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SEINÄJOKI UNIVERSITY OF APPLIED SCIENCES

André Kaufmann & Hannu Ylinen

Preliminary thermodynamic design of a stirling cooler for mobile air conditioning systems

Technical report

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Seinäjoki 2015

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ABSTRACT

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The present discussion on refrigerants used in mobile air conditioning (MAC) units leads to the question whether a stirling cycle based heat pump could replace the present technology. An experimental and simulative study is carried out to determine the design parameters of such a heat pump. It is found that the targeted power density cannot be reached with air as a working fluid. The power require-ments would lead to machine sizes too large for a passenger vehicle.

Keywords: Keywords: Stirling cycle, Stirling cooler, heat pump

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1 INTRODUCTION

Regulations for efficiency in terms of CO_2 emission (European commission, 2009) force automotive manufacturers to change their design in air conditioning units. New regulations limiting the use of refrigerants in cars and trucks narrow the possibilities of air conditioning systems. In electric vehicles, resistance heating in winter time significantly reduces the vehicle range due to limited battery capacity. Presently most activities go into implementation of air conditioning (AC) units and heat pumps (HP) for electric vehicles using the refrigerants R744 or R1234yf. Both refrigerants have got advantages and drawbacks. R1234yf has similar properties when it comes to working pressure and coefficient of performance (COP) as refrigerant R134a. R134a was prohibited by the European Union for the use in newly homologated vehicles starting 01.2011 (European commission, 2009) due to its global warming potential (GWP). Present discussions on the safety of R1234yf use in vehicles (European commission, 2014) put into question the acceptance by the customer. AC units using R744 have to cope with significantly higher pressure levels (up to 130 bars). Furthermore the heat removal at the high pressure level is not in an area of phase change and therefore completely supercritical. This is, compared to other refrigerants, a drawback since heat exchange is not isothermal.

Therefore the use of classical refrigerants is questioned and alternatives are considered. Air has been used as working liquid in aircraft AC units by a Bell Coleman cycle. This cycle is known for poor COP but has the advantage of light components. Alternatively an inverse Stirling cycle can be used as a heat pump. Cryocoolers relying on the inverse stirling process are a common technology used in cooling optoelectronics for instance. Stirling Cryocoolers are commercially available and show a service free lifetime. Other attempts of using the Stirling cycle are made for mobile cooling applications. Coleman (2014) sells a cooler box and TwinbirdFreezer (Twinbird 2014) a freezer based on the Free Piston Stirling Engine (FPSE) developed by Sunpower Inc. and Global Cooling. The FPSE technology has also been used in the development of refrigerator.

The success in using the stirling cycle in cooling devices with low heat flux i.e. smaller 100 Watt are common, leaves the question if such a device can be scaled to the size necessary for the use in mobile air conditioning (MAC) devices with a typical heat flux of 3 kW and COPs competitive with those of present MAC units. This question is addressed in the present paper by attempting a preliminary design.

The literature on stirling cycle analysis, see Finkelstein (Finkelstein 1960) for instance, is vast compared to the literature on the design of stirling cycle machines. Gedeon (Gedeon 1981) gives scaling rules. Existing well working designs can be scaled using this approach to larger or smaller machines. Organ (Organ 1992) (Organ 2014) follows the same approach while giving much more insight on the significance on the non-dimensional parameters. Those approaches are good practice when existing machines have been analyzed. There is however very literature reporting the design parameters of stirling coolers when scaling cannot be applied.

The authors develop a system of non-dimensional numbers partially identical to those define by Finkelstein (1960), Gedeon (1981) and Organ (2014) While Gedeon and Organ derive the non-dimensional numbers by the use of the Buckingham Π -theorem (Barenblatt 1996) the authors derive the non-dimensional values using the underlying differential equations. This has the advantage that the interpretation of the non-dimensional numbers is easier and sensible values for those numbers can be guessed.

A nodal analysis program was used to scan the design space for a good combination of design parameters. Results of the nodal analysis are compared to the ideal adiabatic analysis as a reference. Additionally the nodal analysis program is validated against measurements of a cooler previously designed by third party. The design space does not yet have the physical limitations due to material properties and geometric constraints, but it guides the authors to a sensible choice of parameters.

Physical limitations due to material properties, heat transfer and geometrical constraints are discussed with respect to the manufacturing feasibility.

2 STIRLING COOLER

For the sake of simplicity a α type stirling machine is considered. A sketch of a type α machine described by Urieli and Berchowitz (Urieli & Berchowitz, 1984) is given in fig. 1. From left to right the machine consists of a variable Volume V_1 ,a heat exchanger HE_1, the regenerator Rg, a second heat exchanger HE_2 and a second variable volume V_2 .

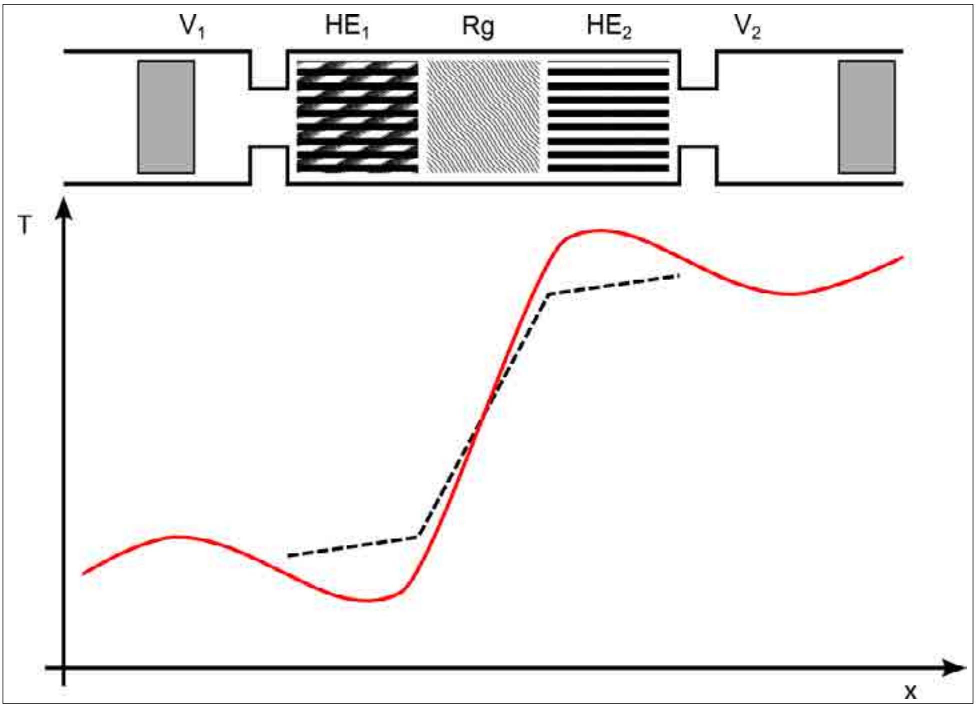


FIGURE 1. Sketch of a Alpha type Stirling machine used as a cooler with the temperature profile of walls (dashed line) and gas (continuous line).

2.1 Stirling cycle

In a textbook stirling cycle, the machine works with two isothermal and two isochoric state changes for the gas (Langeheinecke, et al., 2013). This is a large simplification as the gas has different temperatures in the different volumes and a simplification to one single temperature is not appropriate (Urieli & Berchowitz, 1984). An alternative is however to consider a fluid particle of constant mass. The fluid particle can be considered so small, that it is justified to consider uniform temperature, uniform density and uniform pressure. A fluid particle initially located in V_1 will be pushed into the heat exchanger HE_2. Due to the change in total volume, the volume of the fluid particle will change volume too. In contact with the walls, the fluid particle will exchnange heat and temperature of the fluid particle will change. It is therefore appropriate to consider the p—v and T—s diagrams of fluid particles for the stirling cycle. The fluid particle intially located in V_1 will however not travell all the way through the regenerator Rg heat exchanger HE_2 and into volume V_1 . The location of the fluid particle over one entire cycle depends on the ratio of the sizes of the volumes in V_1,V_2,HE_1,HE_2 and Rg . A sketch of possible fluid particle trajectories in p—v and T—s diagrams is given in fig. 2.

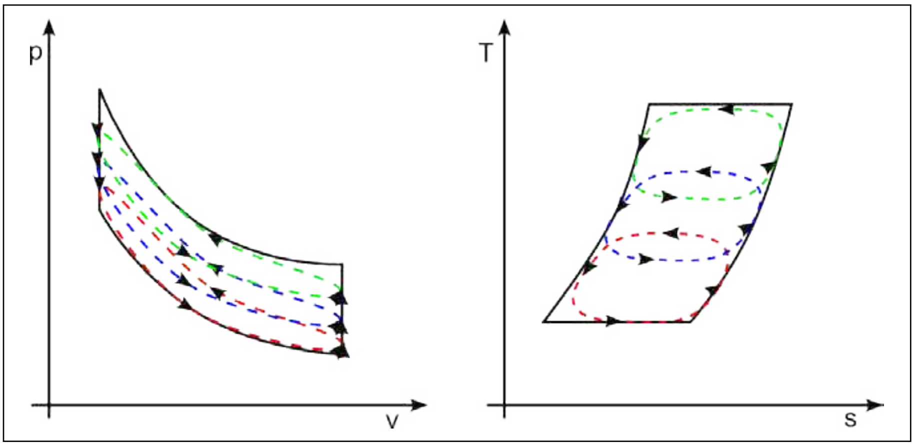


FIGURE 2. Sketch fluid particle trajectories in p—v (left) and T—s (right) diagram. The wall temperatures of the heat exchangers and regenerator are given with dashed lines. HE_1, HE_2 and Rg

From a mechanical point of view it is difficult to achieve isochoric and isothermal state changes. Volumes V_1 and V_2 are typically connected to some kind of crank system to realize the volume change. At this point important design parameters can be identified. The larger the total volume change dV compared to the total volume of the system V_{tot} , the larger is the isothermal exchange in p—v and T—s diagram and thus the total transfered heat. Therefore the ratios of V_1/V_{tot} and V_2/V_{tot} and the phase angle $\Delta\theta$ are important design parameters. For a detailed description of the stirling cycle please refer to the available literature (Urieli & Berchowitz (1984) and others).

2.2 Realisation of stirling engines

With common crank mechanisms it is not possible to achieve isochoric and isothermal textbook cycles. Several mechanisms exist to drive the pistons. Organ (2014) and others give a good review on the different mechanisms. For simplification the volumes are assumed to change as trigonometric functions.

$$V_1 = V_{d1} \cdot \frac{1}{2}(1 + \cos(\theta))$$
$$V_2 = V_{d2} \cdot \frac{1}{2}(1 + \cos(\theta + \Delta\theta)) \tag{1}$$

This is the limit of a very long crankshaft in a practical realization.

3 DESING SPECIFICATION

For the use in MAC systems the lower temperature is typically about $T_{low} = 273\ K$ and the upper temperature $T_{high} = 333\ K$. This is a small temperature ratio compared to typical stirling engines and to cryocoolers. This makes the potential textbook coefficient of performance very attractive but is associated in practice with small temperature differences in the heat exchangers and therefore low heat transfer rates. The typical cooling capacity of passenger vehicle MACs is roughly $3\ kW$. The necessity for heating in electric vehicles is the same order of magnitude. One advantage of the stirling cycle is that a change of phase angle can be used to exchange hot and cold side. Therefore the target is to keep the heat pump symmetrical with respect to displaced volumes and heat exchangers.

4 DETERMINATION OF DESING PARAMETERS

For the development of design criteria, the fluid particle approach gives an idea for the design of the volume ratios. In order to develop criteria for the heat exchanger and regenerator the different volumes of the machine are regarded separately. The situation is simplified by assuming uniform temperature and uniform pressure throughout one volume. According to thermodynamics the conservation of energy and mass must be fulfilled in every volume. For the derivation of the design parameters working fluid parameters like heat capacity c_p , isentropic coefficient $\kappa = c_p/c_v$ and gas constant are assumed to be fixed.

