

Central University of Technology

**Development and characterisation of a Ti6Al4V additive manufactured
compact counterflow heat exchanger for application in
Organic Rankine Cycles**

by

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Built Environment and Information Technology in
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Abstract

Reducing greenhouse gas emissions is a vital component in mitigating the effects of global warming. While engineers across the globe are focused on finding new viable sources of clean energy, increasing the efficiency of current power-producing equipment would alleviate the pressure for finding and implementing these new technologies.

Various technologies exist to achieve greater efficiency, including Organic Rankine Cycles (ORCs). These systems have been successfully implemented in various large-scale industrial sectors, capturing and utilising low-grade waste heat from their processes. With great potential for increased efficiency within the domestic sector (less than 10 kW), intricate components in these small-scale systems require advanced traditional manufacturing processes that drive up cost and lead times.

The study aims to demonstrate the viability of using additive manufacturing (AM) for component fabrication, specifically focusing on developing and characterising an additively manufactured compact counter-flow heat exchanger for application in ORCs.

A theoretical model was developed to size a baseline heat exchanger. The heat exchanger core consisted of 0.5 mm square channels with a length of 30 mm packed in an array. The model was also used to size two more heat exchangers, each with different channel sizes and lengths; one with a core of 1 mm x 40 mm square channels and one with a core of 2 mm x 50 mm square channels. With the preliminary sizing completed, the geometric data from the theoretical model was used to develop CAD models of the heat exchangers for production in Ti-6Al-4V using additive manufacturing.

The heat exchangers were then experimentally characterised using a purpose-built test bench, capturing all relevant data. Each heat exchanger was characterised with 12 different flow settings, running five tests per flow setting. The test results were evaluated for validity and processed into useful data for comparison to the mathematical model results.

Additionally, computational fluid dynamics analyses were conducted and compared to the theoretical and experimental results. An acceptable level of agreement was obtained between the theoretical, experimental and CFD results.

Keywords: Additive Manufacturing, Ti-6Al-4V, Compact Counter-flow Heat Exchanger, Characterisation, Organic Rankine Cycle, Heat Exchanger Test bench, Computational Fluid Dynamics Modelling, Heat Exchanger Design.

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Nomenclature

Roman Letters

Symbol	Description	Units
A	Area	m^2
C	Capacity rate	$\text{W}/^\circ\text{C}$
C^*	Capacity ratio	-
c_p	Specific heat capacity	$\text{kJ}/\text{kg}\cdot\text{K}$
D_h	Hydraulic diameter	m
e	Internal energy	J
f_F	Fanning friction factor	-
G	Mass velocity	$\text{kg}/\text{s}\cdot\text{m}^2$
g	Conversion factor	-
g_i	Gravitational acceleration	m/s^2
H	Height	m
h	Convective heat transfer coefficient	$\text{W}/\text{m}^2\cdot\text{K}$
h	Thermal enthalpy	J
k	Thermal conductivity	$\text{W}/\text{m}\cdot\text{K}$
k	Turbulent kinetic energy	m^2/s^2
K	Von Karman constant	-
K_c	Entrance loss coefficient	-
K_e	Exit loss coefficient	-
L_e	Entrance length	m
L	Length	m
$\dot{m}$	Mass flow rate	kg/s
Nu	Nusselt number	-
P	Pressure	Pa
p	Wetted perimeter	m
Pr	Prandtl number	-
Q	Heat Duty	W
Q_H	Heat source per unit volume	$\text{J}/\text{K}\cdot\text{m}^3$
q	Heat transfer rate	W
q_i	Diffusive heat flux	W/m^2
Re	Reynolds number	-
R_t	Turbulent Reynolds number	-
S_i	Mass-distributed external force per unit area	N/m^2
T	Temperature	$^\circ\text{C}$ or K
t	Time	s
U	Overall heat transfer coefficient	$\text{W}/\text{m}^2\cdot\text{K}$
u	Fluid velocity	m/s^2
v	Specific volume	m^3/kg
W	Width	m
x	Material thickness	m
y_*	$\sqrt{\tau_w/\rho} \cdot y/\nu$	-
y	Distance from wall	m

Greek Letters

Symbol	Description	Units
δ	Boundary layer thickness	m
ε	Isentropic turbulence dissipation rate	W/kg
η	Efficiency	%
μ	Dynamic viscosity	Pa.s
ν	Kinematic viscosity	m ² /s
ρ	Density	kg/m ³
σ	Free-flow to frontal area ratio	-
τ_{ij}	Viscous shear stress tensor	Pa
v	Specific volume	m ³ /kg
ϕ	Electric potential	V

Subscripts

Symbol	Description
--------	-------------

c	Cold
cf	Fin cross-section
cs	Cross-section
f	Fin
h	Hot
i	Inlet
i	i^{th} coordinate direction
j	j^{th} node
o	Outlet
t	Turbulent
t	Total
w	Wall

Abbreviations

AM - Additive Manufacturing

CAD - Computer-Aided design

CAF - Central Analytical Facilities

CRPM - Centre for Rapid Prototyping and Manufacturing

CT - Computerised tomography

DMLS - Direct metal laser sintering

EES - Engineering Equation Solver Software

GHG - Green House Gases

ORC - Organic Rankine Cycle

RAPDASA - Rapid Product Development Association of South Africa

Chapter 1

Introduction

1.1 Background

1.1.1 Heat exchangers

Heat exchangers are devices used to transfer heat between two fluids at different temperatures [1]. Heat exchangers are invaluable in numerous applications for industries ranging from refrigeration to the space industry [2]. 90% of the heat energy used in the production and management of energy is transported via various types of heat exchangers; therefore, various aspects have been investigated intensively to improve the performance of these devices. New materials, manufacturing methods, heat exchange devices and heat transfer surfaces form part of these investigations [3].

1.1.2 Heat exchanger classification

While all heat exchangers operate on the same basic principles, they are generally categorised and classified by flow configuration or construction [4].

Flow configuration

There are four basic flow configurations which refer to the direction of the fluid flow relative to each other within the heat exchanger.

Counterflow heat exchangers have the working fluids flowing parallel in opposite directions. This flow configuration allows for the most significant temperature change between the two fluids and is the most commonly utilised configuration.

Concurrent-flow or parallel-flow heat exchangers have the working fluids flowing parallel in the same direction. The flow configuration results in lower efficiencies compared to the counterflow configuration. However, it is beneficial if the working fluids are required to approach the same temperature.

In *Crossflow heat exchangers*, the working fluids flow perpendicularly to each other. Efficiencies for this flow configuration are generally between that of counterflow and concurrent-flow.

Hybrid flow heat exchangers combine any of the flow as mentioned above configurations within a single heat exchanger. Shell-and-tube, multi-pass and cross-/counterflow heat exchangers are examples of hybrid flow heat exchangers [3, 4].

Construction

We can further categorise heat exchangers based on their construction. The first classification level divides heat exchangers into regenerative or recuperative types, each with subclassifications [3].

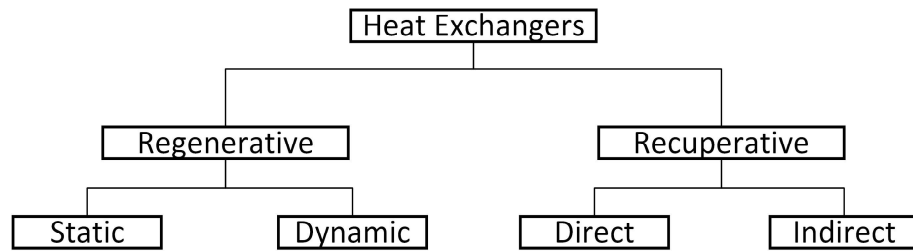


Figure 1.1: Heat exchanger classifications

Regenerative heat exchangers:

Also known as regenerators or capacitive heat exchangers, these transfer heat from one fluid to another by alternately heating and cooling a matrix and can be further categorised as static or dynamic. A *static regenerator* alternates flow through the matrix via a series of pipes and valves. As the hot fluid flows through the matrix, heat is transferred to the matrix, after which the hot fluid flow is shut off, and the cold fluid is allowed to flow through the matrix, absorbing the heat from the matrix.

Dynamic regenerators have a rotating element acting as the heat storage medium. The hot and cold fluids, separated by a gas seal, flow parallel to the rotation axis through the rotating element, transferring heat from one fluid to the other via the rotating element. Both static and dynamic regenerators have the risk of cross-contamination of the fluid streams and this should be taken into consideration. These types of heat exchangers are generally used in gas-gas applications [3, 5].

Recuperative heat exchangers:

Recuperative heat exchangers or recuperators can be categorised into two groups, direct- or indirect-contact heat exchangers. *Direct contact* recuperators rely on the working fluids' direct contact for heat transfer, normally air-water, water-water or water-steam heat exchangers. The most common example of direct contact recuperators is the cooling towers found in power plants.

Indirect contact recuperators have the working fluids flowing simultaneously in separate channels, relying on a combination of convection and conduction for heat transfer. The working fluids are generally separated by tubes or plates, depending on the specific design or application of the heat exchanger [3, 4, 5].

1.1.3 Heat exchanger types

Recuperative indirect heat exchangers are the most common heat exchangers in the industry; the most common types are discussed briefly below [3, 4, 5].

Double-pipe heat exchangers:

Double-pipe heat exchangers are one of the simplest heat exchangers designs and consist of two or more concentric pipes or tubes, one larger in diameter than the other. Fluids flow separately through each pipe in the required configuration, counterflow or parallel flow. These types of heat exchangers are cost-effective, easily maintained and they can be designed flexibly to suit the application.

Shell and tube heat exchangers:

These heat exchangers can be designed as counter, parallel or hybrid flow. They consist of a bundle of parallel tubes enclosed in a sealed pressure vessel known as the shell. One fluid flows through the bundle of tubes while the other flows within the shell over the outside of the tube bundle. They are well known and commonly found in various industries.

Plate-fin or plate-frame heat exchangers:

Thin corrugated plates are stacked together to form these types of heat exchangers. Each pair of stacked plates forms a flow passage for one working fluid, and a flow passage is formed with each additional pair added for a secondary working fluid. These stacks are then permanently joined (brazed or welded for the plate-fin type) or compacted with a frame and bolts (plate-frame type). The popularity of these types of heat exchangers are increasing due to the possible cost, space and weight saving they provide in specific applications.

Compact heat exchangers:

Compact heat exchangers are another type gaining in popularity, especially for high-heat flux applications. They are characterised by their high heat transfer surface area density and can reach more than $650 \text{ m}^2/\text{m}^3$ [6]. These types of heat exchangers can come in various flow and construction arrangements to suit the specific application.

1.1.4 Heat exchanger applications

Heat exchangers form an invaluable part of many processes within various industries. They have therefore been extensively researched to improve the performance of these devices. With new technological developments in material science and manufacturing methods, engineers are innovatively implementing these devices into new and existing systems to improve efficiency and reduce costs [3, 7].

One of the ways in which engineers can increase the efficiency of various processes is by capturing and utilising waste heat either for aiding the process directly (such as pre-heaters) or being used for other processes entirely. Organic Rankine Cycles (ORCs) are a proven technology for capturing waste heat to produce additional power [8].

ORCs comprise at least two heat exchangers: a boiler and condenser, and an expander and a pump [9]. These systems operate on the same basic principles as Rankine cycles. The key difference is that ORCs utilise organic compounds instead of water in order to reduce operating temperatures. Implementing higher efficiency heat exchangers within these cycles could considerably improve the efficiency and applications of these systems.

Organic Rankine Cycles within the industry generally produce anything from a few kW to multi-MWe, and miniaturising these systems would allow domestic users to also benefit from this technology [10]. Miniaturisation, however, introduces new challenges for engineers. One of which is manufacturing the small-scale components required [11].

A promising technology for producing these small-scale components comes in the form of additive manufacturing (AM). This method of manufacture has the ability to produce intricate parts in various materials with incredible accuracy [12]. With numerous studies focusing independently on micro-ORCs and additive manufactured heat exchangers, the

focus for this study will therefore be to characterise an additive manufactured compact counterflow heat exchanger for application in ORCs.

1.2 Motivation for the study

Global warming is one of the biggest challenges we are currently facing, and reducing anthropogenic greenhouse gas (GHG) emissions is vital in reducing the effects of global warming. With the energy sector responsible for approximately three-quarters of the emissions [13], shifting from traditional energy production processes towards renewable energy sources is essential.

In 2021 the International Energy Agency stated that all countries need to do more regarding their GHG reduction goals for 2030. The pledges at that stage covered less than 20% of the gap in GHG reductions to keep on track for a 1.5 °C increase by 2050. Even with the successful pursuit of all pledges, the modelled energy-related GHG emissions fall by only 40% over the period leading to 2050 [13].

With both the end-user and industry able to benefit from increased energy efficiency, more focus is being put on increasing the efficiency of various equipment and processes [14]. Capturing and utilising waste heat is one method to increase efficiency.

A 2018 study by Papapetrou et al. [15] estimated the total waste heat potential in the EU to be 300 TWh/year. One-third of this was low-grade waste heat with temperatures below 200 °C, waste heat in the range of 200–500 °C accounted for 25% and temperatures above 500 °C make up the rest.

Various technologies exist to utilise low-grade waste heat, including the previously mentioned ORCs. Due to technical and manufacturing difficulties, low-grade waste heat is less commonly recovered in industry. However, it should not be disregarded as it still has the potential to be used for power production, as well as heating and cooling processes [16].

Improving existing and new manufacturing techniques allows engineers to effectively downscale these systems for applications previously not deemed viable [17]. Thus the study aims to mitigate the effects of global warming by expanding the knowledge of additive manufactured heat exchangers and their applications.

1.3 Research aim

This research study aims to characterise an additively manufactured Ti6Al4V counterflow compact heat exchanger and to validate the experimental results with numerical simulations.

In order to accomplish the aim mentioned above, a counterflow compact heat exchanger design must be developed to be compatible with additive manufacturing (AM). With the completion of the design, a few configurations differing in size and channel geometry would be produced. An experimental investigation would then be conducted to characterise these configurations, enabling the validation of the numerical simulations against the experimental results obtained.

Thus the primary objectives of the study can be listed as follows:

- Develop a theoretical numerical model for heat exchanger sizing
- Design a compatible compact heat exchanger for additive manufacturing
- Produce different configurations of the compact heat exchanger design
- Characterise compact heat exchangers experimentally
- Validate experimental results against numerical simulations

Secondary objectives could therefore be:

- To apply the design methodology for additively manufactured counterflow compact heat exchangers to Organic Rankine Cycles
- Advance the knowledge of additive manufactured compact heat exchangers
- Assess the performance characteristics of Ti6Al4V for compact heat exchangers

1.4 Scope

The study will focus primarily on the design and experimental characterisation of additive manufactured heat exchangers. The manufacturing parameters and process to obtain the additive manufactured heat exchangers will not be covered. However, critical findings from the manufacturing process will be discussed.

1.5 Structure

This dissertation consists of seven chapters, including the current one. Chapter 2 entails the literature review; the literature of all relevant aspects was investigated and reviewed to obtain up-to-date knowledge on the research subject at hand. Chapter 3 presents the numerical procedure for modelling the compact counterflow compact heat exchanger. Chapter 4 discusses the design process for the heat exchangers and considerations for additive manufacturing design. This chapter also elaborates on the manufacturing difficulties experienced during manufacturing. The experimental setup and procedure are described in Chapter 5. The numerical, experimental and CFD results are presented and discussed in Chapter 6. Chapter 7 summarises and concludes the study, and provides future research suggestions.

Chapter 2

Literature Review

2.1 Introduction

This chapter reviews the technology and research on additive manufactured compact counterflow heat exchangers for application in Organic Rankine Cycles (ORCs). Although numerous studies exist for various applications of additive manufactured heat exchangers, it was found that limited research has been done on the application of additive manufactured heat exchangers in ORCs. The advantages of using additive manufacturing for producing compact heat exchangers are also discussed.

With the current high demand for energy and pressure on the energy sector to reduce their greenhouse gas (GHG) emissions, engineers are tasked with developing more efficient systems for power-producing equipment [18, 19]. With renewable energy sources such as wind and solar growing rapidly and despite the Covid-induced recession, 2021 saw the second-largest annual increase in CO₂ emissions in history [13]. To alleviate GHG production, the efficiency of current energy-producing technology can be increased with existing systems like ORCs [20].

Large-scale ORCs are already implemented in numerous plants worldwide to capture waste heat and produce additional power, thereby increasing efficiency [21, 22]. Micro-ORCs are smaller and more compact systems that can be utilised by domestic users (1-10 kW) [10, 23]. Micro-ORCs have great potential for waste heat recovery on a small scale; however, manufacturing the small-scale components for micro-ORC systems proves to be difficult using traditional methods [24]. The heat exchanger is one of the components that experiences these manufacturing difficulties. Incorporating a smaller heat exchanger into an ORC, such as a micro-channel heat exchanger, has several benefits, one of which is a very high heat transfer coefficient [3, p. 6] [25]. Therefore, these compact heat exchangers are particularly effective when used in power-producing or power-consuming systems [26].

Traditional methods like X-ray lithography (LIGA), chemical and silicon etching, diamond machining or wire machining can produce basic micro-channel heat exchangers [25, 27]. Most of them are time-consuming, restricted to basic designs or require more than one process, like diffusion bonding, to produce a heat exchanger [28].

Additive manufacturing is a well-suited manufacturing method for compact heat exchangers as it allows manufacturing of highly complicated parts with relative ease [29, 30, 31].

2.2 Organic Rankine Cycles

The industrial sector accounted for more than one-third (170 EJ) of the total final energy consumption in 2021 and contributed 45% (9 Gt) of the total direct CO₂ emissions from the end-use sector. The International Energy Agency states, “There is no way to reach zero emissions without strong and coordinated action on emissions reduction in the industry sector” [14].

Organic Rankine Cycle systems are well suited to convert heat into electricity for renewable or industrial energy efficiency applications. From a few kW to multi-megawatt geothermal plants, ORC systems have gained popularity since the 1970s due to surging energy prices and economic incentives [8].

Organic Rankine Cycles are similar to a conventional Rankine Cycle, the difference being that ORCs utilise organic compounds such as refrigerants, hydrocarbons or siloxanes instead of water as the working fluid [9]. Organic refrigerants work well within a temperature range of 100–175 °C, while hydrocarbons are more suitable for temperatures between 175–250 °C and siloxanes for higher temperatures of 250–400 °C [16]. The use of organic compounds in these systems allows them to utilise low temperatures and can be optimised for the best performance with the numerous organic fluids available.

Geothermal energy, solar energy, biomass products, surface seawater and waste heat from thermal processes are some of the possible heat sources for an ORC [21, 16]. Table 2.1 lists some general and industrial waste heat sources and their temperatures.

Table 2.1: Low-grade waste heat sources and temperatures.

Low-grade waste heat source		Temperature (°C)
Generic Unit/Process		
1. Boilers	• Flue gases	110–260
2. Air compressors	• Waste heat from compressor cooling system	30–60
3. Heating/cooling network	• Condensate from steam heating and spent cooling water from cooling systems	60–90
Industrial Sector		
1. Petrochemical	• Stack gas from crude distillation	156
	• Stack gas from vacuum distillation	216
	• Exhaust from ethylene furnace	149
2. Iron/steel making	• Waste gas from coke oven	200
	• Blast furnace gas	93
	• Blast stove exhaust	250
3. Aluminium	• Exhaust from aluminium casting with a stack melter	121
4. Food and drink	• Extracted air from cooking with fryers or ovens	150–200
	• Exhaust from drying with spray/rotary dryers	110–160
	• Water vapour from evaporation and distillation	100
5. Textile	• Dyed wastewater from dyeing	90–94
	• Stenter exhaust for fabric dyeing and finishing	180
	• Waste water rejected from heat exchangers	58–68
6. Paper	• Waste steam from slag flushing in furnace	95–100
	• Waste water from slag flushing in furnace	65–85
	• Cooling water from furnace wall cooling	35–45
7. Cement	• Exhaust from cement kilns using 5- or 6-stage pre-heaters	204–300
	• Hot air discharge from clinker coolers	100

Source: Adapted from [16]

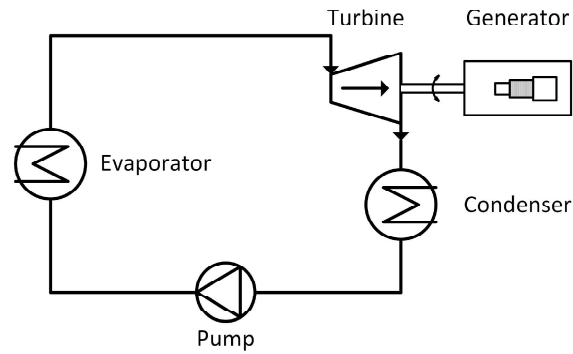


Figure 2.1: Schematic diagram of an ORC

Organic Rankine Cycles usually consist of an evaporator, expander, condenser and pump. The organic fluid is pumped to the evaporator, where the fluid is vaporised using the heat from the waste heat source. The vaporised fluid then travels to the expander, where expansion takes place, and mechanical work is produced. The expander is connected to a generator where the mechanical work is converted to electric energy. The fluid then passes through the condenser, where it is cooled down to a liquid state, ready to be pumped back to the evaporator [9], Figure 2.1 depicts a basic schematic of an ORC.

Tartière and Astolfi published an article in 2017 on “A world overview of the Organic Rankine Cycle market” and analysed more than 700 projects from 27 ORC system manufacturers with a combined installed capacity of 2.7 GW [8]. The +700 projects accounted for 1754 installed plants, with 1073 used to recover waste heat, producing a combined 376 MWe.

A more up-to-date project map of Tartière’s references is available at [32], since the published article in 2017 and the last update of the project map in 2021, the number of projects increased to 1067 with another 83 projects under construction.

The market leader, ORMAT, has installed 1102 ORC units by the end of 2016 with a capacity of 1701 MWe. Approximately 800 of these units installed by ORMAT are used for valve operation and cathodic protection along pipelines in remote areas producing less than 4 kWe [8]. To date, they have 28 plants across the globe used for waste heat recovery producing 147 MWe [33]. Another ORC system manufacturer, Turboden, has to date installed more than 400 ORC plants in 48 countries. Twenty-eight of these are used for waste heat recovery, producing 68.4 MWe, with five more under construction, further increasing the power production with 14.7 MWe [34].

EXERGY, a company based in Italy, which focuses on ORC systems, states that ORCs have experienced significant growth in the last two decades and are becoming the preferred technology for geothermal power plants. Furthermore, the urgency to increase energy efficiency and implement decarbonisation technologies will also drive broader adoption of ORCs for waste heat recovery in various industries. The company currently has 57 active ORC plants producing 485 MWe, of which 22 are implemented for waste heat recovery, producing 36 MWe [35].

Alfa Laval has smaller ORC systems for implementation in the marine sector to capture waste heat and produce additional power on vessels. Waste heat from jacket cooling water

(75 °C) and exhaust gas (550 °C) can be captured with their system, producing up to 200 kWe for larger combined systems [17].

Numerous studies are available for ORCs in domestic applications. Bianchi et al. [36] published a study on the experimental performance of a micro-ORC system for low-grade heat recovery and found the system can produce 300–1200 We with the source temperature ranging from 65–85 °C, the corresponding cycle efficiency at these operating conditions ranged between 2.9 and 4.4%.

A follow-up study done by Bianchi et al. [37] investigated the performance and operation of a micro-ORC energy system using a geothermal heat source. The heat source supplied hot water at a constant ± 55 °C to the ORC system, producing about 2.8 kWe at a cycle efficiency of 4.4%; Bianchi states this is congruent with the state-of-the-art micro-scale ORC systems.

Wajs et al. [11] did a similar study with the heat source being a domestic gas boiler used for space heating. The gas boiler was modified to use thermal oil as a heat carrier and ethanol as the working fluid. The working fluid inlet temperatures to the expander ranged from 115 °C to 126 °C producing ± 710 We with a cycle efficiency between 6 and 7%. Wajs et al. stated in a later study that the performance obtained with the ORC system is auspicious and demonstrates its potential [38].

A small-scale solar ORC system used for domestic micro-cogeneration was experimentally characterised by Fatigati et al. [39]. For characterisation purposes, the thermal availability of a 15 m² solar flat plate collector used for domestic hot water was reproduced with two 12 kW electric heaters. The variability of solar energy was accounted for, and it was found that the system produced anything from 150 to 500 We at an efficiency of 2.4 to 4%, respectively. Optimising the operating strategy could allow such a system to cover up to 11.2% of the yearly required energy of an average European household.

Another solar-driven micro-ORC system for residential application study done by Ancona et al. [40] found that their ORC system could produce approximately 39% (more than 1150 kWh) of the yearly demand for an average single-family user in Italy. The system comprised a 32.25 m² solar collector and a 6000 l storage tank to minimise the solar radiation fluctuations. The overall cycle efficiency for this set-up was just above 2%.

With the large-scale ORC market firmly established and expanding, giving domestic users access to micro-ORC systems for waste heat recovery could substantially benefit the current trends seen in energy consumption. While current studies on micro-ORC systems show <10% cycle efficiencies, optimising the equipment used within the cycles, such as the heat exchanger or expanders, could further increase the efficiency of these cycles. Wajs et al. [11] stated that micro-scale ORC technology (below 10 kWe) is a technical challenge because it should comprise highly efficient small-scale expanders and compact heat exchangers.

2.3 Manufacturing

Producing these small-scale components proves to be challenging. Klonowicz et al. [41] found that higher tolerances decreased the efficiencies of their microturbine design due to larger relative clearances and blade edge thickness. He also stated that the throats of the supersonic nozzles could pose a manufacturing problem.

With traditional methods like X-ray lithography (LIGA), chemical etching, micro-machining or electrical discharge machining able to produce these small-scale components, most of them are time-consuming, restricted to basic designs or requiring more than one process, like diffusion bonding, to produce more complex small-scale components [25, 27, 28].

2.3.1 Traditional manufacturing methods

This section briefly discusses some traditional manufacturing methods focusing on compact heat exchangers.

X-ray lithography (LIGA)

The acronym LIGA is German and stands for lithography, electroplating and moulding. The process consists of three steps: deposition of the resist layer onto a conductive substrate, high-energy X-ray exposure and development, and filling and dissolving the resist mould.

The high-energy X-rays pass through a mask and cure the X-ray sensitive resist layer with the mask, shaping the projection of the X-rays into the desired shape. After that, the unexposed resist is removed using a suitable developer, revealing a mould. The mould and substrate are then electroplated to form the desired parts within the mould. The parts are post-processed, and the mould is dissolved to produce the final part. More complex designs may require the process to be repeated several times and to bond the final products together using processes like diffusion bonding [12, 28].

While compact heat exchangers have been successfully manufactured using LIGA, the complicated processes and procedures coupled with the costly equipment are the main barrier to extensive adoption and development for compact heat exchangers [12].

Chemical etching

Chemical etching, also known as photochemical machining (PCM), utilises strong chemical etchants to remove material controlling the dissolution process to produce parts. The process consists of six steps: cleaning the material, resist coating, UV light exposure, developing, etching and stripping the remaining resist material [12].

Chemical etching is a suitable manufacturing technology for producing compact heat exchangers. It is, however, a tedious process due to the complexity of the related equipment and operation specifications. More complex heat exchanger designs may also require the process to be repeated several times before bonding the components together to form the final heat exchanger [12, 28].

Micro-machining

Micro-machining is similar to traditional machining, where the material is cut, formed, deformed, removed or bonded with tooling to produce the required part. Micro-machining is a suitable technique for various materials and is not limited to materials that etch well or

are good electrical conductors. However, the materials should be soft and ductile enough for machining [28].

Copper, aluminium and steel are the primary materials used with this technique, while in applications for extreme operating conditions, titanium or nickel alloys can be substituted. A study conducted by López de Lacalle et al. [42] found that Ti6Al4V and Inconel 718's material properties severely impact these alloys' machinability. The low thermal conductivity and reactivity to hard metal are the key contributors to the machinability of titanium alloy. As for Inconel 718, the main hurdles for machinability are the hardness, coupled with the mechanical hardening during the cutting process.

The specialised tooling, tool life, material utilisation, material handling and fabrication cost are all obstacles concerning the technology development [12, 25, 43].

Electrical discharge machining (EDM)

Micro EDM is similar to electrical discharge machining (EDM), also known as spark machining, spark eroding, die sinking, wire burning or wire erosion. It shares the same working principle, with the key differences being tool size, fabrication method, gap control and machining technique.

EDM is primarily used to machine electric conductive metals and can be classified into three variants: Drill EDM, wire EDM and die-sinking EDM. The working principle of EDM is to remove material from the workpiece using rapidly recurring current discharges (sparks). These sparks occur between the workpiece (anode) and electrode tool or wire (cathode) separated with a dielectric fluid; either deionised water or erosion oil is used [12].

EDM is a suitable machining technique for hard materials such as titanium or workpieces of particular thickness. Its non-contact aspect gives it the ability to produce extremely thin parts. The main drawbacks of EDM include the material limitation, electrically conductive, and slow material removal rate, making it particularly unsuitable for large-scale production [44].

Microforming

Microforming entails various manufacturing methods, including micro-tube drawing, micro-stamping, micro-embossing, micro-hydroforming, micro-extrusion and micro-rolling. Higher material integrity, productivity and less waste contribute to this manufacturing technique's lower manufacturing costs.

This technique is suitable for prototyping, low- or high-volume production and compact heat exchangers, suitable for manufacturing micro-tubes and micro-channels. Downscaling the conventional process does, however, introduce new manufacturing difficulties. The increase of the frictional resistance between the work piece and tooling causes a decrease in the forming limit, while with micro-hydroforming, the process is heavily dependent on specialised equipment. The technique is also unsuitable for hole or ditch fabrication with large depth to width ratios or difficult-to-deform billets and is also limited to basic designs for micro-channels [12].

2.3.2 Additive manufacturing

From a practical perspective, fabricating complex small-scale components directly in bulk materials tends to be more profitable. Thus additive manufacturing is considered a viable manufacturing technique, especially for ultra-high temperature materials [12].

Additive manufacturing comprises different technologies used for manufacturing physical objects by successive addition of materials layer by layer. Additive manufacturing is used for various applications in engineering, medicine, architecture, toys, entertainment, education and cartography [45, 46].

The global AM market is expanding rapidly, up from USD 6 billion in 2016 to USD 13.7 billion in 2020, with expected growth reaching 63.46 billion USD by 2026, at a compound annual growth rate of 29.48% [47, 48].

With numerous AM technologies available, each with its own processes, advantages, applications and equipment, standards were developed by ISO/TC 261 and ASTM F42 to standardise terminology associated with AM. Seven of these AM processes were categorised and these are listed in Table 2.2.

Table 2.2: Process categories of additive manufacturing

AM Technology	Process	Manufacturing Techniques	Application	Materials
Binder Jetting (BJT)	A liquid bonding agent is selectively deposited to join powder materials.	-	Functional metal parts, low-rate production run of non-critical components	Metals, sands, polymers, ceramics and composites
Direct Energy Deposition (DED)	Focused thermal energy is used to fuse materials by melting as they are deposited.	Laser engineered net shaping (LENS)	Functional metal parts, repairs, adding material to existing parts	Metallic feedstocks, ceramics and polymers
Material Extrusion (MEX)	Material is selectively dispensed through a nozzle or orifice.	Fuse deposition modelling (FDM), Pellet additive manufacturing (PAM), Bound metal deposition (BMD)	Functional metal parts, prototyping	Polymers, food, living cells
Material jetting (MJT)	Droplets of feedstock material are selectively deposited.	Drop-on-demand (DOD), PolyJet technology NanoParticle Jetting (NPJ)	Prototyping, tooling, low-volume production	Polymers, Ceramics, composite, plastics
Powder Bed Fusion (PBF)	Thermal energy selectively fuses regions of a powder bed.	Direct metal laser sintering (DMLS), Electron beam welding (EBM), Selective heat sintering (SLS), Selective laser melting (SLM), Selective laser sintering (SLS)	Functional metal parts, functional prototyping	Metallic feedstocks, ceramics, polymers and composites
Sheet Lamination (SHL)	Sheets of material joined layer by layer using adhesive, welding or thermal bonding.	Laminated object manufacturing (LOM), Ultrasound additive manufacturing (UAM)	Non-functional prototypes, casting moulds, hybrid manufacturing	Metals, Plastic, Papers

Table 2.2: Process categories of additive manufacturing (*continued*)

AM Technology	Process	Manufacturing Techniques	Application	Materials
VAT Photopolymerisation (VPP)	Selectively curing of photo-reactive polymers using a laser, light or ultraviolet light (UV)	Stereolithography (SLA), Digital light processing (DLP), Carbon digital light synthesis (DLS)	Prototyping, engineering applications	Polymers, plastics and resin

Most common materials and techniques listed

Sources: [45, 46, 47, 49]

This paper will not discuss the AM processes mentioned above due to the overabundance of literature in this area. The book by Gibson et al. [50] and reviews done by J.R. McDonough [31] and Ngo et al. [51] give a detailed overview of these processes.

The method of interest for this study is DMLS. The process involves a thin layer of metal powder spread across the build plate and fused with a laser beam. The process is repeated for each layer of the part. The excess powder is removed, and the parts can proceed to post-processing. A basic depiction of the process can be seen in Figure 2.2. The Centre for Rapid Prototyping and Manufacturing (CRPM) at the Central University of Technology also uses the DMLS process to produce parts directly in metal.

Advantages of AM

Conflux Technology, a world-leading AM company pioneering thermal and fluid applications, used the advantages of AM to produce an additive manufactured heat exchanger for application in F1 motorsport. The freedom of design enabled them to optimise their design, offering three times more heat rejection at a third of the pressure drop when benchmarked against traditional manufactured motorsport heat exchangers. A 55 mm reduction in length was also achieved, reducing the weight by 22% [7]. The chief product officer at Conflux Technology, Dan Woodford, stated in a recent article, “AM can offer a significant impact in the energy industry due to its potential for increasing heat transfer efficiency” [19].

Similarly, Zhang et al. [52] produced an additive manufactured manifold micro-channel heat exchanger for high-temperature applications and found an increase of 25% in heat transfer density compared to that of best conventional plate fin heat exchanger designs. Bacellar et al. [53] developed a proof of concept for a 1 kW air-to-water heat exchanger

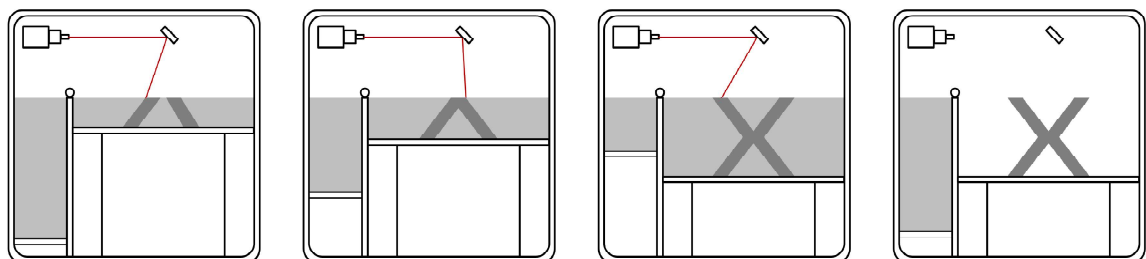


Figure 2.2: Basic schematic of a DMLS process. (Adapted from [31])

design employing small characteristic lengths with novel shapes. Bacellar found that an approximately 20% reduction in size and air pressure drop, an approximate 40% reduction in material volume, and an approximate 2% reduction in face area was achieved, compared to that of their state-of-the-art baseline heat exchanger. With shape optimisation, a potential of 50% reduction in size and 70% reduction in material usage could be achieved.

The advantages AM offers regarding heat transfer are listed and briefly discussed below. An in-depth overview can be found in studies done by McDonough [31] and Ngo [51].

- **Design freedom:**
AM allows for more complex designs as it is not limited to the constraints of conventional manufacturing methods.
- **Volumetric design:**
Designs can be optimised to reduce weight in ways not achievable with conventional manufacturing.
- **New channel designs and secondary/tertiary surfaces:**
More complex geometries for channel designs can be readily achieved.
- **Optimised shapes:**
Heat transfer devices can be designed to fit unusually shaped envelopes.
- **Direct realisation of computer models:**
Physical prototypes can be directly manufactured from CFD-optimised CAD models.
- **Weight reduction and part consolidation:**
Manufacturing complex geometries as a single part could lead to weight reduction. Multi-part integration is also possible with AM, integrating a heat exchanger into other unit operations, like a micro-turbine.
- **Customisation cost and reduced lead time:**
Rapid design iteration review and fabrication time is another advantage of AM. The fabrication cost is also independent of the actual design. The cost remains the same regardless of design complexity.

Challenges with AM

Despite the benefits of additive manufacturing, such as the freedom of design, customisation and the ability to print complex structures, there are a few drawbacks. These include high costs, limited applications in large structures and mass production, inferior and anisotropic mechanical properties, and limitation of materials and defects [12].

The challenges experienced with AM are listed below and briefly discussed; detailed discussions on the challenges of AM are available from studies done by McDonough [31] and Ngo [51].

- **Void formation:**

Void formation can occur between subsequent layers of material, reducing the interfacial bonding and negatively impacting the mechanical performance of the manufactured part.

- **Anisotropic microstructure and mechanical properties:**

The layer-by-layer manufacturing technique of AM results in different microstructures found within the material to that of the boundaries between the layers. These differences cause different mechanical behaviour when the part is subjected to load in the vertical versus the horizontal direction.

- **Divergence from design to execution:**

Tessellation concepts are typically employed by a CAD system to approximate the model when transferring from CAD to an AM format, often resulting in inaccuracies and defects, especially for curved surfaces. This can be resolved to some extent with very fine tessellation; however, the computed processing of the model and manufacturing will be time-consuming and complicated.

- **Surface roughness:**

The typical surface roughness of additive manufactured parts is greater than that of conventional manufacturing technologies. While beneficial for heat transfer in laminar flow, it results in a more significant pressure drop. Post-processing could improve surface finishes but is limited to the outside or accessible areas of the part and increases the processing time and cost.

- **Powder removal:**

Powder removal is crucial, as the trapped powder can sinter during manufacturing, post-processing or high-temperature operation, leading to design deviations, malfunctions, and failures. This is critical in heat transfer applications as the heating or cooling capacity would be altered.

- **Wall thickness:**

Reducing the resistive-thermal loss is crucial for the performance of compact heat exchangers and can be achieved by thinning the walls of the channels. This, however, results in distortion of the walls and wall porosity. Porous walls could lead to fluid leakage or fluid mixture, which is undesirable in most cases. The main factors influencing the achievable wall thickness include the build orientation, warping tolerances, repeatability, and reliability of the manufacturing process. However, optimising the manufacturing process to achieve thin walls can be time-consuming.

Materials

The AM process can utilise various materials, including, but not limited to, metals, alloys, polymers, composites, ceramics, and concrete [51]. The most commonly used materials in metal AM are steel or iron-based alloys, titanium and its alloys, and nickel-based alloys. For heat transfer applications, metals or alloys are preferred due to their higher operating temperatures and thermal conductivities. A comprehensive list of commonly used materials, properties and their respective AM process can be found in [30] and [54].

The high thermal conductivity of copper and aluminium, generally preferred in heat transfer applications, poses additional material handling challenges for AM. As a result of the rapid cooling experienced during the additive manufacturing process, micro-cracks tend to develop when producing parts with high-strength aluminium alloys. Copper's high

thermal conductivity rapidly conducts heat away from the melt pool leading to temperature gradients, potentially inducing additional stress build-up within the part, promoting cracks and delamination. Furthermore, copper typically exhibits low absorption of traditional wavelength heat sources due to its high surface reflectivity [31].

Titanium and its alloys are thoroughly studied and commercially used in various industries, including the aerospace and biomedical fields [51]. Studies have found that additive manufactured components produced with Ti achieved similar mechanical properties compared to those of their cast equivalents [54]. Shi et al. [55], reported relative densities up to 99.95% for Ti6Al4V (Ti64) using the SLM process with an ultimate tensile strength and optimal yield strength of 1150 MPa and 1200 MPa, respectively. Vandenbroucke and Kruth [56] achieved a relative density of 99.98% with Ti64.

Titanium's exceptional corrosion and erosion resistance, coupled with its low thermal expansion coefficient, makes it a highly suitable material for heat transfer equipment [31, 42, 57]. A study conducted by Arie et al. [58] on the experimental characterisation of an additive manufactured heat exchanger for dry cooling of power plants. Three heat exchangers of the same geometric design were produced, each in different materials, one in stainless steel (SS17-4), the second in titanium (Ti64) and lastly in aluminium alloy (AlSi10Mg) using the DMLS process.

The high thermal conductivity of the aluminium alloy led it to have the highest heat exchanger capacity, up to 400 W, followed by stainless-steel - 300 W and titanium - 200 W. Regarding pressure drop, the Ti64 heat exchanger exhibited a significantly lower pressure drop, 50–2000 Pa compared to that of the AlSi10Mg and SS17-4 heat exchangers' pressure drop of 200–7000 Pa. With pressure drop primarily dependent on geometry and not material properties, it was found that the deviation was caused by manufacturing inaccuracies, as mentioned earlier.

The quality of the micro-channels and fins were also assessed for all three heat exchangers and it was found that the Ti64 had a 20% deviation from the design specifications, while the AlSi10Mg and SS17-4 exhibited even more significant deviations from the design specifications.

Although the heat transfer capacity of a heat exchanger manufactured from Ti64 is lower than that of other more suitable heat transfer materials like AlSi10Mg or SS17-4, compared to conventional heat exchangers, the Ti64 heat exchanger showed superior performance. An increase of approximately 27% in the gravimetric heat transfer density compared to that of a state-of-the-art dry cooling heat exchanger for application in power plants was reported. Furthermore, the high strength of Ti64 allows heat exchanger channel walls to be thinned down, decreasing thermal resistance and increasing the heat transfer [59].

2.4 Compact heat exchangers

In heat exchangers, the ratio of the actual heat transferred to the maximum possible heat transfer is known as the effectiveness of the heat exchanger [60, p. 613–622]. The high effectiveness achievable with compact heat exchangers is especially beneficial when implemented with power-producing or power-consuming systems [26].

Compact heat exchangers have a very high ratio of heat transfer area to the overall heat

exchanger volume, typically greater than $700 \text{ m}^2/\text{m}^3$ for gas-gas heat exchangers and greater than $400 \text{ m}^2/\text{m}^3$ for liquid-gas applications. This enables these types of heat exchangers to offer a much higher heating and cooling load due to their higher heat transfer area and heat transfer coefficient compared to their counterparts [3, p. 6].

A widely used channel size classification is proposed by Mehendale et al. [61, 62]:

- Micro-scale - $1 - 100 \text{ } \mu\text{m}$ (micro-structured heat exchanger)
- Meso-scale - $100 \text{ } \mu\text{m} - 1 \text{ mm}$ (meso- or milli-structured heat exchanger)
- Macro-scale - $1 - 6 \text{ mm}$ (compact heat exchanger)
- Conventional - $> 6 \text{ mm}$ (Conventional heat exchanger)

Counterflow heat exchangers have some advantages over parallel-flow heat exchangers (also known as co-flow heat exchangers). As seen with the temperature profiles in Figure 2.3, the temperature of the cooling fluid can be raised to a higher temperature than the hot fluids' exit temperature, in effect having a larger temperature range [60, p. 613–622]. This results in less required surface area for the same amount of heat transfer compared to parallel-flow heat exchangers. Furthermore, fewer thermal stresses are introduced as the temperature difference between the fluids is more uniform throughout the heat exchanger; this also ensures a more uniform heat transfer rate [63]. ALFA LAVAL is a company which uses compact counterflow heat exchangers for heat recovery systems, stating that they are five times more efficient than shell-and-tube heat exchangers, making heat recovery profitable even where the energy sources have traditionally been deemed worthless [17].

With the freedom of design AM offers, engineers have the opportunity to improve the heat transfer process and minimise pressure drop while reducing the size, weight and parts required to produce these types of heat exchangers [19, 24].

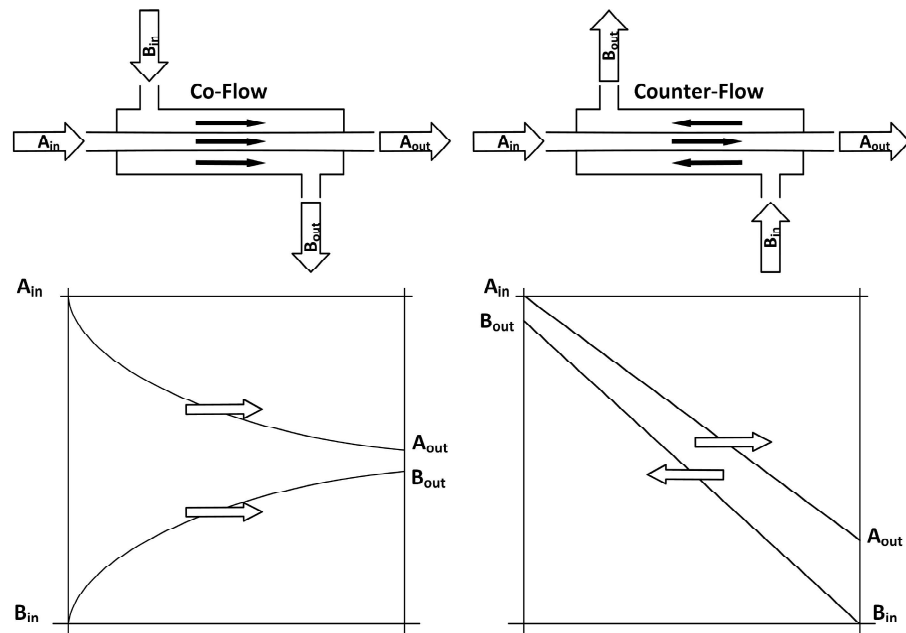


Figure 2.3: Schematic representation of co-flow (*left*) and counterflow (*right*) heat exchangers and their respective temperature profiles. (Adapted from [64])

2.5 Conclusion

This chapter reviewed available literature on ORC and micro-ORC systems and found that even with a cycle efficiency lower than 10%, micro-ORC systems can produce upwards of 11% of an average household's yearly energy usage. Implementing higher-efficiency small-scale expanders and compact heat exchangers within these systems could further improve low-grade waste heat recovery performance. Traditional manufacturing methods for compact heat exchangers either limit the design complexity or require more manufacturing processes, incurring additional costs and time penalties.

Additive manufacturing is the collective term used for various process technologies. With numerous studies reporting positive results for AM within the heat transfer sector, AM is deemed a viable method for producing intricate compact heat exchangers in various materials. The advantages of AM allow engineers and designers to explore new possibilities of designs previously not possible with traditional manufacturing methods. Table 2.2 lists these technologies with their application and viable materials.

Challenges are present with any manufacturing technology and AM is no different. Void formation, powder removal, surface roughness and wall thickness are some of the challenges experienced with AM. These challenges can be overcome with process and design optimisation; however, this leads to additional processing time.

A reasonable amount of materials are available for AM, and the study focused on the use of Ti64 for the manufacture of a compact counterflow heat exchanger. Although titanium is not the first material to come to mind when considering heat transfer, numerous studies have utilised Ti64 for additively manufactured compact heat exchangers and found good results compared with conventional designs. Ti64 is well-studied and optimised for AM, making it a good choice when considering proof-of-concept experimental research using AM as the manufacturing method for compact heat exchangers. Literature on additively manufactured compact heat exchangers and micro-ORC systems is readily available, with no mention of additively manufactured compact heat exchangers for application in ORC systems.

Chapter 3

Modelling and Design

3.1 Introduction

Effective modelling of the heat exchanger enables accurate predictions of the manufactured heat exchanger's performance. These predictions can be used to compare different scenarios and designs so that the optimal designs can be chosen for manufacturing. Modelling the heat exchanger beforehand thus saves time, cost and material. The theoretical procedure for modelling the compact counterflow heat exchanger is described in this chapter.

The model formulated for this research would also guide future research and aid in future design procedures for compact counterflow heat exchangers for additively manufactured ORCs.

3.2 Modelling and Design Overview

The model should accurately predict the heat transfer and pressure drop within the heat exchanger by identifying its material characteristics, channel geometry and input variables. The predicted results would then be compared to the required results for the heat exchanger and, if necessary, be adjusted accordingly.

A study by Hasan et al.[65] states that channel shape, size and length significantly influence fluid flow and heat transfer. In their study, Hasan et al. explored various channel shapes and found that circular channels gave the best overall performance.

The design methodology used is based on the methods presented by [6, 66]. The theoretical design results were compared against the experimental results as discussed in Chapter 6. This would indicate if the formulated theoretical model is valid for designing additive manufactured compact counterflow micro heat exchangers for application in micro ORCs.

The main objectives governing the design of the heat exchanger are to maximise the heat transfer and minimise the pressure drop whilst keeping the heat exchanger to an acceptable size. Raising the heat transfer increases the pressure drop through the heat exchanger. Thus the design should be well balanced between heat transfer and permissible pressure drop.

3.3 Design Theory

The overall heat transfer rate can be calculated using the following equation when both inlet and outlet temperatures are known [6]:

$$q = UA\Delta T \quad (3.1)$$

where U is the overall heat transfer coefficient, A the total heat transfer area and ΔT the overall temperature difference between the fluids. The conductance UA can be calculated by means of the following expression[6, 66]:

$$UA = \frac{1}{\frac{1}{\eta_{o1}h_1A_{wall}} + \frac{\Delta x}{kA_{wall}} + \frac{1}{\eta_{o2}h_2A_{wall}}} \quad (3.2)$$

where $\frac{1}{hA}$ represents the convective resistance and $\frac{\Delta x}{kA}$ the conduction resistance, h as the convective coefficient, Δx the material thickness, k the material's thermal conductivity and η_o the surface efficiency of the fin array. The convective coefficient h can be calculated using the following equation:

$$h = \frac{Nu \cdot k}{D_h} \quad (3.3)$$

where k is the thermal conductivity of the working fluid, Nu the Nusselt number, and D_h the hydraulic diameter.

The Nusselt number correlation should be correctly chosen for the type of flow and roughness of the tube. The following equations can be used for determining whether the flows in the channels are fully developed or developing [67]:

$$L_e = 0.06ReD_h \text{ for laminar flow } (Re < 2300) \quad (3.4a)$$

$$L_e = 4.4Re^{1.6}D_h \text{ for turbulent flow } (Re > 4000) \quad (3.4b)$$

L_e represents the channel's hydrodynamic entrance length. Developing flow occurs up to the point of the entrance length, called the entrance region. Developing flow has a thinner boundary layer which aids in the total heat transferred.

Careful consideration should be taken when selecting a Nusselt number expression, as many equations and correlations exist for calculating the Nusselt number. Sources explored include equations from Sieder and Tate, Gnielinski, Nusselt, Dittus and Boelter and many more, all with different degrees of accuracy [66].

The most common expression used to calculate the Nusselt number for turbulent developed flow in smooth tubes as recommended by Dittus and Boelter is [6]:

$$Nu = 0.023 Re_{D_h}^{0.8} Pr^n \quad (3.5)$$

All properties in the equation are evaluated at the mean bulk temperature, and the exponent n has the following values:

$$n = \begin{cases} 0.4 & \text{for heating of the fluid} \\ 0.3 & \text{for cooling of the fluid} \end{cases}$$

With the Reynolds number also evaluated at the mean bulk temperature and the hydraulic diameter:

$$Re_{D_h} = \frac{D_h G}{\mu} \quad (3.6)$$

D_h represents the hydraulic diameter, G is the mass velocity and μ the dynamic viscosity.

The hydraulic diameter for any given channel can be obtained from the following equation:

$$D_h = \frac{4A_{cs}}{p} \quad (3.7)$$

A_{cs} represents the cross sectional area of the channel and p the wetted perimeter.

G , the mass velocity, can be obtained with the following equation [6]:

$$G = \frac{\dot{m}}{A_c} \quad (3.8)$$

with $\dot{m}$ being the mass flow rate through the channels and A_c the free flow area.

The effectiveness-NTU method was used for the analysis and design as the exit temperatures were unknown. The LMTD method requires an iterative procedure for analysis with unknown temperatures. The heat transfer rate is calculated as follows [6]:

$$q = \dot{m}_h c_{ph} (T_{hi} - T_{ho}) = \dot{m}_c c_{pc} (T_{co} - T_{ci}) \quad (3.9)$$

where q represents the heat transfer rate, $\dot{m}$ the respective mass flow rate, c_p the specific heat of the fluid and T_i ; T_o the inlet and outlet temperatures.

The effectiveness (ε) of a heat exchanger can be defined as the ratio of actual heat transfer (q) to the maximum possible heat transfer (q_{max}), given by [6]:

$$\varepsilon = \frac{q}{q_{max}} \quad (3.10)$$

q_{max} can be calculated with:

$$q_{max} = (\dot{m}c)_{min}(T_{hi} - T_{ci}) \quad (3.11)$$

where the term $\dot{m}c$ is known as the capacity rate (C) and the minimum fluid (C_{min}) can be either the hot or the cold fluid.

$$\varepsilon_h = \frac{C_h(T_{hi} - T_{ho})}{C_h(T_{hi} - T_{ci})} = \frac{(T_{hi} - T_{ho})}{(T_{hi} - T_{ci})} \quad (3.12)$$

$$\varepsilon_c = \frac{C_c(T_{co} - T_{ci})}{C_c(T_{hi} - T_{ci})} = \frac{(T_{co} - T_{ci})}{(T_{hi} - T_{ci})} \quad (3.13)$$

The effectiveness can also be generally expressed as follows:

$$\varepsilon = \frac{\Delta T(\text{minimum fluid})}{\text{Maximum temperature difference in heat exchanger}} \quad (3.14)$$

The minimum fluid always experiences the larger temperature difference [6].

Additionally the effectiveness of a counterflow heat exchanger can also be calculated using the following expression [6]:

$$\varepsilon = \frac{1 - \exp[-NTU(1 - C^*)]}{1 - C^* \exp[-NTU(1 - C^*)]} \quad \text{for } C^* < 1 \quad (3.15a)$$

$$\varepsilon = \frac{NTU}{NTU + 1} \quad \text{for } C^* = 1 \quad (3.15b)$$

Where NTU is the number of transfer units and can be obtained with:

$$NTU = \frac{UA}{C_{min}} \quad (3.16)$$

and C^* the capacity ratio:

$$C^* = \frac{C_{min}}{C_{max}} \quad (3.17)$$

The vertical walls within the core of the heat exchanger act as fins with no convection at the tip, and their efficiency is calculated with:

$$\eta_f = \frac{\tanh(mL)}{mL} \quad (3.18)$$

with L the fin length and m :

$$m = \sqrt{\frac{hP_f}{kA_{cf}}} \quad (3.19)$$

The convective heat transfer coefficient (h) is assumed to be similar to the wall convective coefficient [66]. P_f is the fin perimeter, and A_{cf} the fin cross-sectional area.

The fin efficiency is the ratio of actual heat transferred by the fin (q_f) to the maximum possible heat transfer by the fin (q_{fmax}), calculated with:

$$q_{fmax} = hP_fL(T_b - T_\infty) \quad (3.20)$$

where T_b represents the temperature of the base (in this case, the horizontal wall), and T_∞ the free stream temperature.

The surface efficiency of the fin array η_o can be obtained with:

$$\eta_o = 1 - (1 - \eta_f) \cdot \frac{A_f}{A_t} \quad (3.21)$$

with A_f as the exposed fin area and A_t as the total heat transfer area [3].

For the calculation of the core pressure drop in the compact heat exchanger, the most commonly used expression comes from Kays and London [68]:

$$\frac{\Delta P}{P_1} = \frac{G^2}{2g} \cdot \frac{v^2}{P_1} \left[(K_c + 1 - \sigma^2) + 2 \left(\frac{v_2}{v_1} - 1 \right) + f_F \frac{A_{ht}}{A_c} \cdot \frac{v_m}{v_1} - (1 - \sigma^2 - K_e) \frac{v_2}{v_1} \right] \quad (3.22)$$

with the term $(K_c + 1 - \sigma^2)$ being the entrance effect, K_c represents the entrance loss coefficient and is calculated with:

$$K_c \cong 0.42(1 - \sigma^2)^2 \quad (3.23)$$

σ is the ratio of the free-flow area to the frontal area.

The term, $2 \left(\frac{v_2}{v_1} - 1 \right)$ represents the flow acceleration, where v_2 and v_1 are the entering and exiting specific volumes. The term defines the core friction: $f_F \frac{A_{ht}}{A_c} \cdot \frac{v_m}{v_1}$.

f_F represents the core Fanning friction factor, A_{ht} represents the total heat transfer area, A_c is the free-flow area and v_m is the mean specific volume. Kays and London suggest $v_m = \frac{1}{2}(v_1 + v_2)$ for C^* near unity.

The core Fanning friction factor is calculated with the following equation [69]:

$$f_F = 2 \left[\left(\frac{8}{Re} \right)^{12} + \frac{1}{(A + B)^{1.5}} \right]^{1/12} \quad (3.24)$$

With A and B

$$A = \left[2.457 \ln \frac{1}{(7/Re)^{0.9} + 0.27\varepsilon/d} \right]^{16} \quad (3.25a)$$

$$B = \left(\frac{37530}{Re} \right)^{16} \quad (3.25b)$$

The last term of the expression represents the exit effect, $(1 - \sigma^2 - K_e) \frac{v_2}{v_1}$, with K_e being the exit loss coefficient and calculated by:

$$K_e \cong (1 - \sigma)^2 \quad (3.26)$$

3.4 Numerical Simulation

3.4.1 Engineering Equation Solver

Engineering Equation Solver (EES) was used to implement the above-mentioned theoretical model and find the design parameters for the CAD design. Performance comparisons of different design configurations can be plotted within EES, and the optimal theoretical design parameters can be identified. Channel size, length and wall thickness were altered and compared to the baseline design. The initial baseline design selected for performance comparisons is shown in Table 3.1. The results shown here indicate whether the baseline design changes have a positive or negative effect on the heat exchanger's performance. The theoretical EES model for the baseline design can be seen in Appendix D.

Table 3.1: Baseline characteristics

Controlled Variables		
	Input	Output
Cold Fluid Temperature	± 20 °C	Calculated
Warm Fluid Temperature	± 60 °C	Calculated
Cold Fluid Pressure	± 1 bar	Calculated
Warm Fluid Pressure	± 1 bar	Calculated

Table 3.1: Baseline Characteristics (*continued*)

Fixed Variables	
	Value
Mass Flow Rate	1–10 l/min
Channel Height	0.5 mm
Channel Width	0.5 mm
Channel Length	30 mm
Channel Wall Thickness	0.5 mm
Number of Channels	220

3.4.2 Computational Fluid Dynamics

Computational Fluid Dynamics (CFD) was used to further analyse the heat exchangers and better understand the fluid flow within the core. This, however, did not influence the heat exchanger design and was done post-experimentation. The Flow Simulation add-in package from SOLIDWORKS was used for the CFD analysis. SOLIDWORKS Flow Simulation uses the Navier-Stokes equations, turbulent models, and models for physical phenomena for numerical simulation [70].

Fluid Regions

The Navier-Stokes equations solve the fluid regions with turbulent flow predicted with the Favre-averaged Navier-Stokes equations. The Governing equations are made up of mass, momentum and energy conservation laws:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (3.27)$$

$$\frac{\partial(\rho u_i)}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_i u_j) + \frac{\partial P}{\partial x_i} = \frac{\partial}{\partial x_j}(\tau_{ij} + \tau_{ij}^R) + S_i \quad (3.28)$$

$$\frac{\partial \rho H}{\partial t} + \frac{\partial \rho u_i H}{\partial x_i} = \frac{\partial}{\partial x_i} (u_j(\tau_{ij} + \tau_{ij}^R) + q_i) + \frac{\partial p}{\partial t} - \tau_{ij}^R \frac{\partial u_i}{\partial x_j} + \rho \varepsilon + S_i u_i + Q_H \quad (3.29)$$

with

$$H = h + \frac{u^2}{2}$$

$$\frac{\partial \rho E}{\partial t} + \frac{\partial \rho u_i \left(E + \frac{P}{\rho} \right)}{\partial x_i} = \frac{\partial}{\partial x_i} (u_j(\tau_{ij} + \tau_{ij}^R) + q_i) - \tau_{ij}^R \frac{\partial u_i}{\partial x_j} + \rho \varepsilon + S_i u_i + Q_H \quad (3.30)$$

with

$$E = e + \frac{u^2}{2}$$

where Equation 3.27 represents continuity, Equation 3.28 represents the Navier-Stokes equation for momentum and Equation 3.29 represents the energy conservation law; energy Equation 3.30 is used for calculation of high-speed compressible flows and flows with shock waves.

