

CRANFIELD UNIVERSITY

Florian Willi Jacob

Performance and Preliminary Design of Combined Cycle Aero
Engines utilising Supercritical Carbon Dioxide

School of Aerospace, Transport and Manufacturing
Thermal Power

PhD Thesis
Academic Year: 2015 - 2018

Supervisor: Dr. V. Sethi
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ABSTRACT

Over past decades, the basic thermodynamic concept of the modern turbofan engine has remained largely untouched and its fuel consumption has been reduced through incremental design improvements. That evolutionary approach will show diminishing returns in the short- to medium-term and is expected to have exhausted its improvement potential by 2050. In the face of the predicted exponential growth in air traffic and growing concerns over aviation's contribution to global warming, the development of more radical concepts is imperative. This PhD research provides an original contribution to knowledge through the investigation of a combined cycle turbofan using supercritical CO₂ as the working fluid in the bottoming cycle (CCTF2050). An additional combined cycle engine is investigated for the first time, which also features inter-turbine reheat between the two stages of the high-pressure turbine (CCTF2050_ITR). Both concepts are subsequently compared to a projected year 2050 reference engine (TF2050), representing the limit of the evolutionary engine development approach.

Supercritical CO₂ shows strong real gas behaviour near its critical point, which makes many experience-based design approximation techniques for ideal-gas components infeasible. Several detailed component models were created specifically for this investigation to adequately determine the performance of the combined cycle engines, size them and assess their potential to reduce mission fuel burn. Specific fluid property models were created for CO₂, alongside scalable S-CO₂ turbomachinery maps and multiple heat exchanger preliminary design models, which allow for detailed assessments like the optimum external and internal tube geometry of the heat exchanger tubes. Many of these models are not only applicable to aero engines, but could also be extended to the design of S-CO₂ power cycles for marine or small-scale off-shore applications. Nevertheless, aero engines face some of the most stringent constraints when it comes to fuel consumption, size and weight. These constraints are addressed by integrating the bottoming cycle and the turbofan directly, and creating additional models to specifically capture the component inter-dependencies in an aero engine. For example, significant changes to the engine installation and their effects on fuel burn are addressed by incorporating diffuser models for the heat exchangers and geometry dependent loss models for the bypass flow that does not pass through the pre-cooler.

Overall, the performance of the combined cycle aero engine is dominated by the compromise between the pre-cooler size and the choice of S-CO₂ compressor inlet temperature and pressure. Together they dictate the S-CO₂ mass flow and hence the

power output of the bottoming cycle. The pre-cooler also directly affects the engine's specific thrust and propulsive efficiency by changing the outlet temperature and pressure of part of the bypass flow, and it can potentially lead to significant pressure losses in the bypass flow not passing through it by constricting its flow area. Finally, the pre-cooler accounts for about 40% of the bottoming cycle's weight.

Extensive parametric studies were conducted to establish the possible fuel burn benefit the combined cycles might offer over the TF2050. The modelling approach also enables the quantification of the sensitivities the combined cycles exhibit to manufacturing limitations or technology assumptions. The CCTF2050 promises fuel burn savings up to 3.2 % relative to the TF2050, though with an approximate uncertainty of +0.3 to -4 %. The CCTF2050_ITR may only achieve a 3.3 % reduction in fuel burn, despite a demonstrated synergy between ITR and the bottoming cycle. Unfortunately, the higher number of assumptions has increased the approximate range of uncertainty to +0.5 to - 4.8 %. Both combined cycles would reduce the fuel burn by about 47.6 % relative to an assumed year 2000 state of the art, if the original predictions are met. These improvements are equal to or lower than the benefits claimed for some competing and less complex engine concepts like intercooled-recuperated aero engines for example. However, the uncertainties around the respective claims of those concepts have not been made available publically. Nevertheless, this might disqualify the combined cycle concept for aero application bar further substantial technological advances, such as new extremely compact yet light heat exchanger designs. However, such advances are not inconceivable given that the engines are considered for entry into service in 2050.

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Abruptly, but certainly not unceremoniously comes the end of this journey. For at least three years supercritical carbon dioxide has been a focal point of my daily routine and it seems strange that this will change with the submission of this thesis. First though, I need to thank everyone who helped me in any form along the way.

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“The larger the island of knowledge, the longer the shoreline of wonder”

I stumbled upon Ralph W Sockman's quote above a while ago and could not agree more, especially when it comes to this work. It is just a little unfortunate that this particular part of the island appears to be underwater.

This work contributes to the research efforts in the “Ultra Low emission Technology Innovations for Mid-century Aircraft Turbine Engines” project and has received funding from the European Union's Horizon 2020 research and innovation programme under Grant Agreement No. 633436.

Dedicated to the love of my life,

Lisa

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LIST OF ABBREVIATIONS

ABC	Air Bottoming Cycle
ACARE	Advisory Council for Aviation Research and Innovation in Europe
BC	Bottoming Cycle
BPR	Bypass Ratio
CAD	Computer Aided Design
CCGT	Combined Cycle Gas Turbine
CCTF	Combined Cycle Turbofan
CET	Compressor Entry Temperature
CFD	Computational Fluid Dynamics
EC	European Commission
ECS	Environmental Control System
EIS	Entry Into Service
EOR	End-of-Runway
EQSS	Equivalent Single Stage
FEM	Finite Element Model(ling)
FPR	Fan Pressure Ratio
GUI	Graphical User Interface
HPC	High Pressure Compressor
HPT	High Pressure Turbine
HTC	Heat Transfer Coefficient
HX	Heat Exchanger
H/T	Hub-to-Tip Ratio
IATA	International Air Transport Association
IGT	Industrial Gas Turbine
IPC	Intermediate Pressure Compressor
ITR	Inter-turbine Reheat
LHV	Lower Heating Value
LPT	Low Pressure Turbine
LRTF2000	Baseline year 2000 Turbofan
MC	Mid-Cruise
MHX	Main Heat Exchanger
NTU	Number of Heat Transfer Units
OEM	Original Equipment Manufacturer
OPR	Overall Pressure Ratio

PCHE	Printed Circuit Heat Exchanger
PR	Pressure Ratio
PREOS	Peng-Robinson Equation of State
S-CO ₂	Supercritical CO ₂
SFC	Specific Fuel Consumption
sF _N	Specific Net Thrust
sPwr	Specific Power
SRIA	Strategic Research and Innovation Agenda
SWEOS	Span & Wagner Equation of State
T/O	Take-Off
TET	Turbine Entry Temperature
TF2050	Reference year 2050 Turbofan
TOC	Top Of Climb
ULTIMATE	Ultra Low emission Technology Innovations for Mid-century Aircraft Turbine Engines
UP	Useful Power
USSPP	Ultra Supercritical Steam Power Plant

NOMENCLATURE

ALT	Altitude	[m]
A	Area	[m ²]
AR	Aspect Ratio	[-]
A_{ff}	Free flow area	[m ²]
A_{fr}	Frontal area	[m ²]
a	Major tube axis (ellipse)	[m]
b	Minor tube axis (ellipse)	[m]
c	Velocity	[m/s]
C	Capacity Rate Ratio	[-]
c_p	Specific heat capacity	[J/(kg*K)]
d_x	Diameter at station x	[m]
d_h	Hydraulic diameter	[m]

f_c, f_d	Friction factors	[-]
F_N	Net Thrust	[kN]
h	Specific Enthalpy	[kJ/kg]
ht	Heat Transfer Coefficient	[W/(K*m ²)]
H	Height of heat exchanger module / Annulus	[m]
j	Colburn factor	[-]
K	Loss factor	[-]
k	Conductivity	[W/(m*K)]
L	Length	[m]
LHV	Lower Heating Value	[MJ/kg]
M	Weight	[kg]
M	Mach number	[-]
m	Circumferential Tube Spacing	[-]
N_{tu}	Number of Transfer Units	[-]
N_c	Number of channels	[-]
N_{col}	Number of Columns	[-]
N_{pass}	Number of passes of heat exchanger	[-]
N_{stacks}	Number of Stacks	[-]
N_{tubes}	Number of tubes	
Nu	Nusselt Number	[-]
n	Axial Tube Spacing	[-]
P	Perimeter	[m]
p	Pressure	[Pa]
Pr	Prandtl Number	[-]
$\dot{Q}$	Heat flow	[(J*kg)/s]
R	Gas constant	[J/(kg*K)]
$R_{0,2}$	Yield Strength	[MPa]
Re	Reynolds Number	[-]
s	Entropy	[J/K]
sp	Spacing between heat exchanger tubes	[-]

T	Temperature	[K]
$tube_{th}$	Tube Wall thickness	[m]
U	Overall heat transfer coefficient	[-]
UP	Useful Power	[kW]
V	Volume	[m ³]
ν	Specific volume	[m ³ /kg]
W	Mass flow	[kg/s]
$Width$	Width of heat exchanger module	[m]
x_{inv}	Degree of involute tube shape at inlet	[-]
α	Angle of involute relative to annulus	[rad]
γ	Heat capacity ratio	[-]
ε	Temperature effectiveness	[-]
η_i	Isentropic efficiency	[-]
η_p	Polytropic efficiency	[-]
η_{prop}	Propulsive efficiency	[-]
η_{over}	Overall efficiency	[-]
η_{th}	Thermal efficiency	[-]
λ	Pressure Loss factor	[-]
μ	Dynamic Viscosity	[Pa*s]
ξ_{abs}, ξ_{rel}	Surface roughness (relative / absolute)	[m] / [-]
π	Pi	[-]
ρ	Density	[kg/m ³]
σ_{HX}	Ratio of wetted surface area to flow area	[-]
σ	Van-Mises Stress	[MPa]
φ	Entropy function	[J/(kg*K)]

SUBSCRIPTS

<i>ave</i>	average
<i>BP</i>	Bypass
<i>c</i>	Cold side
<i>ce</i>	Cold end
<i>cmp</i>	Compressor
<i>CO</i>	Core
<i>diff</i>	Diffuser
<i>i</i>	Inner annulus
<i>inv</i>	Involute
<i>e, LPT</i>	Exit of Low Pressure Turbine
<i>h</i>	Hot side
<i>he</i>	Hot end
<i>m</i>	mean
<i>mat</i>	Material
<i>o</i>	Outer annulus
<i>p</i>	Pass (of a heat exchanger)
<i>oBP</i>	Outer Bypass
<i>s</i>	Static (fluid properties)
<i>SC</i>	Sudden contraction
<i>SE</i>	Sudden expansion
<i>sol</i>	Solid Area
<i>t</i>	Total
<i>trb</i>	Turbine
<i>x</i>	Index that can represent hot or cold flow at the same time

The nomenclature and abbreviations are explained again upon first use and where otherwise appropriate.

1 Introduction

Faced with a highly competitive marketplace, continually increasing passenger traffic [1], [2] and ever-more demanding environmental constraints, aircraft and engine design and technology will need to make even greater strides toward higher efficiency in the coming decades. In its “Strategic Research and Innovation Agenda 2050” (SRIA 2050 or Flightpath 2050, see section 1.1.1) [3], the “Advisory Council for Aviation Research and innovation in Europe” (ACARE) has set very ambitious goals for aircraft for entry into service (EIS) in 2050.

In pursuit of those, this thesis presents two novel engine architectures and assesses the feasibility of the concepts by the year 2050. These engines would be applied to future large twin-aisle civil aircraft on inter-continental missions. The first concept is an ultra-high bypass ratio geared turbofan engine with a supercritical carbon dioxide (S-CO₂) bottoming cycle (CCTF2050). The second engine (CCTF2050_ITR) also includes an inter-turbine combustor between the two stages of the high pressure turbine of the main engine to further increase the core specific power and the thermal efficiency of the bottoming cycle. The bottoming cycles of the CCTF2050 and the CCTF2050_ITR deliver additional shaft power to the low pressure shaft of the turbofans and are therefore able to increase the thermal efficiency beyond the limits of a projected year 2050 turbofan engine. The engines are investigated as direct competitors to the intercooled recuperated aero engine, as they also draw on the core exhaust gases as a heat source. The ability of the concepts to lower the mission fuel burn relative to a year 2050 reference engine depends in large parts on the strong real gas behaviour of the S-CO₂, the bottoming cycle’s turbomachinery and especially the heat exchangers involved. This work therefore presents a comprehensive and integrated performance and preliminary design study of the combined cycle engines and their components.

The primary purpose of this chapter is to briefly outline the motivation behind this investigation and highlight the overall objectives in a broader context. The need for such novel engine concepts is thereby explained as well. The main objectives of this thesis are to:

- Create models that allow for the integrated preliminary design and performance evaluation of S-CO₂ components, particularly the heat exchangers

- Apply these detailed component models to highlight the need for such an integrated design and performance estimation approach when dealing with such a complex engine
- Compare the fuel burn reduction potential of the CCTF2050 and the CCTF2050_ITR against each other and what is assumed to be the state of the art in 2050
- Approximate the uncertainty of the obtained performance and design data

Ultimately, the engine's overall feasibility and its ability to meet the Flightpath 2050 goals is assessed and further developments are suggested.

The present investigation of combined cycle aero engines is part of Cranfield Universities' research efforts in the "Ultra Low Emission Technology Innovations for Mid-century Aircraft Turbine Engines" project (ULTIMATE) [4], a collaborative effort of ten leading industrial and research institutions across Europe [5]. ULTIMATE is part of the European Commission's Horizon 2020 programme and is focussed on the development of novel core technology concepts for ultra-efficient engines to enter into service by 2050 and meet the Flightpath goals. Other engine concepts investigated in ULTIMATE include, but are not limited to, intercooled recuperation, pressure rise combustion and advanced propulsor designs, covering the all of the "Big Three" loss sources (see section 1.1.2). ULTIMATE is furthermore concerned with the identification of synergies between the investigated concepts that might be explored in the future.

1.1 Motivation

Air traffic has more than doubled worldwide over the past two decades and is forecast to continue to grow at an annual rate of about 5 % until at least 2030 by the two main civil aircraft manufacturers [1], [2]. That growth has proven to be very resilient to any form of upheaval overall, even if extreme events like the financial crisis of 2009 may lead to short-term deviations. Unfortunately increasing air traffic means increasing fuel consumption and greenhouse gas emissions. While civil air traffic is estimated to account for only about 2 % of man-made CO₂ emissions in 2017 by the International Air Transport Association (IATA) [6], that share can increase rapidly in the face of such strong growth rates. Hence, as environmental awareness and political pressure increase, so does the effort associated with addressing CO₂, NO_x and noise emissions by all parties involved. Reducing the emission footprint of next generation aircraft has therefore joined the

already long list of development challenges such as low mission fuel burn and easy and affordable maintenance.

1.1.1 Flightpath 2050

The “Advisory Council for Aviation Research and innovation in Europe” (ACARE) is a non-legislative committee of industrial, political and research experts. ACARE has repeatedly formulated research and development targets for aviation as a whole, from air traffic management, to research strategies, to precise technology targets. Flightpath 2050 is the most recent and ambitious iteration of those goals. Table 1-1 lists the technological targets of Flightpath 2050 that are relevant to this investigation. The overall Flightpath goals are given relative to an assumed state-of-the-art in the year 2000. Part of the reduction targets are assumed to be achievable via optimised air traffic management systems, but the majority will require specific developments of aircraft and engine technology. The historical pace of about 2 % annual improvement in energy consumption will not be able to meet those targets [7]. Actually, the annual rate of improvement will drop to about 0.5 % by 2020, because the majority of today’s research still focuses on increasingly small loss sources [8]. The remaining gap between the Flightpath goals and what is conceivably achievable with conventional developments therefore needs to be bridged by revolutionary technology concepts.

Table 1-1: Flightpath 2050 goals relative to year 2000 technology and concept target

Category	Overall Flightpath 2050	Aircraft Specific	From Novel Concepts Alone
Energy consumption and CO ₂ emissions	-75%	-68%	-15%
Total Mission and Cruise NO _x	-90%	-87%	-20%
Noise per operation	-65% (15 dB)	-15 dB	-3 dB

Many studies in recent years, in trying to meet similar targets, have focussed on developing novel aircraft concepts [9] with advanced, though conceptually conventional core engine architectures [10]. An alternative approach is the development of advanced tube and wing aircraft together with ultra-efficient core engine technology. The present study follows the latter approach, as a possible upside of this approach may be higher social acceptance of the more conventional looking new generation of aircraft.

1.1.2 Major loss sources in modern turbofans

Conventional aero engine development often pursues fuel burn reductions through higher overall pressure ratios (OPR), turbine entry temperatures (TET), bypass ratios (BPR) and component efficiencies. Unfortunately, the room for improvement in those and related research areas has diminished significantly over the past 70 years, leaving three major loss sources that have not been vigorously addressed to date. The “Big Three” loss sources are the combustor irreversibilities, the waste heat in the core exhaust and the losses in the ever more influential bypass flow (see Figure 1-1).

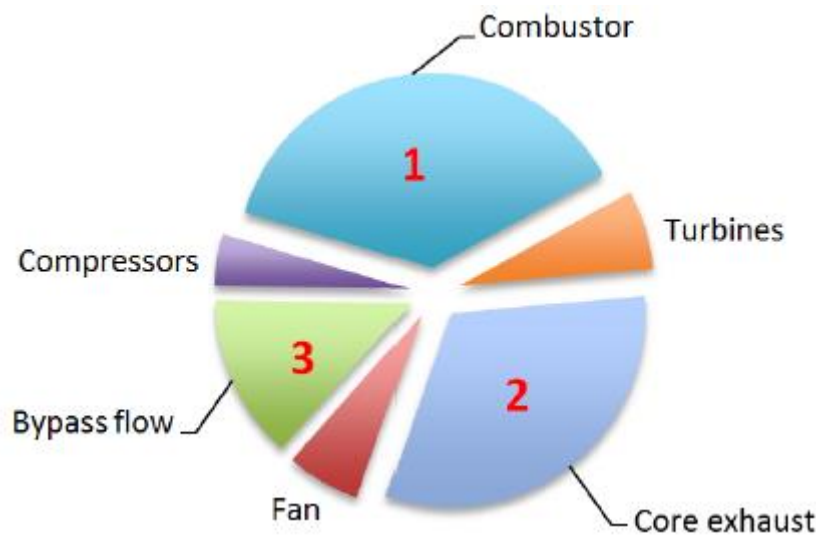


Figure 1-1: The “Big Three” loss sources of modern turbofan engines, from [4]

The “Big Three” are the focus of this and other investigations in ULTIMATE.

1.2 Engine Concept

The residual heat energy in the core exhaust flow of a modern aero engine is one of the “Big Three” [4] loss sources that remain largely unexploited through conventional aero engine development. Previous studies have tried to reduce those losses via direct (recuperation) and indirect (intercooling) methods with differing success. This dissertation investigates the feasibility of what can be described as a combined cycle turbofan engine. A closed Brayton cycle, using supercritical carbon dioxide as its working fluid, draws heat energy from the core exhaust flow of the main engine and uses it to generate further additional power for the low pressure shaft of the turbofan. The combination of lower core exhaust losses and greater shaft power in the LP shaft raise the thermal efficiency of the engine and lower the mission fuel burn. The concept should

furthermore enable a noticeable emission reductions, as CO₂ and partially also NO_x emissions scale with mission fuel burn.

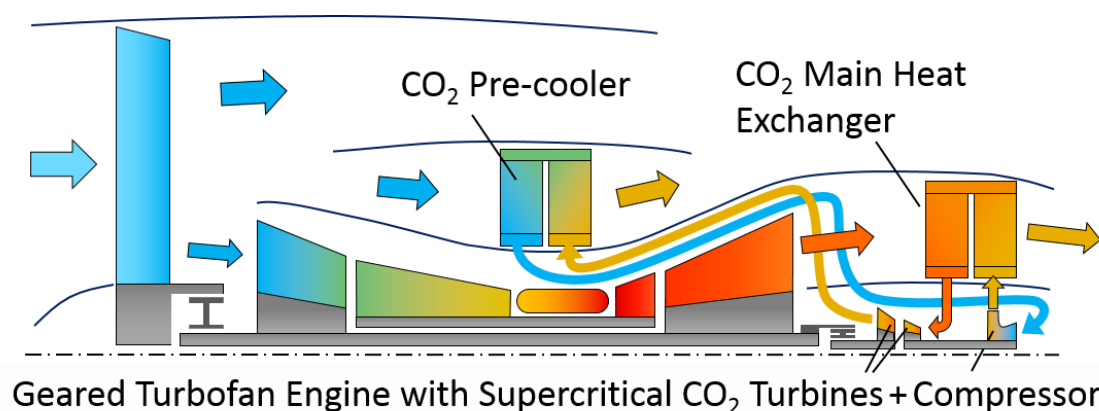


Figure 1-2: Unscaled and simplified engine schematic of CCTF2050

The main engine is a largely conventional geared turbofan engine with an ultra-high bypass ratio (BPR > 16). The closed loop bottoming cycle is located primarily at the back of the engine. Initially, the S-CO₂ is compressed from conditions near the critical point of CO₂. Afterwards, the S-CO₂ passes through the main heat exchanger (MHX), which is located downstream of the low pressure turbine of the main engine. The MHX transfers a considerable portion of the otherwise wasted core exhaust heat to the S-CO₂. Thereafter, the S-CO₂ is expanded first in the high pressure S-CO₂ turbine that drives the S-CO₂ compressor and subsequently in the free-power turbine (FPT). The shaft power generated in the FPT is transferred to the turbofan's LP shaft via a power gearbox.

Eventually, the S-CO₂ passes through the pre-cooler to restore the FPT outlet flow to the desired compressor inlet conditions. The pre-cooler uses part of the bypass flow as the coolant and effectively encapsulates part of the core engine in a similar position as an intercooler would in an intercooled aero engine. The S-CO₂ ducts from the FPT to the pre-cooler and from the pre-cooler to the compressor have to cross the flow path of the

turbofan's core mass flow. However, there will be little to no interaction between the two flows, because the small-diameter ducts should fit inside the support struts behind the LPT.

The pre-cooler only interacts with part of the bypass flow, necessitating a second bypass splitter and two bypass nozzles. Mixing the inner and outer bypass flows before they are expanded through a common nozzle would also be possible, but is not applied in this study to simplify the assumptions and the models.

