and the given dimensional parameters wall thickness, fin thickness, channel length and total transverse area. Using $\dot{m} = n\rho_m V_m w^2$, Eq. 1 can be rewritten as a function of channel width and number of channels:

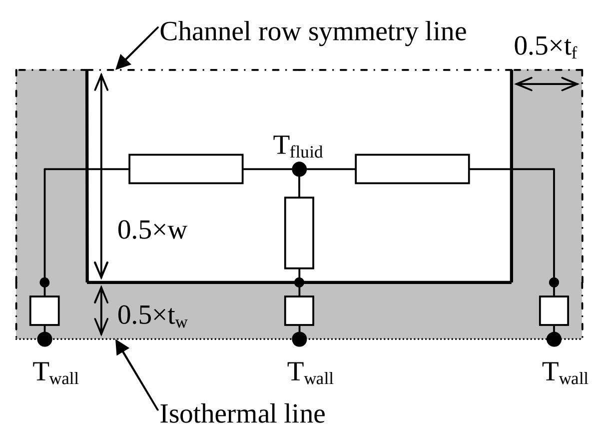


**Figure 2 –** Unit cell with parallel thermal resistances connected to an isothermal plane between hot and cold channels.

$$\Delta p = \frac{1}{2}\frac{\dot{m}^2}{n^2 \rho_m w^4}\left( K_c + 4(f\ \mathrm{Re})_m \frac{n\mu_m L}{\dot{m}} + K_e \right). \qquad (2)$$

This equation holds for hot and cold side of the heat exchanger resulting in two equations with the four unknowns $w_h$, $w_c$, $n_h$, $n_c$. Additional equations can be found from constraints in the transverse plane of the heat exchanger. Indeed, the total transverse area is given by:

$$A = n_h\left(w_h + t_f\right)\left(w_h + t_w\right) + n_c\left(w_c + t_f\right)\left(w_c + t_w\right). \qquad (3)$$

Requiring an equal plate width for hot and cold side plates results in:

$$\frac{n_h}{n_{pl,h}}\left(w_h + t_f\right) = \frac{n_c}{n_{pl,c}}\left(w_c + t_f\right).$$

Or, since the number of hot and cold plates $n_{pl}$ is equal:

$$n_h\left(w_h + t_f\right) = n_c\left(w_c + t_f\right). \qquad (4)$$

Equations (2) to (4) form a system of four non-linear equations which is iteratively solved. Once the internal heat exchanger dimensions are computed the effectiveness is obtained from the heat transfer model.

## 4 – ANALYSIS OF EFFECTIVENESS UNDER DIFFERENT GEOMETRICAL CONSTRAINTS

With the purpose of investigating the effect of outer geometrical constraints, the recuperator volume, transverse area and length are altered independently while the other ones are kept fixed. In order to compare the different cases, a dimensionless shape parameter σ is introduced which is a measure for the heat exchanger slenderness:

$$\sigma = \sqrt{L^2/A} = \sqrt{L^3/V} = \sqrt{V^2/A^3}$$

The baseline recuperator has a length of 6 cm and unity slenderness. The recuperator effectiveness is calculated with the procedure discussed in section 3. Pressure drops are set to $\Delta p_h$ = 8 kPa and $\Delta p_c$ = 12 kPa. Further, following values are used:

$t_f$ = 30 μm ; $t_w$ = 80 μm ; $k_w$ = 21.4 W/mK ;
$\dot{m}_h$ = 20 g/s ; $\rho_h$ = 0.74 kg/m$^3$ ; $c_{p,h}$ = 1035 J/kgK ;
$\mu_h$ = 2.24×10$^{-5}$ Ns/m$^2$ ; $k_h$ = 4.31×10$^{-2}$ W/mK
$\dot{m}_c$ = 20 g/s ; $\rho_c$ = 1.26 kg/m$^3$; $c_{p,c}$ = 1175 J/kgK ;
$\mu_c$ = 2.48×10$^{-5}$ Ns/m$^2$ ; $k_c$ = 6.33×10$^{-2}$ W/mK

The results for three different cases are shown in Fig. 3. In the first case, the slenderness is varied for a constant volume of 108 cm$^3$, 216 cm$^3$ and 432 cm$^3$. The effect on recuperator effectiveness is shown in Fig.3a. It can be seen that there exist an optimal slenderness for each recuperator volume. This optimum shifts towards higher slenderness with increasing volume. Increasing the total volume and keeping the slenderness constant does not guarantee an increase in effectiveness when the slenderness is too low.