Transport equations for the turbulent kinetic energy and its dissipation rate are employed using the modified $k - \varepsilon$ model proposed by Lam and Bremhorst [71]. This model with damping functions describes laminar, turbulent, and transitional flows of homogeneous fluids applying the following turbulence conservation laws:

$$\frac{\partial \rho k}{\partial t} + \frac{\partial \rho k u_i}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_i} \right) + \tau_{ij}^R \frac{\partial u_i}{\partial x_j} - \rho \varepsilon + \mu_t P_B, \quad (3.31)$$

$$\frac{\partial \rho \varepsilon}{\partial t} + \frac{\partial \rho \varepsilon u_i}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_i} \right) + C_{\varepsilon 1} \frac{\varepsilon}{k} \left(f_1 \tau_{ij}^R \frac{\partial u_i}{\partial x_j} + C_B \mu_t P_B \right) - f_2 C_{\varepsilon 2} \frac{\rho \varepsilon^2}{k} \quad (3.32)$$

$$\begin{aligned} \tau_{ij} &= \mu s_{ij}, \\ \tau_{ij}^R &= \mu_t s_{ij} - \frac{2}{3} \rho k \delta_{ij}, \\ s_{ij} &= \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \end{aligned} \quad (3.33)$$

$$P_B = -\frac{g_i}{\sigma_B} \frac{1}{\rho} \frac{\partial \rho}{\partial x_i} \quad (3.34)$$

where:

$$\begin{array}{lll} C_\mu = 0.09 & \sigma_k = 1 & C_B = 1 \quad \text{if } P_B > 0 \\ C_{\varepsilon 1} = 1.44 & \sigma_\varepsilon = 1.3 & C_B = 0 \quad \text{if } P_B < 0 \\ C_{\varepsilon 2} = 1.92 & \sigma_B = 0.9 & \end{array}$$

turbulent viscosity is determined by:

$$\mu_t = f_\mu \cdot \frac{C_\mu \rho k^2}{\varepsilon} \quad (3.35)$$

The Lam and Bremhorst damping functions f_μ , f_1 and f_2 are determined by:

$$f_\mu = (1 - e^{-0.025R_y})^2 \cdot \left(1 + \frac{20.5}{R_t}\right) \quad (3.36)$$

$$f_1 = 1 + \left(\frac{0.05}{f_\mu}\right)^3$$

$$f_2 = 1 - e^{R_t^2} \quad (3.37)$$

with:

$$R_y = \frac{\rho\sqrt{k}y}{\mu}, \quad (3.38)$$

$$R_t = \frac{\rho k^2}{\mu\varepsilon} \quad (3.39)$$

where y represents the distance from point to wall. The approach reverts back to the original $k - \varepsilon$ model if $f_\mu = 1, f_1 = 1, f_2 = 1$.

Heat flux is defined by:

$$q_i = \left(\frac{\mu}{\text{Pr}} + \frac{\mu_t}{\sigma_c}\right) \frac{\partial h}{\partial x_i}, \quad i = 1, 2, 3 \quad (3.40)$$

where $\sigma_c = 0.9$, Pr is the Prandtl Number and h is the thermal enthalpy.

Computational tasks are finally specified by the definition of their geometry, boundary and initial conditions, all of which are defined on the native CAD model.

Solid Regions

Heat conduction and direct electrical current are the two kinds of physical phenomena that SOLIDWORKS Flow Simulation calculates within solid regions, where the resulting Joule heating represents the heat source in the energy equation [72]. Equations 3.29 and 3.30 describe the heat transfer in fluids with Equation 3.40 defining the heat flux.

The solid media heat conduction phenomenon is described by:

$$\frac{\partial \rho e}{\partial t} = \frac{\partial}{\partial x_i} \left(\lambda_i \frac{\partial T}{\partial x_i} \right) + Q_H \quad (3.41)$$

The specific internal energy: $e = c \cdot T$ with c representing the specific heat; Q_H is the specific heat release/absorption rate per unit volume. λ_i are the eigenvalues of the thermal conductivity tensor and are presumed diagonal in the considered coordinate system. $\lambda_1 = \lambda_2 = \lambda_3 = \lambda$ for an isotropic medium. The specific Joule heat release (Q_J) may be included in Q_H with the presence of electric current and is defined by $Q_J = r \cdot j^2$; with j representing the electric current density.

The electric current density vector (i) Equation 3.42 is determined from the steady-state Laplace Equation 3.43 via the electric potential $\phi[V]$:

$$\mathbf{i} = - \left(\frac{1}{r_{11}} \frac{\partial \phi}{\partial x_1}, \frac{1}{r_{22}} \frac{\partial \phi}{\partial x_2}, \frac{1}{r_{33}} \frac{\partial \phi}{\partial x_3} \right) \quad (3.42)$$

$$\frac{\partial}{\partial x_i} \left(\frac{1}{r_{ii}} \frac{\partial \phi}{\partial x_i} \right) = 0 \quad (3.43)$$

where r_{ii} represents the temperature-dependent electrical resistivity in the i -th coordinate direction.

Sub-domains containing electrically conductive materials are solved numerically with the Laplace equation, where dielectric solids and fluid areas are automatically excluded in such sub-domains. A boundary condition specifying the total electric current over a surface area $I[A]$ or electric potential $\phi[V]$ may be set by the user.

Zero resistance is considered by default between two electrically-conductive solids within the sub-domain if no user electrical-contact resistance is specified. Specifying a contact resistance on a surface produces corresponding Joule heating as current passes through, which results in a surface heat source.

Thermal contact resistances between several solid materials within a single solid are considered when calculating heat conduction, and as a result a solid temperature step appears on the contact surfaces. This also applies to a solid in contact with a fluid and is specified via the material's thermal conductivity and layer thickness.

Calculating the energy exchange between a solid and fluid is done via the heat flux in the normal direction of the solid/fluid interface with various factors taken into account, such as surface temperature, fluid boundary layer and radiation heat exchange.

Boundary Layer

An original Two-Scale Function (2SWF) approach is used to couple the boundary layer calculation with the solution of the bulk flow. The approach consists of two methods; namely, the thick-boundary-layer and thin-boundary-layer approach.

The *thick-boundary-layer approach* is used when sufficient cells are within a boundary layer (>10). Laminar boundary layers are solved as part of the core flow calculations with the Navier-Stokes equations, and turbulent boundary layers are solved with a modified wall-function approach. The modified wall-function utilises the full velocity profile instead of the logarithmic profile as proposed by Van Driest [73].

$$u^* = \int_0^{y^*} \frac{2 \cdot dy^*}{1 + \sqrt{1 + 4 \cdot \kappa^2 \cdot y_*^2 \left[1 - \exp \left(-\frac{y_*}{A_v} \right) \right]^2}} \quad (3.44)$$

where $\kappa = 0.4054$ is the Karman constant and $A_v = 26$ is the Van Driest coefficient; the other assumptions are as with the classic wall function approach.

The *thin-boundary-layer approach* is used if the number of cells within a boundary layer is insufficient for flow and thermal profile determination. This approach incorporates the Prandtl boundary layer equations along the wall and are solved within the fluid flow paths along the walls.

Successive approximations based on the Shvets trail functions technology are used to solve laminar boundary layer equations [74]. Whereas, turbulent or transitional boundary layers are solved with a generalisation of this method, where Van Driest's hypothesis on the mixing length in turbulent boundary layers is used [73].

Semi-empirical coefficients are used to account for the impact of surface roughness and turbulence in the external flow on the boundary layer. These coefficients help adjust the calculations of wall shear stress and the heat flux from the fluid to the wall. Corresponding empirical and semi-empirical models are also employed to account for various body forces, fluid compressibility, and turbulent kinetic energy dissipation.

The boundary layer thickness (δ), heat flux from the fluid to the wall (q_w^e), and the wall shear stress (τ_w^e) form part of the boundary conditions for the Navier-Stokes equations, which are obtained via SOLIDWORKS Flow Simulation from the boundary layer calculations.

$$\tau_w = \tau_w^e, \quad (3.45)$$

$$q_w = q_w^e, \quad (3.46)$$

The turbulence equilibrium condition in the near-wall computational mesh cell determines the boundary conditions for k and ε :

$$\frac{\partial k}{\partial y} = 0, \quad (3.47)$$

$$\varepsilon = \frac{C_\mu^{0.75} k^{1.5}}{Ky}. \quad (3.48)$$

A compilation of the approaches are also used in intermediate cases. See the white paper of Sobachkin and Dumnov [72].

The CAD models were set up similarly for CFD, with the same parameters and goals. Table 3.2 shows the summarised setup used for the baseline model. A full report of the CFD set-up can be seen in Appendix E. A basic mesh was used for the heat exchanger, while a local mesh was created for analysing the heat exchanger core (*c.f.* Figure 3.1). Boundary conditions were added for the flow inlet and outlets with their respective goals. Additional goals were added to capture the required data, temperature for the solid and fluids, and velocity for the fluids.

Table 3.2: CFD set-up parameters for baseline model

Computational Domain	
Control planes in X direction	-0.028 mm Min 0.028 mm Max
Control planes in Y direction	-0.007 mm Min 0.007 mm Max
Control planes in Z direction	-0.056 mm Min 0.056 mm Max
Mesh Settings	
Basic Mesh	
Automatic initial mesh	Off
Number of cells in X	20
Number of cells in Y	10
Number of cells in Z	46
Small Solid feature refinement level	2
Local Mesh	
Solid/fluid interface refinement level	1
Narrow channels refinement	On
No. of cells across a narrow channel	10
Maximum channel refinement	4
Number of Cells	
Cells	5 039 158
Fluid Cells	2 712 255
Solid Cells	2 326 903

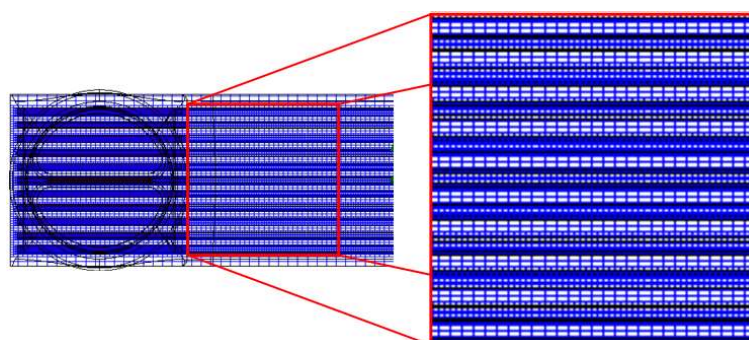


Figure 3.1: Local mesh plot: 0.5 mm design

3.5 Summary and Conclusion

A theoretical model was developed for implementation in EES to numerically simulate the performance characteristics of an additively manufactured compact counterflow heat exchanger. The model was used to identify the design parameters for the heat exchanger design. A baseline model was identified and used to compare different design aspects, discussed in Chapter 6. The numerical simulation was also set up to accommodate different fluids and fluid states. For further analysis of the fluid flow and heat transfer within the heat exchanger core, a summary of the initial setup for CFD was also included in the chapter.

The next chapter discusses the CAD design and design considerations to consider for designing for additive manufacturing. The experimental investigation chapter (Chapter 5) describes the validation for the theoretical model and numerical simulation presented in this chapter.

Chapter 4

Design and considerations for additively manufactured heat exchangers

4.1 Introduction

This section discusses the design of the heat exchangers for additive manufacturing (AM), which includes the design considerations. This chapter also discusses significant difficulties experienced during the manufacturing process and some possible solutions.

4.2 CAD design

The heat exchangers' initial computer aided design (CAD) was developed following the theoretical model described in Chapter 3. With guidance from the Centre for Rapid Prototyping and Manufacturing (CRPM) team at Central University of Technology, Free State, these designs were prepared for AM. The heat exchangers consist of an array of square channels differing in length and size. The channel wall thickness was initially fixed between the designs. After review, they were changed to reduce warpage and ensure the integrity of the channel walls within the heat exchanger core, resulting in different thicknesses chosen for the 1 mm and 2 mm designs. This allowed the author to experimentally compare the effect of channel length, channel wall thickness, and channel size on heat transfer and pressure drop characteristics.

For ease of experimental testing, the fluid inlets and outlets were kept on the same side; the working fluids are split respectively within the manifolds into rows of channels. Figure 4.1 shows the fluid subdomains within the heat exchanger. The target heat transfer capacity for the baseline design was 2 kW. The eventual designs, influenced by cost, additive manufacturing options and constraints, containment of internal leakage and limitations in budget and schedule, led to heat transfer capacities of 2.1 kW, 1.84 kW and 1.9 kW for the 0.5-, 1- and 2 mm channel size heat exchangers respectively.

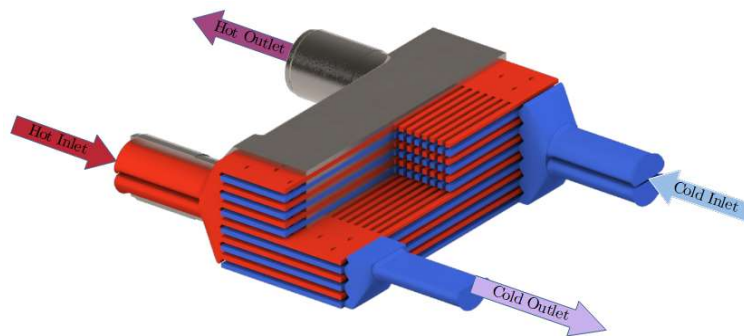


Figure 4.1: Fluid sub-domains

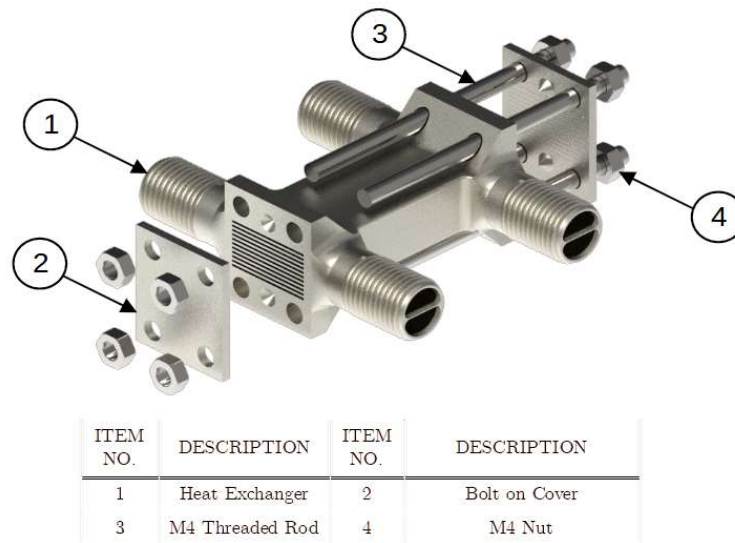


Figure 4.2: Baseline design: exploded view

A bolt-on cover was incorporated into the design for ease of access to remove residual powder after manufacturing. The cover further allows servicing and inspecting the heat exchanger core during its life (*cf.* Figure 4.2).

Vertical overhangs were chamfered 45° where possible to ensure ease of manufacturing and to mitigate the need for supports. $1/4''$ BSP thread was added to the heat exchangers' inlet and outlet nozzles, eliminating the need for thread cutting during post-processing.

4.2.1 0.5 mm heat exchanger

The 0.5 mm heat exchanger core consists of 220 square channels in an array of 11×20 with a channel size of 0.5×0.5 mm; this design's estimated heat transfer rate is 2100 W. The heat exchanger has a 1 mm thick outer wall with 0.5 mm thick walls separating the channels. Overall dimensions for the heat exchanger design are $60 \times 71.5 \times 26.5$ mm with a core of $30 \times 21.5 \times 12.5$ mm ($L \times W \times H$), Figure 4.3 shows the final CAD design for the 0.5 mm heat exchanger.

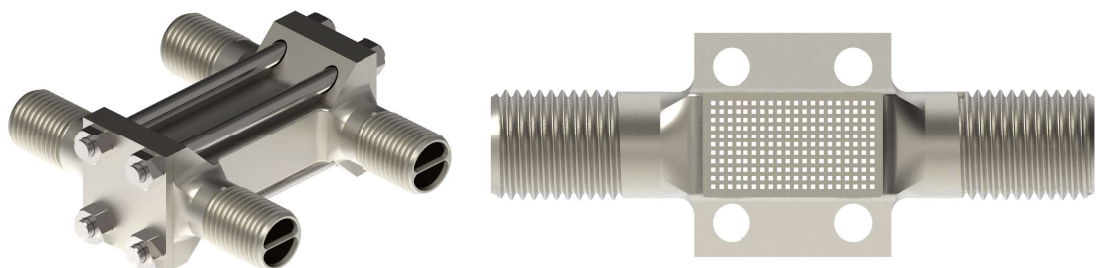


Figure 4.3: 0.5 mm heat exchanger configuration: isometric view (left), core cross-section (right)

4.2.2 1 mm heat exchanger

The 1 mm heat exchanger core consists of 192 square channels in an array of 12x16 with a channel size of 1x1 mm; the estimated heat transfer rate for this design is 1840 W. The heat exchanger has a 1 mm thick outer wall with 0.8 mm thick walls separating the channels. Overall dimensions for the heat exchanger design are 70x80x37.8 mm with a core of 40x30x22.8 mm ($L \times W \times H$). Figure 4.4 shows the final CAD design for the 1 mm heat exchanger.

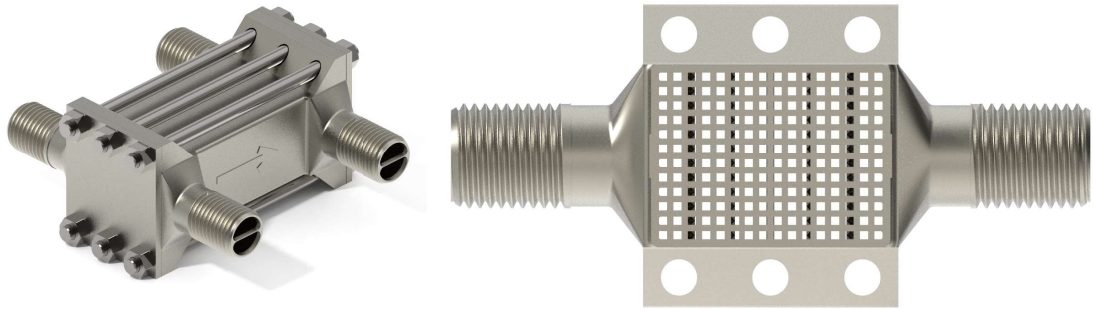


Figure 4.4: 1 mm heat exchanger configuration: isometric view (left), core cross-section (right)

4.2.3 2 mm heat exchanger

The 2 mm heat exchanger core consists of 676 square channels in an array of 26x26 with a channel size of 2x2 mm. The estimated heat transfer rate for this design is 1900 W. The heat exchanger has a 1 mm thick outer wall with 0.8 mm thick walls separating the channels. Overall dimensions for the heat exchanger design are 76x101.4x126.6 mm with a core of 50x76x76 mm ($L \times W \times H$). Figure 4.5 shows the final CAD design for the 2 mm heat exchanger.

A comparison of the heat exchanger design sizes can be seen in Figure 4.6.

4.3 Design considerations for AM

Additive manufacturing broadens our manufacturing capabilities, allowing the manufacture of complex components and systems that was previously impossible with traditional manufacturing methods [24, 75]. Although a versatile and capable technology, there are still limits to consider when developing components/prototypes for AM. Manufacturing orientation, overhangs and post-processing are vital aspects to keep in mind. The manufacturing difficulties experienced during the study are summarised in the section below. See also the paper by Venter, Jacobs, and Du Preez [76] for a detailed discussion on the design considerations for additively manufactured heat exchangers.

Figures 4.7, 4.8, and 4.9 show the design evolution of the 0.5 mm, 1 mm and 2 mm heat exchangers from conceptual design to the manufactured design. Version 1, the initial designs, driven by the results obtained from theoretical calculations, were presented to CRPM for review to assess their manufacturability by AM.

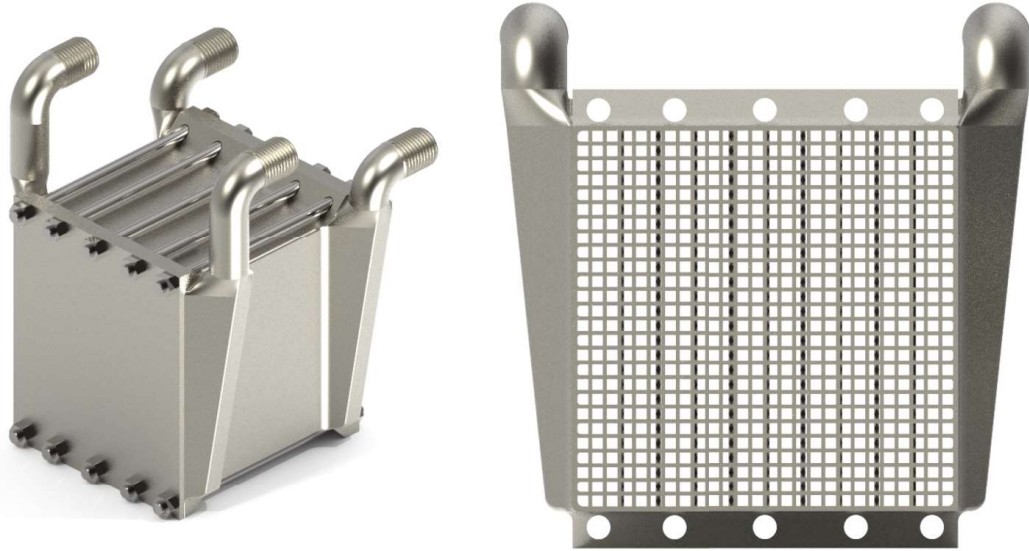


Figure 4.5: 2 mm heat exchanger configuration: isometric view (left), core cross-section (right)

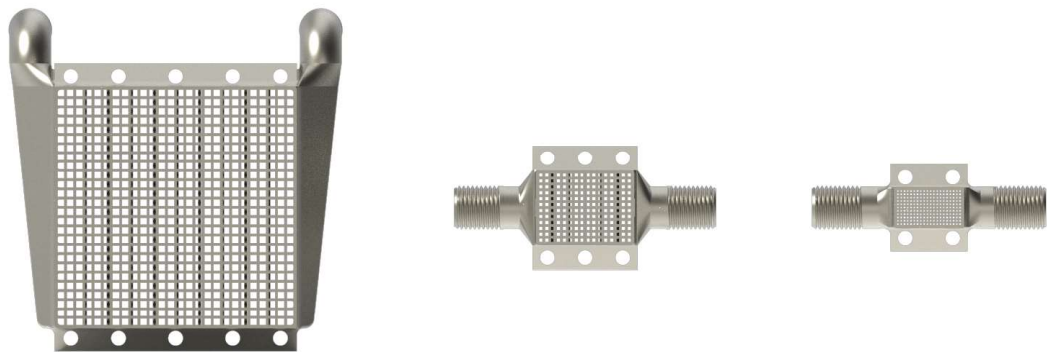


Figure 4.6: Channel size comparison, 2 mm, 1 mm, 0.5 mm

The designs were reviewed and refined for AM. In version 2, chamfers were added to the flanges, creating self-supporting overhangs which eliminated the need for temporary support during manufacturing. Permanent supports were also added to the inside of the nozzle sections, as removing internal supporting material can become tedious, increasing post-processing time and cost. The angle of the transition piece between the nozzle section and manifold was also increased to above 35° . This ensures the section is self-supporting, reducing material usage (no additional supporting material used) and time required for post-processing.

Version 3 shows a refined design. Sharp edges were replaced with filleted corners to reduce stress buildup in corners. Threaded inlets and outlets were also added. Internal manifold supports were added to the 1 mm and 2 mm designs to minimise warpage within the manifold section.

Version 4 is the final design that was successfully manufactured. The radii on all filleted corners were increased, and the channel wall thickness for the 1 mm and 2 mm heat exchanger configuration was increased from 0.5 mm to 0.8 mm.

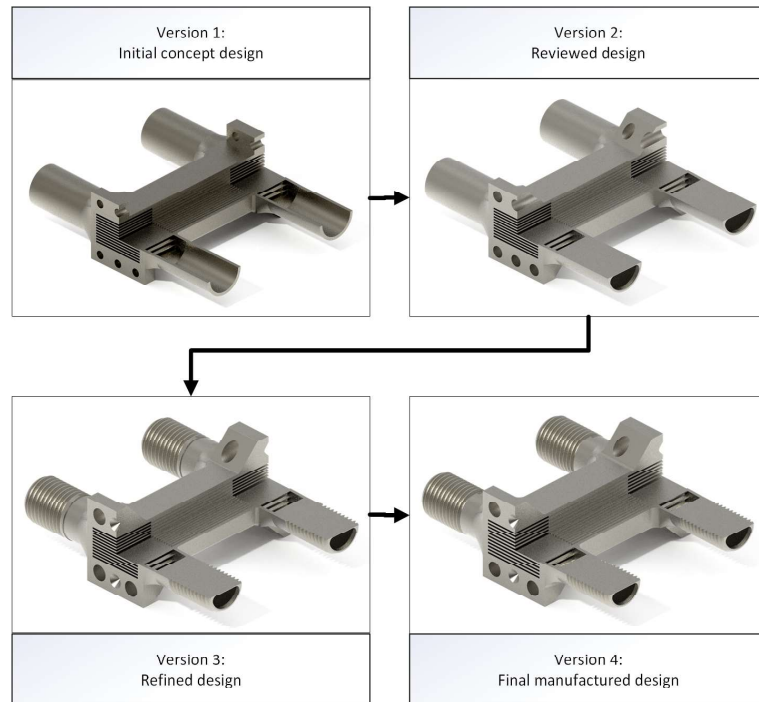


Figure 4.7: 0.5 mm design revisions

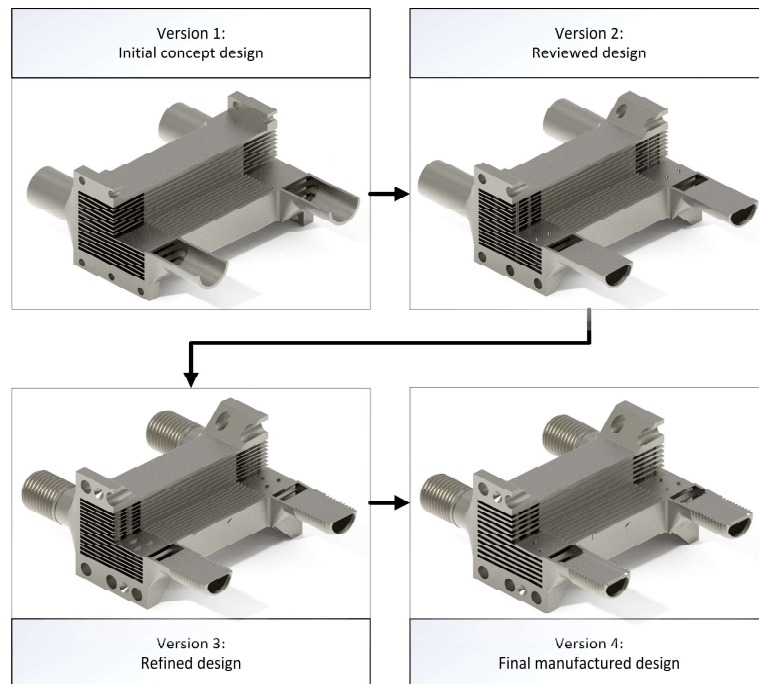


Figure 4.8: 1 mm design revisions

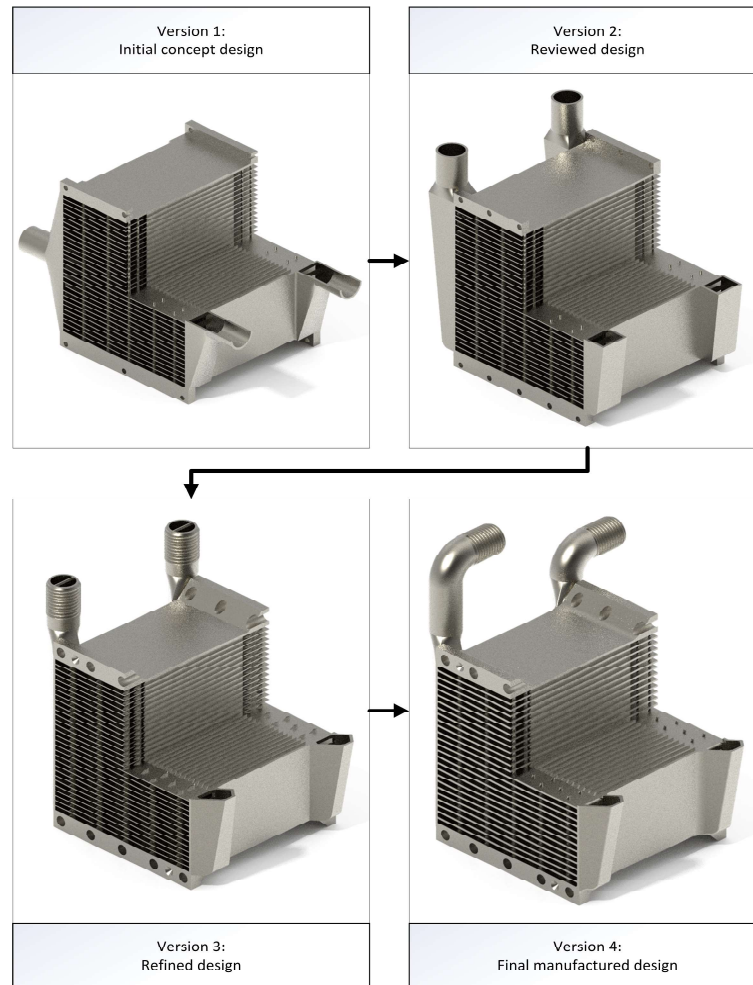


Figure 4.9: 2 mm design revisions

4.4 Development and manufacturing issues

4.4.1 Material buildup

After reviewing the designs and refining them for AM, difficulties were still experienced during the manufacturing process. The first two heat exchanger manufacturing attempts were unsuccessful. Material buildup on some sections of the heat exchanger core caused the AM process to fail. Figure 4.10 shows the material buildup above the red line. The causes for material buildup were identified as the passing of the coater arm, the combination of the thin walls and the heat exchanger core array; the coater arm flicks more material on already coated parts, causing the material buildup. The material buildup was reduced by changing the angle at which the coater arm recoats the heat exchangers. The optimal angle was found to be between 35° and 45° . Several manufacturing attempts were made before the first heat exchanger was successfully produced.

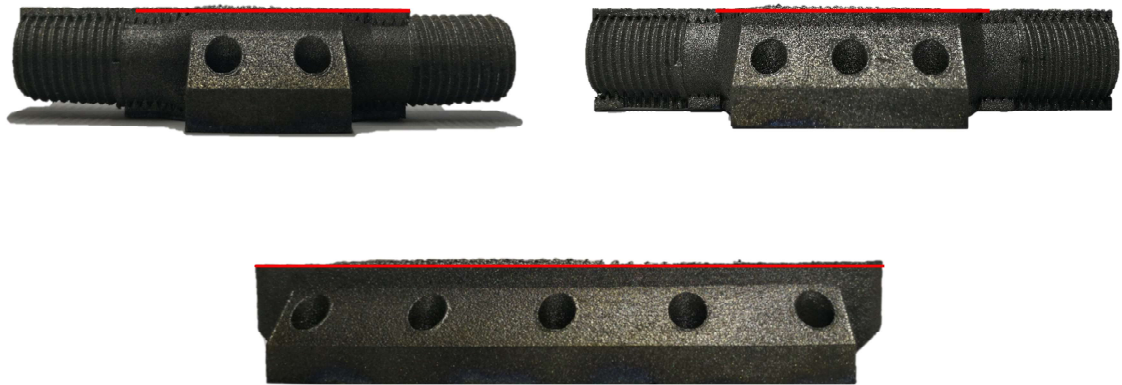


Figure 4.10: Material buildup, 0.5 mm design (top left), 1 mm design (top right), 2 mm design (bottom)

4.4.2 CT scans

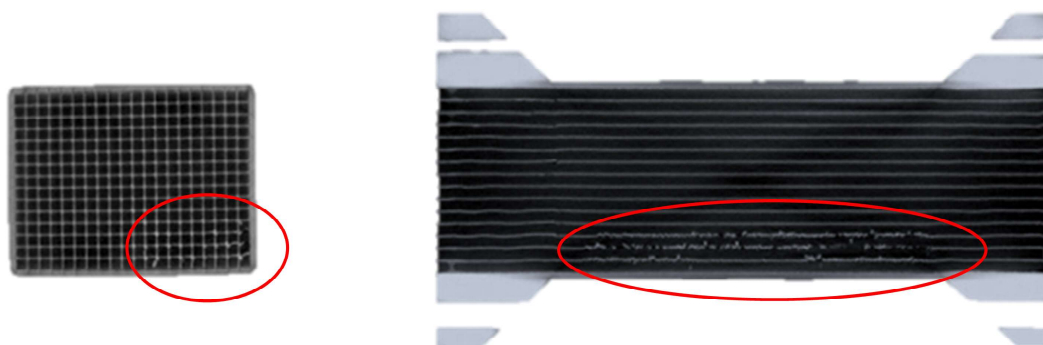
The first successfully produced heat exchangers (0.5 mm & 1 mm designs) were pressure tested and found to have porous walls causing fluid mixture between the hot and cold sides. The heat exchangers were sent for computerised tomography (CT) scans at the Centre for Analytics Facilities (CAF) at Stellenbosch University, South Africa for further internal examination. The CT scans revealed that some of the channel walls within the heat exchanger cores were broken and warped, and abnormalities were also present within the heat exchanger cores.

Heat buildup during the manufacturing process is the likely cause of the broken and warped channels. The thicker outer wall cools more slowly than the thinner channel walls, increasing stress and causing some walls to detach from the outer wall, shown in Figure 4.11. The abnormality seen in Figure 4.12 protrudes through the channel wall, resulting in leakage. The protrusion was presumably part of the broken channel wall that dislodged during the recoating process.

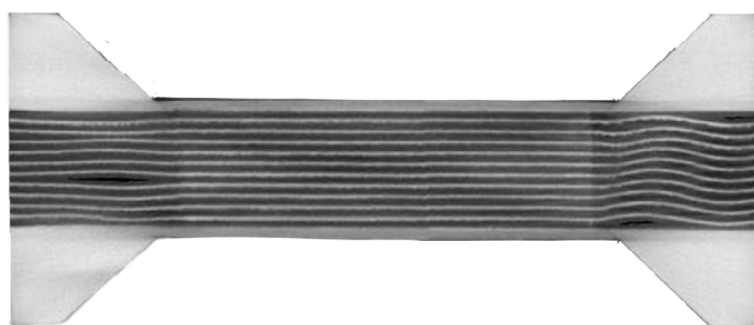
Visual inspection of these heat exchanger cores was severely hampered due to the warping of the manifold walls. The blocked channels in Figure 4.13 were likely caused by stuck residual powder.

The wall thickness of the channels was increased from 0.35 mm to 0.5 mm on the 0.5 mm design and to 0.8 mm on the 1 and 2 mm designs, ensuring the integrity of the channel wall. This, however, introduced another manufacturing complication in the 2 mm design. Added heat from the manufacturing process increased stress, causing a crack to form on the joint of the flange and the heat exchanger core (*cf.* Figure 4.14). Further investigation revealed that the crack propagated into the heat exchanger core running through several channel rows.

The radii of the fillets were increased on the outer and inner corners to mitigate the effects of stress buildup. The increased fillet radii solved the problem, and the 2 mm heat exchanger was manufactured successfully. Increasing the wall thickness does, however, have a negative influence on the heat transfer. Time and budget constraints did not allow for further refinement of the manufacturing parameters for 0.35 mm wall thicknesses and this fell outside the scope of this study. The author presented a detailed article discussing the CT scans at RAPDASA 2019 [77].



1mm Design: broken channels, front & side view



0.5mm Design: warped channels, side view

Figure 4.11: CT scan: Broken and warped channels

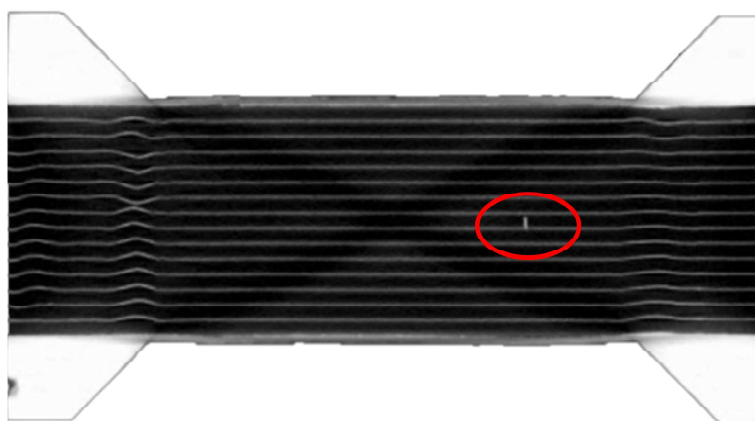


Figure 4.12: CT scan: Abnormality in 0.5 mm design, side view

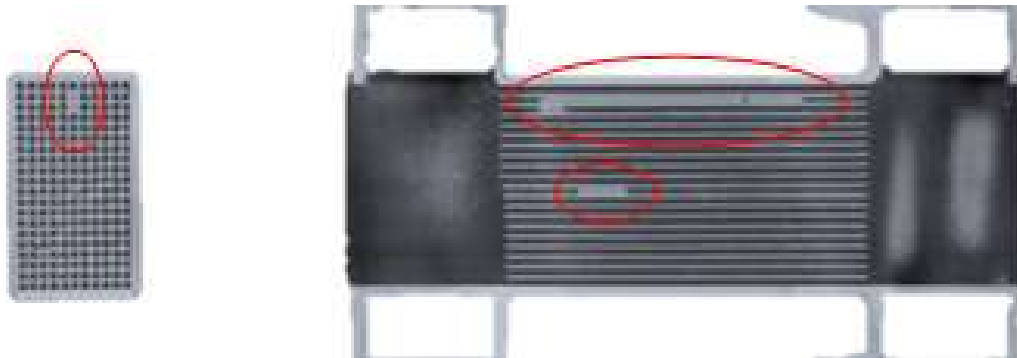


Figure 4.13: CT scan: Blocked channels in 0.5 mm design, front and side view

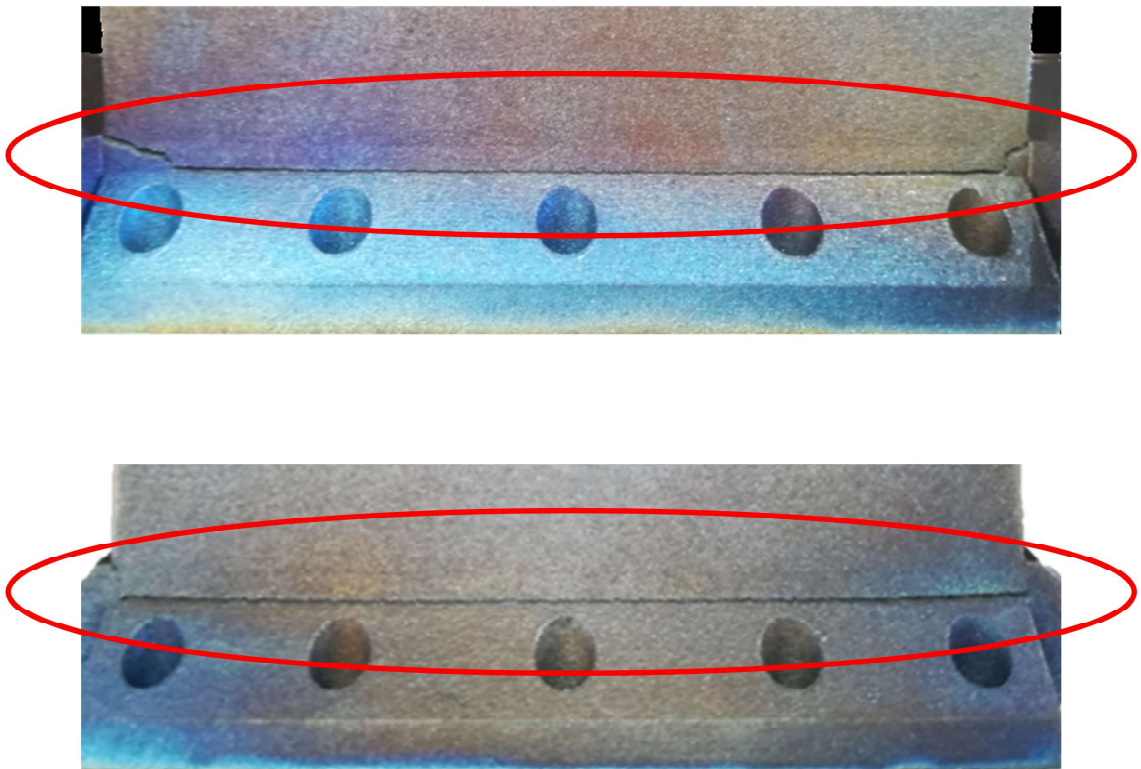


Figure 4.14: Crack along the base of the core, 2 mm design, top and bottom view



The 1/4" BSP threaded nozzle sections had a rough surface finish and it is recommended to clean the thread with either a thread file or thread die before use. One of each heat exchanger design was produced between the four manufacturing attempts (0.5 mm, 1 mm, and 2 mm). The 0.5 mm heat exchanger cracked during post-processing, and it was decided not to re-manufacture that heat exchanger due to time and budget constraints.

4.5 Summary and conclusion

The design for AM of the heat exchangers was discussed in this chapter. The final CAD designs produced had theoretical heat transfer capacities of 2.1 kW, 1.84 kW and 1.9 kW for the 0.5 mm, 1 mm and 2 mm heat exchanger designs, respectively.

Although the design considerations discussed in this chapter focused on the development of heat exchangers, these considerations still apply to any design or development for AM and should be carefully considered. The difficulties experienced while manufacturing the heat exchangers and their solutions were discussed.

Further research should be done on optimising manufacturing parameters for wall thicknesses less than 0.5 mm. Different configurations of the heat exchanger core and channel shapes should also be explored.

The next chapter entails the experimental setup and experimental investigation.

Chapter 5

Experimental Setup and procedure

5.1 Introduction

The experimental setup in the following chapter was constructed to validate the developed theoretical model in Chapter 3. This chapter provides a detailed description of the test procedure followed and the setting up of the experiment.

5.2 Experiment setup

5.2.1 System layout

A test bench was designed, assembled and used to characterise the compact heat exchanger and to validate the theoretical model. The test bench was of a simple design to reduce cost and allow the author to set up and operate it without any assistance. The system, depicted in Figure 5.1, consists of two fluid loops, a hot loop and a cold loop. Both loops utilised water as the working fluid.

The hot fluid was pumped from an external heated reservoir at a user-set pressure, temperature and flow rate through the compact heat exchanger back to the reservoir for reheating. The cold fluid pressure and temperature were as supplied via the water mains with flow rate set by the author. The cold loop outlet fluid was discharged and not returned to the system.

Flow control was achieved via a set of needle valves. Excess fluid is bled from the loop via the off-line valve, while the in-line valve controls the flow to the compact heat exchanger. Fine flow control can be achieved with this configuration. The pressures and temperatures are measured at the inlet and outlet of the compact heat exchanger for both loops respectively. The test bench built for the experimental investigation can be seen in Figure 5.2.

Hot loop

Fluid is pumped from the heated reservoir through the flow control section consisting of a ball valve, two needle valves and a flow meter. The in-line needle valve controls the flow to the compact heat exchanger and the off-line needle valve controls the flow through the hot feedback loop.

The ball valve acts as a shut-off valve for the whole loop when maintenance is required and the flow meter indicates the flow to the compact heat exchanger. This setup allows for accurate fine-tuning of the flow rate through the hot loop.

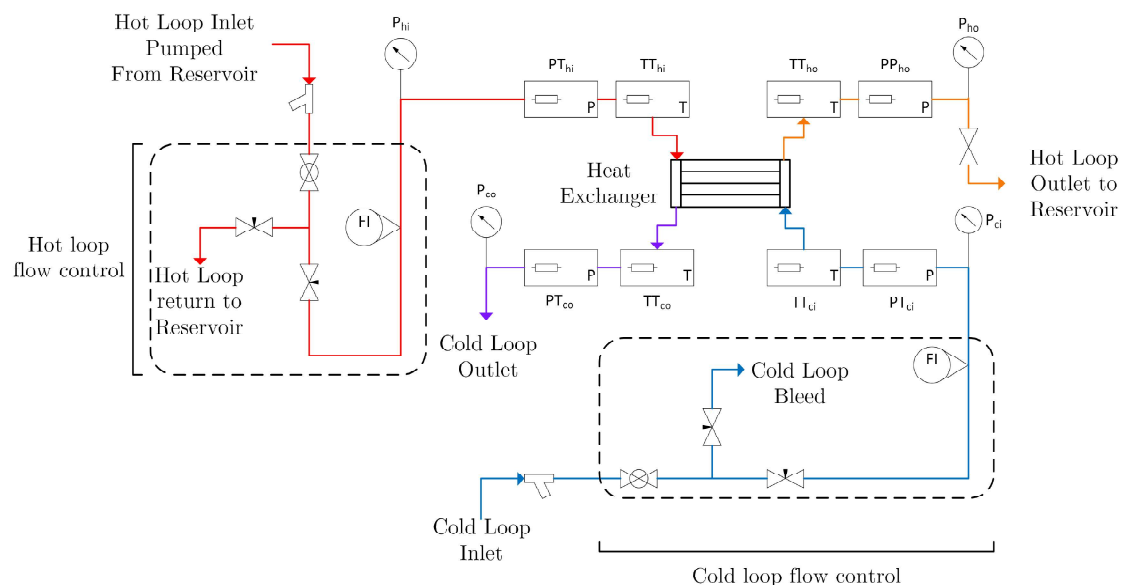
The hot loop heat exchanger section is the last section in the hot loop and consists of the compact heat exchanger (hot side) and two sets of recording instruments. These sets of instruments, which included temperature sensors, pressure indicators and transducers, were placed at the inlet and outlet of the compact heat exchanger to record the necessary

data at these points. Thereafter, the fluid returns to the reservoir via the return line to restart the process.

Cold loop

The cold loop consists of only two sections, the flow control section and the heat exchanger section. The flow control section for the cold loop operates in the same manner as the hot loop. The in-line needle valve controls the flow to the cold loop heat exchanger section and the off-line needle valve controls the flow to the cold bleed. The ball valve acts as the shut-off valve for the loop and the flow meter indicates the flow entering the cold loop heat exchanger section.

The cold loop heat exchanger section consists of the compact heat exchanger (cold side) and two sets of recording instruments. The instrumentation sets were placed at the inlet and outlet of the compact heat exchanger. Each set consists of a temperature sensor, pressure indicator and transducer.



Nomenclature

Symbols		Subscripts	
P	Pressure Gauge	h	Hot
PT	Pressure Transducer	c	Cold
TT	Temperature Transmitter	i	Inlet
FI	Rotameter	o	Outlet

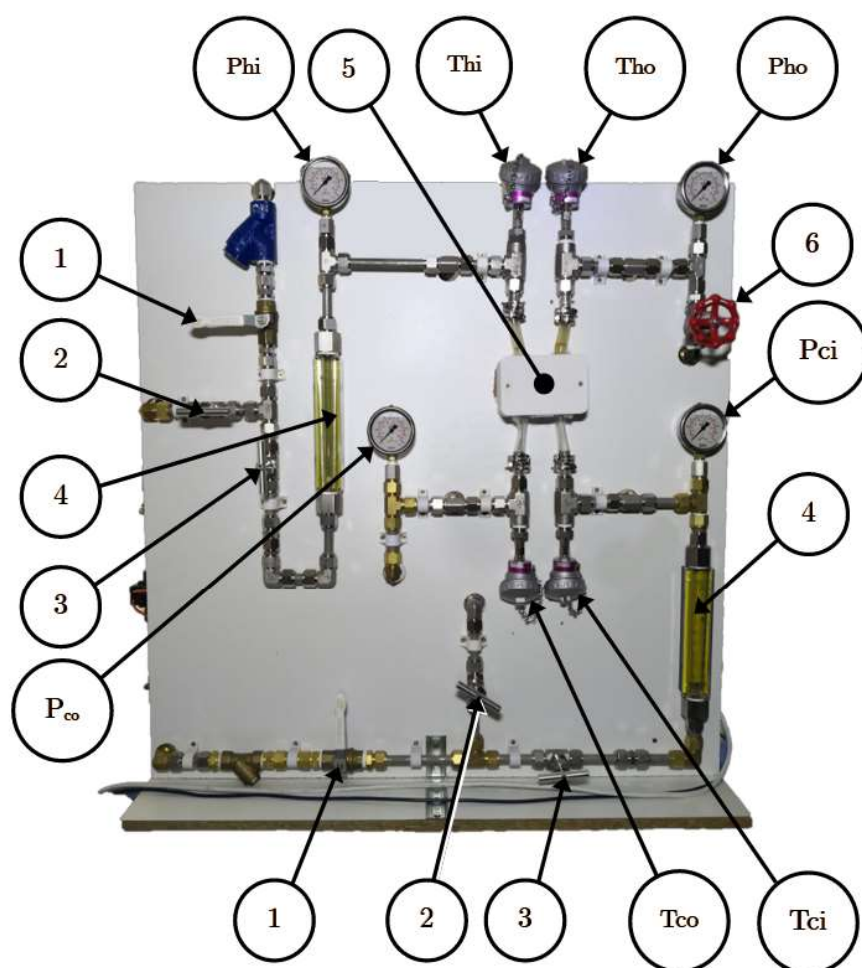
Figure 5.1: Process flow diagram of compact heat exchanger test bench

5.2.2 Instrumentation and data acquisition

Instrumentation

In order to characterise the compact heat exchanger, it is necessary to know the inlet and outlet temperatures, inlet and outlet pressures and mass flow rate through both loops. The temperature and pressure were measured at the inlet and outlet of the compact heat exchanger for both loops (*c.f* Figure 5.2). The mass flow rate of both loops was measured before the inlet of the compact heat exchanger.

The temperatures for the hot and cold loops ranged from 57 °C to 59 °C and 18 °C to 21 °C respectively, and were measured by the PT100 RTD temperature sensors. Pressures ranged from 0 to 3 bar throughout the loops and were measured with Gems 3500 Series Compact Low-Pressure sensors. The hot fluid was pumped through the loop with a Xylem Flojet - Duplex Diaphragm Pump, and both loops' flow rates were measured with Parker variable area flowmeters. Table 5.1 lists all instruments and specifications used for the experiment.



DESCRIPTION

- | | |
|------------------------|-------------------------|
| 1 Ball valve | 2 Off-line needle valve |
| 3 In-line needle valve | 4 Flow meter |
| 5 Heat Exchanger | 6 Gate Valve |

NOMENCLATURE: INSTRUMENTATION

- | Roman Letters | Subscripts |
|---------------------|------------|
| P Pressure gauge | i inlet |
| T Temperature probe | o outlet |

Figure 5.2: Test bench layout: Front



Table 5.1: List of instrumentation and data acquisition system specifications

INSTRUMENT	SPECIFICATION
Temperature Sensors	
Model Name	Jumo 3-wire PT100 Sensor
Range	-50 – 450 °C
Accuracy Class	IEC 60751 Class B
Transmitter	TranTech TT26T 4 – 20 mA
Transmitter Range	0 – 100 °C
Pressure Sensors	
Model Name	Gems 3500 Series Compact Low Pressure Sensor
Range	0 – 40 Bar abs.
Accuracy Class	0.25% of full scale
Transmitter	4 – 20 mA
Pump	
Model Name	Xylem Flojet Duplex Diaphragm Pump
Range	0 – 8.1 l/min max, 7.1 bar max
Data Acquisition System	
Model Name	Picolog 1216
Analog Inputs	16 Channels
Resolution	12 bits
Sampling Rate	1 kS/s per channel
Voltage Range	0 – 2.5 V
Accuracy	0.5% @ 12 bits
Flowmeters	
Model Name	Parker
Range Hot Loop	0.2 – 2 l/min
Range Cold Loop	1.5 – 10 l/min
Accuracy	±2%
Repeatability	±1%

Data acquisition

All data were captured with a PicoLog 1216 series data logger connected to a computer running PicoLog software via a USB 2.0 cable designed to reduce voltage drop and noise. The logger has 16 channels with 12-bit resolution and is used for general-purpose voltage, sensor and transducer data logging. An external terminal board is supplied with the data logger for custom front-end circuitry, which is connected to all electronic instruments necessary to test the compact heat exchangers. Figure 5.3 depicts the wiring diagram for the test bench. Figure 5.6 shows the test bench wiring and instrument placement.

A shunt resistor circuit is required to convert the 4 – 20 mA signals from the instruments to a nominal 0 – 2.5 V signal required by the PicoLog data logger. All instrumentation used was powered by a 4 – 20 mA loop and supplied with 24 V from a DC power supply. The PicoLog external terminal board provides space where such a circuit can be implemented on the required channels 5.5.

120 Ω resistors were soldered to the space provided on the terminal board for the channels used. Channel 15 was used initially. The channel later malfunctioned, and channel 14 was used instead. The required resistor value was calculated with $R_2 = \frac{2.5V}{I_{max}}$.

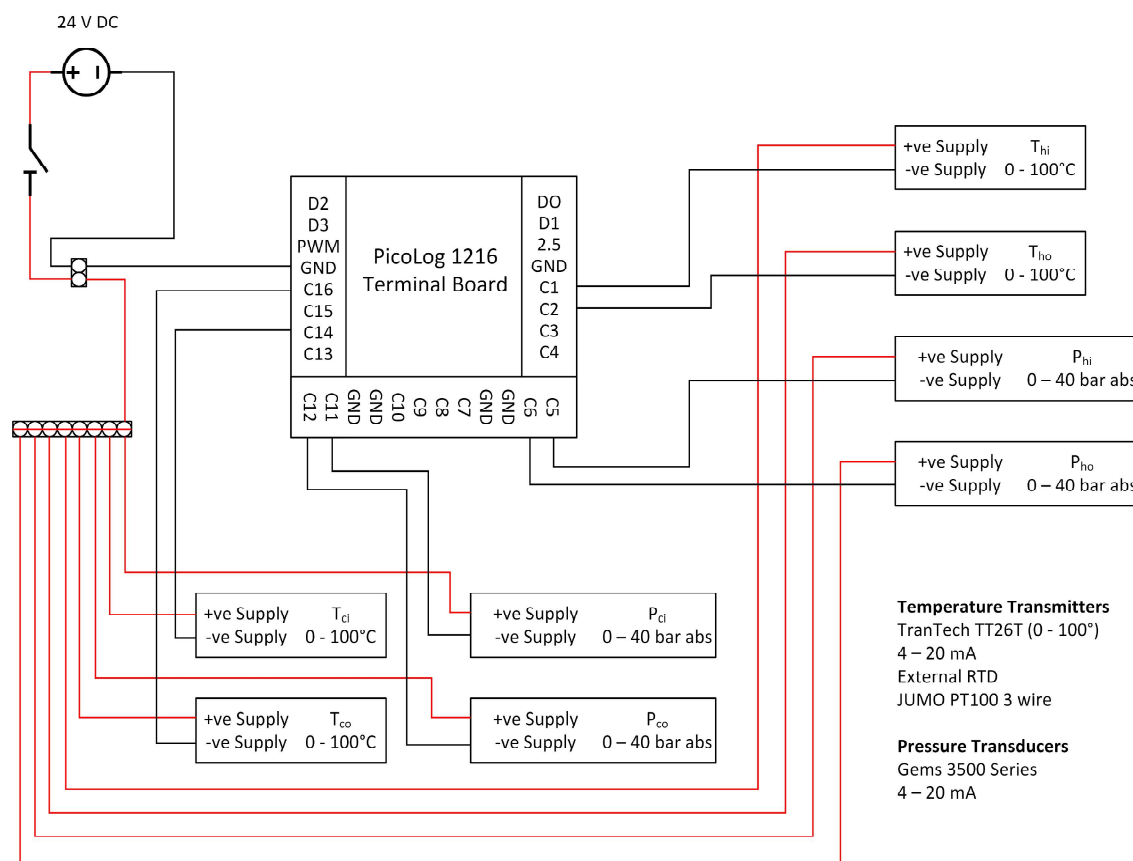


Figure 5.3: PicoLog and instrumentation wiring diagram

Electrical noise is a collection of spontaneous fluctuations in currents and voltages within circuitry, which is generally undesirable [78, 79]. Thus, in an effort to eliminate the effect of electrical noise, a twisted 3-core braid shielded wire was used to connect the instrumentation to the PicoLog data logger (*c.f* Figure 5.4).



Figure 5.4: Instrumentation wire

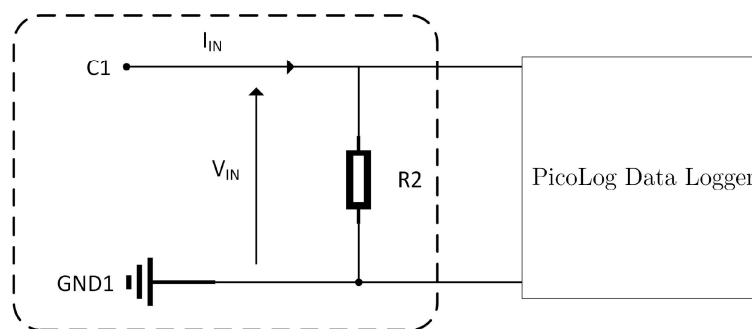
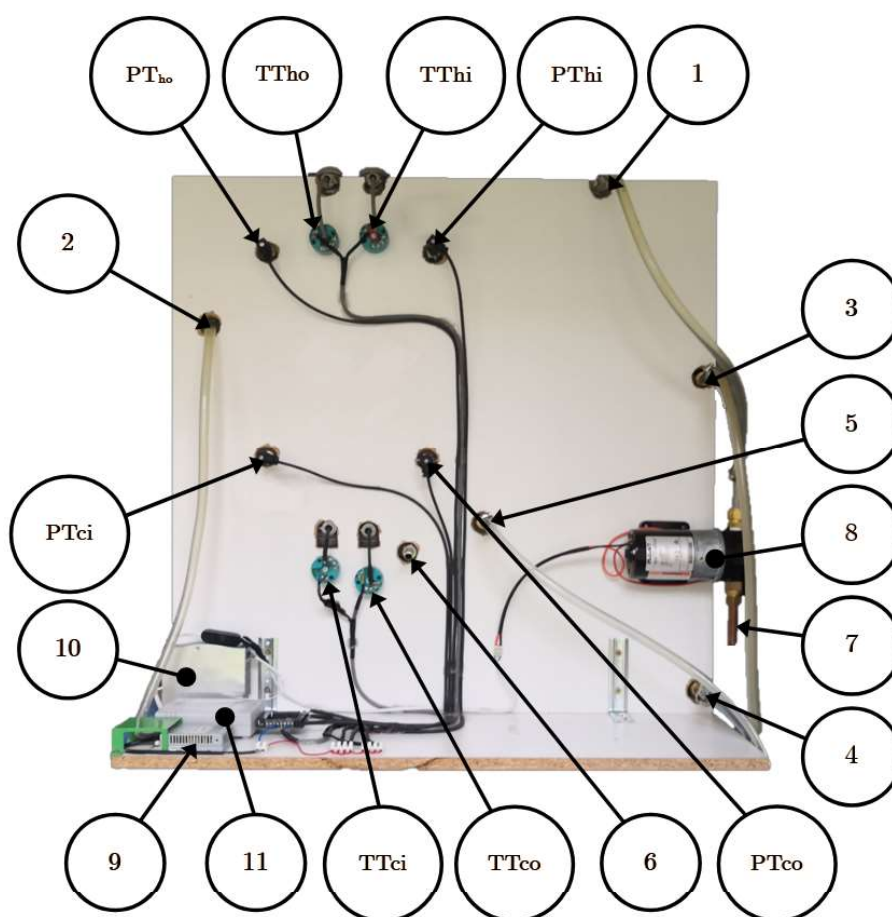


Figure 5.5: Shunt resistor circuit example for channel 1



DESCRIPTION

1	Hot inlet	2	Hot outlet
3	Hot feedback outlet	4	Cold inlet
5	Cold outlet	6	Cold Feedback outlet
7	Pump inlet	8	Electric Pump
9	Instrument power supply	10	Pump power supply
11	PicoLog		

NOMENCLATURE: INSTRUMENTATION

Roman Letters	Subscripts
PT	Pressure Transmitter
TT	Temperature Transmitter
i	inlet
o	outlet

Figure 5.6: Test bench layout: Back



Measurement uncertainties

The Type B uncertainty analysis was used to determine the measurement uncertainties as defined by the ISO [80]. Type B analysis was chosen due to the scarcity of data produced by the experimental procedure. This method is widely used by many international institutions, including the National Physical Laboratory in the UK and the National Institute of Standards and Technology (NIST) [81, 82]. A Type A analysis uses statistical methods and was therefore not used. The influence of instrument accuracy and error, data acquisition system resolution and accuracy and the shunt resistor circuit effect within the 4 – 20 mA instrument current loops were included in the measurement uncertainties and summarised in Table 5.2. A standard uncertainty with a confidence level of 68% was determined and scaled to approximately 95% with a coverage factor of $k = 2$. Appendix A discusses the measurement uncertainties in more detail.