4.1 Energy conservation

Energy conservation is used in differential form.

$$dU = dQ - p \cdot dV + he\,dm \tag{2}$$

The change of internal enery dU is therefore the sum of exchange of heat dQ, Volume work- pdV and enthalpy exchanged HE with mass dm. The change of internal energy due to the exchange of mass dm needs to be taken into account. The internal energy associated to the exchanged mass udm and the work to push the mass $RTdm = d(pV)$ in or out of the volume is therefore added to the energy conservation. With the specific enthalpy $\mathrm{h} = u + pv$ this term can be regrouped to $\mathrm{h}_e dm$. The heat transfer dQ to the gas in the considered volume is assumed to be of convective nature. This can be modeled using a heat transfer coefficient α and the contact surface of the gas to the wall A_w.

$$\frac{dQ}{dt} = \alpha A_w (T_w - T_g) \tag{3}$$

For the sake of generality, it is useful to introduce some reference values so the energy conservation can be analyzed in non-dimensional form.

p_{ref} = average working pressure

V_{ref} = total swept volume

T_{ref} = reference temperature, mostly of lower heat exchanger

$$m_{ref} = \frac{p_{ref}V_{ref}}{RT_{ref}} \tag{4}$$

$$U_{ref} = m_{ref}c_vT_{ref} \tag{5}$$

For further simplification, gas is assumed to follow the ideal gas law $pV = mRT$ and to have constant heat capacity c_v. This assumption is justified by the moderate pressure smaller $10\ bar$ and the moderate temperatures below 100 °C. This enables the use of $U = mc_vT$. Using the reference values, non-dimensional quantities are defined. The non-dimensional quantities always carry a hat symbol.

$$\hat{p} = p/p_{ref} \tag{6}$$

$$\hat{V} = V/V_{ref} \tag{7}$$

$$\hat{T} = T/T_{ref} \tag{8}$$

$$\hat{m} = m/m_{ref} \tag{9}$$

$$\hat{U} = U/U_{ref} \tag{10}$$

The ideal gas law and the state equation can be formulated in non-dimensional quantities by dividing by the reference values.

$$\frac{pV}{p_{ref}V_{rev}} = \frac{mRT}{m_{ref}RT_{ref}} \qquad pV = mT \tag{11}$$

The differential equation for the energy is time dependent. Since a periodic cycle is to be analyzed, it is more appropriate to write the energy equation as a function of angle θ. The angle is related to time by the rotational frequency $\theta = \omega_0 t$..

$$\frac{dU}{d\theta} = \frac{1}{\omega_0}\frac{dQ}{dt} - p \cdot \frac{dV}{d\theta} + \mathrm{h}_e \frac{dm}{d\theta} \tag{13}$$

Introduction of the non-dimensional variables lead to a non-dimensional version of the energy equation.

$$\frac{dU}{d\theta} = \frac{1}{\omega_0 m_{ref} c_v T_{ref}}\frac{dQ}{dt} - \frac{p}{m_{ref} c_v T_{ref}} \cdot \frac{dV}{d\theta} + \frac{\mathrm{h}}{m_{ref} c_v T_{ref}}\frac{dm}{d\theta} \tag{14}$$

The terms on the right hand side (rhs) can be simplified using relations for the ideal gas.

$$c_p - c_v = R \qquad \frac{c_p}{c_v} = \kappa \qquad \frac{R}{c_v} = \kappa - 1$$

$$u = c_v T \qquad \mathrm{h} = c_p T$$

Together with the convective heat transfer assumption the non-dimensional energy equation is dependent only on one parameter.

$$\frac{dU}{d\theta} = \underbrace{\frac{\alpha A_w}{\omega_0 m_{ref} c_v}}_{NTU}(T_w - T_g) - (\kappa - 1)p\frac{dV}{d\theta} + \kappa T_e \frac{dm}{d\theta} \tag{15}$$

This parameter is the number of transfer units (NTU). Physically it is the ratio of heat transferred by convection compared to the heat transport capacity of the gas traveling through the volume. This parameter was first identified by Finkelstein (1960) and is identified by Organ (2014) as a key design parameter.

The non-dimensional heat transfer per unit crank angle depends on the NTU parameter and the temperature difference between gas T_g and wetted surface T_w.

$$\frac{dQ}{d\theta} = \underbrace{\frac{\alpha A_w}{\omega_0 m_{ref} c_v}}_{NTU}(T_w - T_g) \tag{16}$$

The heat transfer coefficient α is constant for laminar flow. In this case the NTU number can be taken as constant. Heat transfer depends only on the temperature difference.

In the case of turbulent flow, α is function of the Reynolds-Number. The NTU number is then a function of crank angle. This makes the use of NTU as a design parameter more difficult. For the design process the NTU number will therefore be held independent of crank angle.

4.2 Mass conservation

As the volumes V_1 and V_2 change as a function of θ, mass is exchanged between the individual volumes. The amount of mass exchanged per unit time depends on the pressure gradient. For the sake of simplicity the pressure in every volume is considered uniform. Mass exchanged between volumes depends on the pressure difference. The velocity at the volume intersection is estimated using the Bernoulli energy relation.

$$c \approx \sqrt{\frac{2\Delta p}{\rho(1+\zeta)}} \tag{17}$$

This approach neglects all effects due to temporal change of velocity and inertia. ζ is the non-dimensional parameter taking into account the pressure loss due to flow friction in the system. For the pressure loss in duct or tube of length l and diameter d the loss coefficient is determined using the Moody diagram and the loss parameter λ.

$$\zeta = \lambda \frac{l}{d} \tag{18}$$

The mass flow rate between the volumes is proportional to cross section A_F, density ρ, and the velocity c at the cross section.

$$\frac{dm}{dt} = \rho A_F c \approx \rho A_F \sqrt{\frac{2\Delta p}{\rho(1+\zeta)}} \tag{19}$$

The non-dimensional relation for the mass exchange as a function of angle is determined using the reference quantities.

$$\frac{d\hat{m}}{d\theta} = \underbrace{\frac{A_F\sqrt{RT_{ref}}}{\omega_0 V_{ref}}\sqrt{\frac{2}{1+\zeta}}}_{\Pi_M}\sqrt{\hat{\rho}\Delta\hat{p}}b \tag{20}$$

The first part of the non-dimensional coefficient Π_M is homogeneous to the inverse of flow velocity, the second term homogeneous to the speed of sound.

$$\underbrace{\left(\frac{A_F}{\omega_0 V_{ref}}\right)}_{\text{flow velocity}} \cdot \underbrace{\left(\sqrt{\kappa R T_{ref}}\right)}_{\text{speed of sound}} = \frac{1}{Ma}$$

The non-dimensional parameter Π_M can be considered the product of the inverse of a Mach-number with an inverse flow friction coefficient. When Π_M is large, the mass flow rate per unit angle will be “large” and the flow is not hindered to pass through the volumes. This is the case at low friction coefficients and Mach-numbers. “Small” Π_M will constrain the mass flow rate and result in larger pressure differences.

The flow friction coefficient ζ is a function of Reynolds number. This makes dependent Π_M on crank angle. For the design process Π_M will be considered independent of crank angle.

4.3 Regenerator temperature

In a simplified view the average wall temperature of the volumes V_1 and V_2 can be considered fixed and equal to the average gas temperature if good isolation is applied. The wall temperatures of the heat exchangers HE_1 and HE_2 can also be considered fixed as they are the temperatures at which heat is supplied to or extracted from the system. This still leaves the temperature of the regenerator material variable over the cycle. An open design parameter is still the regenerator mass since surface area is defined by the NTU-number. A “small” regenerator mass will cause large wall temperature changes in a cycle and “small” pressure loss, while a “large” regenerator mass will result in “small” wall temperature amplitudes and “large” pressure loss. The temperature change of the regenerator material can be quantified by a heat transfer balance from the gas to the regenerator.

$$m_{rg} c_{p,rg} \frac{dT_{w,rg}}{dt} = \frac{dQ}{dt} \tag{21}$$

The non-dimensional version of the wall Temperature fluctuation is obtained by assuming convective heat transfer and division by the reference temperature.

$$\frac{d\hat{T}_{w,rg}}{dt} = \underbrace{\frac{\alpha A_w}{\omega_0 m_{rg} c_{p,rg}}}_{\Pi_{rg}} (\hat{T}_g - \hat{T}_{w,rg}) \tag{22}$$

The resulting non-dimensional parameter Π_{rg} is interpreted as the ratio of heat transfered to the regenerator divided by the capacity of the regenerator to store heat. As the rhs of equation 22 should remain as small as possible, this parameter should take small values as this limits the regenerator wall temperature amplitude.

Essential for the heat transfer to the regenerator is as well the temperature difference $(\hat{T}_g - \hat{T}_w)$. The differential for the energy conservation (eq. 15) can be transfered into a differential equation for the gas temperature in the regenerator.

$$\frac{d\hat{T}_g}{d\theta} = \frac{NTU}{\hat{m}} (\hat{T}_{w,rg} - \hat{T}_g) + \frac{1}{\hat{m}} (\kappa \hat{T}_e - \hat{T}_g) \frac{d\hat{m}}{d\theta} \tag{23}$$

The difference between the temperatures $\Delta\hat{T} = (\hat{T}_w - \hat{T}_g)$ can be formulated in a differential equation taking the difference between eq. 22 and eq. 23.

$$\frac{d\Delta\hat{T}}{d\theta} = (\Pi_{rg} - \frac{NTU}{\hat{m}}) \Delta\hat{T} + \frac{1}{\hat{m}} (\kappa \hat{T}_e - \hat{T}_g) \frac{d\hat{m}}{d\theta} \tag{24}$$

4.4 Summary of design parameters

The simplified model has 14 independent major design parameters listed below.

4.4.1 Geometric design parameters

The geometric design parameters listed here are of importance for the thermodynamic design. Other equally important parameters like bore to stroke ratio for the volumes V_1 and V_2 are neglected at this point.

$$\Delta\theta = \text{phase angle between displacement volumes} \tag{25}$$

$$\hat{V}_{d1} = \frac{V_{d1}}{V_{ref}} \tag{26}$$

$$\hat{V}_{d2} = \frac{V_{d2}}{V_{ref}} \tag{27}$$

$$\hat{V}_{Rg} = \frac{V_{Rg}}{V_{ref}} \tag{28}$$

$$\hat{V}_{HE_1} = \frac{V_{HE_1}}{V_{ref}} \tag{29}$$

$$\hat{V}_{HE_2} = \frac{V_{HE_2}}{V_{ref}} \tag{30}$$

The geometric design parameters are not independent. The reference Volume V_{ref} can not be chosen independently of the sum of all volumes since this would alter all other reference quantities by the reference mass m_{ref}. Therefore the sum of all non-dimensional volumes is here unity.

$$\hat{V}_1 + \hat{V}_{HE_1} + \hat{V}_{Rg} + \hat{V}_{HE_2} + \hat{V}_2 = 1$$

In other publications different volume ratio are taken as independent parameters. Such ratios can be easily transfered to the non-dimensional volumes.

4.4.2 Thermodynamic design parameters

In order to reduce the number of design parameters, further assumptions are made. Since the wall temperature of volumes V_1 and V_2 are assumed to be identical to average gas temperature heat transfer is neglected in V_2 and V_1 . This corresponds to the assumptions of the "adiabatic" model of Urieli (Urieli, 2014), and results in three NTU parameters for the volumes with heat transfer.

(31)

$$NTU_{HE\,1} = \frac{\alpha A_{w,HE\,1}}{\omega_0 m_{ref}\, c_v} \tag{31}$$

$$NTU_{HE\,2} = \frac{\alpha A_{w,HE\,2}}{\omega_0 m_{ref}\, c_v} \tag{32}$$

$$NTU_{Rg} = \frac{\alpha A_{w,Rg}}{\omega_0 m_{ref}\, c_v} \tag{33}$$

$$\Pi_{rg} = \frac{\alpha A_{w,Rg}}{\omega_0 m_{rg} c_{p,rg}} \tag{34}$$

$$\Pi_{M,V_1 HE\,1} = \frac{A_{F,V_1\,HE\,1}\sqrt{RT_{ref}}}{\omega_0 V_{ref}}\sqrt{\frac{2}{1+\zeta_{V_1\leftrightarrow HE\,1}}} \tag{35}$$

$$\Pi_{M,V_2 HE\,2} = \frac{A_{F,V_1 HE\,1}\sqrt{RT_{ref}}}{\omega_0 V_{ref}}\sqrt{\frac{2}{1+\zeta_{HE\,1\leftrightarrow Rg}}} \tag{36}$$

$$\Pi_{M,HE\,1 Rg} = \frac{A_{F,V_1 HE\,1}\sqrt{RT_{ref}}}{\omega_0 V_{ref}}\sqrt{\frac{2}{1+\zeta_{Rg\leftrightarrow HE\,2}}} \tag{37}$$

$$\Pi_{M,HE\,2 Rg} = \frac{A_{F,V_1 HE\,1}\sqrt{RT_{ref}}}{\omega_0 V_{ref}}\sqrt{\frac{2}{1+\zeta_{HE\,2\leftrightarrow V_2}}} \tag{38}$$

The major design parameters can only in theory be chosen independently. In real geometric design the flow cross section A_F, the wetted area for heat transfer A_w, the volume V, the heat transfer coefficient α and the loss coefficient ζ can not be chosen completely independently. This will be adressed in the mechanical design section 6.