In the second case the slenderness is varied for a constant surface area of the recuperator of 18 cm$^2$, 36 cm$^2$ and 72 cm$^2$. Figure 3b shows that the effectiveness increases with

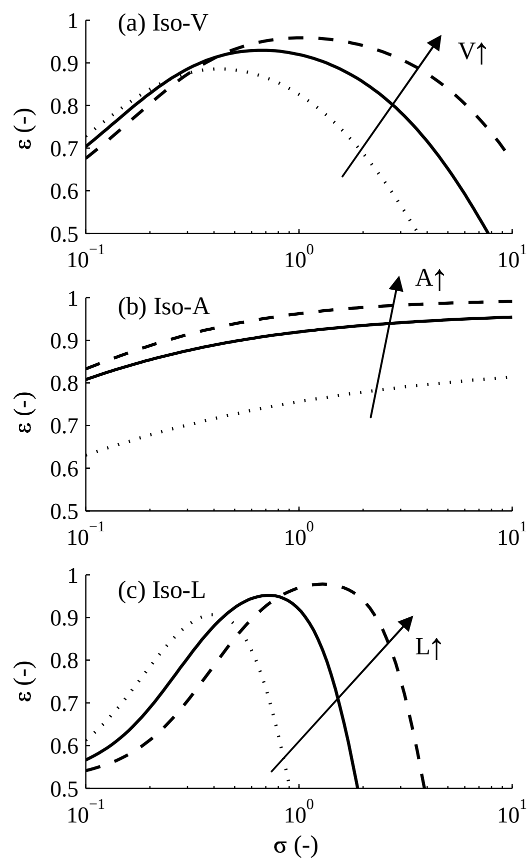


**Figure 3** – Iso-V, Iso-A and Iso-L curves for baseline dimensions (–), halve baseline dimensions (···) and double baseline dimensions (–·–) for $\Delta p_h$ = 8 kPa and $\Delta p_c$ = 12 kPa

increasing slenderness. Finally in Fig. 3c, the effectiveness is plotted for constant a constant length of 3 cm, 6 cm and 12 cm. As with the constant volume case, there exists an optimum which shifts towards higher slenderness with increasing volume.

All plots exhibit a similar behaviour: for large slenderness values an increase in volume, transverse area or length induces a higher efficiency. At low slenderness values a contra-intuitive trend is induced by high axial conductivity effects.

## 5 – OPTIMAL HEAT EXCHANGER CONFIGURATION

The cube-like configuration is now piecewise integrated in a proposed gas turbine design. The target recuperator consists of 6 micro channel core blocks arranged in an annular shape. Figure 4 gives a front view of this annular volume. The inner diameter is fixed to 6 cm and the length (in z-direction) is fixed to 6.7 cm leaving the outer diameter as a design parameter. Two alternative configurations are considered as depicted in Fig. 4. In the first one the flow passes axially

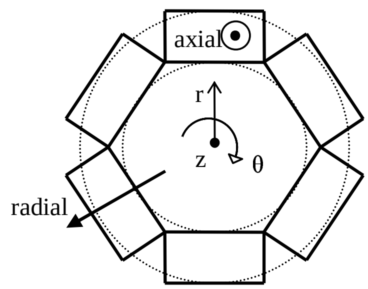


**Figure 4 –** Constant z section of the annular heat exchanger with axial or radial flow directions

through the blocks. In this case, increasing the outer diameter is the same as decreasing the slenderness for constant flow length. In the second case the flow passes radially through the blocks. Changing the outer diameter is the same as increasing the slenderness at constant transverse area.

The configuration that leads to the highest cycle efficiency will now be determined in an optimization. For the present case one single design variable $\Delta p_h$ is optimized for every outer diameter using the inverse pressure drop model, the heat transfer model and a cycle model that calculates the total cycle efficiency. Since it is known that the pressure drop is preferably located at the cold side of the heat exchanger [2] a fixed ratio $\Delta p_{,c} / \Delta p_{,h}$ equal to 1.5 is set. The Brayton cycle in which the recuperator is used has following characteristics: $PR$ = 3; $TIT$ = 1200 K; $\eta_c$ = 0.8; $\eta_t$ = 0.87; $\Delta p_{,outlet}$ = 2 kPa; $\Delta p_{,combust}$ = 18 kPa; $p_1$ = 1 bar; $T_1$ = 290 K ; $\dot{m}$ = 20 g/s. For this cycle the unrecuperated and ideally recuperated cycle efficiencies amounts respectively to 16.4 % and 49.7 %.