The heat exchangers and piping leading to and from the temperature probes were well insulated to reduce heat loss. Thus, the heat loss to the environment can be omitted during calculations, reducing the uncertainties coupled to the heat loss estimation [3]. The heat exchangers were placed in custom-made boxes with insulating material to ensure minimal heat loss (*c.f* Figure 5.7).

Table 5.2: Measurement uncertainties

MEASUREMENT	RANGE	UNCERTAINTY ¹ (95% confidence)
Hot inlet temperature	0 – 100 °C	± 0.52 °C ($\pm 0.52\%$ of FS)
Hot outlet temperature	0 – 100 °C	± 0.52 °C ($\pm 0.52\%$ of FS)
Cold inlet temperature	0 – 100 °C	± 0.52 °C ($\pm 0.52\%$ of FS)
Cold outlet temperature	0 – 100 °C	± 0.52 °C ($\pm 0.52\%$ of FS)
Hot inlet pressure	0 – 4000 kPa a	± 53.6 kPa a ($\pm 1.34\%$ of FS)
Hot outlet pressure	0 – 4000 kPa a	± 53.6 kPa a ($\pm 1.34\%$ of FS)
Cold inlet pressure	0 – 4000 kPa a	± 53.6 kPa a ($\pm 1.34\%$ of FS)
Cold outlet pressure	0 – 4000 kPa a	± 53.6 kPa a ($\pm 1.34\%$ of FS)

¹Expanded uncertainty with a coverage factor of $k = 2$ which gives a 95% confidence level. Measurement uncertainty includes influences from instrumentation and data acquisition system.

5.3 Experimental procedure

5.3.1 Test preparation

To ensure reliable test results, a test sheet was used to ensure consistency throughout the tests. The data from each test run was entered on a separate sheet; time, date, ambient temperatures and pressures were recorded before each initial test run. Appendix F shows a sample of the test sheet.

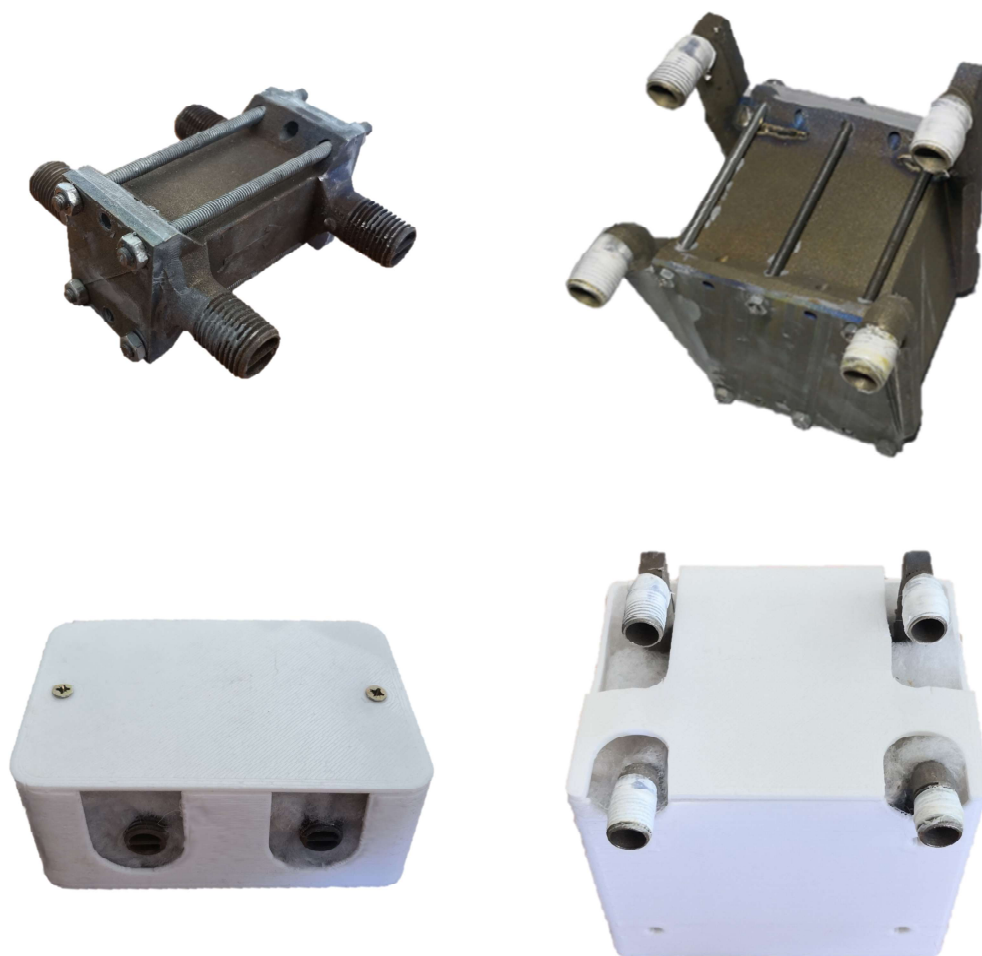


Figure 5.7: Heat exchangers in insulated boxes

The external heated reservoir was inspected for leaks and filled if necessary; all flexible pipes were inspected for any signs of deterioration or kinks that might lead to increased pressure drop. If necessary, the pipes were replaced with new ones. All external electrical connections and functions were checked.

A multi-monitor set-up was used to simultaneously fill the data sheet while monitoring the PicoLog software during tests. The PicoLog data logger was connected to the computer, after which the PicoLog software was started. This is displayed on monitor one closest to the test bench. Monitor two was used for completing the test sheet during testing, and monitor three displayed the local time to keep track of test times(*c.f* Figure 5.8).

5.3.2 Test procedure

Once all preliminary checks were completed, the external heated reservoir was switched on, and the ball valves and off-line needle valves were fully opened while keeping the in-line needle valves shut. Once the external heated reservoir was close to operating temperature (53–55 °C), the cold inlet was opened and allowed to run until the external heated reservoir reached operating temperature (56.5–58.5 °C).

The hot loop pump was then started, and the in-line needle valves for both loops were opened fully to bleed any air within the system. After all air was removed from the system,

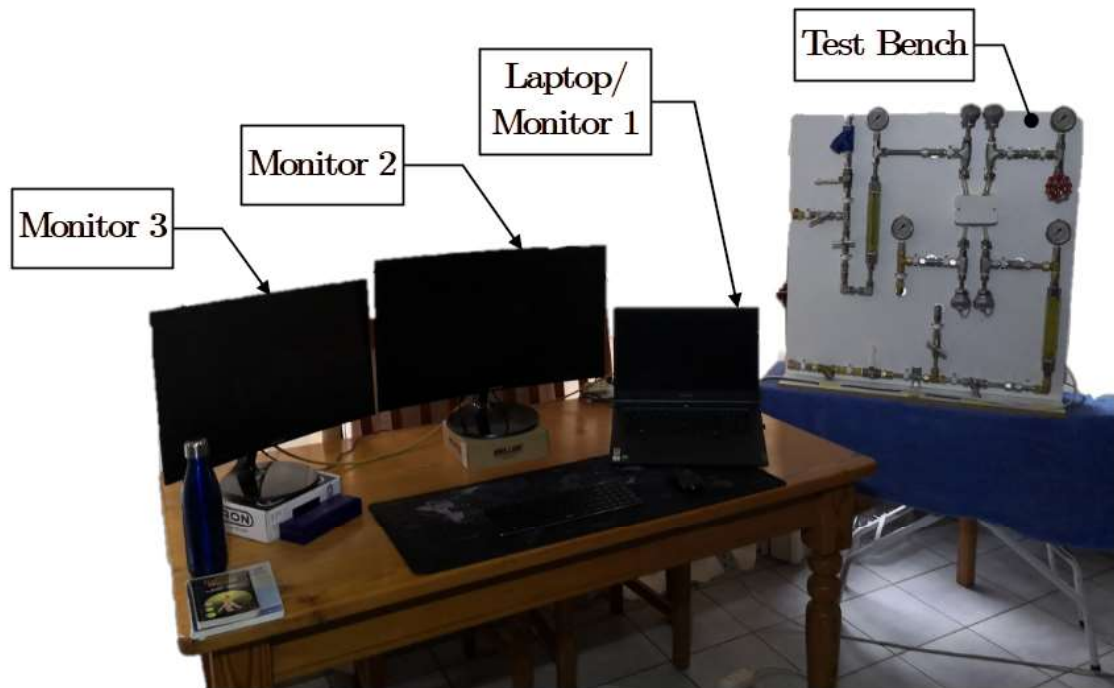


Figure 5.8: Experimental test set-up

the flow rate was adjusted to the required setting using a combination of the in-line and off-line needle valves.

The test was run until stable operating conditions were reached. The external heated reservoir is manually controlled to keep the temperature from either increasing too much, switched off at $\pm 57.8^\circ\text{C}$, or dropping too low, switched on at $\pm 57.6^\circ\text{C}$. Supply pressures vary during test runs; thus, constant flow rate monitoring is necessary and was adjusted for stability.

The test runs were based on the heat exchangers' capacity ratio (C^*), discussed in Section 5.3.3 below. For each chosen capacity ratio, three flow rates were selected for the minimum capacity rate fluid ($C_{min} = \text{hot fluid}$), resulting in 12 tests per heat exchanger.

5.3.3 Test matrix

A test matrix was set up to ensure proper coverage of the intended test envelope and to keep track of tests. The flow rate for the minimum capacity rate fluid (hot fluid) was varied per test between 50%, 75% and 100% of full scale on the flow meter; the flow rate for the maximum capacity rate fluid (cold fluid) was then calculated according to a selected value for the capacity ratio, $C^* = 1, 0.75, 0.5, 0.25$. A total of 12 flow rate settings were run using the test matrix, each consisting of at least five tests per flow rate setting. After steady state, the test ran for at least six minutes, recording one sample every three seconds. A sample of the PicoLog recording is attached in Appendix G. The test matrix used is summarised in Table 5.3.



Table 5.3: Test matrix

Capacity Ratio	Flow Setting	Flow Rate (l/min)		No. of Tests	
		Hot Side*	Cold Side**	1 mm	2 mm
$C^* = 1$	1	1	1	6	6
	2	1.5	1.5	5	5
	3	2	2	6	5
$C^* = 0.75$	1	1	1.3	5	5
	2	1.5	2	5	5
	3	2	2.6	5	5
$C^* = 0.5$	1	1	2	5	5
	2	1.5	3	5	6
	3	2	3.9	5	5
$C^* = 0.25$	1	1	3.9	6	5
	2	1.5	5.9	5	5
	3	2	7.9	5	5

* User Set (C_{min})** Calculated (C_{max})

5.4 Data reduction

Data reduction was required to convert the experimental data into a proper format. A Microsoft Excel™ spreadsheet was set up for each test and used to interpret the experimental data. Appendix B shows an example of the Excel test sheet used.

5.4.1 Stable operating conditions

Background

Reliable heat transfer analysis within the heat exchanger requires steady state operation. Steady state is defined as a state or condition of a system or process that does not change in time or only changes negligibly over a specific time [83].

Method

Temperatures and pressure were measured only after the specified operating conditions were set and allowed to reach steady state. With this testing technique, the transient terms can be omitted in the energy equations for the fluid and material, and the thermal capacity of the wall can be ignored [3]. An average of the captured data was used to smooth the signal fluctuations.



The energy balance error is a good indication of steady state. For a valid test run, the error should fall within 5% [3] and this was evaluated for each flow rate settings test. The energy balance error can be calculated with the following:

$$\varepsilon = \left| \frac{q_h - q_c}{Q} \right| \cdot 100\% \quad (5.1)$$

Where Q is the heat duty of the heat exchanger and q the heat transfer rate of the respective fluid. The heat duty can be determined by

$$Q = \frac{(q_h + q_c)}{2} \quad (5.2)$$

The steady state data of each test were then checked to conform to the limiting criteria in Table 5.4 to determine if the test run was valid. The Excel test sheet was set up to automatically check the first minute of each test to comply with the steady state criteria.

Table 5.4: Limit criteria for steady state

Measured Variable	Maximum deviation from average permissible	
	Range	% Deviation
Inlet Pressures	± 22 kPa	$\pm 2\%$
Outlet Pressures	± 22 kPa	$\pm 2\%$
Cold Inlet Temperature	± 0.35 °C	$\pm 2\%$
Hot Inlet Temperature	± 1 °C	$\pm 2\%$
Energy Balance Error	Calculated	$\pm 5\%$

5.4.2 Summary and conclusion

The experimental setup and test procedure to test and characterise the heat exchangers were discussed in this chapter. The purpose-built test bench layout was described, and the technical data for the instrumentation was listed (*c.f* Figure 5.2 and Table 5.1). A PicoLog 1216 was used for data acquisition capturing data at a sample rate of 3S/s; the instrument wiring and PicoLog setup can be seen in Figures 5.3 and 5.6.

The steady state limiting criteria were identified with a type B uncertainty analysis and implemented on the Excel test spreadsheet to check the validity of each test run. The tests consisted of five minutes of data capturing and at least one minute steady state before data capture. A test matrix was developed to ensure good data variety. The test matrix was based on the capacity ratio of the heat exchanger, adjusting flow for each test with the hot fluid being the minimum fluid.

The captured data was then analysed using an Excel spreadsheet, converting it into useful data. The results of the experimental tests, theoretical model and CFD are presented and discussed in the next chapter.

Chapter 6

Results and Discussion

6.1 Introduction

With the previous chapters having discussed the experimental procedure and data reduction, this chapter contains the experimental results.

Comparisons are made between the theoretical calculations (EES), CFD, and experimental results. A total of 12 tests were run per heat exchanger according to the test matrix discussed in Chapter 5. Graphs for each test were generated for data analysis and comparison purposes.

The total heat transfer rate, maximum theoretical heat transfer, effectiveness, overall heat transfer coefficient, and number of transfer units (NTU) were calculated using the Excel test sheet with the data captured from each test. The five runs per test flow setting were averaged to achieve a representative result for the chosen flow rate setting.

6.2 Experimental results

With the inlet and outlet temperatures and pressures captured for the chosen flow rate setting, the heat transfer rate for the heat exchanger can be obtained using Equation 3.9. Values for density (ρ), used to calculate the mass flow rate at the user-set temperature and pressure, and specific heat capacity (c_p) were automatically recalled by the *Water97V13* Excel add-in and checked periodically to ensure accuracy of results. Heat transfer rates for both sides were calculated and checked to fall below the limiting criteria ($<5\%$) with Equation 5.1.

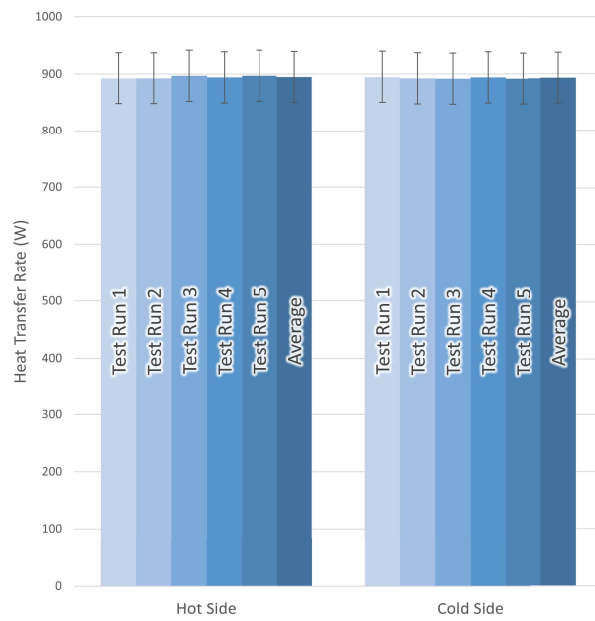
Using the mass flow rate and c_p values, the capacity rate is determined (hot fluid chosen to be C_{min}) and the capacity ratio checked to correlate with the test matrix. The maximum possible heat transfer is determined by the inlet temperatures and the value obtained for C_{min} using Equation 3.11. The effectiveness is then determined using Equation 3.10.

With known heat transfer areas obtained from the CAD models, the overall heat transfer coefficient was determined using Equation 3.1. The number of transfer units (NTU) can then be obtained by means of Equation 3.16.

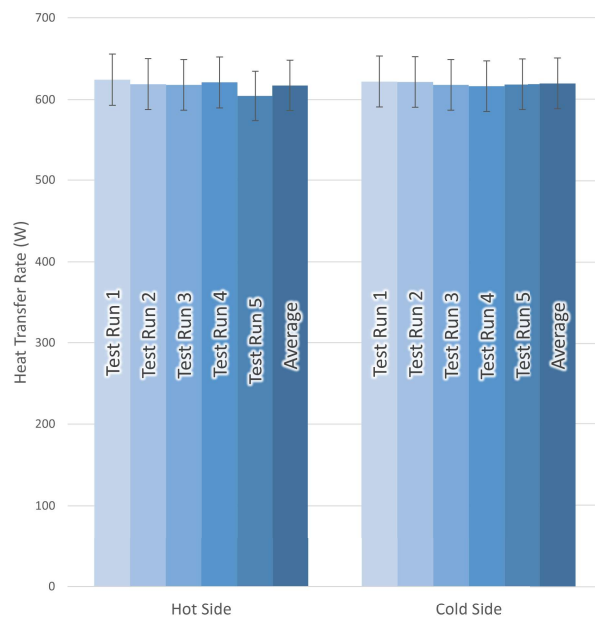
The maximum heat transfer rate achieved for the 1 mm design heat exchanger is 1.29 kW with an effectiveness of 24.93% at a capacity ratio of $C^* = 0.25$ on the third flow setting. The maximum effectiveness of 34.86% was achieved with the first flow setting of the $C^* = 0.25$ capacity ratio test.

A similar trend can be seen with the 2 mm design; the maximum heat transfer rate of 1.234 kW was obtained with a capacity ratio of $C^* = 0.25$ at the third flow setting with a effectiveness of 23.98%. The maximum effectiveness (30.77%) was achieved at a capacity ratio of $C^* = 0.25$ on the first flow rate setting.

A summary of the experimental results can be seen in Table 6.1, and the results for all test runs can be viewed in Appendix C. The heat transfer rate data for each respective test run was compared to analyse the repeatability. Figure 6.1 shows a typical best-case and worst-case scenario.



(a) Typical best scenario



(b) Typical worst scenario

Figure 6.1: Test run comparisons

Table 6.1: Summary of test results: Averages for each test group are shown. Each test group consisted of three flow settings and has a minimum of 5 tests per flow setting

Test Group		Temperature			Pressure			Temperature			Pressure			Flow Rate			Maximum Theoretical			Effectiveness			Number of		
HX Size	Capacity Ratio C*	Hot Inlet	Hot Outlet	T _{ho} (°C)	Hot Inlet	Hot Outlet	P _{hi} (kPa a)	P _{ho} (kPa a)	Cold Inlet	Cold Outlet	T _{co} (°C)	T _{ci} (°C)	Cold Inlet	Cold Outlet	P _{co} (kPa a)	PS _{co} (l/min)	PS _{ci} (l/min)	Cold Inlet	Cold Outlet	Q _h (W)	Q _c (W)	Q _{max} (W)	ε	NTU	Energy Balance
		T _{hi} (°C)	T _{ho} (°C)		P _{hi} (kPa a)	P _{ho} (kPa a)			P _{ci} (kPa a)	P _{co} (kPa a)								Q _c vs Q _c %							
1mm	1	57,78	47,96	30,25	130,07	126,03	20,45	126,38	125,79	1,0	1,0	1,0	675,24	671,95	2563,87	26,34%	0,357	0,52%							
		57,76	49,75	28,38	127,28	123,04	20,36	166,89	165,33	1,5	1,5	1,5	824,88	824,64	3850,31	21,42%	0,273	0,37%							
		57,79	50,87	27,16	128,91	122,55	20,18	147,86	145,24	2,0	2,0	2,0	951,50	956,85	5161,75	18,43%	0,226	0,56%							
	0,75	57,84	47,09	28,10	124,41	120,76	20,06	115,26	115,45	1,0	1,3	1,3	738,63	735,05	2597,23	28,44%	0,379	0,52%							
		57,93	49,25	26,66	127,14	123,24	20,16	169,56	166,85	1,5	2,0	2,0	894,80	893,11	3894,29	22,98%	0,288	0,29%							
		57,91	50,51	25,79	122,26	116,66	20,23	184,95	180,73	2,0	2,6	2,6	1016,27	1018,77	5177,91	19,63%	0,237	0,26%							
	0,5	57,78	46,26	25,94	127,02	123,69	20,17	141,18	139,82	1,0	2,0	2,0	792,67	791,31	2586,18	30,65%	0,399	0,21%							
		57,82	48,48	25,03	125,81	121,13	20,35	164,44	159,28	1,5	3,0	3,0	963,66	963,86	3863,09	24,95%	0,308	0,26%							
		57,78	49,70	24,29	124,02	118,10	20,26	191,37	182,07	2,0	3,9	3,9	1111,24	1107,11	5157,10	21,55%	0,257	0,37%							
0,25	57,84	44,78	23,63	127,18	123,83	20,38	155,58	148,37	1,0	3,9	3,9	898,43	894,27	2576,92	34,86%	0,450	0,52%								
	57,76	47,04	22,94	121,81	118,39	20,26	189,99	172,09	1,5	5,9	5,9	1106,04	1105,16	3868,09	28,59%	0,350	0,24%								
	57,80	48,39	22,40	122,16	117,60	20,05	214,91	178,55	2,0	7,9	7,9	1293,66	1290,58	5189,59	24,93%	0,296	0,42%								
2mm	1	57,88	50,05	28,23	129,98	125,96	20,42	157,12	156,97	1,0	1,0	1,0	537,80	535,82	2573,28	20,90%	0,264	0,43%							
		57,90	52,59	25,34	121,19	117,21	20,10	137,21	137,02	1,5	1,5	1,5	546,92	539,93	3894,07	14,05%	0,163	1,42%							
		57,76	52,21	27,12	127,86	123,47	21,59	187,83	186,44	2,0	2,0	2,0	763,11	759,86	4968,30	15,36%	0,181	0,63%							
	0,75	58,05	51,63	26,63	117,71	114,24	21,79	116,55	116,55	1,0	1,3	1,3	441,56	442,57	2490,85	17,73%	0,210	0,62%							
		57,83	50,88	25,72	120,56	116,24	20,49	181,30	180,08	1,5	2,0	2,0	715,83	718,29	3848,22	18,60%	0,222	0,54%							
		57,77	52,28	24,26	116,48	111,71	20,17	137,73	135,80	2,0	2,6	2,6	753,79	749,19	5165,84	14,59%	0,167	0,61%							
	0,5	57,92	48,94	24,73	124,51	121,92	20,22	150,75	149,74	1,0	2,0	2,0	617,49	619,46	2591,29	23,83%	0,291	0,77%							
		57,82	50,11	24,18	123,70	119,50	20,29	156,36	153,33	1,5	3,0	3,0	794,67	800,72	3867,96	20,54%	0,243	0,86%							
		57,78	51,09	23,37	122,88	118,60	20,02	171,60	167,09	2,0	3,9	3,9	918,72	920,42	5188,06	17,71%	0,204	0,60%							
0,25	57,77	46,22	23,13	128,43	124,81	20,24	156,85	153,34	1,0	3,9	3,9	794,18	794,71	2580,67	30,77%	0,384	0,62%								
	57,66	47,32	22,76	125,89	121,77	20,19	197,00	186,90	1,5	5,9	5,9	1066,93	1059,78	3865,28	27,60%	0,335	0,75%								
	57,50	48,52	22,27	125,18	120,99	20,03	196,40	175,82	2,0	7,9	7,9	1235,51	1227,80	5151,49	23,98%	0,283	0,63%								

6.3 Theoretical results

After developing and implementing the theoretical model in Engineering Equation Solver (EES), as previously discussed, the heat exchanger size was altered until the baseline design parameters were met. To better understand the effect the channel size, length, and wall thickness have on the performance of the heat exchanger, parametric tables were implemented within EES to simulate different design configurations.

Figure 6.2(a) shows the effect of channel size on the calculated heat transfer rate and pressure drop per unit volume for various flow rates, and Table 6.2 summarises the data of the graph. Increasing channel size positively affects pressure drop at the cost of less heat transfer. This approach can be used to find the optimal design for heat transfer vs pressure drop. Increasing the channel size from 0.5 mm to 1 mm decreases the heat transfer rate and pressure drop by -56% and -84%, respectively, for a flow rate of 4 l/min. Increasing the channel size from 0.5 mm to 2 mm decreases the heat transfer and pressure drop by -86% and -98%, respectively, for a flow rate of 4 l/min.

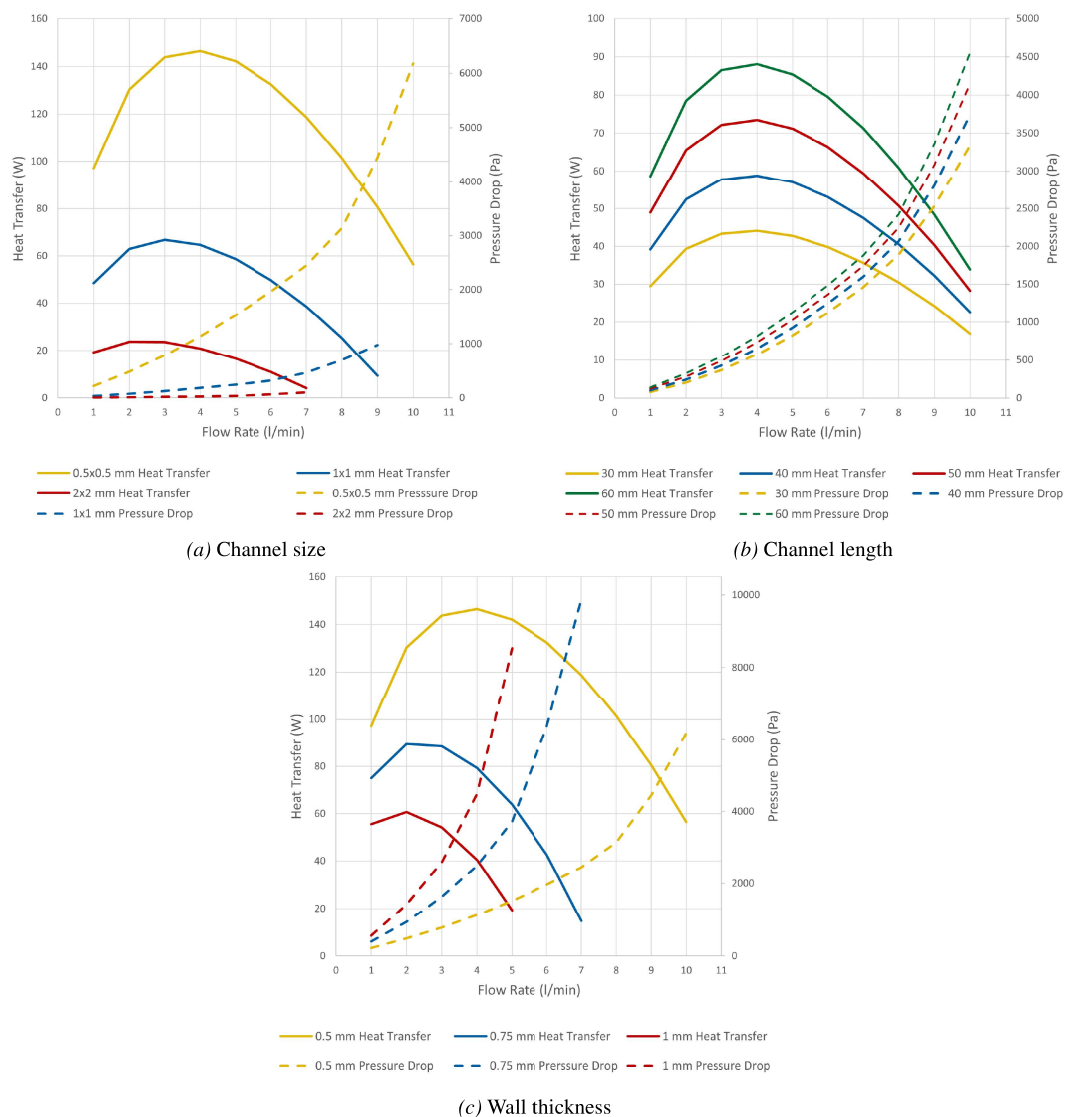


Figure 6.2: Influence of channel size, channel length, and wall thickness on calculated heat transfer and pressure drop

Table 6.2: Theoretical effect of channel size on calculated heat transfer rate, effectiveness, and pressure drop

Channel Size (mm)	Flow Rate (l/min)	Calculated Heat Transfer Rate (W)	Effectiveness (%)	Pressure Drop (kPa)
0.5	1	97.19	3.73	221.77
	4	146.5	1.4	1137
	7	118.7	0.65	2444
1	1	48.46	1.86	34.37
	4	65.66	0.62	182.4
	7	38.62	0.21	463.1
2	1	19.37	0.74	5.31
	4	20.83	0.2	27.57
	7	4.08	0.02	99.93

Furthermore, increasing the channel length also increases the heat transfer rate and pressure drop (*c.f* Figure 6.2(b)). Increasing the channel length by 33.33% (+10 mm on base-line model, 30 mm), the heat transfer rate and pressure drop increase by 33% and 12%, respectively, for a flow rate of 4 l/min, while increasing the channel length by 66.66% (+20 mm) increases the heat transfer rate and pressure drop by 66% and 24%. Moreover, increasing the channel length by 100% (+30 mm) increases the heat transfer rate and pressure drop by 100% and 36%, respectively.

The effect of the channel wall thickness on heat transfer and pressure drop can be seen in Figure 6.2(c). An increase of 50% in wall thickness leads to a -46% decrease in heat transfer rate and a 119% increase in pressure drop. Increasing the wall thickness +100% leads to a decrease of -72% in heat transfer rate and a 294% increase in pressure drop. As the wall thickness increases, fewer channels are present per unit volume; thus, the free flow to frontal area ratio decreases and velocity within the channels increases, negatively impacting pressure drop.

Finding the right channel size, length, and wall thickness combination would lead to optimal heat transfer rate and minimal pressure drop. Table 6.3 summarises all the data discussed.

Table 6.3: Data comparison summary*

Change	% Change	Calculated Heat Transfer Rate	Pressure Drop
Channel Size			
0.5 mm → 1 mm	+100%	-59%	-84%
0.5 mm → 2 mm	+200%	-88%	-98%
Channel Length			
30 mm → 40 mm	+33.33%	+33%	+12%
30 mm → 50 mm	+66.66%	+67%	+24%
30 mm → 60 mm	+100%	+100%	+36%
Wall Thickness			
0.5 mm → 0.75 mm	+50%	-55%	+143%
0.5 mm → 1 mm	+100%	-86%	+294%

*Flow Rate = 4 l/min

6.4 CFD results

The inlet temperatures and flow rates achieved with the test runs were used for the CFD simulation inputs, which ensures a sound basis for result comparisons. Mesh parameters were optimised for each heat exchanger design to ensure accurate results with minimum calculation time.

6.4.1 0.5 mm Heat exchanger design

Although no physical model for the 0.5 mm heat exchanger was tested experimentally, its performance can still be compared to the theoretical model. The maximum heat transfer rate was achieved with the third flow rate setting, full-scale flow on the hot side (2 l/min), and a capacity ratio of $C^* = 0.25$. The heat transfer rate for this setting was 1.098 kW with an effectiveness of 21.03%. The maximum effectiveness (32.24%) was achieved with a capacity ratio of $C^* = 0.25$ and the first flow rate setting, 50% of full-scale flow on the hot side (1 l/min).

Figure 6.3 shows the CFD results plots for the third flow rate setting with a capacity ratio of $C^* = 0.75$, Table 6.4 shows the relevant data for Figure 6.3.

The maximum velocity occurs within the channels while the minimum velocity occurs in the top left-hand corner of the manifold in Figure 6.3(b). The minimum and maximum pressure occurs at the manifold inlet and outlet, respectively, Figure 6.3(d).

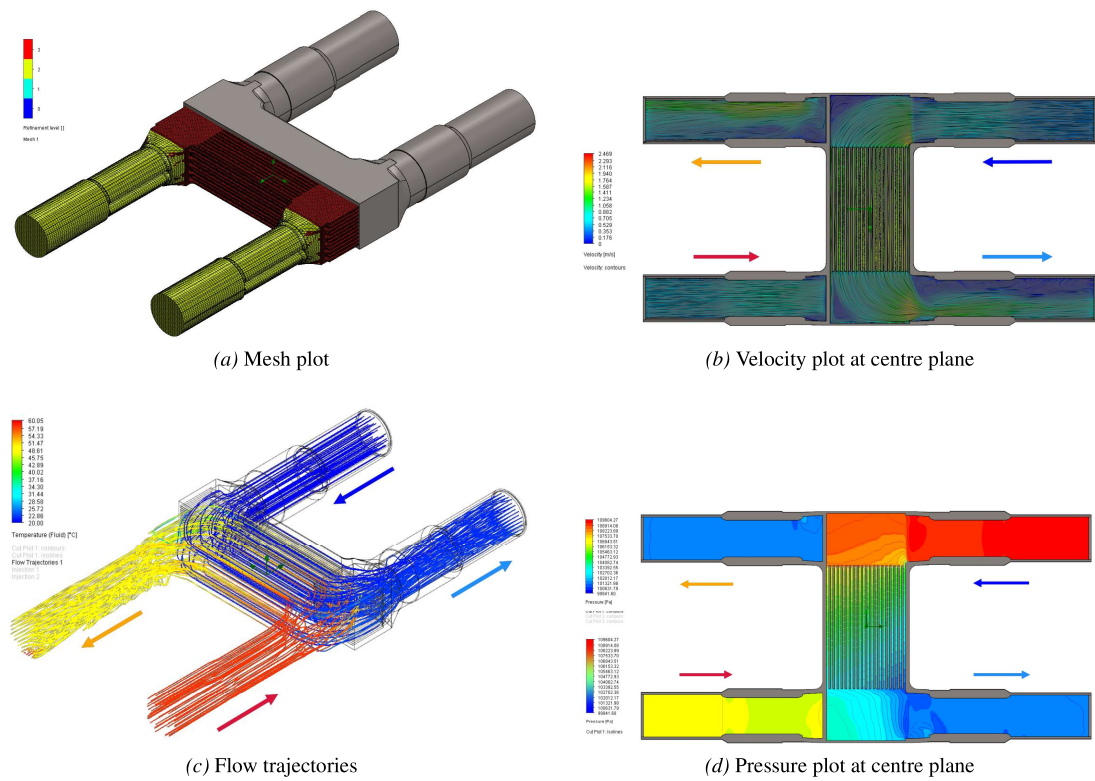


Figure 6.3: CFD plots: 0.5 mm design

Table 6.4: CFD Results 0.5 mm Design: $C^*=0.75$, Third Flow Setting

Input Variables			
Flow Rate	Hot Inlet	2 l/min	
	Cold Inlet	2.6 l/min	
Temperature	Hot Inlet	58 °C	
	Cold Inlet	20 °C	
Figure	Description	Value	Units
6.3(a)	No. of Fluid Cells	551 834	
	No. of Solid Cells	983 398	
6.3(b)	Velocity Min	3.293×10^{-4}	m/s
	Velocity Max	2.47	m/s
6.3(c)	Temperature Min	20	°C
	Temperature Max	58	°C
6.3(d)	Pressure Min	100.77	kPa
	Pressure Max	109.41	kPa

6.4.2 1 mm heat exchanger design

The input values from the 1 mm design experimental test were used as the input values for the CFD simulation. The maximum heat transfer rate of 1.165 kW with an effectiveness of 21.2% was achieved at the third flow rate setting, full-scale flow on the hot side (2 l/min), at a capacity ratio of $C^* = 0.25$. The maximum effectiveness (33.1%) occurred at the first flow rate setting, 50% of full-scale flow on the hot side (1 l/min), and a capacity ratio of $C^* = 0.25$ with a heat transfer rate of 0.91 kW.

Figure 6.4 shows the CFD results plots for a capacity ratio of $C^* = 0.75$ at the third flow setting. Table 6.5 shows the relevant data for Figure 6.4.

The maximum velocity occurs right before the manifold exit, where the flow converges, and the minimum velocity occurs in the bottom manifold at the fourth manifold support from the left. Flow disturbance caused by the support structures can be seen in Figure 6.4(b). The minimum pressure occurs at the heat exchanger exit. Maximum pressure occurs in the bottom manifold at the second-from-left manifold support, Figure 6.4(d).

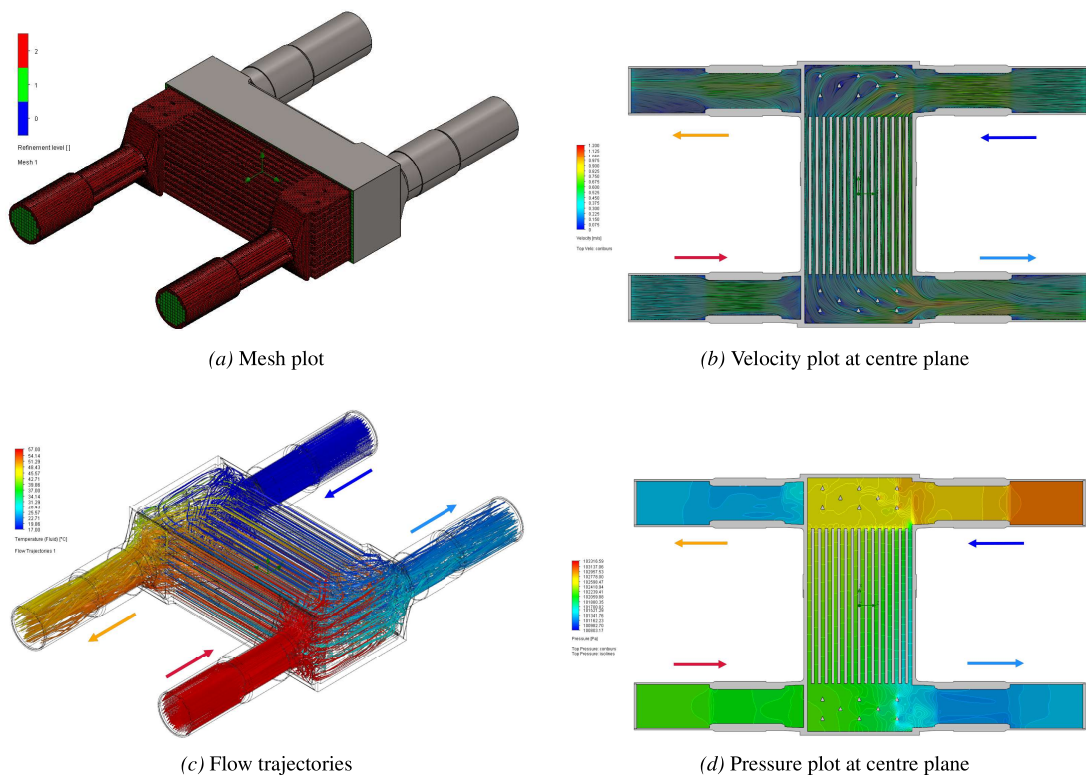


Figure 6.4: CFD plots: 1 mm design

Table 6.5: CFD Results 1 mm design: $C^* = 0.75$, Third flow setting

Input Variables			
Flow Rate	Hot Inlet	2 l/min	
	Cold Inlet	2.6 l/min	
Temperature	Hot Inlet	57.91 °C	
	Cold Inlet	20.23 °C	

Figure	Description	Value	Units
6.3(a)	No. of Fluid Cells	478 519	
	No. of Solid Cells	555 037	
6.3(b)	Velocity Min	0.006	m/s
	Velocity Max	1.11	m/s
6.3(c)	Temperature Min	20.23	°C
	Temperature Max	57.91	°C
6.3(d)	Pressure Min	101.17	kPa
	Pressure Max	103.01	kPa

6.4.3 2 mm heat exchanger design

The respective input values from the 2 mm design experimental test were used as the input values for the CFD simulation. The maximum heat transfer rate (1.73 kW) was achieved with the third flow rate setting, full-scale flow on the hot side (2 l/min), and a capacity rate of $C^* = 0.25$, an effectiveness of 33.49% was achieved with these settings. A maximum effectiveness of 46.1% was achieved with the first flow setting, 50% of full-scale flow on the hot side (1 l/min), and a capacity ratio of $C^* = 0.25$. The heat transfer rate with this setting is 1.19 kW.

The CFD result plots for the 2 mm design can be seen in Figure 6.5. The third flow rate setting was used for the plots at a capacity rate of $C^* = 0.25$. The relevant data is summarised in Table 6.6.

The minimum and maximum velocity occurs at the inlet and outlet of the manifold, respectively, with the minimum occurring right before entering the manifold and the maximum right after exiting the manifold shown in Figure 6.5(b). Maximum pressure occurs at the manifold inlet and minimum pressure within the exit section leading to the heat exchanger outlet in Figure 6.5(d).

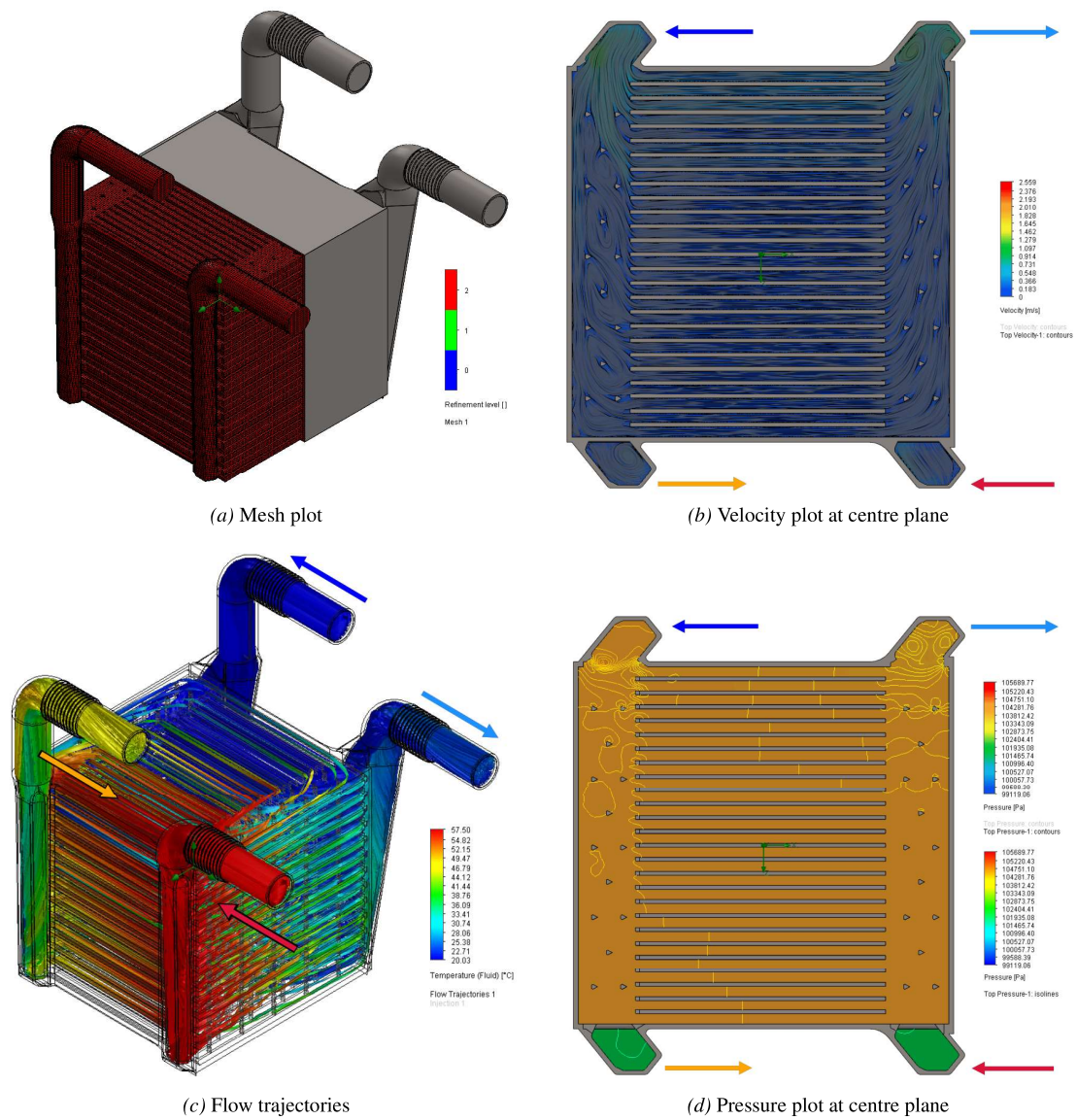


Figure 6.5: CFD plots: 2 mm design

Table 6.6: CFD results 2 mm design: $C^* = 0.75$, Third flow setting

Input Variables			
Flow Rate	Hot Inlet	2 l/min	
	Cold Inlet	2.6 l/min	
Temperature	Hot Inlet	57.77 °C	
	Cold Inlet	20.17 °C	

Figure	Description	Value	Units
6.3(a)	No. of Fluid Cells	1 184 650	
	No. of Solid Cells	1 208 849	
6.3(b)	Velocity Min	0.005	m/s
	Velocity Max	1.82	m/s
6.3(c)	Temperature Min	20.17	°C
	Temperature Max	57.77	°C
6.3(d)	Pressure Min	101.44	kPa
	Pressure Max	104.01	kPa

6.5 Results Discussion

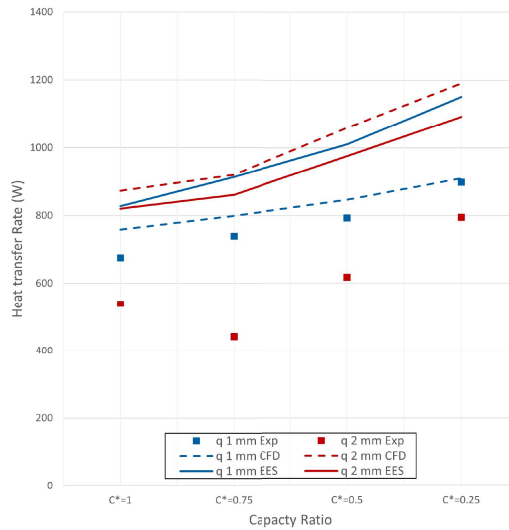
This section discusses and compares all data from the experimental test, theoretical, and CFD simulations.

6.5.1 Heat transfer

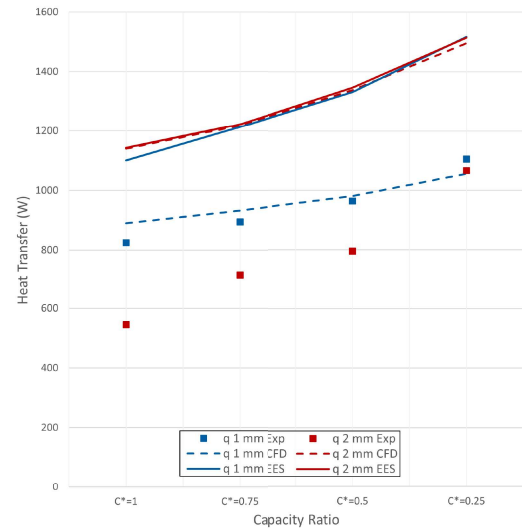
A comparison of the heat transfer rate between the numerical simulations and experimental data can be seen in Figure 6.6. The general trend of the graphs shows good agreement between the results, with the heat transfer rate increasing with respect to the increase in mass flow rate. The numerical models, however, overestimate the heat transfer rate compared to the experimental results.

The theoretical model overestimates the heat transfer rate by 35% on average. At the same time, the CFD overestimates the heat transfer by 2% on average for the 1 mm design. The 2 mm design shows an overestimation of 72% and 71% on average, respectively. A direct comparison of the experimental heat transfer rate vs the theoretical heat transfer rate can be seen in Figure 6.7.

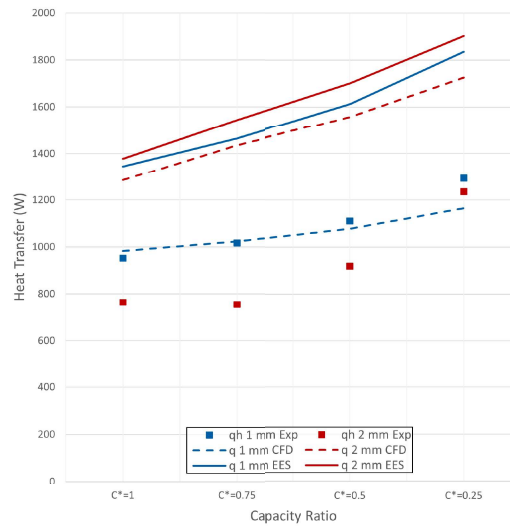
Numerous factors could lead to the high deviation of the results. This could be a combination of the uncertainty in the measured experimental data and the theoretical model, with the assumption of uniform flow within the channels and the CFD simulated in the best-case scenario, assuming uniform surface roughness. The CFD results from the 2 mm design showed the effect of the non-uniform flow across the heat exchanger.



(a) 50% of full-scale flow rate - Hot side



(b) 75% of full-scale flow rate - Hot side



(c) 100% of full-scale flow rate - Hot side

Figure 6.6: Comparison of heat transfer vs capacity ratio

The low flow rates used for testing (limited to the maximum flow rate on the rota meter) resulted in the underutilisation of the heat exchanger (*c.f* Figure 6.8). Figure 6.8(a) shows the velocity cut plot in the second row of channels from the top. The velocity vectors show clearly that only the first few rows experience flow with a higher velocity. Further down the heat exchanger, at the third row of channels from the bottom, the velocity vectors are almost invisible (*c.f* Figure* 6.8(b)). This indicates that only part of the heat exchanger core was utilised during testing. Higher flow rates are expected to result in a more equal velocity distribution, resulting in more accurate results.

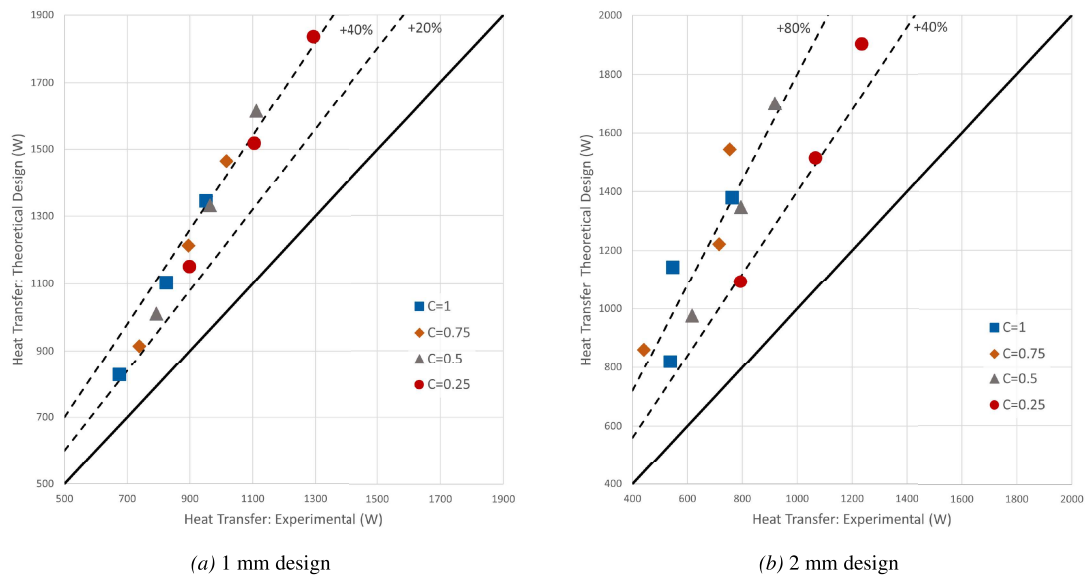


Figure 6.7: Comparison of heat transfer: Theoretical vs experimental

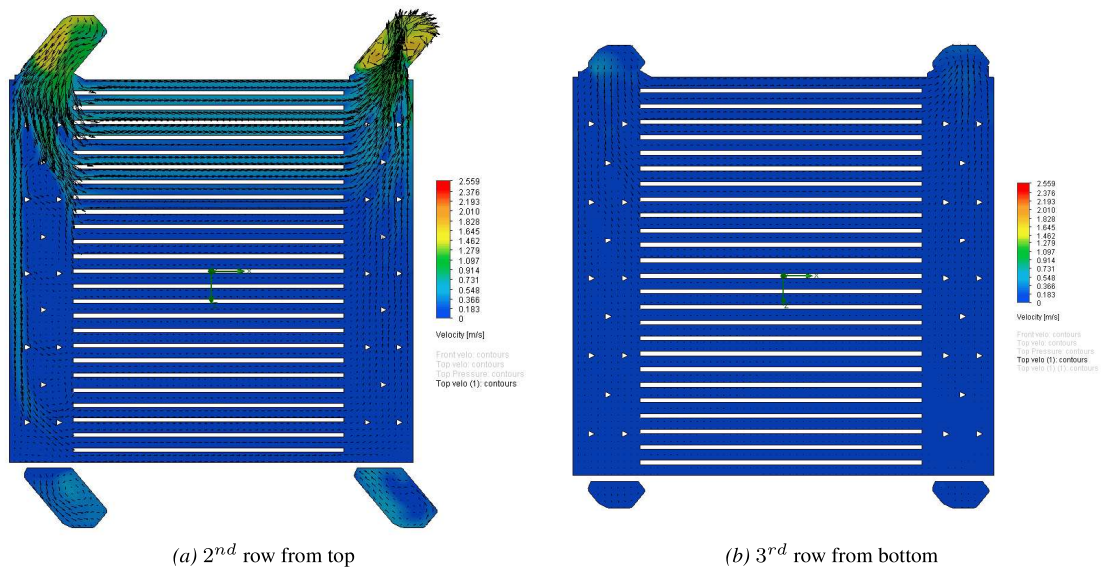


Figure 6.8: CFD velocity plots: 2 mm design, unequal flow distribution

6.5.2 Effectiveness

With effectiveness based on the heat transfer, the effectiveness graphs (*c.f.* Figure 6.9) show similar trends to the heat transfer rate graphs, increasing as the flow rate increases. Hence, the numerical models show the same overestimation of effectiveness.

The experimental results are plotted on an effectiveness, NTU graph for a counterflow heat exchanger in Figure 6.10. The graph is obtained using the equation described in Chapter 3 for effectiveness, Equation 3.15. The experimental results overestimate the effectiveness for the corresponding theoretical NTU. For the 1 mm design, the difference is, on average 11.51% higher and for the 2 mm design, on average, 16.53% higher.