4.5 Grouping of design parameters

Recalling the design specification and taking the lower temperature T_{low} as reference temperature, the non-dimensional temperatures will be $\widehat{T}_{low} = 1$ and $\widehat{T}_{high} = 1.22$. Since the non-dimensional temperatures differ not too much, a symmetrical design of the heat exchangers is tempted. This feature needs to be validated within the design process. With the constraint of symmetrical volume displacement and symmetrical heat exchangers, the number of independent design parameters can be reduced.

$$\widehat{V}_1 = \widehat{V}_2 \qquad \widehat{V}_{He,1} = \widehat{V}_{He,2} \tag{39}$$

$$NTU_{He,1} = NTU_{He,2} \tag{40}$$

$$\Pi_{M,V_1 HE_1} = \Pi_{M,V_2 HE_2} \qquad \Pi_{M,HE_1 Rg} = \Pi_{M,RgHE_2} \tag{41}$$

This set of parameters is used in the simulation to investigate the possible design space.

5 NUMERICAL EXPLORATION OF DESIGN SPACE

The 14 non-dimensional quantities leave a very large design space. Using the symmetry constrains still 6 independent design parameter remain. This can not be explored by experiment or simulation. Simulation allows however to investigate in a cheap way trends for some parameters when sensible engineering choices are made. For the exploration of the design space a nodal analysis program using the design parameters as input is used. The nodal design program is briefly described in the following. The nodal program is validated against a stirling cooler available for measurement and compared to the ideal adiabatic analysis of Urieli (Urieli 2014).

The validated nodal simulation program is then used to explore the design space of the non-dimensional parameters.

5.1 Conservation equations

Based on the previously identified design parameters a physical model for the computation of the non-dimensional energy, temperature, pressure, mass and heat transfer for every volume is used. The model is based on solving the differential equations for the non-dimensional volume energy (eq. 15), the non-dimensional volume mass (eq. 20) and the non-dimensional wall temperature (eq. 23). In the adiabatic volumes (V_1 ,V_2) and the heat exchangers (HE_1,HE_2) the wall temperature equation is not solved. Non-dimensional temperature T and pressure p are computed using the state equations 11,12.

1. mass conservation:

$$\frac{d\widehat{m}_{V1-He1}}{d\theta} = \Pi_{M,V1-He1}\sqrt{\hat{\rho}\Delta\hat{p}_{V1-He1}} \tag{42}$$

$$\frac{d\widehat{m}_{He1-Rg}}{d\theta} = \Pi_{M,He1-Rg}\sqrt{\hat{\rho}\Delta\hat{p}_{He1-Rg}} \tag{43}$$

$$\frac{d\widehat{m}_{Rg-He2}}{d\theta} = \Pi_{M,Rg-He2}\sqrt{\hat{\rho}\Delta\hat{p}_{Rg-He2}} \tag{44}$$

$$\frac{d\hat{m}_{He2-V2}}{d\theta} = \Pi_{M,He2-V2}\sqrt{\hat{\rho}\Delta\hat{p}_{He2-V2}} \tag{45}$$

The pressure difference needs to be positive under the square root. Therefore the absolute value of the pressure difference is taken. Density p and temperature T are taken as conditional values from the volume with the higher pressure.

2. energy conservation:

$$\frac{d\widehat{U_{V1}}}{d\theta} = -(\kappa - 1)\hat{p}\frac{d\hat{V}_{V1}}{d\theta} + \kappa\hat{T}_{V1-He1}\frac{d\hat{m}_{V1-He1}}{d\theta} \tag{46}$$

$$\frac{d\hat{U}_{He1}}{d\theta} = NTU_{He1}(\hat{T}_{He1,w} - \hat{T}_{He1,g}) - \kappa\hat{T}_{V1-He1}\frac{d\hat{m}_{V1-He1}}{d\theta} + \kappa\hat{T}_{He1-Rg}\frac{d\hat{m}_{He1-Rg}}{d\theta} \tag{47}$$

$$\frac{d\hat{U}_{Rg}}{d\theta} = NTU_{Rg}(\hat{T}_{Rg,w} - \hat{T}_{Rg,g}) - \kappa\hat{T}_{He1-Rg}\frac{d\hat{m}_{He1-Rg}}{d\theta} + \kappa\hat{T}_{Rg-He2}\frac{d\hat{m}_{Rg-He2}}{d\theta} \tag{48}$$

$$\frac{d\hat{U}_{He2}}{d\theta} = NTU_{He2}(\hat{T}_{He2,w} - \hat{T}_{He2,g}) - \kappa\hat{T}_{Rg-He2}\frac{d\hat{m}_{Rg-He2}}{d\theta} + \kappa\hat{T}_{He2-V2}\frac{d\hat{m}_{He2-V2}}{d\theta} \tag{49}$$

$$\frac{d\widehat{U_{V2}}}{d\theta} = -(\kappa - 1)\hat{p}\frac{d\hat{V}_{V1}}{d\theta} + \kappa\hat{T}_{He2-V2}\frac{d\hat{m}_{He2-V2}}{d\theta} \tag{50}$$

3. regenerator wall temperature

For the design space sweep only the wall temperature of the regenerator is considered. For the comparison to the experimental setup, all wall temperature equations are solved.

$$\frac{d\hat{T}_{w,rg}}{dt} = \Pi_{rg}(\hat{T}_g - \hat{T}_{w,rg}) \tag{51}$$

Dimensional quantities are obtained by multiplication with the reference quantities (eq. 4).

5.2 Simulation platform

The differential equations are solved with a 4th order runge-kutta scheme with fixed angle advance (Deuflhard 2002). Depending on the chosen non-dimensional parameters from a few tenth to several hundred cycles need to be computed to reach a steady cycle solution. Commercial software packages usually do not allow performing non-dimensional simulations. Therefore the model is implemented in ANSI-C and run on a notebook computer. A simulation time for several hundred cycles is in the order of minutes.

5.3 Validation of simulation on experimental setup

The experimental setup consists of an D-type machine with a phasing of $\delta\theta = 90°$. Temperature and pressure are measured between the variable volumes and the heat exchangers, see fig. 3.

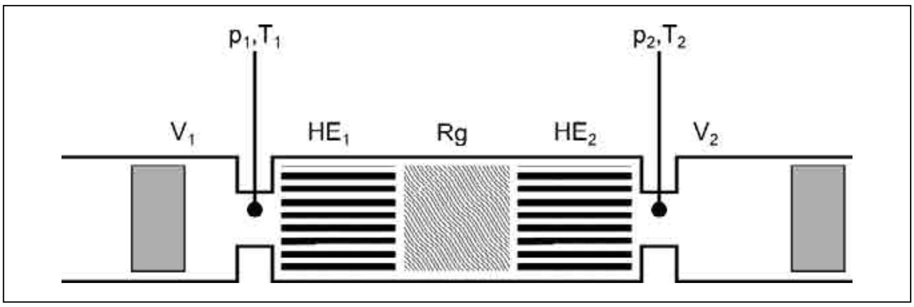


FIGURE 3. Measurement points (1) and (2) to obtain pressure and temperature of the working gas.

FIGURE 4. Image of experimental setup. The crank shaft is driven by a belt from the electric motor. Cylinders of V_1 and V_2 are mounted on top of heat exchangers and regenerator. First heat exchanger HE_1, regenerator $\boldsymbol{Rg}$ and second heat exchanger HE_2 are seperatated by insulation material (white).

The simulation is validated at two operating points of the machine. The non-dimensional parameters of the experimental operating points are given in table 2. On the contrary to the design parameters, the heat transfer to the volumes V_1 and V_2 is included in the simulation of the experimental setup.

TABLE 1. Non-dimensional volumes of experimental setup

Vol.	$\widehat{V_d}$	$\widehat{V_r}$
V_1	0.1625	0.01
HE_1		0.1725
$\boldsymbol{Rg}$		0.3250
HE_2		0.1725
V_2	0.1625	0.01

TABLE 2. Non-dimensional values for the experimental setup operating at frequencies $\boldsymbol{f = 5\,Hz}$ and $\boldsymbol{f = 10\,Hz}$

Vol.	NTU	Π_{rg}	NTU	Π_{rg}
	(5 Hz)	(5 Hz)	(10 Hz)	(10 Hz)
V_1	0.00034	0.0002	0.00017	0.0001
HE_1	0.002	0.002	0.017	0.001
$\boldsymbol{Rg}$	0.088	0.012	0.044	0.006
HE_2	0.002	0.002	0.017	0.001
V_2	0.00034	0.0002	0.00013	0.0001

Flow	Π_M	Π_M
	(5 Hz)	(10 Hz)
$V_1 \leftrightarrow HE_1$	2.4	1.2
$HE_1 \leftrightarrow Rg$	0.6	0.3
$Rg \leftrightarrow HE_2$	0.6	0.3
$HE_2 \leftrightarrow V_2$	2.4	1.2

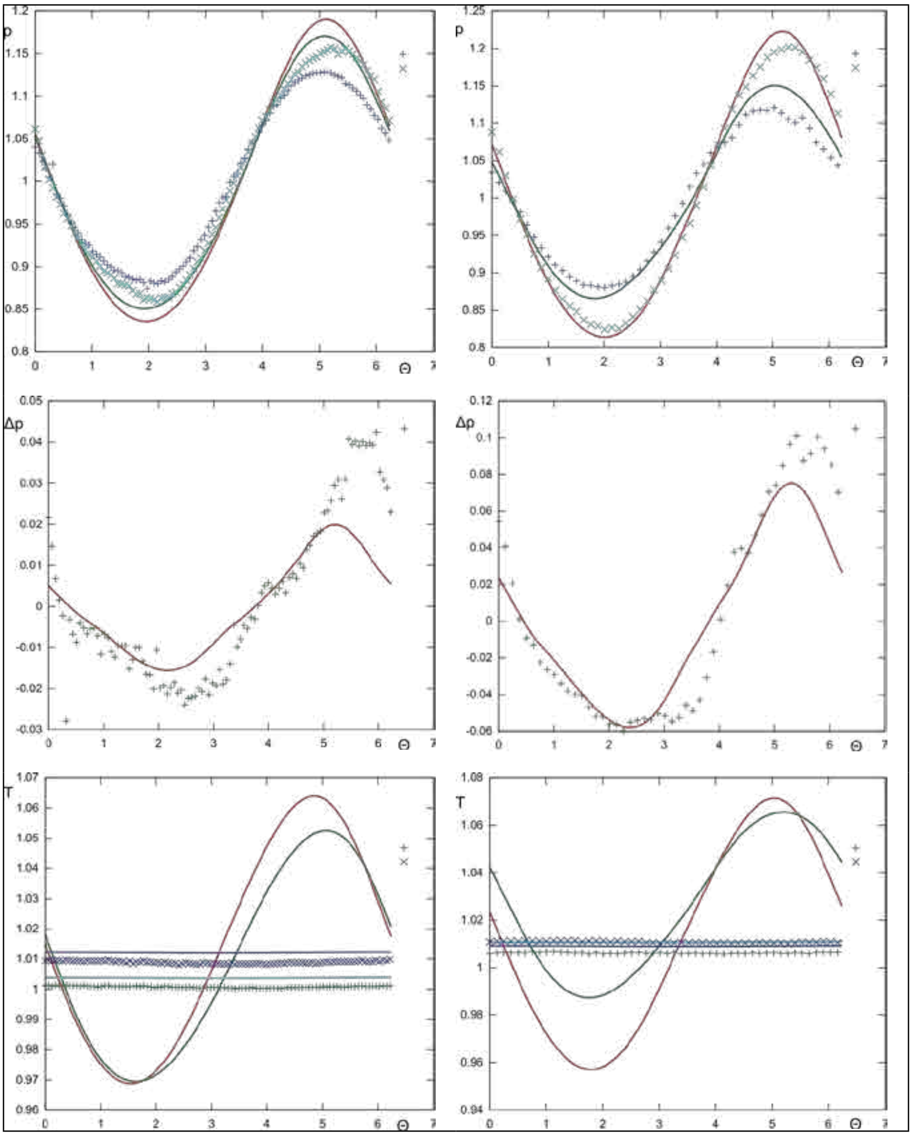


FIGURE 5. Comparison of measured and simulated values in position 1 and position 2. Top figures: non-dimensional pressure, 5 Hz left, 10 Hz right. Middle figures: non-dimensional pressure difference across heat exchangers and regenerator, 5 Hz left, 10 Hz right. Bottom figures: non-dimensional temperature and wall temperature , 5 Hz left, 10 Hz right. Simulation: continous line, Experiment: symbols