Figure 5 shows the maximal cycle efficiency as a function of outer diameter for the two configurations under investigation. For small values of the outer diameter a radial design can give an efficiency improvement of 5 % points over an axial design and is therefore preferable. For values of $D_{out}$ between 9 cm and 19 cm the axial configuration is the better one. For high values of the outer diameter the axial design suffers from an increase in conduction parameter due to a low slenderness and the cycle efficiency starts to decrease. A constant length design problem for axial heat exchangers thus exhibits an optimum that is not necessarily constraint by the

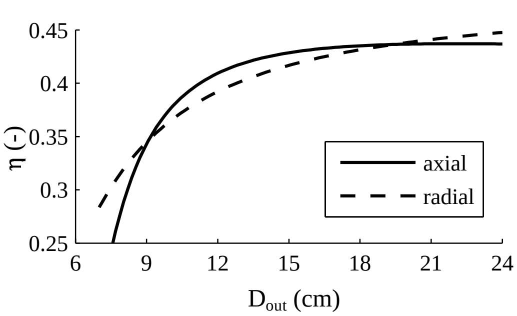


**Figure 5 –** Cycle efficiency vs. outer diameter for an axial design (–) and a radial design (--)

maximal allowable radial size of the component.

## 5 - CONCLUSION

In this paper it is shown that the inverse pressure drop model leads to a convenient procedure to explore geometrical constraints of a micro heat exchanger. Its combination with the heat transfer model that accounts for axial conductivity reveals that the slenderness, which is a measure for the heat exchanger outer shape, is a determinant for the effectiveness. For a given volume or flow length there exists an optimal slenderness. Furthermore, the effect of axial conductivity leads to contra-intuitive trends at small slenderness values: e.g. an increase of volume does not necessarily improve the efficiency. Further, increasing the length or transverse area of a micro heat exchanger core and keeping the total volume equal will generally not yield the same rate of performance improvement. The effectiveness can even become worse by increasing the transverse area of the heat exchanger because of a decrease in axial wall resistance.

For the envisaged application of a micro gas turbine radial design is favorable for very small or high outer diameter values, whereas an axial design for intermediate values is preferred. Further, a constant length design problem for axial heat exchangers thus exhibits an optimum that is not necessarily constraint by the maximal allowable radial size of the component


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## REFERENCES


[1] S. Sullivan, X. Zhang, A. A. Ayon and J. G. Brisson, “Demonstration of a microscale heat exchanger for a silicon gas turbine engine, “*The 11th International Conference on Solid-State Sensors and Actuators*, Munich, Germany, June 2001.

[2] T. Stevens, F. Verplaetsen, M. Baelmans, Requirements for recuperators in micro gas turbines, *Proceedings of PowerMEMS conference*, Kyoto, Japan, pp.96-99, 2004.

[3] J. Peirs, D. Reynaerts, F. Verplaetsen, Development of an axial microturbine for a portable gas turbine generator, *J. Micromechanics and Microengineering*, Vol. 13, No. 5, pp. 190-195, 2003.

[4] R.B Peterson. Numerical modeling of conduction effects in microscale counterflow heat exchangers, Microscale Thermophysical Engineering, 3:17-30, 1999.

[5] T. Stief, O. Langer, K. Schubert, “Numerical Investigations of Optimal Heat Conductivity in Micro Heat Exchangers,” *Chem. Eng. Technol.*, Vol. 21, No. 4, pp.297-303, 1999

[6] W. M. Kays, A. L. London, “Compact Heat Exchangers,” 2nd ed., *McGraw-Hill Book Company*, New York, pp. 33, 1964.

[7] F.M. White, “Fluid Mechanics,” 5th ed., *McGraw-Hill*, New York, pp. 390, 2003.

[8] S. Kakac, R. K. Shah, W. Aung, “Handbook of Single-Phase Convective Heat Transfer,” *Wiley*, New York, 1987.