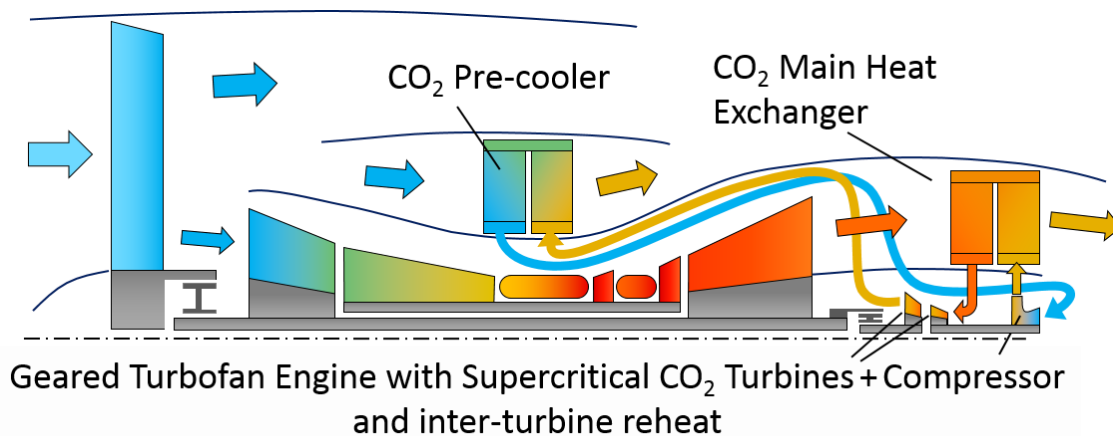


Figure 1-3: Unscaled and simplified engine schematic of CCTF2050_ITR

Adding an inter-turbine reheat (ITR) combustor to the combined cycle concept (see Figure 1-3) increases the core specific power and reduces the core mass flow of the main engine. The LPT exhaust gas temperature (EGT) will also be increased unless the overall pressure ratio of the turbofan can be increased significantly, which means that the MHX sees fewer, but hotter core exhaust gases. Hence, the peak temperatures of the bottoming cycle are increased alongside its thermal efficiency and useful power output. The combined cycle engine with inter-turbine reheat (CCTF2050_ITR) is the second configuration investigated in this dissertation. Whether the CCTF2050_ITR can outperform the CCTF2050 will depend on the heat input and pressure losses in the reheat combustor.

Initially, an S-CO₂ to S-CO₂ recuperator was included in the CCTF2050 simulations, which transferred heat from the S-CO₂ turbine exhaust flow to the compressor exit flow. This resulted in an increased peak temperature in the cycle at a given mass flow and a reduced heat duty for the pre-cooler. However, the chosen recuperator design was too heavy to offer a fuel burn improvement over the non-recuperated configurations, which is why the component was dropped from the final investigations.

1.3 Contribution to Knowledge

The present research evaluates the feasibility and development of a novel turbofan engine concept aimed at entry into service in 2050. The main objective is the conceptualisation and evaluation of a geared turbofan engine featuring state-of-the-art 2050 technology and a closed S-CO₂ Brayton Cycle utilising the main engine core's exhaust waste heat (CCTF2050). Inter-turbine reheat is introduced to the cycle in another configuration (CCTF2050_ITR) to further improve the performance of the combined cycle engine. The investigation is motivated by trying to identify an engine cycle that can meet the aircraft specific Flightpath 2050 [3] goals on an advanced tube and wing aircraft. The Flightpath 2050 goals are referenced against a year 2000 engine technology baseline, but the conventional evolutionary engine development approach will still deliver considerable improvements relative to that year 2000 baseline until the year 2050. The CCTF2050 and the CCTF2050_ITR will therefore be assessed against a baseline TF2000 engine and a reference TF2050 engine, to highlight the specific contributions of the novel components.

The contribution to knowledge of this project has three interlinked parts:

- I. The creation of integrated preliminary design and performance evaluation tools for S-CO₂ power cycles for aero application including the development of:
 - a. Extended fluid property tables and thermodynamic functions of used simulation environment
 - b. Non-dimensional and scalable S-CO₂ turbomachinery maps
 - c. Component models for heat exchangers (MHX, pre-cooler, recuperator) accounting for size, weight, heat transfer and pressure losses
 - d. Accompanying component models for inlet diffusers and outer bypass pressure losses
 - e. Integrated combined cycle models
- II. The application of these tools to assess the performance of the CCTF2050 and CCTF2050_ITR relative to the state of the art
- III. The distinction between the individual contributions of the novel technologies (ITR, bottoming cycle) and their synergistic effects

So far only one relatively recent study is publicly available on S-CO₂ bottoming cycles for aero engines [11]. The application of this technology to an aero engine with all the inherent size, weight and safety constraints is therefore not only challenging but almost completely uncharted territory. Furthermore, the short-comings of the study of De Servi

et al. (see [11] and section 2.4.3) have helped highlight the need for a much more detailed preliminary design and modelling approach, even for conceptual performance studies such as this. Assuming the effectiveness and pressure losses of multiple heat exchangers not only introduces large uncertainty to the thermal performance estimate one attains, it also makes size and weight estimations all but impossible. This is especially true for working fluids with strong real gas behaviour like S-CO₂, as the empirical data to feed first order surrogate models does not exist in the public domain. The first contribution to knowledge is therefore indispensable in order to achieve the others. The created models, particularly for the heat exchangers, enable detailed design and trade-off studies of the effects of varying heat exchanger tube geometries, arrangements and inlet Mach numbers on the overall performance and packaging of the combined cycle engine. Additional side-studies were conducted along the way that investigated whether a larger number of passes in the MHX and/or the pre-cooler would be beneficial, or how the relative weight of the MHX would improve if the aspect ratio of the elliptical tubes were lowered, for example. The obtained data also demonstrates the necessity of such a detailed approach when investigating such a complex concept.

The application of these detailed tools allowed for parametric performance and design studies to be conducted, which ultimately led to the identification of engine configurations that may offer fuel burn benefits over the assumed year 2050 reference engines. The fidelity of these models lends credibility to the obtained results, even though some uncertainty obviously remains, and the models may serve as a building block for further developments in the future. Additionally, no one has applied both an S-CO₂ bottoming cycle and an inter-turbine reheat combustor to an aero engine, not even with lower fidelity models. The second major contribution to knowledge is therefore the application of the newly created tools to investigate these two concepts in appropriate detail.

The simultaneous increase in core specific power and main engine exhaust gas temperature due to inter-turbine reheat is likely to offer synergies with the bottoming cycle. The separate quantification of the individual benefits of these technologies validates the synergy assumption, as the fuel burn benefit of the CCTF2050_ITR exceeds the sum of the concepts' individual benefits. Other studies have investigated the improvement potential of inter-turbine reheat added to state-of-the-art turbfans before, but none of them have compared the resulting engine to a combined cycle engine using a consistent modelling approach to date. The third major contribution to knowledge then stems from the quantification of those individual and synergistic effects.

1.4 List of Publications

The presented work and further side studies led to a number of publications, which the author of this work either contributed to or lead himself. The following list of publications also contains those that are submitted and will be published very soon, or are currently being written.

Lead authorship:

- I. **Jacob, F., Rolt, A., Sebastampillai, J., Sethi, V., Belmonte, M., Cobas, P.** “Performance of a Supercritical CO₂ Bottoming Cycle for Aero Applications”, MDPI Applied Sciences Special Issue Gas Turbines Propulsion and Power, vol. 7, no. 3, p. 27, 2017.
- II. **Jacob, F., Rolt, A., Sethi, V.** “Performance and Preliminary Design of a Combined Cycle Aero Engine”, work in progress, submission targeted by August 2018, target journal either ‘Elsevier - Applied Thermal Engineering’ or “MDPI – Aerospace”

Co-authorship:

- III. **Rengel, J., Jacob, F., Rolt, A., Sethi, V.** “Structural Analysis of Profiled Tubes for a Turbofan Engine Supercritical CO₂ Bottoming Cycle Heat Exchanger”, GT2018-76208, accepted, to be presented at ASME Turbo Expo 2018 in Oslo
- IV. **Rolt, A., Sethi, V., Jacob, F., Sebastampillai, J., Xisto, C., Grönstedt, T., Raffaelli, L.** “Scale Effects on Conventional and Intercooled Turbofan Engine Performance”, ISABE 2017-22659, published and presented
- V. **Rolt, A., Sethi, V., Jacob, F., Sebastampillai, J., Xisto, C.** “Selecting Combinations of Advanced Aero Engine Technologies”, ISABE 2017-22660, published and presented
- VI. **Fakhre, A., Samuelsson, S., Xisto, C., Heinemann, P., Kaiser, S., Jacob, F., Rolt, A.,** „Civil Transport Aircraft Fuel Efficiency – A Long-Term Technology Maturation Scenario“, GTP-14-1147, Journal of Engineering for Gas Turbines and Power, in peer-review

Additionally, the author contributed to or led the writing of multiple EC-reviewed project deliverables, as most of this work was part of Cranfield University’s ongoing efforts in the European research programme ULTIMATE. It was furthermore an honour to be one of the recipients of the ASME IGTI Student Scholarships in 2017.

1.5 Thesis Structure

This chapter provides a general introduction to the challenges engine and aircraft development will face in the future and how they might be tackled. The inner workings of the combined cycle aero engine concept are also explained and the distinct contribution to knowledge of this work is highlighted.

Chapter 2 provides the background information necessary to appreciate the novelty of the concept, what previous information can be built upon and where the short-comings of other studies may lie. Alongside the recent progress of the most relevant competing engine concepts, the chapter also illustrates relevant component level developments that could be built upon in this work. The whole chapter is built to reflect the reasoning behind the specific engine configurations investigated here.

In chapter 3, the general project methodology as well as the details of the individual component models are described. The potential limitations of the models and their implications with regard to the predicted values are pointed out throughout the explanations, to enable a better appreciation of the data presented further on.

Chapter 4 presents the performance of the individual engine configurations as determined using the methods of chapter 3. The results are presented as largely independent arguments of each configuration to enable an in-depth understanding of the observed trends.

Afterwards, the new engine configurations are compared against each other, the Flightpath 2050 goals and other competing concepts in chapter 5. Chapter 5 also approximates the uncertainty around the fuel burn benefits determined in this work, by explaining the most influential modelling assumptions and how they might contribute to unrealistic results. Qualitative discussions of the concept's ability to lower noise and gaseous emissions are also presented alongside a cautious argument of the concept's economic viability.

Finally, chapter 6 provides an overall feasibility assessment based on the obtained results. However, the chapter also acknowledges the remaining uncertainty and discusses possible design and modelling changes that may improve their accuracy and the feasibility of the concept.

2 Literature Review

The concept of combined cycle aero engines has been investigated a few times in the past [11]–[13], but never to the level of detail of the present study, especially when utilising supercritical CO₂ as the bottoming cycle's working fluid. As such, the information required to create the models and draw the assumptions for the present study, was based on research on other working fluids and concepts. This chapter outlines the state of the art regarding concepts and technology related to this work, describes knowledge gaps and explains how and why this study aims to address these gaps. Prior to the presentation of concept-related research from the past, the reference points of this study are presented. The present work aims to quantify the fuel burn reduction potential of the combined cycle aero engine not only against year 2000 technology, but also against an assumed mature engine scenario for the year 2050. Section 2.1 presents these two reference points and briefly outlines how they were identified.

Subsequently, section 2.2 compares some of the most popular novel core concepts for aero engines investigated in the past and in parallel to this work: intercooling, recuperation, inter-turbine reheat and bottoming cycles. The section is limited to concepts that are either immediate alternatives or those that may offer synergies with combined cycle aero engines. Including other engine core concepts such as constant volume combustors may offer further benefits, but these technologies are not discussed, as they address other loss sources and are therefore beyond the scope of this study.

Thereafter, section 2.3 introduces a range of working fluids that could potentially be considered for aero application. Following a brief presentation of the state of the art for each working fluid's power cycles, the reasoning behind choosing S-CO₂ for this study is outlined.

Carbon dioxide exhibits strong real gas behaviour near its critical point, which affects the cycle design and operation. Hence, section 2.4 explains the influence of S-CO₂ fluid properties near the critical point before outlining the current research status regarding S-CO₂ power cycles for other applications. The particular needs of aero application and the limited amount of previous, publicly available, research are discussed as well.

Sections 2.5 and 2.6 then introduce heat exchanger and turbomachinery design concepts relevant for the present study. Finally, section 2.7 briefly summarises the reviewed literature to accentuate the originality of this work.

2.1 Reference and Baseline Engine and Aircraft Performance

This study aims to address the specific propulsion challenges set by the Advisory Council for Aviation Research and Innovation in Europe (ACARE) in 'Flightpath 2050' [3] through novel aero engine cycles. Year 2000 state-of-the-art technology was assumed as the reference point in Flightpath 2050 for technology targets such as energy consumption, and hence CO₂, reductions by 68 % and mission NO_x reductions by up to 90 %.

The novel combined cycle engine concept using supercritical CO₂ as the working fluid in the bottoming cycle therefore needs to be compared against an assumed representative year 2000 engine model, in order to adequately quantify its benefits. The presumably high complexity and weight of the combined cycle engine will most likely only be feasible when applied to long-range intercontinental missions, where the fuel savings at cruise are more influential than on a short-range mission. This section details the engine performance and weight estimation of the Long-Range Turbofan 2000 (LRTF2000), an engine similar to the Trent 772B-60 used on an Airbus A330-300. The modelling assumptions and final engine specifications for the LRTF2000 are briefly highlighted in section 2.1.1 to establish the year 2000 baseline.

It can be reasonably assumed that conventional means of aero engine development, such as raising the overall pressure ratio and turbomachinery efficiencies, and introducing new materials, will no longer be offering significant improvements by 2050. The historical pace of engine development and the individual contributions of the different drivers behind that development have been presented many times before [8], [14] and shall not be regurgitated here. The aforementioned development plateau may not be reached for 15 –20 years or more, but it is necessary to predict the performance and design of the intercontinental reference engine incorporating full development of these conventional technologies. Applying the bottoming cycle and inter-turbine reheat to this reference engine, rather than the baseline year 2000 engine, enables the isolation of the achievable performance benefits of these novel technologies over the assumed state of the art in the year 2050. The modelling assumptions, the performance data of the engine and its general arrangement and weight predictions are briefly presented in section 2.1.2.

Any performance modelling presented in this study was done in version 3.6.14 of the 'Propulsion Object Oriented Simulation Software' (PROOSIS). The consistent modelling of the baseline, reference and the novel engines in the same simulation environment ensures that the results are reasonably comparable.

2.1.1 The year 2000 baseline

Flightpath 2050 specifies several performance and emissions improvements for aero engines relative to year 2000 technology. To quantify any benefits that arise from the novel engine technologies investigated herein, the baseline characteristics of those year 2000 engines need to be established. For intercontinental missions, an engine similar to the Trent 772B-60 has been identified as representative of year 2000 technology and hence a model similar to the Trent772B-60 has been produced (LRTF2000). The 772B-60 is used on the A330-343, so an aircraft model with traits similar to the A330-300 was created by ULTIMATE project partner Bauhaus Luftfahrt.

The Trent 772B-60 has been chosen as the baseline engine to represent year 2000 engine technology in a number of previous research programs such as NEWAC [15], LEMCOTEC [16] or VITAL [17]. The model was created to mirror the specifications and performance of those baseline models, because more detailed data was available for those than for the original Trent 772B-60. The LRTF2000 reproduces the engine performance of those models to within 1 to 2% for most thermodynamic parameters, with a maximum deviation of about 5 % at End-of-Runway conditions. Figure 2-1 shows the difference in engine performance for selected parameters as exhibited by the LRTF2000.

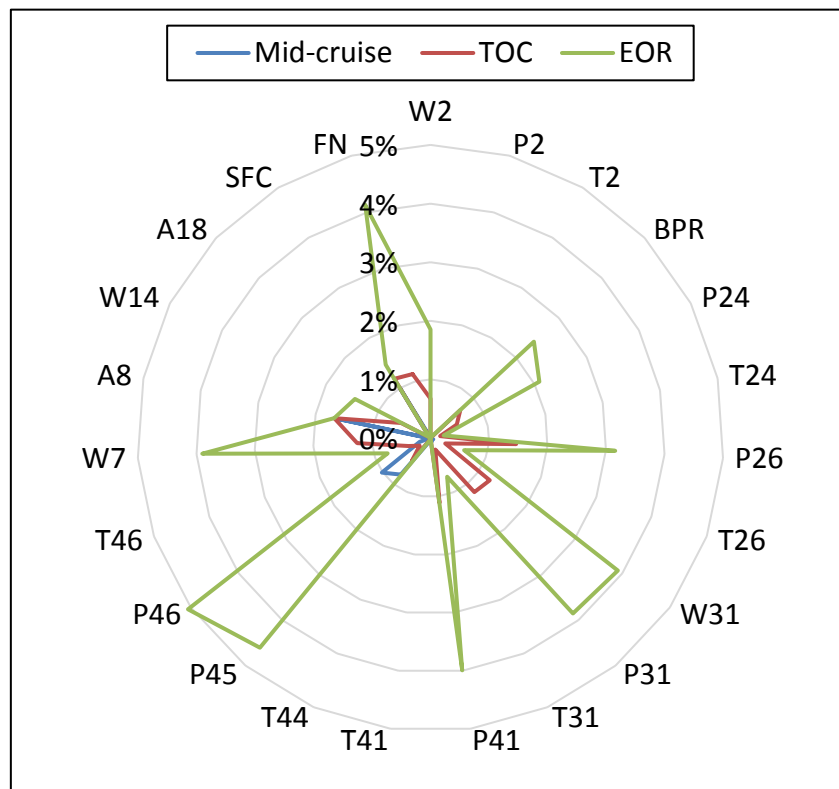


Figure 2-1: Remaining errors after LRTF2000 matching

The Trent 772B-60 consists of:

- A single-stage high bypass ratio fan
- An eight-stage intermediate pressure compressor
- A six-stage high pressure compressor
- An annular RQL combustor
- A single-stage high pressure turbine
- A single-stage intermediate pressure turbine
- A four-stage low pressure turbine

The weight estimation and dimensions software WeiCo [18] was originally developed by Chalmers University during the VITAL programme and has been developed further ever since. WeiCo has been used in this investigation to assess the size weight of all conventional turbofans and their components. Among other things, WeiCo determines the dimensions and weight of an engine based on a modular approach that evaluates each major component such as fan, compressors, burner, nacelle, shafts and more.

WeiCo has been used to estimate the total engine weight (7972 kg) and replicate the engine general arrangement of the Trent 772B-60 as closely as possible (see Figure 2-2). The mission fuel burn of the LRTF2000 was estimated to approximately 86.4 t, when applied to the intercontinental year 2000 aircraft model. Some overall performance data of the LRTF2000 model are presented in Appendix A.

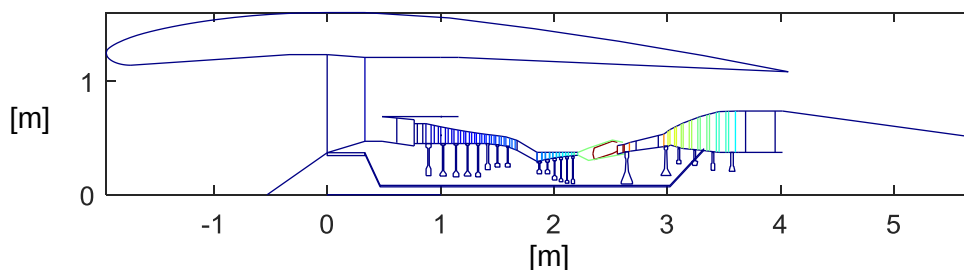


Figure 2-2: Engine cross-section of the LRTF2000 baseline engine from [19]

2.1.2 The year 2050 reference

The envisioned year 2050 advanced tube and wing (ATW) aircraft for long-range intercontinental missions is substantially more aerodynamic, and therefore produces less drag, than the baseline aircraft (with an A330 like configuration) due to developments including improved materials, fly-by-light and hybrid laminar flow control. As such, the

thrust requirements of the aircraft are considerably lower for the same payload and even higher range capabilities.

The modelling of the reference engines and aircraft was done in parallel, with Bauhaus Luftfahrt being the custodian of the aircraft modelling exercises. The initial technology parameters of the reference engines were determined by the industrial partners in the ULTIMATE program, based on their expertise and future projections. An Ultra-High Bypass Ratio (UHBPR) geared turbofan engine is assumed to represent the evolutionary limit of existing engine architectures for the year 2050 (TF2050). The design features a reduction gearbox between the high-speed low pressure turbine (LPT) and the fan. The intermediate pressure compressor is driven directly by the LPT without any speed reduction. The HPT and the HPC both feature tip clearance control to reduce over-tip leakage (OTL) losses and maintain high component efficiencies despite the high overall pressure ratio and small engine core. The maximum flow temperature of 880 K at the HPC exit was imposed when deriving the optimum cruise OPR of the TF2050. Further increases in HPC delivery air temperature (T_3) may be possible, but would require the increased use of expensive and heavy nickel super-alloys in the HPC. An increase in T_3 would also give rise to the NO_x emissions produced by the engine.

Likely developments in material technology are projected based on public domain data and experiences of the industrial partners. Further development of ceramic matrix composites (CMC) beyond what is already being tested in the GE9X [20], and cooled cooling air heat exchangers are assumed to enable CMC NGVs without film-cooling in the HPT of the TF2050. When modelling the engine, the mass flow percentage used to cool the HPT was assumed to be constant throughout the flight envelope. The cooling air requirements were significantly reduced relative to the year 2000 and were derived by enforcing a blade metal temperature maximum for the CMC HPT NGVs. The combustor is assumed to be a lean direct injection (LDI) design, which is assumed to significantly reduce the emissions footprint of the engine [21], [22]. The TF2050 is furthermore assumed to be bleed-less, like the Trent 1000 for example, though the TF2050 does not feature shaft power off-takes for the aircraft environmental control system (ECS). The ECS power is supplied by hydrogen fuel cells, eliminating the need for generators on the engines and the auxiliary power unit. The fuel cells are furthermore used during taxi to enable fully electric taxiing to reduce emissions during the landing-and-take-off (LTO) cycle.