Fig. 5 shows a comparison of non dimensional pressure $\hat{p}$, non dimensional pressure difference $\delta\hat{p}$ and non dimensional temperature $\hat{T}$ between the experimental setup and the nodal simulation on the operating points given in table 2. When comparing the results, it has to be kept in mind that the volume displacement is not modeled correctly as the numerical displacement (eq.1) differs from the physical displacement (eq.52) with a real crank law including the conrod length L (A_1 piston surface, D displacement).

$$V_1(\theta) = A_1 \left(\frac{D}{2} \cos\theta + L \sqrt{1 - \left(\frac{D}{2L}\right)^2 \sin^2\theta} \right) \tag{52}$$

Quantities for the flow parameter Π_M in table 2 could only be estimated as the loss coefficient ζ was not measured for the experimental setup before.

The top figures show that the simulation predicts an increase in flow resistance at higher frequencies resulting in a larger pressure difference. The magnitude of the pressure difference is the same in simulation as in experiment. Figures in the middle give the detail of the pressure difference.

The bottom figures show the temperatures from the PT100 sensors implemented to measure the temperature (symbols). The lines show the simulated gas temperature and the simulated wall temperature. The gas temperature amplitude largely exceeds the temperature difference between the two heat exchangers.

The simulation results with the working parameters 2 show an interesting behavior of the experimental setup. Table 3 shows that the mechanical work input is given to the two heat exchangers and variable volume walls. The intention of a cooler to lift heat from lower temperature to higher temperature is not reached here.

TABLE 3. Results of simulation with experimental setup, $\boldsymbol{d\widehat{W}} = \Sigma(\boldsymbol{\kappa - 1}) \int \boldsymbol{\hat{p}\, d\hat{V}}$ is the non-dimensional cycle integrated work balance, and $\boldsymbol{d\hat{Q}}$ is the cycle integrated transfered heat of each volume.

operating point	$d\hat{W}$	$d\hat{Q}_1$	$d\hat{Q}_2$	$d\hat{Q}_{V1}$	$d\hat{Q}_{V2}$
5 Hz	0.00377	-0.00138	-0.00135	-0.000521	-0.000523
10 Hz	0.0105	-0.00718	-0.00218	-0.000249	-0.000920

This behavior can be confirmed by running the experimental setup over a certain time and comparing to the equivalent number of cycles in the simulation.

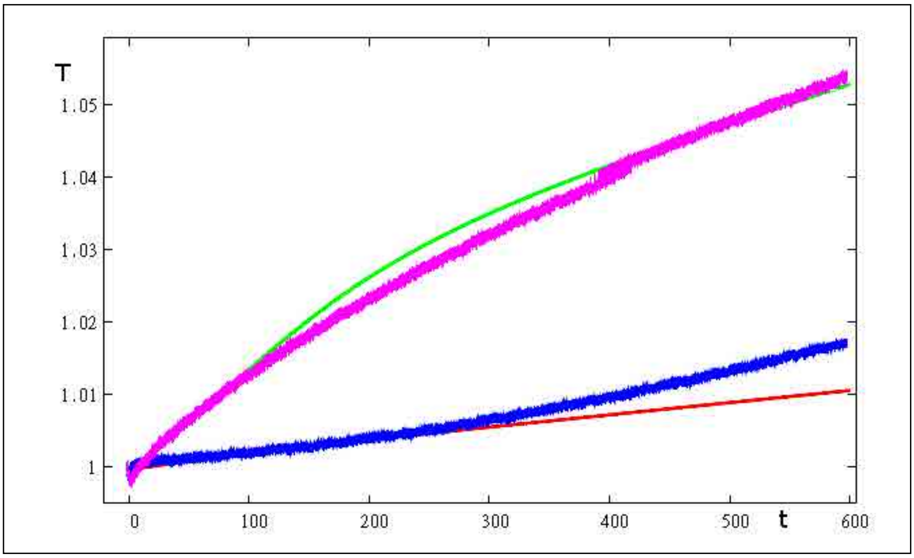


FIGURE 6. temporal heat up of experimental setup compared to simulation (green line wall temperature simulation T1, red line wall temperature simulation T2,magenta and blue corresponding experimental values, time axis in seconds)

The result is shown in fig. 6. The curves from simulation and measurement follow the same trends. In the simulation conduction from hot to cold heat exchanger is not included and leads to different results.

5.4 Exploration of design space

During the numerical exploration of the design space, the underlying constraints and relations limiting the realization are not considered. This will be done in the mechanical design section. The temperature ratio is fixed to $T_{high}/T_{low} = 1.2$. The phase lag between the two volumes $\hat{V}_1$ and $\hat{V}_2$ is fixed to $90°$.

5.4.1 Design range for NTU

With the non-dimensional energy conservation (eq. 15) and the small temperature difference ($\hat{T}_{low} = 1$ and $\hat{T}_{high} = 1.22$) between hot and cold side the NTU has to be at least unity to reach a significant heat transfer. Organ (Organ, 2014) states that NTU of at least 2.5 should be considered. The experimental units has significantly smaller NTU numbers (see table 2).

Therefore the explored range of NTU is defined from 0.1 to 10. Identical NTU numbers are chosen for the regenerator and the heat exchangers.

5.4.2 Design range for $\boldsymbol{\Pi_{rg}}$

The differential equations for the regenerator material temperature (eq. 22) and for the temperature difference between regenerator gas and material (eq. 24 suggest that small Π_{rg} are beneficial for the operation of the cooler.
The explored range of Π_{rg} is defined from 0.001 to 0.01.

5.4.3 Design range for $\boldsymbol{\Pi_M}$

Mass transfer is limited when Π_M becomes small. Values of Π_M larger than unity are beneficial for the operation of the cooler since it limits the pressure losses. It is however not possible to build a machine without losses. Therefore it is interesting what values of Π_M are tolerable for the operation of the Stirling cooler. The differential equations become stiff for Π_M larger than one. In the $\Pi_M >> 1$ limit, pressure in the system is uniform. In the limit of very small pressure losses the simulation model can be changed accordingly and uniform pressure can be assumed through all volumes. The range of Π_M is defined from 0.1 to 1.

5.4.4. Design range for $\boldsymbol{\widehat{V}_d}$

As a geometric parameter for the design the non-dimensional total displaced volume is considered.

$$\widehat{V}_d = \widehat{V}_{d1} + \widehat{V}_{d2} \tag{53}$$

Low temperature stirling engines have a small $\widehat{V}_d$ whereas high temperature ratio stirling engines tend to rather high values. A range of $\widehat{V}_d$ from 0.1 to 0.8 is considered here. The remaining volume is split equally in thirds for the two heat exchangers and the regenerator.

5.4.5 Simulation of cycles

For the scan of the design space a combination of non-dimensional parameters is chosen and the simulation is carried out to conversion. The combination of non-dimensional parameters is chosen by two ways. First, a sweep of design parameters on a grid for NTU, Π_{rg}, Π_M and $\hat{V}_d$ is carried out. In order not to leave out systematically an interesting design area, in a second step the combination of design parameters is chosen with a random function. A total of 3000 combinations of non-dimensional parameters were simulated to convergence.

This approach was chosen since a fine grained sweep of all parameter combinations exceeds the computational resources of a desktop computer.

5.4.6. Evaluation of design parameters

For the choice of the design parameters, target parameters need to be evaluated. In order for the cooler to be competitive, it needs to have a coefficient of performance (COP) superior to 2 when it comes to cooling. The coefficient of performance is defined as the ratio of heat extracted at lower temperature dQ_0 compared to the work input dW.

$$d\hat{Q}_0 = \int NTU_{He,2}(\hat{T}_g - \hat{T}_w) \tag{54}$$

$$d\hat{W} = d\hat{W}_1 + d\hat{W}_2 = -(\kappa - 1)\int \hat{p}_1\, d\hat{V}_1 - (\kappa - 1)\int \hat{p}_2\, d\hat{V}_2 \tag{55}$$

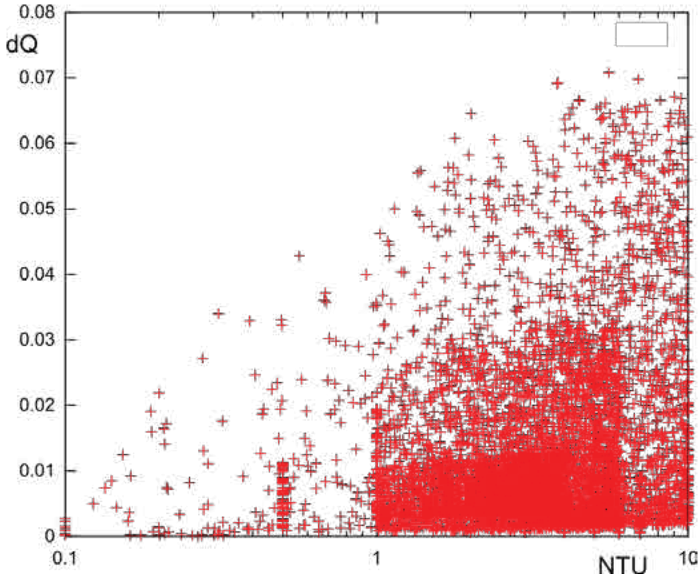
$$COP = \frac{d\hat{Q}_0}{d\hat{W}} \tag{56}$$

Another important factor is the power density of the machine, that is how much heat it can lift per cycle. This corresponds to $d\hat{Q}_0$. Since this quantity is already non-dimensional with reference to energy content of the working gas at lower temperature, it is well suited for choosing the design parameters.

The results of the design space sweep is presented by projecting the results for COP and $d\hat{Q}$ on one non-dimensional parameter. The less informing plots of for the parameters Π_{rg} and Π_M are presented in appendix 8

5.4.7. Results of design space sweep

For the ease of visualization, the results are given as scatter plots as a function of the non dimensional parameter. In a second step design criteria are applied to limit possible choices of parameters.



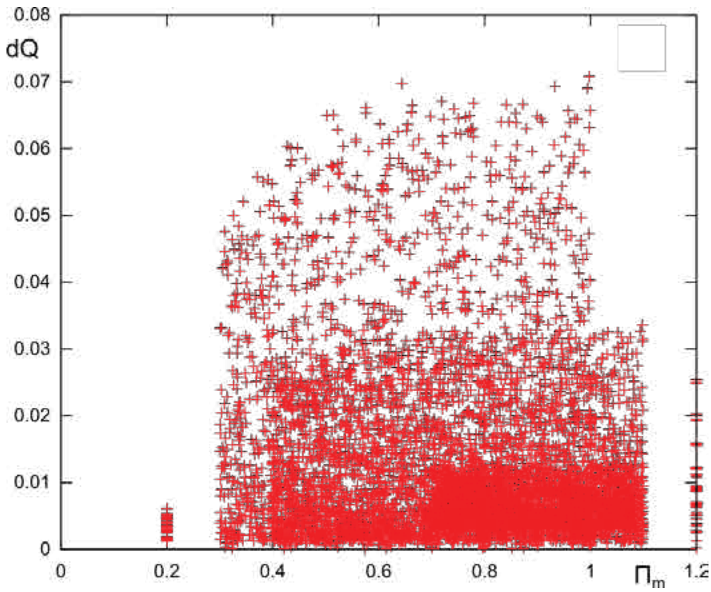


Figure 7. Scatter plot of COP (top) and dQ_0 (bottom) vs NTU.