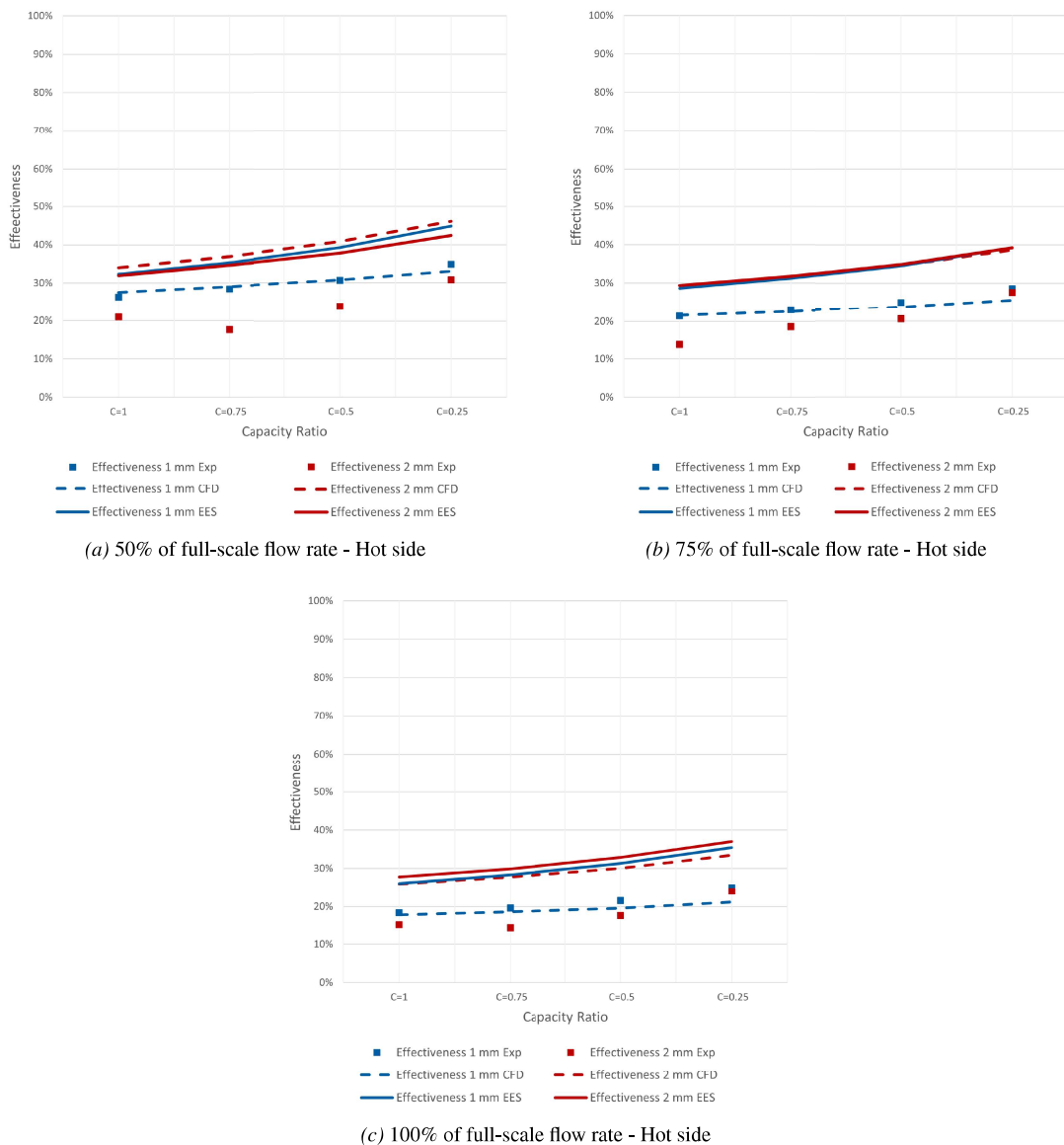


Figure 6.9: Comparison of effectiveness vs capacity ratio

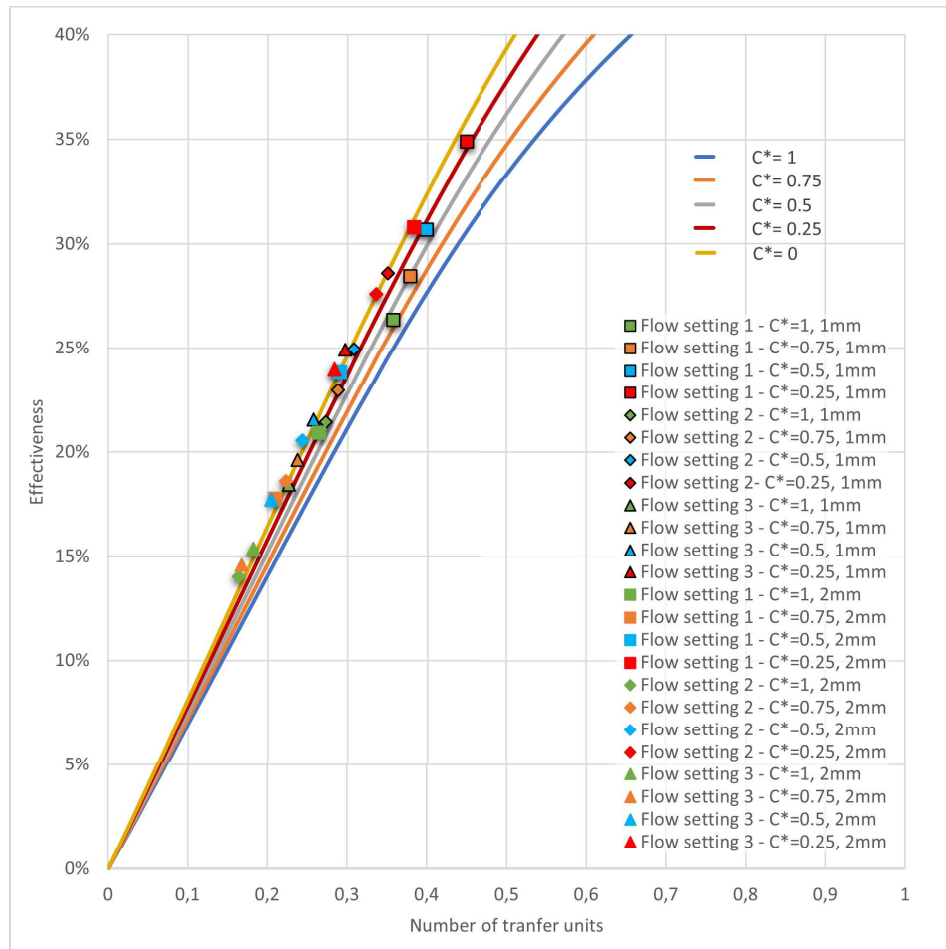


Figure 6.10: Effectiveness vs Number of Transfer Units (NTU)

6.5.3 Pressure drop

Figure 6.11 shows the pressure drop across the heat exchangers. A similar trend can be seen between the experimental and numerical results; the pressure drop increases as the flow rate increases. The numerical models do, however, underpredict the pressure drop except for the 2 mm design cold side flow (*c.f* Figure 6.11(c)). For the 1 mm design, the theoretical model underpredicts the pressure drop by 45% on average for the cold side and 413% on average for the hot side. On the 2 mm design, the experimental and theoretical model shows good agreement with an 15% difference on average for the cold side. There is, however, an underprediction in pressure drop of 573% on average for the hot side of the heat exchanger.

The deviation in pressure drop from the experimental and theoretical model can be mainly attributed to the pressure drop caused by the manifold supports and a lesser degree to the uncertainty in calculations and data captured. The manifold supports split the incoming flow, acting as an obstruction in the flow path and this was not accounted for in the theoretical model. The effect can be seen in the manifolds of Figure 6.4. The effect is more prominent in the 1 mm design, as flow velocities are higher within the manifolds. The 2 mm design resembles the theoretical model more closely due to the lower velocities within the manifold of the 2 mm design, decreasing the effect the manifold supports have on pressure drop. This can, however, still be seen in Figure 6.5.

The CFD results underestimate the pressure drop by 113% and 770% for the 1 mm design cold and hot sides, respectively, and 395% and 2993% for the 2 mm design cold and hot sides, respectively. The deviation is mainly due to the fact that the CFD model does not simulate the losses within the test bench setup. The inlet and outlet pressures were taken directly at the inlet and outlet of the heat exchanger rather than where the pressures were recorded on the experimental tests. The CFD model also applies a uniform surface roughness throughout the model. In contrast, the experimental model has different roughness depending on the surface and the manufacturing orientation.

Furthermore, the pressure drop was calculated and simulated with the assumption of a steady flow. The test bench, however, utilised a diaphragm pump to circulate the hot side fluid, inducing flow pulsations within the loop. Numerous studies have found that pulsating flow can have a negative effect on pressure drop within a system. Blythman [84] found that both the wall shear stress and pressure gradient were augmented when introducing pulsations compared to steady flow. However, the pressure gradient increases more significantly due to increased internal losses.

Similarly, a study done by Khosravi-Bizhaem [85] investigated experimentally the effects of pulsating flow for a range of Reynolds numbers (2000–9500) and Womersley numbers (16–40). The study reported an increase in pressure drop of 3–7% for pulsating flow compared to steady flow. An experimental study done by Davletshin et al. [87] found that the flow resistance values for pulsating flow were double that of the steady flow results within circular channels with semicircular annular ribs. Ribas [86] attributed this phenomenon to the relaminarisation and inertia effects of pulsating flows. The strong acceleration experienced gives rise to the relaminarisation process, resulting in the flow not maintaining its turbulent regime even at high Reynolds numbers. Furthermore, the transients' substantial effect on the wall shear stress is to the extent that it invalidates the stationary condition correlations developed for friction factor calculations.

6.5.4 0.5 mm design

Various attempts were made before successfully manufacturing the heat exchangers. As previously discussed, the 0.5 mm design proved especially difficult as the combination of thin walls and small form factor led to various challenges during the manufacturing process. The fourth manufacturing attempt yielded a successful 0.5 mm heat exchanger; however, during post-processing, a crack developed, rendering the heat exchanger nonfunctional. Time and budget limitations prevented the remanufacture of the heat exchanger. Therefore, the 0.5 mm heat exchanger results only entail the values obtained from the numerical simulations and the CFD modelling.

The numerical simulation results for the 0.5 mm heat exchanger design can be seen in Figure 6.12. Results are plotted for heat transfer, effectiveness and pressure drop. The deviation seen in heat transfer (*c.f.* Figure 6.12(a)) is due to the theoretical model assuming equal flow distribution within the channels, where the CFD model represents a more realistic unequal flow profile throughout the heat exchanger. The varying pressure within the heat exchanger core, left to right, shown in Figure 6.3(d); indicates non-uniform flow within the core.

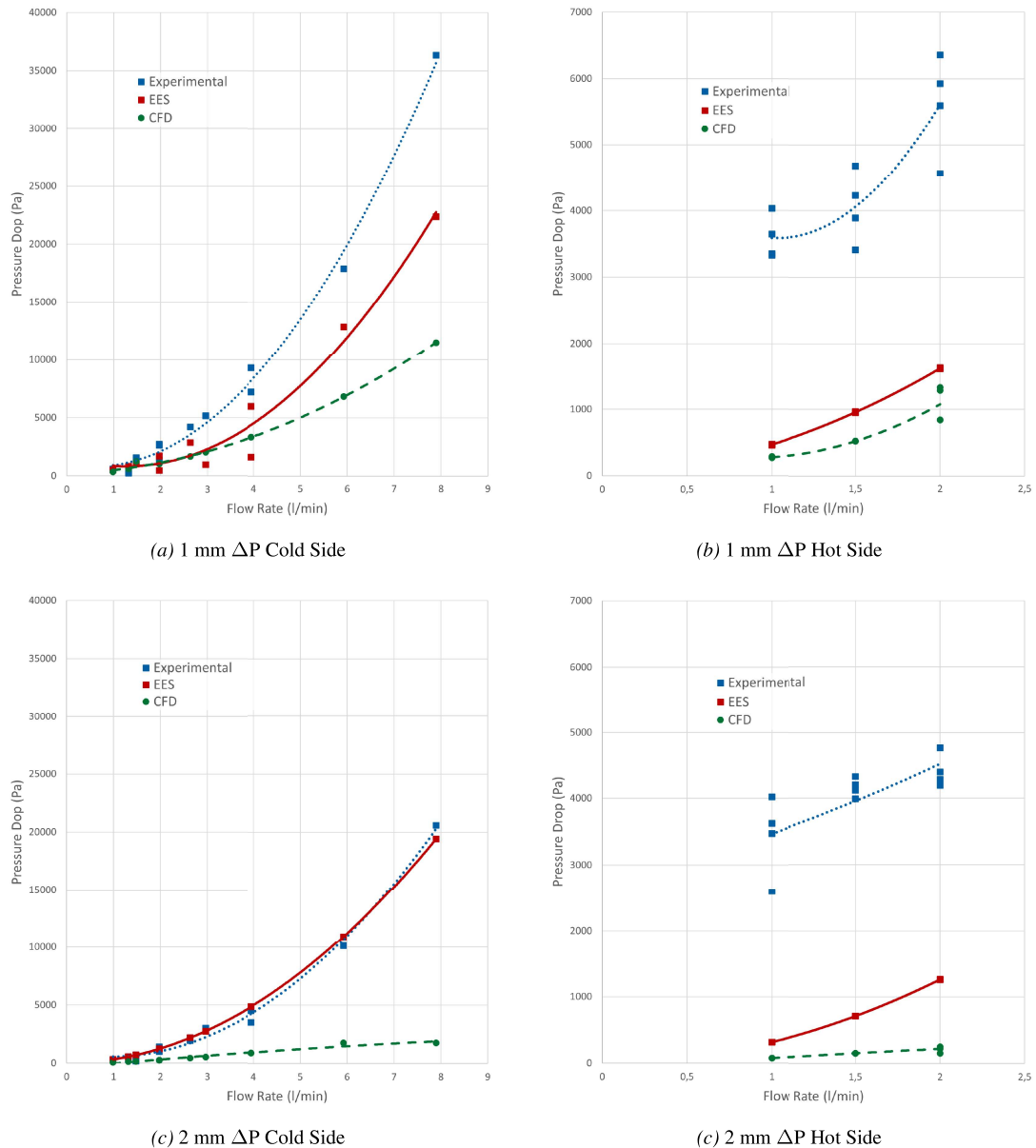


Figure 6.11: Comparison of pressure drop vs flow rate

The effectiveness of the heat exchanger is coupled to the heat transfer and shows the same trends. The theoretical model overestimates the heat transfer and effectiveness by 78% on average.

The deviation in pressure drop is believed to occur at the manifold inlets (*c.f.* Figure 6.12(c)). The theoretical model simulates pressure loss at the manifold entrance as sudden contractions; this is a basic representation of the pressure loss occurring at the manifold entrance. The CFD model, however, can simulate this loss with greater accuracy. In Figure 6.3(d), there is a prominent line where the pressure drops at the manifold entrance. On the other heat exchanger designs, the effect is less noticeable as the flow restriction at the manifold entrance is smaller, becoming more noticeable at higher flow rates. In this case, the theoretical model underpredicts the pressure drop by 26% on average.

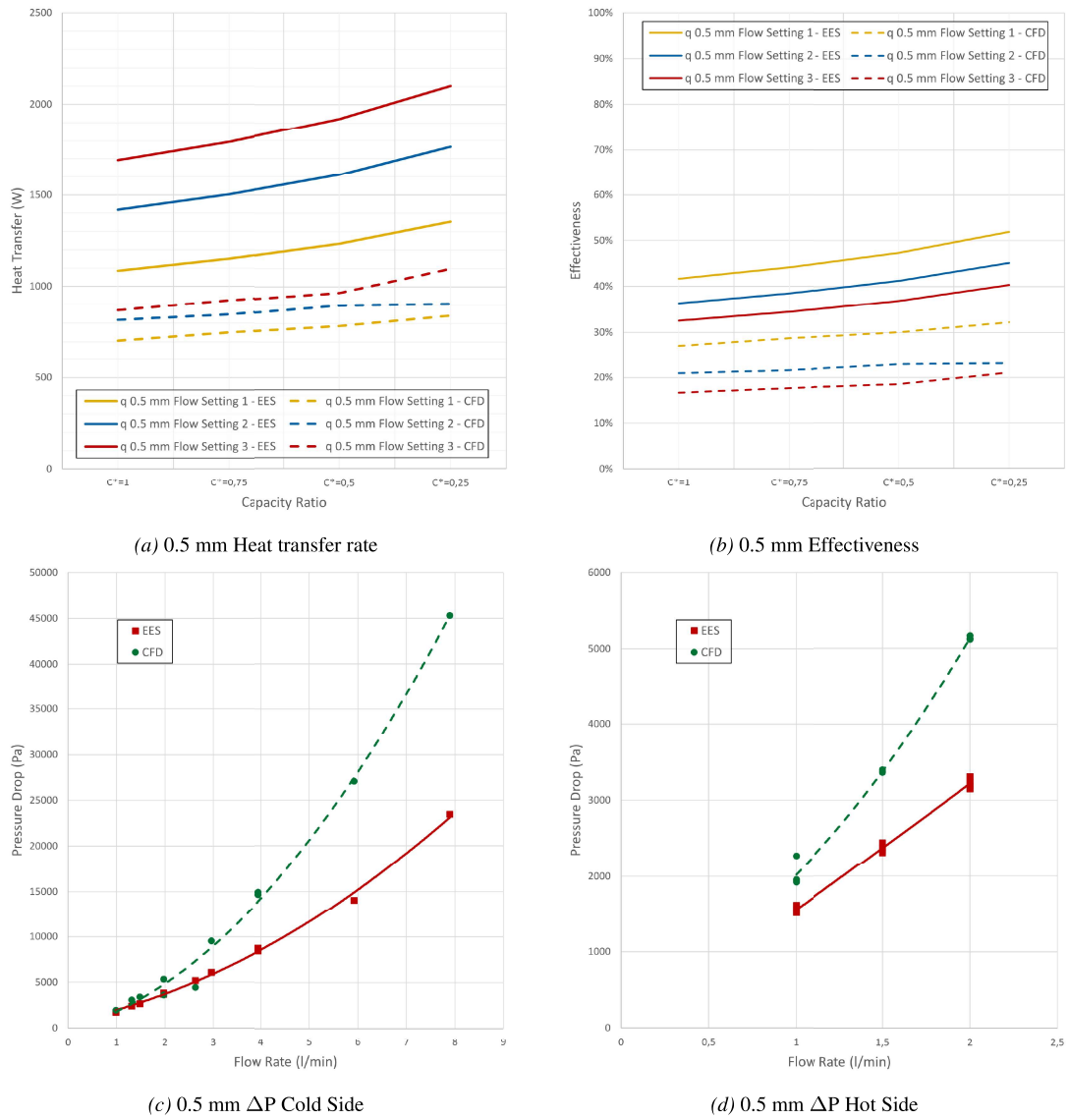


Figure 6.12: 0.5 mm design comparison: Theoretical vs CFD

6.5.5 R-134a numerical simulations

The organic fluid 1,1,1,2-Tetrafluoroethane (R-134a) is one of the working fluids used in Organic Rankine Cycles. Hence, the numerical model was also used to simulate results for R-134a - water application. The simulations were based on the same test matrix used for the water-water calculations, however, the temperatures and pressures were adjusted to keep the R-134a within its liquid state. The inlet parameters for the simulations are summarised in Table 6.7.

Similar trends can be seen for heat transfer rate, effectiveness and pressure drop compared to that of the water - water simulations.

Figure 6.13(a) shows the graph for heat transfer rate vs the capacity ratio, as with the previous results, the heat transfer rate increases with increased mass flow rate. The maximum heat transfer rate was achieved with the third flow setting and a capacity ratio of 0.25 for all three heat exchanger designs.

The maximum heat transfer rate achieved was 1236, 980.9 and 977.4 W for the 0.5, 1 and 2 mm heat exchanger designs, respectively. The corresponding effectiveness achieved was 51.37, 40.99 and 40.78% for the 0.5, 1 and 2 mm heat exchanger designs.

The maximum effectiveness for all heat exchanger designs occurred with the first flow setting and a capacity ratio of 0.25. A maximum effectiveness of 62.82, 49.16 and 46.56% was achieved for the 0.5, 1 and 2 mm heat exchanger designs, with a heat transfer rate of 761.9, 591.3 and 560 W, respectively.

Table 6.7: Test matrix R-134a

Capacity Ratio	Flow Rate (l/min)		Inlet Temperature (°C)		Inlet Pressure (Bar)	
	R-134a*	Water**	R-134a	Water	R-134a	Water
$C^* = 1$	$\begin{cases} 1 \\ 1.5 \\ 2 \end{cases}$	$\begin{cases} 0.43 \\ 0.646 \\ 0.86 \end{cases}$	0	40	6.5	2
$C^* = 0.75$	$\begin{cases} 1 \\ 1.5 \\ 2 \end{cases}$	$\begin{cases} 0.57 \\ 0.86 \\ 1.15 \end{cases}$	0	40	6.5	2
$C^* = 0.5$	$\begin{cases} 1 \\ 1.5 \\ 2 \end{cases}$	$\begin{cases} 0.86 \\ 1.29 \\ 1.72 \end{cases}$	0	40	6.5	2
$C^* = 0.25$	$\begin{cases} 1 \\ 1.5 \\ 2 \end{cases}$	$\begin{cases} 1.72 \\ 2.59 \\ 3.45 \end{cases}$	0	40	6.5	2

* User Set (C_{min})

** Calculated (C_{max})

An effectiveness - capacity ratio graph can be seen in Figure 6.13(b). The graph illustrates the same trends seen with the water - water simulations, an increase in mass flow rate results in increased effectiveness.

The pressure drop across the heat exchanger for R-134a and water side show similar results compared to the results obtained from the water - water simulations, (*c.f* Figure 6.13(C), (D)). The results do however indicate a slightly higher pressure drop for the 1 and 2 mm heat exchanger designs with R-134a. On average, the pressure drop is 11 and 28% higher for the 1 and 2 mm heat exchanger designs, respectively.

In contrast, the 0.5 mm heat exchanger experiences lower pressure drop for the first two flow settings and a higher pressure drop for the third flow setting, -21% average for the first two flow settings and an 52% increase for the third flow setting. Graph (e) in Figure 6.13 shows the comparison between the pressure drop results for the R-134a (R-134a - water) and the hot side water (water - water) simulations. The above-mentioned results are summarised in Table 6.8 and Table 6.9.

Table 6.8: R134-a results summary

Heat Exchanger	Results	Maximum Heat Transfer	Effectiveness ¹	Maximum Effectiveness	Heat Transfer ²
0.5 mm Design	Theoretical	1236 W	51.37%	62.82%	761.9 W
1 mm Design	Theoretical	980.9 W	40.99%	49.16%	591.3 W
2 mm Design	Theoretical	977.4 W	40.78%	46.56%	560 W

¹ at maximum heat transfer

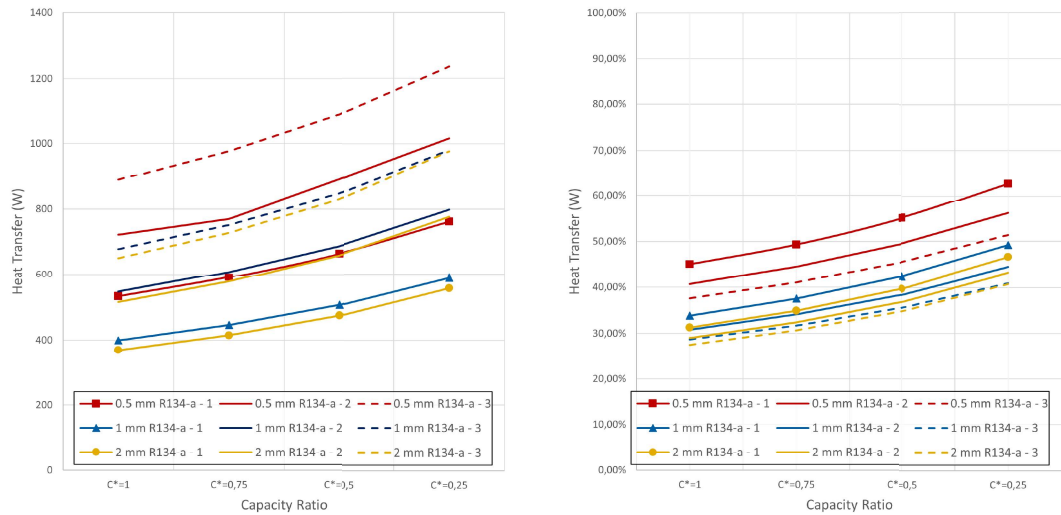
² at maximum effectiveness

Table 6.9: Pressure drop comparison summary

Heat Exchanger	Flow Setting	Pressure Drop (Pa)		% Difference
		R-134a ¹	Water ²	
0.5 mm Design	1	1023.25	1562.25	-34%
	2	2195.00	2373.25	-8%
	3	4913.25	3222.75	52%
1 mm Design	1	491.18	468.83	5%
	2	1068.75	957.30	12%
	3	1868.00	1615.75	16%
2 mm Design	1	404.95	316.45	28%
	2	911.35	709.85	28%
	3	1620.50	1259.75	29%

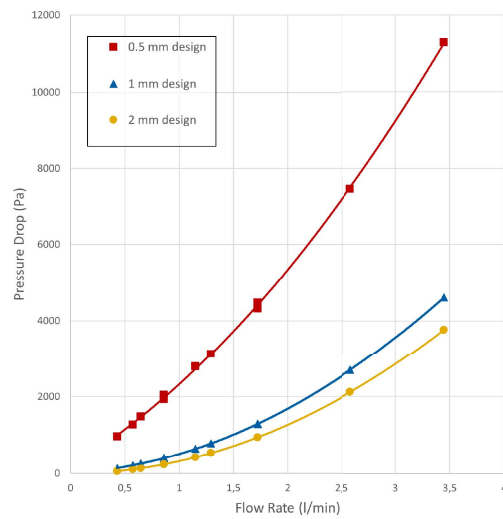
¹ results from R-134a - water numerical simulations

² results from water - water numerical simulations

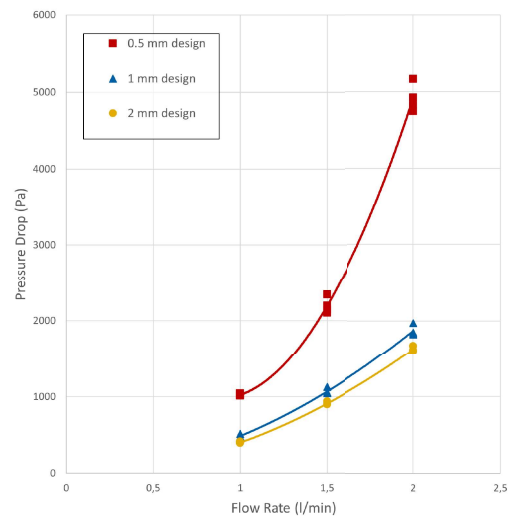


(a) Heat transfer rate vs capacity ratio

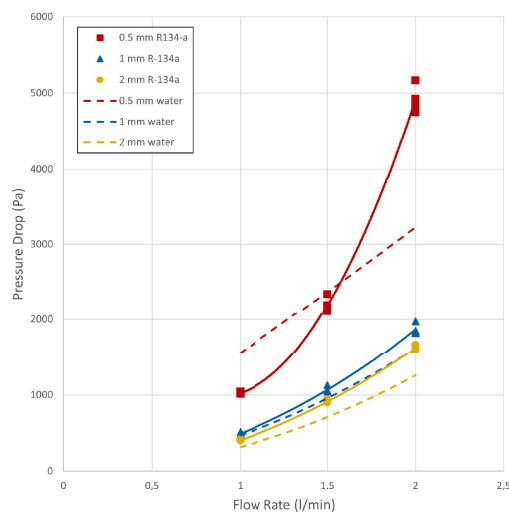
(b) Effectiveness vs capacity ratio



(c) 0.5 mm ΔP Water Side



(d) 0.5 mm ΔP R-134a Side



(e) Pressure drop comparison

Flow Setting 1 - 50% full-scale flow - R-134a
 Flow Setting 2 - 75% full-scale flow - R-134a
 Flow Setting 3 - 100% full-scale flow - R-134a

Figure 6.13: R-134a numerical simulation results

6.6 Summary and conclusion

6.6.1 Summary

The results from the 12 experimental tests per heat exchanger were presented, compared and discussed. The tests were run with three flow settings at different capacity ratios according to the test matrix. The theoretical and CFD simulations also used the test matrix to ensure comparisons were based on the same flow conditions. All results follow the same trend throughout the results; as the flow rate increases, the heat transfer and pressure drop also increases. The experimental results did show deviations from the numerical simulations, and the deviations and possible causes were discussed.

The theoretical model was used to size the initial heat exchanger design, and the effects of channel size, length and wall thickness on heat transfer were compared and discussed. It was found that by increasing the channel size and wall thickness, a decrease in heat transfer can be expected. While increasing the channel wall thickness and length has a negative effect on the pressure drop, the channel length shows the most promising results in terms of the heat transfer as a 100% increase in channel length amounts to a 100% increase in heat transfer and only a 36% increase in pressure drop (*c.f* Table 6.3).

The CFD simulations were run after the experimental testing; this gave good insight into the flow distribution through the heat exchanger. Similar results to the theoretical and experimental results were obtained, showing the same trends. The deviation from these results was discussed and possible causes were identified. The heat transfer rate, effectiveness and pressure drop from the experimental results were compared to the numerical results.

Compared to the CFD results for the heat transfer rate, the experimental results differs 2% and 71% for the 1 mm and 2 mm designs respectively. While the theoretical results differ 33% and 5% for the 1 mm and 2 mm designs compared to that of the CFD results. Possible causes for this deviation were discussed, with the leading cause being the theoretical model assuming uniform flow.

The pressure drop graphs showed good agreement with only 30% (1 mm design) and 11% (2 mm design) deviation from the theoretical model. However, the CFD model underpredicted the pressure drop for the 1 mm and 2 mm designs by 113% and 395%, respectively. The CFD model only accounts for the pressure drop within the heat exchanger. In contrast, the experimental and theoretical model includes more pressure drops coupled to the test bench setup.

Time limitations and computer resources governed the CFD simulations; adding piping and fittings as per the test bench setup increases the computation time and resources required for accurate simulation. The current CFD setup for the 0.5 mm heat exchanger ran approximately 110 minutes per test; 12 tests amount to 22 hours of simulation. Thus, it was decided to run CFD only on the heat exchangers.

The numerical simulation results for the 0.5 mm design were also discussed; compared to the CFD simulation, the theoretical model overpredicted the heat transfer rate by 78% and underpredicted the pressure drop by 26%. The causes for these deviations include the uniform flow the theoretical model assumes and the error in calculating the manifold inlet pressure losses.

Furthermore, numerical simulation results were discussed for an R-134a - water application on all three heat exchanger designs. A liquid only state was considered for the R-134a - water numerical simulations and the initial inlet conditions were adjusted accordingly. The results showed acceptable agreement compared to the water - water numerical simulations with the same trends in heat transfer rate, effectiveness and pressure drop.

6.6.2 Conclusion

The numerical simulation models presented in Chapter 3 were validated against the experimental data presented in this chapter. The results compared the heat transfer rate, effectiveness and pressure drop for each heat exchanger design. The data showed acceptable agreement between the numerical models and experimental data results. The R-134a - water numerical simulations results shows the capability of incorporating various working fluids within the model to find optimal operating conditions.

The deviations in results were identified and can be addressed by means of design changes to the experimental model. With the acquired knowledge from the CFD simulations, the flow paths can be optimised to ensure a more uniform flow distribution through the heat exchanger. Furthermore, pressure drops can be minimised by optimising the manifold supports.

The numerical model can be used to initially size a heat exchanger; after that, preliminary CFD simulations can be implemented to identify possible unwanted phenomena. This method would decrease the computational time required to size a heat exchanger with CFD simulation and produce a better heat exchanger design than would have been obtained by using only the theoretical model.

Chapter 7

Summary and conclusions

The feasibility of producing compact heat exchangers with additive manufacturing was demonstrated in this study. The results correlate well with the developed theoretical model, and tests show promising results for incorporating these devices into micro-ORC systems.

With the study serving as a basis for future research, further refining the theoretical model, optimising the manufacturing parameters and investigating different heat exchanger designs would prove beneficial. Furthermore, the study showcased Ti6Al4V as a viable material for compact heat exchangers, and the results were as expected.

This chapter will summarise the key findings, discuss some challenges found during the study and suggest recommendations for future research.

7.1 Modelling and design

Using existing theoretical methods and correlations, a theoretical model was developed to predict the performance of Ti6Al4V additive manufactured compact counter-flow heat exchangers. The model predicts the theoretical heat transfer and pressure drop of the chosen heat exchanger geometry with the input data known or selected.

The effects of channel length, size, shape and wall thickness on the heat transfer and pressure drop can be easily modelled with the model. The model can also be adapted to accommodate various heat exchanger configurations, including flow orientation, fluids, fluid states and materials.

The theoretical model provided a good baseline for heat exchanger design and was used to produce the heat exchangers tested in this study.

7.2 Design and manufacturing

Three initial CAD models of the heat exchangers were developed with the parameters and results obtained from the theoretical model. The designs were proposed to the engineers at CRPM and with guidance, they were optimised for additive manufacturing.

The heat exchanger designs comprised of different channel sizes packed in a rectangular array with alternating flow in each row of channels. The selected channel sizes for manufacturing were 0.5 mm, 1 mm, and 2 mm square channels, each with a theoretical heat transfer rate of about 2 kW.

The material buildup was the first challenge that arose during the manufacturing process. After several manufacturing attempts, it was found that changing the manufacturing orientation minimised the buildup of material, and two heat exchangers were successfully produced.



After preliminary testing of the heat exchangers, it was found that the heat exchanger cores were porous and they were sent for further analysis via CT scans. It was found that thermal stresses during manufacturing had caused the heat exchanger channel walls to warp and eventually fracture.

Increasing the wall thickness of the channels and adding additional supports within the heat exchanger reduced warping effects. Three heat exchangers models were successfully produced after implementing the aforementioned solutions, one of which unfortunately fractured during post-process machining. The other two were used for experimental characterisation.

Further optimisation of the manufacturing parameters and heat exchanger design should be considered for improved results.

7.3 Experimental setup

A test bench was designed and constructed to test and characterise the heat exchangers. The system is set up to use water as the working fluid. It allows the user to independently adjust the flow rate and pressure through the heat exchangers.

The test bench was designed to enable one user to easily operate the whole bench. The bench can accommodate various sizes and configurations of heat exchangers. Changing the flow meters would allow the test bench to work with various other fluids and gases.

The designed system was set up to accommodate a variety of testing parameters whilst ensuring the accuracy of the results. The heat exchangers and their piping were thermally insulated to ensure negligible heat loss during operation. The pressure and temperature sensors were placed as close as possible to the inlet and outlet of the heat exchangers to ensure accurate results were captured via the data acquisition system.

All instrumentation was tested and calibrated before testing the heat exchangers and everything performed as expected.

7.4 Experimental procedure and results

The testing of the heat exchangers was done as described in Chapter 5. A total of 12 flow rate settings were tested per heat exchanger, each consisting of at least five tests per flow rate setting. It resulted in 60 tests per heat exchanger. The tests ran for 6 minutes, recording a sample every three seconds. The first 20 samples of each test were used to check the test run's validity using a set-up Excel test sheet.

The results obtained from testing were accurate enough to characterise the heat exchangers successfully. Some discrepancies were found in the heat transfer and pressure drop compared to the calculated and CFD results. The leading cause for the deviation was the assumption of uniform flow through the heat exchanger core in the theoretical model, impacting both the heat transfer and pressure drop results.

The added supports within the heat exchanger manifolds complicated the pressure loss calculations, and extra attention should be applied to model these pressure losses more accurately in future. Furthermore, the effects of pulsating flow on pressure loss should be included for greater accuracy.



Table 7.1: Results summary

Heat Exchanger	Results	Maximum Heat Transfer	Effectiveness ¹	Maximum Effectiveness	Heat Transfer ²
<u>Water - Water</u>					
0.5 mm Design	Experimental	-	-	-	-
	Theoretical	2.103 kW	40.36 %	51.96 %	1.354 kW
	CFD	1.098 kW	21.03 %	32.24 %	0.842 kW
1 mm Design	Experimental	1.290 kW	24.93 %	34.86 %	0.898 kW
	Theoretical	1.836 kW	35.46 %	44.81 %	1.151 kW
	CFD	1.165 kW	21.20 %	33.10 %	0.909 kW
2 mm Design	Experimental	1.234 kW	23.98 %	30.77 %	0.794 kW
	Theoretical	1.903 kW	37.01 %	42.42 %	1.092 kW
	CFD	1.730 kW	33.49 %	46.10 %	1.191 kW
<u>R-134a - Water</u>					
0.5 mm Design	Theoretical	1.236 kW	51.37%	62.82%	0.762 kW
1 mm Design	Theoretical	0.981 kW	40.99%	49.16%	0.591 kW
2 mm Design	Theoretical	0.977 kW	40.78%	46.56%	0.560 kW

¹ at maximum heat transfer² at maximum effectiveness

Combining the theoretical model and CFD would yield more accurate results. Preliminary sizing can be done with the theoretical model and refined with CFD before manufacturing.

The R-134a - water numerical simulations shows promising results with similar trends compared to the results of the experimental, theoretical and CFD. The maximum heat transfer was achieved at 100% flow rate (hot fluid) and a capacity ratio of 0.25, whereas the maximum effectiveness was found at 50% flow rate (hot fluid) with a capacity ratio of 0.25. A summary of these results can be found in Table 7.1.

7.5 Recommendations and future research

During the study, it was observed that more research should be done on optimising Ti6Al4V heat exchangers for additive manufacturing. The channel wall thickness, size, shape, and manufacturing orientation all impact the ease of manufacturing, heat transfer and pressure drop and should be carefully considered to reduce costs whilst maintaining good performance characteristics.

Stress build-up during the manufacturing process should also be kept in mind as this could lead to unwanted warping or fractures in the channel walls. Further research on optimising the channel geometry, layout and wall thickness is required in this regard.



Future research can also focus on implementing the heat exchangers into ORCs, as time and budget constraints did not allow testing the additively manufactured heat exchangers with organic fluids. The effects of fouling were omitted in the study and could be a valuable topic of research, particularly focusing on the fouling effects with organic fluids in additively manufactured Ti6Al4V heat exchangers.

Furthermore, refining the theoretical model would allow more accurate predictions of the heat exchanger performance. CFD simulations provide beneficial insights into flow distribution within the heat exchanger and should be used to optimise the model before manufacturing.

Improvements to the test bench can also be made by replacing the current flow meters with more accurate digital flow meters, upgrading the external heated reservoir for more stable operating conditions or changing the diaphragm pump to eliminate the effect of pulsating flow.

Pulsating flow can, however, have a positive effect on heat transfer and should be considered for future research. Khosravi-Bizhaem [85] found a 39% increase in convective heat transfer during experimentation, with more enhancement at lower Reynolds numbers. These improvements or changes depend on the required testing parameters and should be reviewed during the design phase.

Valuable lessons were learnt during the execution of the research which may aid future researchers working in the field of additive manufactured compact heat exchangers. These lessons learnt are contained in Table 7.2.

Table 7.2: Lessons learnt for the design, manufacturing and experimentation of additive manufactured Ti6Al4V heat exchangers

No.	Lessons learnt	Why	Suggested further improvement
Heat exchanger thermal design and simulation			
1	Heat exchanger core configuration: The heat exchanger core should be of a simple configuration when initially developing the design methodology for additively manufactured heat exchangers.	To ease calculations and simulations. The 1-D design of heat exchangers is complex. Simple configurations ease the development of the design for AM.	Explore different and more complex channel shapes and configurations in future that hold the promise of optimised performance.
2	Value of CFD simulation: Conduct CFD analyses of preliminary 1-D designs to identify unequal flow distributions.	The 1-D design approach assumes equal flow distribution between flow channels. Unequal flow distribution will cause the design rating and/or effectiveness of the heat exchanger to not be reached, in return reducing the heat exchanger's efficiency.	

Table 7.2: Lessons learnt for the design, manufacturing and experimentation of additive manufactured Ti6Al4V heat exchangers (*cont.*)

No.	Lessons learnt	Why	Suggested further improvement
3	Heat exchanger manifold design: The inlet manifold configuration caused uneven flow distribution through the heat exchanger core. CFD analysis should be used to design the inlet manifold to ensure even flow distribution.	Uneven flow distribution in the core causes reduced heat exchanger performance.	
4	Heat exchanger manifold warpage: Long unsupported walls warp during the AM manufacturing process. Supports hinder flow within the manifold, negatively impacting pressure drop.	Warped heat exchanger walls are defects that cause a mismatch of mating parts.	Reduce the length of manifold walls.

Design for additive manufacturing

1	Overhang optimisation: Overhangs should be designed to have an angle greater than 35° with the horizontal to ensure a self-supporting structure.	This minimises the amount of post processing and machining required.	
2	Cut-out supports: Hole cut-outs or tube sections with their axis parallel to the horizontal should be internally supported.	These sections can either have removable supports or be designed with permanent supports where the removal of internal support poses a problem.	
3	Threaded holes, drilling and tapping: Threaded holes, pipes and nozzles should be additively manufactured with the intended screw thread to minimise screw-cutting operations.	Screw threads added during the additive manufacturing process reduce overall manufacturing time. Drilling and tapping smaller than M6 is not recommended, especially for difficult-to-machine alloys such as Ti6Al4V.	Screw threads smaller than M10 have poor definition.
4	Parts reduction: Functions should be combined to reduce the parts count.	Multiple parts can be combined to reduce parts count. A reduction in parts count reduces post processing time.	Functional analysis should be employed during the design process to optimise the function allocation of parts.

Table 7.2: Lessons learnt for the design, manufacturing and experimentation of additive manufactured Ti6Al4V heat exchangers (*cont.*)

No.	Lessons learnt	Why	Suggested further improvement
5	Sagging: The top surfaces of horizontal flow channels with circular or square cross-sections will sag if unsupported. For example, to achieve a circular flow channel, the top section must be designed to have a slight teardrop profile so that a circular cross-section is achieved after sagging has occurred.	If sagging is not accounted for then the AM process will produce distorted horizontal flow channels.	Research is required to characterise sagging of hollow horizontal channels and the establishment of design and manufacturing rules for the achievement of desired flow channels after sagging has occurred.

Additive manufacturing of heat exchanger

1	Manufacturing orientation: Manufacturing orientation yields different surface roughness depending on the manufacturing axis. Generally, the Z-axis surface roughness is greater than the roughness on the X- and Y-axis.	Rough surfaces can be advantageous; a rough surface finish can increase heat transfer at the cost of increased pressure drop.	
2	Manufacturing orientation: The manufacturing orientation should be carefully considered to eliminate or limit the number of supports required.	Optimal manufacturing orientation ensures less material being used during manufacturing and lessens the amount of post processing required.	The optimal manufacturing orientation may limit design. Internal supporting structures should be avoided due to the difficulty of removal during post processing.
3	Wall thickness: The achievable wall thickness with the selected AM process or machine might not be optimal for the heat exchanger design. Heat exchanger channels with a wall thickness of 0.35 mm were porous whereas a thickness of 0.5 mm was leak-proof.	Mixing between the cold and hot streams is not acceptable. Leaks between the cold and hot streams are not acceptable. Thicker walls reduce heat exchanger effectiveness.	Further research is required on the leak tightness of thin walls to determine the minimum leak-tight wall thickness for specific process conditions and materials.

Table 7.2: Lessons learnt for the design, manufacturing and experimentation of additive manufactured Ti6Al4V heat exchangers (*cont.*)

No.	Lessons learnt	Why	Suggested further improvement
4	Warping of thin wall sections: Thin wall sections might warp or fracture. This typically occurs with wall thicknesses less than 0.5 mm thick.		Increasing wall thickness or adapting the design to lessen straight unsupported walls should improve manufacturability.
5	Powder removal: The heat exchanger must be designed to allow metal powder removal within internal structures and channels during post processing.	Powder removal within the internal structures of the heat exchanger ensures proper functioning of the heat exchanger.	Inspection holes should be incorporated into the design to allow powder removal and servicing of the heat exchanger core.
6	Material buildup during the AM manufacturing process: The manufacturing process and selected heat exchanger core geometry caused material buildup during the manufacturing process. Successful manufacturing was accomplished by changing the recoating angle to 35° - 45° so that the recoating arm passes the heat exchanger core diagonally.	A combination of the heat exchanger core geometry and thin walls flicked extra material on already coated parts during the recoating process and changing the angle at which the recoating arm coats the core had positive effects.	Selecting a different geometry for the heat exchanger core (circular or triangular channels) or selecting different AM technology could eliminate the phenomenon.

Additive manufacturing materials

- | | | |
|---|--|--|
| 1 | Heat exchanger strength: The mechanical properties of Ti6Al4V allow the manufacturing of strong thin sections. | Thermally induced stress during manufacturing can cause fractures in thin sections and should be sufficiently reinforced. Thin sections joined to thicker sections may detach, joining these sections with chamfers or rounds reduces stress concentrations. |
| 2 | Ti6Al4V mechanical properties: The properties of Ti6Al4V make it difficult to machine, thus post-process machining should be limited. | |

Table 7.2: Lessons learnt for the design, manufacturing and experimentation of additive manufactured Ti6Al4V heat exchangers (*cont.*)

No.	Lessons learnt	Why	Suggested further improvement
Quality assurance and diagnostics			
1	Computed tomography (CT scans): CT scans give valuable insight into the heat exchanger core where visual inspection is not possible. It should form part of the development of new heat exchanger core configurations.	CT scans can be beneficial for fault finding or checking intricate heat exchanger core details. It is a valuable quality assurance technique.	
Experimental investigation			
1	Hot side reservoir temperature stability: The hot side reservoir should be designed to ensure stable hot side temperatures either by employing better temperature control and/or using a larger reservoir.	Unstable hot side temperatures lead to the inaccurate characterisation of the heat exchanger performance.	
2	Hot fluid pump overheating: The hot fluid pump overheated and consequently had to be operated intermittently, although the maximum hot fluid temperature did not exceed the maximum fluid temperature limit of 60°C.	Intermittent pump operation hampers testing.	Exchange the current pump with a pump capable of pumping high-temperature fluids continuously.



7.6 Conclusion

The capability of additive manufacturing for producing compact heat exchangers in Ti6Al4V was demonstrated in this study. A theoretical model was successfully developed for determining the performance of the heat exchangers before manufacturing and provided baseline parameters for detailed design and CAD modelling. The heat exchangers were characterised experimentally, and the results were compared to that of the theoretical model and CFD simulations.

A test bench was designed and built to characterise the heat exchangers. The bench was designed to accommodate various configurations of heat exchangers and can be modified with relative ease to fit specific applications for future research. High-quality instruments and materials were used to ensure the test bench's accuracy and durability. The heat exchangers performed as expected, with the experimental results correlating with the theoretical and CFD results.

With further research and optimisation, these devices can be incorporated into numerous processes and systems to increase efficiency.

References

- [1] C. Balaji, B. Srinivasan, and S. Gedupudi, *Heat exchangers*. Elsevier, 2020, pp. 199–231. [Online]. Available: <https://linkinghub.elsevier.com/retrieve/pii/B9780128185032000071>
- [2] S. Hall, *Heat Exchangers*, 5th ed. Elsevier, 2012, ch. 2, pp. 27–57.
- [3] W. Roetzel, X. Luo, and D. Chen, *Design and Operation of Heat Exchangers and Their Networks*, M. Convey, Ed. ELSEVIER, 2020.
- [4] R. Ronquillo. (2023, 1) Understanding heat exchangers - types, designs, applications and selection guide. [Online]. Available: <https://www.thomasnet.com/articles/process-equipment/understanding-heat-exchangers/>
- [5] R. J. Brogan. (2011, 1) Heat exchangers. [Online]. Available: <https://www.thermopedia.com/content/832/>
- [6] J. P. Holman, *Heat Transfer*, 10th ed. McGraw-Hill, 2010.
- [7] Conflux-Technology, “Case study: Our first patented design,” 11 2021. [Online]. Available: <https://www.confluxtechnology.com/article/case-study-conflux-s-first-patented-heat-exchanger>
- [8] T. Tartière and M. Astolfi, “A world overview of the organic rankine cycle market,” *Energy Procedia*, vol. 129, pp. 2–9, 2017.
- [9] F. Vélez, J. J. Segovia, M. C. Martín, G. Antolín, F. Chejne, and A. Quijano, “A technical, economical and market review of organic rankine cycles for the conversion of low-grade heat for power generation,” *Renewable and Sustainable Energy Reviews*, vol. 16, pp. 4175–4189, 8 2012. [Online]. Available: <https://linkinghub.elsevier.com/retrieve/pii/S1364032112002055>
- [10] M. A. Ancona, M. Bianchi, L. Branchini, A. D. Pascale, F. Melino, V. Orlandini, S. Ottaviano, A. Peretto, M. Pinelli, P. R. Spina, and A. Suman, “A micro-orc energy system: Preliminary performance and test bench development,” *Energy Procedia*, vol. 101, pp. 814–821, 2016.
- [11] J. Wajs, D. Mikielewicz, M. Bajor, and Z. Kneba, “Experimental investigation of domestic micro-chp based on the gas boiler fitted with orc module,” *Archives of Thermodynamics*, vol. 37, pp. 79–93, 9 2016.
- [12] B. Meng, M. Wan, R. Zhao, Z. Zou, and H. Liu, “Micromanufacturing technologies of compact heat exchangers for hypersonic precooled airbreathing propulsion: A review,” *Chinese Journal of Aeronautics*, vol. 34, pp. 79–103, 2 2021.
- [13] International-Energy-Agency, “World energy outlook 2021,” 2021. [Online]. Available: www.iea.org
- [14] —, “World energy outlook 2022,” 2022. [Online]. Available: www.iea.org
- [15] M. Papapetrou, G. Kosmadakis, A. Cipollina, U. L. Commare, and G. Micale, “Industrial waste heat: Estimation of the technically available resource in the eu per

industrial sector, temperature level and country,” *Applied Thermal Engineering*, vol. 138, pp. 207–216, 6 2018.

- [16] J. Ling-Chin, H. Bao, Z. Ma, W. Taylor, and A. P. Roskilly, *State-of-the-Art Technologies on Low-Grade Heat Recovery and Utilization in Industry*. IntechOpen, 1 2019. [Online]. Available: <https://www.intechopen.com/books/energy-conversion-current-technologies-and-future-trends/state-of-the-art-technologies-on-low-grade-heat-recovery-and-utilization-in-industry>
- [17] Alfa-Laval, “Heat exchanger designs,” 12 2022. [Online]. Available: <https://www.alfalaval.com/microsites/waste-heat-recovery/heat-exchanger-designs/>
- [18] International-Energy-Agency, “Climate change,” *Nature*, vol. 479, pp. 267–268, 11 2011. [Online]. Available: <https://www.nature.com/articles/479267b>
- [19] D. Woodford, “Large scale heat exchangers for energy, oil and gas: The growing role of additive manufacturing,” *Heat Exchanger World*, pp. 52–55, 11 2022. [Online]. Available: <https://heat-exchanger-world.com/large-scale-heat-exchangers-for-energy-oil-gas-the-growing-role-of-additive-manufacturing/>
- [20] G. Phylipsen, “Energy efficiency indicators,” *Berkeley Lab*, p. 102, 2016. [Online]. Available: <http://escholarship.org/uc/item/6tz8873z>
- [21] B. F. Tchanche, G. Lambrinos, A. Frangoudakis, and G. Papadakis, “Low-grade heat conversion into power using organic rankine cycles - a review of various applications,” *Renewable and Sustainable Energy Reviews*, vol. 15, pp. 3963–3979, 2011. [Online]. Available: <http://dx.doi.org/10.1016/j.rser.2011.07.024>
- [22] S. Lemmens and S. Lecompte, “Case study of an organic rankine cycle applied for excess heat recovery: Technical, economic and policy matters,” *Energy Conversion and Management*, vol. 138, pp. 670–685, 4 2017. [Online]. Available: <https://linkinghub.elsevier.com/retrieve/pii/S0196890417300936>
- [23] N. Yamada, Y. Tominaga, and T. Yoshida, “Demonstration of 10-wp micro organic rankine cycle generator for low-grade heat recovery,” *Energy*, vol. 78, pp. 806–813, 2014.
- [24] C. Klahn, F. Bechmann, S. Hofmann, M. Dinkel, and C. Emmelmann, “Laser additive manufacturing of gas permeable structures,” *Physics Procedia*, vol. 41, pp. 873–880, 2013. [Online]. Available: <http://dx.doi.org/10.1016/j.phpro.2013.03.161>
- [25] S. Tsopanos, C. J. Sutcliffe, and I. Owen, “The manufacture of micro cross-flow heat exchangers by selective laser melting,” *Proc. of the 5th Int. Conference on Enhanced, Compact and Ultra-Compact Heat Exchangers: Science, Engineering and Technology*, pp. 410–417, 2005.
- [26] P. A. Kew and D. A. Reay, “Compact/micro-heat exchangers - their role in heat pumping equipment,” *Applied Thermal Engineering*, vol. 31, pp. 594–601, 2011. [Online]. Available: <http://dx.doi.org/10.1016/j.applthermaleng.2010.08.003>
- [27] T. Popescu, M. Marinescu, H. Pop, G. Popescu, and M. Feidt, “Microchannel heat exchangers-present and perspectives,” *U.P.B. Sci. Bull., Series D*, vol. 74, 2012.
- [28] S. Ashman and S. G. Kandlikar, “A review of manufacturing processes for

- microchannel heat exchanger fabrication,” in *A Review of Manufacturing Processes for Microchannel Heat Exchanger Fabrication*, vol. 2006. ASME, 6 2006, pp. 855–860. [Online]. Available: <http://proceedings.asmedigitalcollection.asme.org/proceeding.aspx?articleid=1601433>
- [29] M. A. Arie, A. H. Shooshtari, S. V. Dessiatoun, and M. M. Ohadi, “Performance characterization of an additively manufactured titanium (ti64) heat exchanger for an air-water cooling application,” *ASME*, p. 1059, 2016. [Online]. Available: <https://proceedings.asmedigitalcollection.asme.org/proceeding.aspx?articleid=2584530>
- [30] A. Zhakeyev, P. Wang, L. Zhang, W. Shu, H. Wang, and J. Xuan, “Additive manufacturing: Unlocking the evolution of energy materials,” *Advanced Science*, vol. 4, 10 2017.
- [31] J. R. McDonough, “A perspective on the current and future roles of additive manufacturing in process engineering, with an emphasis on heat transfer,” *Thermal Science and Engineering Progress*, vol. 19, 10 2020.
- [32] T. Tartière, “Orc world map,” 12 2022. [Online]. Available: <https://orc-world-map.org/>
- [33] Ormat-Technologies-Inc., “Global projects.” [Online]. Available: <https://www.ormat.com/en/projects/all/main/?Country=0&Seg=0&Tech=8&pageNum=1>
- [34] Turboden, “Installation of orc plants all over the world,” 12 2022. [Online]. Available: <https://www.turboden.com/references?countries=&solutions=waste-heat-recovery&countries=&casolutions=#filters>
- [35] Exergy, “Enabling energy efficiency and sustainability,” 12 2022. [Online]. Available: <https://www.exergy-orc.com/>
- [36] M. Bianchi, L. Branchini, A. D. Pascale, V. Orlandini, S. Ottaviano, M. Pinelli, P. R. Spina, and A. Suman, “Experimental performance of a micro-orc energy system for low grade heat recovery,” *Energy Procedia*, vol. 129, pp. 899–906, 2017.
- [37] M. Bianchi, L. Branchini, A. D. Pascale, F. Melino, S. Ottaviano, A. Peretto, N. Torricelli, and G. Zampieri, “Performance and operation of micro-orc energy system using geothermal heat source,” *Energy Procedia*, vol. 148, pp. 384–391, 2018.
- [38] J. Wajs, D. Mikielwicz, and B. Jakubowska, “Performance of the domestic micro orc equipped with the shell-and-tube condenser with minichannels,” *Energy*, vol. 157, pp. 853–861, 8 2018.
- [39] F. Fatigati, D. Vittorini, M. D. Bartolomeo, and R. Cipollone, “Experimental characterization of a small-scale solar organic rankine cycle (orc) based unit for domestic microcogeneration,” *Energy Conversion and Management*, vol. 258, 4 2022.
- [40] M. A. Ancona, M. Bianchi, L. Branchini, A. D. Pascale, F. Melino, A. Peretto, C. Polletto, and N. Torricelli, “Solar driven micro-orc system assessment for residential application,” *Renewable Energy*, vol. 195, pp. 167–181, 8 2022.
- [41] P. Klonowicz, L. Witanowski, L. Jedrzejewski, T. Suchocki, and P. Lampart, “A turbine based domestic micro orc system,” *Energy Procedia*, vol. 129, pp. 923–930, 2017.

- [42] L. N. L. D. Lacalle, J. Pérez, J. I. Llorente, and J. A. Sánchez, “Advanced cutting conditions for the milling of aeronautical alloys,” *Journal of Materials Processing Technology*, vol. 100, pp. 1–11, 2000.
- [43] D. Dornfeld, S. Min, and Y. Takeuchi, “Recent advances in mechanical micromachining,” *CIRP Annals - Manufacturing Technology*, vol. 55, pp. 745–768, 2006.
- [44] Ruhlamat, “Die-sinking edm - contactless machining of hardest materials,” 12 2022. [Online]. Available: <https://www.ruhlamat.com/en/wiki/die-sinking-edm#metalworking-with-electric-charge>
- [45] ISO, “Iso/astm 52900(en), additive manufacturing - general principles - terminology,” International Organisation of Standardization, 12 2022. [Online]. Available: <https://www.iso.org/obp/ui/#iso:std:iso-astm:52900:dis:ed-2:v1:en>
- [46] M. Armstrong, H. Mehrabi, and N. Naveed, “An overview of modern metal additive manufacturing technology,” *Journal of Manufacturing Processes*, vol. 84, pp. 1001–1029, 12 2022.
- [47] T. C. Dzogbewu, S. K. Fianko, N. Amoah, S. A. Jnr, and D. de Beer, “Additive manufacturing in south africa: critical success factors,” *Heliyon*, vol. 8, 11 2022. [Online]. Available: <https://linkinghub.elsevier.com/retrieve/pii/S2405844022031401>
- [48] Mordor-Intelligence-Inc, “3d printing market research reports and analysis,” 12 2022. [Online]. Available: <https://www.marketresearch.com/Mordor-Intelligence-LLP-v4018/3D-Printing-c1970/1.html>
- [49] Siemens, “Sheet lamination — siemens software.” [Online]. Available: <https://www.plm.automation.siemens.com/global/en/our-story/glossary/sheet-lamination/55512>
- [50] I. Gibson, D. W. Rosen, and B. Stucker, *Additive Manufacturing Technologies*. Springer US, 2010. [Online]. Available: <http://link.springer.com/10.1007/978-1-4419-1120-9>
- [51] T. D. Ngo, A. Kashani, G. Imbalzano, K. T. Q. Nguyen, and D. Hui, “Additive manufacturing (3d printing): A review of materials, methods, applications and challenges,” *Composites Part B: Engineering*, vol. 143, pp. 172–196, 6 2018.
- [52] X. Zhang, R. Tiwari, A. H. Shooshtari, and M. M. Ohadi, “An additively manufactured metallic manifold-microchannel heat exchanger for high temperature applications,” *Applied Thermal Engineering*, vol. 143, pp. 899–908, 10 2018.
- [53] D. Bacellar, V. Aute, Z. Huang, and R. Radermacher, “Design optimization and validation of high-performance heat exchangers using approximation assisted optimization and additive manufacturing,” *Science and Technology for the Built Environment*, vol. 23, pp. 896–911, 8 2017.
- [54] C. Y. Yap, C. K. Chua, Z. L. Dong, Z. H. Liu, D. Q. Zhang, L. E. Loh, and S. L. Sing, “Review of selective laser melting: Materials and applications,” *Applied Physics Reviews*, vol. 2, 12 2015.
- [55] W. Shi, Y. Liu, X. Shi, Y. Hou, P. Wang, and G. Song, “Beam diameter dependence of performance in thick-layer and high-power selective laser melting of ti-6al-4v,”

Materials, vol. 11, 7 2018.

- [56] B. Vandenbroucke and J. Kruth, “Selective laser melting of biocompatible metals for rapid manufacturing of medical parts,” *Rapid Prototyping Journal*, vol. 13, pp. 196–203, 8 2007.
- [57] C. Veiga, J. P. Devim, and A. J. R. Loureiro, “Properties and applications of titanium alloys: a brief review,” *Reviews on Advanced Materials Science*, vol. 32, pp. 133–148, 2012.
- [58] M. A. Arie, A. H. Shooshtari, and M. M. Ohadi, “Experimental characterization of an additively manufactured heat exchanger for dry cooling of power plants,” *Applied Thermal Engineering*, vol. 129, pp. 187–198, 2018.
- [59] Q. Liu, Y. Wang, H. Zheng, K. Tang, L. Ding, H. Li, and S. Gong, “Microstructure and mechanical properties of lmd-slm hybrid forming ti6al4v alloy,” *Materials Science and Engineering A*, vol. 660, pp. 24–33, 2016. [Online]. Available: <http://dx.doi.org/10.1016/j.msea.2016.02.069>
- [60] T. D. Eastop and A. McConkey, *Applied Thermodynamics: For Engineering Technologists*. European Journal of Engineering Education, 1993, vol. 18.
- [61] S. S. Mehendale, A. M. Jacobi, and R. K. Shah, “Fluid flow and heat transfer at micro- and meso-scales with application to heat exchanger design,” *Applied Mechanics Reviews*, vol. 53, pp. 175–193, 7 2000. [Online]. Available: <https://experts.illinois.edu/en/publications/fluid-flow-and-heat-transfer-at-micro-and-meso-scales-with-applic>
- [62] Y. Fan and L. Luo, “Recent applications of advances in microchannel heat exchangers and multi-scale design optimization,” *Heat Transfer Engineering*, vol. 29, pp. 461–474, 5 2008.
- [63] Engineers-Edge, “Parallel and counter flow designs heat exchangers,” 11 2022. [Online]. Available: https://www.engineersedge.com/heat_transfer/parallel_counter_flow_designs.htm
- [64] G. Sidebotham, *Heat Transfer Modeling*. Springer International Publishing, 2015. [Online]. Available: <http://link.springer.com/10.1007/978-3-319-14514-3>
- [65] M. I. Hasan, A. A. Rageb, M. Yaghoubi, and H. Homayoni, “Influence of channel geometry on the performance of a counter flow microchannel heat exchanger,” *International Journal of Thermal Sciences*, vol. 48, pp. 1607–1618, 2009. [Online]. Available: <http://dx.doi.org/10.1016/j.ijthermalsci.2009.01.004>
- [66] B. K. Hodge and R. P. Taylor, *Analysis and design of energy systems*. Prentice Hall, 1999.
- [67] S. B. Amirault, “Fluid mechanics: Entrance length,” 6 2023. [Online]. Available: <https://sbainvent.com/fluid-mechanics/fully-developed-flow/>
- [68] W. M. Kays and A. L. London, *Compact Heat Exchangers*, 3rd ed. Scientific International, 2018.
- [69] C. F. Meyer, *Applications of Fluid Mechanics Part 3*, 1st ed. CM TEK Lecture materials cc, 2003.