Due to its very low specific thrust design, the engine may require either a variable pitch fan or a variable area nozzle. The variable pitch fan may also be used to provide reverse thrust during landing, eliminating the need for thrust reversers in the nacelle and reducing weight. Detailed studies of VPF technology and its capabilities are currently being conducted confidentially by ISAE within the ULTIMATE programme [23] to determine if these benefits may actually manifest in this particular engine. For the purposes of this study only a variable area cold nozzle is assumed to be in place on the TF2050. The most important performance data of the TF2050 is listed in Appendix B.

The TF2050 consists of:

- A single-stage ultra-high bypass ratio fan with a reduction gearbox to
- A three-stage intermediate pressure compressor
- A ten-stage high pressure compressor
- An annular LDI combustor
- A two-stage high pressure turbine
- A four-stage low pressure turbine (driving IPC and fan)

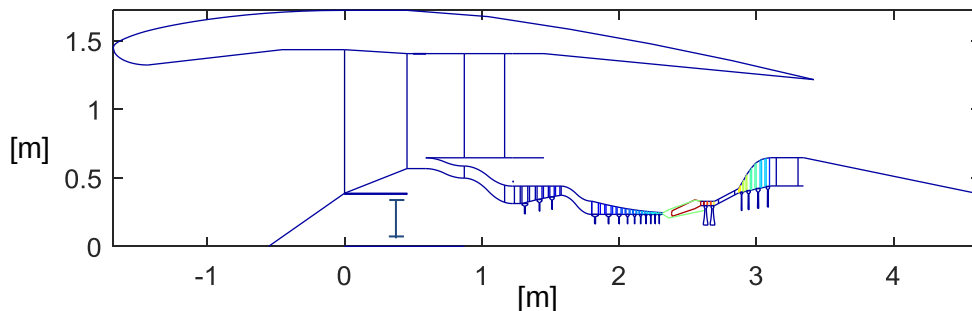


Figure 2-3: Engine cross-section of the geared TF2050 reference engine from [19]

The weight and general arrangement estimation tool WeiCo, developed by Chalmers University, was calibrated to include the material assumptions for the year 2050. Figure 2-3 shows a schematic engine cross-section of the TF2050. The total engine weight was estimated at 5161 kg. The engine weight and cross-section were deduced from WeiCo while keeping parameters such as the stage counts of the turbomachinery components constant during the engine-aircraft design iterations. The engine has therefore not been optimised with respect to the flow cross-section. Further improvements of the engine design may therefore be possible through variation of the stage counts and the coupling of the component efficiencies to the flow path designs. However, those improvements

were assumed to be negligible for the present study, given the amount of technology assumptions already incorporated in the engine.

The iterative design of the TF2050 and the year 2050 advanced tube-and-wing (ATW) aircraft furthermore led to the derivation of non-linear trade factors (NLTFs). The NLTFs can be seen as a simplified aircraft design platform, in that they represent a host of different aircraft designs with different fuel burn characteristics. The NLTFs are used in conjunction with the PROOSIS engine models and WeiCo to evaluate the novel engine configurations in this investigation. Using changes in mission average specific fuel consumption (SFC_{ave}) (see equation 2-1), total engine weight and fan diameter, the NLTFs can be used to calculate the mission fuel burn improvement relative to the reference design (TF2050). The NLTFs also account for necessary changes to the airframe such as stronger support beams in the wings for heavier engines or longer landing gears for larger engines that would otherwise violate ground clearance minima.

$$SFC_{ave} = SFC_{cruise} * 0.929 + SFC_{TOC} * 0.061 + SFC_{EOR} * 0.01 \quad 2-1$$

Any change in mission fuel burn obviously also affects the aircraft weight and hence, thrust requirements. The NLTFs can therefore also be used to determine those new thrust requirements for a new engine, triggering an iterative design approach that closely matches engine and aircraft specifications. The payload of the aircraft is unaffected by the NLTFs. The number of design iterations necessary to reach convergence depends on the change relative to the TF2050, but usually three to five iterations suffice, making this a powerful yet computationally inexpensive tool. The mission fuel burn of the TF2050 was calculated to approximately 46.8 t using the NLTFs, representing a reduction of about 45.8 % over the LR2000. The simulation and optimisation of the LR2000 and the TF2050 were part of this author's research efforts for this thesis and ULTIMATE.

2.2 Rivalling Concepts

The evolutionary engine development process will probably have exploited most of the improvement potential of the conventional turbofan engine by the year 2050. However, three major loss sources still remain: the combustor pressure loss, the bypass losses and the excess heat in the engine exhaust gases. Adding a bottoming cycle to a standard turbofan engine is one possible concept that may help reduce the exhaust gas temperature and produce additional shaft power, attacking the third loss source. Each component in the combined cycles affects the performance of the engines in multiple ways. For example, the second engine configuration investigated in this work, the

CCTF2050_ITR, also includes a secondary combustor. The reheat combustor will increase the combustor outlet temperature and improve the bottoming cycle's performance, but it also introduces further combustor irreversibilities through additional pressure losses. The pre-cooler on the other hand will raise the mean temperature and reduce the mean pressure of the bypass flow. The exhaust losses are the primary focus of the combined cycle engine. Whether or not the combined cycles offer a fuel burn benefit over the TF2050 will depend on the balance of these and more effects.

Other concepts, such as intercooling and recuperation, also aim to raise the thermal efficiency of the turbofan and mitigate exhaust losses, making them direct competitors for bottoming cycles. This section briefly describes the competition, starting with intercooling in section 2.2.1. Section 2.2.2 moves on to recuperated and intercooled-recuperated engine concepts. Section 2.2.3 gives an overview of past and ongoing research with regard to inter-turbine reheat. Finally, section 2.2.4 gives a brief overview of the state of the art when it comes to bottoming cycle research and development.

2.2.1 Intercooling

The introduction of an intercooler in the compressor system of a large civil turbofan enables an increase in overall pressure ratio of the turbofan at a constant compressor delivery temperature. This in turn reduces the exhaust gas temperature of the engine for a given turbine inlet temperature, addressing the exhaust gas losses of the engine.

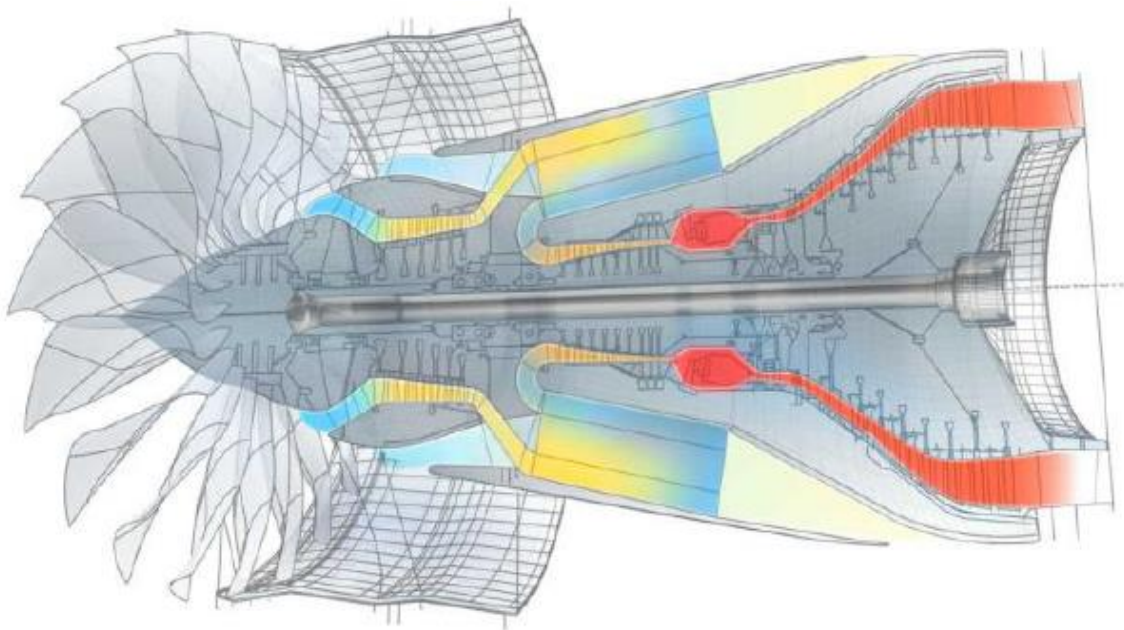


Figure 2-4: Schematic cross-section of NEWAC intercooled engine, from [24]

Part of the bypass flow of an intercooled engine is used to cool the core flow between the intermediate and the high pressure compressors. Intercooling reduces the compression work by reducing the compressor inlet temperature of the HPC and the volume flow rate passing through it. For stationary applications, the reduction in compressor work and subsequent drop in compressor delivery air temperature may be used to increase the fuel flow in the combustor and increase the net power output of the engine. In a modern turbofan, this will lead to an SFC increase however and is therefore not pursued. Instead, the OPR is increased without exceeding the HPC delivery air temperature limits. The OPR does not necessarily have to be increased until the HPC delivery air temperature limit is reached though. That way, the SFC and the NO_x emissions of the engine could be reduced at the same time. NEWAC [15] showed that intercooling could reduce NO_x emissions relative to the conventional cycle in high-OPR engines, because the combustor inlet air temperature may be reduced. The NO_x reduction is most noticeable with conventional combustors, where higher inlet temperatures would result in higher flame temperatures. Intercooling also reduces NO_x emissions with lean-burn combustors, but the benefit is not as large because their flame temperatures are less sensitive to combustor inlet temperature.

The OPR of an intercooled engine is furthermore no longer primarily limited by the HPC delivery temperature, but by excessively small blade heights in the rear stages of the HPC. The reduced volume flow rate and higher overall pressure at the back of the compressor lead to very small compressor blades. Small compressor blades are prone to proportionally large over-tip leakage, which affects the efficiency of the whole compressor. Initial intercooler studies assumed a three-shaft design, as can be seen from the intercooled engine schematic from the NEWAC project in Figure 2-4. Having three concentric shafts limits the rotational speed of the shaft and the minimum possible bore diameter of the HPC, leading to higher hub-to-tip ratios in the compressor. Later designs in NEWAC and LEMCOTEC assumed a geared engine design rather than a direct-drive design, alleviating some of the constraints on the inner shaft diameter and henceforth allowing for a faster shaft with smaller hub-to-tip ratios and larger blades. Hence, the HPC efficiency was improved and the optimum OPR could be increased further. The geared drive design inherently leads to a large HP pressure ratio for a given OPR, which further improves the mission fuel burn of an intercooled engine by introducing the intercooler earlier in the compression process.

William Camilleri and Eduardo Anselmi Palma investigated a geared intercooled reverse flow core (GIRFC) concept in the LEMCOTEC project to enable further increases in OPR [25], [26]. The HP and LP shafts are not arranged concentrically but sequentially in the GIRFC. This configuration eliminates the inner shaft diameter limit for the HPC and increases blade heights, enabling further OPR increases. However, the GIRFC also features a complex, heavy and pressure loss inducing exhaust system that eliminates much of the benefit. The GIRFC is also likely longer than the GISFC. The detailed mechanical design of the system showed, that the GIRFC only holds approximately 1 % of fuel burn benefit over a more conventional straight flow core arrangement [26].

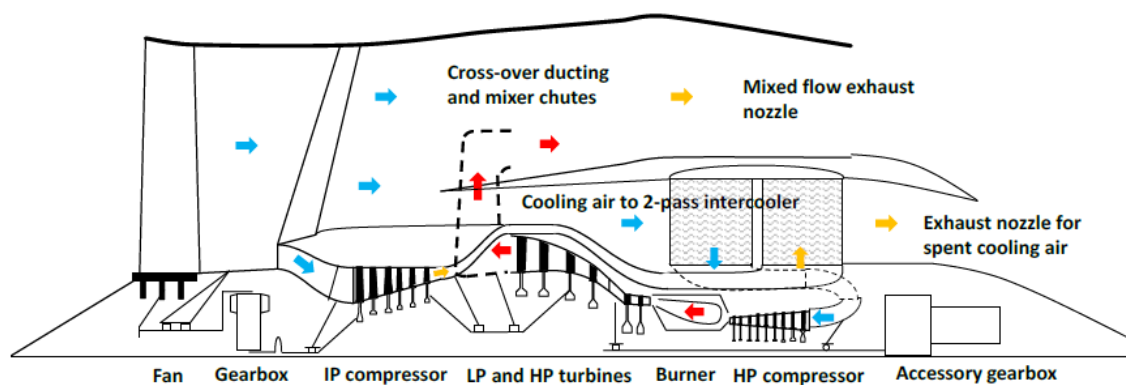


Figure 2-5: Schematic of Geared Intercooled Reverse Flow Core concept of Camilleri et al. [27]

The reduced volume flow of the HPC also saves weight by reducing cross-sectional flow areas in the whole HPC. Unfortunately, the intercooler itself adds weight and increases drag, offsetting much of the potential fuel burn benefit. The intercooler enables the greatest benefits during hot conditions such as take-off and climb. During cruise, excessive cooling of the core flow will lead to increased fuel consumption and hence increased SFC. From an operational standpoint it is therefore favourable to modulate the coolant mass flow with a variable area nozzle at the back of the intercooler bypass duct [28] or by bypassing some of the IPC delivery air around the pre-cooler [29]. This reduces the temperature drop in the core flow and the pressure losses of the bypass air at the same time, reducing the SFC. The additional weight of a variable area nozzle will however offset a considerable portion of those benefits yet again.

Overall, intercooling has been shown to improve the cruise SFC of a large turbofan similar to the TF2050 by around 3% [15] and the mission fuel burn by up to 4.9% [29]. Relative to a large turbofan engine model representative of year 2000 technology, the

fuel burn benefit may be as high as 21.6 % [25]. For smaller engines, the benefits are somewhat less due to even smaller engine cores.

2.2.2 Recuperation

In a recuperated engine, part of the core exhaust heat is transferred to the high pressure compressor delivery air. McDonald et al. presented the concept, its historical development and the potential performance in detail [30]–[32].

In previous concepts and existing engines, such as the Rolls-Royce WR21 [33], the air is transferred in a non-mixed separate flow recuperator, but in small gas turbines a regenerator often replaces the recuperator due to size constraints. In a regenerator, the two separate flows alternately pass through the same heat exchanger channels. Civil aero engines are large enough to favour recuperators over regenerators. The improvement potential due to the recuperator depends on the temperature difference between the turbine exhaust gases and the compressor delivery air, automatically favouring lower OPR designs. Combining recuperation and intercooling raises the optimum OPR, because the HPC delivery air temperature is not excessively increased. However the turbine exhaust temperature is reduced, so an ‘optimum’ intercooled-recuperated aero engine (IRA) will still feature a lower OPR than an intercooled non-recuperated engine. The Rolls-Royce WR21 engine was developed for marine propulsion and was developed based on the RB211 aero engine. During the redesign, the OPR was lowered from around 30 to about 20 for optimum performance.

Previous research programs such as NEWAC [34] and LEMCOTEC [35] have shown that intercooled and recuperated engines can offer significant performance improvements over purely intercooled engines, particularly if the core size is significantly reduced as in ultra-high bypass ratio engines or in engines with lower thrust ratings. NEWAC also showed that the intercooled and recuperated engine could enable lower NO_x emissions than a conventional engine, due to the often-reduced combustor inlet temperatures, even after recuperation. In NEWAC the lower inlet temperatures were utilised by a Lean Premixed Prevaporized (LPP) combustor to further reduce NO_x emissions. Depending on legislative boundaries and operating costs, SFC and NO_x emissions may be traded against each other to identify the optimum configuration. The recuperator in the NEWAC engine (see Figure 2-6) was based on the earlier work by MTU as presented by Boggia et al. [36]. Eight modules of profiled tubes were placed around the engine nozzle cone to form the recuperator. During the subsequent research

programmes LEMCOTEC and especially ULTIMATE, that design was further improved upon to improve the recuperator's effectiveness while reducing its weight, pressure loss and installation penalties [35].

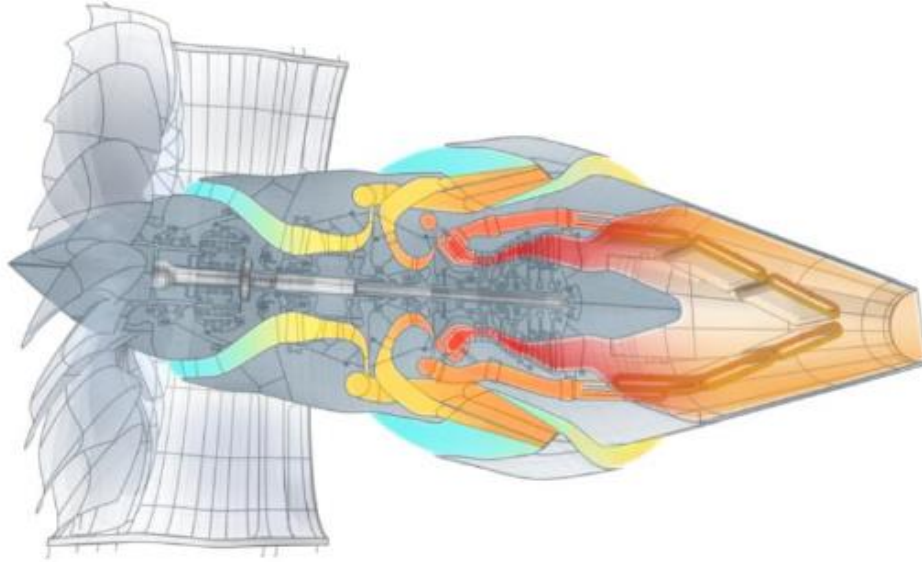


Figure 2-6: Schematic engine cross-section of NEWAC geared, intercooled and recuperated engine concept [24]

Theoretically, the SFC of the intercooled and recuperated engine could be further reduced by extracting more heat at an earlier point in the expansion process. This would result in an increased combustor delivery temperature which would aid fuel efficiency but may increase NO_x emissions. A different combustor design may be necessary to accompany such a concept change to avoid auto-ignition or flash-back scenarios. The additional installation difficulties, the pressure losses and the reduced inlet temperature to the turbine stages downstream of the high temperature recuperator will impede the concept's improvement potential however, necessitating a careful optimisation to prove the overall improvement potential of the redesign.

Ultimately, the performance of the IRA engine will largely depend on the relative importance of low emissions and specific fuel consumption. NEWAC and LEMCOTEC emphasized low NO_x emissions during the LTO cycle, sacrificing some of the SFC potential. However, these engines still promised up to 13% SFC improvement over a conventional cycle with EIS 2000. The technology levels of the reference engine will also affect the actual benefit of the concept, but ongoing studies based on projected year 2050 conventional reference engines indicate that significant improvements can still be achieved. One of the designs currently favoured by Salpingidou et al. [37] eliminates the

need to recirculate the HPC delivery air from the back of the engine and to the combustor by employing a secondary heat transfer fluid, reducing the cold-side pressure losses.

2.2.3 Inter-turbine Reheat

Different people use the term “inter-turbine reheat” (ITR) in different ways. In a multi-shaft engine the second combustor can be positioned between different turbines on different shafts (e.g. between the HPT and the LPT) or between different stages of the same turbine (e.g. HPT1 and HPT2). The former is most commonly referred to as inter-turbine reheat, while the latter is sometimes also referred to as sequential combustion or intra-turbine reheat. Whether the secondary combustor requires altitude relight capabilities will depend on the operation of the second combustor at cruise. However, the main combustor would always be relit first, making ignition in the second combustor much easier without additional design changes. If the combustor is only used at take-off and climb, which is likely in an otherwise conventional turbofan, no altitude relight would be necessary. On the other hand, in a combined cycle engine, the ITR may be used to increase the LPT exhaust gas temperature and hence the bottoming cycle efficiency, so reliable and continuous operation of the second combustor should be guaranteed.

Thermodynamically it would be most effective to burn all the fuel in a primary combustor, rather than diverting some of it to the reheat combustor. However, temperature limitations of the materials in the combustor and HPT, cooling requirements and the exponential increase in NO_x emissions beyond a TET of approximately 1900 K pose practical limits to this approach. ITR increases the thermal efficiency of the cycle by raising the mean temperature of heat addition and by increasing the core specific power. The increase in core specific power means the losses induced during compression and expansion, which experience a relative increase due to the smaller turbomachinery, are applied to a smaller core mass flow, reducing their impact. To compensate for the reduced core mass flow, the LP turbine may need a higher pressure ratio and an extra stage, but with a high-speed LP turbine and a geared fan, the smaller core mass flow should still tend to reduce overall engine weight.

Introducing the second combustor late in the expansion process raises the exhaust gas temperature (EGT) of the engine, which can be useful when coupled with a recuperator or a bottoming cycle as in a combined cycle power plant. However, in a conventional aero engine, a number of studies [27], [38], [39] agree that the reheat should take place as early in the expansion as possible to grant the biggest improvements. Hence, the reheat

combustor should be placed between HPT1 and HPT2. In his study of a medium bypass ratio turbofan engine, Vogeler [38] determined that it would be thermodynamically ideal to place the reheat combustor after 7-10 % of the expansion process has taken place. However, this would result in an unbalanced turbine design. A more reasonable assumption seems to be that of Koehler [39], where the turbine work split between HPT1 and HPT2 is kept in the range of 45 to 55%, resulting in a more realistic and balanced HPT design. The MSc study of Koehler is one of the most detailed, publicly available, design and off-design performance investigations of inter-turbine reheat. Additionally, the study is based on the UltraFan concept, an engine with more similar performance characteristics to the TF2050 engine, than any other publicly available study on ITR. His models include an effective representation of the secondary air system of the engine to capture the increased cooling requirements of an ITR engine. Additionally, the operating freedom of the ITR engine is highlighted through the discretisation of the HPC and the subsequent demonstration of the different effects ITR has on the operating point of the individual sections. Adjusting the fuel-flow-split between the primary and secondary combustors provides an additional off-design variable, with a potential benefit arising from preferentially reducing the secondary combustor fuel flow at part-power conditions. As a result, the engine may run with a higher OPR at cruise. Koehler determined, that ITR could help improve the SFC of the UltraFan engine by approximately 1.5% or that it could be used to increase the specific thrust of the engine by up to 22%. Even though that increase in specific thrust would not grant any SFC benefit, it may reduce the engine weight and drag, improving the mission fuel burn of the engine. Unfortunately, Koehler did not include a rigorous mission fuel burn assessment of his configuration. He did acknowledge that the reheat combustor would increase the SFC at TOC and EOR, which meant that the mission fuel burn benefit would be reduced relative to what the cruise data may indicate. However, without an accurate portrayal of the weight and drag implications of introducing ITR to the engine, these figures are only approximations. Additionally, Koehler did not include a penalty function for excessively small blade heights in the high pressure turbomachinery components, which will lead to an overestimation of the performance improvements that can be gained from greater core size reductions and higher OPRs. The lifing implications of the higher OPR and TET at cruise were furthermore neglected. Finally, the outlet temperature of the second combustor was allowed to exceed the outlet temperature of the main combustor during the investigation, which seems an unlikely scenario considering the cooling and lifing implications this would have on the HPT and especially the LPT.