Figure.7 shows the COP and $d\hat{Q}$ as a function of the heat transfer number NTU. The white areas on the plot indicate that for lower NTU $(NTU < 2)$ it is not possible to achieve good COP values. The lower graph shows higher NTU are beneficial for the power density of the unit.

Non-dimensional heat transfer is limited by the temperature swing $\Delta\hat{T}$ and NTU.

$$d\hat{Q} \approx NTU\Delta\hat{T} \tag{57}$$

In common stirling engines the temperature difference between heat exchanger wall and working fluid is roughly 1/10 of the working fluid temperature different $T_{high} - T_{low}$ (see Lane (N.W. Lane, 1997)). This makes $\hat{T} \approx 0.02$ in the case of the stirling cooler as an upper limit. For an NTU of order of unity this limits the heat transfer $d\hat{Q} < 0.02$. Fig.12 does not allow a conclusion on the choice of Π_{rg}. Π_{rg} concerns mainly the regenerator mass whereas NTU concerns the heat transfer to the regenerator. This result may be biased due to the choice of identical heat transfer numbers for regenerator and heat exchangers.

The influence of the mass transfer by Π_m is shown in fig.13. The scatter plot suggests the larger values of Π_m seem to be beneficial for the COP.

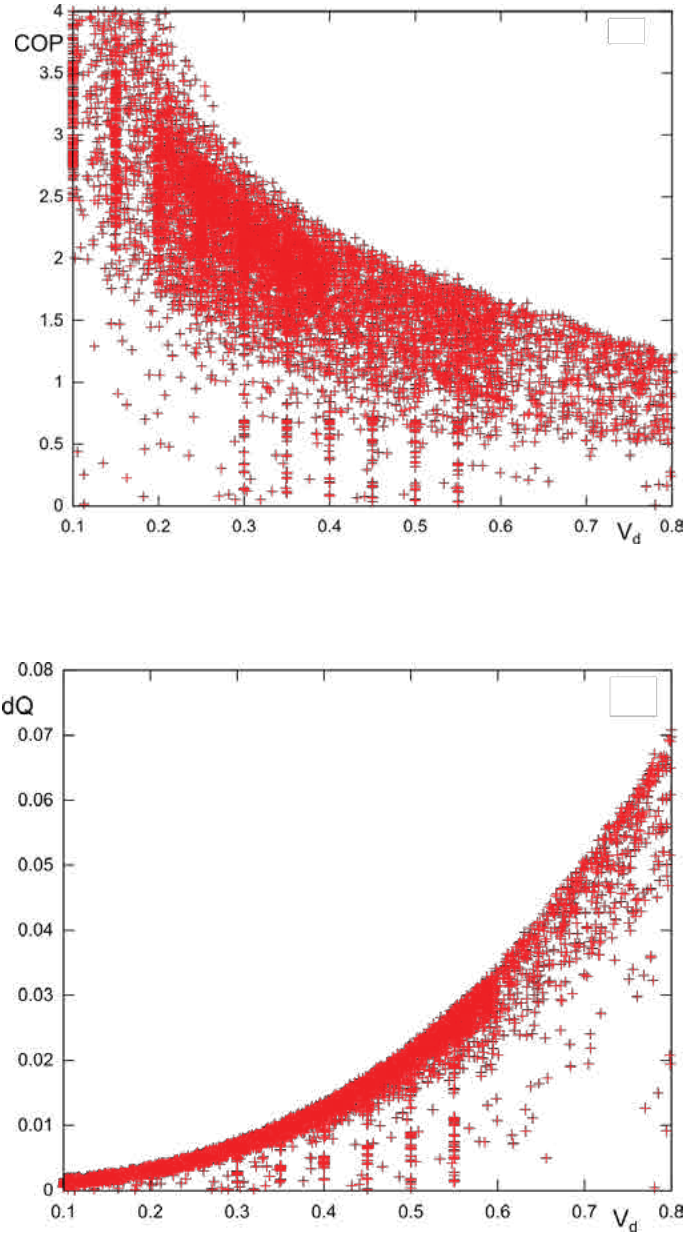


Figure 8. Scatter plot of COP (top) and dQ_0 (bottom) vs V_d .

The influence of the geometric design parameter $\hat{V}_d$ is illustrated in fig.8. Concerning the coefficient of performance, small values of $\hat{V}_d$ are beneficial. For the power density the opposite is the case. For the design a tradeoff between COP and power density has to be made. For a COP of 2, a non-dimensional heat transfer $d\hat{Q}_0 > 0.01$ seems to be realistic.

Considering the Beale number (see Organ (Organ, 2014)) for a temperature ratio of $T_{high}/T_{low} = 3$

$$Be = \frac{P}{p_{ref}V_{swept}\omega} \approx 0.15 \tag{58}$$

and making some additional assumptions on the mechanical efficiency of the systems a power density can be estimated.

$$\frac{P}{U_{ref}\omega} = 0.15(\kappa - 1.0)/2.0 = 0.03 \tag{59}$$

In the present case the temperature ratio $T_{high}/T_{low} = 1.2$ and a Carnot factor scaling would lead to possible power density of

$$\frac{P}{U_{ref}\omega} \approx 0.01 \tag{60}$$

This makes the non-dimensional heat transfer $d\hat{Q} = 0.01$ a reasonable choice.

5.4.8. Conditional scatter plots with design criteria

When the simulation results are filtered with respect to $COP > 2$ and $d\hat{Q}_0 > 0.01$ conditional scatter plots help to visualize the remaining design space.

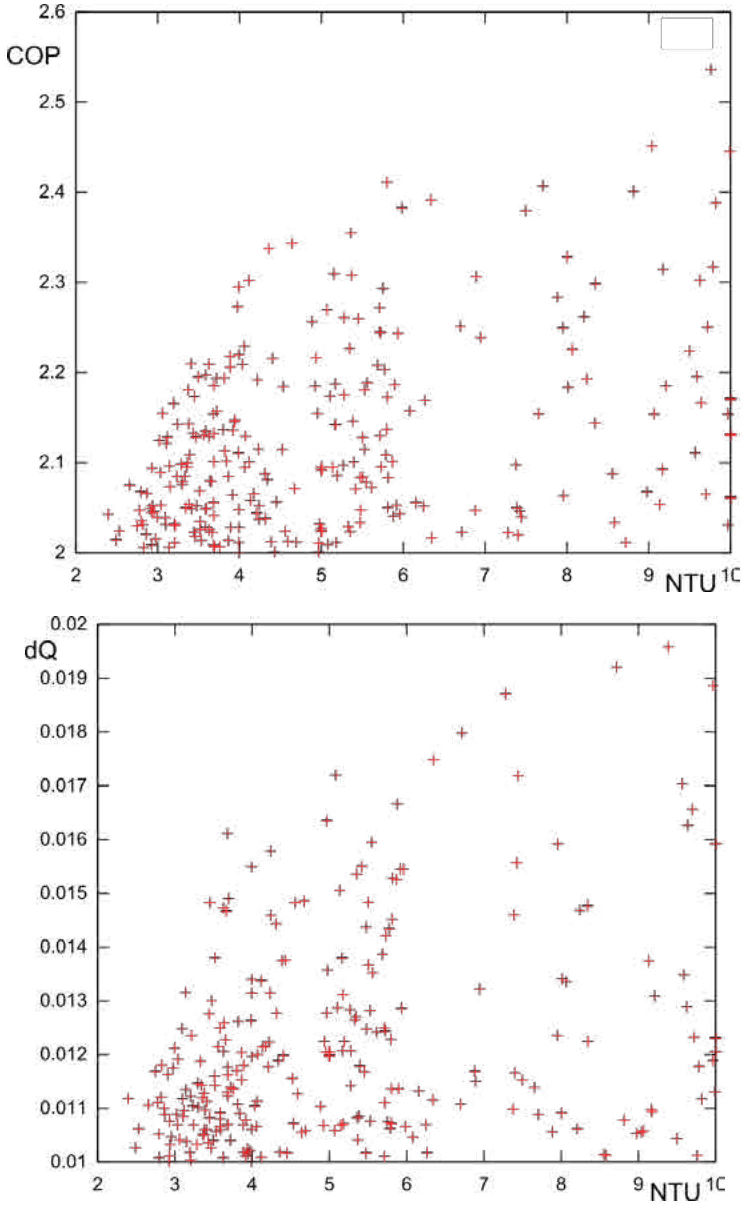


Figure 9. Conditional $(\boldsymbol{COP} > \mathbf{2}, \boldsymbol{d\hat{Q}_0} > \mathbf{0.01})$, scatter plot of COP and $d\hat{Q}$ (bottom) vs NTU.

Fig.9 shows that according to the simulation, $NTU >= 3$ are necessary to achieve beneficial COP and $d\hat{Q}_0$. This gives a first design criteria for the NTU range. The result supports the statement of Organ (Organ, 2014) that $NTU > 2.5$ should be considered.

The choice of Π_{rg} in the design of the machine is difficult to make based on the scatter plots in fig.14. The density of points suggest however that values smaller 0.1 might be beneficial.

The white areas of the conditional scatter plots in fig. 15 suggest that the mass flow parameter Π_m should take values larger 0.9 during operation. For the extension of the operating range targeting higher Π_m should not harm the performance.

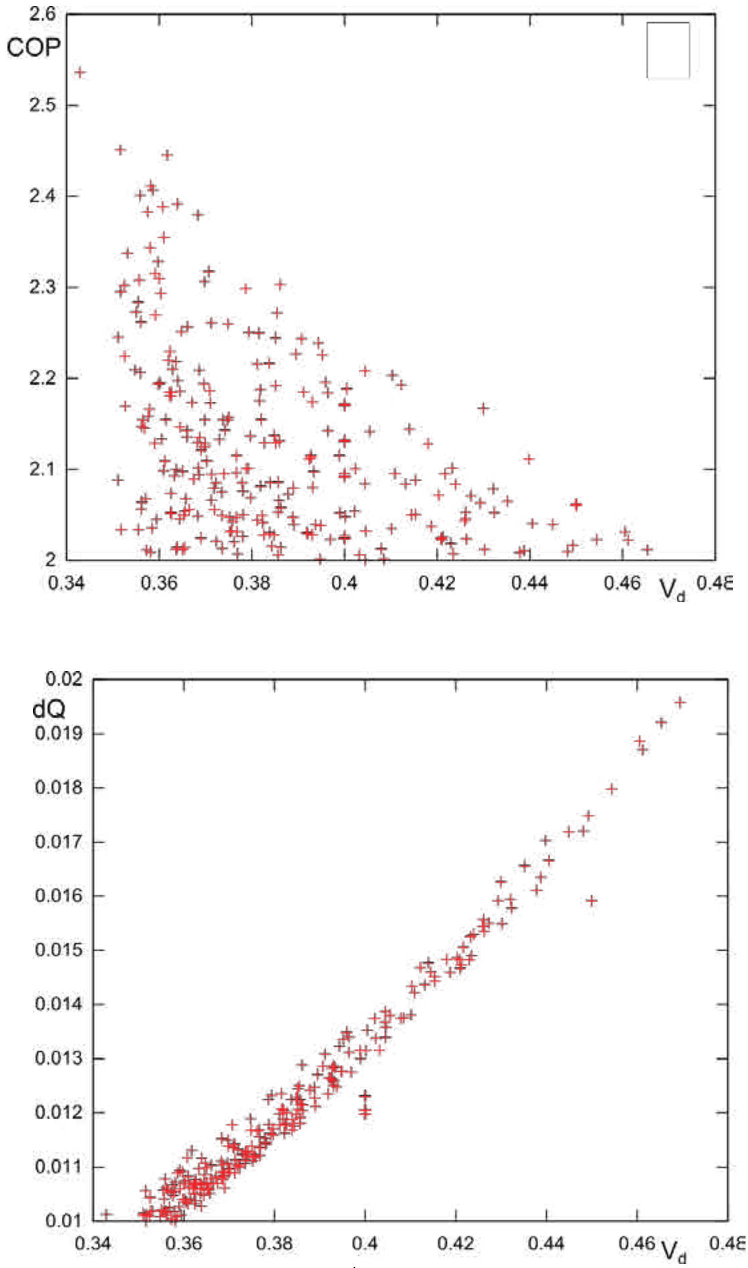


Figure 10. Conditional $(COP > 2, d\hat{Q}_0 > 0.01)$ scatter plot of COP and $d\hat{Q}_0$ (bottom) vs $\hat{V}_d$.