- [70] Hawk-Ridge-Systems, “Enhanced turbulence modeling in solidworks flow simulation.”
- [71] C. K. G. Lam and K. Bremhorst, “A modified form of the k-e model for predicting wall turbulence,” *ASME Journal of Fluids Engineering*, vol. 103, pp. 456–460, 1983.
- [72] A. Sobachkin, *Numerical Basis of CAD-Embedded CFD*, 2014.
- [73] E. R. V. Driest, “On turbulent flow near a wall,” *Journal of the Aeronautical Sciences*, vol. 23, pp. 1007–1011, 11 1956.
- [74] I. P. Ginzburg, “Theory of drag and heat transfer,” Pushkin Leningrad State University, Leningrad, 1970.
- [75] T. McMahan, “Nasa 3d printed fuel rocket pump,” 2015. [Online]. Available: <https://www.nasa.gov/centers/marshall/news/news/releases/2015/successful-nasa-rocket-fuel-pump-tests-pave-way-for-3-d-printed-demonstrator-engine.html>
- [76] S. C. Venter, G. G. Jacobs, and J. J. D. Preez, “Design considerations for developing an additive manufactured ti-6al-4v compact counter-flow heat exchanger for application in organic rankine cycles,” in *19th RAPDASA Annual International Conference*, 2018, pp. 192–200.
- [77] S. C. Venter and G. G. Jacobs, “Development and characterisation of a ti6al4v additive manufactured compact counter-flow heat exchanger for application in organic rankine cycles,” in *20th RAPDASA Annual International Conference*, 2019, pp. 262–275.
- [78] P. Regtien and E. Dertien, *Sensors for Mechatronics*, 2nd ed. Elsevier, 2018. [Online]. Available: <https://linkinghub.elsevier.com/retrieve/pii/C20160050593>
- [79] R. Keim, “What is electrical noise and where does it come from?” *All About Circuits*, 2018. [Online]. Available: <https://www.allaboutcircuits.com/technical-articles/electrical-noise-what-causes-noise-in-electrical-circuits/>
- [80] ISO and IEC, “Uncertainty of measurement – part 1: Introduction to the expression of uncertainty in measurement,” International Organization for Standardization, 9 2009. [Online]. Available: <https://www.iso.org/standard/46383.html>
- [81] B. N. Taylor and C. E. Kuyatt, “Guidelines for evaluating and expressing the uncertainty of nist measurement results,” National Institute of Standards and Technology, 1994.
- [82] S. Bell, *Good Practice Guide No. 11 The Beginner’s Guide to Uncertainty of Measurement*, 2nd ed., 2001.
- [83] Merriam-Webster, “Steady state definition,” 10 2022. [Online]. Available: <https://www.merriam-webster.com/dictionary/steadystate>
- [84] R. Blythman, T. Persoons, N. Jeffers, and D. B. Murray, “Effect of oscillation frequency on wall shear stress and pressure drop in a rectangular channel for heat transfer applications,” in *Effect of oscillation frequency on wall shear stress and pressure drop in a rectangular channel for heat transfer applications*, vol. 745. Institute of Physics Publishing, 10 2016.

- [85] H. Khosravi-Bizhaem, A. Abbassi, and A. Z. Ravan, “Heat transfer enhancement and pressure drop by pulsating flow through helically coiled tube: An experimental study,” *Applied Thermal Engineering*, vol. 160, 9 2019.
- [86] F. A. Ribas and C. J. Deschamps, “Friction factor under transient flow condition,” in *Friction Factor Under Transient Flow Condition*. Purdue e-Pubs, 2004, pulsating flow. [Online]. Available: <https://docs.lib.purdue.edu/icec/1665>
- [87] I. A. Davletshin, O. A. Dushina, N. I. Mikheev, and S. A. Kolchin, “Estimation of friction loss under forced flow pulsations in a channel with discrete roughness elements,” vol. 891. Institute of Physics Publishing, 11 2017.
- [88] J. P. Holman, *Experimental Methods for Engineers*, 8th ed. McGraw-Hill, 2012.
- [89] NIST, “Measurement uncertainty,” National Institute of Standards and Technology, 9 2021. [Online]. Available: <https://www.nist.gov/itl/sed/topic-areas/measurement-uncertainty>
- [90] V. F. Dean, “Guide to the expression of uncertainties,” International Criticality Safety Benchmark Evaluation Project, 9 2008.

Appendix A

Uncertainty analysis

A.1 Introduction

The uncertainty of measurement tells us something about its quality; it is the doubt that exists about the result of any measurement. This margin of doubt can be quantified by the width of the margin (interval) and the confidence level and may be taken as the possible value the error may have [82, 88].

Making a measurement at least three times is considered wise. A better estimate of a “true” value can be obtained by using more measurements. A popular choice for a number of measurements is ten. Twenty yields a slightly better estimate, and 50 is a slightly better estimate than 20. [82]

Uncertainties arise from multiple sources and can be categorised as gross, fixed, systematic, and random errors [88]. These include uncertainty in the measuring equipment, the object, the process, “imported” uncertainties, operator skill, sampling issues, and the environment.

The range of a set of values can be characterised to have either a normal or a rectangular distribution. While there are other distributions, these are the most common. In a normal distribution, the values will likely fall near the average with some deviation. In a rectangular distribution the values are more evenly spread throughout the range. [82]

Identifying the source of uncertainty is the first step in calculating the uncertainty of a measurement. Each source adds a certain magnitude of uncertainty which is then added together to obtain an overall figure of uncertainty.

Methods utilised by various international organisations to estimate uncertainty are Type **A** and Type **B** evaluations [82, 89]. Type **A** evaluations are performed using statistical methods, and Type **B** evaluations make use of other means [90].

Type **A** uses simple statistical methods for calculating uncertainty. The standard uncertainty can be estimated using the following formula:

$$u = \frac{s}{\sqrt{n}} \quad (\text{A.1})$$

where u represents the estimated standard uncertainty, s the estimated standard deviation and n the number of measurements within the set.

The estimated standard deviation can be obtained through the following formula:

$$s = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{(n - 1)}} \quad (\text{A.2})$$

Where x_i is the result of the i^{th} measurement and $\bar{x}$ the mean of the data set.

Type **B** evaluation is used where information scarcity persists [82]. In these cases, only an upper and a lower limit of uncertainty are likely to be estimated. Assuming a rectangular distribution, the standard uncertainty is obtained from:

$$\frac{a}{\sqrt{3}} \quad (\text{A.3})$$

with a being the half width of the resolution. Data from manufacturers' calibration certificates can be assumed to be normally distributed, thus the divisor = 2.

Individual uncertainties are then, as stated, combined to obtain the combined standard uncertainty u_c .

$$u_c = \sqrt{u_1^2 + u_2^2 + u_3^2 \dots + u_n^2} \quad (\text{A.4})$$

The combined standard uncertainty links a confidence level of approximately 68%. However, the combined standard uncertainty can be scaled to include a higher level of confidence by using a coverage factor.

$$U = k u_c \quad (\text{A.5})$$

Where U represents the expanded uncertainty, k is the coverage factor for which a particular level of confidence is given by a particular coverage factor value.

$k = 1$: confidence level = 68%

$k = 2$: confidence level = 95%

$k = 2.58$: confidence level = 99%

$k = 3$: confidence level = 99.7%

For normally distributed combined standard uncertainty, a value of $k = 2$ can be assumed, giving us a 95% confidence level.

Gathering enough data to ensure an accurate representation of a Type **A** analysis requires time but due to the lack of time, a Type **B** analysis was used.

The uncertainties of pressure, temperature and flow measurements were incorporated into a spreadsheet program using the methods described above (*c.f.* Tables A.1, A.2 and A.3).



Table A.1: Summary of calculations for uncertainty of temperature measurement

Variable	Units	Range min	Range max	Instrument Description	Source of Uncertainty	± Value	Probability distribution	Divisor	Standard uncertainty	(u _i) ²
Hot Inlet Temperature	°C	0	100	400 PT100 Transmitter Jumo 3 wire PT100, connected to Transmitter > Picolog 1216	Picolog 1216 resolution	0,012	Rectangular	1,732	0,007 °C	0,000050
					Picolog 1216 accuracy	0,500	Normal	2,000	0,250 °C	0,063
					Shunt resistor circuit	0,090	Normal	2,000	0,045 °C	0,002
					Jumo PT100 instrument error:	0,040	Normal	2,000	0,020 °C	0,0004
					TranTech Temperature Transmitter	0,100	Normal	2,000	0,050 °C	0,0025
					Combined standard uncertainty (68% confidence level)		Normal	n/a	0,255 °C	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Normal (k = 2)	n/a	0,520 °C	0,520%
Cold Inlet Temperature	°C	0	100	400 PT100 Transmitter Jumo 3 wire PT100, connected to Transmitter > Picolog 1216	Picolog 1216 resolution	0,012	Rectangular	1,732	0,007 °C	0,000050
					Picolog 1216 accuracy	0,500	Normal	2,000	0,250 °C	0,063
					Shunt resistor circuit	0,090	Normal	2,000	0,045 °C	0,002
					Jumo PT100 instrument error:	0,040	Normal	2,000	0,020 °C	0,0004
					TranTech Temperature Transmitter	0,100	Normal	2,000	0,05 °C	0,0025
					Combined standard uncertainty (68% confidence level)		Normal	n/a	0,260 °C	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Normal (k = 2)	n/a	0,520 °C	0,520%
Hot Outlet Temperature	°C	0	100	400 PT100 Transmitter Jumo 3 wire PT100, connected to Transmitter > Picolog 1216	Picolog 1216 resolution	0,012	Rectangular	1,732	0,007 °C	0,000050
					Picolog 1216 accuracy	0,500	Normal	2,000	0,250 °C	0,063
					Shunt resistor circuit	0,090	Normal	2,000	0,045 °C	0,002
					Jumo PT100 instrument error:	0,040	Normal	2,000	0,020 °C	0,0004
					TranTech Temperature Transmitter	0,100	Normal	2,000	0,05 °C	0,0025
					Combined standard uncertainty (68% confidence level)		Normal	n/a	0,260 °C	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Normal (k = 2)	n/a	0,520 °C	0,520%
Cold Outlet Temperature	°C	0	100	400 PT100 Transmitter Jumo 3 wire PT100, connected to Transmitter > Picolog 1216	Picolog 1216 resolution	0,012	Rectangular	1,732	0,007 °C	0,000050
					Picolog 1216 accuracy	0,500	Normal	2,000	0,250 °C	0,063
					Shunt resistor circuit	0,090	Normal	2,000	0,045 °C	0,002
					Jumo PT100 instrument error:	0,040	Normal	2,000	0,020 °C	0,0004
					TranTech Temperature Transmitter	0,100	Normal	2,000	0,05 °C	0,0025
					Combined standard uncertainty (68% confidence level)		Normal	n/a	0,260 °C	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Normal (k = 2)	n/a	0,520 °C	0,520%

Table A.2: Summary of calculations for uncertainty of pressure measurement

Variable	Units	Range min	Range max	Instrument Description	Source of Uncertainty	± Value	Probability distribution	Divisor	Standard uncertainty	(u _i) ²
Hot Inlet Pressure	Bar abs	0	40	Gems Sensor 3500 transmitter, connected to Picolog 1216	Picolog 1216 resolution	0,005	Rectangular	1,732	0,003	0,000008
					Picolog 1216 accuracy	0,500	Normal	2,000	0,250	0,063
					Shunt resistor circuit	0,090	Normal	2,000	0,045	0,002
					Gems Sensor 3500 instrument error: 0.25% of FS	0,100	Normal	2,000	0,050	0,003
					Combined standard uncertainty (68% confidence level)		Normal	n/a	0,259	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Normal (k = 2)	n/a	0,518	1,295%
Cold Inlet Pressure	Bar abs	0	40	Gems Sensor 3500 transmitter, connected to Picolog 1216	Picolog 1216 resolution	0,005	Rectangular	1,732	0,003	0,000008
					Picolog 1216 accuracy	0,500	Normal	2,000	0,250	0,063
					Shunt resistor circuit	0,090	Normal	2,000	0,045	0,002
					Gems Sensor 3500 instrument error: 0.25% of FS	0,100	Normal	2,000	0,050	0,003
					Combined standard uncertainty (68% confidence level)		Normal	n/a	0,259	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Normal (k = 2)	n/a	0,518	1,295%
Hot Outlet Pressure	Bar abs	0	40	Gems Sensor 3500 transmitter, connected to Picolog 1216	Picolog 1216 resolution	0,005	Rectangular	1,732	0,003	0,000008
					Picolog 1216 accuracy	0,500	Normal	2,000	0,250	0,063
					Shunt resistor circuit	0,090	Normal	2,000	0,045	0,002
					Gems Sensor 3500 instrument error: 0.25% of FS	0,100	Normal	2,000	0,050	0,003
					Combined standard uncertainty (68% confidence level)		Normal	n/a	0,259	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Normal (k = 2)	n/a	0,518	1,295%
Cold Outlet Pressure	Bar abs	0	40	Gems Sensor 3500 transmitter, connected to Picolog 1216	Picolog 1216 resolution	0,005	Rectangular	1,732	0,003	0,000008
					Picolog 1216 accuracy	0,500	Normal	2,000	0,250	0,063
					Shunt resistor circuit	0,090	Normal	2,000	0,045	0,002
					Gems Sensor 3500 instrument error: 0.25% of FS	0,100	Normal	2,000	0,050	0,003
					Combined standard uncertainty (68% confidence level)		Normal	n/a	0,259	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Normal (k = 2)	n/a	0,518	1,295%

Table A.3: Summary of calculations for uncertainty of flow measurement

Variable	Units	Range min	Range max	Instrument Description	Source of Uncertainty	± Value	Probability distribution	Divisor	Standard uncertainty		$(u_i)^2$
Hot Flow Rate	l/min	0,2	2	Parker S Series Variable Area Flowmeter	Flowmeter resolution = 0,05 l/min	0,025	Rectangular	1,732	0,014	l/min	0,000208
					Flowmeter accuracy = ± 2% of FS	0,040	Normal	2,000	0,020	l/min	0,000400
					Combined standard uncertainty (68% confidence level)		Normal	n/a	0,02	l/min	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Normal (k = 2)	n/a	0,05	l/min	2,466%
Cold Flow Rate	l/min	1,5	10	Parker S Series Variable Area Flowmeter	Flowmeter resolution = 0,05 l/min	0,025	Rectangular	1,732	0,014	l/min	0,000208
					Flowmeter accuracy = ± 2% of FS	0,200	Normal	2,000	0,100	l/min	0,010000
					Combined standard uncertainty (68% confidence level)		Normal	n/a	0,10	l/min	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Normal (k = 2)	n/a	0,20	l/min	2,021%

Appendix B

Excel test sheet sample



Test No:		C025-75-5										
Date:		17-10-2022										
	Samples	C1 Hot Inlet	C2 Hot Outlet	C5 Hot Inlet	C6 Hot Outlet	C11 Cold	C12 Cold	C14 Cold	C16 Cold Outlet	Flow Rate	Flow Rate Cold Side	Atmospheric
		Degrees Celsius		Bar (AP)		Bar (AP)		Bar (AP)		per Meter	per Minute	Bar
		Thi	Tho	Phi	Pho	Pci	Pco	Tci	Tco	PS_h	PS_c	Patm
		Filled Manually										
Total/Average	100	57,81	47,05	1,22	1,18	1,90	1,72	20,19	22,87	1,5	5,922	891
Steady State Check Values		57,77595	46,9994	1,21825	1,1853	1,90025	1,7235	20,177	22,8561			
STEADY STATE CHECK	2022-10-18T01:05:04+02:00	57,748	46,981	1,217	1,186	1,898	1,72	20,178	22,854	1,5	5,922	891
	2022-10-18T01:05:07+02:00	57,752	46,979	1,217	1,185	1,899	1,722	20,178	22,854	1,5	5,922	891
	2022-10-18T01:05:10+02:00	57,751	46,979	1,217	1,185	1,9	1,722	20,178	22,854	1,5	5,922	891
	2022-10-18T01:05:13+02:00	57,748	46,982	1,217	1,185	1,9	1,723	20,177	22,852	1,5	5,922	891
	2022-10-18T01:05:16+02:00	57,75	46,981	1,217	1,185	1,902	1,725	20,178	22,856	1,5	5,922	891
	2022-10-18T01:05:19+02:00	57,748	46,98	1,219	1,184	1,901	1,723	20,177	22,851	1,5	5,922	891
	2022-10-18T01:05:22+02:00	57,75	46,982	1,22	1,186	1,901	1,725	20,174	22,857	1,5	5,922	891
	2022-10-18T01:05:25+02:00	57,751	46,985	1,217	1,185	1,901	1,726	20,175	22,858	1,5	5,922	891
	2022-10-18T01:05:28+02:00	57,756	46,986	1,217	1,184	1,901	1,726	20,175	22,856	1,5	5,922	891
	2022-10-18T01:05:31+02:00	57,756	46,992	1,216	1,185	1,9	1,725	20,173	22,852	1,5	5,922	891
	2022-10-18T01:05:34+02:00	57,765	46,996	1,218	1,186	1,899	1,725	20,176	22,857	1,5	5,922	891
	2022-10-18T01:05:37+02:00	57,77	47,001	1,218	1,186	1,901	1,726	20,178	22,856	1,5	5,922	891
	2022-10-18T01:05:40+02:00	57,778	47,005	1,218	1,185	1,901	1,725	20,179	22,857	1,5	5,922	891
	2022-10-18T01:05:43+02:00	57,788	47,008	1,218	1,185	1,901	1,725	20,178	22,856	1,5	5,922	891
	2022-10-18T01:05:46+02:00	57,799	47,011	1,218	1,185	1,9	1,724	20,178	22,856	1,5	5,922	891
	2022-10-18T01:05:49+02:00	57,808	47,016	1,218	1,185	1,9	1,724	20,177	22,859	1,5	5,922	891
	2022-10-18T01:05:52+02:00	57,817	47,023	1,218	1,186	1,899	1,718	20,18	22,858	1,5	5,922	891
	2022-10-18T01:05:55+02:00	57,824	47,029	1,217	1,185	1,901	1,723	20,178	22,859	1,5	5,922	891
	2022-10-18T01:05:58+02:00	57,827	47,034	1,22	1,184	1,9	1,722	20,177	22,86	1,5	5,922	891
	2022-10-18T01:06:01+02:00	57,833	47,038	1,228	1,189	1,9	1,721	20,178	22,86	1,5	5,922	891
	2022-10-18T01:06:04+02:00	57,839	47,042	1,218	1,184	1,9	1,723	20,177	22,859	1,5	5,922	891
	2022-10-18T01:06:07+02:00	57,844	47,047	1,217	1,184	1,899	1,72	20,177	22,861	1,5	5,922	891
	2022-10-18T01:06:10+02:00	57,85	47,058	1,216	1,184	1,899	1,721	20,175	22,862	1,5	5,922	891
	2022-10-18T01:06:13+02:00	57,854	47,064	1,217	1,183	1,9	1,722	20,178	22,862	1,5	5,922	891
	2022-10-18T01:06:16+02:00	57,865	47,071	1,217	1,184	1,899	1,722	20,176	22,864	1,5	5,922	891
	2022-10-18T01:06:19+02:00	57,87	47,075	1,216	1,184	1,901	1,723	20,173	22,861	1,5	5,922	891
	2022-10-18T01:06:22+02:00	57,873	47,078	1,217	1,184	1,9	1,721	20,175	22,861	1,5	5,922	891
	2022-10-18T01:06:25+02:00	57,877	47,079	1,216	1,183	1,901	1,724	20,17	22,859	1,5	5,922	891
	2022-10-18T01:06:28+02:00	57,875	47,078	1,216	1,183	1,899	1,721	20,168	22,86	1,5	5,922	891
	2022-10-18T01:06:31+02:00	57,871	47,076	1,217	1,183	1,9	1,722	20,171	22,856	1,5	5,922	891
	2022-10-18T01:06:34+02:00	57,86	47,073	1,216	1,183	1,9	1,723	20,171	22,854	1,5	5,922	891
	2022-10-18T01:06:37+02:00	57,845	47,066	1,216	1,182	1,9	1,722	20,168	22,852	1,5	5,922	891
	2022-10-18T01:06:40+02:00	57,832	47,062	1,216	1,183	1,901	1,723	20,172	22,85	1,5	5,922	891
	2022-10-18T01:06:43+02:00	57,818	47,057	1,216	1,183	1,9	1,723	20,172	22,853	1,5	5,922	891
	2022-10-18T01:06:46+02:00	57,803	47,046	1,219	1,184	1,9	1,723	20,168	22,852	1,5	5,922	891
	2022-10-18T01:06:49+02:00	57,793	47,041	1,217	1,184	1,901	1,723	20,173	22,853	1,5	5,922	891
	2022-10-18T01:06:52+02:00	57,776	47,032	1,216	1,184	1,901	1,722	20,171	22,85	1,5	5,922	891
	2022-10-18T01:06:55+02:00	57,758	47,024	1,216	1,184	1,9	1,722	20,17	22,852	1,5	5,922	891
	2022-10-18T01:06:58+02:00	57,743	47,016	1,218	1,185	1,901	1,724	20,174	22,854	1,5	5,922	891
	2022-10-18T01:07:01+02:00	57,728	47,006	1,216	1,184	1,902	1,724	20,173	22,853	1,5	5,922	891
	2022-10-18T01:07:04+02:00	57,721	46,995	1,217	1,185	1,902	1,723	20,175	22,853	1,5	5,922	891
	2022-10-18T01:07:07+02:00	57,714	46,989	1,217	1,184	1,901	1,723	20,18	22,854	1,5	5,922	891
	2022-10-18T01:07:10+02:00	57,702	46,984	1,217	1,184	1,9	1,723	20,181	22,857	1,5	5,922	891
	2022-10-18T01:07:13+02:00	57,687	46,977	1,217	1,183	1,899	1,723	20,184	22,858	1,5	5,922	891
	2022-10-18T01:07:16+02:00	57,673	46,971	1,217	1,184	1,9	1,723	20,188	22,862	1,5	5,922	891
	2022-10-18T01:07:19+02:00	57,66	46,965	1,217	1,184	1,899	1,722	20,191	22,863	1,5	5,922	891
	2022-10-18T01:07:22+02:00	57,647	46,956	1,217	1,183	1,898	1,72	20,194	22,865	1,5	5,922	891
	2022-10-18T01:07:25+02:00	57,639	46,949	1,217	1,183	1,9	1,72	20,197	22,868	1,5	5,922	891
	2022-10-18T01:07:28+02:00	57,632	46,942	1,217	1,183	1,901	1,724	20,201	22,869	1,5	5,922	891
	2022-10-18T01:07:31+02:00	57,627	46,937	1,218	1,184	1,902	1,725	20,201	22,875	1,5	5,922	891
	2022-10-18T01:07:34+02:00	57,622	46,934	1,217	1,184	1,901	1,725	20,203	22,874	1,5	5,922	891
	2022-10-18T01:07:37+02:00	57,622	46,927	1,218	1,184	1,902	1,726	20,208	22,877	1,5	5,922	891
	2022-10-18T01:07:40+02:00	57,62	46,924	1,217	1,184	1,9	1,724	20,21	22,878	1,5	5,922	891
	2022-10-18T01:07:43+02:00	57,624	46,923	1,218	1,184	1,9	1,724	20,212	22,881	1,5	5,922	891
	2022-10-18T01:07:46+02:00	57,629	46,923	1,218	1,183	1,898	1,721	20,215	22,882	1,5	5,922	891
	2022-10-18T01:07:49+02:00	57,637	46,925	1,226	1,189	1,899	1,72	20,217	22,884	1,5	5,922	891
	2022-10-18T01:07:52+02:00	57,647	46,928	1,232	1,19	1,899	1,721	20,217	22,884	1,5	5,922	891

Figure B.1: Excel test sheet example

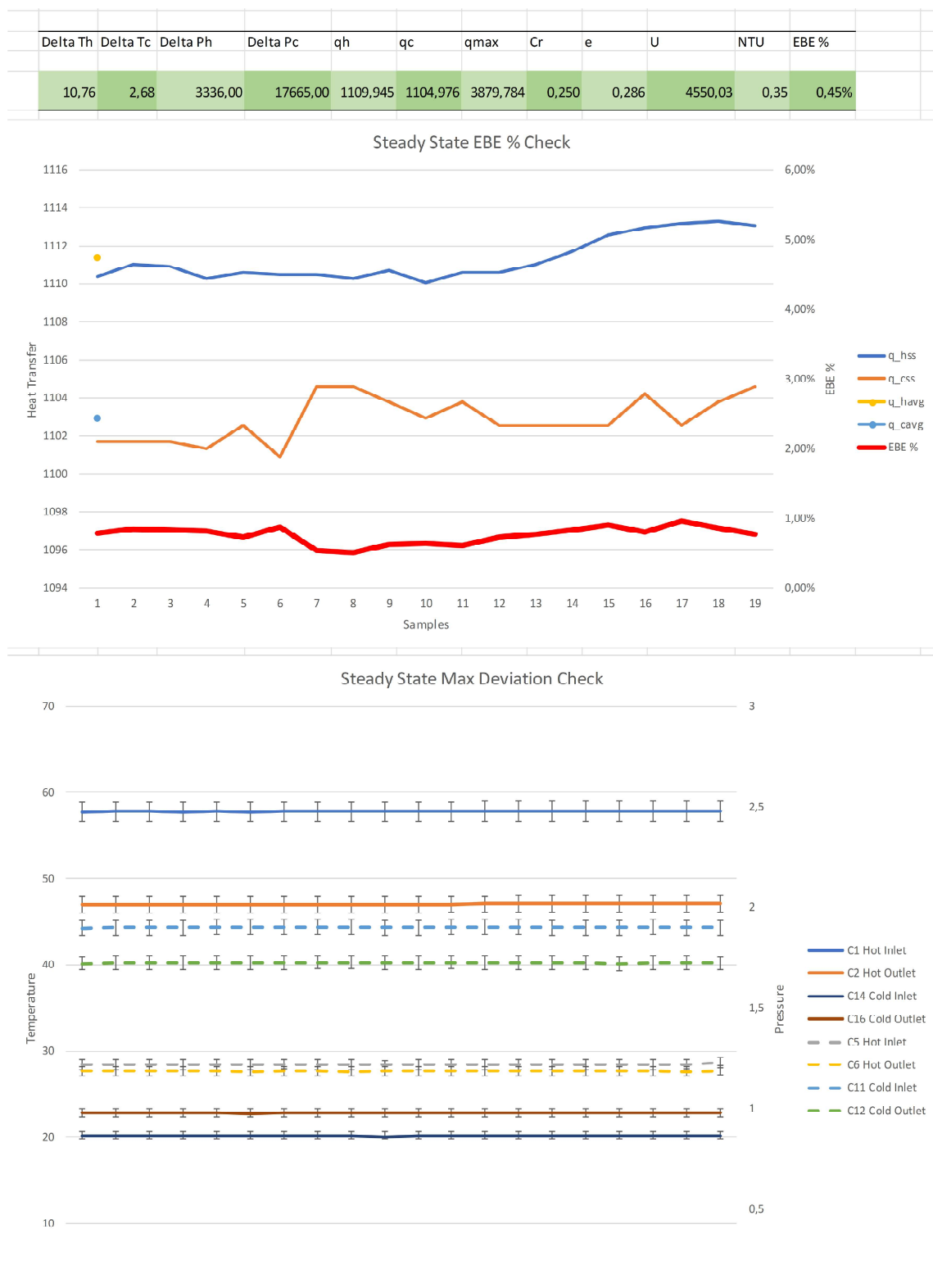


Figure B.2: Excel test sheet example, steady state check graphs



Hot Side							
Recorded Values				Calculated			
	Temperature	System Pressure	Flow Rate	Density (Rho)	Specific Heat Capacity (cp)	Mass Flow	Heat Transfer (q)
	Kelvin	Pa	l/s	kg/m ³	kJ/kg K	kg/s	kW
Inlet	330,96	32657,00	0,0250	984,29	4,18	0,025	-
Outlet	320,20	29321,00	0,0250	989,32	4,18	0,025	-
D	10,76	3336,00	Average	986,81	4,18	0,02467	1,10994 kW
Thi-Tho		Phi-Pho					1109,94452 W
Cold Side							
Recorded Values				Calculated			
	Temperature	System Pressure	Flow Rate	Density (Rho)	Specific Heat Capacity (cp)	Mass Flow	Heat Transfer (q)
	Kelvin	Pa	l/s	kg/m ³	kJ/kg K	kg/s	kW
Inlet	293,34	100708,00	0,099	998,17	4,18	0,099	-
Outlet	296,02	83043,00	0,099	997,56	4,18	0,098	-
D	2,68	17665,00	Average	997,86	4,18	0,09849	1,10498 kW
Tco-Tci		Pci-Pco					1104,97552 W
Capacity Rates							
C _h	0,103 kW/°C			C _{max}	0,412 kW/°C		
C _c	0,412 kW/°C				412,0644417		
Cr	0,250	Capacity Ratio		C _{min}	0,103 kW/°C		
					103,1364829		
Maximum Heat Transfer							
					3,87978 kW		
					3879,78409 W		
Effectiveness							
	1			2			
HOT	e=	0,286		Manually done	0,286		
Cold	e=	0,285					
Overall Heat Transfer Coefficient							
				Thi		Tho	
				330,96		320,20	
ΔT _m	30,72 °C			2		1	
				296,02		293,34	
				Tco		Tci	
Data from model							
Area	1323,5 mm ²	Per Row		12 Rows		6 Per Fluid	
	0,0013235 m ²					1 Heat transfer Area per row	
Total	0,007941 m ²	Per Fluid				6 Total	
U=q/ΔT _m							
U	4550,03 W/m ² K						
	4,55 kW/m ² K						
Number of Transfer Units							
NTU =	0,35						

Figure B.3: Excel calculation sheet example

Appendix C

Results summary

C.1 Experimental results

Table C.1: Summary of all test results for the 1 mm heat exchanger

C*	Flow Setting	Test Group Data		T _{hi} (°C)	T _{ho} (°C)	P _{hi} bar (abs)	P _{ho} bar (abs)	P _d bar (abs)	P _{co} bar (abs)	T _{ci} (°C)	T _{co} (°C)	P _{sh} LPM	P _{sc} LPM	P _{am} Bar	ΔT _h (°C)	ΔT _c (°C)	ΔP _h Pa	ΔP _c Pa	q _h W	q _c W	Q _{max} W	ε %	U W/m ² K	NTU	q _h vs q _c %
		Test No.																							
1	1	1	57.42	47.71	1.30	1.26	1.27	1.26	1.26	20.52	30.26	1	1.0	870	9.71	9.74	3934.00	316.00	667.58	668.18	2530.74	26.38%	879.75	0.358	0.09%
		2	57.78	47.95	1.30	1.26	1.26	1.25	1.25	20.46	30.31	1	1.0	870	9.83	9.85	4252.00	1133.00	675.68	675.48	2559.78	26.40%	3095.68	0.358	0.03%
		3	57.79	48.00	1.30	1.26	1.27	1.26	1.26	20.45	30.22	1	1.0	870	9.80	9.77	3981.00	461.00	673.38	669.89	2566.74	26.23%	3077.32	0.355	0.52%
		4	58.00	48.13	1.30	1.26	1.26	1.26	1.26	20.43	30.24	1	1.0	870	9.87	9.81	4042.00	101.00	678.67	672.67	2582.55	26.28%	3082.10	0.356	0.89%
		5	57.91	48.01	1.30	1.26	1.26	1.25	1.25	20.39	30.21	1	1.0	870	9.90	9.82	3965.00	916.00	680.89	673.53	2579.52	26.40%	3099.77	0.358	1.09%
	Average	57.78	47.96	1.30	1.26	1.26	1.26	1.26	1.26	20.45	30.25	1	1.0	870	9.82	9.80	4034.80	585.40	675.24	671.95	2563.87	26.34%	2646.92	0.357	0.52%
1	2	1	57.63	49.67	1.27	1.23	1.67	1.65	1.65	20.42	28.44	1.5	1.5	870	7.96	8.01	3900.00	1795.00	820.37	824.36	3828.88	21.43%	3535.16	0.273	0.49%
		2	57.66	49.68	1.27	1.23	1.67	1.65	1.65	20.38	28.38	1.5	1.5	870	7.98	8.01	3991.00	1300.00	822.43	823.78	3836.16	21.44%	3536.43	0.273	0.16%
		3	57.76	49.77	1.27	1.23	1.67	1.66	1.66	20.35	28.36	1.5	1.5	870	7.99	8.02	4260.00	1543.00	823.12	824.84	3849.73	21.38%	3524.60	0.272	0.21%
		4	57.86	49.84	1.27	1.23	1.67	1.66	1.66	20.35	28.36	1.5	1.5	870	8.02	8.01	4456.00	1186.00	826.45	824.77	3866.71	21.37%	3528.30	0.272	0.20%
		5	57.87	49.80	1.27	1.23	1.67	1.65	1.65	20.33	28.35	1.5	1.5	870	8.07	8.02	4570.00	1988.00	832.06	825.42	3870.09	21.50%	3551.99	0.274	0.80%
	Average	57.76	49.75	1.27	1.23	1.67	1.65	1.65	20.36	28.38	1.5	1.5	870	8.00	8.01	4235.40	1562.40	824.88	824.64	3850.31	21.42%	3535.30	0.273	0.37%	
1	3	1	57.87	50.97	1.30	1.24	1.48	1.45	1.45	20.27	27.29	2	2.0	870	6.91	7.03	6245.00	2244.00	949.29	964.18	5161.55	18.39%	3901.28	0.226	1.56%
		2	57.80	50.88	1.30	1.23	1.48	1.45	1.45	20.20	27.14	2	2.0	870	6.92	6.94	6701.00	2931.00	950.67	952.51	5160.59	18.42%	3903.21	0.226	0.19%
		3	57.78	50.86	1.30	1.23	1.48	1.45	1.45	20.17	27.14	2	2.0	870	6.92	6.96	6758.00	2495.00	950.96	955.80	5161.58	18.42%	3905.17	0.226	0.51%
		4	57.90	50.95	1.28	1.22	1.48	1.45	1.45	20.15	27.14	2	2.0	870	6.95	6.99	5875.00	2859.00	954.91	959.59	5181.22	18.43%	3906.81	0.226	0.49%
		5	57.61	50.69	1.27	1.21	1.48	1.45	1.45	20.14	27.07	2	2.0	870	6.93	6.94	6216.00	2589.00	951.66	952.16	5143.80	18.50%	3923.29	0.227	0.05%
	Average	57.79	50.87	1.29	1.23	1.48	1.45	1.45	20.18	27.16	2	2.0	870	6.92	6.97	6359.00	2623.60	951.50	956.85	5161.75	18.43%	3907.95	0.226	0.56%	
0.75	1	1	57.77	47.02	1.24	1.21	1.17	1.17	1.17	20.10	28.05	1	1.3	873.75	10.75	7.94	3538.00	89.00	738.96	726.81	2589.81	28.53%	3288.64	0.380	1.66%
		2	57.78	47.13	1.24	1.21	1.16	1.16	1.16	20.07	28.06	1	1.3	873.75	10.64	8.00	3519.00	319.00	731.82	731.61	2592.89	28.22%	3248.39	0.375	0.03%
		3	57.84	47.09	1.24	1.21	1.15	1.15	1.15	20.06	28.14	1	1.3	873.75	10.74	8.08	3601.00	222.00	738.63	739.20	2597.26	28.44%	3281.80	0.379	0.08%
		4	57.93	47.14	1.25	1.21	1.15	1.15	1.15	20.04	28.14	1	1.3	873.75	10.79	8.10	3819.00	19.00	741.85	740.68	2605.22	28.48%	3286.28	0.380	0.16%
		5	57.87	47.08	1.24	1.21	1.15	1.15	1.15	20.04	28.10	1	1.3	873.75	10.79	8.06	3746.00	471.00	741.92	736.93	2600.98	28.52%	3291.58	0.380	0.67%
	Average	57.84	47.09	1.24	1.21	1.15	1.15	1.15	20.06	28.10	1	1.3	873.75	10.74	8.03	3644.60	224.00	738.63	735.05	2597.23	28.44%	3279.34	0.379	0.52%	
0.75	2	1	57.93	49.27	1.27	1.23	1.70	1.67	1.67	20.19	26.71	1.5	2.0	873.75	8.66	6.52	3880.00	3236.00	892.81	895.07	3890.87	22.95%	3730.28	0.287	0.25%
		2	57.93	49.27	1.27	1.23	1.70	1.67	1.67	20.17	26.67	1.5	2.0	873.75	8.66	6.50	4038.00	3263.00	892.71	892.46	3892.94	22.93%	3726.15	0.287	0.03%
		3	57.93	49.23	1.27	1.23	1.70	1.67	1.67	20.15	26.65	1.5	2.0	873.75	8.70	6.50	3772.00	2756.00	897.03	891.83	3894.58	23.03%	3744.63	0.288	0.58%
		4	57.94	49.26	1.27	1.23	1.69	1.67	1.67	20.14	26.65	1.5	2.0	873.75	8.68	6.51	3783.00	1929.00	894.37	894.12	3896.84	22.95%	3730.17	0.287	0.03%
		5	57.93	49.22	1.27	1.23	1.69	1.67	1.67	20.13	26.63	1.5	2.0	873.75	8.70	6.50	3984.00	2400.00	897.11	892.08	3896.21	23.03%	3743.17	0.288	0.56%
	Average	57.93	49.25	1.27	1.23	1.70	1.67	1.67	20.16	26.66	1.5	2.0	873.75	8.68	6.51	3891.40	2716.80	894.80	893.11	3894.29	22.98%	3734.88	0.288	0.29%	
0.75	3	1	57.93	50.53	1.22	1.17	1.84	1.80	1.80	20.30	25.88	2	2.6	873.75	7.40	5.58	5030.00	4112.00	1017.09	1020.93	5170.56	19.67%	4114.56	0.238	0.38%
		2	57.91	50.52	1.22	1.17	1.85	1.81	1.81	20.24	25.81	2	2.6	873.75	7.39	5.57	5587.00	4017.00	1015.66	1019.80	5176.41	19.62%	4102.08	0.237	0.41%
		3	57.90	50.52	1.22	1.17	1.86	1.81	1.86	20.22	25.78	2	2.6	873.75	7.38	5.56	5683.00	4597.00	1014.72	1018.43	5178.70	19.59%	4095.13	0.237	0.37%
		4	57.88	50.49	1.22	1.16	1.85	1.81	1.85	20.22	25.77	2	2.6	873.75	7.40	5.55	5903.00	4014.00	1016.54	1016.10	5175.27	19.64%	4105.89	0.237	0.04%
		5	57.91	50.50	1.22	1.17	1.85	1.81	1.85	20.15	25.71	2	2.6	873.75	7.40	5.56	5800.00	4326.00	1017.36	1018.59	5188.64	19.61%	4097.64	0.237	0.12%
	Average	57.91	50.51	1.22	1.17	1.85	1.81	1.81	20.23	25.79	2	2.6	873.75	7.40	5.57	5600.60	4213.20	1016.27	1018.77	5177.91	19.63%	4103.06	0.237	0.26%	

Table C.1: Summary of all test results for the 1 mm heat exchanger (continued)

Test Group Data		T _{hi}	T _{ho}	P _{hi}	P _{ho}	P _{ci}	P _{co}	T _{ci}	T _{co}	PS _h	PS _c	P _{ann}	ΔT _h	ΔP _h	ΔP _c	q _h	q _c	Q _{max}	ε	U	q _h vs q _c		
C*	Flow Setting	Test No.	(°C)	bar (abs)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	Pa	Pa	W	W	W	%	W/m ² K	NTU	%	
0,5	1	1	57,55	46,10	1,27	1,24	1,41	20,23	25,97	1	2,0	873,75	11,45	5,74	2923,00	853,00	787,43	787,81	2566,68	30,68%	3463,33	0,400	0,05%
		2	57,87	46,35	1,27	1,24	1,42	20,17	25,95	1	2,0	873,75	11,53	5,78	3180,00	1516,00	792,60	792,98	2592,84	30,57%	3446,67	0,398	0,05%
		3	57,93	46,37	1,27	1,23	1,42	20,17	25,94	1	2,0	873,75	11,56	5,77	3881,00	1231,00	794,63	791,81	2596,19	30,61%	3451,00	0,399	0,36%
		4	57,81	46,27	1,27	1,24	1,40	20,15	25,93	1	2,0	873,75	11,54	5,77	3476,00	872,00	793,51	792,31	2589,80	30,64%	3456,59	0,399	0,15%
		5	57,74	46,18	1,27	1,23	1,40	20,15	25,92	1	2,0	873,75	11,56	5,77	3171,00	2292,00	795,16	791,62	2585,38	30,76%	3472,91	0,401	0,45%
Average		57,78	46,26	1,27	1,24	1,41	20,17	25,94	1	2,0	873,75	11,53	5,76	3326,20	1352,80	792,67	791,31	2586,18	30,65%	3458,10	0,399	0,21%	
0,5	2	1	57,67	48,31	1,26	1,21	1,64	20,41	25,08	1,5	3,0	873,75	9,36	4,67	4088,00	5200,00	965,26	961,04	3842,16	25,12%	4026,71	0,310	0,44%
		2	57,87	48,49	1,26	1,21	1,64	20,36	25,06	1,5	3,0	873,75	9,38	4,70	4631,00	4829,00	966,97	967,01	3867,36	25,00%	4004,02	0,308	0,00%
		3	57,86	48,54	1,26	1,21	1,65	20,34	25,03	1,5	3,0	873,75	9,32	4,69	5183,00	5431,00	961,37	966,13	3868,64	24,85%	3975,15	0,306	0,49%
		4	57,82	48,49	1,26	1,21	1,65	20,33	24,99	1,5	3,0	873,75	9,33	4,66	5325,00	4910,00	961,75	960,28	3865,48	24,88%	3979,28	0,306	0,15%
		5	57,88	48,54	1,25	1,21	1,64	20,33	25,01	1,5	3,0	873,75	9,34	4,68	4207,00	5467,00	962,97	964,82	3871,82	24,87%	3978,39	0,306	0,19%
Average		57,82	48,48	1,26	1,21	1,64	20,35	25,03	1,5	3,0	873,75	9,35	4,68	4686,80	5167,40	963,66	963,86	3863,09	24,95%	3992,71	0,308	0,26%	
0,5	3	1	57,97	49,84	1,23	1,18	1,91	20,30	24,37	2	3,9	891	8,14	4,07	4729,00	9113,00	1118,53	1116,59	5177,85	21,60%	4467,45	0,258	0,17%
		2	57,76	49,68	1,25	1,18	1,92	20,31	24,33	2	3,9	891	8,07	4,02	6354,00	9353,00	1109,44	1103,94	5146,23	21,56%	4456,01	0,257	0,50%
		3	57,67	49,61	1,24	1,18	1,92	20,26	24,28	2	3,9	891	8,06	4,01	5868,00	9348,00	1107,43	1102,30	5141,56	21,54%	4451,47	0,257	0,46%
		4	57,64	49,59	1,25	1,18	1,92	20,22	24,23	2	3,9	891	8,05	4,01	6448,00	9234,00	1106,08	1100,99	5142,72	21,51%	4443,81	0,257	0,46%
		5	57,88	49,77	1,24	1,18	1,90	20,21	24,26	2	3,9	891	8,11	4,05	6234,00	9455,00	1114,70	1111,72	5177,14	21,53%	2427,14	0,257	0,27%
Average		57,78	49,70	1,24	1,18	1,91	20,26	24,29	2	3,9	891	8,09	4,03	5926,60	9300,60	1111,24	1107,11	5157,10	21,55%	4049,18	0,257	0,37%	
0,25	1	1	57,76	44,77	1,28	1,24	1,57	20,43	23,66	1	3,9	891	12,98	3,23	3274,00	6734,00	893,13	886,50	2567,62	34,78%	3885,48	0,449	0,75%
		2	57,87	44,82	1,27	1,24	1,57	20,41	23,65	1	3,9	891	13,06	3,24	3415,00	8437,00	898,08	890,69	2577,23	34,85%	3894,23	0,450	0,83%
		3	57,87	44,79	1,27	1,24	1,56	20,39	23,65	1	3,9	891	13,07	3,26	3366,00	7568,00	899,15	894,19	2577,72	34,88%	3899,84	0,450	0,55%
		4	57,84	44,75	1,27	1,24	1,54	20,36	23,62	1	3,9	891	13,09	3,27	3359,00	6747,00	900,73	897,62	2578,46	34,93%	3907,83	0,451	0,35%
		5	57,87	44,77	1,27	1,24	1,54	20,31	23,59	1	3,9	891	13,10	3,29	3348,00	6571,00	901,07	902,35	2583,56	34,88%	3900,60	0,450	0,14%
Average		57,84	44,78	1,27	1,24	1,56	20,38	23,63	1	3,9	891	13,06	3,26	3352,40	7211,40	898,43	894,27	2576,92	34,86%	3897,59	0,450	0,52%	
0,25	2	1	57,89	47,15	1,22	1,18	1,90	20,40	23,08	1,5	5,9	891	10,74	2,68	3484,00	17944,00	1107,26	1106,15	3866,48	28,64%	4556,15	0,351	0,10%
		2	57,77	47,04	1,22	1,18	1,90	20,29	22,97	1,5	5,9	891	10,73	2,68	3490,00	17803,00	1106,78	1104,55	3866,17	28,63%	4554,22	0,351	0,20%
		3	57,70	47,01	1,22	1,18	1,90	20,19	22,91	1,5	5,9	891	10,69	2,69	3383,00	18389,00	1102,64	1107,20	3865,22	28,53%	4535,84	0,349	0,41%
		4	57,64	46,94	1,22	1,18	1,90	20,19	22,86	1,5	5,9	891	10,70	2,68	3367,00	17706,00	1103,57	1102,91	3862,81	28,57%	4543,30	0,350	0,06%
		5	57,81	47,05	1,22	1,18	1,90	20,19	22,87	1,5	5,9	891	10,76	2,68	3336,00	17665,00	1109,94	1104,98	3879,78	28,61%	4550,03	0,350	0,45%
Average		57,76	47,04	1,22	1,18	1,90	20,26	22,94	1,5	5,9	891	10,72	2,68	3412,00	17901,40	1106,04	1105,16	3868,09	28,59%	4547,91	0,350	0,24%	
0,25	3	1	57,77	48,39	1,22	1,18	2,15	20,15	22,50	2	7,9	891	9,38	2,35	4019,00	36257,00	1289,86	1293,49	5172,63	24,94%	5135,89	0,297	0,28%
		2	57,74	48,36	1,23	1,18	2,15	20,07	22,41	2	7,9	891	9,38	2,35	4704,95	36159,41	1288,94	1291,23	5179,45	24,89%	5123,29	0,296	0,18%
		3	57,80	48,39	1,22	1,18	2,15	20,05	22,40	2	7,9	891	9,41	2,35	4763,37	36438,61	1293,20	1290,15	5189,50	24,92%	5130,82	0,296	0,24%
		4	57,84	48,41	1,22	1,17	2,15	20,01	22,36	2	7,9	891	9,44	2,35	4416,83	36426,73	1297,56	1289,60	5200,74	24,95%	5137,44	0,297	0,62%
		5	57,84	48,39	1,22	1,17	2,15	19,97	22,32	2	7,9	891	9,45	2,34	4935,00	36491,00	1298,72	1288,43	5205,60	24,95%	5136,88	0,297	0,80%
Average		57,80	48,39	1,22	1,18	2,15	20,05	22,40	2	7,9	891	9,41	2,35	4567,83	36354,55	1293,66	1290,58	5189,59	24,93%	5132,86	0,296	0,42%	

Table C.2.: Summary of all test results for the 2 mm heat exchanger

C*	Flow Setting	Test Group Data		T _{hi}	T _{no}	P _{hi}	P _{ho}	P _{ci}	P _{co}	T _{ci}	T _{co}	PS _h	PS _c	P _{atm}	ΔT _h	ΔT _c	ΔP _h	ΔP _c	q _h	q _c	q _{max}	ε	U	NTU	q _h vs q _c
		Test No.	(°C)	(°C)	bar (abs)	bar (abs)	bar (abs)	bar (abs)	(abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	W/m ² K	%	%
1	1	1	57.86	50.14	1.29	1.25	1.58	1.58	1.58	20.47	28.19	1	1.0	892.5	7.72	7.71	4050.00	366.00	530.42	529.29	2568.96	20.65%	314.33	0.260	0.21%
		2	57.92	50.09	1.30	1.26	1.57	1.57	1.57	20.43	28.19	1	1.0	892.5	7.83	7.76	3933.00	163.00	538.07	532.25	2576.19	20.89%	165.66	0.264	1.09%
		3	57.81	49.99	1.30	1.26	1.57	1.57	1.57	20.43	28.23	1	1.0	892.5	7.82	7.80	3927.00	311.00	537.15	535.12	2568.80	20.91%	319.35	0.264	0.38%
		4	57.96	50.07	1.31	1.27	1.57	1.57	1.57	20.39	28.26	1	1.0	892.5	7.89	7.88	4090.00	385.00	542.04	540.29	2581.59	21.00%	321.02	0.266	0.32%
		5	57.84	49.97	1.31	1.27	1.57	1.56	1.56	20.37	28.27	1	1.0	892.5	7.88	7.90	4102.00	265.00	541.31	542.15	2570.86	21.06%	321.74	0.267	0.15%
Average		57.88	50.05	1.30	1.26	1.57	1.57	1.57	20.42	28.23	1	1.0	892.5	7.83	7.81	4020.40	298.00	537.80	535.82	2573.28	20.90%	288.42	0.264	0.43%	
1	2	1	57.85	52.56	1.20	1.16	1.37	1.37	1.37	20.12	25.44	1.5	1.5	866.25	5.29	5.31	3986.00	325.00	545.43	547.28	3885.20	14.04%	295.75	0.163	0.34%
		2	57.96	52.61	1.20	1.16	1.38	1.38	1.38	20.11	25.38	1.5	1.5	866.25	5.36	5.26	4280.00	97.00	551.66	541.96	3899.40	14.15%	298.06	0.165	1.77%
		3	57.90	52.54	1.21	1.17	1.37	1.37	1.37	20.10	25.32	1.5	1.5	866.25	5.36	5.21	4202.00	217.00	552.57	536.95	3894.31	14.19%	298.83	0.165	2.87%
		4	57.76	52.50	1.22	1.19	1.37	1.37	1.37	20.09	25.29	1.5	1.5	866.25	5.26	5.21	3707.00	191.00	542.31	536.05	3881.27	13.97%	293.94	0.162	1.16%
		5	58.03	52.76	1.22	1.18	1.37	1.37	1.37	20.07	25.29	1.5	1.5	866.25	5.27	5.22	3764.00	111.00	542.66	537.41	3910.15	13.88%	291.65	0.161	0.97%
Average		57.90	52.59	1.21	1.17	1.37	1.37	1.37	20.10	25.34	1.5	1.5	866.25	5.31	5.24	3987.80	188.20	546.92	539.93	3894.07	14.05%	295.65	0.163	1.42%	
1	3	1	57.65	52.12	1.25	1.21	1.92	1.90	21.60	27.07	2	2.0	883.5	5.53	5.47	4349.00	1561.00	759.49	751.09	4952.01	15.34%	437.18	0.181	1.11%	
		2	57.72	52.16	1.24	1.20	1.90	1.89	21.60	27.14	2	2.0	883.5	5.55	5.54	4254.00	1159.00	762.80	760.72	4961.59	15.37%	438.75	0.182	0.27%	
		3	57.70	52.14	1.26	1.22	1.89	1.88	21.64	27.24	2	2.0	883.5	5.57	5.60	4129.00	1284.00	764.60	768.46	4948.04	15.45%	441.07	0.183	0.50%	
		4	57.77	52.12	1.26	1.22	1.90	1.89	21.60	27.23	2	2.0	883.5	5.65	5.63	4081.00	1157.00	776.04	773.02	4969.18	15.62%	446.91	0.185	0.39%	
		5	57.97	52.49	1.38	1.33	1.78	1.76	21.49	26.93	2	2.0	883.5	5.48	5.44	5135.00	1781.00	752.65	746.03	5010.68	15.02%	426.60	0.177	0.88%	
Average		57.76	52.21	1.28	1.23	1.88	1.86	21.59	27.12	2	2.0	883.5	5.56	5.54	4389.60	1388.40	763.11	759.86	4968.30	15.36%	438.11	0.181	0.63%		
0.75	1	1	57.94	51.42	1.18	1.14	1.17	1.17	21.90	26.82	1	1.3	883.5	6.51	4.92	3820.00	651.00	447.35	450.26	2475.83	18.07%	259.45	0.215	0.65%	
		2	58.16	51.68	1.17	1.13	1.17	1.16	21.86	26.77	1	1.3	883.5	6.48	4.91	3750.00	1107.00	445.09	449.31	2493.74	17.85%	255.73	0.212	0.94%	
		3	58.03	51.60	1.18	1.14	1.16	1.16	21.79	26.65	1	1.3	883.5	6.43	4.85	3245.00	600.00	441.65	443.88	2489.27	17.74%	253.84	0.210	0.50%	
		4	58.10	51.72	1.18	1.14	1.17	1.17	21.72	26.50	1	1.3	883.5	6.39	4.77	3340.00	111.00	438.74	436.79	2498.98	17.56%	250.52	0.207	0.45%	
		5	58.04	51.71	1.18	1.15	1.17	1.17	21.69	26.42	1	1.3	883.5	6.33	4.73	3194.00	131.00	434.96	432.60	2496.46	17.42%	248.25	0.206	0.54%	
Average		58.05	51.63	1.18	1.14	1.17	1.17	21.79	26.63	1	1.3	883.5	6.43	4.84	3469.80	520.00	441.56	442.57	2490.85	17.73%	253.56	0.210	0.62%		
0.75	2	1	57.90	51.03	1.21	1.16	1.82	1.81	20.53	25.75	1.5	2.0	879.75	6.87	5.22	4197.00	962.00	707.69	716.22	3850.82	18.38%	397.31	0.219	1.20%	
		2	57.85	50.92	1.21	1.16	1.82	1.80	20.51	25.71	1.5	2.0	879.75	6.93	5.20	4402.00	1271.00	714.43	714.35	3848.41	18.56%	401.75	0.222	0.01%	
		3	57.75	50.83	1.21	1.17	1.81	1.79	20.49	25.71	1.5	2.0	879.75	6.92	5.22	3891.00	1651.00	712.97	717.17	3840.38	18.57%	401.98	0.222	0.59%	
		4	57.76	50.81	1.20	1.16	1.80	1.79	20.46	25.70	1.5	2.0	879.75	6.95	5.24	4564.00	1161.00	716.10	719.11	3844.16	18.63%	403.57	0.223	0.42%	
		5	57.87	50.81	1.20	1.16	1.82	1.81	20.44	25.72	1.5	2.0	879.75	7.06	5.28	4558.00	1054.00	727.97	724.58	3857.34	18.87%	409.59	0.226	0.47%	
Average		57.83	50.88	1.21	1.16	1.81	1.80	20.49	25.72	1.5	2.0	879.75	6.95	5.23	4322.40	1219.80	715.83	718.29	3848.22	18.60%	402.84	0.222	0.54%		
0.75	3	1	57.80	52.30	1.15	1.10	1.39	1.37	20.21	24.31	2	2.6	879.75	5.50	4.10	4926.00	2303.00	755.78	750.37	5163.18	14.64%	405.37	0.168	0.72%	
		2	57.74	52.26	1.15	1.10	1.38	1.36	20.18	24.26	2	2.6	879.75	5.49	4.09	5035.00	2439.00	753.73	748.45	5160.55	14.61%	404.37	0.167	0.70%	
		3	57.80	52.29	1.18	1.13	1.38	1.36	20.15	24.26	2	2.6	879.75	5.51	4.11	5077.00	1931.00	752.54	752.54	5171.67	14.64%	405.54	0.168	0.62%	
		4	57.72	52.24	1.18	1.13	1.37	1.36	20.14	24.23	2	2.6	879.75	5.48	4.09	4311.00	1399.00	752.19	748.32	5162.02	14.57%	403.33	0.167	0.52%	
		5	57.79	52.33	1.17	1.13	1.37	1.35	20.14	24.22	2	2.6	879.75	5.46	4.08	4513.00	1583.00	750.01	746.26	5171.79	14.50%	401.11	0.166	0.50%	
Average		57.77	52.28	1.16	1.12	1.38	1.36	20.17	24.26	2	2.6	879.75	5.49	4.09	4772.40	1931.00	753.79	749.19	5165.84	14.59%	403.94	0.167	0.61%		

Table C.2: Summary of all test results for the 2 mm heat exchanger (continued)