One of the very few existing commercial gas turbines featuring inter-turbine reheat is the GT24/GT26 family originally developed by Alstom Power. Alstom called the technology sequential combustion and even though this engine was developed for stationary and preferably combined cycle applications, the reheat combustor is placed early in the expansion system, behind HPT1. The GT36 of Ansaldo Energia [40] is the newest iteration of that design and is supposed to extend the configurations' outstanding part-load performance and hence operating flexibility.

The limitations of aero application and the resulting small engine cores may limit the SFC improvements that may be gained from ITR alone, but the potential for weight reduction could make a significant contribution to reducing fuel burn. Combining ITR with technologies that capitalise on the increased LPT EGT, such as recuperation or in this case a bottoming cycle, may enable significant SFC improvements as well. Approximating the extent of this synergistic effect is part of the present study.

If possible, the secondary combustor would be of an 'oxy-poor-flame' design made possible by the high inlet temperature, as was developed by Levy et. al [41] in the AHEAD project [42]. Martinez Bueno [43] investigated the design of a similar combustor in the ULTIMATE project and Mirko Romanelli is presently developing the concept further. Due to the novel combustor design and the reduced heat input relative to the main combustor, it may be assumed that the pressure losses in the secondary combustor are lower than in the main combustor (e.g. 2% total pressure loss rather than 3%). The pressure losses would furthermore be relatively small, because the projected year 2050 models assume that the downstream NGV does not need film cooling. The novel design may also help address the potential increase in NO_x emissions of the engine with ITR, because it may improve the temperature distribution in the combustor and because it works with fluid mixtures closer to stoichiometric conditions. On the other hand, some issues with regard to temperature distributions and increased UHC emissions still need to be addressed.

2.2.4 Bottoming Cycles

Bottoming cycles are secondary power cycles that capitalise on the exhaust gas temperature of the main engine to produce further power or, in the case of combined heat and power plants, steam. Bottoming cycles therefore directly compete with recuperators in that they both draw heat from the engine's core exhaust. A combination of the two concepts may offer even greater SFC reductions than each technology

individually, though the incurring complexity of the engine and the additional weight and drag of the components are likely to eliminate any benefit for an aero engine.

Most bottoming cycles are closed power cycles using a different working fluid than the main engine, although an integrated open air bottoming cycle is also conceivable. Air behaves like an ideal gas, resulting in a significant temperature rise and power consumption during compression. Most closed cycle working fluids on the other hand are compressed close to their critical temperatures, resulting in more efficient cycles due to the real gas behaviour of the fluids or they are compressed in a saturated state, which reduces the necessary compression work. The higher efficiency closed cycles usually require large pre-coolers or condensers however, which are not necessary in an open air cycle, making it impossible to definitively declare closed cycles the more attractive option for any application. Many closed power cycles operate at very high overall pressure levels, which lead to very compact and power dense cycles, especially when operating near the critical point of the working fluid. Steam cycles are most common in industrial power plants, representing over 90% of the existing Rankine cycles worldwide. However, steam cycles commonly feature very large and heavy components. The different working fluids and the respective design implications are explained in detail in section 2.3. The section also explains why supercritical CO₂ was chosen for the current investigation.

The current study anticipates the provision of additional shaft power to the main engine, but that is not necessarily the case for all bottoming cycles. Depending on the application, the produced power could be used to drive auxiliary systems such as the Environmental Control System (ECS) of the aircraft for example. Alternatively, the closed power cycle could be used as an alternative intercooling system, that also produces shaft power, as described by Petit et al. [44], or to provide anti-icing capabilities [12].

2.3 Bottoming Cycle Working Fluids

Bottoming cycles for aero application have been a largely overlooked concept to date, as the few conducted studies predicted the machinery and heat exchangers involved would be too large and heavy. This assessment however depends largely on the working fluid involved. To that end, five potential types of working fluid have been considered: Water, Air, Organic fluids (e.g. R245fa), Ammonia-Water solutions and Supercritical CO₂. Cycles with each of these fluids have not only been theorised but tested for different applications, with steam (water) being the most common working fluid in bottoming cycles for the commercial power generation market.

2.3.1 Water / Steam

State-of-the-art large scale power generation plants with steam as the working fluid in the bottoming cycles have recently breached the 62 % thermal efficiency line for the combined cycle [45]. That represents an approximate 20 % increase in power plant efficiency from the bottoming cycle, which would be impossible to achieve in a simple gas turbine cycle. Steam bottoming cycles offer very good standalone efficiencies at medium to high temperatures, because the pumps only consume about 3% of the generated turbine shaft power. Steam is also a largely unreactive, non-toxic fluid, which reduces health and safety concerns. However, any working fluid impurities such as salt can quickly result in corrosion and failure problems for the cycle. Moreover, steam cycles are very large and heavy due to huge pressure ratios, values above 5000 are not uncommon, and extremely large and heavy heat exchangers. While the heat exchanger size could be reduced if a lower effectiveness and higher pressure loss is accepted, it will never be as small as even an air-to-air heat exchanger, because the water undergoes an isothermal phase change. Consequently, a substantial portion of the heat energy introduced to the bottoming cycle is used to evaporate the water without raising the average fluid temperature. Hence, the heat exchanger has to be large enough to maximise the heat recovery to the steam cycle for a given turbine entry temperature in the bottoming cycle. Multi-pressure level designs have remedied that limitation for stationary applications, but the inherent complexity of those cycles eliminates them as an option for aero engines for the time being. Additionally, steam cycles require relatively high exhaust gas temperatures of around 900 K to reach these outstanding efficiency values. These temperatures are well beyond the EGT of a modern turbofan at cruise, where the engine concept is supposed to provide the biggest fuel burn reduction.

2.3.2 Organic Fluids

Organic Rankine Cycles (ORC) are a popular field of study for small scale power cycles (typically between 0.02 and 2 MW) with a low-grade heat source. The inclination of the saturation curves toward higher entropies (see Figure 2-7) means that the entire expansion process takes place in the superheated region, which reduces droplet erosion, increases reliability and allows for faster start-ups. One of the reasons ORCs are an attractive option for low-grade heat sources is their relatively low condensation temperature. Clemente et al. [46] showed ORC can feature good cycle efficiencies with low grade heat sources.

In an aero engine however, the temperature of the LPT exhaust gases, the heat source of the bottoming cycle, varies considerably throughout the flight envelope. Even during cruise, a relatively cold steady-state condition, the EGT is considerably above the temperature limits of most ORCs for stable operation. Clemente et al. note that the critical points of most organic fluids are below 600 K, for many even below 450 K. The mass flow of an ORC bottoming cycle would therefore need to be sufficiently high to avoid the fluid's critical point, increasing the cycle weight. Alternatively, the heat exchangers' effectiveness could be limited, which meant that only a fraction of the available heat energy would be recovered. In addition, while ORC cycles are considered reliable and capable of good part-load behaviour, the temperature variation throughout a mission will likely require a large capacity fluid tank in order to alter the bottoming cycle's mass flow at hot conditions adding further weight. Moreover, the cooling from the turbine exit to the compressor or pump inlet conditions would once again require an isothermal phase change, which would increase the pre-cooler size significantly.

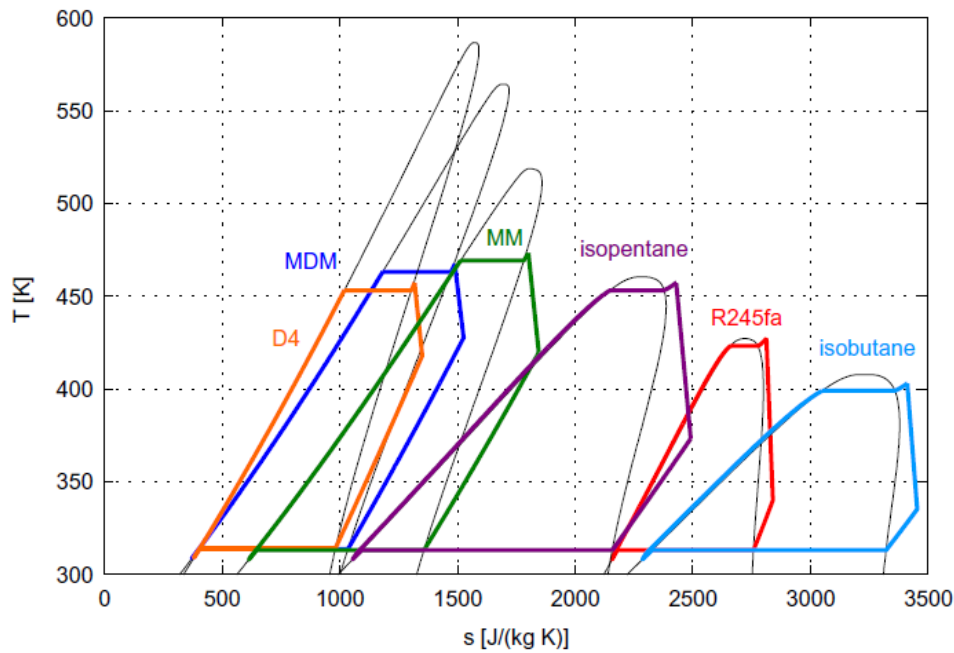


Figure 2-7: Representation of real ORC cycles by Clemente et al. [46]

Despite these limitations, the power density (the power output per unit cycle weight) of an ORC is still considerably higher than that of a steam cycle for low-to-medium grade heat sources. That is one of the reasons why Perullo et al. [12] have conceptualised an ORC bottoming cycle for application to a CFM56. The bottoming cycle did not deliver additional shaft power to the main engine, but was designed to replace the conventional power generators for the environmental control system (ECS) of the aircraft and provide

anti-icing. The study only predicted a 0.9% fuel burn reduction though this might have risen to 2.3%, if the new components could have been designed to be weight neutral to the reference engine, an unlikely scenario. Perullo et al. appear to have neglected the pressure losses induced by the heat exchangers, which means the given estimates may be too optimistic still. It should be noted that long-range aircraft are generally less sensitive to changes in engine weight and Perullo et al. may have chosen an unfavourable baseline engine in their study.

The more recent study of Petit et al. [44] investigated ORCs as an alternative intercooling concept that also produces shaft power. The study estimated a possible 3% benefit in cruise SFC over conventional intercooling concepts, though the likely considerable impact of the installation's weight on the mission fuel burn was not determined. The concept's effect on mission fuel burn is therefore not obvious.

The greatest argument against an organic bottoming cycle on an aero engine has not been addressed appropriately in either study and concerns health and safety. Organic fluids are highly flammable and therefore pose a danger to aircraft and passengers in the case of a failure that would allow the working fluid to enter the main engine's gas path. Most organic fluids are also toxic and have an ozone depletion and global warming potential (GWP) when released at ground level [47], and their impact at altitude is not assessed yet. R245fa (1,1,1,3,3-Pentafluoropropane) for example has approximately 850 times the GWP of CO₂ [48]. Because of these concerns, the safety factors on the component designs would have to be substantial to reach certification, reducing their power density and worsening maintainability and thermo-mechanical integrity.

2.3.3 Ammonia-Water Mix (Kalina Cycle)

Alexander Kalina and his colleagues had been working on the Kalina cycle since the 1970's by the time they took out the patent on a novel 'Low Temperature Geothermal System' [49]. The Kalina cycle removes the limitation of a constant temperature phase change in the steam boiler and the cooler, by mixing two fluids with different boil-off points. At a given pressure, ammonia changes phase from liquid to gaseous at a lower temperature than water, which means that the mixture is essentially distilled until the boiling point of water is reached. During that distillation, the ammonia gas is already being superheated, so the overall temperature of the mixture rises continuously (see Figure 2-8a) from [50]). Hence, the temperature curves of the hot gases and the ammonia-water mixture in a heat transfer diagram are closer together (see Figure 2-8b)

than in a steam cycle. This reduces the heat exchanger's size and weight and improves the thermal efficiency of the cycle. The claims of 20 to 50 % higher efficiency than a comparable steam Rankine cycle of Mlcak et al. [50] are rather audacious and unsubstantiated through calculations or models. The estimate was furthermore based on less than state-of-the-art steam cycle designs as it was assumed that the thermal efficiency of an equivalent standalone steam cycle would not exceed 35%. However, Kalina Cycle plants in service today do support the claim of very high efficiencies, even if the benefit over complex USSPP's is lower than advertised. Ammonia-water mixtures are not excessively toxic, non-flammable and cheap fluids.

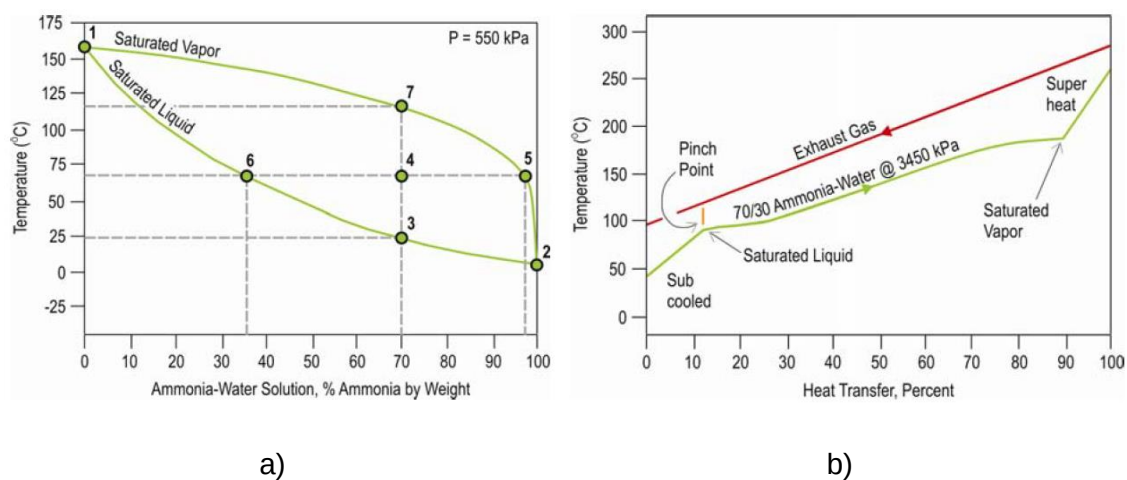


Figure 2-8: a) phase diagram of ammonia-water mixtures [50]; b) schematic of heat transfer in Kalina cycle [50]

There are three main concerns regarding the Kalina cycle for aero application. First, the simulation of a fluid mixture requires a very complex and detailed fluid property model (even more than that of S-CO_2), which will affect the calculation time of any cycle dramatically. Additionally, any models and the subsequently obtained results would be very difficult to validate, as the amount of information in the public domain is extremely limited. This limitation spans across all of the necessary fields of enquiry from fluid property models to turbomachinery and from heat exchanger design guidelines to reports on the operation of the cycle. The final obstacle that would need to be overcome is the cycle's high sensitivity to small changes in coolant temperatures. In order for the cycle to run at optimum efficiency, the ammonia concentration in the fluid that passes through the cooler needs to be actively regulated. This requires bypass and backflow ducts around the cooler and a complex control system, neither of which are ideal for aero application.

2.3.4 Air

The most obvious upside to using air in a bottoming cycle is the opportunity to have an open cycle configuration, which eliminates the need for a pre-cooler. However, it is easily compressible and the specific heat capacity does not vary very much in the typical peak operating temperature range of a bottoming cycle (300 - 800 K). The power density of the cycle will be helped by the lower part count compared to a closed cycle, but the operating pressures may be smaller, which could increase flow areas and hence system size. Though intended for off-shore application, the study of Bolland et al. [51] offers some insight into the challenges of an open air bottoming cycle (ABC) coupled to an aero-derivative LM2500PE. While successful in raising the efficiency from 36.1% for the LM2500PE to 46.6% in the combined cycle, the bottoming cycle had a “power density” of only 43 kW/t, approximately two orders of magnitude less than the main engine. As the cycle was designed for a stationary application, Bolland et al. prioritised cycle efficiency over weight and designed high effectiveness heat exchangers, which would be too big for aero applications.

Lundblad et al. [13] recently investigated a turbofan with an integrated open air bottoming cycle, which promised a potential efficiency gain of 5% – 8% relative to a projected EIS2025 turbofan. This translated to a reduction in specific fuel consumption (SFC) of 3.7% at cruise, but the study did not specify a weight increase or a fuel burn benefit. Lundblad et al. did however estimate the weight for a steam bottoming cycle option and found it cancelled out all of the efficiency benefits.

As mentioned before, the biggest limitation of an ABC is the peak temperature in the bottoming cycle. Yet, if the main heat exchanger were to be moved further upstream in the main engine’s flow path, the concept becomes much more competitive due to the higher peak temperatures and the higher hot side pressures that improve heat transfer. The elimination of the pre-cooler from the cycle will furthermore aid the power density and overall competitiveness of the cycle. This is why an ABC is being investigated by Andrew Rolt in his PhD research [27].

2.3.5 Supercritical CO₂

Supercritical carbon dioxide (S-CO₂) was originally investigated as a possible working fluid in power cycles by Feher [52] and Angelino [53], [54] in the 1960’s. Feher and Angelino conceptualised various different cycle configurations that could best exploit the real gas effects around the critical point and henceforth feature very high cycle

efficiencies. The development of these concepts never came to fruition, because the more conventional steam and gas cycles still held significant improvement potential and the computational and technological efforts to develop S-CO₂ components were too great. The concept was reinvigorated around the turn of the millennium, when studies such as that of Dostál [55] revisited the concept and indicated that the combination of fuel prices, investment and development cost of conventional technologies, and rising ecological awareness might make it a lucrative endeavour. Significant technological developments with respect to the design and operation of real gas turbomachinery and heat exchangers over the past two decades have further stimulated the interest in S-CO₂ cycles. Over the years, S-CO₂ power cycles were conceptualised for a number of different applications, as section 2.4.2 explains.

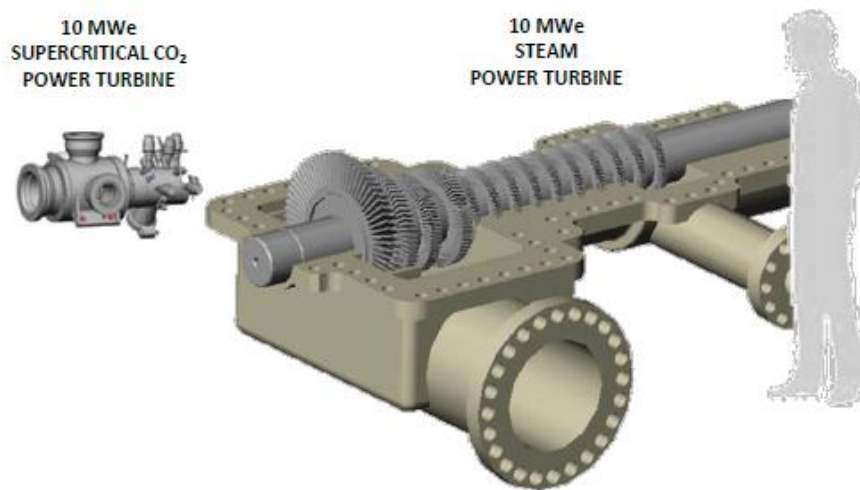


Figure 2-9: Comparing the scale of S-CO₂ power turbine with Steam turbine [56]

The main reason why S-CO₂ is such an attractive working fluid is the combination of possibly very high standalone cycle efficiencies (higher than steam or helium at medium to high temperatures) with a very high power density (see Figure 2-9). The critical point of S-CO₂ is at about 7.38 MPa and 304.13 K, values that can be realised in modern machinery. According to Cengel [57], “*The critical point is the point at which the saturated liquid and the saturated vapour states are identical.*”. Around the critical point, the fluid exhibits strong real gas behaviour, meaning that small changes in temperature and/or pressure result in substantial changes in fluid properties such as specific heat capacity, viscosity and thermal conductivity. Further beyond the critical point the fluid behaves similar to an ideal gas again. The S-CO₂ power cycle can utilise that variation in fluid behaviour in Brayton Cycles, whose compressor inlet conditions are just above the critical point while the turbine works in the near ideal gas region.

Alternatively, S-CO₂ could be used in a possibly even more efficient Rankine cycle, though those are more challenging to cool and cavitation in the pump may arise. Either way, the turbine generates a large power surplus relative to the power consumed by the compressor or pump, resulting in a very efficient cycle. The low power consumption of the compressor coupled with the high overall pressure levels in the system enable the S-CO₂ cycle to be much more compact and power dense than for example a steam Rankine cycle. S-CO₂ cycles feature relatively low pressure ratios, partly due to material limitations, but primarily due to most S-CO₂ cycles being highly recuperated [58]. Low pressure ratios lead to reduced numbers of turbomachinery stages. The final S-CO₂ bottoming cycles presented in this investigation feature a power density of 1.0 and 1.2 MW/t respectively, over 23 times the power density of the air bottoming cycle of Bolland et al. CO₂ is also not flammable and therefore no risk to the passenger's safety in case of leakage to the flow path of the main engine. An immediate environmental concern with CO₂ is the emission of the gas into the atmosphere in the case of a cycle failure. However, CO₂ is an unwanted by-product of many industrial processes and if the working fluid were sequestered from those, an accidental emission of the fluid to the environment would not trigger a net increase in CO₂ in the atmosphere.