Fig.10 shows that the filtering condition strictly limits the choice of non-dimensional displacement volume. A choice of $\hat{V}_d = 0.4$ looks like a good compromise concerning COP and power density.

6 PRELIMINARY MECHANICAL DESIGN

For the mechanical design, the design parameters extracted from simulation are summarized. The consequences of chosen NTU to different design options are discussed in the following sections.

6.1 Implications of design space parameters on tube heat exchangers

For a heat exchanger consisting of nbr identical tubes of diameter d and length l the following relations for heat exchanger surface and volume hold.

$$A_w = \pi d \cdot l \cdot nbr \quad (61)$$

$$V_{He} = \frac{\pi}{4} d^2 \cdot l \cdot nbr \quad (62)$$

$$A_w / V_{He} = \frac{4}{d} \quad (63)$$

The following thermodynamical relation for the working gas is used in the determination of tube diameter.

$$m_{ref} c_v = \frac{c_v}{R} \frac{p_{ref} V_{ref}}{T_{ref}} = \frac{1}{\kappa - 1} \frac{p_{ref} V_{ref}}{T_{ref}} \quad (64)$$

The definition of NTU can be used to determine the necessary tube diameter.

$$\begin{aligned} NTU &= \frac{\alpha A_W}{\omega_0 m_{ref} c_v} \\ &= \frac{\alpha}{\omega_0} \frac{A_W}{V_{He}} \frac{(\kappa - 1) T_{ref} \hat{V}_{HE}}{p_{ref}} \\ &= \frac{\alpha}{\omega_0} \frac{4}{d} \frac{(\kappa - 1) T_{ref} \hat{V}_{HE}}{p_{ref}} \end{aligned} \quad (65)$$

For laminar flow in circular tubes and ducts the Nusselt number is roughly $Nu = 4$. The expression for the heat transfer coefficient α is:

$$\alpha = Nu \cdot \frac{\lambda}{d} \quad (66)$$

The resulting NTU relation for laminar tube flow only depends on the tube diameter d as a geometric variable. A more general view on the characteristic length scale is found in the appendix 9.

$$NTU = \frac{Nu\lambda}{\omega_0} \frac{4}{d^2} \frac{(\kappa - 1) T_{ref} \hat{V}_{HE}}{p_{ref}} \tag{67}$$

The cooler should be designed to work at a certain operating frequency $\omega_0 = 2\pi f$. The relation of tube diameter as a function of operating frequency is given in fig. 11. Conductivity $\lambda = 0.0279\ W/mK$ of ambient air $p_{ref} = 10^5\ Pa$ $T_{ref} = 293\ K$ has been used to obtain the data.

$$d = \sqrt{\frac{4Nu\lambda}{\omega_0 NTU} \frac{(\kappa - 1) T_{ref} \hat{V}_{HE}}{p_{ref}}} \tag{68}$$

$$d = \frac{4\alpha}{\omega_0 NTU} \frac{(\kappa - 1) T_{ref} \hat{V}_{HE}}{p_{ref}} \tag{69}$$

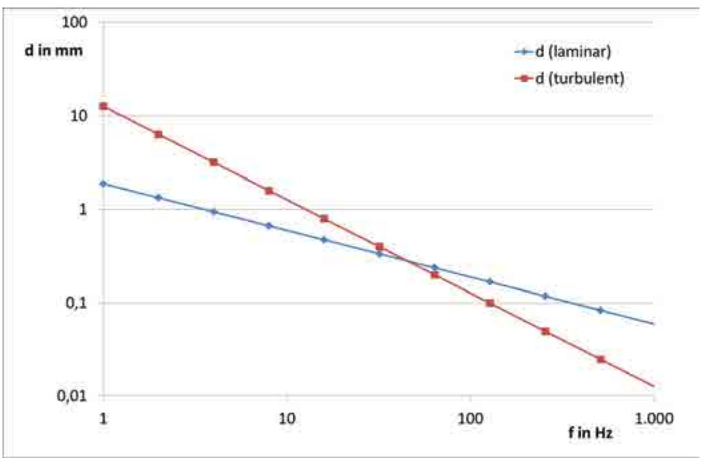


Figure 11. Tube diameter as a function of operating frequency for laminar and turbulent (fixed $\boldsymbol{\alpha} = \mathbf{400}\, \boldsymbol{W/mK)}$) flow

Karabulut (2009) claims for his stirling engine heat transfer coefficients of $\alpha \approx 447\ W/m^2K$. The tube diameter can be computed from equation 69 and is displayed in fig. 11.

For engines running at operating frequencies superior to $10\ Hz$ this leads to very small tube diameters not suitable for manufacturing. Alternative heat exchanger designs need to be considered to obtain designs that can be manufactured.
Starting with diameters d that can be manufactured and operating frequencies suitable for electric machines, a relation for the other depending parameters is needed. The relation for NTU can written as a function for pressure.

$$p_{ref} = \left(\frac{\alpha}{\omega_0}\right) \underbrace{\left(\frac{A_{He}}{V_{HE}}\right)}_{4/d} \widehat{V}_{He}(\kappa - 1) \left(\frac{RT_{ref}}{NTU}\right) \tag{70}$$

Assuming reasonable values leads to low reference pressures. This leads on the other hand to very low power densities with big displacement machines due to the state equation and the definition of specific heat lift.

$$U_{ref} = m_{ref} c_v T_{ref} = (\kappa - 1) p_{ref} V_{ref} \tag{71}$$

$$dQ_0 = \frac{\dot{Q}_0}{\omega_0 U_{ref}} = \frac{\dot{Q}_0}{\omega_0 (\kappa - 1) p_{ref} V_{ref}} \tag{72}$$

This small diameter can be confirmed with existing machines. Organ (Organ, 2014) calculates hydraulic radii $R_H = A_w/V_{swept}$ of $0.16\ mm$ for the GPU-3 and $0.21\ mm$ for the Phillips MP1002CA stirling engine.

6.2 Heat exchanger flow length

The mass flow parameter Π_M can be used to determine the maximum flow length of heat exchanger and regenerator. For the sake of simplicity tube heat exchangers of nbr identical tubes of diameter d are considered. This can be replaced by any other type of exchanger geometry.

$$\Pi_M = \frac{A_F \sqrt{RT_{ref}}}{\omega_0 V_{ref}} \sqrt{\frac{2}{1 + \zeta}} \tag{73}$$

$$A_F = \frac{\pi}{4} d^2\, nbr \tag{74}$$

$$V_{ref} = V_{HE}/\hat{V}_{He} \tag{75}$$

$$V_{HE} = \frac{\pi}{4} d^2 l\, nbr \tag{76}$$

$$\frac{A_F}{V_{ref}} = \hat{V}_{HE} \frac{\pi/4d^2\, nbr}{\pi/4d^2 l\, nbr} = \hat{V}_{HE} \frac{1}{l} \tag{77}$$

This leads to a maximum flow length l.

$$l = \frac{\hat{V}_{HE}}{\Pi_M \omega_0} \sqrt{RT_{ref}} \sqrt{\frac{2}{1+\zeta}} \tag{78}$$

The loss coefficient ζ depends on the friction coefficient λ_M, tube diameter d and flow length l.

$$\zeta = \lambda_M \frac{l}{d} \tag{79}$$

For laminar flow the friction coefficient can be analytically expressed as a function of Reynolds number.

$$\lambda_M = 64/Re \tag{80}$$

Assuming values of $\zeta >> 1$ and taking sensible values for velocity v, diameter d and viscosity ν the maximum length can be approximated.

$$l \approx \left(\frac{\hat{V}_{HE}}{\Pi_M \omega_0} \sqrt{RT_{ref}} \sqrt{\frac{vd^2}{32\nu}} \right)^{2/3} \tag{81}$$

This expression yields maximum heat exchanger length of roughly $0{,}08\ m$ for $v = 10\ m/s$, $d = 1\ mm$, $\nu = 1{,}5 \cdot 10^{-5}\ m^2/s$ and $\hat{V}_{HE} = 0{,}2$..
The constraints concerning diameter d and length l of tube heat exchangers make it difficult to realize a stirling cooler with ordinary tube heat exchangers.

6.3 Alternatives to tube heat exchangers

Modern bar extrusion production allows to use finned tubes or microchannel inexpensively as heat exchangers.

Length and thickness of the fins can easily be computed depending on the material properties. Heat exchanger literature (see for instance Marek (R. Marek, 2010)) defines a finn parameter μ.

$$\mu = \sqrt{\frac{\alpha U}{\lambda A}} \tag{82}$$

U is the circumference of the fin
A is the foot cross section
λ_f is the conductivity of the fin material
α is the heat transfer coefficient
L is the hight of the fin
l is the length of the fin

The heat transfer efficiency of the fin is defined using height L and the fin parameter μ.

$$\eta_f = \frac{1}{\mu L} \tag{83}$$

For a very long rectangular fin, the minimum fin thickness δ can be obtained by considering a minimum efficiency of the fin.

$$\delta \approx L^3 \frac{2\alpha}{\lambda_f l} \tag{84}$$

6.4 Regenerator design

The regenerator design can be based on the non-dimensional parameters $NTU_{Rg} = 4$ and $\Pi_{rg} = 0.1$. Since both parameters contain heat transfer coefficient wetted area and frequency, the expression can be simplified by taking the quotient.

$$\frac{\Pi_{Rg}}{NTU_{Rg}} = \frac{\alpha A_w}{\omega_0 m_{rg} c_{p,rg}} \cdot \frac{\omega_0 m_{ref} c_v}{\alpha A_w} = \frac{m_{ref} c_v}{m_{rg} c_{p,rg}} \tag{85}$$

The product heat capacity of the regenerator $m_{rg}c_{p,rg}$ can be evaluated with the reference energy U_{ref} and reference temperature T_{ref}.

$$m_{rg}c_{p,rg} = \frac{NTU_{Rg}}{\Pi_{Rg}} \frac{U_{ref}}{T_{ref}} \tag{86}$$

With knowledge of wire gauze material, for instance stainless steel with cp = $500 J/kgK$, density $\rho = 7880\, kg/m^3$ the minimal necessary regenerator mass and volume can be determined.

6.5 Design space parameters and assumptions

The chosen design parameters are summarized in table 4.

Table 4. Design parameters obtained from simulation results

volume	$\hat{V}$	NTU	Π_{rg}
V_1	0.2		
V_2	0.2		
$V_{He,1}$	0.2	4	
$V_{He,2}$	0.2	4	
V_{Rg}	0.2	4	0.1

Flow	Π_M
$V_1 \leftrightarrow HE_1$	1.0
$HE_1 \leftrightarrow Rg$	1.0
$Rg \leftrightarrow HE_2$	1.0
$HE_2 \leftrightarrow V_2$	1.0

Table 5. Chosen design parameters with air at ambient pressure

$R_{gas} =$	$287\ J/kgK$
$c_v =$	$717\ J/kgK$
$\lambda =$	$0.0279\, W/mK$
$\kappa =$	1.4
$\nu =$	$1.5 \cdot 10^{-5}\ m^2/s$

$p_{ref} =$	$10^5\ Pa$
$T_{ref} =$	$273.0\ K$
$dQ =$	0.01
$f =$	$50\ Hz$
$v_{He} =$	$50\ m/s$

Air is chosen as working fluid for cost sensitive for a cost sensitive application target. Without supercharging ambient pressure is considered as reference pressure. The working frequency of $50\ Hz$ corresponds to some $3000\ RPM$ and is a typical rotational frequency used in vehicle motors. A flow velocity of $50 m/s$ is well below limiting Mach number and still not limiting the geometry sizes. This choice of design parameter is considered as a typical operating point. Altering the parameters within the possible limits of vehicle applications does significantly alter the findings.