Test Group Data	T _{hi}	T _{ho}	P _{hi}	P _{ho}	P _d	P _{co}	T _{ci}	T _{co}	PS _h	PS _c	P _{atm}	ΔT _h	ΔT _c	ΔP _h	ΔP _c	q _h	q _c	Q _{max}	ε	U	NTU	q _h vs q _c
C* Flow Setting	Test No.	(°C)	(°C)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	W/m ² K		%
0.5	1	58.00	48.91	1.24	1.21	1.54	20.28	24.81	1	2.0	883.5	9.09	4.53	2178.00	775.00	624.43	622.32	2592.31	24.09%	355.86	0.294	0.34%
	2	57.85	48.84	1.27	1.24	1.51	20.23	24.76	1	2.0	883.5	9.01	4.53	3471.00	844.00	619.14	621.67	2585.44	23.95%	353.51	0.293	0.41%
	3	58.02	49.02	1.23	1.20	1.51	20.20	24.70	1	2.0	883.5	8.99	4.50	2325.00	983.00	618.11	618.08	2599.02	23.78%	350.41	0.290	0.00%
	4	57.91	48.87	1.24	1.22	1.51	20.21	24.70	1	2.0	883.5	9.04	4.49	2246.00	1383.00	621.21	616.47	2591.22	23.97%	353.70	0.293	0.77%
	5	57.85	49.05	1.24	1.22	1.47	20.18	24.69	1	2.0	883.5	8.80	4.51	2736.00	1054.00	604.56	618.74	2588.44	23.36%	343.33	0.284	2.32%
Average		57.92	48.94	1.25	1.22	1.51	20.22	24.73	1	2.0	883.5	8.98	4.51	2591.20	1007.80	617.49	619.46	2591.29	23.83%	351.36	0.291	0.77%
0.5	1	57.74	50.15	1.25	1.21	1.57	20.36	24.24	1.5	3.0	883.5	7.59	3.88	3803.00	2970.00	781.96	800.00	3853.09	20.29%	434.95	0.240	2.28%
	2	57.79	50.14	1.24	1.19	1.57	20.31	24.19	1.5	3.0	883.5	7.65	3.88	4371.00	3095.00	788.52	798.73	3852.72	20.41%	437.71	0.242	1.29%
	3	57.85	50.11	1.24	1.20	1.57	20.29	24.17	1.5	3.0	883.5	7.74	3.88	3994.00	3281.00	797.52	799.51	3871.36	20.60%	442.19	0.244	0.25%
	4	57.86	50.10	1.23	1.19	1.56	20.25	24.14	1.5	3.0	883.5	7.76	3.89	4365.00	2752.00	800.01	801.93	3876.69	20.64%	443.10	0.245	0.24%
	5	57.84	50.03	1.22	1.18	1.56	20.24	24.14	1.5	3.0	883.5	7.81	3.90	4453.00	3035.00	805.37	803.44	3875.94	20.78%	446.62	0.246	0.24%
Average		57.82	50.11	1.24	1.20	1.56	20.29	24.18	1.5	3.0	883.5	7.71	3.89	4197.20	3026.60	794.67	800.72	3867.96	20.54%	440.91	0.243	0.86%
0.5	1	57.81	51.28	1.23	1.19	1.71	19.97	23.29	2	3.9	891	6.54	3.33	4107.00	4482.00	898.17	913.62	5199.89	17.27%	480.18	0.199	1.71%
	2	57.65	51.01	1.21	1.17	1.72	19.99	23.29	2	3.9	891	6.64	3.30	3961.00	4392.00	912.21	906.78	5174.54	17.63%	491.10	0.203	0.60%
	3	57.77	51.05	1.23	1.19	1.71	20.04	23.39	2	3.9	891	6.72	3.34	4328.00	4502.00	922.70	918.81	5183.51	17.80%	496.66	0.206	0.42%
	4	57.82	51.04	1.24	1.19	1.72	20.02	23.41	2	3.9	891	6.78	3.39	4272.00	4520.00	931.65	932.23	5194.72	17.93%	501.10	0.207	0.06%
	5	57.83	51.07	1.23	1.18	1.72	20.07	23.46	2	3.9	891	6.76	3.39	4719.00	4666.00	928.85	930.68	5187.66	17.91%	500.17	0.207	0.20%
Average		57.78	51.09	1.23	1.19	1.72	20.02	23.37	2	3.9	891	6.69	3.35	4277.40	4512.40	918.72	920.42	5188.06	17.71%	493.84	0.204	0.60%
0.25	1	57.50	46.21	1.28	1.24	1.54	20.25	23.12	1	3.9	891	11.30	2.86	3349.54	3158.37	776.86	786.72	2561.83	30.32%	455.72	0.377	1.26%
	2	57.96	46.47	1.29	1.25	1.57	20.38	23.26	1	3.9	891	11.48	2.88	3894.00	3532.00	789.58	791.65	2584.05	30.56%	459.82	0.380	0.26%
	3	57.95	46.35	1.29	1.25	1.57	20.21	23.12	1	3.9	891	11.59	2.91	3730.00	3465.00	797.15	798.79	2594.76	30.72%	462.94	0.383	0.21%
	4	57.77	46.12	1.29	1.25	1.58	20.17	23.08	1	3.9	891	11.65	2.91	3658.00	3419.00	801.30	800.05	2586.09	30.98%	467.90	0.387	0.16%
	5	57.66	45.94	1.28	1.25	1.58	20.20	23.10	1	3.9	891	11.72	2.90	3481.00	3978.00	806.01	796.32	2576.65	31.28%	473.35	0.391	1.21%
Average		57.77	46.22	1.28	1.25	1.57	20.24	23.13	1	3.9	891	11.55	2.89	3622.51	3510.47	794.18	794.71	2580.67	30.77%	463.95	0.384	0.62%
0.25	1	57.68	47.52	1.25	1.22	1.98	20.30	22.82	1.5	5.9	891	10.16	2.51	3638.00	10109.00	1048.28	1034.78	3854.55	27.20%	596.88	0.329	1.30%
	2	57.81	47.54	1.26	1.22	1.96	20.18	22.75	1.5	5.9	891	10.27	2.57	3997.00	10018.00	1059.10	1059.37	3881.37	27.29%	599.58	0.331	0.03%
	3	57.68	47.36	1.26	1.22	1.97	20.16	22.75	1.5	5.9	891	10.32	2.59	4579.00	10215.00	1064.57	1066.44	3869.12	27.51%	605.77	0.334	0.18%
	4	57.75	47.27	1.26	1.22	1.98	20.18	22.77	1.5	5.9	891	10.48	2.59	4338.00	10043.00	1081.09	1066.44	3874.55	27.90%	615.86	0.340	1.36%
	5	57.40	46.91	1.26	1.22	1.96	20.10	22.70	1.5	5.9	891	10.49	2.60	4026.00	10155.00	1081.62	1071.88	3846.79	28.12%	621.88	0.343	0.90%
Average		57.66	47.32	1.26	1.22	1.97	20.19	22.76	1.5	5.9	891	10.35	2.57	4115.60	10108.00	1066.93	1059.78	3865.28	27.60%	607.99	0.335	0.75%
0.25	1	57.58	48.67	1.25	1.21	1.95	20.11	22.32	2	7.9	891	8.90	2.22	4213.00	20016.00	1224.15	1220.75	5151.17	23.76%	677.08	0.280	0.28%
	2	57.38	48.50	1.25	1.21	1.95	20.15	22.32	2	7.9	891	8.88	2.22	4218.00	20148.00	1221.13	1221.41	5125.93	23.82%	679.16	0.281	0.02%
	3	57.49	48.50	1.25	1.21	1.97	20.05	22.29	2	7.9	891	8.99	2.23	4210.00	20679.00	1236.10	1227.81	5147.29	24.01%	685.47	0.284	0.67%
	4	57.56	48.48	1.25	1.21	1.97	19.99	22.23	2	7.9	891	9.08	2.24	4082.00	21061.00	1248.31	1231.21	5165.26	24.17%	690.48	0.286	1.38%
	5	57.51	48.43	1.25	1.21	1.98	19.92	22.17	2	7.9	891	9.08	2.25	4232.00	20993.00	1247.86	1237.84	5167.81	24.15%	689.92	0.285	0.81%
Average		57.50	48.52	1.25	1.21	1.96	20.03	22.27	2	7.9	891	8.99	2.23	4191.00	20579.40	1235.51	1227.80	5151.49	23.98%	684.42	0.283	0.63%

C.2 CFD results

Table C.3: Summary of all CFD results for the 0.5 mm heat exchanger

Test Group Data		T _{hi}	T _{ho}	P _{hi}	P _{ho}	P _{ci}	P _{co}	T _{ci}	T _{co}	PS _{hi}	PS _c	P _{atm}	ΔT _h	ΔT _c	ΔP _h	ΔP _c	q _h	q _c	q _{max}	ε	U	q _h vs q _c	
C*	Flow Setting	Test No.	(°C)	(°C)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	W/m ² ·K	%	
1	1	1	58	42,3	3,01	3,00	3,01	3,00	36,00	1	0,99	1,0133	15,82	15,81	1515,00	1714,00	1085,00	1085,00	2606,00	41,64%	36,07	0,53	0,00%
	2	1	58	44,4	3,01	2,99	3,01	2,99	33,90	1,5	1,48	1,0133	13,78	13,77	2314,00	2673,00	1418,00	1418,00	3910,00	36,26%	45,58	0,44	0,00%
	3	1	58	45,8	3,01	2,98	3,01	2,98	32,50	2	1,97	1,0133	12,35	12,34	3153,00	3685,00	1694,00	1695,00	5213,00	32,51%	53,24	0,39	0,00%
0,75	1	1	58	41,4	3,01	3,00	3,01	2,99	32,70	1	1,32	1,0133	16,78	12,57	1541,00	2405,00	1151,00	1151,00	2606,00	44,15%	38,86	0,57	0,00%
	2	1	58	43,5	3,01	2,99	3,01	2,98	31,10	1,5	1,97	1,0133	14,60	10,94	2346,00	3758,00	1503,00	1503,00	3910,00	38,43%	48,94	0,48	0,00%
	3	1	58	45,1	3,01	2,98	3,01	2,96	30,00	2	2,63	1,0133	13,08	9,80	3191,00	5194,00	1794,00	1794,00	5213,00	34,42%	57,02	0,42	0,00%
0,5	1	1	58	40,1	3,01	3,00	3,01	2,97	29,10	1	1,97	1,0133	18,00	8,99	1573,00	3868,00	1235,00	1235,00	2606,00	47,38%	42,58	0,62	0,00%
	2	1	58	42,5	3,01	2,99	3,01	2,95	28,00	1,5	2,96	1,0133	15,66	7,82	2387,00	6080,00	1611,00	1611,00	3909,00	41,21%	53,40	0,52	0,00%
	3	1	58	44,1	3,01	2,98	3,01	2,93	27,10	2	3,95	1,0133	14,01	7,03	3239,00	8457,00	1922,00	1922,00	5213,00	36,87%	62,00	0,45	0,00%
0,25	1	1	58	38,4	3,01	3,00	3,01	2,93	25,10	1	3,95	1,0133	19,74	4,93	1620,00	8749,00	1354,00	1354,00	2606,00	51,96%	48,14	0,70	0,00%
	2	1	58	41	3,01	2,99	3,01	2,87	24,40	1,5	5,92	1,0133	17,16	4,23	2446,00	14049,00	1766,00	1766,00	3909,00	45,17%	60,02	0,58	0,00%
	3	1	58	42,8	3,01	2,98	3,01	2,78	24,00	2	7,90	1,0133	15,33	3,83	3308,00	23443,00	2103,00	2104,00	5212,00	40,36%	69,34	0,51	0,00%

Table C.4: Summary of all CFD results for the 1 mm heat exchanger

Test Group Data		T _{hi}	T _{ho}	P _{hi}	P _{ho}	P _{ci}	P _{co}	T _{ci}	T _{co}	PS _h	PS _c	P _{atm}	ΔT _h	ΔT _c	ΔP _h	ΔP _c	q _h	q _c	q _{max}	ε	U	q _h vs q _c		
C*	Flow Setting	Test No.	(°C)	(°C)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	W/m ² ·K	NTU %		
1	1	1	57,00	45,98	1,02	1,01	1,02	1,01	17,00	27,99	1	0,99	1,1325	11,02	10,99	272,30	344,80	757,99	754,51	2751,34	27,55%	3292,06	0,380	0,46%
	2	1	57,00	48,37	1,02	1,01	1,02	1,03	17,00	25,62	1,5	1,48	1,1325	8,63	8,62	521,00	1240,00	889,97	888,08	4121,04	21,60%	3572,05	0,275	0,21%
	3	1	57,00	49,85	1,02	1,01	1,02	1,01	17,00	24,14	2	1,97	1,1325	7,15	7,14	843,00	1039,00	982,83	981,08	5496,22	17,88%	3767,07	0,218	0,18%
0,75	1	1	57,00	45,39	1,02	1,01	1,02	1,01	17,00	25,69	1	1,32	1,1325	11,61	8,69	289,00	592,00	798,67	795,81	2751,66	29,03%	3372,05	0,389	0,36%
	2	1	57,00	47,96	1,02	1,01	1,02	1,01	17,00	23,77	1,5	1,97	1,1325	9,04	6,77	521,89	1046,00	932,33	930,30	4125,36	22,60%	3659,65	0,282	0,22%
	3	1	57,00	49,55	1,03	1,01	1,03	1,01	17,00	22,59	2	2,63	1,1325	7,45	5,59	1329,00	1664,00	1024,13	1024,42	5498,71	18,63%	3853,08	0,223	-0,03%
0,5	1	1	57,00	44,70	1,02	1,01	1,02	1,01	17,00	23,15	1	1,97	1,1325	12,30	6,15	276,00	1058,90	846,25	845,20	2752,04	30,75%	3474,39	0,401	0,12%
	2	1	57,00	47,49	1,02	1,01	1,03	1,01	17,00	21,75	1,5	2,96	1,1325	9,51	4,75	521,00	2037,00	980,90	979,45	4125,75	23,78%	3764,52	0,290	0,15%
	3	1	57,00	49,16	1,03	1,01	1,05	1,01	17,00	20,92	2	3,95	1,1325	7,84	3,92	1291,40	3329,00	1077,83	1077,90	5499,15	19,60%	3982,41	0,230	-0,01%
0,25	1	1	57,00	43,78	1,02	1,01	1,05	1,01	17,00	20,30	1	3,95	1,1325	13,22	3,30	277,00	3334,99	909,72	907,52	2752,54	33,05%	3639,12	0,420	0,24%
	2	1	57,00	46,76	1,02	1,01	1,08	1,01	17,00	19,56	1,5	5,92	1,1325	10,24	2,56	525,00	6836,00	1056,35	1056,17	4126,36	25,60%	3976,44	0,306	0,02%
	3	1	57,00	48,53	1,02	1,01	1,13	1,01	17,00	19,12	2	7,90	1,1325	8,47	2,12	842,00	11539,00	1164,59	1166,28	5499,85	21,18%	4237,63	0,245	-0,15%

Table C.5: Summary of all CFD results for the 2 mm heat exchanger

Test Group Data		T _{hi}	T _{ho}	P _{hi}	P _{ho}	P _{di}	P _{co}	T _{di}	T _{co}	PS _h	PS _c	P _{atm}	ΔT _h	ΔT _c	ΔP _h	ΔP _c	q _h	q _c	q _{max}	ε	U	q _h vs q _c	
C*	Flow Setting	Test No.	(°C)	bar (abs)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	W/m ² K	NTU %	
1	1	1	57,88	45,19	1,01	1,01	1,01	20,42	33,10	1	0,99	1,1325	12,69	12,68	76,00	80,00	872,84	869,14	2567,67	33,99%	35,23	0,514	0,43%
	2	1	57,90	46,84	1,02	1,01	1,02	20,10	31,16	1,5	1,48	1,1325	11,06	11,05	152,00	157,00	1140,71	1137,63	3888,10	29,34%	5330,68	0,412	0,27%
	3	1	57,76	48,41	1,02	1,01	1,02	21,59	30,93	2	1,97	1,1325	9,35	9,34	248,00	256,00	1285,42	1280,65	4959,42	25,92%	6034,33	0,349	0,37%
0,75	1	1	58,05	44,67	1,01	1,01	1,01	21,79	31,83	1	1,32	1,1325	13,38	10,04	77,00	128,00	920,36	917,56	2494,20	36,90%	4728,28	0,546	0,31%
	2	1	57,83	46,03	1,02	1,01	1,02	20,49	29,34	1,5	1,97	1,1325	11,80	8,85	152,00	259,00	1217,24	1214,07	3851,86	31,60%	5679,76	0,437	0,26%
	3	1	57,77	47,33	1,02	1,01	1,02	20,17	28,00	2	2,63	1,1325	10,44	7,83	249,00	436,00	1435,58	1432,63	5170,28	27,77%	6355,43	0,367	0,21%
0,5	1	1	57,92	42,52	1,01	1,01	1,02	20,22	27,92	1	1,97	1,1325	15,40	7,70	77,00	260,00	1059,79	1056,64	2594,42	40,85%	40,82	0,593	0,30%
	2	1	57,82	44,85	1,02	1,01	1,02	20,29	26,77	1,5	2,96	1,1325	12,97	6,48	153,00	533,00	1338,26	1334,10	3872,38	34,56%	6088,70	0,469	0,31%
	3	1	57,78	46,45	1,02	1,01	1,03	20,02	25,68	2	3,95	1,1325	11,33	5,66	219,80	892,00	1558,23	1554,10	5193,19	30,01%	6726,25	0,388	0,27%
0,25	1	1	57,77	40,47	1,01	1,01	1,03	20,24	24,56	1	3,95	1,1325	17,30	4,32	76,74	893,30	1191,06	1186,36	2583,85	46,10%	5727,82	0,661	0,40%
	2	1	57,66	43,17	1,02	1,01	1,04	20,19	23,81	1,5	5,92	1,1325	14,49	3,62	154,00	1770,00	1495,64	1491,41	3867,61	38,67%	6710,99	0,516	0,28%
	3	1	57,50	44,95	1,02	1,01	1,04	20,03	23,17	2	7,90	1,1325	12,55	3,14	154,00	1770,00	1726,64	1725,13	5155,16	33,49%	7402,19	0,427	0,09%

C.3 Theoretical results: water - water

Table C.6: Summary of water - water theoretical results for the 0.5 mm heat exchanger

Test Group Data		T _{hi}	T _{ho}	P _{hi}	P _{ho}	P _{ci}	P _{co}	T _{ci}	T _{co}	PS _h	PS _c	P _{atm}	ΔT _h	ΔT _c	ΔP _h	ΔP _c	q _h	q _c	q _{max}	ε	U	q _h vs q _c		
C*	Flow Setting	Test No.	(°C)	(°C)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	W/m ² ·K	%		
1	1	1	58	42,3	3,01	3,00	3,01	3,00	20,00	36,00	1	0,99	1,0133	15,82	15,81	1515,00	1714,00	1085,00	1085,00	2606,00	41,64%	36,07	0,53	0,00%
	2	1	58	44,4	3,01	2,99	3,01	2,99	20,00	33,90	1,5	1,48	1,0133	13,78	13,77	2314,00	2673,00	1418,00	1418,00	3910,00	36,26%	45,58	0,44	0,00%
	3	1	58	45,8	3,01	2,98	3,01	2,98	20,00	32,50	2	1,97	1,0133	12,35	12,34	3153,00	3685,00	1694,00	1695,00	5213,00	32,51%	53,24	0,39	0,00%
0,75	1	1	58	41,4	3,01	3,00	3,01	2,99	20,00	32,70	1	1,32	1,0133	16,78	12,57	1541,00	2405,00	1151,00	1151,00	2606,00	44,15%	38,86	0,57	0,00%
	2	1	58	43,5	3,01	2,99	3,01	2,98	20,00	31,10	1,5	1,97	1,0133	14,60	10,94	2346,00	3758,00	1503,00	1503,00	3910,00	38,43%	48,94	0,48	0,00%
	3	1	58	45,1	3,01	2,98	3,01	2,96	20,00	30,00	2	2,63	1,0133	13,08	9,80	3191,00	5194,00	1794,00	1794,00	5213,00	34,42%	57,02	0,42	0,00%
0,5	1	1	58	40,1	3,01	3,00	3,01	2,97	20,00	29,10	1	1,97	1,0133	18,00	8,99	1573,00	3868,00	1235,00	1235,00	2606,00	47,38%	42,58	0,62	0,00%
	2	1	58	42,5	3,01	2,99	3,01	2,95	20,00	28,00	1,5	2,96	1,0133	15,66	7,82	2387,00	6080,00	1611,00	1611,00	3909,00	41,21%	53,40	0,52	0,00%
	3	1	58	44,1	3,01	2,98	3,01	2,93	20,00	27,10	2	3,95	1,0133	14,01	7,00	3239,00	8457,00	1922,00	1922,00	5213,00	36,87%	62,00	0,45	0,00%
0,25	1	1	58	38,4	3,01	3,00	3,01	2,93	20,00	25,10	1	3,95	1,0133	19,74	4,93	1620,00	8749,00	1354,00	1354,00	2606,00	51,96%	48,14	0,70	0,00%
	2	1	58	41	3,01	2,99	3,01	2,87	20,00	24,40	1,5	5,92	1,0133	17,16	4,29	2446,00	14049,00	1766,00	1766,00	3909,00	45,17%	60,02	0,58	0,00%
	3	1	58	42,8	3,01	2,98	3,01	2,78	20,00	24,00	2	7,90	1,0133	15,33	3,83	3308,00	23443,00	2103,00	2104,00	5212,00	40,36%	69,34	0,51	0,00%

Table C.7: Summary of water - water theoretical results for the 1 mm heat exchanger

Test Group Data		T_{hi}	T_{ho}	P_{hi}	P_{ho}	P_{ci}	P_{co}	T_{di}	T_{co}	PS_h	PS_c	P_{atm}	ΔT_h	ΔT_c	ΔP_h	ΔP_c	q_h	q_c	q_{max}	ε	U	q_h vs q_c
C^*	Flow Setting	(°C)	(°C)	bar (abs)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	$W/m^2 K$	NTU
1	1	58	45,9	1,30	1,30	1,26	1,26	20,00	32,70	1	0,99	1,0133	12,06	12,05	464,50	513,10	827,40	827,50	2561,00	32,31%	26,43	0,385
																					0,385	-0,01%
1	2	58	47,2	1,27	1,27	1,67	1,67	20,00	31,20	1,5	1,48	1,0133	10,71	10,70	951,70	1027,00	1102,00	1102,00	3847,00	28,64%	34,40	0,334
																					0,334	0,00%
3	1	58	48,1	1,29	1,29	1,48	1,47	20,00	30,10	2	1,97	1,0133	9,80	9,80	1609,00	1711,00	1345,00	1345,00	5160,00	26,07%	41,12	0,300
																					0,300	0,00%
1	1	58	44,7	1,24	1,24	1,15	1,15	20,00	30,20	1	1,32	1,0133	13,33	9,99	466,80	831,90	914,20	914,30	2591,00	35,28%	28,62	0,417
																					0,417	-0,01%
0,75	2	58	46,3	1,27	1,27	1,70	1,69	20,00	29,10	1,5	1,97	1,0133	11,79	8,84	954,40	1693,00	1213,00	1213,00	3886,00	31,22%	37,20	0,36
																					0,36	0,00%
3	1	58	47,4	1,22	1,22	1,85	1,84	20,00	28,40	2	2,63	1,0133	10,58	8,00	1612,00	2850,00	1465,00	1465,00	5169,00	28,34%	44,38	0,32
																					0,32	0,00%
1	1	58	43,2	1,27	1,27	1,41	1,41	20,00	27,70	1	1,97	1,0133	14,74	7,36	469,90	1686,00	1011,00	1011,00	2580,00	39,19%	31,55	0,46
																					0,46	0,00%
0,5	2	58	45	1,26	1,26	1,64	1,64	20,00	27,00	1,5	2,96	1,0133	12,95	6,47	958,50	3512,00	1332,00	1332,00	3855,00	34,55%	40,87	0,40
																					0,40	0,00%
3	1	58	46,2	1,24	1,24	1,91	1,91	20,00	26,30	2	3,95	1,0133	11,75	5,87	1617,00	5997,00	1613,00	1613,00	5155,00	31,28%	48,63	0,35
																					0,35	0,00%
1	1	58	41,2	1,27	1,27	1,56	1,55	20,00	24,70	1	3,95	1,0133	16,79	4,19	474,10	5981,00	1151,00	1151,00	2569,00	44,81%	35,85	0,52
																					0,52	0,00%
0,25	2	58	43,2	1,22	1,22	1,90	1,89	20,00	24,10	1,5	5,92	1,0133	14,74	3,68	964,60	12872,00	1517,00	1517,00	3858,00	39,31%	46,21	0,45
																					0,45	0,00%
3	1	58	44,6	1,22	1,22	2,15	2,13	20,00	23,50	2	7,90	1,0133	13,38	3,34	1625,00	22365,00	1836,00	1836,00	5176,00	35,46%	54,80	0,40
																					0,40	0,00%

Table C.8: Summary of water - water theoretical results for the 2 mm heat exchanger

Test Group Data	T_{hi}	T_{ho}	P_{hi}	P_{ho}	P_{ci}	P_{co}	T_{ci}	T_{co}	PS_h	PS_c	P_{atm}	ΔT_h	ΔT_c	ΔP_h	ΔP_c	q_h	q_c	q_{max}	ε	U	q_h vs q_c
Flow Setting	(°C)	(°C)	bar (abs)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	$W/m^2 K$	%
1	58	46,1	1,30	1,30	1,57	1,57	20,00	32,50	1	0,99	1,0133	11,95	11,94	316,30	312,70	819,90	819,90	2569,00	31,90%	26,04	0,38
2	58	46,9	1,21	1,21	1,37	1,37	20,00	31,40	1,5	1,48	1,0133	11,11	11,10	709,50	700,80	1143,00	1143,00	3889,00	29,39%	35,45	0,34
3	58	47,9	1,28	1,28	1,88	1,88	20,00	31,80	2	1,97	1,0133	10,05	10,04	1259,00	1243,00	1378,00	1378,00	4962,00	27,78%	44,25	0,32
1	58	45,6	1,18	1,18	1,17	1,17	20,00	31,40	1	1,32	1,0133	12,55	9,41	316,30	548,90	861,00	861,00	2487,00	34,62%	28,71	0,42
0,75	58	46,1	1,21	1,21	1,81	1,81	20,00	29,50	1,5	1,97	1,0133	11,87	8,90	709,70	1232,00	1222,00	1222,00	3841,00	31,80%	38,90	0,38
3	58	46,7	1,16	1,16	1,38	1,38	20,00	28,80	2	2,63	1,0133	11,25	8,43	1259,00	2187,00	1544,00	1544,00	5158,00	29,92%	48,27	0,35
1	58	43,8	1,25	1,25	1,51	1,51	20,00	27,50	1	1,97	1,0133	14,23	7,11	316,50	1224,00	976,10	976,10	2586,00	37,74%	31,91	0,47
0,5	58	44,9	1,24	1,24	1,56	1,56	20,00	27,00	1,5	2,96	1,0133	13,09	6,54	709,90	2748,00	1347,00	1347,00	3861,00	34,89%	43,47	0,42
3	58	45,5	1,23	1,23	1,72	1,72	20,00	26,40	2	3,95	1,0133	12,41	6,20	1260,00	4881,00	1702,00	1702,00	5179,00	32,86%	53,93	0,39
1	58	42	1,28	1,28	1,57	1,57	20,00	24,40	1	3,95	1,0133	15,92	3,98	316,70	4862,00	1092,00	1092,00	2574,00	42,42%	36,92	0,54
0,25	58	43,1	1,26	1,26	1,97	1,97	20,00	24,00	1,5	5,92	1,0133	14,71	3,67	710,30	10927,00	1514,00	1514,00	3855,00	39,26%	50,26	0,49
3	58	43,8	1,25	1,25	1,96	1,96	20,00	23,60	2	7,90	1,0133	13,87	3,46	1261,00	19416,00	1903,00	1903,00	5140,00	37,01%	62,30	0,45

C.4 Theoretical results: R-134a - water

Table C.9: Summary of R-134a - water theoretical results for the 0.5 mm heat exchanger

Test Group Data	T_{hi}	T_{ho}	P_{hi}	P_{ho}	P_{ci}	P_{co}	T_{ci}	T_{co}	PS_h	PS_c	P_{atm}	ΔT_h	ΔT_c	ΔP_h	ΔP_c	q_h	q_c	q_{max}	ε	U	NTU	q_h vs q_c
Flow Setting	(°C)	(°C)	bar (abs)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	$W/m^2 K$	NTU	%
1	40	27,6	3,0133	3,0034	7,5133	7,5073	0,00	17,9	0	1	1,0133	17,87	17,86	963,30	1021,00	532,00	526,70	1183,00	44,99%	17,16	0,58	1,01%
2	40	28,5	3,0133	2,9999	7,5133	7,5017	0,00	16,2	0,6	1,5	1,0133	16,10	16,18	1486,00	2104,00	721,60	714,50	1774,00	40,69%	22,65	0,51	0,99%
3	40	29,1	3,0133	2,9955	7,5133	7,4826	0,00	15,0	1	2	1,0133	14,93	14,97	2044,00	4745,00	888,90	880,30	2364,00	37,60%	27,37	0,46	0,98%
1	40	29,6	3,0133	3,0018	7,5133	7,5073	0,00	19,7	1	1	1,0133	14,87	19,37	1269,00	1016,00	590,30	583,50	1197,00	49,30%	18,83	0,62	1,17%
0,75	40	30,3	3,0133	2,9959	7,5133	7,5014	0,00	17,8	0,9	1,5	1,0133	13,38	17,80	1994,00	2139,00	769,70	787,70	1791,00	44,48%	24,74	0,54	-2,29%
3	40	30,8	3,0133	2,9900	7,5133	7,4820	0,00	16,4	1	2	1,0133	12,31	16,41	2787,00	4820,00	977,40	966,70	2383,00	41,02%	29,79	0,49	1,11%
1	40	32	3,0133	0,2996	7,5133	7,5074	0,00	22,1	1	1	1,0133	11,14	22,06	1928,00	1014,00	663,50	654,40	1204,00	55,12%	20,98	0,68	1,39%
0,5	40	32,6	3,0133	2,9879	7,5133	7,5010	0,00	19,8	1,3	1,5	1,0133	9,97	19,81	3122,00	2195,00	890,70	878,90	1799,00	49,50%	27,40	0,60	1,34%
3	40	33	3,0133	2,9789	7,5133	7,4812	0,00	18,2	2	2	1,0133	9,14	18,20	4478,00	4923,00	1088,00	1074,00	2393,00	45,47%	32,85	0,54	1,30%
1	40	35,3	3,0133	2,9805	7,5133	7,5075	0,00	25,2	2	1	1,0133	6,40	25,15	4319,00	1042,00	761,90	749,00	1213,00	62,82%	23,88	0,77	1,72%
0,25	40	35,7	3,0133	2,9619	7,5133	7,5003	0,00	22,5	2,6	1,5	1,0133	5,69	22,47	7471,00	2342,00	1017,00	1000,00	1811,00	56,15%	31,00	0,67	1,70%
3	40	35,9	3,0133	2,9418	7,5133	7,4803	0,00	20,6	3	2	1,0133	5,19	20,56	11288,00	5165,00	1236,00	1217,00	2406,00	51,37%	36,98	0,60	1,56%

Table C.10: Summary of R-134a - water theoretical results for the 1 mm heat exchanger

Test Group Data		T _{hi}	T _{ho}	P _{hi}	P _{ho}	P _{ci}	P _{co}	T _{ci}	T _{co}	PS _h	PS _c	P _{atm}	ΔT _h	ΔT _c	ΔP _h	ΔP _c	q _h	q _c	q _{max}	ε	U	q _h vs q _c		
C*	Flow Setting	Test No.	(°C)	bar (abs)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	W/m ² K	NTU %		
1	1	1	40	26,6	3,0133	3,0125	7,5133	7,5127	0	13,5	0,430	1	1,0133	13,42	13,49	138,70	481,00	399,60	395,80	1182,00	33,81%	12,02	0,404	0,96%
	2	1	40	27,8	3,0133	3,0122	7,5133	7,5123	0	12,3	0,646	1,5	1,0133	12,23	12,31	254,90	1044,00	546,30	541,10	1773,00	30,82%	16,14	0,362	0,96%
	3	1	40	28,6	3,0133	3,0115	7,5133	7,5119	0	11,5	0,861	2	1,0133	11,39	11,47	402,80	1823,00	678,10	671,70	2363,00	28,70%	19,79	0,332	0,95%
0,75	1	1	40	28,8	3,0133	3,0123	7,5133	7,5127	0	15,0	0,574	1	1,0133	11,24	15,00	209,80	481,70	445,90	441,00	1190,00	37,49%	13,34	0,443	1,11%
	2	1	40	29,8	3,0133	3,0118	7,5133	7,5123	0	13,6	0,861	1,5	1,0133	10,00	13,64	399,10	1047,00	607,50	600,80	1781,00	34,10%	17,85	0,397	1,12%
	3	1	40	30,5	3,0133	3,0113	7,5133	7,5119	0	12,7	1,149	2	1,0133	9,47	12,68	645,00	1828,00	751,90	743,80	2372,00	31,71%	21,85	0,365	1,09%
0,5	1	1	40	31,5	3,0133	3,0119	7,5133	7,5127	0	17,0	0,861	1	1,0133	8,51	16,96	394,10	485,90	506,40	499,80	1195,00	42,38%	15,07	0,498	1,32%
	2	1	40	32,3	3,0133	3,0112	7,5133	7,5132	0	15,4	1,292	1,5	1,0133	7,69	15,37	782,20	1057,00	687,00	678,10	1789,00	38,41%	20,09	0,444	1,31%
	3	1	40	32,9	3,0133	3,0104	7,5133	7,5119	0	14,2	1,723	2	1,0133	7,12	14,24	1298,00	1848,00	847,50	836,80	2381,00	35,60%	24,47	0,407	1,28%
0,25	1	1	40	35	3,0133	3,0105	7,5133	7,5127	0	19,7	1,723	1	1,0133	4,97	19,67	1285,00	516,10	591,30	581,70	1203,00	49,16%	17,54	0,574	1,65%
	2	1	40	35,5	3,0090	3,0090	7,5133	7,5124	0	17,8	2,585	1,5	1,0133	4,47	17,76	2695,00	1127,00	798,20	785,70	1799,00	44,38%	23,24	0,510	1,59%
	3	1	40	35,9	3,0074	3,0074	7,5133	7,5119	0	16,4	3,447	2	1,0133	4,12	16,40	4617,00	1973,00	980,90	966,00	2393,00	40,99%	28,20	0,466	1,54%

Table C.11: Summary of R-134a - water theoretical results for the 2 mm heat exchanger

Test Group Data		T _{hi}	T _{ho}	P _{hi}	P _{ho}	P _{ci}	P _{co}	T _{ci}	T _{co}	PS _h	PS _c	P _{atm}	ΔT _h	ΔT _c	ΔP _h	ΔP _c	q _h	q _c	q _{max}	ε	U	q _h vs q _c		
C*	Flow Setting	Test No.	(°C)	bar (abs)	bar (abs)	bar (abs)	bar (abs)	(°C)	(°C)	LPM	LPM	Bar	(°C)	(°C)	Pa	Pa	W	W	W	%	W/m ² K	NTU %		
1	1	1	40	27,6	3,0133	3,0133	7,5133	7,5133	0	12,5	0,430	1	1,0133	12,41	12,48	60,29	400,30	369,60	365,90	1182,00	31,27%	10,95	0,368	1,01%
	2	1	40	28,5	3,0133	3,0133	7,5133	7,5133	0	11,6	0,646	1,5	1,0133	11,53	11,60	134,40	900,70	514,80	509,60	1773,00	29,04%	15,06	0,337	1,02%
	3	1	40	29,1	3,0133	3,0133	7,5133	7,5133	0	11,0	0,861	2	1,0133	10,92	11,00	237,80	1601,00	650,20	643,70	2364,00	27,51%	18,85	0,317	1,01%
0,75	1	1	40	29,6	3,0133	3,0133	7,5133	7,5133	0	14,0	0,574	1	1,0133	10,45	13,96	106,30	400,60	414,60	409,80	1189,00	34,88%	12,23	0,407	1,17%
	2	1	40	30,3	3,0133	3,0133	7,5133	7,5133	0	13,0	0,861	1,5	1,0133	9,87	12,96	237,60	901,50	576,70	570,10	1781,00	32,39%	16,80	0,374	1,16%
	3	1	40	30,8	3,0133	3,0133	7,5133	7,5133	0	12,3	1,149	2	1,0133	9,17	12,27	421,20	1603,00	727,70	719,30	2372,00	30,68%	21,01	0,351	1,17%
0,5	1	1	40	32	3,0133	3,0133	7,5133	7,5133	0	15,9	0,861	1	1,0133	7,97	15,89	237,50	402,50	474,30	467,70	1194,00	39,72%	13,95	0,461	1,41%
	2	1	40	32,6	3,0133	3,0133	7,5133	7,5133	0	14,7	1,292	1,5	1,0133	7,38	14,74	532,00	906,00	658,80	649,70	1789,00	36,83%	19,13	0,423	1,40%
	3	1	40	33	3,0133	3,0132	7,5133	7,5133	0	13,9	1,723	2	1,0133	6,97	13,94	943,60	1611,00	830,30	818,90	2382,00	34,85%	23,90	0,397	1,39%
0,25	1	1	40	35,3	3,0133	3,0132	7,5133	7,5133	0	18,6	1,723	1	1,0133	4,70	18,63	943,00	416,30	560,00	550,30	1203,00	46,56%	16,44	0,539	1,76%
	2	1	40	35,7	3,0133	3,0132	7,5133	7,5133	0	17,3	2,585	1,5	1,0133	4,35	17,26	2117,00	937,20	776,60	763,30	1800,00	43,13%	22,51	0,493	1,74%
	3	1	40	35,9	3,0133	3,0132	7,5133	7,5133	0	16,3	3,447	2	1,0133	4,10	16,32	3760,00	1667,00	977,40	960,80	2397,00	40,78%	28,08	0,463	1,73%

Appendix D

EES calculations

The following mathematical model was developed for the preliminary sizing of the heat exchangers with a target heat transfer rate of 2 kW.

EES Equations

Input

$$T_{\infty;h;in} = \text{ConvertTemp}(C; K; 58 [C]) \quad \text{Hot Inlet} \quad (\text{D.1})$$

$$P_{h;in} = 301330 [\text{Pa}] \quad 2 \text{ bar} \quad (\text{D.2})$$

$$T_{\infty;c;in} = \text{ConvertTemp}(C; K; 20 [C]) \quad \text{Cold Inlet} \quad (\text{D.3})$$

$$P_{c;in} = 301330 [\text{Pa}] \quad 2 \text{ bar} \quad (\text{D.4})$$

$$T_{Room} = \text{ConvertTemp}(C; K; 20 [C]) \quad \text{Room Temp} \quad (\text{D.5})$$

$$Q_h = 1 \cdot \left| 1,66667 \times 10^{-5} \frac{\text{m}^3/\text{s}}{\text{l/min}} \right| \quad \text{Flow Rate [l/s]} \quad (\text{D.6})$$

$$Q_c = 1 \cdot \left| 1,66667 \times 10^{-5} \frac{\text{m}^3/\text{s}}{\text{l/min}} \right| \quad (\text{D.7})$$

HX Geometry

$$HX_{o;w} = 0,001 [\text{m}] \quad \text{outer wall thickness} \quad (\text{D.8})$$

$$HX_{chl;w} = 0,0005 [\text{m}] \quad \text{channel wall thickness} \quad (\text{D.9})$$

$$H_{chl} = 0,0005 [\text{m}] \quad \text{channel height} \quad (\text{D.10})$$

$$W_{chl} = 0,0005 [\text{m}] \quad \text{channel width} \quad (\text{D.11})$$

$$N_{chl} = 20 \quad \text{number of channels per row} \quad (\text{D.12})$$

$$N_r = 11 \quad \text{number of rows} \quad (\text{D.13})$$

$$L_{chl;L1} = 0,015 [\text{m}] \quad \text{channel lenght L1} \quad (\text{D.14})$$

$$L_{chl;L2} = 0,015 [\text{m}] \quad \text{channel lenght L2} \quad (\text{D.15})$$

$$L_{chl;Total} = L_{chl;L1} + L_{chl;L2} \quad (\text{D.16})$$

$$L_{MF} = 0,018 [\text{m}] \quad \text{Manifold length} \quad (\text{D.17})$$

$$W_{MF} = 0,013 [\text{m}] \quad \text{Manifold width} \quad (\text{D.18})$$

Mean Temperatures

Point 1 is hot in/cold out point 2 is hot out/cold in

$$\bar{T}_{\infty;h;L1} = \frac{T_{\infty;h;in} + T_{\infty;h;mid}}{2} \quad \text{Hot Free-stream Mean Temperature} \quad (D.19)$$

$$\bar{T}_{\infty;c;L1} = \frac{T_{\infty;c;out} + T_{\infty;c;mid}}{2} \quad \text{Cold Free-stream Mean Temperature} \quad (D.20)$$

$$\bar{T}_{\infty;h;L2} = \frac{T_{\infty;h;mid} + T_{\infty;h;out}}{2} \quad (D.21)$$

$$\bar{T}_{\infty;c;L2} = \frac{T_{\infty;c;mid} + T_{\infty;c;in}}{2} \quad (D.22)$$

Wall Temperatures

$$T_{w;h;i} = \left(\left(\left(\frac{T_{w;c;o} - T_{\infty;c;out}}{Rc_{c;L1}} \right) \cdot R w_{L1} \right) + T_{w;c;o} \right) \quad \text{Wall Temperature at Hot Inlet} \quad (D.23)$$

$$T_{w;h;mid} = \left(\left(\left(\frac{T_{w;c;mid} - T_{\infty;c;mid}}{Rc_{c;L1}} \right) \cdot R w_{L1} \right) + T_{w;c;mid} \right) \quad \text{Wall Temperature at Hot Mid} \quad (D.24)$$

$$T_{w;h;o} = \left(\left(\left(\frac{T_{w;c;i} - T_{\infty;c;in}}{Rc_{c;L2}} \right) \cdot R w_{L2} \right) + T_{w;c;i} \right) \quad \text{Wall Temperature at Hot Outlet} \quad (D.25)$$

$$T_{w;c;i} = \left(T_{w;h;o} - \left(\left(\frac{T_{\infty;h;out} - T_{w;h;o}}{Rc_{h;L2}} \right) \cdot R w_{L2} \right) \right) \quad \text{Wall Temperature at Cold Inlet} \quad (D.26)$$

$$T_{w;c;mid} = \left(T_{w;h;mid} - \left(\left(\frac{T_{\infty;h;mid} - T_{w;h;mid}}{Rc_{h;L2}} \right) \cdot R w_{L2} \right) \right) \quad \text{Wall Temperature at Cold Mid} \quad (D.27)$$

$$T_{w;c;o} = \left(T_{w;h;i} - \left(\left(\frac{T_{\infty;h;in} - T_{w;h;i}}{Rc_{h;L1}} \right) \cdot R w_{L1} \right) \right) \quad \text{Wall Temperature at Cold Outlet} \quad (D.28)$$

Film Temperatures

$$T_{f;h;i} = \frac{T_{w;h;i} + T_{\infty;h;in}}{2} \quad \text{hot in} \quad (\text{D.29})$$

$$T_{f;h;mid} = \frac{T_{w;h;mid} + T_{\infty;h;mid}}{2} \quad \text{hot mid} \quad (\text{D.30})$$

$$T_{f;h;o} = \frac{T_{w;h;o} + T_{\infty;h;out}}{2} \quad \text{hot out} \quad (\text{D.31})$$

$$T_{f;c;i} = \frac{T_{w;c;i} + T_{\infty;c;in}}{2} \quad \text{cold in} \quad (\text{D.32})$$

$$T_{f;c;mid} = \frac{T_{w;c;mid} + T_{\infty;c;mid}}{2} \quad \text{cold mid} \quad (\text{D.33})$$

$$T_{f;c;o} = \frac{T_{w;c;o} + T_{\infty;c;out}}{2} \quad \text{cold out} \quad (\text{D.34})$$

$$\bar{T}_{f;h;L1} = \frac{T_{f;h;i} + T_{f;h;mid}}{2} \quad \text{mean film temp hot} \quad (\text{D.35})$$

$$\bar{T}_{f;c;L1} = \frac{T_{f;c;mid} + T_{f;c;o}}{2} \quad \text{mean film temp cold} \quad (\text{D.36})$$

$$\bar{T}_{f;h;L2} = \frac{T_{f;h;mid} + T_{f;h;o}}{2} \quad (\text{D.37})$$

$$\bar{T}_{f;c;L2} = \frac{T_{f;c;i} + T_{f;c;mid}}{2} \quad (\text{D.38})$$

Mean Temperatures

$$\bar{T}_{h;L1} = \frac{\bar{T}_{\infty;h;L1} + \bar{T}_{f;h;L1}}{2} \quad \text{mean temp hot} \quad (\text{D.39})$$

$$\bar{T}_{c;L1} = \frac{\bar{T}_{\infty;c;L1} + \bar{T}_{f;c;L1}}{2} \quad \text{mean temp cold} \quad (\text{D.40})$$

$$\bar{T}_{h;L2} = \frac{\bar{T}_{\infty;h;L2} + \bar{T}_{f;h;L2}}{2} \quad (\text{D.41})$$

$$\bar{T}_{c;L2} = \frac{\bar{T}_{\infty;c;L2} + \bar{T}_{f;c;L2}}{2} \quad (\text{D.42})$$

$$T_{opr;mat} = \frac{T_{w;h;mid} + T_{w;c;mid}}{2} \quad \text{material operating temp} \quad (\text{D.43})$$

Mean Pressures

$$\bar{P}_{h;L1} = \frac{P_{h;in} + P_{h;mid}}{2} \quad \text{mean pressure hot} \quad (D.44)$$

$$\bar{P}_{c;L1} = \frac{P_{c;mid} + P_{c;out}}{2} \quad \text{mean pressure cold} \quad (D.45)$$

$$\bar{P}_{h;L2} = \frac{P_{h;mid} + P_{h;out}}{2} \quad (D.46)$$

$$\bar{P}_{c;L2} = \frac{P_{c;in} + P_{c;mid}}{2} \quad (D.47)$$

PROPERTIES

Material Ti64

$$\rho_{mat} = \rho(Ti_{6Al;4V}; T = 300 \text{ [K]}) \quad \text{Density material} \quad (D.48)$$

$$cp_{mat} = c_p(Ti_{6Al;4V}; T = 300 \text{ [K]}) \quad \text{Specific Heat material} \quad (D.49)$$

$$k_{mat} = k(Ti_{6Al;4V}; T = T_{opr;mat}) \quad \text{Conductivity material} \quad (D.50)$$

$$SR = 0,000014 \text{ [m]} \quad \text{Surface Roughness} \quad (D.51)$$

$$RR = SR/D_h \quad \text{Relative Roughness} \quad (D.52)$$

All fluid properties are evaluated at the mean temperature of the fluid

HOT-SIDE - WATER

$$\rho_{h;in} = \rho(Water; T = T_{\infty;h;in}; P = P_{h;in}) \quad (D.53)$$

$$\rho_{h;mid} = \rho(Water; T = T_{\infty;h;mid}; P = P_{h;mid}) \quad (D.54)$$

$$\rho_{h;out} = \rho(Water; T = T_{\infty;h;out}; P = P_{h;out}) \quad (D.55)$$

$$\rho_{h;m;L1} = \frac{\rho_{h;in} + \rho_{h;mid}}{2} \quad (D.56)$$

$$\rho_{h;m;L2} = \frac{\rho_{h;mid} + \rho_{h;out}}{2} \quad (D.57)$$

$$cp_{h;mid} = c_p(Water; T = T_{\infty;h;mid}; P = P_{h;mid}) \quad (D.58)$$

$$s_{h;in} = s(Water; T = T_{\infty;h;in}; P = P_{h;in}) \quad (D.59)$$

$$s_{h;mid} = s(Water; T = T_{\infty;h;mid}; P = P_{h;mid}) \quad (D.60)$$

$$s_{h;out} = s(Water; T = T_{\infty;h;out}; P = P_{h;out}) \quad (D.61)$$

$$\rho_{h;L1} = \rho(Water; T = \bar{T}_{h;L1}; P = \bar{P}_{h;L1}) \quad \text{Density hot} \quad (D.62)$$

$$cp_{h;L1} = c_p(Water; T = \bar{T}_{h;L1}; P = \bar{P}_{h;L1}) \quad \text{Specific heat hot} \quad (D.63)$$

$$\mu_{h;L1} = \mu(Water; T = \bar{T}_{h;L1}; P = \bar{P}_{h;L1}) \quad \text{Dynamic viscosity hot} \quad (D.64)$$

$$k_{h;L1} = k(Water; T = \bar{T}_{h;L1}; P = \bar{P}_{h;L1}) \quad \text{Conductivity hot} \quad (D.65)$$

$$Pr_{h;L1} = P_r (Water; T = \bar{T}_{h;L1}; P = \bar{P}_{h;L1}) \quad \text{Prandtl number} \quad (D.66)$$

$$\rho_{h;L2} = \rho (Water; T = \bar{T}_{h;L2}; P = \bar{P}_{h;L2}) \quad (D.67)$$

$$cp_{h;L2} = c_p (Water; T = \bar{T}_{h;L2}; P = \bar{P}_{h;L2}) \quad (D.68)$$

$$\mu_{h;L2} = \mu (Water; T = \bar{T}_{h;L2}; P = \bar{P}_{h;L2}) \quad (D.69)$$

$$k_{h;L2} = k (Water; T = \bar{T}_{h;L2}; P = \bar{P}_{h;L2}) \quad (D.70)$$

$$Pr_{h;L2} = P_r (Water; T = \bar{T}_{h;L2}; P = \bar{P}_{h;L2}) \quad (D.71)$$

COLD-SIDE - WATER

$$\rho_{c;in} = \rho (Water; T = T_{\infty;c;in}; P = P_{c;in}) \quad (D.72)$$

$$\rho_{c;mid} = \rho (Water; T = T_{\infty;c;mid}; P = P_{c;mid}) \quad (D.73)$$

$$\rho_{c;out} = \rho (Water; T = T_{\infty;c;out}; P = P_{c;out}) \quad (D.74)$$

$$\rho_{c;m;L1} = \frac{\rho_{c;out} + \rho_{c;mid}}{2} \quad (D.75)$$

$$\rho_{c;m;L2} = \frac{\rho_{c;mid} + \rho_{c;in}}{2} \quad (D.76)$$

$$cp_{c;mid} = c_p (Water; T = T_{\infty;c;mid}; P = P_{c;mid}) \quad (D.77)$$

$$\rho_{c;L1} = \rho (Water; T = \bar{T}_{c;L1}; P = \bar{P}_{c;L1}) \quad (D.78)$$

$$cp_{c;L1} = c_p (Water; T = \bar{T}_{c;L1}; P = \bar{P}_{c;L1}) \quad (D.79)$$

$$\mu_{c;L1} = \mu (Water; T = \bar{T}_{c;L1}; P = \bar{P}_{c;L1}) \quad (D.80)$$

$$k_{c;L1} = k (Water; T = \bar{T}_{c;L1}; P = \bar{P}_{c;L1}) \quad (D.81)$$

$$Pr_{c;L1} = P_r (Water; T = \bar{T}_{c;L1}; P = \bar{P}_{c;L1}) \quad (D.82)$$

$$\rho_{c;L2} = \rho (Water; T = \bar{T}_{c;L2}; P = \bar{P}_{c;L2}) \quad (D.83)$$

$$cp_{c;L2} = c_p (Water; T = \bar{T}_{c;L2}; P = \bar{P}_{c;L2}) \quad (D.84)$$

$$\mu_{c;L2} = \mu (Water; T = \bar{T}_{c;L2}; P = \bar{P}_{c;L2}) \quad (D.85)$$

$$k_{c;L2} = k (Water; T = \bar{T}_{c;L2}; P = \bar{P}_{c;L2}) \quad (D.86)$$

$$Pr_{c;L2} = P_r (Water; T = \bar{T}_{c;L2}; P = \bar{P}_{c;L2}) \quad (D.87)$$

COLD-SIDE - R-134a (used for the R-134a calculations)

$$\rho_{c;in} = \rho(R134a; T = T_{\infty;c;in}; P = P_{c;in}) \quad (D.88)$$

$$\rho_{c;mid} = \rho(R134a; T = T_{\infty;c;mid}; P = P_{c;mid}) \quad (D.89)$$

$$\rho_{c;out} = \rho(R134a; T = T_{\infty;c;out}; P = P_{c;out}) \quad (D.90)$$

$$\rho_{c;m;L1} = \frac{\rho_{c;out} + \rho_{c;mid}}{2} \quad (D.91)$$

$$\rho_{c;m;L2} = \frac{\rho_{c;mid} + \rho_{c;in}}{2} \quad (D.92)$$

$$cp_{c;mid} = c_p(Water; T = T_{\infty;c;mid}; P = P_{c;mid}) \quad (D.93)$$

$$\rho_{c;L1} = \rho(R134a; T = \bar{T}_{c;L1}; P = \bar{P}_{c;L1}) \quad (D.94)$$

$$cp_{c;L1} = c_p(R134a; T = \bar{T}_{c;L1}; P = \bar{P}_{c;L1}) \quad (D.95)$$

$$\mu_{c;L1} = \mu(R134a; T = \bar{T}_{c;L1}; P = \bar{P}_{c;L1}) \quad (D.96)$$

$$k_{c;L1} = k(R134a; T = \bar{T}_{c;L1}; P = \bar{P}_{c;L1}) \quad (D.97)$$

$$Pr_{c;L1} = P_r(R134a; T = \bar{T}_{c;L1}; P = \bar{P}_{c;L1}) \quad (D.98)$$

$$\rho_{c;L2} = \rho(R134a; T = \bar{T}_{c;L2}; P = \bar{P}_{c;L2}) \quad (D.99)$$

$$cp_{c;L2} = c_p(R134a; T = \bar{T}_{c;L2}; P = \bar{P}_{c;L2}) \quad (D.100)$$

$$\mu_{c;L2} = \mu(R134a; T = \bar{T}_{c;L2}; P = \bar{P}_{c;L2}) \quad (D.101)$$

$$k_{c;L2} = k(R134a; T = \bar{T}_{c;L2}; P = \bar{P}_{c;L2}) \quad (D.102)$$

$$Pr_{c;L2} = P_r(R134a; T = \bar{T}_{c;L2}; P = \bar{P}_{c;L2}) \quad (D.103)$$

$$s_{c;out} = s(R134a; T = T_{\infty;c;out}; P = P_{c;out}) \quad (D.104)$$

$$h_{c;out} = h(R134a; T = T_{\infty;c;out}; P = P_{c;out}) \quad (D.105)$$

$$P_c = P(R134a; T = T_{\infty;c;out}; P = P_{c;out}) \quad (D.106)$$

$$cp_{c;in} = c_p(R134a; T = T_{\infty;c;in}; P = P_{c;in}) \quad (D.107)$$

$$x_c = x(R134a; T = T_{\infty;c;out}; P = P_{c;out}) \quad (D.108)$$

Calculations

AREA

$$W_{HX} = ((HX_{o;w} \cdot 2) + (W_{chl} \cdot N_{chl}) + (HX_{chl;w} \cdot (N_{chl} - 1))) \quad (D.109)$$

$$H_{HX} = ((HX_{o;w} \cdot 2) + (H_{chl} \cdot N_r) + (HX_{chl;w} \cdot (N_r - 1))) \quad (D.110)$$

$$p_{Dh} = (W_{chl} + H_{chl}) \cdot 2 \quad \text{Wetted perimeter} \quad (D.111)$$

$$A_{fr} = (W_{chl} \cdot N_{chl} + (HX_{chl;w} \cdot (N_{chl} - 1))) \cdot (H_{chl} \cdot N_r + (HX_{chl;w} \cdot (N_r - 1))) \quad (D.112)$$

Frontal area of hx total core

$$A_c = (W_{chl} \cdot H_{chl}) \cdot (N_{chl} \cdot N_r) \quad \text{Free-flow area} \quad (D.113)$$

$$A_{fr;h} = ((N_{chl} \cdot W_{chl}) + ((N_{chl} - 1) \cdot HX_{chl;w})) \cdot (N_{r;h} \cdot H_{chl}) \quad (D.114)$$

Fontal area hot

$$A_{fr;c} = ((N_{chl} \cdot W_{chl}) + ((N_{chl} - 1) \cdot HX_{chl;w})) \cdot (N_{r;c} \cdot H_{chl}) \quad (D.115)$$

Fontal area cold

$$A_{c;h} = (W_{chl} \cdot H_{chl}) \cdot (N_{chl} \cdot N_{r;h}) \quad (D.116)$$

$$A_{c;c} = (W_{chl} \cdot H_{chl}) \cdot (N_{chl} \cdot N_{r;c}) \quad (D.117)$$

$$N_{r;h} = \text{Trunc}(N_r/2) \quad \text{Free-flow area hot} \quad (D.118)$$

$$N_{r;c} = \text{Round}(N_r/2) \quad \text{Free-flow area cold} \quad (D.119)$$

$$\sigma_h = A_{c;h}/A_{fr;h} \quad \text{Ratio of free-flow to fronal hot} \quad (D.120)$$

$$\sigma_c = A_{c;c}/A_{fr;c} \quad \text{Ratio of free-flow to fronal cold} \quad (D.121)$$

$$A_{chl;cr} = W_{chl} \cdot H_{chl} \quad \text{Channel cross section} \quad (D.122)$$

$$D_h = 4 \cdot A_{chl;cr}/p_{Dh} \quad \text{Channel hyd Diameter} \quad (D.123)$$

$$A_{chl;s;L1} = W_{chl} \cdot L_{chl;L1} \quad \text{Channel HT area L1} \quad (D.124)$$

$$A_{ht;L1} = ((A_{chl;s;L1} \cdot N_{chl}) \cdot (N_r - 1)) + A_{MF} + A_{f;L1} \quad \text{Total HT area L1} \quad (D.125)$$

$$A_{chl;s;L2} = W_{chl} \cdot L_{chl;L2} \quad (D.126)$$

$$A_{ht;L2} = ((A_{chl;s;L2} \cdot N_{chl}) \cdot (N_r - 1)) + A_{MF} + A_{f;L2} \quad (D.127)$$

$$A_{ht} = (A_{ht;L1} + A_{ht;L2}) \quad (D.128)$$

Manifold Area - Both sides are the same

$$A_{MF} = (L_{MF} \cdot W_{MF}) \cdot (N_r - 1) \quad \text{Manifold Heat Transfer Area} \quad (D.129)$$

Flow Calculations

$$\dot{m}_{h;PS} = ((\rho_{h;in}) \cdot Q_h) \quad \text{Water - Hot mass flow rate} \quad (\text{D.130})$$

$$\dot{m}_{c;PS} = ((\rho_{c;in}) \cdot Q_c) \quad \text{Water - Cold mass flow rate} \quad (\text{D.131})$$

$$\dot{m}_{h;chl} = \frac{\dot{m}_{h;PS}}{(N_{chl} \cdot N_{r;h})} \quad \text{Mass flow per channel} \quad (\text{D.132})$$

$$\dot{m}_{c;chl} = \frac{\dot{m}_{c;PS}}{(N_{chl} \cdot N_{r;c})} \quad \text{Mass flow per channel} \quad (\text{D.133})$$

$$v_{chl;h;L1} = \frac{\dot{m}_{h;chl}}{(\rho_{h;L1} \cdot A_{chl;cr})} \quad \text{Flow velo in channel} \quad (\text{D.134})$$

$$v_{chl;c;L1} = \frac{\dot{m}_{c;chl}}{\rho_{c;L1} \cdot A_{chl;cr}} \quad (\text{D.135})$$

$$v_{chl;h;L2} = \frac{\dot{m}_{h;chl}}{\rho_{h;L2} \cdot A_{chl;cr}} \quad (\text{D.136})$$

$$v_{chl;c;L2} = \frac{\dot{m}_{c;chl}}{\rho_{c;L2} \cdot A_{chl;cr}} \quad (\text{D.137})$$

$$G_h = \dot{m}_{h;PS} / A_{c;h} \quad \text{Mass velo} \quad (\text{D.138})$$

$$G_c = \dot{m}_{c;PS} / A_{c;c} \quad (\text{D.139})$$

$$Re_{h;L1} = \frac{\rho_{h;L1} \cdot v_{chl;h;L1} \cdot D_h}{\mu_{h;L1}} \quad \text{Reynolds number} \quad (\text{D.140})$$

$$Re_{c;L1} = \frac{\rho_{c;L1} \cdot v_{chl;c;L1} \cdot D_h}{\mu_{c;L1}} \quad (\text{D.141})$$

$$Re_{h;L2} = \frac{\rho_{h;L2} \cdot v_{chl;h;L2} \cdot D_h}{\mu_{h;L2}} \quad (\text{D.142})$$

$$Re_{c;L2} = \frac{\rho_{c;L2} \cdot v_{chl;c;L2} \cdot D_h}{\mu_{c;L2}} \quad (\text{D.143})$$

Entrance Length

Lam flow <2300

$$Le_{h;Lam} = 0,06 \cdot Re_{h;L1} \cdot D_h \quad (\text{D.144})$$

$$Le_{c;Lam} = 0,06 \cdot Re_{c;L2} \cdot D_h \quad (\text{D.145})$$

Turbulent flow >4000

$$Le_{h;Tur} = 4,4 \cdot \left(Re_{h;L1}^{1/6} \right) \cdot D_h \quad (\text{D.146})$$

$$Le_{c;Tur} = 4,4 \cdot \left(Re_{c;L2}^{1/6} \right) \cdot D_h \quad (\text{D.147})$$

Nusselt Number Calculations

$$\nu_{h;L1} = 0,023 \cdot (Re_{h;L1}^{0,8}) \cdot Pr_{h;L1}^{0,3} \quad (D.148)$$

$$\nu_{c;L1} = 0,023 \cdot (Re_{c;L1}^{0,8}) \cdot Pr_{c;L1}^{0,4} \quad (D.149)$$

$$\nu_{h;L2} = 0,023 \cdot (Re_{h;L2}^{0,8}) \cdot Pr_{h;L2}^{0,3} \quad (D.150)$$

$$\nu_{c;L2} = 0,023 \cdot (Re_{c;L2}^{0,8}) \cdot Pr_{c;L2}^{0,4} \quad (D.151)$$