2.3.6 Down selection

Table 2-1 gives an indication of the qualitative working fluid down-selection process. Each fluid is assessed on a scale from 1 (very poor) to 5 (very good) in terms of the expected power density of the bottoming cycle, its implications for health and safety, the amount and quality of available and relevant literature, and the cruise design point cycle efficiency that can be expected. The assessments of each category are then accumulated according to their suitability S :

$$S = Pwr_{dens} * 0.5 + Safety * 0.15 + Lit * 0.15 + eff * 0.2 \quad 2-2$$

Each of the four categories is assigned a weight in equation 2-2, based on assumed priorities during the preliminary design process of an aero engine. The power output per unit weight is by far the most important parameter of the bottoming cycle for an aero engine and therefore accounts for 50% of the final assessment. Passenger and aircraft safety is obviously important, but it is possible to design a system with larger factors of safety for riskier fluids as long as they are power dense enough, while a safe fluid with a low power density is plain unfeasible. The literature available is an important parameter, as it will directly affect the quality of this investigation. However, it is not crucial this early

in the process and there is enough material on most fluids from which to derive models. The standalone efficiency of the bottoming cycle is not the primary objective for a combined cycle engine, but it directly affects the size of the pre-cooler and hence the power density. Hence, it is given slightly more weight than safety and literature. Based on these assumptions, S-CO₂ is currently assumed to perform best overall.

Table 2-1: Working fluid options and their feasibility for Bottoming Cycles for aero application

<i>Fluid</i>	Power density	Safety	Literature available	Eff. in temp. range	Suitability	Comments
<i>S-CO₂</i>	4	5	3	4	4	+ High power density - Complex controls
<i>Air</i>	3	5	5	2	3.4	- Low efficiency + Open Cycle
<i>H₂O</i>	1	5	5	3	2.6	- Very heavy
<i>Organic</i>	3	1	3	5	3.1	- Flammable - Thermally unstable
<i>NH₃</i>	-	3	2	1	4	2.75 - Difficult to model
<i>H₂O mix</i>						- Complex controls

2.4 Supercritical Carbon Dioxide

This section provides an overview of the state of the art in S-CO₂ fluid property and power cycle modelling. Additionally, the specific requirements for an application in a modern turbofan engine are listed and the resulting boundary conditions for the bottoming cycle derived.

2.4.1 Fluid Properties

The industrial and academic interest in supercritical CO₂ power cycles largely results from the fluid properties of carbon dioxide when in a state close to its critical point. While the fluid properties of CO₂ can be determined assuming ideal gas behaviour to a reasonable degree of accuracy for a wide array of operating conditions, the same is not true for the range of operating conditions of interest in these power cycles. Supercritical CO₂ exhibits strong real gas behaviour near its critical point, which entails radical

changes in properties such as density, viscosity or specific heat capacities for small changes in operating pressure and temperature (see Figure 2-10 and Figure 2-11). These changes directly affect heat transfer coefficients, the enthalpy rise in the turbomachinery and more. The operating conditions therefore have a significant effect on the cycle performance, which does not apply for ideal gases.

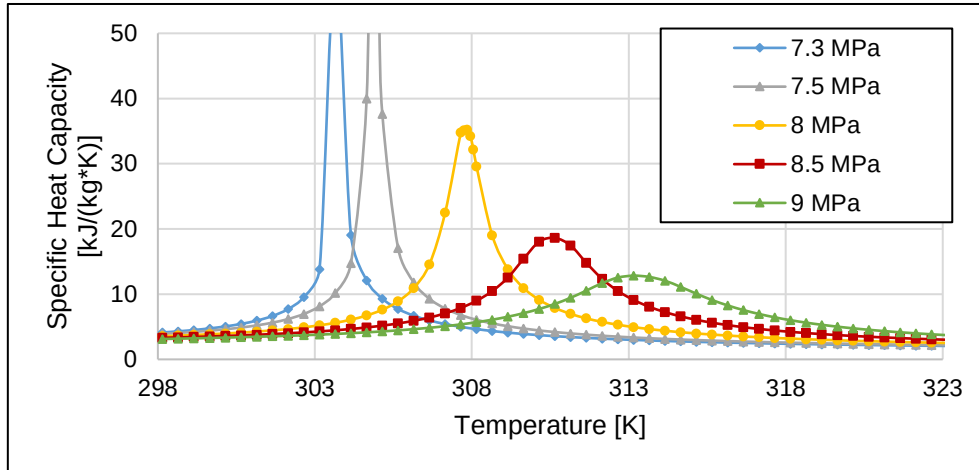


Figure 2-10: Specific heat capacity of S-CO₂ depending on pressure and temperature

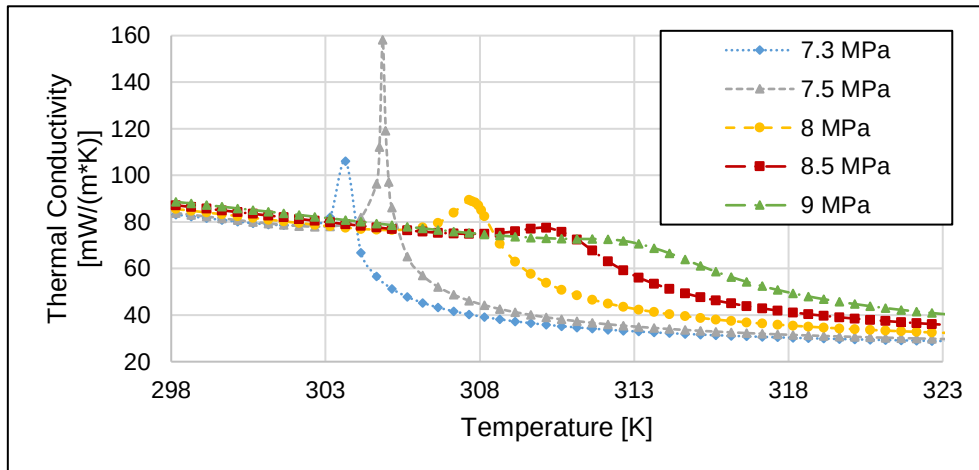


Figure 2-11: Thermal conductivity of S-CO₂ depending on pressure and temperature

Such fluid behaviour near the critical point is not unique to CO₂, but the reason why S-CO₂ is of special interest for power generation purposes is its favourable combination of temperature and pressure at the critical point. For many fluids, the critical point lies at very high or low temperatures and pressures, which disqualify them for application in a combined cycle engine arrangement. However, operating conditions around 304.13 K

and 7.38 MPa have been realised in steam cycles before and no longer pose a prohibitive design challenge.

The compressor work spent to achieve a given pressure ratio is significantly reduced due to the real gas behaviour of the CO₂ near the critical point. The fluid is barely compressible near the critical point, meaning the density change for a given pressure ratio is relatively small. Figure 2-12 illustrates how these changes affect the compressor work directly. The graph shows how the specific inlet and outlet enthalpy, and the specific compressor power vary for a compressor with a fixed inlet temperature of 305 K, a pressure ratio of 3 and an efficiency of 90%. Once the fluid has passed into the supercritical region (around 7.5 MPa at that temperature), the inlet and outlet enthalpy drop significantly. The compressor work is reduced, because the outlet enthalpy is reduced more. The turbine however works in near ideal gas conditions, where the turbine work is primarily dependent on the pressure ratio and not the operating conditions. Hence, the power output is nearly constant for all relevant operating conditions as long as the pressure ratio remains the same. This is the reason why S-CO₂ power cycles can achieve a higher thermal efficiency than an ideal Brayton cycle.

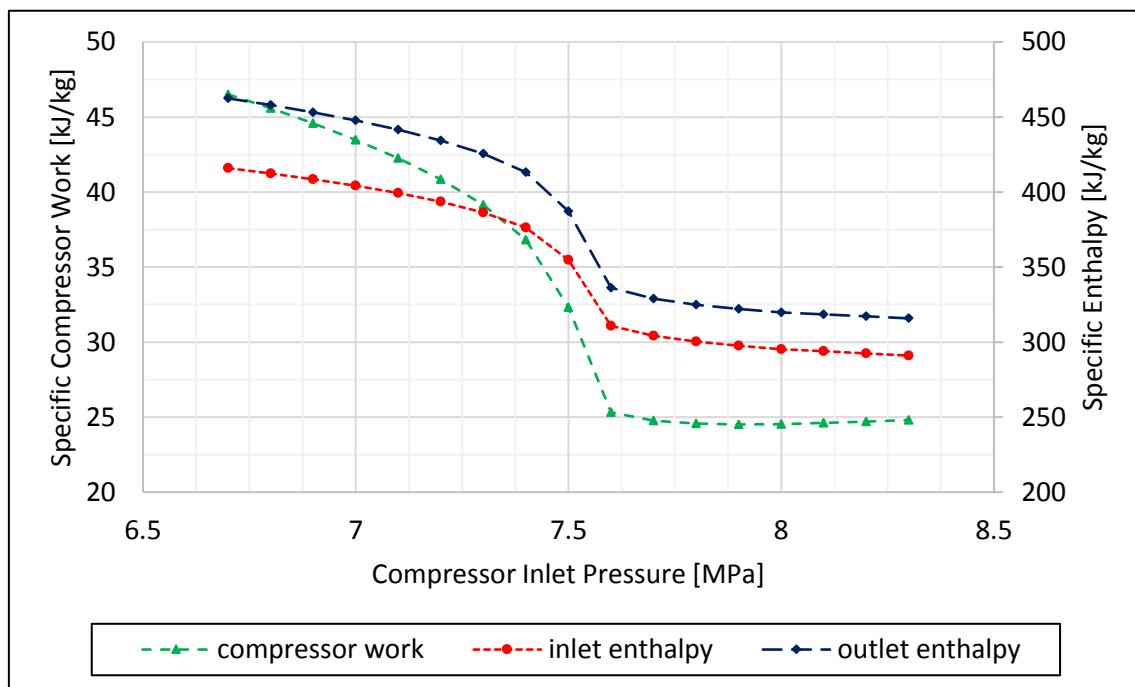


Figure 2-12: Effect of inlet pressure on compressor work

The rapid change in fluid properties has competing beneficial and adverse effects on S-CO₂ power cycles. On the one hand, the increase in thermal conductivity near the critical

point means, that the heat transfer coefficients in a given channel rise and therefore the heat transfer gets easier. On the other hand, estimation of a heat exchanger's effectiveness operating near the critical point is complicated and can lead to gross overestimations if not conducted carefully. A pinch point is the location inside a heat exchanger with the lowest temperature difference between the two fluids. In ideal gas heat exchangers, the pinch point is usually at either end of the heat exchanger, depending on the mass flow rate and heat capacity rates of the flows. The rapid change in specific heat capacity in the S-CO₂ flow may mean however, that that pinch point exists inside either the pre-cooler or the recuperator, limiting the maximum heat transfer possible. In order to capture this possibly serious limitation on the cycle's performance, the recuperator and the pre-cooler need to be discretised into multiple shorter sections whose performance is calculated in sequence. The details of this approach are explained in chapter 3 of this work.

Overall, the accurate portrayal of the fluid properties across the whole operating range, particularly near the critical point, is crucial when estimating the performance of an S-CO₂ power cycle. A number of different equations of state (EOS) have been proposed to capture these effects. Two of the most commonly implemented are the EOS of Peng-Robinson [59] and that of Span & Wagner [60]. The Span & Wagner EOS (SWEOS) is adopted by most researchers in the industry and Belmonte illustrated its superior ability to predict S-CO₂ behaviour near the critical point in his MSc thesis [61]. The SWEOS is therefore utilised in the current study as explained in section 3.3.

2.4.2 State of the Art for other Applications

Feher [52] and Angelino [53] presented a host of different cycle configurations that are applicable to S-CO₂. Their research determined that the more complex configurations like the recompression or pre-compression cycles would be most efficient and can potentially outperform Helium or Steam cycles at medium to high turbine inlet temperatures. Further research over the past decades identified further possible configurations, each uniquely qualified for certain applications and boundary conditions. The recuperated (RB) and the recompression Brayton cycles (RCBC) are particularly popular for a range of applications and exhibit high cycle efficiencies at moderate pressure ratios as indicated in Figure 2-13. The figure shows that consistently higher cycle efficiencies are attainable in the S-CO₂ RCBC relative to the S-CO₂ RB. The underlying data of White et al. [62] is based on constant component efficiencies for all cycles and a common turbine entry temperature of 700 °C.

The majority of research on S-CO₂ power cycles is motivated by the development of next generation nuclear power plants [55] or concentrated solar power plants (CSP) [63], [64]. Both concepts benefit from the smaller footprint of the S-CO₂ power cycles, but it is seldom a first design priority. The total weight of the cycle components is rarely a concern during concept or preliminary design studies, as these cycles are land based. The primary design objective of these studies is the maximisation of the cycle efficiency and the development of materials and manufacturing techniques that enable a reduction of the cost of electricity in order to compete with established power sources. Compound cycles, as Dostál dubbed the more complex configurations, are therefore frequently applied in these cases. It is important to note that many of the compound cycles entail that over 50% of the heat added to the compressor exit flow is transferred from the S-CO₂ turbine's exhaust flow via a recuperator. These recuperators are frequently anticipated to be printed circuit heat exchangers, which are very compact, but also heavy. A similarly high degree of recuperation is therefore unlikely to be ideal for aero application.

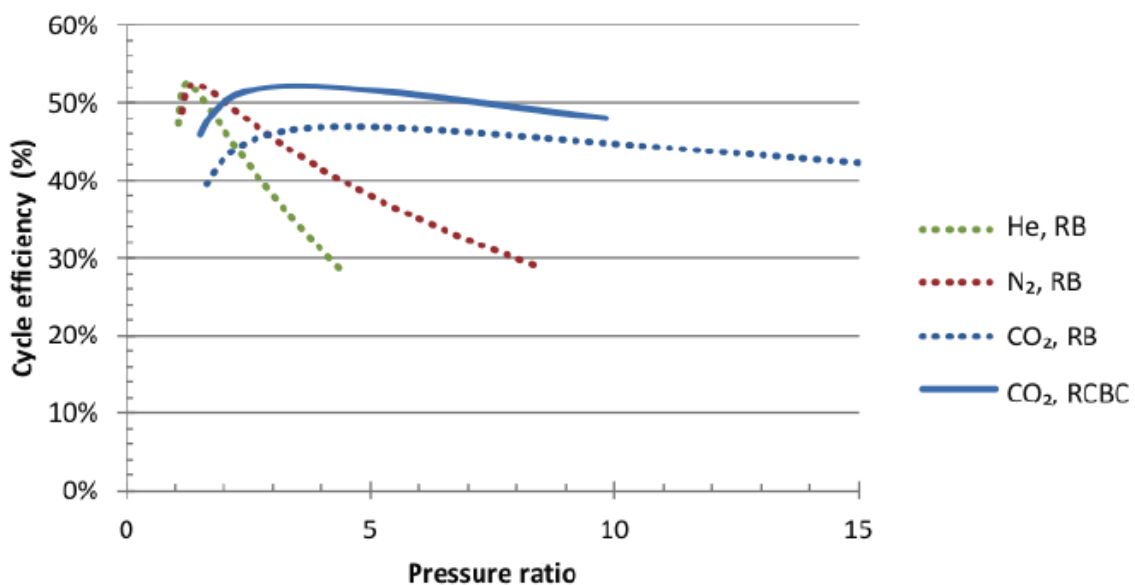


Figure 2-13: Comparison of S-CO₂ power cycles with HE and N₂ cycles, from [62]

Exhaust gas recovery units have also been developed, constructed and tested, with Echogen Power Systems at the forefront of those efforts. Little information is available on the early designs by Echogen Power Systems, though they are likely based on a recuperated cycle configuration [56]. Data on the commercially available EPS100 [64] units producing approximately 7-8 MWe is equally scarce, but Brun et al. [65] deduced that the dual-shaft design featured turbomachinery components with isentropic

efficiencies between 80 and 85%. The packaging volume and weight of the system is a much more prominent design objective for exhaust gas recovery units, which explains the simpler cycle layout in these systems relative to nuclear or CSP applications.

S-CO₂ power cycles have furthermore been investigated as alternative propulsion systems for marine vessels [66], increasing the importance of power to weight ratio and unit size. Brun et al. present an overview of the different potential applications S-CO₂ power cycles that have been considered to date together with their respective design priorities, scale and likely range of operating conditions. The pressure and temperature levels in the resulting Table 2-2 describe the turbine inlet conditions of the S-CO₂ cycles. Brun et al. furthermore provide insight to the cycle developments for most applications other than aero applications.

Table 2-2: Potential applications for S-CO₂ power cycles from Brun et al. [65]

Application	Cycle type	Motivation	Size (MWe)	Temperature (°C)	Pressure (MPa)
Nuclear	Indirect sCO ₂	Efficiency, size, water reduction	10–300	350–700	20–35
Fossil fuel (PC, CFB, ...)	Indirect sCO ₂	Efficiency, water reduction	300–600	550–900	15–35
Concentrating solar power	Indirect sCO ₂	Efficiency, size, water reduction	10–100	500–1000	35
Shipboard propulsion	Indirect sCO ₂	Efficiency, size	<10–10	200–300	15–25
Shipboard house power	Indirect sCO ₂	Efficiency, size	<1–10	230–650	15–35
Waste heat recovery	Indirect sCO ₂	Efficiency, size, simple cycles	1–10	<230–650	15–35
Geothermal	Indirect sCO ₂	Efficiency	1–50	100–300	15
Fossil fuel (syngas, natural gas)	Direct sCO ₂	Efficiency, water reduction, CO ₂ cap	300–600	1100–1500	35

In 2017, Crespi et al. [67] canvassed over 150 different references on S-CO₂ power cycles to investigate the differences, monitor the respectively achievable cycle efficiencies and propose a reasonable categorisation, resulting in a great source of further information. The different operating conditions and cycle designs lead to an enormous range of thermal efficiency estimates from 6.4 to 63 %. For aero applications, it can be expected that the thermal efficiency of the bottoming cycle has to be higher than 15 % improve whole engine performance.

First S-CO₂ test rigs have been designed and in use for a few years now with some of the best documented facilities located at Sandia National Labs [68], Bechtel Marine Propulsion Corporation (BMPC) [69], [70], Southwest Research Institute (SWRI) [63], [71] and the Korean Advanced Institute of Science and Technology (KAIST) [72]. These

test loops range from laboratory scale 10 kWe units to 1MWe units in case of the SWRI test loop. The Sandia test loop is one of the earliest and best documented test loops, including component maps for the turbomachinery and some cycle data, both of which have been used to inform the models of this work. The test loop at SWRI is part of the collaborative effort of multiple research institutions in the US under the umbrella of the SunShot programme [73] and was originally intended for a 10 MWe scale. A redesign of the test loop has reduced the scale to 1MWe. The data available from that test rig is still very limited. BMPC identified some significant discrepancies between the predicted and measured performance of their test loop, highlighting a need to better understand the effects of gas foil bearings and off-nominal performance of S-CO₂ turbomachinery. The investigation of the part-load performance [69] of the test rig was not among the initial objectives of the test campaign at BMPC, but the gathered data is crucial nonetheless. Cycles for nuclear or CSP power plants will spend very little time at such part-power conditions, but aero engines experience a large range of operating conditions. Further investigation of the off-design performance of S-CO₂ cycles, especially through tests, is vital for the continued development of the concept of a combined cycle aero engine, but this is outside the scope of the present work.

BMPC also used their test configuration to investigate transient behaviour of S-CO₂ bottoming cycles and validate their turbomachinery models to that regard [70]. The investigation of transient performance of S-CO₂ cycles as a whole and of their heat exchangers in particular has not yet been comprehensively studied. Moiseyev [74] showed that the simulation software at Argonne National Laboratory is mature enough to depict the transient performance data provided to them by BMPC with reasonable accuracy. However, the estimation of transient performance of whole cycles is particularly difficult due to the volume packaging effect [14] of large components such as the heat exchangers, and further development and validation of methods is required from future studies.

2.4.3 Aero Application: Design Implications and Knowledge Gap

None of the aforementioned applications faces such strict limitations as aero application when it comes to size, weight and off-design performance. The components of the S-CO₂ bottoming cycle, the auxiliary systems necessary to connect them to main engine and the S-CO₂ itself accumulate to a substantial weight increase of the engine. The increased specific power of the engine will help alleviate that affect somewhat through a size reduction of the core and maybe even the fan, but overall the combined cycle engine

will still be significantly heavier than the TF2050 reference engine. The physical dimensions of the components, especially the heat exchangers, will affect the efficiency of the cycle, but also the pressure losses inside the engine and/or the drag of the engine. If the engine diameter is increased in order to minimise pressure losses in the bypass stream for example, the nacelle drag will increase, as will the weight of the load bearing structures of the aircraft, the landing gear and more. Any increase in weight or drag will require a higher thrust rating of the engine, leading to higher fuel burn values and offsetting some of the gained SFC improvements. The engine drag is also accounted for in the non-linear aircraft trade factors. The SFC reduction achieved by the combined cycle engine, the prime objective of many concept studies, is therefore only one of three major contributors to the mission fuel burn of the aircraft.

The only other publication on S-CO₂ bottoming cycles for aero application identified so far is the work of De Servi et al. from Delft University [11], which approached the subject in two steps. An initial (and optimistic) overall assessment of the technologies' potential to improve a modern turbofan engine's performance promised a maximum of a 20% lower SFC than the baseline turbofan and a 6% SFC improvement over an intercooled recuperated engine. The bottoming cycle in question was a recuperated arrangement. When the authors applied the concept on a component level to an engine model representing a GE90-94b, the SFC improvement over the reference was reduced to 2.8%. The reduction occurred due to the incorporation of more complex phenomena such as the heat exchanger induced pressure losses, the thrust reduction due to heat removal from the core exhaust and the physical dimensions of the components. Finally, the authors stated that the cycle weight (estimated to over 3 t) would outweigh any performance benefits on a mission basis. However, De Servi et al. did not investigate the substantial effect the compressor inlet conditions have on the cycle design and performance. The compressor inlet conditions were fixed to 283 K and 7.4 MPa, making this a Rankine rather than a supercritical Brayton cycle. The low compressor inlet temperature is the biggest driver behind the extreme weight of the cooler (1.6 t). It is doubtful, that the external pressure loss across the cooler could really be limited to 6% given its size. The authors also acknowledge that the proposed heat exchanger geometry is just a first attempt and a sub-optimal design. A mission fuel burn analysis accounting for the aforementioned effects of engine drag and weight on the engine scale and aircraft performance was not included. This study clearly highlights the need for preliminary design tools that allow for trade-off studies of physical component parameters, such as tube geometry and count.