6.6 Application of design parameters to 3kW MAC

For the MAC application a heat flux of $\dot{Q} = 3kW$ is assumed. In the following steps the sizing of the components will be performed step by step.

1. Reference energy U_{ref}:

$$U_{ref} = \frac{\dot{Q}}{\omega_0 d\hat{Q}} \tag{87}$$

2. Reference mass m_{ref}:

$$m_{ref} = \frac{U_{ref}}{c_v T_{ref}} \tag{88}$$

3. Reference volume V_{ref}:

$$V_{ref} = \frac{m_{ref} R_{gas} T_{ref}}{p_{ref}} \tag{89}$$

4. Heat exchanger pipe diameter d_{He}: (90)

$$d_{He} = \sqrt{\frac{4Nu\lambda}{\omega_0 NTU} \frac{(\kappa - 1) T_{ref} \hat{V}_{He}}{p_{ref}}}$$

5. Heat exchanger pipe length l_{He}: (91)

$$l_{He} = \left(\frac{\hat{V}_{He}}{\Pi_M \omega_0} \sqrt{R T_{ref}} \sqrt{\frac{v d^2}{32 \nu}} \right)^{2/3}$$

6. Heat exchanger surface A_{He}: (92)

$$A_{He} = \frac{Nu\lambda}{d_{He} \omega m_{ref} NTU}$$

7. number of identical pipes nbr: (93)

$$nbr = \frac{A_{He}}{\pi d_{He} l_{He}}$$

8. volume of heat exchanger V_{He}: (94)

$$V_{He} = nbr \cdot \frac{\pi}{4} d_{He}^2 l_{He}$$

Application of the design procedure with the parameters given in tables 4 and 5 leads to the following parameters for the heat exchangers.

$U_{ref} =$	$954.9\,J$
$m_{ref} =$	$0.00488\ kg$
$V_{ref} =$	$0.00382\ m^3$
$d_{He} =$	$0.0002785\ m$
$l_{He} =$	$0.00179\ m$
$A_{He} =$	$65.35\,m^2$
$nbr =$	41814000
$V_{He} =$	$0.00455\,m^3$

It can be seen that the volume of the heat exchanger V_{He} is larger than the reference volume V_{ref}. The number of 41.8 million tubes can not be manufactured with reasonable cost. Therefore this combination of design parameters does not lead to a design that can be realized. Much smaller $d\hat{Q}$ need to be taken to achieve a combination of design parameters that lead to design that can be realized. With the estimated $d\hat{Q}$ of the twinbird stirling cooler, the mac application would have a reference volume of $0.096\ m^3$. Such a combination leads to volume sizes that are too large for a MAC application.

6.7 Inverse design considerations

Taking for instance $d = 1mm$ as a reasonable pipe diameter for manufacturing, eq. 101 requires for equal Nusselt number the NTU number to be roughly 20 times smaller. This would result in $NTU = 0.2$. Simulation results given in the scatter plot in fig. 7 suggest that this would lead to COP smaller then unity and to much smaller power densities.

7. CONCLUSION

The possiblity of a stirling cylce based cooler for a MAC application was examined using experimental and simulative approaches. The identified non-dimensional parameters are explored by simulation for a possible design space. The resulting possible combinations of parameters were considered for a mechanical design. Due to the poor conductivity of the working fluid air, the resulting heat exchangers need to have a very large surface which is physically difficult to place into the available volume. Stirling cycle applications with a small power density and a favorable surface to volume ratio are less limited in the possible design. Since the physical space in the vehicle is limited on the one side and the power density of the stirling cycle is limited on the other side, a MAC application based on a stirling cycle is not possible.

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APPENDIX

Appendix A. Additional plots of non-dimensional parameters

Appendix B. Characteristic length of heat exchangers

A. Additional plots of non-dimensional parameters

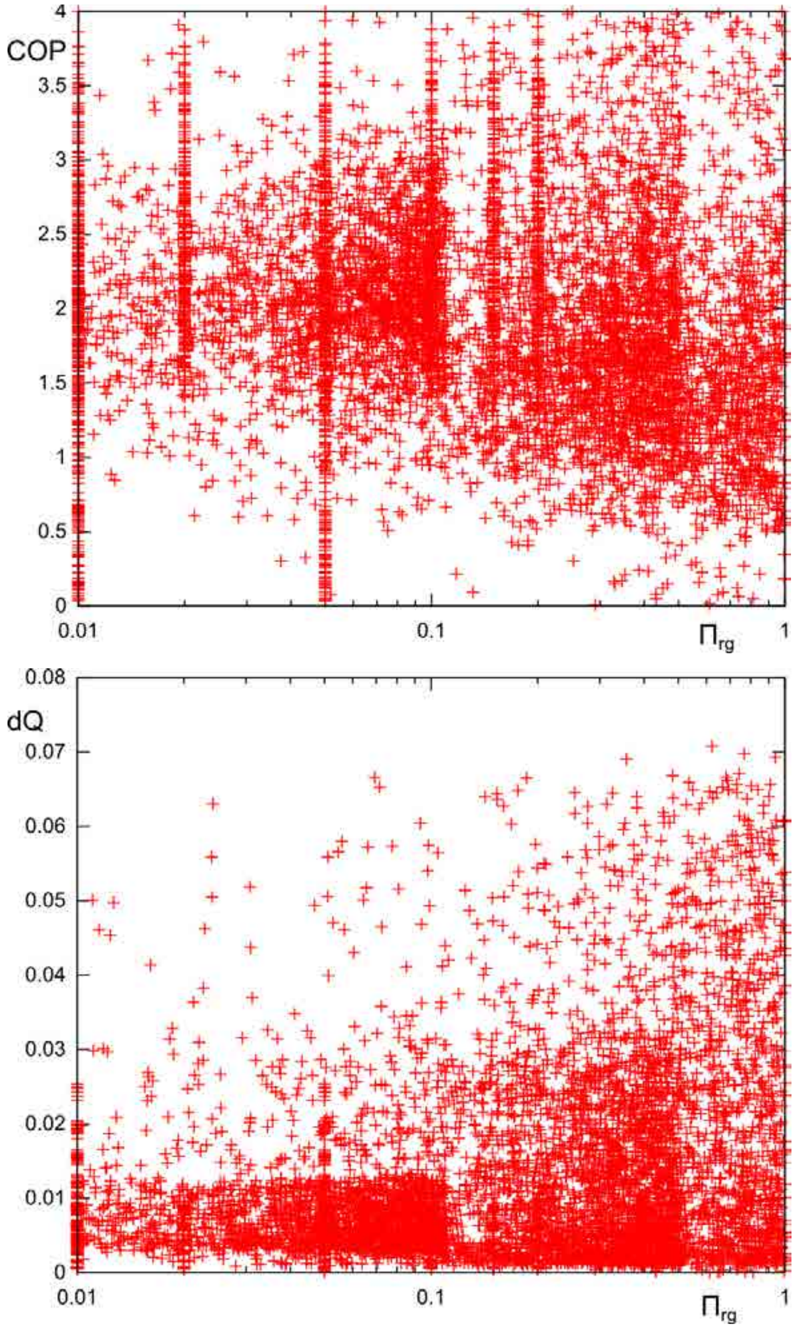

Figure 12. Scatter plot of COP (top) and dQ_0 (bottom) vs Π_{rg}.

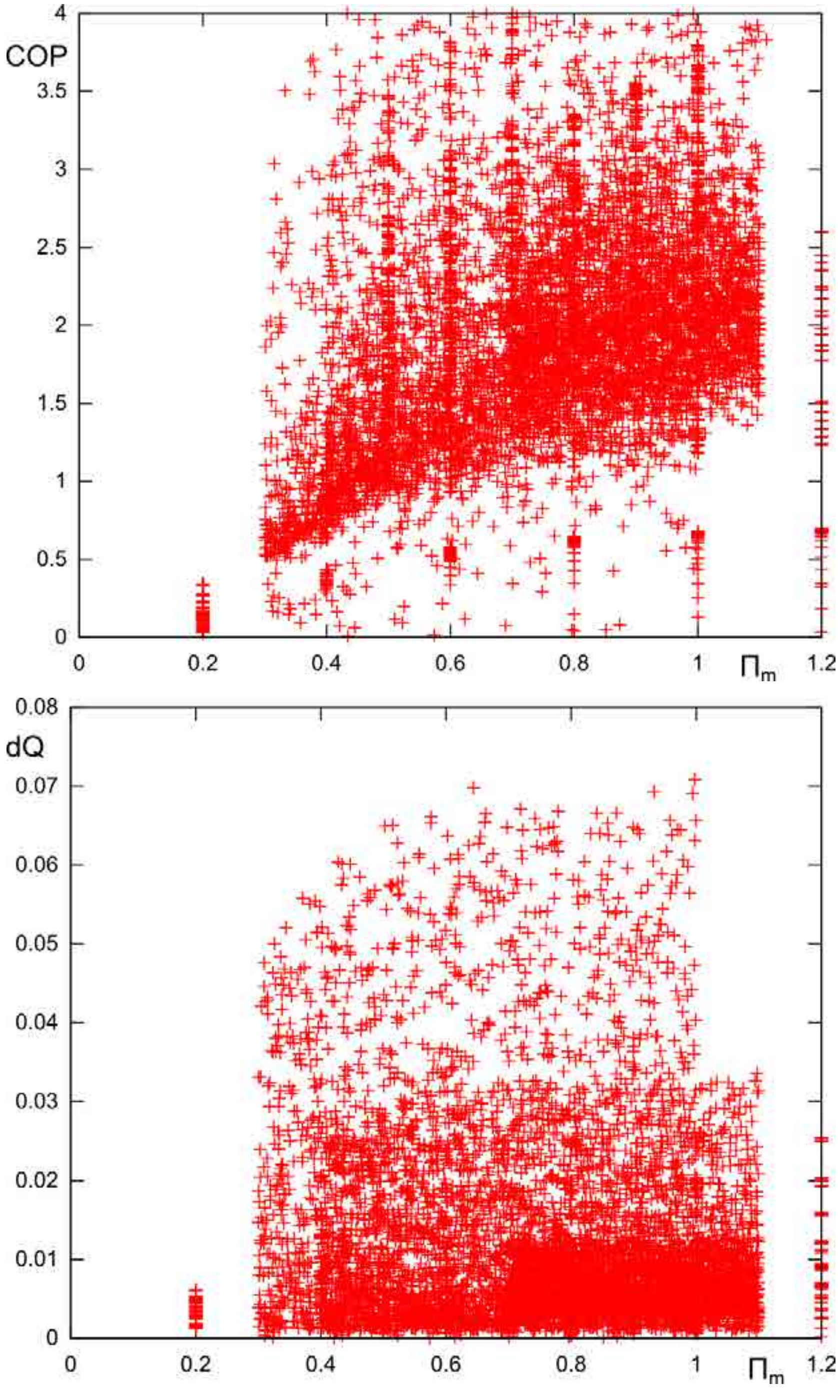

Figure 13. Scatter plot of COP (top) and dQ (bottom) vs Π_M.