$Re > 10^4$, $0,7 < Pr < 120$, $L/dh > 10$, 0.3 for cooling, 0.4 for heating

Overall Heat Transfer Coefficient

$$UA_{L1} = \left(\frac{1}{\left(\frac{1}{\eta_{t;h;L1} \cdot h_{i;L1} \cdot A_{ht;L1}} \right) + \left(\frac{HX_{chl;w}}{k_{mat} \cdot A_{ht;L1}} \right) + \left(\frac{1}{\eta_{t;c;L1} \cdot h_{i;c;L1} \cdot A_{ht;L1}} \right)} \right) \quad (D.152)$$

Overall Heat Transfer coefficient

$$U_{L1} = \frac{1}{(R_{T;L1} \cdot A_{ht;L1})} \quad \text{Overall HT coefficient based on R} \quad (D.153)$$

$$UA_{L2} = \left(\frac{1}{\left(\frac{1}{\eta_{t;h;L2} \cdot h_{i;L2} \cdot A_{ht;L2}} \right) + \left(\frac{HX_{chl;w}}{k_{mat} \cdot A_{ht;L2}} \right) + \left(\frac{1}{\eta_{t;c;L2} \cdot h_{i;c;L2} \cdot A_{ht;L2}} \right)} \right) \quad (D.154)$$

$$U_{L2} = \frac{1}{R_{T;L2} \cdot A_{ht;L2}} \quad (D.155)$$

$$UA = UA_{L1} + UA_{L2} \quad (D.156)$$

Fin Efficiency

Fin Area

Full Fin

$$A_{c,f} = H_{chl} \cdot W_{chl} \quad \text{Fin Cross Sectional Area} \quad (\text{D.157})$$

$$A_{f,L1} = 2 \cdot H_{chl} \cdot L_{chl,L1} \cdot (N_{chl} - 1) \cdot N_r \quad \text{Exposed fin area per channel} \quad (\text{D.158})$$

$$A_{f,L2} = 2 \cdot H_{chl} \cdot L_{chl,L2} \cdot (N_{chl} - 1) \cdot N_r \quad (\text{D.159})$$

$$A_{L1} = A_{ht,L1} \quad (\text{D.160})$$

$$A_{L2} = A_{ht,L2} \quad (\text{D.161})$$

$$P_f = H_{chl} \cdot 2 \quad \text{Fin Perimeter, No conv. at tip} \quad (\text{D.162})$$

$$L = H_{chl} \quad \text{Height of Fin} \quad (\text{D.163})$$

Fin Hot L1

$$\eta_{f,h;L1} = \frac{\tanh(m_{h;L1} \cdot L)}{(m_{h;L1} \cdot L)} \quad \text{Fin Efficiency Hot Side - L1} \quad (\text{D.164})$$

$$m_{h;L1} = \sqrt{\frac{hi_{f,h;L1} \cdot P_f}{k_{mat} \cdot A_{c,f}}} \quad h_f = h_i, \text{ Performance Factor} \quad (\text{D.165})$$

$$hi_{f,h;L1} = (hi_{h;L1}) \quad (\text{D.166})$$

$$q_{max,f;h;L1} = hi_{f,h;L1} \cdot (P_f \cdot L) \cdot \text{abs}(T_{b;h;L1} - T_{\infty;h;L1}) \quad (\text{D.167})$$

$P_f \cdot L = \text{Exposed Fin area, qmax for fin}$

$$T_{b;h;L1} = \frac{T_{w;h;i} + T_{w;h;mid}}{2} \quad \text{Base temp fin} \quad (\text{D.168})$$

$$T_{\infty;h;L1} = \bar{T}_{\infty;h;L1} \quad \text{Free stream Temp} \quad (\text{D.169})$$

$$q_{f;h;L1} = \eta_{f;h;L1} \cdot q_{max,f;h;L1} \quad \text{Fin Heat Transfer Rate} \quad (\text{D.170})$$

$$M1_{h;L1} = \frac{q_{f;h;L1}}{(\tanh(m_{h;L1} \cdot L))} \quad q_f = M \cdot \tanh(ml) \quad (\text{D.171})$$

$$\theta_{h;L1} = \frac{M1_{h;L1}}{(\sqrt{hi_{f,h;L1} \cdot P_f \cdot k_{mat} \cdot A_{c,f}})} \quad M = \sqrt{hi_{f,h;L1} \cdot P_f \cdot k_{mat} \cdot A_{c,f}} \cdot \theta \quad (\text{D.172})$$

$$T_{fin;h;L1} = \theta_{h;L1} + T_{\infty;h;L1} \quad (\text{D.173})$$

$$\eta_{t;h;L1} = 1 - (1 - \eta_{f;h;L1}) \cdot A_{f;L1} / A_{L1} \quad (\text{D.174})$$

Fin Hot L2

$$\eta_{f,h;L2} = \frac{\tanh(m_{h;L2} \cdot L)}{m_{h;L2} \cdot L} \quad (\text{D.175})$$

$$m_{h;L2} = \sqrt{\frac{hi_{f,h;L2} \cdot P_f}{k_{mat} \cdot A_{c,f}}} \quad (\text{D.176})$$

$$hi_{f,h;L2} = (hi_{h;L2}) \quad (\text{D.177})$$

$$q_{max;f;h;L2} = hi_{f;h;L2} \cdot (P_f \cdot L) \cdot abs(T_{b;h;L2} - T_{\infty;h;L2}) \quad (D.178)$$

$$T_{b;h;L2} = \frac{T_{w;h;mid} + T_{w;h;o}}{2} \quad (D.179)$$

$$T_{\infty;h;L2} = \bar{T}_{\infty;h;L2} \quad (D.180)$$

$$q_{f;h;L2} = \eta_{f;h;L2} \cdot q_{max;f;h;L2} \quad (D.181)$$

$$M1_{h;L2} = \frac{q_{f;h;L2}}{(\tanh(m_{h;L2} \cdot L))} \quad (D.182)$$

$$\theta_{h;L2} = \frac{M1_{h;L2}}{(\sqrt{hi_{f;h;L2} \cdot P_f \cdot k_{mat} \cdot A_{c;f}})} \quad (D.183)$$

$$T_{fin;h;L2} = \theta_{h;L2} + T_{\infty;h;L2} \quad (D.184)$$

$$\eta_{t;h;L2} = 1 - (1 - \eta_{f;h;L2}) \cdot A_{f;L2}/A_{L2} \quad (D.185)$$

Fin Cold L1

$$\eta_{f;c;L1} = \frac{\tanh(m_{c;L1} \cdot L)}{m_{c;L1} \cdot L} \quad (D.186)$$

$$m_{c;L1} = \sqrt{\frac{hi_{f;c;L1} \cdot P_f}{k_{mat} \cdot A_{c;f}}} \quad (D.187)$$

$$hi_{f;c;L1} = (hi_{c;L1}) \quad (D.188)$$

$$q_{max;f;c;L1} = hi_{f;c;L1} \cdot (P_f \cdot L) \cdot abs(T_{b;c;L1} - T_{\infty;c;L1}) \quad (D.189)$$

$$T_{b;c;L1} = \frac{T_{w;c;o} + T_{w;c;mid}}{2} \quad (D.190)$$

$$T_{\infty;c;L1} = \bar{T}_{\infty;c;L1} \quad (D.191)$$

$$q_{f;c;L1} = \eta_{f;c;L1} \cdot q_{max;f;c;L1} \quad (D.192)$$

$$M1_{c;L1} = \frac{q_{f;c;L1}}{(\tanh(m_{c;L1} \cdot L))} \quad (D.193)$$

$$\theta_{c;L1} = \frac{M1_{c;L1}}{(\sqrt{hi_{f;c;L1} \cdot P_f \cdot k_{mat} \cdot A_{c;f}})} \quad (D.194)$$

$$T_{fin;c;L1} = \theta_{c;L1} + T_{\infty;c;L1} \quad (D.195)$$

$$\eta_{t;c;L1} = 1 - (1 - \eta_{f;c;L1}) \cdot A_{f;L1}/A_{L1} \quad (D.196)$$

Fin Cold L2

$$\eta_{f;c;L2} = \frac{\tanh(m_{c;L2} \cdot L)}{m_{c;L2} \cdot L} \quad (D.197)$$

$$m_{c;L2} = \sqrt{\frac{hi_{f;c;L2} \cdot P_f}{k_{mat} \cdot A_{c;f}}} \quad (D.198)$$

$$hi_{f;c;L2} = (hi_{c;L2}) \quad (D.199)$$

$$q_{max;f;c;L2} = hi_{f;c;L2} \cdot (P_f \cdot L) \cdot abs(T_{b;c;L2} - T_{\infty;c;L2}) \quad (D.200)$$

$$T_{b;c;L2} = \frac{T_{w;c;mid} + T_{w;c;i}}{2} \quad (D.201)$$

$$T_{\infty;c;L2} = \bar{T}_{\infty;c;L2} \quad (\text{D.202})$$

$$q_{f;c;L2} = \eta_{f;c;L2} \cdot q_{max;f;c;L2} \quad (\text{D.203})$$

$$M1_{c;L2} = \frac{q_{f;c;L2}}{(\tanh(m_{c;L2} \cdot L))} \quad (\text{D.204})$$

$$\theta_{c;L2} = \frac{M1_{c;L2}}{(\sqrt{hi_{f;c;L2} \cdot P_f \cdot k_{mat} \cdot A_{c,f}})} \quad (\text{D.205})$$

$$T_{fin;c;L2} = \theta_{c;L2} + T_{\infty;c;L2} \quad (\text{D.206})$$

$$\eta_{t;c;L2} = 1 - (1 - \eta_{f;c;L2}) \cdot A_{f;L2}/A_{L2} \quad (\text{D.207})$$

Resistance

Convective Resistance

$$hi_{h;L1} = \frac{\nu_{h;L1} \cdot k_{h;L1}}{D_h} \quad \text{Estimate convective coef} \quad (\text{D.208})$$

$$hi_{c;L1} = \frac{\nu_{c;L1} \cdot k_{c;L1}}{D_h} \quad (\text{D.209})$$

$$Rc_{h;L1} = \frac{1}{(\eta_{t;h;L1} \cdot hi_{h;L1} \cdot A_{ht;L1})} \quad \text{Convective resistance} \quad (\text{D.210})$$

$$Rc_{c;L1} = \frac{1}{\eta_{t;c;L1} \cdot hi_{c;L1} \cdot A_{ht;L1}} \quad (\text{D.211})$$

$$hi_{h;L2} = \frac{\nu_{h;L2} \cdot k_{h;L2}}{D_h} \quad \text{Estimate convective coef} \quad (\text{D.212})$$

$$hi_{c;L2} = \frac{\nu_{c;L2} \cdot k_{c;L2}}{D_h} \quad (\text{D.213})$$

$$Rc_{h;L2} = \frac{1}{(\eta_{t;h;L2} \cdot hi_{h;L2} \cdot A_{ht;L2})} \quad \text{Convective resistance} \quad (\text{D.214})$$

$$Rc_{c;L2} = \frac{1}{\eta_{t;c;L2} \cdot hi_{c;L2} \cdot A_{ht;L2}} \quad (\text{D.215})$$

$$St_{h;L1} = \frac{\nu_{h;L1}}{(Re_{h;L1} \cdot Pr_{h;L1})} \quad \text{Stanton Number} \quad (\text{D.216})$$

$$hi_{h;L1;st} = St_{h;L1} \cdot \rho_{h;L1} \cdot cp_{h;L1} \cdot v_{chl;h;L1} \quad \text{Convective coefficient - St no.} \quad (\text{D.217})$$

Conductive Resistance

$$Rw_{L1} = \frac{HX_{chl;w}}{(k_{mat} \cdot A_{ht;L1})} \quad \text{Conductive resistance} \quad (\text{D.218})$$

$$Rw_{L2} = \frac{HX_{chl;w}}{k_{mat} \cdot A_{ht;L2}} \quad (\text{D.219})$$

Total Resistance

$$R_{T;L1} = R_{c_{h;L1}} + R_{w_{L1}} + R_{c_{c;L1}} \quad (D.220)$$

$$R_{T;L2} = R_{c_{h;L2}} + R_{w_{L2}} + R_{c_{c;L2}} \quad (D.221)$$

Capacity Rate

$$C_{h;L1} = \dot{m}_{h;PS} \cdot cp_{h;L1} \quad (D.222)$$

$$C_{c;L1} = \dot{m}_{c;PS} \cdot cp_{c;L1} \quad (D.223)$$

$$C_{min;L1} = \min(C_{h;L1}; C_{c;L1}) \quad (D.224)$$

$$C_{max;L1} = \max(C_{h;L1}; C_{c;L1}) \quad (D.225)$$

$$C_{L1}^* = C_{min;L1}/C_{max;L1} \quad \text{Capacity Ratio} \quad (D.226)$$

$$C_{h;L2} = \dot{m}_{h;PS} \cdot cp_{h;L2} \quad (D.227)$$

$$C_{c;L2} = \dot{m}_{c;PS} \cdot cp_{c;L2} \quad (D.228)$$

$$C_{min;L2} = \min(C_{h;L2}; C_{c;L2}) \quad (D.229)$$

$$C_{max;L2} = \max(C_{h;L2}; C_{c;L2}) \quad (D.230)$$

$$C_{L2}^* = C_{min;L2}/C_{max;L2} \quad (D.231)$$

Effectiveness

$$\epsilon_{L1} = \frac{NTU_{L1}}{(NTU_{L1} + 1)} \quad C^*=1 \quad (D.232)$$

$$\epsilon_{L2} = \frac{NTU_{L2}}{(NTU_{L2} + 1)} \quad C^*=1 \quad (D.233)$$

$$\epsilon = \frac{q_{L1} + q_{L2}}{q_{max}} \quad (D.234)$$

$$NTU_{L1} = UA_{L1}/C_{min;L1} \quad (D.235)$$

$$NTU_{L2} = UA_{L2}/C_{min;L2} \quad (D.236)$$

$$NTU = UA/C_{min;L1} \quad (D.237)$$

Heat Transfer

$$q_{hot} = \dot{m}_{h;PS} \cdot cp_{h;mid} \cdot (T_{\infty;h;in} - T_{\infty;h;out}) \quad \text{q based on mass flow full length} \quad (D.238)$$

$$q_{cold} = \dot{m}_{c;PS} \cdot cp_{c;mid} \cdot (T_{\infty;c;out} - T_{\infty;c;in}) \quad (D.239)$$

$$q_{max} = \left(\frac{C_{min;L1} + C_{min;L2}}{2} \right) \cdot (T_{\infty;h;in} - T_{\infty;c;in}) \quad (D.240)$$

Maximum temperature difference

$$q_{L1} = \epsilon_{L1} \cdot q_{max} \quad \text{Solving } q_{L1} \quad (D.241)$$

$$q_{L2} = \epsilon_{L2} \cdot q_{max} \quad \text{Solving } q_{L2} \quad (D.242)$$

$$T_{\infty;h;mid} = T_{\infty;h;in} - \left(\frac{q_{L1}}{\dot{m}_{h;PS} \cdot cp_{h;L1}} \right) \quad \text{Solving } T_{h;mid} \quad (D.243)$$

$$T_{\infty;h;out} = T_{\infty;h;mid} - \left(\frac{q_{L2}}{\dot{m}_{h;PS} \cdot cp_{h;L2}} \right) \quad \text{Solving } T_{h;out} \quad (D.244)$$

$$T_{\infty;c;mid} = T_{\infty;c;in} + \left(\frac{q_{L2}}{\dot{m}_{c;PS} \cdot cp_{c;L2}} \right) \quad \text{Solving } T_{c;mid} \quad (D.245)$$

$$T_{\infty;c;out} = T_{\infty;c;mid} + \left(\frac{q_{L1}}{\dot{m}_{c;PS} \cdot cp_{c;L1}} \right) \quad \text{Solving } T_{c;out} \quad (D.246)$$

$$\Delta T_h = T_{\infty;h;in} - T_{\infty;h;out} \quad \text{Solving Delta } T_h \quad (D.247)$$

$$\Delta T_c = T_{\infty;c;out} - T_{\infty;c;in} \quad \text{Solving Delta } T_c \quad (D.248)$$

$$q_{LMTD} = UA \cdot \Delta T_m \quad \text{q based on LMTD} \quad (D.249)$$

$$\Delta T_m = \frac{(T_{\infty;h;out} - T_{\infty;c;in}) - (T_{\infty;h;in} - T_{\infty;c;out})}{\ln \left(\frac{T_{\infty;h;out} - T_{\infty;c;in}}{T_{\infty;h;in} - T_{\infty;c;out}} \right)} \quad (D.250)$$

Pressure Drop

Hot Side

$$\Delta P_{h;L1} = \left(\frac{G_h^2}{2 \cdot \rho_{h;in}} \right) \cdot (1 - \sigma_h^2 + K_{c;h}) + \left(f_{F;h;L1} \cdot \left(\frac{4 \cdot L_{chl;L1}}{D_h} \right) \right) \cdot (\rho_{h;in} / \rho_{h;m;L1}) + 2 \cdot ((\rho_{h;in} / \rho_{h;mid}) - 1) \quad (D.251)$$

$$\Delta P_{h;L2} = \left(\frac{G_h^2}{2 \cdot \rho_{h;mid}} \right) \cdot \left(f_{F;h;L2} \cdot \left(\frac{4 \cdot L_{chl;L2}}{D_h} \right) \right) \cdot (\rho_{h;mid} / \rho_{h;m;L2}) + 2 \cdot ((\rho_{h;mid} / \rho_{h;out}) - 1) - ((1 - \sigma_h^2 - K_{e;h}) \cdot (\rho_{h;mid} / \rho_{h;out})) \quad (D.252)$$

$$\Delta P_h = \Delta P_{90;h;L1} \cdot 2 + \Delta P_{GE;h} + \Delta P_{MI;h} + \Delta P_{h;L1} + \Delta P_{h;L2} + \Delta P_{MO;h} + \Delta P_{GC;h} + h_{f;90;h;L2} \cdot 2 + \Delta P_{U;h} \cdot 8 + \Delta P_{EL;h} \cdot 2 \quad (D.253)$$

$$f_{F;h;L1} = \frac{MoodyChart(Re_{h;L1}; RR)}{4} \quad (D.254)$$

$$f_{F;h;L2} = \frac{MoodyChart(Re_{h;L2}; RR)}{4} \quad (D.255)$$

$$P_{h;mid} = P_{h;in} - \Delta P_{h;L1} \quad (D.256)$$

$$P_{h;out} = P_{h;mid} - \Delta P_{h;L2} \quad (D.257)$$

$$K_{c;h} = 0,42 \cdot (1 - \sigma_h^2)^2 \quad (D.258)$$

$$K_{e;h} = (1 - \sigma_h)^2 \quad (D.259)$$

Cold Side

$$\Delta P_{c;L1} = \left(\frac{G_c^2}{2 \cdot \rho_{c;mid}} \right) \cdot \left(f_{F;c;L1} \cdot \left(\frac{4 \cdot L_{chl;L1}}{D_h} \right) \right) \cdot (\rho_{c;mid} / \rho_{c;m;L1}) + 2 \cdot ((\rho_{c;mid} / \rho_{c;out}) - 1) - (1 - \sigma_c^2 - K_{e;c}) \cdot (\rho_{c;mid} / \rho_{c;out}) \quad (D.260)$$

$$\Delta P_{c;L2} = \left(\frac{G_c^2}{2 \cdot \rho_{c;in}} \right) \cdot (1 - \sigma_c^2 + K_{c;h}) + \left(f_{F;c;L2} \cdot \left(\frac{4 \cdot L_{chl;L1}}{D_h} \right) \right) \cdot (\rho_{c;in} / \rho_{c;m;L2}) + 2 \cdot ((\rho_{c;in} / \rho_{c;mid}) - 1) \quad (D.261)$$

$$\Delta P_c = \Delta P_{90;c;L1} \cdot 2 + \Delta P_{GE;c} + \Delta P_{MI;c} + \Delta P_{c;L1} + \Delta P_{c;L2} + \Delta P_{MO;c} + \Delta P_{GC;c} + h_{f;90;c;L2} \cdot 2 + \Delta P_{U;c} \cdot 8 + \Delta P_{EL;c} \cdot 2 \quad (D.262)$$

$$f_{F;c;L1} = \frac{MoodyChart(Re_{c;L1}; RR)}{4} \quad (D.263)$$

$$f_{F;c;L2} = \frac{MoodyChart(Re_{c;L2}; RR)}{4} \quad (D.264)$$

$$P_{c;mid} = P_{c;in} - \Delta P_{c;L2} \quad (D.265)$$

$$P_{c;out} = P_{c;mid} - \Delta P_{c;L1} \quad (D.266)$$

$$K_{c;c} = 0,42 \cdot (1 - \sigma_c^2)^2 \quad (D.267)$$

$$K_{e;c} = (1 - \sigma_c)^2 \quad (D.268)$$

Minor Losses

$$f_t = 0,027 \quad \text{Friction factor nominal pipe size} \quad (D.269)$$

$$g_{c1} = 1 \quad (D.270)$$

$$V_{90;h} = \frac{\dot{m}_{h;PS}}{\rho_{h;in} \cdot a_{1;L1}} \quad (D.271)$$

$$V_{90;c} = \frac{\dot{m}_{c;PS}}{\rho_{c;in} \cdot a_{1;L1}} \quad (D.272)$$

$$V_{p;h} = \frac{\dot{m}_{h;PS}/N_{r;h}}{\rho_{h;in} \cdot (0,001 \cdot 0,028)} \quad (D.273)$$

$$V_{p;c} = \frac{\dot{m}_{c;PS}/N_{r;c}}{\rho_{c;in} \cdot (0,001 \cdot 0,028)} \quad (D.274)$$

$$V_{GE;h} = V_{90;h} \quad (D.275)$$

$$V_{GE;c} = V_{90;c} \quad (D.276)$$

$$V_{GC;h} = V_{90;h} \quad (D.277)$$

$$V_{GC;c} = V_{90;c} \quad (D.278)$$

$$K_{90} = 30 \cdot f_t \quad (D.279)$$

$$K_{MI} = 0,5 \quad (D.280)$$

$$K_{MO} = 1 \quad (D.281)$$

$$K_U = 0,04 \quad (D.282)$$

$$K_{EL} = 60 \cdot f_t \quad (D.283)$$

$$K_{GE} = \frac{(1 - \beta^2)^2}{\beta^4} \quad (D.284)$$

$$\beta^2 = a_{1;L1}/a_{2;L1} \quad (D.285)$$

$$a_{1;L1} = \pi \cdot 0,00485^2 \quad (D.286)$$

$$a_{2;L1} = 0,011 \cdot 0,02 \quad (D.287)$$

$$K_{GC} = \frac{0,5 \cdot (1 - \beta^2) \cdot \sqrt{\sin(\theta_p/2)}}{\beta^4} \quad (D.288)$$

$$\theta_p = 103 \quad (D.289)$$

HOT

$$h_{f;90;h;L1} = K_{90} \cdot \left(\frac{V_{90;h}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss 90 hot inlet} \quad (D.290)$$

$$\Delta P_{90;h;L1} = \rho_{h;in} \cdot h_{f;90;h;L1} \quad \text{Pressure Drop 90 hot inlet} \quad (\text{D.291})$$

$$h_{f;90;h;L2} = K_{90} \cdot \left(\frac{V_{90;h}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss 90 hot outlet} \quad (\text{D.292})$$

$$\Delta P_{90;h;L2} = \rho_{h;out} \cdot h_{f;90;h;L2} \quad \text{Pressure Drop 90 hot outlet} \quad (\text{D.293})$$

$$h_{f;GE;h;L1} = K_{GE} \cdot \left(\frac{V_{GE;h}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss Gradual Enlargement} \quad (\text{D.294})$$

$$\Delta P_{GE;h} = \rho_{h;in} \cdot h_{f;GE;h;L1} \quad \text{Pressure Drop Gradual Enlargement} \quad (\text{D.295})$$

$$h_{f;GC;h;L2} = K_{GC} \cdot \left(\frac{V_{GC;h}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss Gradual Contraction} \quad (\text{D.296})$$

$$\Delta P_{GC;h} = \rho_{h;out} \cdot h_{f;GC;h;L2} \quad \text{Pressure Drop Gradual Contraction} \quad (\text{D.297})$$

$$h_{f;h;L1} = K_{MI} \cdot \left(\frac{V_{p;h}^2}{2 \cdot g_{c1}} \right) \cdot N_{r;h} \quad \text{Head Loss Manifold inlet} \quad (\text{D.298})$$

$$\Delta P_{MI;h} = \rho_{h;in} \cdot h_{f;h;L1} \quad \text{Pressure Drop Manifold inlet} \quad (\text{D.299})$$

$$h_{f;h;L2} = K_{MO} \cdot \left(\frac{V_{p;h}^2}{2 \cdot g_{c1}} \right) \cdot N_{r;h} \quad \text{Head Loss Manifold outlet} \quad (\text{D.300})$$

$$\Delta P_{MO;h} = \rho_{h;out} \cdot h_{f;h;L2} \quad \text{Pressure Drop Manifold outlet} \quad (\text{D.301})$$

$$h_{f;u;h} = K_U \cdot \left(\frac{V_{90;h}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss Coupling} \quad (\text{D.302})$$

$$\Delta P_{U;h} = \rho_{h;mid} \cdot h_{f;u;h} \quad \text{Pressure Drop Coupling} \quad (\text{D.303})$$

$$h_{f;EL;h} = K_{EL} \cdot \left(\frac{V_{90;h}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss Elbow} \quad (\text{D.304})$$

$$\Delta P_{EL;h} = \rho_{h;mid} \cdot h_{f;EL;h} \quad \text{Pressure Drop Elbow} \quad (\text{D.305})$$

COLD

$$h_{f;90;c;L1} = K_{90} \cdot \left(\frac{V_{90;c}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss 90 cold inlet} \quad (\text{D.306})$$

$$\Delta P_{90;c;L1} = \rho_{c;in} \cdot h_{f;90;c;L1} \quad \text{Pressure Drop 90 cold inlet} \quad (\text{D.307})$$

$$h_{f;90;c;L2} = K_{90} \cdot \left(\frac{V_{90;c}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss 90 hot outlet} \quad (\text{D.308})$$

$$\Delta P_{90;c;L2} = \rho_{c;out} \cdot h_{f;90;c;L2} \quad \text{Pressure Drop 90 hot outlet} \quad (\text{D.309})$$

$$h_{f;GE;c;L1} = K_{GE} \cdot \left(\frac{V_{GE;c}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss Gradual Enlargement} \quad (\text{D.310})$$

$$\Delta P_{GE;c} = \rho_{c;in} \cdot h_{f;GE;c;L1} \quad \text{Pressure Drop Gradual Enlargement} \quad (\text{D.311})$$

$$h_{f;GC;c;L2} = K_{GC} \cdot \left(\frac{V_{GC;c}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss Gradual Contraction} \quad (\text{D.312})$$

$$\Delta P_{GC;c} = \rho_{c;out} \cdot h_{f;GC;c;L2} \quad \text{Pressure Drop Gradual Contraction} \quad (\text{D.313})$$

$$h_{f;c;L1} = K_{MI} \cdot \left(\frac{V_{p;c}^2}{2 \cdot g_{c1}} \right) \cdot N_{r;c} \quad \text{Head Loss Manifold inlet} \quad (\text{D.314})$$

$$\Delta P_{MI;c} = \rho_{h;in} \cdot h_{f;h;L1} \quad \text{Pressure Drop Manifold inlet} \quad (\text{D.315})$$

$$h_{f;c;L2} = K_{MO} \cdot \left(\frac{V_{p;c}^2}{2 \cdot g_{c1}} \right) \cdot N_{r;c} \quad \text{Head Loss Manifold outlet} \quad (\text{D.316})$$

$$\Delta P_{MO;c} = \rho_{c;out} \cdot h_{f;c;L2} \quad \text{Pressure Drop Manifold outlet} \quad (\text{D.317})$$

$$h_{f;u;c} = K_U \cdot \left(\frac{V_{90;c}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss Coupling} \quad (\text{D.318})$$

$$\Delta P_{U;c} = \rho_{c;mid} \cdot h_{f;u;c} \quad \text{Pressure Drop Coupling} \quad (\text{D.319})$$

$$h_{f;EL;c} = K_{EL} \cdot \left(\frac{V_{90;c}^2}{2 \cdot g_{c1}} \right) \quad \text{Head Loss Elbow} \quad (\text{D.320})$$

$$\Delta P_{EL;c} = \rho_{c;mid} \cdot h_{f;EL;c} \quad \text{Pressure Drop Elbow} \quad (\text{D.321})$$

END

Appendix E

CFD: Full report

Heat Exchanger CFD



SOLIDWORKS Flow Simulation Project Report

September 11, 2023



Fluid Flow Simulation Report

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Fluid Flow Simulation Report

1 General Information

Objective of the simulation: To get a better understanding of the fluid flow within the heat exchanger. Results used to validate theoretical model and characterize the heat exchanger.

1.1 Analysis Environment

Software Product:	Flow Simulation 2021 SP3.0. Build: 5075
CPU Type:	Intel(R) Core(TM) i5-10300H CPU @ 2.50GHz
CPU Speed:	2496 MHz
RAM:	16291 MB / 9731 MB
Operating System:	Windows 10 (or higher) (Version 10.0.22621)

1.2 Model Information

Model Name:	1mm Design, Optimized for CFD.SLDPRT
Project Name:	C=0.75 100% Flow

1.3 Project Comments:

Unit System:	SI (m-k-g-s)
Analysis Type:	Internal

1.4 Size of Computational Domain

Size

X min	-0.034 m
X max	0.034 m
Y min	-0.013 m
Y max	0.013 m
Z min	-0.060 m
Z max	0.060 m
X size	0.068 m
Y size	0.026 m
Z size	0.120 m

Fluid Flow Simulation Report

1.5 Simulation Parameters

1.5.1 Mesh Settings

1.5.1.1 Basic Mesh

Basic Mesh Dimensions

Number of cells in X	38
Number of cells in Y	14
Number of cells in Z	66

1.5.1.2 Analysis Mesh

Total Cell count:	1033556
Fluid Cells:	478519
Solid Cells:	555037
Partial Cells:	300920
Trimmed Cells:	0

1.5.1.3 Additional Physical Calculation Options

Heat Transfer Analysis:	Heat conduction in solids: On Heat conduction in solids only: Off
Flow Type:	Laminar and turbulent
Time-Dependent Analysis:	Off
Gravity:	Off
Radiation:	Off
Default Wall Roughness:	14.0 micrometer

1.5.2 Material Settings

Fluids: Water

Solids: Ti6Al4V

1.5.3 Initial Conditions

Initial Conditions

Thermodynamic parameters	Static Pressure: 101325.00 Pa Temperature: 20.05 °C
Velocity parameters	Velocity vector Velocity in X direction: 0 m/s Velocity in Y direction: 0 m/s Velocity in Z direction: 0 m/s
Solid parameters	Default material: Ti-6Al-4V Initial solid temperature: 20.05 °C
Turbulence parameters	Turbulence intensity and length Intensity: 2.00 % Length: 2.564e-04 m

Fluid Flow Simulation Report

1.5.4 Boundary Conditions

Boundary Conditions

Cold Inlet 17

Type	Inlet Mass Flow
Faces	LID4//Face
Coordinate system	Face Coordinate System
Reference axis	X
Flow parameters	Flow vectors direction: Normal to face Mass flow rate: 0.0439 kg/s Fully developed flow: No Inlet profile: 0
Thermodynamic parameters	Temperature type: Temperature of initial components Temperature: 20.00 °C
Turbulence parameters	Turbulence intensity and length Intensity: 2.00 % Length: 2.564e-04 m
Boundary layer parameters	Boundary layer type: Turbulent

Cold Outlet

Type	Environment Pressure
Faces	LID3//Face
Coordinate system	Face Coordinate System
Reference axis	X
Thermodynamic parameters	Environment pressure: 101325.00 Pa Temperature type: Temperature of initial components Temperature: 20.00 °C
Turbulence parameters	Turbulence intensity and length Intensity: 2.00 % Length: 2.564e-04 m
Boundary layer parameters	Boundary layer type: Turbulent

Hot Inlet 57

Type	Inlet Mass Flow
Faces	LID1//Face
Coordinate system	Face Coordinate System
Reference axis	X
Flow parameters	Flow vectors direction: Normal to face Mass flow rate: 0.0329 kg/s Fully developed flow: No Inlet profile: 0

Fluid Flow Simulation Report

Thermodynamic parameters	Temperature type: Temperature of initial components Temperature: 57.00 °C
Turbulence parameters	Turbulence intensity and length Intensity: 2.00 % Length: 2.564e-04 m
Boundary layer parameters	Boundary layer type: Turbulent

Hot Outlet

Type	Environment Pressure
Faces	LID2 // Face
Coordinate system	Face Coordinate System
Reference axis	X
Thermodynamic parameters	Environment pressure: 101325.00 Pa Temperature type: Temperature of initial components Temperature: 57.00 °C
Turbulence parameters	Turbulence intensity and length Intensity: 2.00 % Length: 2.564e-04 m
Boundary layer parameters	Boundary layer type: Turbulent

1.5.5 Engineering Goals

Goals

Global Goals

GG Bulk Av Total Pressure 2

Type	Global Goal
Goal type	Total Pressure
Calculate	Average value
Coordinate system	Global Coordinate System
Criteria	1.00 Pa
Use in convergence	On

GG Bulk Av Temperature (Fluid) 4

Type	Global Goal
Goal type	Temperature (Fluid)
Calculate	Average value
Coordinate system	Global Coordinate System
Criteria	-272.15 °C
Use in convergence	On

Fluid Flow Simulation Report

GG Bulk Av Density (Fluid) 8

Type	Global Goal
Goal type	Density (Fluid)
Calculate	Average value
Coordinate system	Global Coordinate System
Criteria	1.00 kg/m ³
Use in convergence	On

GG Mass Flow Rate 9

Type	Global Goal
Goal type	Mass Flow Rate
Coordinate system	Global Coordinate System
Criteria	1.0000 kg/s
Use in convergence	On

GG Average Velocity 10

Type	Global Goal
Goal type	Velocity
Calculate	Average value
Coordinate system	Global Coordinate System
Criteria	1.000 m/s
Use in convergence	On

GG Average Heat Transfer Coefficient 12

Type	Global Goal
Goal type	Heat Transfer Coefficient
Calculate	Average value
Coordinate system	Global Coordinate System
Criteria	1.000 W/m ² /K
Use in convergence	On

GG Average Wall Temperature 13

Type	Global Goal
Goal type	Wall Temperature
Calculate	Average value
Coordinate system	Global Coordinate System
Criteria	-272.15 °C
Use in convergence	On

Fluid Flow Simulation Report

GG Bulk Av Temperature (Solid) 17

Type	Global Goal
Goal type	Temperature (Solid)
Calculate	Average value
Coordinate system	Global Coordinate System
Criteria	-272.15 °C
Use in convergence	On

Surface Goals

SG Cold Inlet 17 Total Pressure Av

Type	Surface Goal
Goal type	Total Pressure
Calculate	Average value
Faces	Face<1>
Coordinate system	Global Coordinate System
Criteria	1.00 Pa
Use in convergence	On

SG Cold Inlet 17 Velocity Av

Type	Surface Goal
Goal type	Velocity
Calculate	Average value
Faces	Face<1>
Coordinate system	Global Coordinate System
Criteria	1.000 m/s
Use in convergence	On

SG Cold Outlet Mass Flow Rate

Type	Surface Goal
Goal type	Mass Flow Rate
Faces	Face<1>
Coordinate system	Global Coordinate System
Criteria	1.0000 kg/s
Use in convergence	On

SG Cold Outlet Velocity Av

Type	Surface Goal
Goal type	Velocity
Calculate	Average value
Faces	Face<1>
Coordinate system	Global Coordinate System
Criteria	1.000 m/s

Fluid Flow Simulation Report

Use in convergence	On
--------------------	----

SG Cold Outlet Temperature (Fluid) Bulk Average

Type	Surface Goal
Goal type	Temperature (Fluid)
Calculate	Average value
Faces	Face<1>
Coordinate system	Global Coordinate System
Criteria	-272.15 °C
Use in convergence	On

SG Hot Inlet 57 Total Pressure Av

Type	Surface Goal
Goal type	Total Pressure
Calculate	Average value
Faces	Face<1>
Coordinate system	Global Coordinate System
Criteria	1.00 Pa
Use in convergence	On

SG Hot Inlet 57 Velocity Av

Type	Surface Goal
Goal type	Velocity
Calculate	Average value
Faces	Face<1>
Coordinate system	Global Coordinate System
Criteria	1.000 m/s
Use in convergence	On

SG Hot Outlet Mass Flow Rate

Type	Surface Goal
Goal type	Mass Flow Rate
Faces	Face<1>
Coordinate system	Global Coordinate System
Criteria	1.0000 kg/s
Use in convergence	On

Fluid Flow Simulation Report

SG Hot Outlet Velocity Av

Type	Surface Goal
Goal type	Velocity
Calculate	Average value
Faces	Face<1>
Coordinate system	Global Coordinate System
Criteria	1.000 m/s
Use in convergence	On

SG Hot Outlet Temperature (Fluid) Bulk Average

Type	Surface Goal
Goal type	Temperature (Fluid)
Calculate	Average value
Faces	Face<1>
Coordinate system	Global Coordinate System
Criteria	-272.15 °C
Use in convergence	On

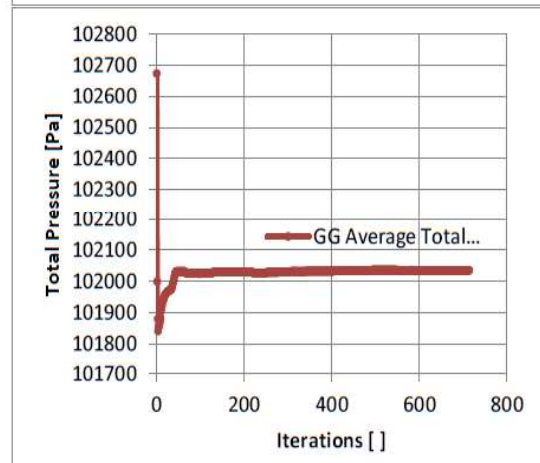
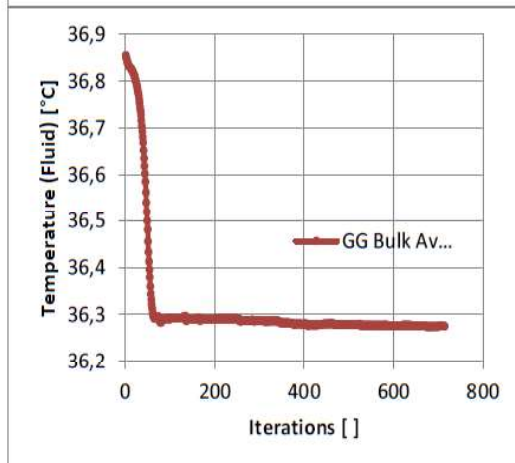
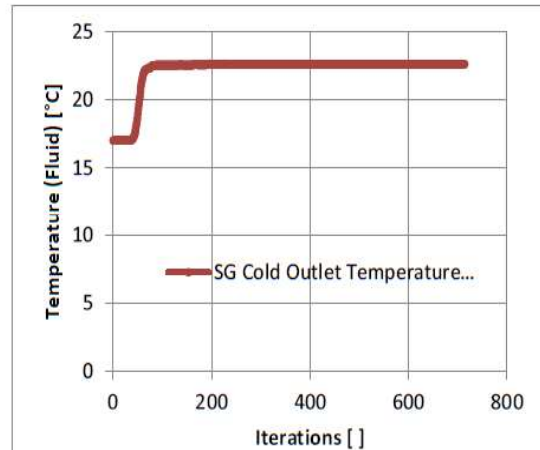
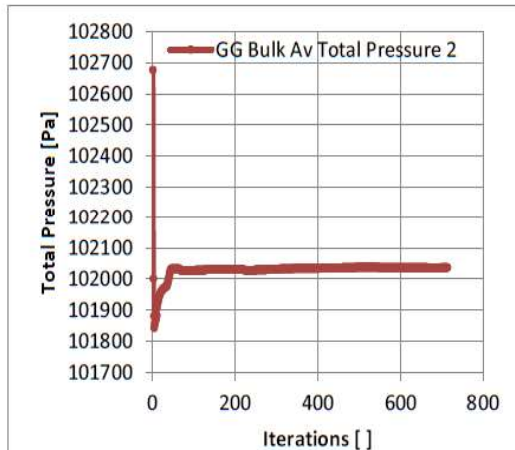
Fluid Flow Simulation Report

1.6 Analysis Time

Calculation Time: 16472 s

Number of Iterations: 714

Warnings:



Fluid Flow Simulation Report

2 Results

2.1 Analysis Goals

Goals

Name	Unit	Value	Progress	Criteria	Delta	Use in convergence
GG Bulk Av Total Pressure 2	Pa	102035.68	100	5.71049301	0.484964186	On
GG Bulk Av Temperature (Fluid) 4	°C	36.28	100	0.0168914038	0.00118866826	On
GG Average Density (Fluid) 7	kg/m ³	991.92	100	0.0180642463	0.000513694741	On
GG Bulk Av Density (Fluid) 8	kg/m ³	991.95	100	0.0175721731	0.0005092503	On
GG Average Velocity 10	m/s	0.401	100	0.000660486823	0.000168798277	On
GG Bulk Av Velocity 11	m/s	0.401	100	0.000658122616	0.000168281801	On
GG Average Heat Transfer Coefficient 12	W/m ² /K	5418.853	100	16174.982	1191.5588	On
GG Average Wall Temperature 13	°C	36.51	100	0.3360885	0.00135145568	On
GG Average Temperature (Solid) 16	°C	36.54	100	0.40275593	0.00322585786	On
GG Bulk Av Temperature (Solid) 17	°C	36.54	100	0.40275593	0.00322585787	On
SG Cold Inlet 17 Total Pressure Av	Pa	103090.27	100	15.3331086	0.407857384	On
SG Cold Inlet 17 Velocity Av	m/s	0.428	100	4.27548924e-09	3.88578059e-16	On
SG Cold Outlet Mass Flow Rate	kg/s	-0.0439	100	4.38665278e-05	4.74324347e-08	On

Fluid Flow Simulation Report

SG Cold Outlet Velocity Av	m/s	0.431	100	0.000197950925	0.000130522915	On
SG Cold Outlet Temperature (Fluid) Bulk Average	°C	22.59	100	0.167247621	0.00243449022	On
SG Hot Inlet 57 Total Pressure Av	Pa	102220.75	100	8.12602918	1.63093043	On
SG Hot Inlet 57 Velocity Av	m/s	0.325	100	3.2520895e-09	2.77555756e-16	On
SG Hot Outlet Mass Flow Rate	kg/s	-0.0329	100	3.29000299e-05	6.13278365e-08	On
SG Hot Outlet Velocity Av	m/s	0.327	100	0.000150696257	0.000148698153	On
SG Hot Outlet Temperature (Fluid) Bulk Average	°C	49.55	100	0.223026605	0.0041540413	On

Fluid Flow Simulation Report

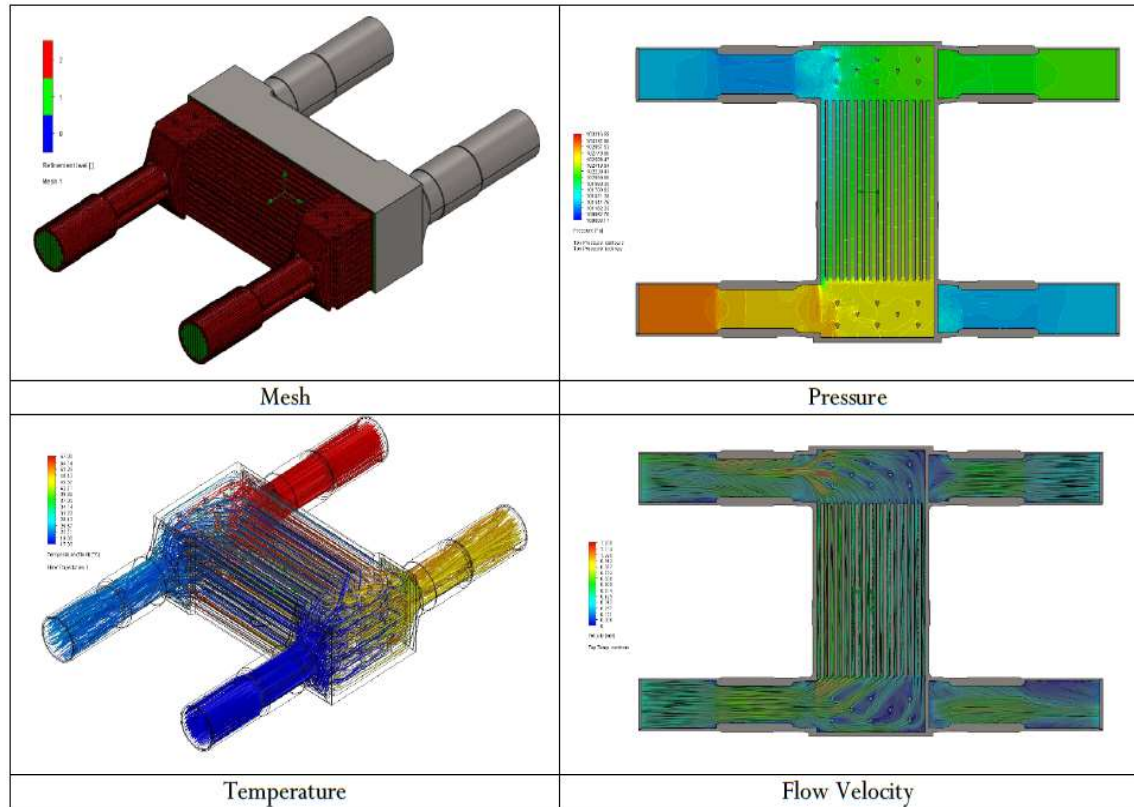
2.2 Global Min-Max-Table

Min/Max Table

Name	Minimum	Maximum
Density (Fluid) [kg/m ³]	984.33	998.29
Density (Solid) [kg/m ³]	4428.78	4428.78
Pressure [Pa]	100803.17	103316.59
Temperature [°C]	17.00	57.00
Temperature (Fluid) [°C]	17.00	57.00
Temperature (Solid) [°C]	17.00	57.00
Velocity [m/s]	0	1.200
Velocity (X) [m/s]	-1.015	0.783
Velocity (Y) [m/s]	-0.823	0.733
Velocity (Z) [m/s]	-1.121	1.152
Velocity RRF (X) [m/s]	-1.015	0.783
Velocity RRF (Y) [m/s]	-0.823	0.733
Velocity RRF (Z) [m/s]	-1.121	1.152
Relative Pressure [Pa]	-521.83	1991.59
Shear Stress [Pa]	0	52.42
Bottleneck Number []	2.4758579e-23	1.0000000
Heat Flux [W/m ²]	3.378e-06	231253.191
Heat Transfer Coefficient [W/m ² /K]	0	2.306e+08
Overheat above Melting Temperature [°C]	-9709.850	-9669.850
ShortCut Number []	6.2571588e-26	1.0000000
Surface Heat Flux [W/m ²]	-270680.743	202918.986
Surface Heat Flux (Conductive) [W/m ²]	0	0
Surface Heat Flux (Convective) [W/m ²]	-270680.743	202918.986
Total Enthalpy Flux [W/m ²]	-8.902e+08	5.225e+08
Acoustic Power [W/m ³]	2.379e-63	9.743e-15
Acoustic Power Level [dB]	0	0

Fluid Flow Simulation Report

2.3 Results



2.4 Conclusion

The results converged on iteration 714 successfully. The data will be analysed and compared to the theoretical calculations results obtain from EES. Similar CFD's was run for the other flow rates and heat exchangers.

Fluid Flow Simulation Report

3 Appendix

3.1 Material Data

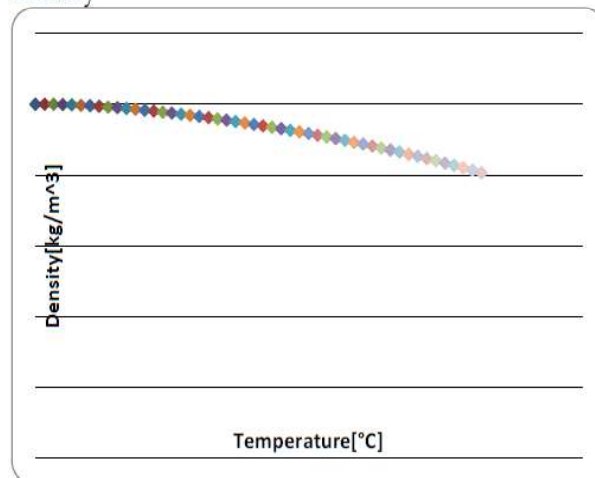
Engineering Database

Liquids

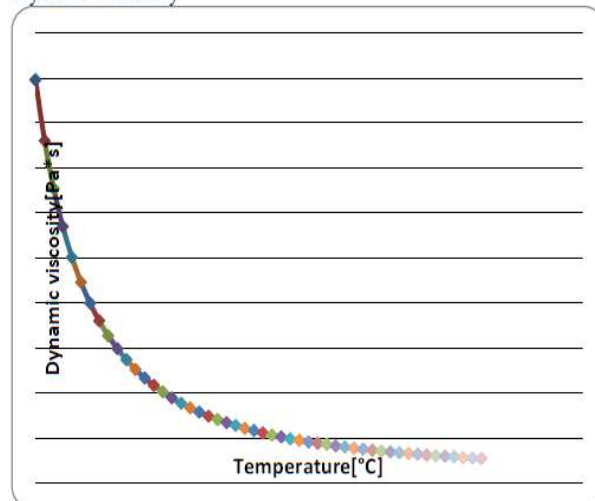
Water

Path: Liquids Pre-Defined

Density

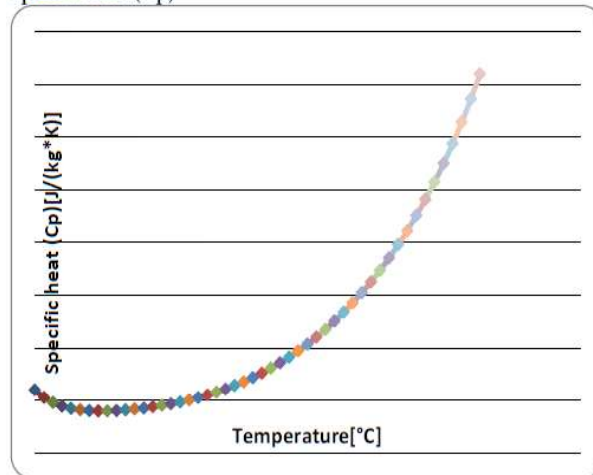


Dynamic viscosity

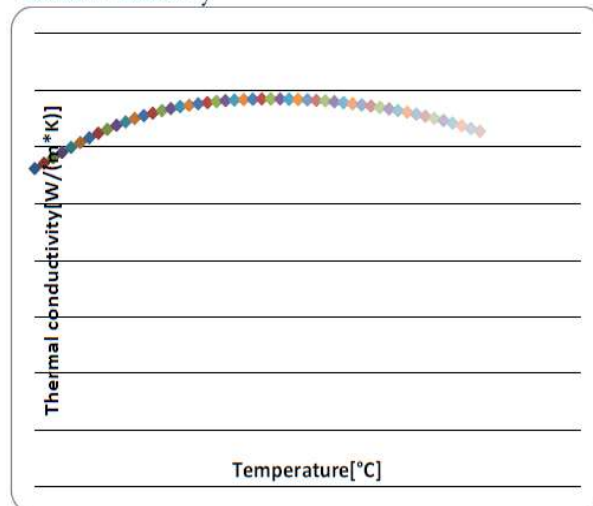


Fluid Flow Simulation Report

Specific heat (C_p)



Thermal conductivity



Cavitation effect: Yes

Temperature: -273.15 °C

Saturation pressure: 0 Pa

Radiation properties: No

Solids

Ti-6Al-4V

Path: Solids User Defined\SOLIDWORKS Material

Density: 4428.78 kg/m³

Specific heat: 586.0 J/(kg*K)

Conductivity type: Isotropic

Thermal conductivity: 6.7000 W/(m*K)

Electrical conductivity: Dielectric

Radiation properties: No

Melting temperature: No

Appendix F

Test sheet sample

Test sheet: 75 C=0.25

Test No	1			Checklist	
Date	17/10/2022			Ball Valves Open	Yes Yes
T _{ambient}	22.67/22.38/ 22.34/22.66	Celsius		Needle Valve Return Open	Yes Yes
P _{atmosphere}	888/908/ 883/885	mbar		Needle Valve in-line Closed	Yes
Target Flow	Hot	1.5	l/min	Pipe and Fittings Checked	Yes
	Cold	5.9	l/min	Connections Secure	Yes
Target Heat Transfer (EES)	Hot	1517	W	Instruments Power On	Yes
	Cold	1517	W	Computer Connected	Yes
Hot Loop				PicoLog Start-up	Yes
T _{in}	57.89	Celsius		Switch On Heated Reservoir	Yes
T _{out}	47.15	Celsius			
P _{in}	1.22			Cold Inlet	Yes
P _{out}	1.18			Hot Inlet	Yes
Flow Rate	1.5	l/min		Air Bleed	Yes
Cold Loop				Adjust to Target Flow	Yes
T _{in}	20.40	Celsius			
T _{out}	23.08	Celsius		Steady State Time Start	00:38
P _{in}	1.90	bar		Steady State Time End	00:44
P _{out}	1.72	bar			
Flow Rate	5.9	l/min		Hot/Cold Feed Shut-off	Yes
Capture Time Start	00:44			Save PicoLog Data	Yes
Capture Time End	00:50	min		Shut-off Instruments	Yes
Capture Count	100	Samples		Shut-off PicoLog	Yes

Figure F.1: Test sheet sample

Appendix G

PicoLog sample recording

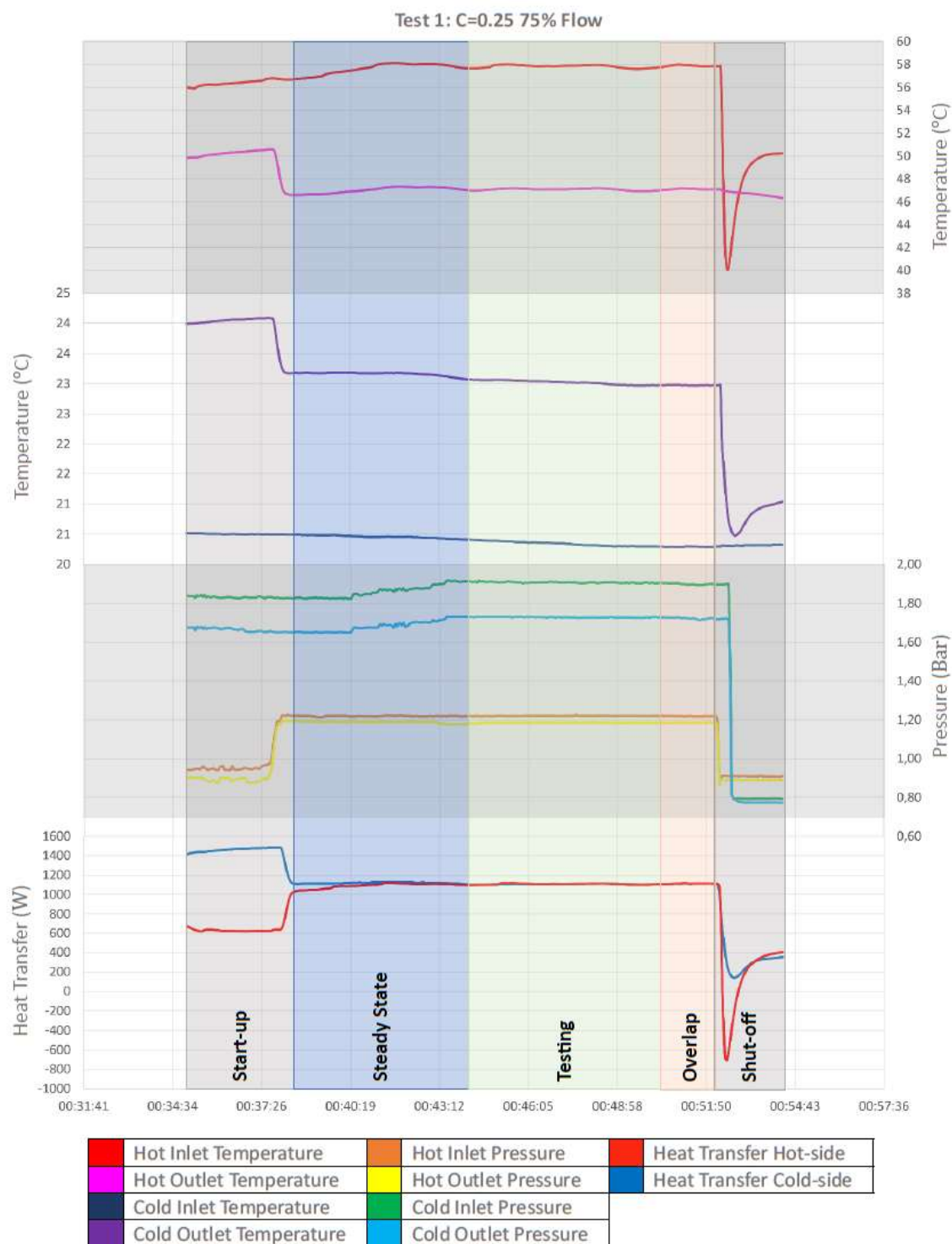


Figure G.1: PicoLog sample recording

Appendix H

RAPDASA papers

H.1 RAPDASA Paper 2018

S.C. Venter, G.G. Jacobs and J. du Preez, DESIGN CONSIDERATIONS FOR DEVELOPING AN ADDITIVE MANUFACTURED TI-6AL-4V COMPACT COUNTER-FLOW HEAT EXCHANGER FOR APPLICATION IN ORGANIC RANKINE CYCLES, 19th RAPDASA Annual International Conference, Parktonian Hotel, Johannesburg, 7-9 November 2018, Conference Proceedings, pp 192 - 200, ISBN: 978-0-620-80987-0

DESIGN CONSIDERATIONS FOR DEVELOPING AN ADDITIVE MANUFACTURED TI-6AL-4V COMPACT COUNTER-FLOW HEAT EXCHANGER FOR APPLICATION IN ORGANIC RANKINE CYCLES

S.C. Venter^{1*}, G.G. Jacobs² and J. du Preez³

ABSTRACT

Additive manufacturing has the ability to produce parts not previously possible with conventional methods. While this technology enables the production of highly complicated parts, there are still design considerations to conform to. A literature study indicated that there is little information available on how to design compact heat exchangers for additive manufacturing. This paper discusses the design considerations for producing compact counter-flow heat exchangers by means of additive manufacturing for application in Organic Rankine Cycles.

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1. INTRODUCTION

With the current high demand for energy, engineers all over the world are aiming to increase the efficiency of energy producing equipment [1]. Increasing the efficiency will reduce the production of greenhouse gasses, lessening the effect of global warming. Various methods exist to accomplish this. The method of interest in this study is the Organic Rankine Cycle (ORC). An ORC operates on a similar base as a conventional steam cycle for energy conversion, this cycle however uses organic fluid instead of water. It is implemented to improve the efficiency by utilising waste heat, captured by means of a heat exchanger (evaporator) and used to produce additional power via a generator [2,3].

Large scale ORCs are already implemented in numerous power plants over the world [4]. Micro ORCs are more compact systems that can be utilised by domestic users (1-10kW) [5]. Miniaturising such system would be possible by using additive manufacturing (AM) processes.

The AM process involves material being joined layer upon layer to produce the desired part [6]. This technology opens a world of opportunities to develop intricate parts as one solid, which was not previously possible with traditional manufacturing methods [7,8]. One technology that could substantially benefit from additive manufacturing is micro-channel heat exchangers. These devices achieve a very high surface area per unit volume due to small channel sizes.