The size of the cooler estimated by De Servi et al. illustrates one of the main reasons why the bottoming cycles in the present work are designed to be supercritical Brayton cycles rather than Rankine Cycles. Another important driver to that end is the cavitation risk for the CO₂ compressor/pump [75] in a Rankine cycle, given that the operating conditions of the component will probably vary greatly throughout the flight envelope. The bottoming cycles investigated herein are also either simple or recuperated cycles rather than complex compound cycles. Earlier studies at Cranfield University [66], [76] considered the impact of more efficient compound cycles and dismissed their feasibility for aero application. The standalone efficiency is only a secondary priority for the bottoming cycle via its influence on the pre-cooler dimensions. The useful power output is the focus, as it supplements the topping cycle's performance and increases the combined cycle efficiency. The addition of inter-turbine reheat to the main engine of the combined cycle will aid the useful power production and has not been analysed in the public domain to date.

In summary, S-CO₂ cycles have been researched and even tested for a range of different applications, however the unique requirements of aero applications have been largely overlooked so far. The current work addresses that need and gives an assessment of the performance potential of simple S-CO₂ Brayton cycles for aero application. This study accounts for the effects of:

- The compressor inlet conditions on cycle performance and heat exchanger size, particularly regarding the pre-cooler
- The pressure losses and heat transfer across the main heat exchanger and the pre-cooler on the thrust produced by the engine using component models
- The radial dimensions of the heat exchangers on the installation, bypass pressure losses and the nacelle drag
- The diffuser designs of the pre-cooler and the main heat exchanger on whole engine performance
- The bottoming cycle's performance on whole engine and aircraft performance

The coupling of preliminary design and performance calculations in this work is one of the main differentiators from the investigation of De Servi et al. [11]. The application of the technology to a 'fully mature' year 2050 reference engine to adequately identify the technology's potential in a realistic timeframe is another. This study is therefore primarily focused on establishing the concept's viability based on design point data, even though some off-design estimates are used to calculate potential mission fuel burn reductions.

The investigation of the off-design behaviour of these cycles and the variety of options to control them should be explored in more detail in the future, as only a simplified off-design performance estimation method could be employed herein (see section 3.7.3). The operation of the main heat exchanger and the pre-cooler ahead of one, or between two, choked nozzles poses a significant modelling challenge and an opportunity to control the bottoming cycle's performance at the same time. The study of off-design and transient performance of S-CO₂ cycles is presently a popular research area, with one of the most comprehensive studies being presented by Sanchez et al. [77]. However, to the best of the author's knowledge, no results have been presented on combined cycle engine control for nuclear, CSP or waste heat recovery scenarios, not to mention aero application to date.

A whole range of different methods could be used to control the combined cycle's performance. Dostál [55] and Brighenti [66] discussed the possibility of a mass flow control scheme, where the S-CO₂ mass flow would be modulated at off-design to control the non-dimensional mass flow to the turbomachinery components. While theoretically possible, this concept would entail the addition of a central S-CO₂ fluid tank and additional ducts and valves, not to mention a sophisticated control system. The small size of the S-CO₂ turbomachinery means that their moments of inertia would be relatively small, promising a rapid response to mass flow control. Furthermore, a variable area nozzle in the pre-cooler duct similar to that envisioned for intercooled engines may be used to control the cooling of the S-CO₂. Variable nozzle guide vanes in the main engine LPT on the other hand could be used to modulate the core exhaust gas temperature to a certain degree and therefore the bottoming cycle's peak temperature as proposed for recuperated engines [36].

Alternatively, the bottoming cycle could be largely left to operate on its own accord. The higher fan outlet temperatures at take-off and climb conditions would lead to a further departure of the compressor inlet conditions from the critical point, which in turn would lead to a reduction in thermal efficiency of the bottoming cycle. The higher LPT EGT at these operating points will offset a considerable portion of those penalties however. Leaving the bottoming cycle to self-regulate would require close monitoring and fast emergency control systems to avoid surge and/or over-speed scenarios.

Finally, the bottoming cycle could be disengaged from the main engine completely at hot conditions, as the cycle will be designed primarily to improve cruise performance and large bypass ratio turbofans seldom have trouble meeting thrust requirements at take-

off and early climb anyway. Again, the flow conditions of the bottoming cycle would have to be monitored closely at those conditions. Whether or not the shaft speed of the bottoming cycle would have to be synchronised to the LPT shaft speed before engaging it would depend on the transmission system.

2.5 Heat Exchangers

The heat exchangers are by far the largest new components of the combined cycle engine. They are responsible for the heat transfer from the LPT exhaust gases to the supercritical CO₂ in the main heat exchanger (MHX) and from the S-CO₂ to part of the bypass flow in the pre-cooler. In case of a recuperated cycle, a third heat exchanger is responsible for the heat transfer from the S-CO₂ turbine exhaust to the compressor exit flow of the S-CO₂ bottoming cycle. One of the main goals of the investigation is to keep the cycle as compact and as light as possible. The compactness of a heat exchanger is commonly characterised by the ratio of total heat transfer surface area to the volume of the heat exchanger. A design is usually classified as a compact heat exchanger, if that ratio is greater than 1000 m²/m³ [78], [65]. Previously, only plate-fin type heat exchangers (PFHEs) and printed circuit heat exchangers (PCHEs) were considered in that class, because of the extremely small hydraulic diameters that can be realised with them. However, both concepts are relatively heavy, lessening their feasibility for aero application. By 2050, the production and successful use of tube heat exchangers with extremely small diameter tubes will likely also be possible. These microtube heat exchangers can also reach over 1000 m²/m³, as the heat exchanger designs of Thar Energy are showing [79], and hence qualify as compact heat exchangers. However, in this study only PCHEs and PFHEs will be referred to as compact heat exchangers. Developing these fine tube designs and manufacturing capabilities would make tube heat exchangers a prime concept for this application, because they are inherently lighter than PFHEs or PCHEs. Hence, a thorough screening of heat exchanger concepts that were previously deemed applicable to an aero engine is necessary, including the required developments to make currently unfeasible designs feasible.

The performance of tube heat exchangers may be further improved through the addition of secondary surfaces as explained by Kays and London [78], Schulenberg [80] or Kraus et al. [81]. Secondary surfaces, often referred to as fins, are commonly applied in oil-to-gas heat exchangers for example to increase the heat transfer surface area on the gaseous side. Usually, gases feature noticeably lower heat transfer coefficients than liquids, creating an imbalance in the overall conductance of the heat exchanger. By

increasing the heat transfer surface area on only one side of the heat exchanger, that imbalance in heat transfer coefficients may be offset. The S-CO₂-side heat transfer coefficients in the MHX and the pre-cooler of the combined cycle concepts are between 15 and 20 times larger than those on the exhaust gas/air side. These ratios may be overcome, as fins have been used to offset ratios of 30:1 and more [82]. However, the actual ratio that can be offset depends on the specific design and application of the heat exchanger. The secondary surfaces not only increase heat transfer surface areas, but also the weight and drag of the heat exchanger, both of which are crucial for aero applications. Detailed preliminary design and performance models for secondary surfaces were too complex for the current models and were therefore not included.

Further, more unconventional heat exchanger design concepts would be airfoil heat exchangers [83] or surface heat exchangers on the engine nacelle [44] or aircraft fuselage. Airfoil heat exchangers could be used as novel intercooler arrangements, making them a theoretical option as full or partial replacements of the air-pre-cooler. In fact, airfoil heat exchangers were originally conceptualised to work with carbon dioxide as one of the possible cooling media. However, the flow around the stator vanes of the IPC or the first stages of the HPC is too hot and the heat transfer surface areas are too small to warrant the additional complexity and pressure losses for this investigation.

Using the nacelle of the engine as a primary surface heat exchanger [44] is a further possible design concept for the pre-cooler. The uncompressed air flowing around the nacelle would lower the cold side inlet temperature of the pre-cooler, possibly leading to higher heat transfer rates. However, the heat transfer figures stated by Petit et al. are only initial estimates and detailed proof-of-concept studies would be required to show the feasibility of the arrangement. In their study some necessarily simple assumptions such as the constant wall temperatures of the nacelle will have affected the results significantly. Furthermore, the surface area of the nacelle is very limited, limiting the amount of heat that can be rejected. The problem is aggravated by increasingly short nacelles with short inlets for mixed nozzle engines. The space limitation may be overcome, if the wetted surface area of the wing and the fuselage were used as well. Using the aircraft fuselage to reject the excess heat from the bottoming cycle to the boundary layer of the fuselage might have the additional benefit of lowering the drag of the aircraft as investigated by NASA [84]. However, this arrangement would entail a lot more integration and heat management issues, which most likely outweigh any performance benefit that may be gained from the engine. Additionally, the outside of the

nacelle is exposed to the environment. Foreign object damage (FOD) or dirt on the surfaces could easily render the heat exchangers less effective or completely unusable. Due to all this, surface heat exchangers were not considered further in this investigation, though detailed studies may reveal beneficial designs yet.

Previous attempts to further improve the power to weight ratio of heat exchangers for aero application included the investigation of heat pipes taking the place of an intercooler [85] or a recuperator [86]. Heat pipes are relatively easy to manufacture and would be safer to employ than tube heat exchangers, because any failures of the tubes would not relay leakage of S-CO₂ from the bottoming cycle. Additionally, Camilleri et al. claimed that the heat pipe heat exchangers would be approximately 40% lighter and feature significantly lower pressure losses for either fluid than their design reference. However, the tube heat exchanger design of Xu et al. [87] has since been significantly improved upon by Zhao et al. [88]. Additionally, Camilleri et al. do not seem to have taken the weight of the installation of the heat pipes into account. The claimed benefits may therefore be partially or completely offset. Employing heat pipes in the combined cycle engine would entail a number of difficulties. Firstly, the high density of the S-CO₂ means that the flow area is very small even at extremely low Mach numbers, which limits the amount of heat transfer surface. Having a larger number of rows behind each other may alleviate that problem, but it also increases S-CO₂ pressure losses. Secondly, the pre-cooler uses bypass air as a coolant and sits around the high pressure components of the core engine to minimise interaction with the outer bypass flow. Hence, the S-CO₂ would still need to be directed to the pre-cooler. This not only negates some of the theoretical benefits of having fewer ducts and pressure losses when working with heat pipes, it also means, that the S-CO₂ piping would once again have to cross the core flow. Finally, the choice of transfer fluid inside the heat pipes and its thermal behaviour have a significant effect on the performance of the heat pipes. Incorporating these affects would further complicate the models of this investigation and reduce stability. Due to all these concerns it was decided not to model heat pipe heat exchangers herein, though the concept may be revisited in future studies.

Overall, it appears that only compact heat exchangers and tube heat exchangers have been researched in sufficient detail to warrant their application in the current investigation. Other concepts may help improve upon the findings of this study, but they also introduce further modelling uncertainties and instabilities that were deemed prohibitive for this initial feasibility study. Hence, sections 2.5.1 and 2.5.2 present some

exemplary developments of compact and tube heat exchangers. Section 2.5.3 discusses additional technologies that may be necessary for the heat exchangers to exploit their full potential. Finally, possible material options and their implications are briefly discussed in section 2.5.4.

2.5.1 Compact Heat Exchangers

Compact heat exchangers have been in use for years for different applications, often when the working fluids are gaseous and therefore feature inherently poor heat transfer capabilities. Compact heat exchanger designs may be split into three categories: plates, fins and matrices.



Figure 2-14: Plate Fin Heat Exchanger core section (from Brayton Energy in [89])

The plate fin heat exchanger (PFHE) design (see Figure 2-14) is the logical culmination of adding secondary surfaces to plate type heat exchangers to improve their performance. Brayton Energy [89] offers PFHEs for different applications and manufactured from a range of materials, including high temperature steels and titanium.

Through the preliminary design studies of his heat exchangers, Dostál [55] concluded that PFHEs may be designed to work at the necessary operating pressures and temperatures, but their dimensions and weight would be excessive once the maximum operating pressure is higher than approximately 20 MPa. Especially the parting plates need to be relatively thick for those cases. This conclusion is corroborated by the data presented by Heatric about compact heat exchangers in operation to date. Figure 2-15 illustrates the approximate design ranges PFHEs and PCHEs have been successfully employed at and there appears to be a much stronger dependency of the PFHE's viability on pressure than there is for PCHEs. Naturally, the pressure differential between the two fluids would be smaller in an S-CO₂ to S-CO₂ recuperator than in an S-CO₂ to water cooler. Nonetheless, it is assumed that PFHE's may be the less compact and heavier design option than PCHEs.

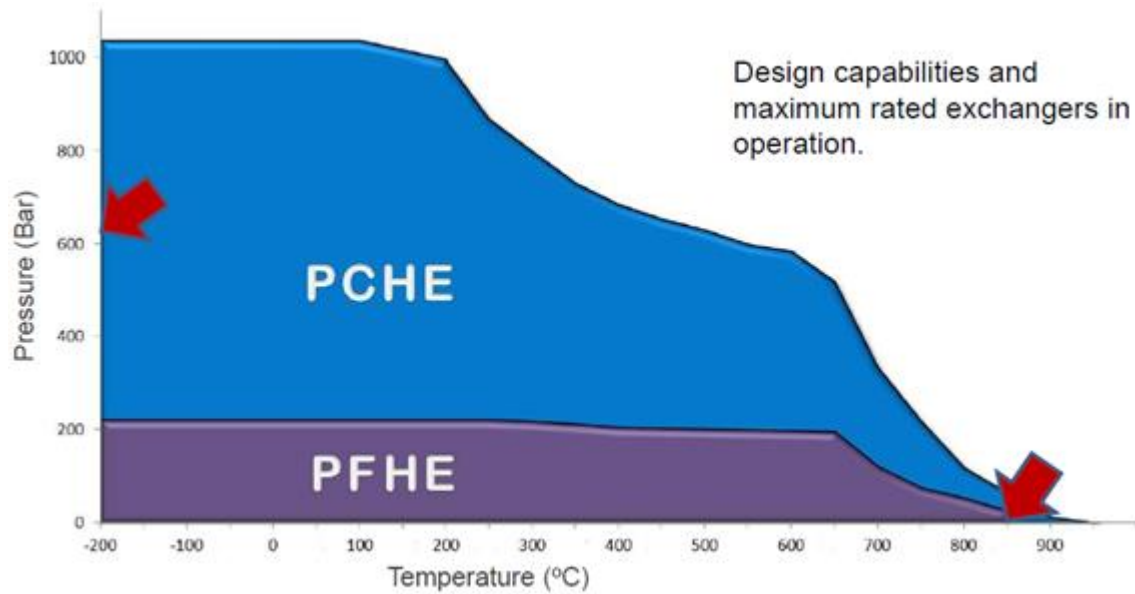


Figure 2-15: Design ranges of operating PFHEs and PCHEs (from Heatric in [89])

PCHEs are created from many separate flow plates in which the flow channels are chemically etched. The flow channels can take many forms, with straight and wavy continuous channels being the most common. After the plates are stacked on top of each other, they are diffusion bonded to form a monolithic heat exchanger matrix. This modular manufacturing procedure is the reason for the design flexibility available for PCHEs. The current maximum dimensions of the heat exchanger modules according to Heatric [90] are 600 mm in width, 600 mm in height and 1500 mm in length. The widest available photo film necessary for the etching process is the reason why the plates are limited. However, multiple modules can be coupled in sequence or in parallel to create large components with an effectiveness of up to 99%. The channel width can be customised to the specific application, with 2 mm being a common value and 0.2 mm being the lowest width currently offered by Heatric [90]. The PCHEs are designed so the pressure drop is of the order of around 1 % unless otherwise specified by the customer. Heatric's PCHEs typically feature heat transfer surface to volume ratios between 650 and 1300 m²/m³, depending on the operating conditions. Finally, PCHEs appear to be relatively fail-safe. Heatric has reported no header failures and subsequent leaks of the working fluid to the environment to date. Also, any potential leaks inside the heat exchanger matrix can be addressed easily by removing the headers, identifying the leaking channel and spot welding it shut. This technique has been successfully applied several times already.

PCHEs are only considered as design options for the S-CO₂ recuperator, as they are too heavy to be considered an option for the main heat exchanger or the pre-cooler. Depending on the proximity of the compressor inlet conditions to the critical point, the S-CO₂ may still exhibit strong real gas behaviour in the recuperator. To a certain degree, the corresponding fluid property variations would aid the performance of the recuperator through increased heat transfer coefficients as demonstrated by Dostál and mentioned before. Unfortunately, the turbine exit flow will not show such strong fluid property variations and the pinch point of the heat exchanger may lie somewhere along the flow path through the recuperator rather than at one end. In order to realistically assess the performance of S-CO₂ recuperators, it is therefore necessary to discretise them into multiple adjacent sections, whose performance is evaluated iteratively using established methods like the ϵ -NTU method. This is a commonly applied approach in the industry and is also applied herein and explained further on. Depending on the material the PCHE is manufactured from, longitudinal conduction through the heat exchanger matrix may theoretically affect the performance of the component. As part of his MSc research, Zhu [91] investigated the influence longitudinal conduction would have on a cryogenic PCHE pre-cooler manufactured from aluminium. His case study concluded that the influence was negligible despite the high total temperature difference between the two inflow streams and the high thermal conductivity of aluminium. Ti-6Al-4V has a relatively low thermal conductivity, especially when compared to aluminium, which means longitudinal conduction should only have a marginal effect on the performance of the recuperator as well.

When applied to an aero engine, it appears likely that a PCHE recuperator does not reach the necessary power to weight ratio to warrant its incorporation, despite the compactness of the design. A simple cycle arrangement with only a compressor, turbine, main heat exchanger and pre-cooler appears to be more attractive for a combined cycle aero engine.

2.5.2 Tube Heat Exchangers

The relatively large heat exchangers of most aero engine concept studies (intercoolers, recuperators) are often tube heat exchangers, due to their low weight and drag characteristics. Additionally, the tube heat exchangers can also reach high surface area to volume ratios, if the tube diameter is sufficiently small. Typically, the respective manufacturing technique is the limiting factor for the minimal tube diameter. Over the years, a range of different tube heat exchanger geometries have been proposed for

different aero engine components. Some of the most promising designs with regard to the present study are discussed in the following sub-sections.

2.5.2.1 MTU Lancette Heat Exchanger

The Lancette Heat exchanger was originally developed as a gas turbine recuperator for a tank. MTU has continuously developed the design of that recuperator for aero application for about two decades, both internally and through European research projects such as CLEAN, NEWAC and LEMCOTEC. Individual modules of the Lancette Heat exchanger design have been produced and tested in the past. The recuperator of MTU's intercooled recuperated aero engine concept consists of eight heat exchanger modules placed around the nozzle cone of the core exhaust as shown in section 2.2.2, Figure 2-6. Each individual module consists of hundreds of small near-elliptical tubes as illustrated in Figure 2-16. The relatively cold compressor exit flow enters and exits the individual tubes via cylindrical headers. An internal support web ensures the tubes' mechanical integrity. These tubes were folded from sheet metal and their ends subsequently brazed, which is why the tubes deviate from a pure elliptical cross-section to something closer to a double circular arc design. The manufacturing technique and its application to a more intricate tube design with a higher web count was patented by MTU in 1993 [92].

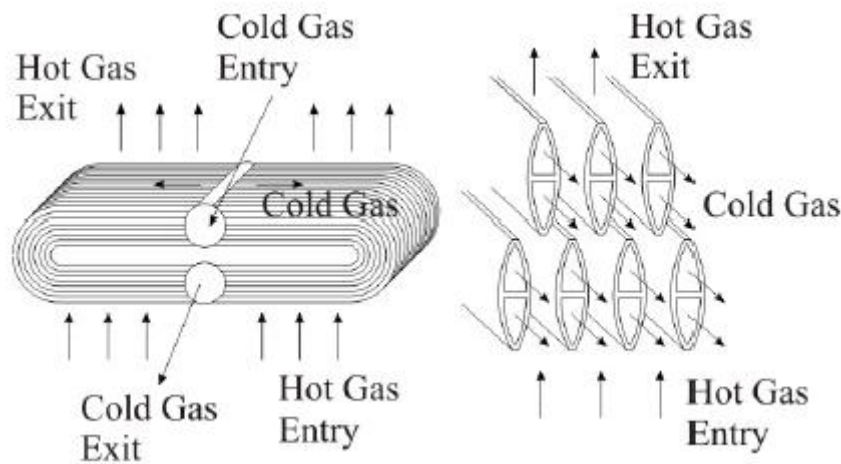


Figure 2-16: Schematic of Lancette Heat Exchanger design from [93]

The recuperator modules were designed to withstand the considerable thermal cycling inherent to aero application, where they are repeatedly exposed to fluid temperatures above 900 K. It was therefore crucial to ensure unrestricted thermal expansion, good resistance to thermal fatigue and to corrosion. The Lancette heat exchanger has been

shown to reach a heat exchanger effectiveness of about 75 % at a pressure loss below 8 %. The installation of the modules around the nozzle cone is relatively complex and requires a change in flow direction from the LPT exhaust stream. However, the modules' inclined position relative to the engine axis also reduces the necessary diffuser length and the radial dimensions of the heat exchanger, resulting in a more compact engine.

Concept studies based on the Lancette design have shown that SFC improvements of more than 15% are possible over conventional EIS 2000 engine configurations, though this includes improvements in TET and OPR. The technology level of the respective reference engine will impact the relative improvement achieved through the recuperator, but recent studies have shown that significant improvements can be expected even when applied to projected year 2050 engines [37]. Unfortunately, the potential of the design and the resulting commercial interest in it are the reason, why the detailed design data available in the public domain is very limited. The available information is not sufficient to enable the detailed preliminary design models and trade-off studies that are the objective of this study. Hence, the Lancette design is not applied to the present combined cycle simulations.

2.5.2.2 Advanced Recuperator designs from Aristotle University of Thessaloniki

In recent years, Misirlis, Salpingidou et al. have investigated a number of different alternative recuperator designs [35], [37]. Starting from the Lancette recuperator, the CORN, STARTREC and SFR concepts were studied in cooperation with MTU Aero Engines. The Conical Recuperative Nozzle (CORN) is illustrated in Figure 2-17 a) and may be seen as the continuation of the Lancette design. The compressor exit air flows around the cone nozzle of the engine inside very small elliptical tubes. The concept essentially combines the heat transfer of all eight Lancette modules in one or two modules depending on the design of the inlet and outlet manifolds (not shown).