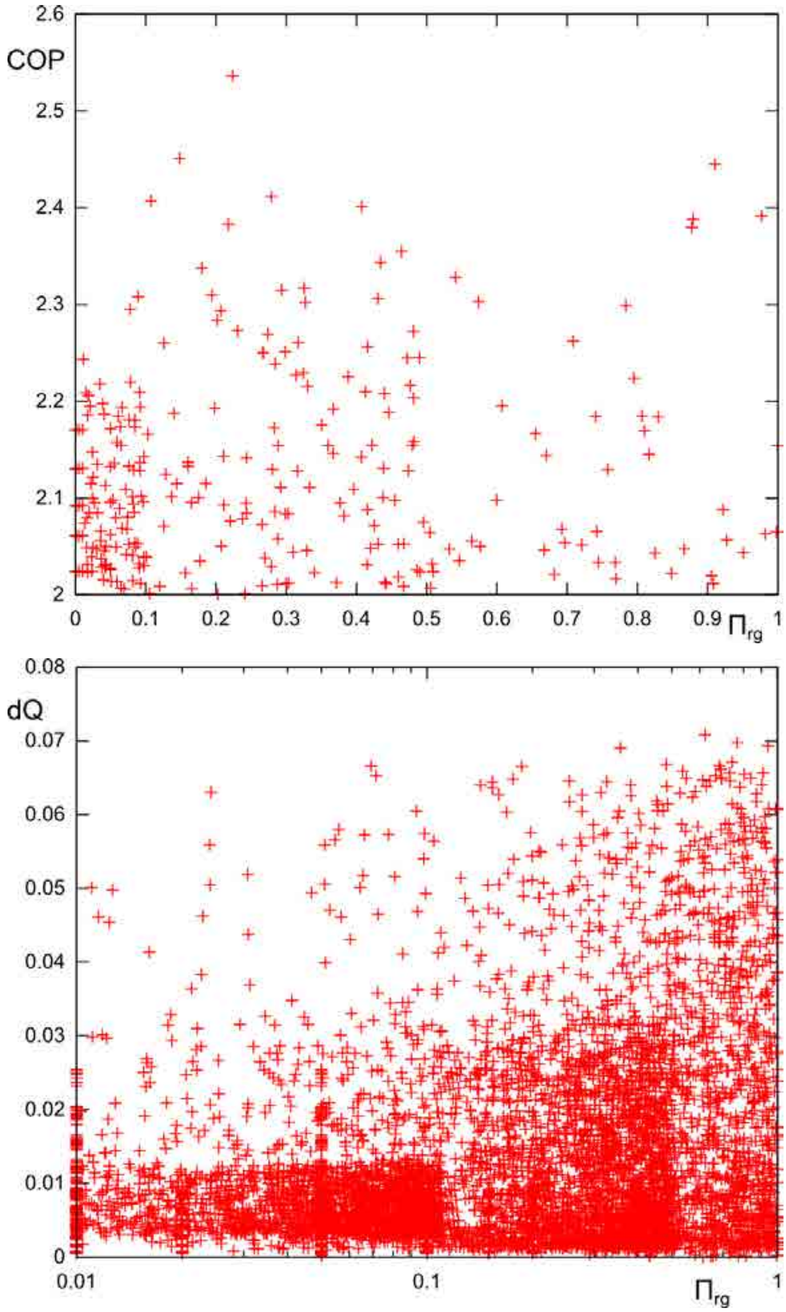

Figure 14. Conditional ($COP > 2, dQ_0 > 0.01$) scatter plot of COP and dQ_0 (bottom) vs Π_{rg}.

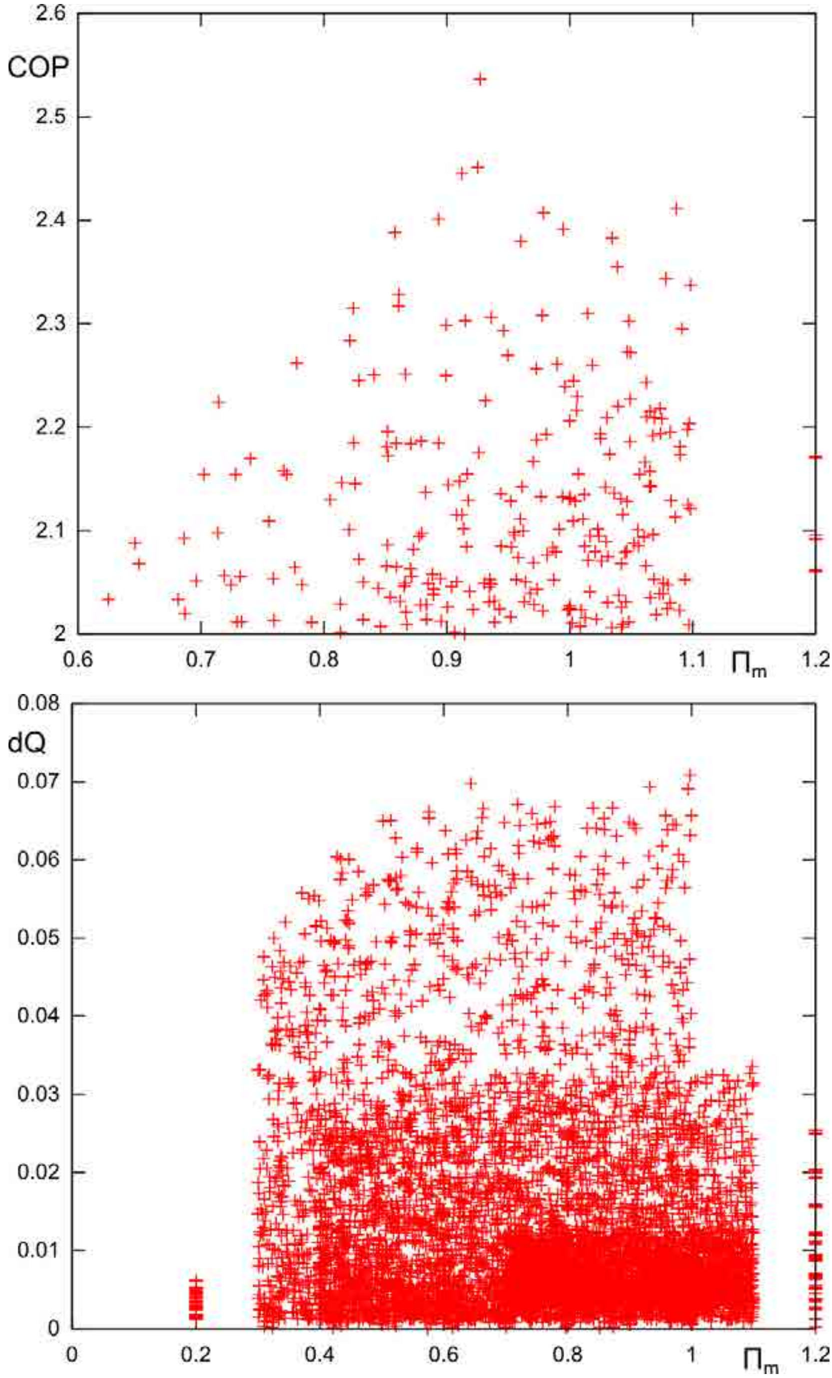

Figure 15. Conditional ($COP > 2$,$dQ_0 > 0.01$) scatter plot of COP and dQ_0 (bottom) vs Π_{rg}.

B. Characteristic length of heat exchangers

Some implications on the characteristic length scale of heat exchangers can be defined using the definition of NTU.

$$NTU = \frac{\alpha A_w}{\omega_0 m_{ref} c_v} \tag{95}$$

For the estimation of a length scale a geometry type has to be chosen. Most geometries can be simplified to cylinders of diameter d and length l or flat plates of width b and length l.

$$A_{cyl} = \pi d l \qquad A_{plate} = bl \tag{96}$$

Heat exchangers are used with fluids. Therefore either a liquid or a gaseous state can be assumed. In both cases the mass m can be related to the volume making either the assumption of an ideal gas or constant density.

$$V = \frac{m}{\rho} \qquad V = \frac{mRT}{p} \tag{97}$$

The volumes are function of either tube diameter d or the distance δ between the flat plates.

$$V_{cyl} = \frac{\pi}{4} d^2 l \qquad V_{box} = \delta d l \tag{98}$$

This leads to an expression for characteristic diameter or plate distance.

$$d = \frac{4\alpha}{\omega_0 NTU c_v \rho} \qquad \delta = \frac{\alpha}{\omega_0 NTU c_v \rho} \tag{99}$$

For the evaluation of the expressions values for the transfer number NTU and the heat transfer coefficient α are needed as the operating frequency, heat capacity c_v. and density ρ are typically given.
The heat transfer coefficient is expressed as a function of Nusselt number Nu, thermal conductivity λ_F of the fluid and the characteristic length scale.

$$\alpha_{cyl} = Nu\, \lambda_F / d \qquad \alpha_{plate} = Nu\, \lambda_F / \delta \tag{100}$$

This leads to an expression for the characteristic length scale as a function of Nusselt number and thermal conductivity.

$$d = \sqrt{\frac{4 Nu \lambda_F}{\omega_0 NTU c_v \rho}} \qquad \delta = \sqrt{\frac{Nu \lambda_F}{\omega_0 NTU c_v \rho}} \tag{101}$$

For laminar flow, a constant Nusselt number $Nu = 4$ (see Marek (R. Marek, For laminar flow, a constant Nusselt number $Nu = 4$ (see Marek (R. Marek, 2010)) can be assumed. With given properties for the air $\lambda_F = 0{,}0279\ W/mK$, heat capacity $c_v = 717\ J/kgK$, and density at ambient conditions $\rho = 1{,}15\ kg/m^3$, the characteristic length scale can be estimated for $NTU = 4$ as a function of the operating frequency. The resulting characteristic length scale is given in fig.16.

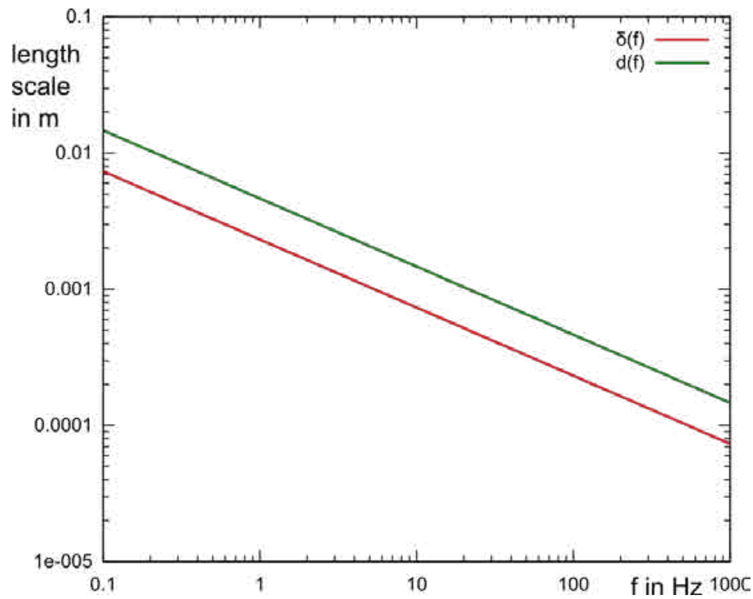

Figure 16. Characteristic length scale for laminar heat transfer with air for $NTU = 4$ at ambient conditions

This small length scales are mainly due to the small conductivity of air. Fig. 17 shows that air is at the bottom range of thermal conductivities.

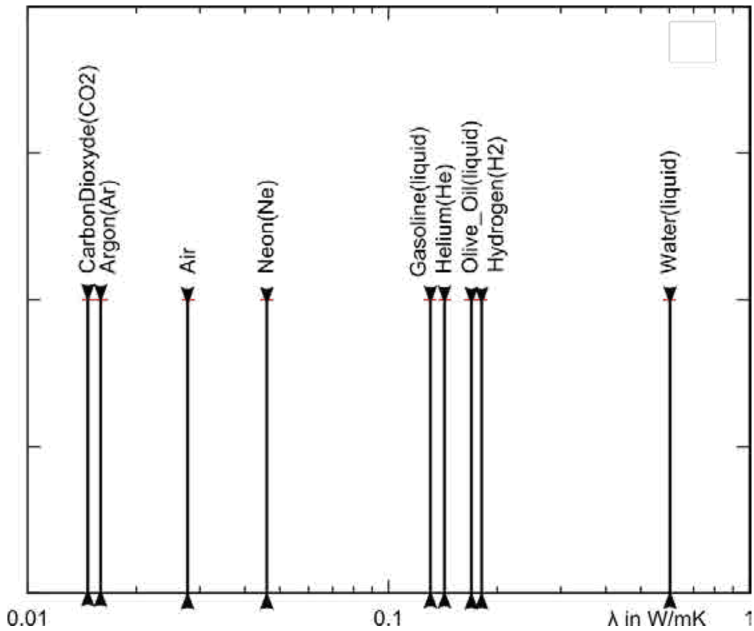

Figure 17. Thermal conductivities of some gases and liquids.

Using helium or hydrogen is beneficial when in comes to the characteristic length scale of the heat exchanger.

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