Traditional methods for producing micro-channel heat exchangers is time consuming and are limited to basic designs [9]. With the ability to produce walls with thicknesses of less than 200 μm these devices can evolve from its former conventional manufactured designs to increase its effectiveness [10].

Implementing high effectiveness micro-channel heat exchangers in ORC allows a reduction in size of these cycles to implement on systems producing low grade waste heat such as flue gas. The main focus of the study is the development and characterisation of Ti-6Al-4V compact counter-flow heat exchangers for production with AM with application in ORC's.

Although additive manufacturing is a versatile technology, there still remain limitations to which a design should adhere to. This paper outlines the design considerations that were considered during the design of a Ti-6Al-4V compact heat exchanger.

2. LITERATURE REVIEW

2.1 Heat exchangers

Heat exchangers are devices used to transfer heat between two or more fluids, either to capture or reject excess heat. This study focused on compact counter-flow heat exchangers. The above-mentioned heat exchangers have a high effectiveness due to its ability to operate with close approach temperatures [11]. The ratio of the actual heat transferred to the maximum possible heat transfer is known as the effectiveness of a heat exchanger. High effectiveness is particularly beneficial in power-producing systems [12].

A simple heat exchanger design was considered for this study to minimise the effects of uncertainty and simplify the characterisation process. The characterisation of these heat exchangers would allow better understanding of the possibilities and limitations of additive manufactured compact heat exchangers.

2.2 Production methods

X-ray lithography (LIGA), chemical and silicon etching and diamond or wire machining are traditionally used to produce compact heat exchangers. These methods either limit the design complexity or require more than one process to produce a heat exchanger [9,10].

Additive manufacturing is a well suited method to create complex parts with great accuracy ($\pm 50 \mu\text{m}$) [13]. A well suited AM technology for creating metallic parts is selective laser sintering (SLS). Therefore, SLS is considered to be a viable method to produce intricate parts such as compact heat exchangers in a variety of metallic materials.

2.3 Material selection

Additive manufacturing possess the ability to produce parts in a variety of materials [6], with Titanium (Ti-6Al-4V) being the material of interest in this study. Ti-6Al-4V is a well suited material for producing compact heat exchangers, with its high specific strength ($\pm 930 \text{ MPa}$) the heat exchangers' channel walls can be thinned down significantly to reduce heat transfer resistance. Furthermore, Ti-6Al-4V has exceptional corrosion and good fluid erosion resistance and with its low thermal expansion coefficient ($\pm 8.6 \mu\text{m}/\text{m} \cdot ^\circ\text{C}$, $0 - 100^\circ\text{C}$), warpage and fatigue effects are minimised during thermal cycling [14].

3. METHODOLOGY AND DESIGN CONSIDERATIONS

Table 1: List of symbols.

Symbol	Description
ε	Heat exchanger effectiveness
q	Heat transfer
$\dot{m}$	Mass flow rate
c	Specific heat
T	Temperature
P	Pressure
G	Mass velocity
ρ	Fluid density
σ	Ratio of free-flow area to frontal area
$K_{c/e}$	Entrance/Exit loss coefficient
f_F	Fanning's friction factor
L	Length
D_h	Hydraulic diameter

3.1 Theoretical Design

A mathematical model for the preliminary design was developed with the effectiveness-NTU method. This method is generally preferred for analysis and design of heat exchangers [15].

The effectiveness of a heat exchanger is determined with equation 1:

$$\text{Effectiveness} = \varepsilon = \frac{\text{Actual Heat Transfer}}{\text{Maximum Heat Transfer}} \quad (1)$$

The actual heat transfer is calculated by either determining the energy lost from the hot fluid or the energy gained by the cold fluid, using Equation 2. Maximum heat transfer may be calculated with the maximum temperature difference within the heat exchanger, as shown in equation 3 [15].

$$q = \dot{m}_h c_h (T_{h_i} - T_{h_o}) = \dot{m}_c c_c (T_{c_o} - T_{c_i}) \quad (2)$$

$$q_{max} = (\dot{m}c)_{min} (T_{h_i} - T_{c_i}) \quad (3)$$

The core pressure drop across the heat exchanger is calculated by Equation 4. This equation includes entrance-, core friction-, flow acceleration- and exit- losses [16].

$$\Delta p = \frac{G^2}{2\rho_i} \left[(1 - \sigma_i^2 + K_c) + f_F \frac{4L}{D_h} \left(\frac{\rho_i}{\rho_m} \right) + 2 \left(\frac{\rho_i}{\rho_e} - 1 \right) - (1 - \sigma_e^2 - K_e) \left(\frac{\rho_i}{\rho_e} \right) \right] \quad (4)$$

With these equations a theoretical outcome can be predicted for a specific heat exchanger design. The aim of this study is to produce and characterise heat exchangers with three different channel sizes, 0.5, 1 and 2 mm which can transfer 3 kW of heat.

Mathematical models of the heat exchangers were created using Engineering Equation Solver (EES), using the above-mentioned equations. These models were then used as a guide line for the CAD models shown in Figures 1 to 9. The CAD models of the heat exchangers were sent to the Centre of Rapid Prototyping and Manufacturing (CRPM) at Central University of Technology, Free State for review.

3.1.1 Design 1

Details:

Heat exchanger width: 18.65 mm
Heat exchanger height: 11.00 mm
Channel length: 30.00 mm
Channel size: 0.50 mm

Number of channels: 20
Number of rows: 11
Estimated heat transfer: 3012 W
20 3.012 kW

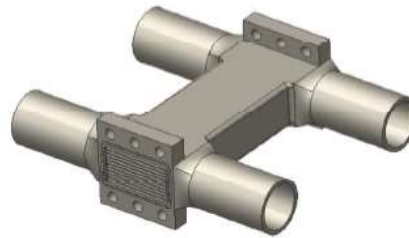


Figure 1: 3-D CAD model of design 1.

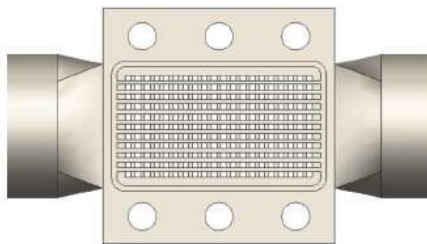


Figure 2: Side view of design 1.

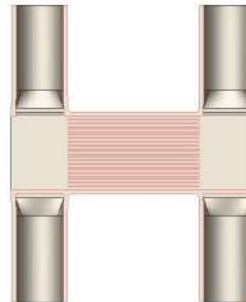


Figure 3: Top cross-section view of design 1.

3.1.2 Design 2

Details:

Heat exchanger width: 28.65 mm
Heat exchanger height: 21.90 mm
Channel length: 40.00 mm
Channel size: 1.00 mm

Number of channels: 20
Number of rows: 15
Estimated heat transfer: 3071 W
20 3.012 kW

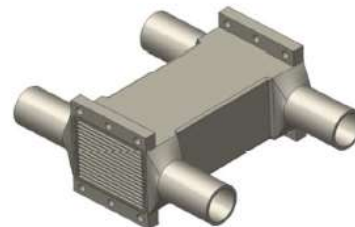


Figure 4: 3-D CAD model of design 2.

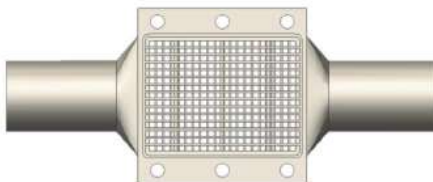


Figure 5: Side view of design 2.

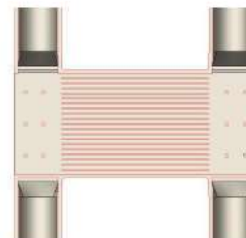


Figure 6: Top cross-section view of design 2.

3.1.3 Design 3

Details:

Heat exchanger width:	72.15	mm
Heat exchanger height:	72.15	mm
Channel length:	50.00	mm
Channel size:	2.00	mm
Number of channels:	30	
Number of rows:	30	
Estimated heat transfer:	3008	W
	20	3.012
		kW

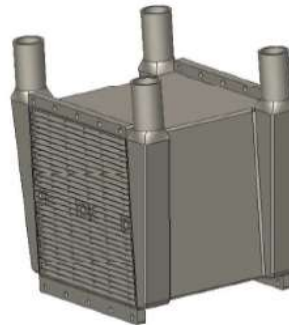


Figure 7: 3-D CAD model of design 3.



Figure 8: Side view of design 2.



Figure 9: Top cross-section view of design 2.

3.2 Design Considerations

Design reviews are of high importance as incompatible designs will not be manufactured as desired, some of these are described below. Process engineers of the CRPM reviewed and suggested improvements to the designs to make it compatible with SLS. The following section provides design considerations which should be considered when designing similar heat exchangers for production with additive manufacturing.

3.2.1 Manufacturing orientation

Selecting the correct manufacturing orientation is an important consideration for any AM product. Therefore it was the first consideration that was addressed by the CRPM. Manufacturing of heat exchangers with the channels in a horizontal orientation, (the arrows indicate manufacturing direction shown in Figure 10), will cause the channels to deform without adding supports. Figure 11 shows, indicated in red, how the channels will deform when the heat exchanger is manufactured in the horizontal orientation. With over 200 channels per heat exchanger, adding supports within the channels would unnecessarily complicate the cleaning process afterward while increasing build time and cost. Consequently, the orientation was changed from horizontal to vertical (*as shown in Figure 12*), which ensures the channels will be manufactured square as required (*cf. Figure 13*).

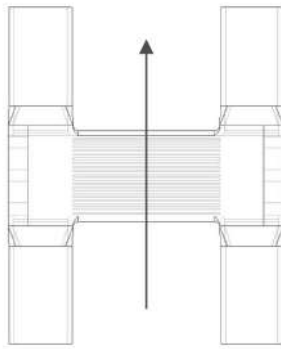


Figure 10: Horizontal Orientation.

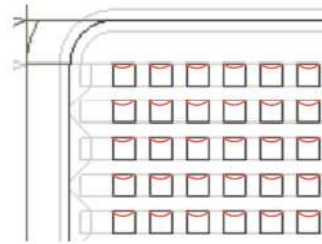


Figure 11: Result of horizontal manufacturing.

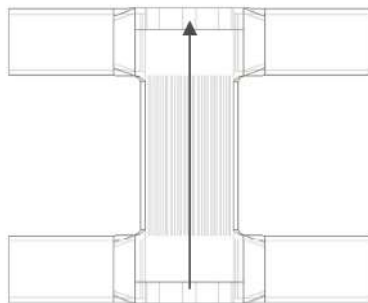


Figure 12: Vertical Orientation.

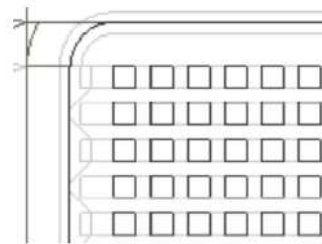


Figure 13: Result of vertical manufacturing.

3.2.2 Holes and circular sections

It is also preferred to manufacture circular sections like pipes, with their axis in a vertical direction. If it is not possible, supports should be added to sections with diameters larger than 6 mm to ensure that the circular profile manufactures correctly. Figure 14 shows a circular section manufactured without a support and it can be seen (shown in red) how the top part of the section will sag during manufacturing, indicated in red. While supports are automatically added by the software used by CRPM, one may incorporate optimized supports within the CAD model to eliminate the need for removal. Software generated supports are removed after manufacturing, thus less supports equals less labour and time. Figure 15 shows how the same profile would be manufactured with an optimized support.

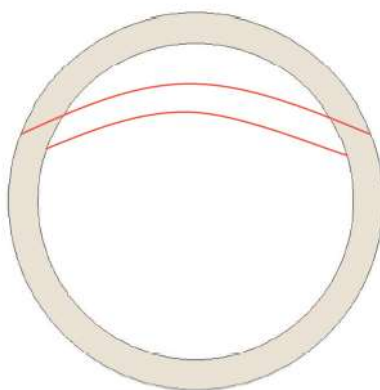


Figure 14: Circular section manufactured in a horizontal position without support.



Figure 15: Circular section manufactured in a horizontal position with support.

Holes within a body orientated in a horizontal direction also requires supports for diameters larger than 8 mm, however the CAD model can be adapted to create holes without supports. This can be achieved by changing the shape from round to a teardrop as shown in Figure 16). During the manufacturing process the teardrop shape acts as a self-supporting structure, allowing the hole to be manufactured without support.

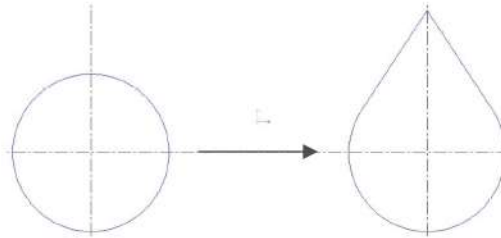


Figure 16: Shape adapted to create hole without supports.

3.2.3 Sections at an angle

Sections such as transitioning sections, where a circular section becomes square, or sections forming an angle with the horizontal should be carefully considered. If not supported, these sections could deform during the AM process. With the heat exchanger design the supports within the transition sections would not be easily accessible for removal. A solution is to keep the angle, created with the horizontal plane, larger than 35° . If the angle to the horizontal plane is less than 35° , without supports, the material could curl up during manufacturing. Figure 17 shows how a section could deform during manufacturing (indicated in red), without support structures added to an angle less than 35° . Figure 18 shows the revised section of the design with an angle greater than 35° [17]. This does however increase the entrance loss experienced within the heat exchanger and should be carefully considered. Thus, supports could not be added to this section due to access restriction.

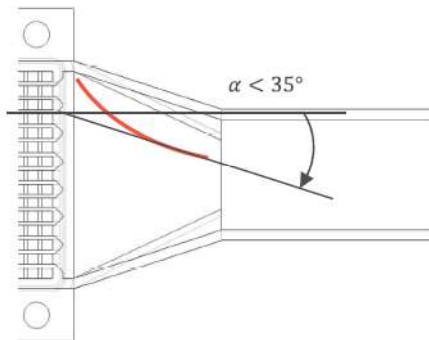


Figure 17: Transitioning section with α smaller than 35° .

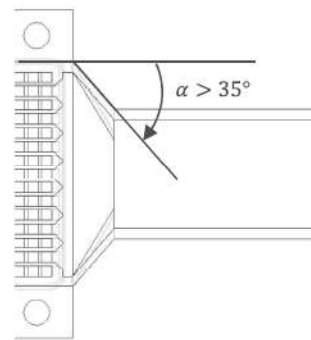


Figure 18: Transitioning section with α larger than 35° .

3.2.4 Threaded holes, drilling and tapping

The ability additive manufacturing possesses to create intricate parts as one is very appealing, although not always preferred. For simplicity it is sometimes necessary to use fasteners to join the manufactured part to existing parts.

The heat exchanger design consists of three parts, two end flanges and a body. These parts will be joined with M4 screws. Threaded holes are thus required within the body of the heat exchanger. The initial idea was to manufacture the heat exchanger body with threaded holes. Process engineers of the CRPM advised that holes smaller than M10 has poor thread definition and should be added after the manufacturing process by drilling and tapping the holes manually.

After a consultation with the CRPM machinist, he indicated that it is possible to drill and tap M4 threaded holes into Ti-6Al-4V, but it is a difficult task. The hardness of the material may cause the taps to break when tapping holes smaller than M6. He suggested manufacturing the holes as clearance holes and using standard nuts and bolts to fix the flanges to the heat exchangers' body.

3.2.5 Cleaning and examination

After manufacturing, the part needs to be cleaned. This entails the removal of excess powder, support structures and rough edges that might have been created during the AM built process. After production, un-sintered powder would be present within the heat exchangers' core. This excess powder can then be removed with compressed air. The difficulty of this process would increase greatly if the core of the heat exchanger is completely enclosed. Therefore, the heat exchangers were designed with open ends which would allow visual inspection of the heat exchanger internal structure. During assembly, these ends are closed with flanges, as mentioned in the previous section and shown in Figure 19). This can also be used for service inspections and cleaning during the life cycle of the heat exchangers.

The supports created during the AM process were removed using conventional methods such as wire cutting, grinding, sawing, and drilling. The position of internal supports should be carefully considered as they need to be accessible for removal after manufacturing. The heat exchanger design has minimal internal supports and is designed to form part of the heat exchanger as they are not accessible for removal.

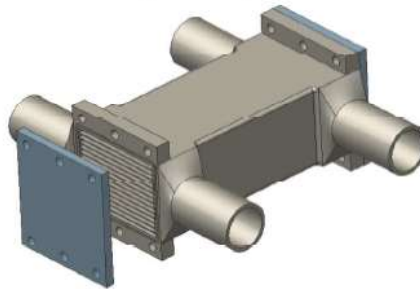


Figure 19: 3-D model with flanges.

The channel walls are extremely thin (0.35 mm) and it would be difficult to visually inspect their integrity throughout the channels. Thus, the heat exchangers will be sent to Stellenbosch University for internal examination of the channels' wall integrity by means of a computerized axial tomography (CAT) scan. Channel wall porosity present within the walls which would cause fluid mixture between the primary and secondary fluid streams, which is highly undesirable.

3.2.6 Minimum thickness and surface roughness

The achievable limitations should be kept in mind when designing any part for additive manufacturing. One of the main limitations is the minimum wall thickness that can be achieved using the DMLS AM process. The minimum wall thickness recommended for Ti-6Al-4V is 0.35 mm, as thinner thicknesses are too fragile according to the CRPM process engineers. Using a minimum thickness for the heat exchanger channel walls reduces the heat transfer resistance which improves the heat transfer.

The surface roughness of additive manufactured parts varies along its axis's and should thus be considered when designing a part. A study done at the CRPM revealed that the top surface, regarding the build direction, of a part is considerably smoother than the side surfaces, with the average surface roughness of $Ra = 5.9 \mu m$ and $Ra = 14.5 \mu m$ respectively [18]. According to Kandlikar SG [19], a pipe with a higher surface roughness yields greater heat transfer. The velocity boundary layer is broken, increasing the heat transfer within the channel [20]. Thus, manufacturing the heat exchangers with the channel walls to have a greater surface roughness would improve the heat transfer. This consideration refers back to the manufacturing orientation.

4. CONCLUSION

Using design considerations provided by CRPM and other sources, heat exchangers will be produced for testing and characterisation. This research will produce design considerations, design lessons and thermo-hydraulic characteristics for additive manufactured compact counter-flow heat exchangers, produced from Ti-6Al-4V, for application in Organic Rankine Cycles. Although these considerations were guided by the heat exchanger design, they are still applicable to other additive manufacturing designs.

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REFERENCES

- [1] Taisce, A. 2015. Energy Matters - How COP21 can shift the energy sector onto a low-carbon path that supports economic growth and energy access, COP 21, 2015 United Nations Climate Change Conference, pp 1-8.
- [2] Engel, D. 2008. Closed-loop, organic Rankine-cycle plants are adding multi megawatts –without additional fuel, Distributed Energy, The Journal for Onsite Power Solutions, pp 1-7.
- [3] Tchanche, B.F.; Lambrinos, G.; Frangoudakis, A.; Papadakis, A. 2011. Low-grade heat conversion into power using organic Rankine cycles - A review of various applications, Renewable and Sustainable Energy Reviews, 15(8), pp 3963-3979.
- [4] Lemmens, S.; Lecompte, S. 2017. Case study of an organic Rankine cycle applied for excess heat recovery: Technical, economic and policy matters, Energy Conversion and Management, 138, pp 670-685.
- [5] Yamada, N.; Tominaga, Y.; Yoshida, T. 2014. Demonstration of 10-Wp micro organic Rankine cycle generator for low-grade heat recovery, Energy, 78(Nov), pp 806-813.
- [6] Frazier, W.E. 2014. Metal additive manufacturing: A review, Journal of Materials Engineering and Performance, 6, pp 1917-1928.
- [7] McMahan, T. 2015. NASA 3D printed fuel rocket pump, Available: <https://www.nasa.gov/centers/marshall/news/news/releases/2015/successful-nasa-rocket-fuel-pump-tests-pave-way-for-3-d-printed-demonstrator-engine.html>.
- [8] Arie, M.A.; Shooshtari, A.H.; Dessiatoun, S.V.; Al-Hajri, E.; Ohadi, M.M. 2015. Numerical modeling and thermal optimization of a single-phase flow manifold-microchannel plate heat exchanger, International Journal of Heat and Mass Transfer, 81, pp 478-489.
- [9] Ashman, S.; Kandlikar, S.G. 2006. A Review of Manufacturing Processes for Microchannel Heat Exchanger Fabrication, ASME 4th international Conference on Nanochannels, Microchannels and Minichannels, 2006, pp 855-860.
- [10] Tsopanos, S.; Sutcliffe C.J.; Owen, I. 2005. The Manufacture of Micro Cross-Flow Heat Exchangers by Selective Laser Melting, Proc. 5th International Conference on Enhanced Compact and Ultra-Compact Heat Exchangers: Science, Engineering and Technology, Sept, pp 410-417.
- [11] Kew, P.A.; Reay, D.A. 2011. Compact/micro-heat exchangers - Their role in heat pumping equipment, Applied Thermal Engineering, 31(5), pp 594-601.
- [12] Eastop, T.D.; McConkey A. 1993. Applied Thermodynamics: For Engineering Technologists, 5th Edition, Pearson.
- [13] Furumoto, T.; Koizumi, A.; Alkahari, M.; Anayama, R.; Hosokawa, A.; Tanaka, R.; Ueda, T. 2015. Permeability and strength of a porous metal structure fabricated by additive manufacturing, Journal of Materials Processing and Technology, 219, pp 10-16.
- [14] Liu, Q.; Wang, Y.; Zheng, H.; Ding, L.; Li, H.; Gong, S. 2016. Microstructure and mechanical properties of LMD-SLM hybrid forming Ti6Al4V alloy, Materials Science and Engineering: A, 660, pp 23-33.
- [15] Holman, J.P. 2010. Heat Transfer 10th Edition, 10th Edition, McGraw-Hill.
- [16] Hodge, B.K.; Taylor, R.P. 1990. Analysis and design of energy systems, 3rd Edition, Pearson.
- [17] Stratasys, Design guidelines: Direct metal laser sintering (DMLS), Available: <https://www.stratasysdirect.com/resources/design-guidelines/direct-metal-laser-sintering>
- [18] Kouprianoff, D.; du Plessis, A.; Yadroitsava, I.; Yadroitsev, I. 2016. Non-destructive testing of the parts manufactured by direct metal laser sintering. SUN Conferences, 17th Annual Conference of the Rapid Product Development Association of South Africa, 17, pp 1-10.
- [19] Kandlikar, S.G. 2001. Effect of channel roughness on heat transfer and fluid flow characteristics at low reynolds numbers in small diameter tubes, Heat Transfer Engineering, 24(3), PP 4-16.
- [20] Everts, M.; Ayres, S.R.; Houwer, F.A.M.; Vanderwagen, C.P. 2014. The influence of surface roughness in the transitional flow regime, IHTC-15, The 15th International Heat Transfer Conference, 15, pp 1-13.

H.2 RAPDASA Paper 2019

S.C. Venter, G.G. Jacobs, DEVELOPMENT AND CHARACTERISATION OF A
Ti6Al4V ADDITIVE MANUFACTURED COMPACT COUNTER-FLOW HEAT
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Micro Organic Rankine Cycles possess the ability to increase the efficiency of various systems by utilising their waste heat. The waste heat is captured by means of a heat exchanger and converted into useful energy to aid the existing system. Traditional manufacturing methods for heat exchangers limit their size or design. Additive manufacturing bridges the gap with its ability to produce intricate designs with great accuracy. The paper discusses the development and characterisation of an additive manufactured compact counter-flow heat exchanger for application in Organic Rankine Cycles.

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1. INTRODUCTION

The high energy demand of the modern world is pushing engineers to improve the efficiency of current energy producing equipment [1]. An increase in efficiency will alleviate the effect of global warming by lessening the production of greenhouse gasses. Various technologies exist to accomplish this and the technology of interest in this study is the Organic Rankine Cycle (ORC). An ORC operates on the same principles as a conventional steam Rankine cycle with the difference being it utilises organic fluids instead of water. These systems are implemented to capture the waste heat from existing systems, in return improving efficiency. The waste heat is captured by means of a heat exchanger (evaporator) and used to produce additional power via a generator [2,3].

Reducing the size of conventional ORCs would allow domestic users to benefit from this technology [4]. However, reducing the size decreases the manufacturability of the various components. Traditional methods used to produce these components are either limiting their size or design [5]. Additive manufacturing (AM) is considered to be a viable method for manufacturing these components [6,7]. The AM process involves material being joined layer upon layer to produce the desired part [8]. This technology opens a world of opportunities to develop intricate parts as one solid body.

One technology that could substantially benefit from AM is micro-channel heat exchangers. Traditional methods for producing microchannel heat exchangers are time-consuming and are limited to basic designs [5]. With the ability to produce walls with thicknesses of less than 200 μm these devices can evolve from its former conventional manufactured designs to increase its effectiveness [9]. Implementing high effectiveness small-channel and micro-channel heat exchangers in ORCs allows a reduction in the size of these cycles to implement on systems producing low-grade waste heat, such as flue gas.

The purpose of this study is to develop, characterise and refine the design and AM methodologies for additively manufactured Ti6Al4V compact counter-flow heat exchangers for application in ORCs. A mathematical model was developed to aid in the preliminary heat exchanger design. The preliminary design was adapted for AM with design considerations provided by the Centre for Rapid Prototyping and Manufacturing (CRPM) at the Central University of Technology (CUT). Validation of the mathematical model against experimental results will be undertaken in future.

This paper discusses the process followed to design compact counter-flow heat exchangers for AM. It includes design lessons learnt for additive manufactured Ti6Al4V heat exchangers. Results from the one-dimensional mathematical model are also discussed and an outline of the planned experimental setup is given.

2. LITERATURE REVIEW

2.1 Heat exchangers

Heat exchangers are devices used to transfer heat between two or more fluids, either to capture or reject excess heat. This study focused on compact counter-flow heat exchangers. The above-mentioned heat exchangers have high effectiveness due to its ability to operate with close approach temperatures [10]. The ratio of the actual heat transferred to the maximum possible heat transfer is known as the effectiveness of a heat exchanger. High effectiveness is particularly beneficial in power-producing systems [11].

A simple heat exchanger design was considered for this study to minimise the effects of uncertainty and simplify the characterisation process. The characterisation of these heat

exchangers would allow a better understanding of the possibilities and limitations of additive manufactured Ti6Al4V compact heat exchangers.

2.2 Production methods

Conventional production methods like X-ray lithography (LIGA), chemical and silicon etching and diamond or wire machining are used to produce contemporary compact heat exchangers. These methods either limit the design complexity or require more than one process to produce a heat exchanger [5,9].

Additive manufacturing, particularly selective laser melting (SLM), is a well-suited technology for the creation of complex parts with good accuracy ($\pm 50 \mu\text{m}$) [12]. Components manufactured from Ti-6Al-4V through SLM also achieve ultimate tensile stresses close to that obtained through conventional processes making SLM a viable method to produce intricate parts such as compact heat exchangers in a variety of metallic materials [13, 14].

2.3 Material selection

Additive manufacturing possesses the ability to produce parts in a variety of materials [8], with the titanium alloy Ti-6Al-4V being the material of interest in this study. Ti-6Al-4V is a well-suited material for producing compact heat exchangers, with its high specific strength the heat exchangers' channel walls can be thinned down significantly to reduce heat transfer resistance. Furthermore, Ti-6Al-4V has exceptional corrosion and good fluid erosion resistance and with its low thermal expansion coefficient, warpage and fatigue effects are minimised during thermal cycling [15].

3. METHODOLOGY AND DESIGN CONSIDERATIONS

Table 1: List of symbols

Symbol	Description
ε	Heat Exchanger Effectiveness
q	Heat transfer rate
$\dot{m}$	Mass flow rate
c	Specific heat capacity
T	Temperature
P	Pressure
G	Mass velocity
ρ	Fluid density
σ	Ratio of free-flow area to frontal area
$K_{c/e}$	Entrance/exit loss coefficient
f_F	Fanning's friction factor
L	Length
D_h	Hydraulic diameter
Subscripts	
i	Inlet
e	Exit
m	Mean

3.1 Theoretical design

For the preliminary design of the heat exchanger, the effectiveness-NTU method was used; the method is generally preferred for analysis and design of heat exchangers [16,17,18].

The effectiveness of a heat exchanger is determined with equation 1:

$$\text{Effectiveness} = \varepsilon = \frac{\text{Actual Heat Transfer}}{\text{Maximum Heat Transfer}} \quad 1$$

The actual heat transfer is calculated by either determining the energy lost from the hot fluid or the energy gained by the cold fluid, equation 2. Maximum heat transfer may be calculated with the maximum temperature difference within the heat exchanger using equation 3:

$$q = \dot{m}_h c_h (T_{hi} - T_{ho}) = \dot{m}_c c_c (T_{co} - T_{ci}) \quad 2$$

$$q_{\max} = (\dot{m}c)_{\min} (T_{hi} - T_{ci}) \quad 3$$

The core pressure drop across the heat exchanger is calculated with equation 4; which incorporates entrance-, core friction-, flow acceleration-, and exit- losses:

$$\Delta p = \frac{G^2}{2\rho_i} \left[\left(1 - \sigma_i^2 + K_e \right) + f_F \frac{4L}{D_h} \left(\frac{\rho_i}{\rho_m} \right) + 2 \left(\frac{\rho_i}{\rho_e} - 1 \right) - \left(1 - \sigma_e^2 - K_e \left(\frac{\rho_i}{\rho_e} \right) \right) \right] \quad 4$$

A one-dimensional mathematical model was created implementing the above-mentioned equations in Engineering Equation Solver (EES). With the mathematical model, the theoretical performance can be predicted for a specific heat exchanger design. For this study three compact counter-flow heat exchangers were produced, each with different channels sizes namely 0.5, 1 and 2mm, transferring 3kW heat theoretically, when water is used as the fluid being heated. Initial testing and characterisation will be done with water, due to its ease of use. Later testing and characterisation will be conducted with an organic fluid.

3.2 Computer-aided design models

The results from EES gave a good indication of how many channels and what length of channels are required to attain a theoretical 3kW of heat transfer for each of the selected channel sizes. The inlet conditions for all the heat exchangers remained the same with the only difference being the channel's size, number of channels and the length. The design results are listed below.

3.2.1 Design 1

Heat exchanger width	72	mm
Heat exchanger height	27	mm
Channel length	30	mm
Channel size	0.5x0.5	mm
Wall thickness	0.35	mm
Number of channels per row	20	
Number of rows	11	
Estimated heat transfer with steam and water	3012	W

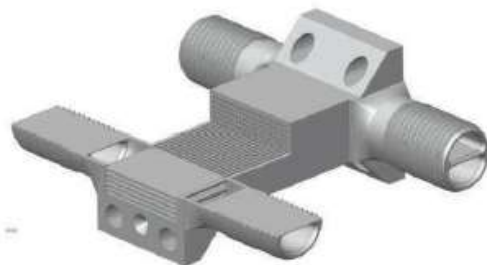


Figure 1: Design 1: 3-D Clipped Section

3.2.2 Design 2

Heat exchanger width	80	mm
Heat exchanger height	38	mm
Channel Length	40	mm
Channel size	1x1	mm
Wall thickness	0.35	mm
Number of channels per row	16	
Number of rows	12	
Estimated heat transfer with steam and water	3071	W

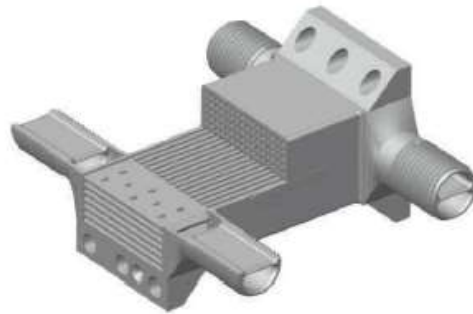


Figure 2: Design 2: 3-D Clipped Section

3.2.3 Design 3

Heat exchanger width	100	mm
Heat exchanger height	109	mm
Channel Length	50	mm
Channel size	2x2	mm
Wall thickness	0.35	mm
Number of channels per row	26	
Number of rows	26	
Estimated heat transfer with steam and water	3008	W



Figure 3: Design 3: 3-D Clipped Section

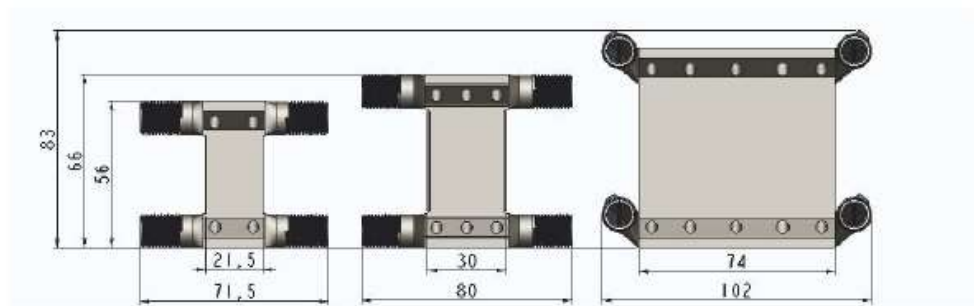


Figure 4: Size comparison of the three heat exchanger designs, front view

As the channel size is increased the total heat transfer diminish, requiring more and longer channels to achieve the 3kW of heat transfer (cf. Figure 4).

3.3 Design Considerations

Numerous design and manufacturing considerations informed the eventual design and configuration of the heat exchangers discussed in this paper. The design considerations resulted from literature and consultation with expert process engineers at CRPM. Optimising designs for any AM technology reduces time and/or cost by eliminating the need for unnecessary supports and cutting down on post-processing. A detailed article discussing the design considerations was written by Venter, Jacobs and Du Preez [19]. A List of design improvements suggested by the CRPM engineers are listed below:

- **Manufacturing orientation:**
Selecting the correct manufacturing orientation can eliminate the need for unnecessary supports.
- **Holes and circular sections:**
Holes and circular sections are preferably manufactured with its axis in the vertical direction, otherwise supports or scaffolding are required.
- **Sections at an angle:**
Any sections that form an included angle with the horizontal of less than 35° should be supported.
- **Threaded holes, drilling and tapping:**
Carefully consider manufacturing parts with threads smaller than M10 as the thread definition is poor. Tapping or drilling holes smaller than M6 is not advised as the hardness of Ti6Al4V poses a problem.
- **Cleaning and examination:**
The design must facilitate post-processing such as removal of supports and removal of excess powder.
- **Minimum thickness and surface roughness:**
Achievable limitations should be considered for the designs, keeping in mind the minimum wall thickness and surface roughness of 'as manufactured' parts. The minimum wall thickness for SLM of Ti-6Al-4V is in the order of 0.35 mm.

3.3.1 *Unsuccessful prints*

Although every precaution was taken to ensure that the heat exchangers will be manufactured correctly, there are always unforeseen difficulties with new technology. The first two manufacturing attempts failed. Further investigation revealed build-up of material on the top layer of the part causing the recoating arm, of the SLM machine, to collide with the part and stopping the printing process. Build-up problems were experienced during the first two manufacturing attempts.

Design 1

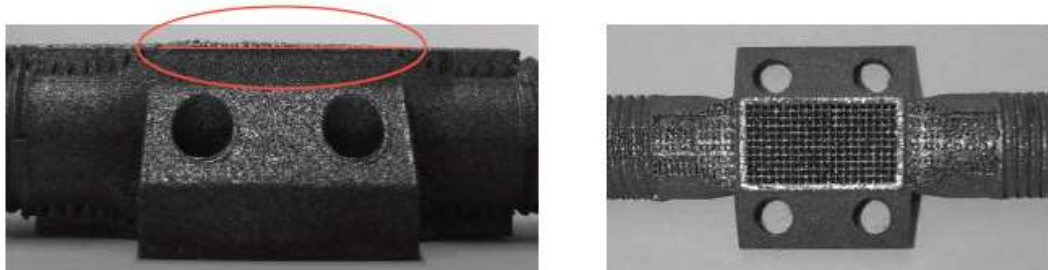


Figure 5: Design 1, 1st build attempt, side and top view

Design 2 -First attempt

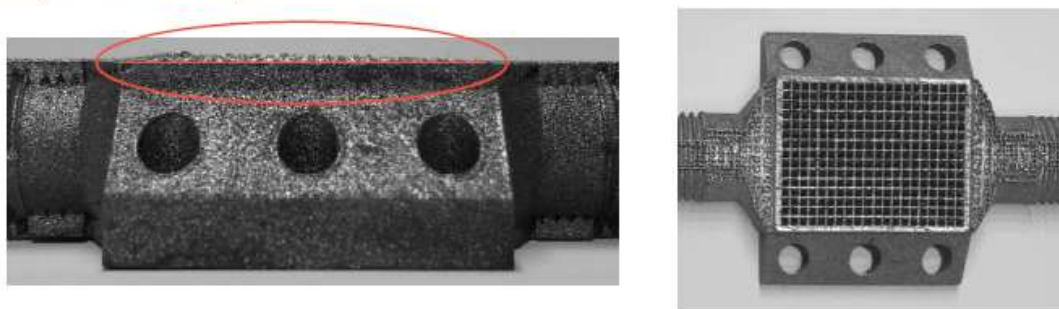


Figure 6: Design 2, 1st build attempt, side and top view

Design 2 - Second attempt

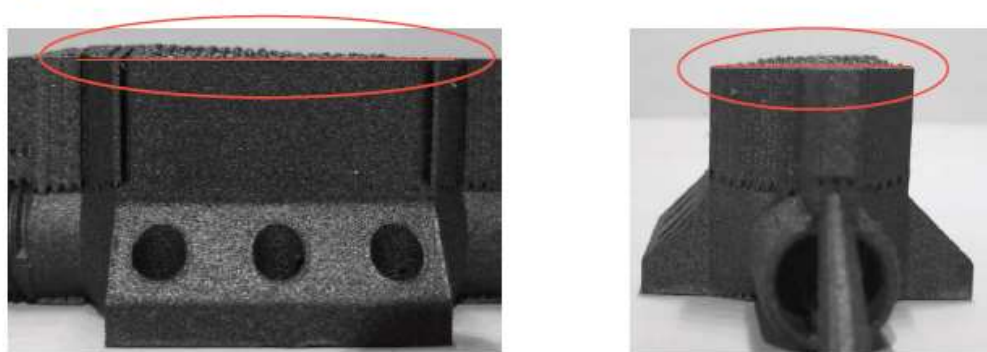


Figure 7: Design 2, 2nd build attempt, side and side 2 view

Design 3

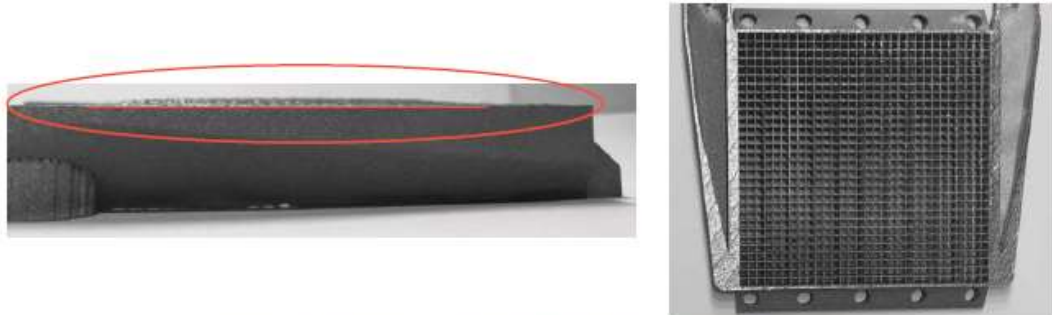


Figure 8: Design 3, 1st build attempt, side and top view

Inspecting the unsuccessful prints shows that material builds-up on one side during the manufacturing process. Looking at figure 5, it can be seen that there is approximately 0.3mm of material build-up on the left side. Figure 6, the first manufacturing attempt for design 2, shows material build-up of approximately 0.7mm on the left. Figure 7, the second manufacturing attempt of design 2, shows material build-up of approximately 1.3mm. The first manufacturing attempt of design 3 (cf. figure 8) shows a material build-up of 1mm approximately.

Although the printing of design 1 completed during the second manufacturing attempt, the channel walls were found to be porous. It was sent to the centre for analytics facilities (CAF) at Stellenbosch for an internal examination by means of a computerised tomography (CT) scan.

Designs 1 and 2 were considered for a second manufacturing attempt. The angle at which the recoating arm recoats the parts was changed. This enabled design 1 to print entirely with minimal material build-up, design 2 however failed once again (cf. Figure 12 & 13) building only 10mm higher before failing

3.3.2 Micro-CT scans

Three models were scanned, two of design 1 and one of design 2. With the scans the internals of the heat exchangers can be inspected. Discussed in this section are areas identified with faults within the heat exchangers.

3.3.2.1 Scan 1

The general build direction is indicated with yellow arrows and the coater direction with blue arrows in figures 9 to 17.

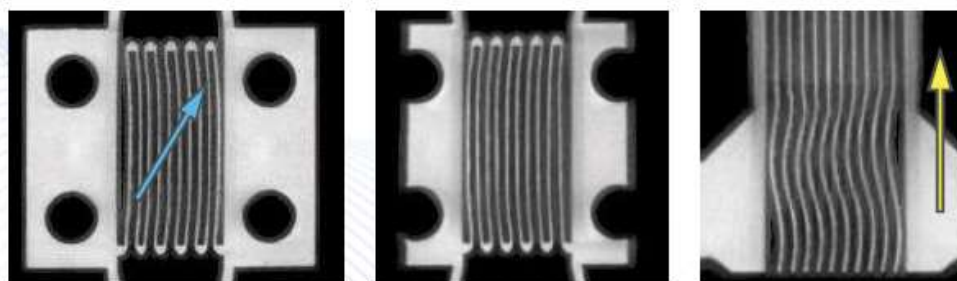


Figure 9: Scan 1, bottom manifold. Top and side view.



Figure 10: Scan 1, top manifold. Top and side view.

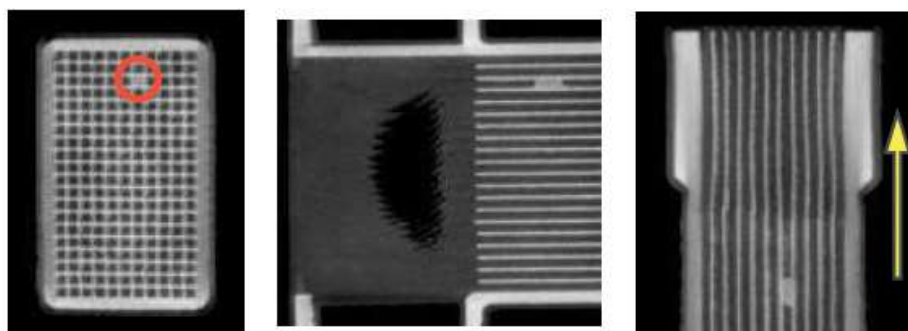


Figure 11: Scan 1, blocked channel. Top, front and side view.

3.3.2.2 Scan 2

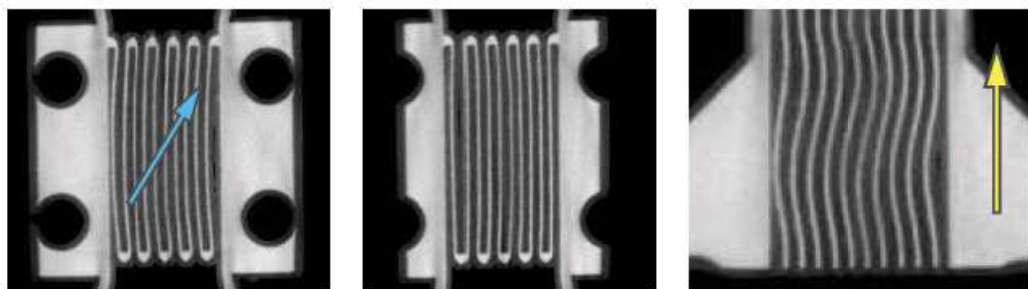


Figure 12: Scan 2, bottom manifold. Top and side view.

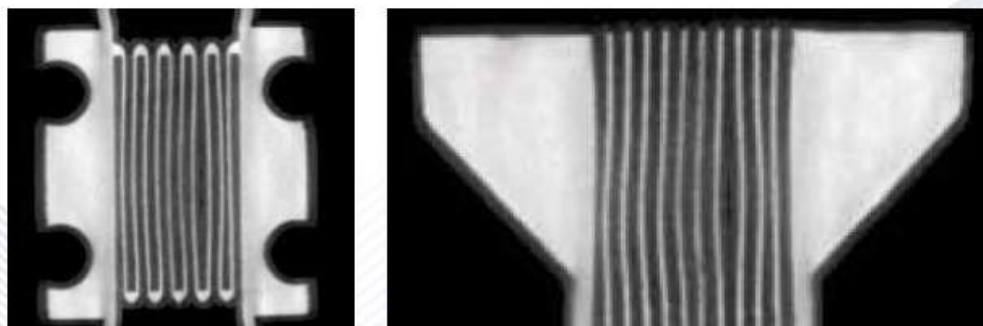


Figure 13: Scan 2, top manifold. Top and side view.

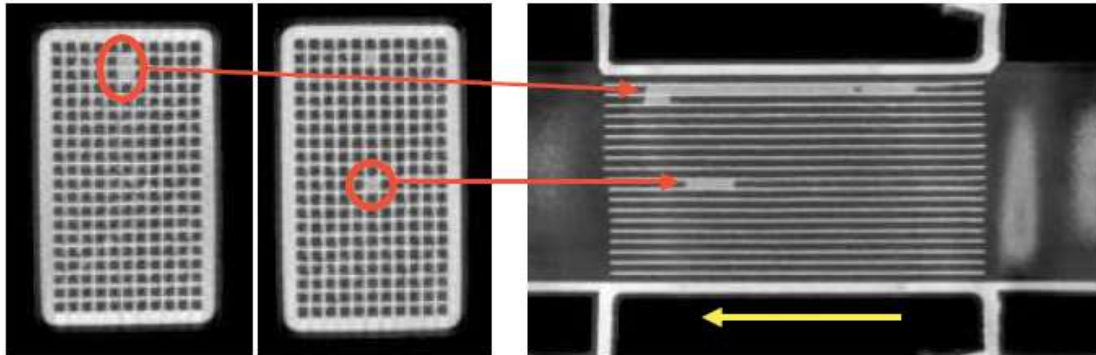


Figure 14: Scan 2, blocked channels. Top and front view.

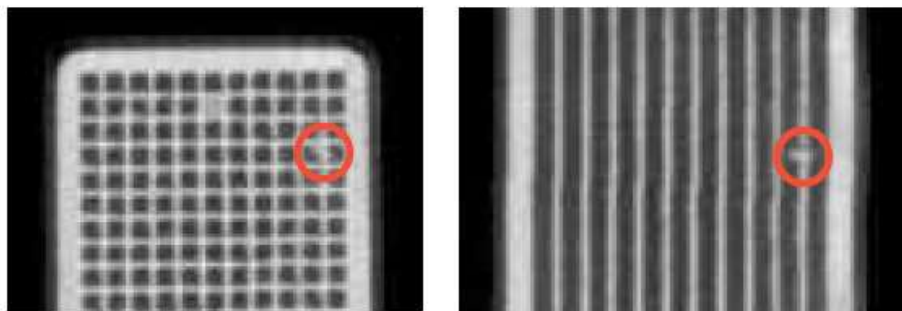


Figure 15: Scan 2, Abnormality. Top and side view.

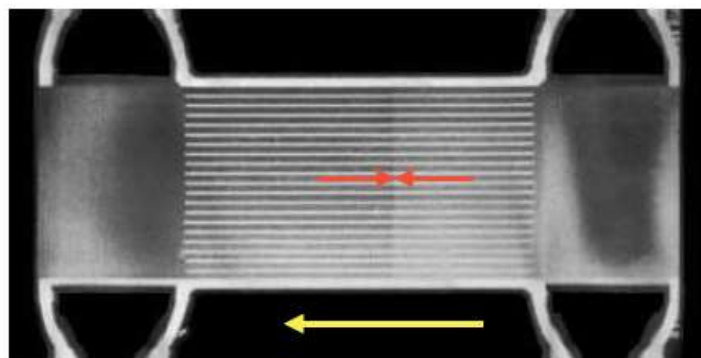


Figure 16: Scan 2, manufacturing line (indicated with red arrows). Front view.

3.3.2.3 Scan 3

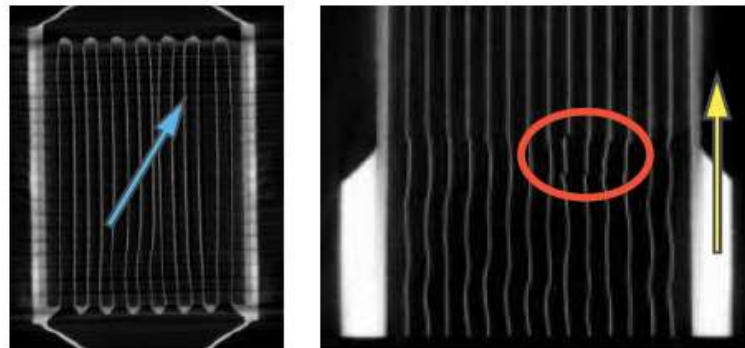


Figure 17: Scan 3, bottom manifold. Top and side view.

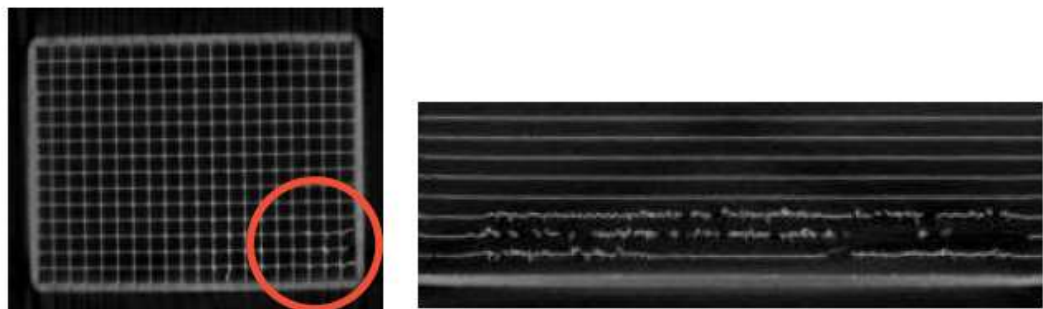


Figure 18: Scan 3, broken channels. Top and front view.

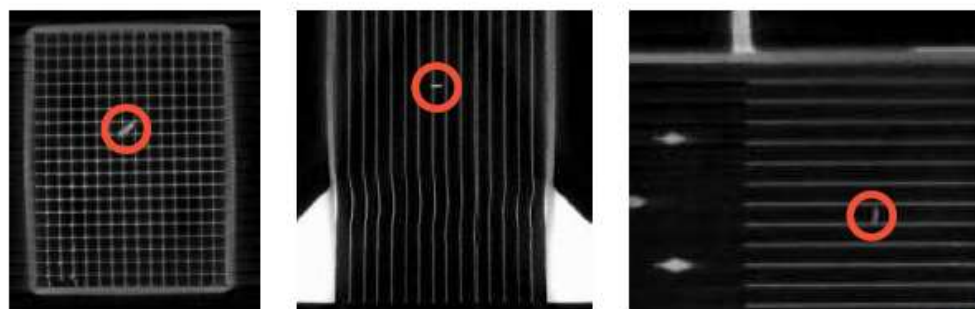


Figure 19: Scan 3, abnormality 1. Top, side and front view.

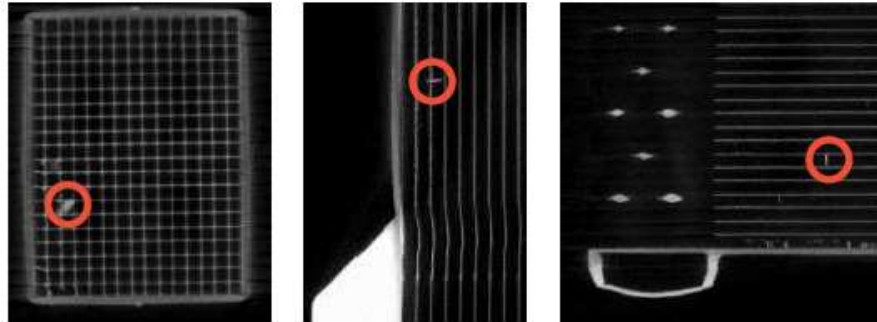


Figure 20: Scan 3, abnormality 2. Top, side and front view.

Scan 1 shows minor imperfections throughout the heat exchanger. Manifold deformation can be seen in figures 9 and 10. High stress within the manifold walls is likely the cause of this deformation. Heat build-up during manufacturing can cause the thin walls to buckle without support, thus the bottom of the heat exchanger is more deformed than the top. Figure 11 depicts a blocked channel within the heat exchanger, likely to be powder stuck within the channel.

Scan 2 shows the same minor imperfections within the heat exchanger as scan 1. Also present are two other more serious imperfections (*cf.* Figures 15 and 16). Figure 15 shows an abnormality within the heat exchanger that passes through the channel wall. This is likely some material build up (splatter) during the manufacturing process. Scan 1 and 2 was performed on the heat exchangers from the second manufacturing attempt. Figure 16 shows a faint line throughout the heat exchanger where the manufacturing process stopped and continued.

Design 2 was manufactured on the 3rd attempt and was also sent for CAT scans. Figures 17-20 shows the results for scan 3. Figure 17 shows the same manifold deformation as in the previous scans despite that design 2 has manifold channel supports. It is however worse in this case as the stress caused the channels to break. Broken channel walls can be seen in figure 18, the thicker outer wall causes a higher heat build-up inducing more stress within the channels attaching to it. Two abnormalities were found within the heat exchanger that passes through walls, likely causing fluid mixture within the heat exchanger (*cf.* Figures 19 and 20).

4. SUMMARY & CONCLUSION

There are various obstacles identified that prevented a successful heat exchanger build. However, to ensure a successful build in the future, an achievable minimum wall thickness should be selected to prevent fluid mixture and ensure pressure integrity. Additionally, the channel wall breakage and warping visible in the scans was due to a combination of the thin channel walls, heat build-up and stresses during the manufacturing process. Furthermore, channel blockage occurred only on design 1 which is a result of the small channel size. Further investigation is necessary to determine if channel blockage is due to powder stuck within the channel or if the channel was blocked during the melting process.

Moreover, it was evident that the placement of supports had a paramount effect on channel warping and breakage. Careful consideration should be given to ensure practicality of supports, minimising the obstruction of fluid flow and the increase of pressure drops. Consequently, optimum placement of supports will alleviate the effects of channel warping

and breakage. In conclusion, optimal parameters should still be investigated to alleviate the above-mentioned obstacles identified when manufacturing micro-channel heat exchangers for application in ORC's.

Further investigation is required to resolve the material build-up during the manufacturing process. Changing the angle at which the recoating arm passes had a positive effect during the manufacturing process. The second design manufactured completely by increasing the angle. The porosity and broken walls found within the heat exchangers could possibly be eliminated by increasing the channel wall thickness. Another solution would be to alter the laser scan parameters to obtain a better build quality and eliminate wall porosity within the heat exchangers. Further research should be done in this regards.

Additional future research should be done to optimise the heat exchanger channels, looking for the best size, shape and length to achieve the highest possible heat transfer with minimum pressure drop.

The characterisation of these heat exchangers forms the basis for further investigation in AM heat exchanger design for application in micro ORC's at CUT.

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6. REFERENCES

- [1] Taisce, A. 2015. Energy Matters - How COP21 can shift the energy sector onto a low-carbon path that supports economic growth and energy access, COP 21, 2015 United Nations Climate Change Conference, pp 1-8.
- [2] Engel, D. 2008. Closed-loop, organic Rankine-cycle plants are adding multi megawatts –without additional fuel, Distributed Energy, The Journal for Onsite Power Solutions, pp 1-7.
- [3] Tchanche, B.F.; Lambrinos, G.; Frangoudakis, A.; Papadakis, A. 2011. Low-grade heat conversion into power using organic Rankine cycles - A review of various applications, Renewable and Sustainable Energy Reviews, 15(8), pp 3963-3979.
- [4] Yamada, N.; Tominaga, Y.; Yoshida, T. 2014. Demonstration of 10-Wp micro organic Rankine cycle generator for low-grade heat recovery, Energy, 78(Nov), pp 806-813.
- [5] Ashman, S.; Kandlikar, S.G. 2006. A Review of Manufacturing Processes for Microchannel Heat Exchanger Fabrication, ASME 4th international Conference on Nanochannels, Microchannels and Minichannels, 2006, pp 855-860.
- [6] McMahan, T. 2015. NASA 3D printed fuel rocket pump, Available: <https://www.nasa.gov/centers/marshall/news/news/releases/2015/successful-nasa-rocket-fuel-pump-tests-pave-way-for-3-d-printed-demonstrator-engine.html>.
- [7] Arie, M.A.; Shooshtari, A.H.; Dessiatoun, S.V.; Al-Hajri, E.; Ohadi, M.M. 2015. Numerical modelling and thermal optimization of a single-phase flow manifold-microchannel plate heat exchanger, International Journal of Heat and Mass Transfer, 81, pp 478-489.

- [8] Frazier, W.E. 2014. Metal additive manufacturing: A review, *Journal of Materials Engineering and Performance*, 6, pp 1917-1928.
- [9] Tsopanos, S.; Sutcliffe C.J.; Owen, I. 2005. The Manufacture of Micro Cross-Flow Heat Exchangers by Selective Laser Melting, *Proc. 5th International Conference on Enhanced Compact and Ultra-Compact Heat Exchangers: Science, Engineering and Technology*, Sept, pp 410-417.
- [10] Kew, P.A.; Reay, D.A. 2011. Compact/micro-heat exchangers - Their role in heat pumping equipment, *Applied Thermal Engineering*, 31(5), pp 594-601.
- [11] Eastop, T.D.; McConkey A. 1993. *Applied Thermodynamics: For Engineering Technologists*, 5th Edition, Pearson.
- [12] Furumoto, T.; Koizumi, A.; Alkahari, M.; Anayama, R.; Hosokawa, A.; Tanaka, R.; Ueda, T. 2015. Permeability and strength of a porous metal structure fabricated by additive manufacturing, *Journal of Materials Processing and Technology*, 219, pp 10-16.
- [13] Mertens, A.; Reginster, S.; Paydas, H.; Contrepolis, Q.; Dormal, T.; Lemaire, O.; Lecomte-Beckers, J. 2014. Mechanical properties of alloy Ti-6Al-4V and of stainless steel 316L processed by Selective Laser Melting: Influence of out-of-equilibrium microstructures, *Powder Metallurgy*, 57(3), pp 184-189.
- [14] Veiga, C.; Davim, J.P.; Loureiro, A.J.R. 2012. Properties and applications of titanium alloys: a brief review, *Rev. Adv. Mater. Sci.*, 32, pp 133-148.
- [15] Liu, Q.; Wang, Y.; Zheng, H.; Ding, L.; Li, H.; Gong, S. 2016. Microstructure and mechanical properties of LMD-SLM hybrid forming Ti6Al4V alloy, *Materials Science and Engineering: A*, 660, pp 23-33.
- [16] Holman, J.P. 2010. *Heat Transfer 10th Edition*, 10th Edition, McGraw-Hill.
- [17] Hodge, B.K.; Taylor, R.P. 1990. *Analysis and design of energy systems*, 3rd Edition, Pearson.
- [18] Muzychka, Y. 1998. "Chapter 5 Heat Exchangers." p. 71-90.
- [19] Venter, S.C.; Jacobs, G.G.; du Preez, J. 2018. "Design considerations for developing an additive manufactured compact counter-flow heat exchanger for application in Organic Rankine Cycles", in 19th Intern. Conf. on Rapid Product Dev. Assoc. of South Africa (RAPDASA), Protea Parktonian South Africa, Nov. 2018, p. 195-200.