The CORN recuperator is a single-pass cross-flow heat exchanger, rather than a cross-counter flow arrangement as the Lancette, which may reduce its effectiveness if the heat transfer surface area were kept constant from the Lancette. However, the arrangement of the tubes means that the CORN utilises the whole annulus area, while the Lancette design required the exhaust gases to be channelled toward the modules positioned in small annulus sections. The CORN recuperator may therefore feature a larger total heat transfer surface area than the Lancette. In Figure 2-17 a), the LPT exhaust gases pass through the matrix of tubes from the right to the left. Unfortunately, further design details for the CORN recuperator are scarce especially when it comes to flow and heat transfer

characteristics or the mechanical design. This is primarily due to the continued development of the concept in the ongoing ULTIMATE programme.

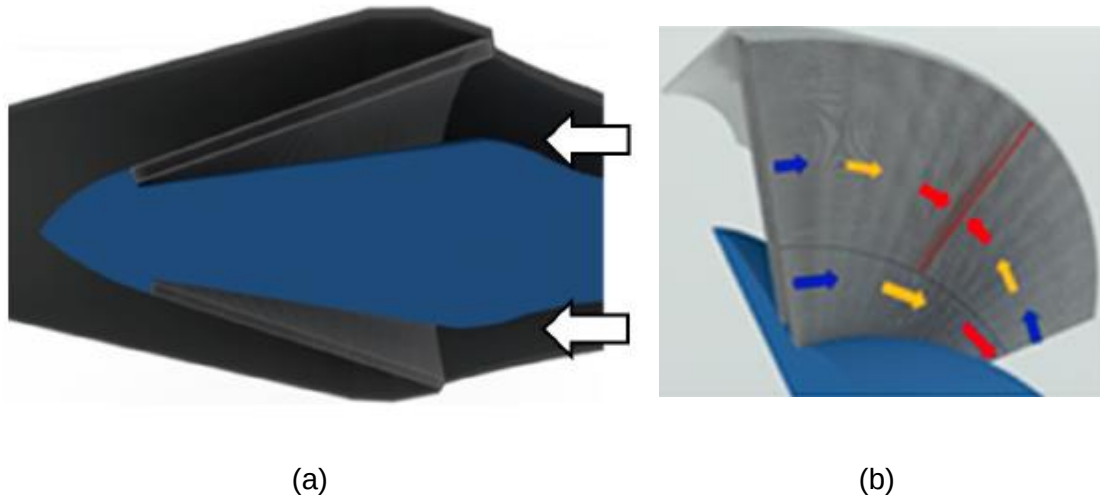


Figure 2-17: CORN and STARTREC recuperator concepts from [35]

The Straight Annular Thermal Recuperator (STARTREC) design is shown in Figure 2-17 b). The STARTREC recuperator also consists of a large number of annular tubes with an elliptical cross-section. The STARTREC recuperator was not developed as an alternative to the CORN or Lancette design, but as an additional heat exchanger module that is placed further upstream in the expansion process. The compressor exit flow would first pass through the CORN and then through the STARTREC for example. The STARTREC recuperator may be located between LPT 2 and 3 for example. This way, the combustor inlet temperature can be raised, reducing the fuel consumption. However, the pressure ratio of the engine may have to be lowered in this case, to reduce the effect of the additional pressure losses in compressor exit flow. Additionally, the reduction in OPR may be necessary to mitigate the risk of increased NO_x emissions due to higher combustor inlet and flame temperatures. Again, the design of the inlet and outlet manifolds is not depicted, but it can be assumed that one or two collectors would feed the STARTREC recuperator. Furthermore, the details of the heat transfer and pressure loss characteristics of the design are under lock and keep due to the ongoing development of the concept. Extracting heat earlier in the expansion process will undoubtedly raise the combustor inlet flow temperature, but it will also reduce the inlet flow temperature to the later LPT stages. This will reduce the turbine work of these stages, which in turn would require larger expansion ratios. Hence, it may reduce the thermal efficiency of the core. The additional hot flow pressure losses induced by the

recuperator will furthermore penalise engine performance. In order to reduce those pressure losses, the hot flow would have to be diffused before passing through the STARTREC recuperator, resulting in a longer LP shaft and increased engine weight. A longer shaft is also more prone to problems with rotor dynamics. The weight penalty may be offset by the higher specific power of the core. Additionally, the pressure losses will still be significant even if a diffuser is in place.

The ideal trade-off of these effects is not trivial and requires detailed models, which are currently only available to MTU and AUTH. Without these models it is not obvious that the additional complexity and losses of the STARTREC recuperator warrant any performance improvement. Misirlis et al. [35] determined, that the original Lancette design, as specified at the end of the NEWAC programme [94], may enable an SFC reduction of about 12.3 % when applied to an engine similar to the TF2050. The CORN design offered a slightly higher SFC reduction at 13.1 % and a simultaneous reduction in heat exchanger weight by about 5 %. However, those figures are less certain, due the concept's earlier development stage. The SFC benefit over the reference engine was reduced to 9.1 % when the STARTREC recuperator was simulated, though the weight of the heat exchanger was simultaneously reduced by approximately 50 %, which will help reduce mission fuel burn. Again, the margin of error for the STARTREC results is likely higher than it is for the Lancette recuperator. The authors concluded that the optimum recuperator design will always depend on the engine configuration, thrust rating and mission profile at hand.

Further investigation of the new recuperator concepts by Salpingidou et al. [37] has been accompanied by the development of a fourth recuperator concept at AUTH: the secondary fluid recuperator (SFR). The SFR eliminates the need to channel the compressor delivery air to the back of the engine before being returned to combustor. This should reduce the cold side pressure losses. The SFR uses two heat exchanger modules, whose geometry can be optimised largely independently, and a heat transfer fluid that circulates between the two components. The heat transfer fluid is anticipated to be a molten salt; with sodium solutions deemed the most attractive by the authors. The most recent results suggest that the SFR may offer an SFC improvement of about 8% over an engine similar to the TF2050, while the CORN and STARTREC configurations did not yield any benefits at all. The fuel burn benefits will depend on the total engine weight, which has not been discussed in [37]. The additional components (two heat exchangers, one pump, ducting) and the secondary fluid itself will introduce a

considerable penalty in terms of weight. The original Lancette design was determined to offer an SFC reduction of approximately 2.6%. The SFR concept is still under development and little data is currently available.

The limited amount of publicly available data on any of these concepts makes it virtually impossible to create a reliable heat exchanger model that incorporates preliminary design and performance evaluation. Hence, none of the advanced recuperator concepts are implemented in this study. However, especially the CORN concept might be of interest for further improvements to the main heat exchanger design, due to its inclination to the engine axis and subsequently reduced diffuser length.

2.5.2.3 The involute spiral intercooler from Chalmers University

The involute spiral heat exchanger shown in Figure 2-18 was originally conceptualised as an aero engine intercooler by Zhao et al. at Chalmers University [88]. The design has been proven to be very compact, while simultaneously avoiding excessive pressure losses and weight, therefore enabling significant mission fuel burn improvements up to 3.2% [28]. Further optimisation of the hot and cold air flow path and their operation may further increase those benefits to 4.9% [29].

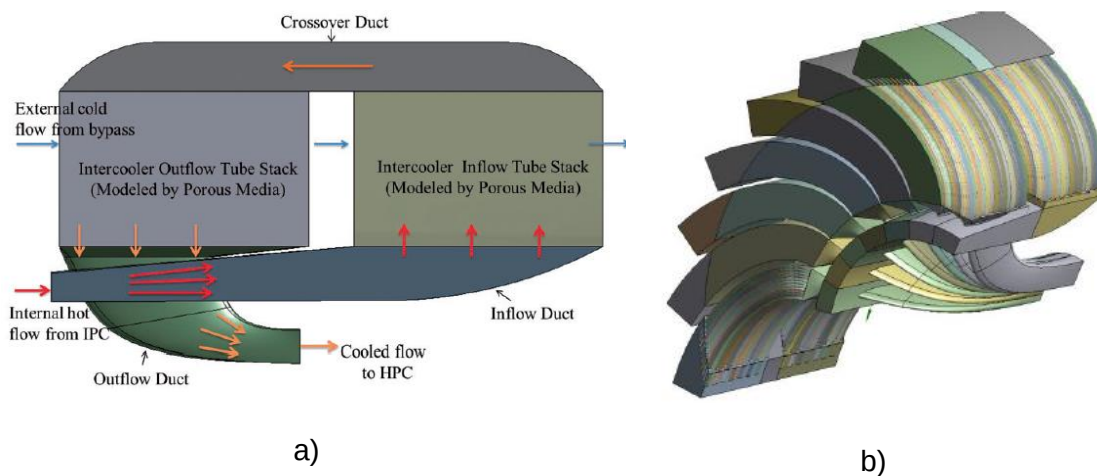


Figure 2-18: Involute Spiral Intercooler Design by Zhao et al. [28]

The IPC delivery air enters the inlet manifold of the intercooler as axial flow, before being turned radially outward into the first pass of the two-pass counter-crossflow heat exchanger (see Figure 2-18 a) for an illustration of the flow path). Following the heat transfer, the high-pressure air is axially discharged from the intercooler to the high-pressure compressor at a lower engine diameter. The continuous circumferential turning of the tubes means that the distance between individual tubes does not increase with

increasing annulus diameter. Hence, the involute spiral design requires a reduced annulus height for a given tube length relative to a straight tube design. The heat exchanger is therefore more compact, reducing installation problems and avoiding the bypass air that does not pass through the intercooler to be pushed radially outward too much.

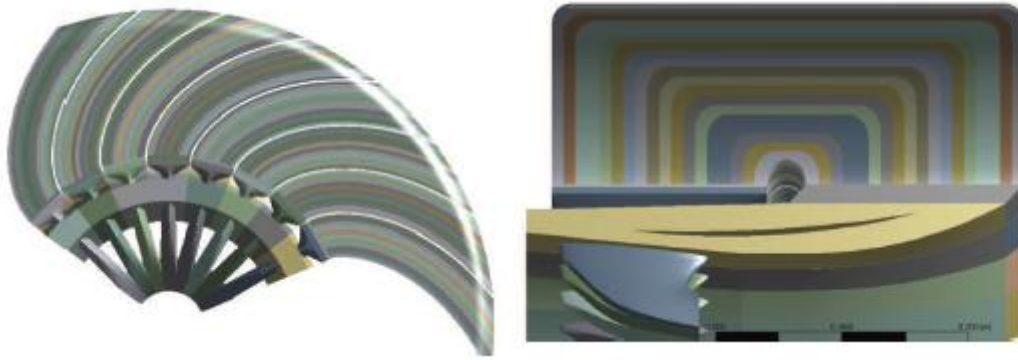


Figure 2-19: U-bent tube design of involute spiral intercooler by Zhao et al. [95]

Zhao investigated a two-pass heat exchanger design with a cross-over duct as shown in Figure 2-18 a) and b), as well as a design with continuous u-bent tubes as shown in Figure 2-19. The flow mixing in the cross-over duct was found to aid the performance and is likely to simplify manufacturing. It was therefore chosen by Zhao as the preferred option and has been adopted in the present work as well.

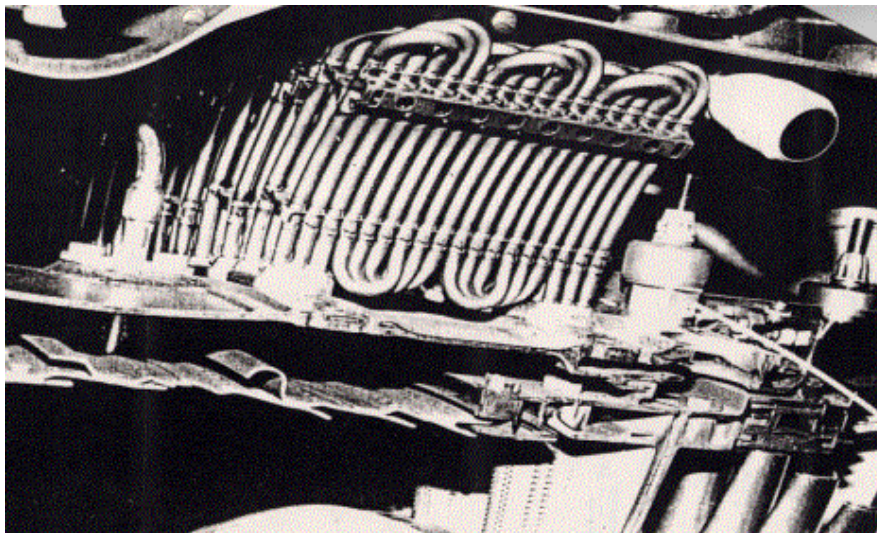


Figure 2-20: Cooled cooling air heat exchanger of the Lyulka AL-31 engine [96]

However, a different arrangement of the u-bent tubes as in the cooled-cooling air heat exchanger of the Russian Lyulka AL-31 engine [97] shown in Figure 2-20 may overcome

some of the performance benefits of the cross-over duct design and would half the amount of tube brazing necessary to manufacture the component. Such a design may therefore prove superior after all and should be reconsidered during a detailed component design procedure should the concept of a combined cycle turbofan engine prove to be lucrative overall.

The pre-cooler of the S-CO₂ bottoming cycle is likely required to deliver a similar heat duty as an aero engine intercooler and would be installed in approximately the same position as well. As such, the adaptation of the involute spiral design was deemed realistic. The thermal efficiency of the bottoming cycle is likely only around 20 %, meaning the Main Heat Exchanger (MHX) of the S-CO₂ bottoming cycle will face different working conditions, but a similar heat duty as the pre-cooler. It should therefore be feasible to employ the involute spiral heat exchanger design for the MHX as well.

2.5.2.4 Internal tube design of profiled tubes containing S-CO₂

The tubes of Zhao's involute spiral heat exchanger have an elliptical cross-section with an aspect ratio of eight, meaning the major semi-axis of the tube is eight times as long as the minor semi-axis. That configuration was deemed ideal for the intended application as an air-to-air intercooler [95]. Elliptical tubes inherently offer lower pressure losses for the outside flow at a given inlet Mach number and a greater heat transfer surface area for a given internal flow area [98]. Parts of those benefits can be offset through optimised spacing and relative positioning of adjacent tubes when working with circular cross-sections. Nonetheless, elliptical tubes very likely provide better performance overall than plain circular tubes.

However, circular tubes are usually easier to manufacture and offer greater mechanical integrity for a given wall thickness. Zhao et al. estimated that the high aspect ratio elliptical tubes would be able to withstand the internal pressure levels of the IPC exit air with a wall thickness of approximately 0.2 mm (for a 30.7 mm chord length) and without any additional strengthening measures as seen in MTU's Lancette heat exchanger [94], [99]. However, the internal pressures of the pre-cooler and the MHX are more than an order of magnitude higher than those in an aero engine intercooler. A dedicated study of the mechanical integrity of the MHX tubes was conducted by Jon Rengel-Jimenez [100],[101], and guided by Andrew Rolt and the author of this work. The study confirmed the assumption that hollow elliptical tubes would not withstand the working pressures of the MHX and that an internal support structure is necessary.

Rengel assessed the structural integrity of the tubes using the finite element modelling (FEM) capabilities of the commercially available tool ANSYS. Figure 2-21 illustrates the concept of a multi-webbed double-circular arc tube design. The departure from the original elliptical shape is due to two reasons. Firstly, the additional material at the leading and trailing edges of the tubes strengthens the design, as the point of maximum stress is otherwise likely to be on the inside of those wall sections. Secondly, a parametric elliptical tube geometry including the internal features would have required coupling an external CAD suite such as Catia with the FEM simulations, which would have increased computational time and therefore reduced the amount of designs that could be evaluated in the time available.

The change in external tube shape from an elliptical to a double-circular arc design affects the heat transfer and pressure loss characteristics of the tube and hence the performance of the heat exchanger and the whole engine. Those flow effects have not been captured by Rengel, as the focus of that particular study was on the mechanical design of the tubes. The assessment of the implications on the external flow is presently being assessed in another MSc research project conducted by Giuseppe D’Antino, which will conclude in August 2018 and is also guided by Andrew Rolt and the author. For the present work it is assumed that the influence of the change in external tube shape on the whole engine performance is negligible given the preliminary stage of the design process.

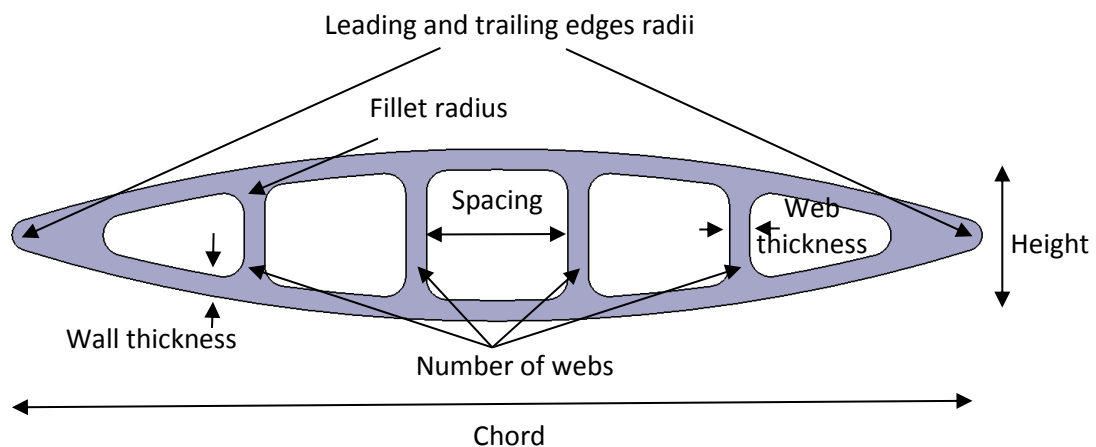


Figure 2-21: Idealised internal design of high aspect ratio double-circular arc tube

Rengel evaluated a total of 1350 combinations of the parameters illustrated in Figure 2-21, except for the chord length and tube height. All of those cases were calculated for a fixed chord length of 10 mm, as that was close to the approximate optimal tube chord length determined in earlier performance studies [102]. Additionally, all design parameters were non-dimensionalised relative to the chord length, meaning simply scaling the designs will give a reasonable answer. The height of the tube was determined through the aspect ratio of the tubes. Rengel included aspect ratios six, seven and eight in his studies, to allow for possible trade-off studies during the preliminary design process of the combined cycle engine. However, due to the lack of external flow correlations for such aspect ratios until the relevant CFD studies are concluded by Giuseppe D'Antino, those designs were not included in the parametric design. A sensitivity analysis of the combined cycle performance without a correction of the correlations of Zhao et al. will be presented in chapter 4.2.2.1 to illustrate the possible effect such a design choice may have on engine weight and therefore whole engine performance. The internal and external fluid temperatures, heat transfer coefficients and the external pressure were kept constant for most of Rengel's studies. A separate investigation confirmed that slight variations of those boundary conditions would only have a second order effect on the peak stress levels in the tube walls. The internal pressures were furthermore kept constant, as the resulting change in peak stress is directly proportional to the internal pressures and the results of the studies are therefore scalable again.

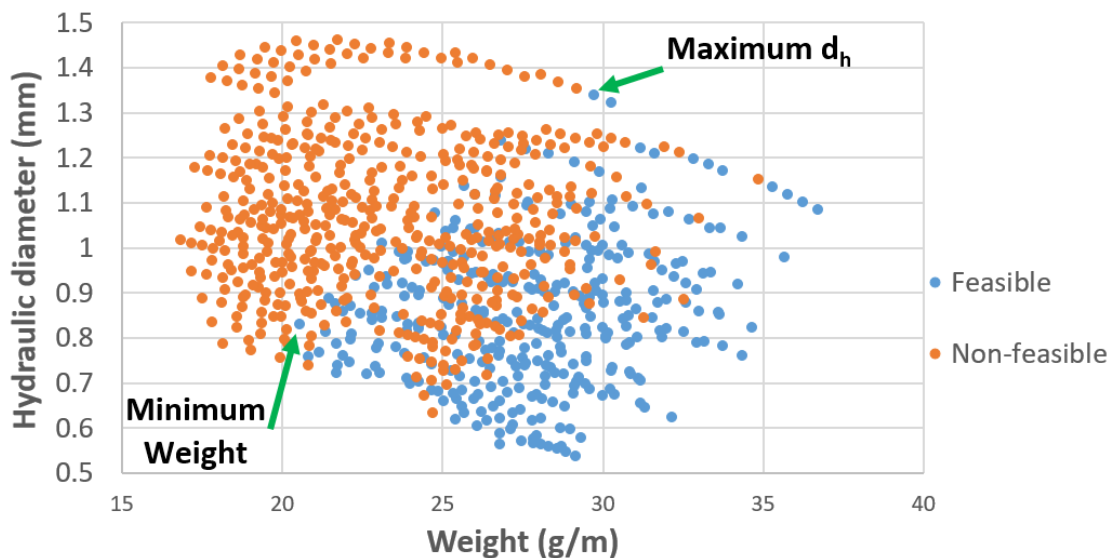


Figure 2-22: Example results of Rengel's design space study of MHX tubes [100]

Figure 2-22 illustrates two of the main output parameters of Rengel's simulations, where all assessed designs are depicted in terms of their specific weight in grams per meter and the total hydraulic diameter of the channels inside the tube. Those are the two most important output parameters of Rengel's studies as they directly affect the internal pressure losses, heat transfer coefficients and weight of the design. The tubes are assumed to be manufactured from Ti-6Al-4V. The designs are divided into 'feasible' and 'non-feasible' based on their respective peak stress levels. A maximum allowable stress of 400 MPa was chosen as the feasibility criterion by Rengel, as it corresponds to a Factor-of-Safety (FOS) of two for Ti-6Al-4V at an approximate material temperature of 532 K. The maximum allowable stress is temperature dependent for any material, because the yield strength changes with temperature. A simplified correlation of the temperature dependency of the yield strength of Ti-6Al-4V has been implemented in the design studies presented further on.

It is important to note that Rengel only simulated a tube section of approximately 10 cm, instead of the whole involute tube in most cases. This was done to reduce computational time and resources, while acknowledging that the problem was oversimplified and further optimisation would be necessary at a later stage of the design process. For example, the high pressure of the internal flow creates a bending moment forcing the tube to unwind, an effect that obviously cannot be captured in only a short part of the tube. However, a separate simulation of a complete involute tube confirmed the assumption that those longitudinal stresses are at least an order of magnitude lower than the stresses induced by the internal working pressures, making it a second order affect. Fixing the tubes' relative position to each other via hoops around the cross-over ducts may furthermore alleviate this effect. Additional effects, for example due the tube fixings, were also neglected at this stage.

The detailed design of the inlet and outlet manifolds of the heat exchanger will likely have a greater effect on the concept feasibility due to their weight than some of these simplifications. Rengel presents some initial thoughts on a redesign of the manifolds [100] relative to those envisioned in Zhao's involute spiral heat exchanger. Figure 2-23 illustrates some of those ideas, specifically designed for a tail-mounted MHX with high-density S-CO₂ inside. Further work is necessary to mature the design.

The limitations of the made assumptions were acknowledged and discussed in [100], but do not reduce the value of the work. The primary target was to create a design space of internal tube configurations and evaluate their approximate effect, so their results could

be integrated into the preliminary design and performance evaluation process of the combined cycle.

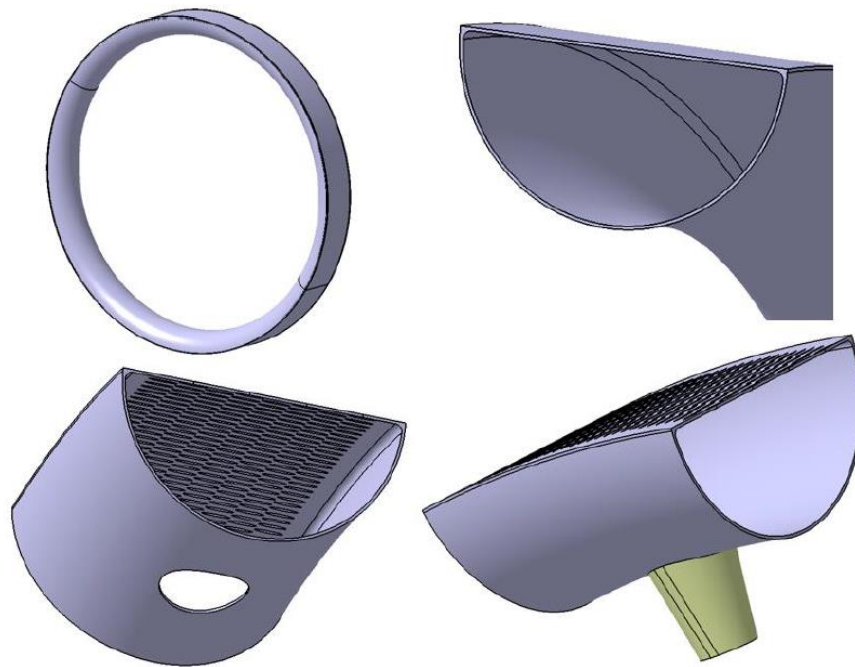


Figure 2-23: Initial design ideas for the MHX inlet and outlet manifolds by Rengel

Figure 2-22 highlights the large range of output values, underlining the necessity of Rengel's study in order to inform the design process of the combined cycle engine and enable a realistic feasibility analysis. The design space investigated by Rengel is not large enough to ensure that second order surrogate models, such as response surfaces models (RSMs), would be able to capture the effects of all inputs adequately and allow for interpolation or even extrapolation of the designs with reasonable confidence. Hence, the discrete results of the study were scaled and adapted as explained in section 3.4.1.3.

Rengel et al. also briefly discussed possible manufacturing scenarios for the complex tube geometries that proved sensible with respect to integrity and performance [101]. Whether the tubes would be drawn as those of the heat exchanger Reaction Engines Ltd. [103], folded and welded from sheet metal as the tubes for MTU's Lancette heat exchanger [92] or created through additive manufacturing is yet to be determined. Given the 2050 timeframe and the constant improvement of manufacturing processes, the authors believe that manufacturing the tubes from Ti-6Al-4V will be possible.

2.5.3 Enabling Technologies

The performance of a heat exchanger is not exclusively determined by the design of the actual heat exchanger matrix. Instead, additional components such as the inlet diffusers and possibly a variable area nozzle behind the heat exchanger have a considerable effect as well and all of these components need to be designed in unison as a system in order to really identify the optimal configuration. Hence, the following gives a brief background on previous diffuser research and its impact on whole engine performance. It is conceivable that variable area nozzles are necessary in order to minimise mission fuel burn, but the design and performance modelling of such an intricate system was beyond the scope of this work. However, a brief review of previous developments regarding variable area nozzles can be found in Appendix C.

Diffusers

The exit Mach number of the bypass air flow or the LPT exhaust gases is usually in the range of 0.4 – 0.6, depending on the application, configuration and importance of downstream flow profile [14]. Heat exchangers block a significant amount of the meridional flow area of the duct they are installed in, resulting in considerable pressure losses. In order to mitigate those pressure losses, the average flow Mach number at the front of the heat exchanger has to be reduced in a diffuser. The optimal Mach number depends on a number of parameters such as the blockage factor or the shape of the heat exchanger tubes on an tube heat exchanger, but typically it is in the range from 0.1 to 0.2 [78], [14]. Ultimately, the Mach number at the inlet of the heat exchanger is a trade-off between the pressure losses in the heat exchanger itself, the pressure losses in the preceding diffuser and the maximum available frontal area.

Diffusers can take different shapes depending on the available space and target performance. One of the most common diffusers is the conical diffuser as illustrated at the top of Figure 2-24, which uses a continuously diverging flow area to reduce the flow velocity and increase static pressure. The conical diffuser is defined by two design parameters: the area ratio (d_1/d_2) and the axial length. However, the relationship between diffuser length and area ratio is commonly represented by the cone angle, in an effort to describe as many different designs as possible with the same mathematical model. The dynamic head losses across the diffuser are a combination of non-ideal pressure recovery (flow separation) and friction losses on the diffuser wall (see equation 2-3).

$$K = 2.61 * \sin \theta * \left(1 - \left(\frac{d_1}{d_2}\right)^2\right)^2 + f_{av} * \frac{L}{d_{av}}$$

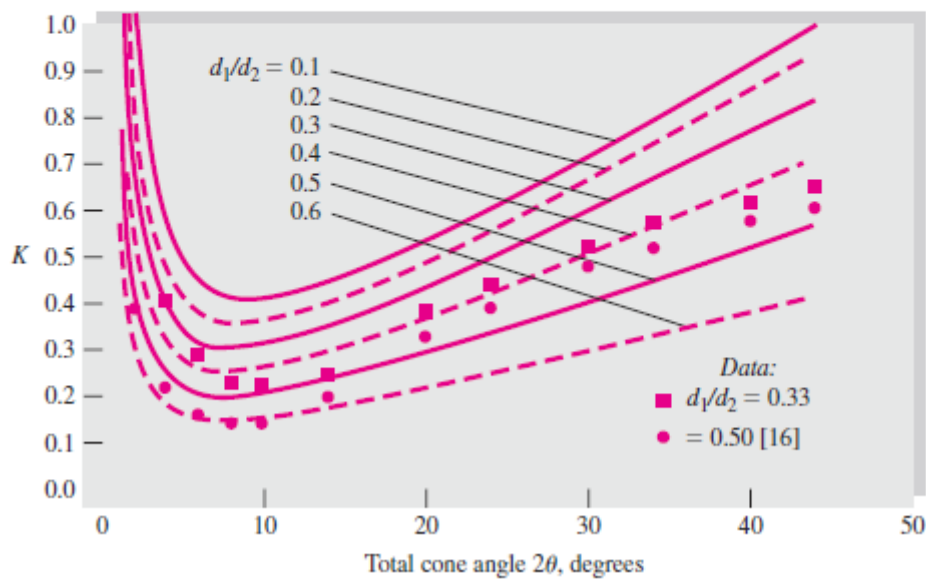
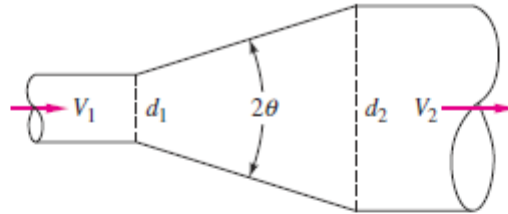


Figure 2-24: Dynamic head loss characteristic of a conical diffuser, from [104]

During the NEWAC programme, A'Barrow [105] investigated the design and performance of the coolant delivery system to an aero engine intercooler (see Figure 2-25) in great detail using computational models and component tests. Among others, he evaluated the effects of:

- inlet turbulence and flow field distortion
- the relative size and position of the diffuser to the fan outlet guide vane (mechanical design and aerodynamic performance are closely linked)
- the shape of the off-take and the forebody geometry (especially important at off-design)
- pre-diffusion
- conventional annular diffusers (controlled diffuser)
- boundary layer bleeds from the diffuser (hybrid diffuser)

transferrable to many other designs, as he investigated the diffuser design independent of the heat exchanger downstream.

Anselmi Palma [26] successfully adapted and extended the simplified methodology of A'Barrow in his investigation of the design of the novel geared intercooled reverse flow core (GIRFC) engine for example (see Figure 2-5 in section 2.2.1). Anselmi Palma had to modify A'Barrow's method, because the reverse flow core arrangement entails exhaust pipes protruding radially from the engine core into the bypass. The pipes therefore cross the inlet flow of the diffuser and change the inlet geometry. However, Anselmi Palma found that the cross-sectional shape of the inlet stream tube does not have a dominant effect on the diffuser's performance. Additionally, Anselmi Palma assumed higher pre-diffusion levels to be attainable than investigated by A'Barrow. In fact, Anselmi Palma has investigated much larger diffusion rates overall, which may be impossible to achieve in the limited space available.

A'Barrow and Anselmi Palma both assume that the pre-diffusion levels would be controlled using a variable area nozzle at the back of the intercooler duct. However, neither of their preliminary design models appears to account for the changing flow conditions of the outer bypass and the resulting change in bypass pressure losses. Unfortunately, A'Barrow's and Anselmi Palma's design methods are too complex to be incorporated in the present studies. However, their findings are still used to inform a simplified method as described in section 3.4.1.2.

2.5.4 Materials

Heat exchangers for S-CO₂ power cycles are often manufactured from high-temperature resistant steels such as austenitic stainless steel 347H [106] or nickel-base alloys like Haynes 282 [107]. The combination of enhanced fatigue strength, creep resistance and oxidation resistance make them particularly attractive. The identification of corrosion characteristics of these materials when employed on S-CO₂ power cycles is a prime objective of numerous ongoing investigations [108], [109]. S-CO₂ power cycles have not exhibited particularly strong corrosive behaviour as long as the working fluid contains few contaminants. Even stronger alloys capable of withstanding higher temperatures and pressures are constantly being developed for S-CO₂ application for high temperature applications like concentrated solar power plants. Unfortunately, most of these high temperature alloys are very expensive and feature a high density.

Lewinsohn et al. [110] propose the use of ceramic materials for S-CO₂ heat exchangers. Their studies suggest that these designs may offer improved heat transfer characteristics, due to the use of high-conductivity ceramics ($k > 50 \text{ W/(m}\cdot\text{K)}$). The main advantages of ceramic heat exchangers are that they would be more easily customisable and feature a lower component weight due to the reduced density. However, ceramic heat exchangers are not yet commercially ready. Furthermore, the pre-cooler and the main heat exchanger should probably not be made from ceramics, due to the risk of foreign object damage and soot deposition in case of the MHX.

Titanium-aluminium alloys are frequently used in medium to high temperature environments on aircraft, due to their relatively low density, good fatigue strength and high yield strength at elevated temperatures (about 650 MPa at 700 K). Ti-6Al-4V [111] is a commonly used aircraft material with well documented characteristics and approximately half the density of Haynes 282 (4.3 g/cm^3 vs 8.3 g/cm^3). Hence, Ti-6Al-4V is assumed to be used for the all heat exchangers of the combined cycle engine. Ti-6Al-4V is not going to lead to a significant cost reduction relative to nickel-base alloys however. It may also be possible to manufacture the pre-cooler from aluminium due to the relatively low working temperatures, but a detailed mechanical design study would be necessary to prove that concept. Hence, aluminium was dismissed as a possible material for the purposes of this study.

2.6 Turbomachinery in S-CO₂ Cycles

The design of S-CO₂ turbomachinery components, especially compressors, differs from the procedures for ideal gas components, due to the near incompressibility of the fluid near the critical point. A number of studies have been published over the years that investigated the real gas effects on turbomachinery design and performance [75], [112], [113]. For example, it is possible that the thermodynamic conditions reach the vapour-pressure curve due to local flow accelerations near the leading edge of the impeller. Baltadjev et al. [112] also determined that the surge margin of a compressor can be reduced by up to 9 % when operated near the critical point. Most of these studies utilised high-fidelity tools such as CFD to validate their findings against the data available in the public domain [113] or from in-house test rigs. These kinds of investigations are beyond the scope of the present study. Lower fidelity 1 and 2D models as presented by Monge [113] or Qi et al. [114] have been shown to produce reasonable performance estimates and component maps at lower computational cost, though these techniques are still too complex to be integrated in the current investigation.

One of the biggest disputes is what kind of turbomachinery efficiencies are attainable for S-CO₂ turbomachinery components. Qi et al. [114] concluded that the radial turbines of their design would feature total-to-static efficiencies of approximately 78 to 82 %. Those turbines were designed for the 100 – 200 kW range and featured extremely small blades with a minimum height as low as 0.9 mm. At the other end of the spectrum (100 MW), Moroz et al. [115] identified 2D turbine designs with isentropic efficiencies around 95 %. For a similar power scale, Dostál estimated the component efficiencies of his compressor to 89%, while the turbine efficiency was fixed to 90%. However, a subsequent 2D design study of the turbomachinery components with the commercially available AXIAL code [116] revealed that the compressor efficiencies would be around 95 % and the turbine efficiency would be approximately 93 %. These figures match those of Moroz et al. well.

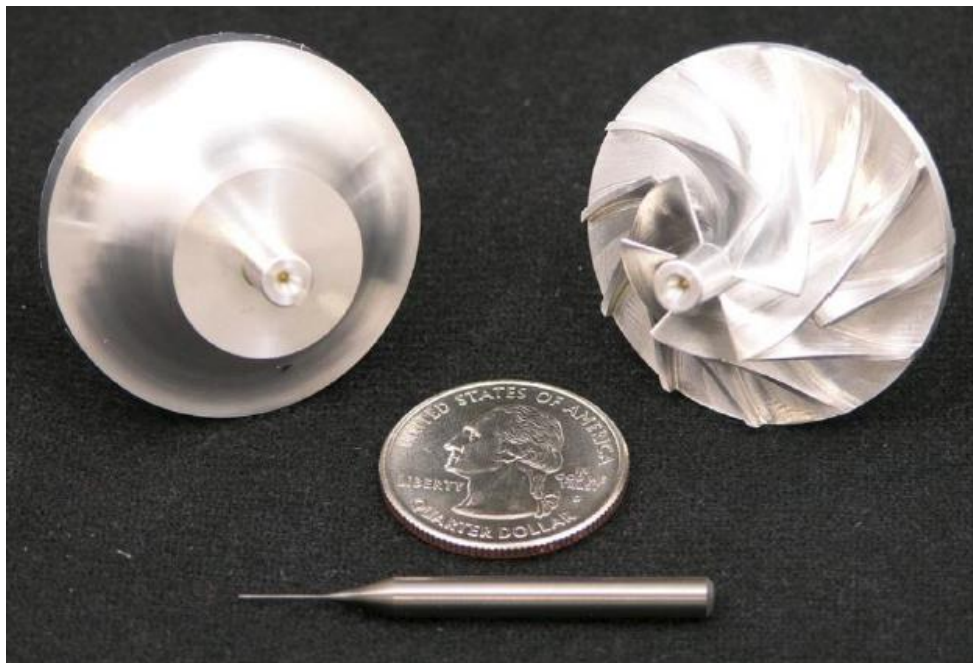


Figure 2-26: Size comparison of finished 50 kW (3.5 kg/s) compressor wheel for Sandia test rig, one of the manufacturing tools and a US dime, from [68]

The scale of the components has such a significant influence on their efficiency, because loss mechanisms like over-tip leakage are crucial in smaller components [117], but less important for large machines. Unfortunately, not all efficiency claims can be compared directly to each other, as some studies do not include the losses in the stator vanes (e.g. Qi et al. [114]), while others even take losses due to shaft seals and gas foil bearings into account [118], [119]. Based on these studies, it seems reasonable to assume isentropic efficiencies between 83 and 89 % for compressors and between 85 and 90 %

for turbines for an S-CO₂ power cycle that enters into service in 2050 and has a net shaft power output of around 1 MW.

Very little data is currently available due to the limited amount of operational S-CO₂ power cycles. The compressor on the test rig at Sandia National Labs (see Figure 2-26) reportedly features a design pressure ratio of 1.8 with an isentropic efficiency of 67 %. Barber-Nichols Inc. (BNI) delivered the compressor, the turbine and the corresponding component maps to Sandia. Similar parts were delivered to Bechtel Marine Propulsion Corporation (BMPC) [69], but their test data showed considerably better performance than predicted. The BMPC test rig was initially unable to reach full power conditions due to some technical issues, but the compressor efficiency at half-load exceeded even the design point estimate and it can therefore reasonably assumed, that the initial assumption of around 67% isentropic efficiency was overly conservative.

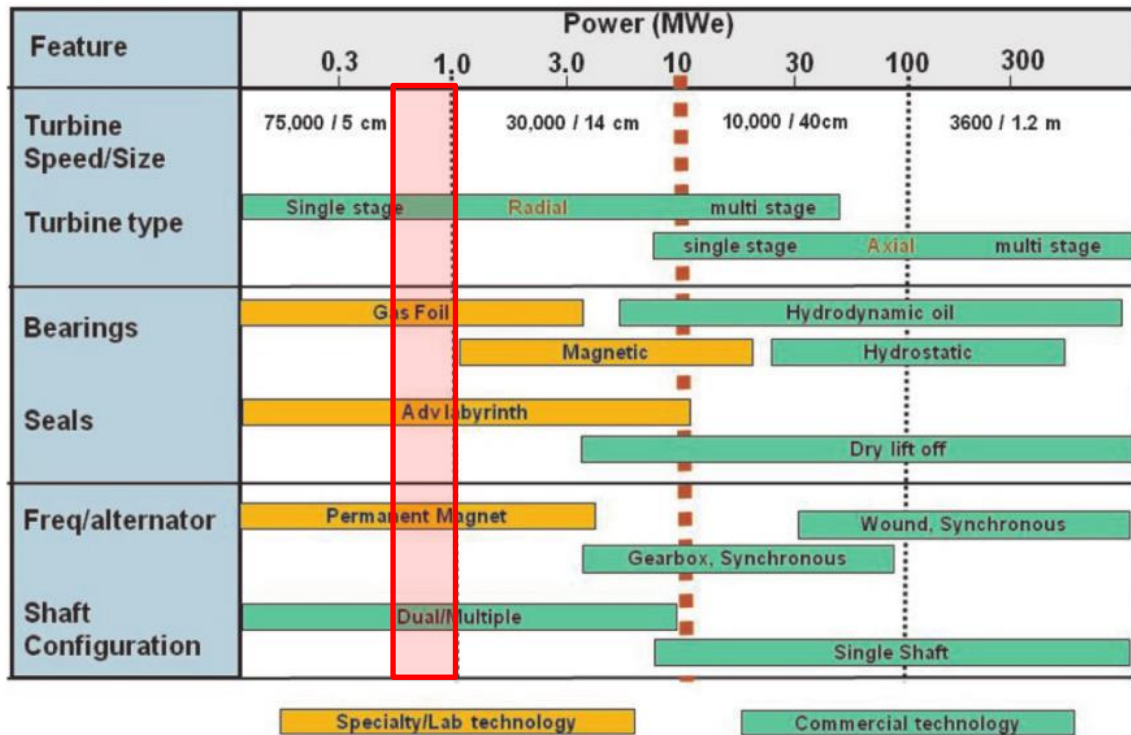


Figure 2-27: Application ranges for key S-CO₂ turbomachinery design features, adapted from [62]

The Department of Energy [62] provides a generalisation of the main design choices for S-CO₂ turbomachinery based on the design experience gathered by US institutions such as Sandia, BMPC and NETL until 2015. The resulting graphic can be seen as a best practice guide and is shown in Figure 2-27. A red box highlights the approximate power

range of the S-CO₂ bottoming cycles of the present investigation. The overview includes guidelines with regard to bearing and sealing technology as well. Bearing and sealing technology is a key research area to date. Small-scale power cycles, to which even the present bottoming cycle for aero application would count according to Figure 2-27, usually feature gas foil bearings, because only labyrinth seals are really applicable, resulting in a necessary pressurisation of the bearing chamber. If dry lift off seals were developed further so that they could be applied on such small components, hydrodynamic journal bearings may become feasible, which might alleviate the windage losses presently dominating S-CO₂ turbomachinery performance.

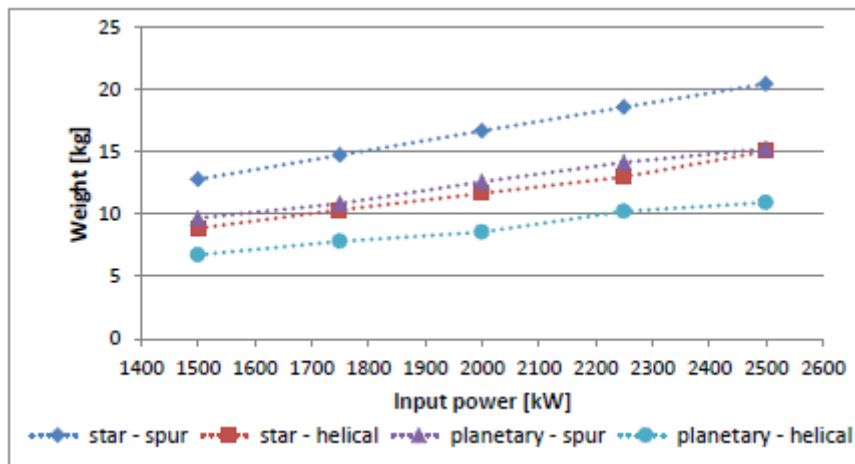


Figure 2-28: Weight estimation of different transmission systems, from [120]

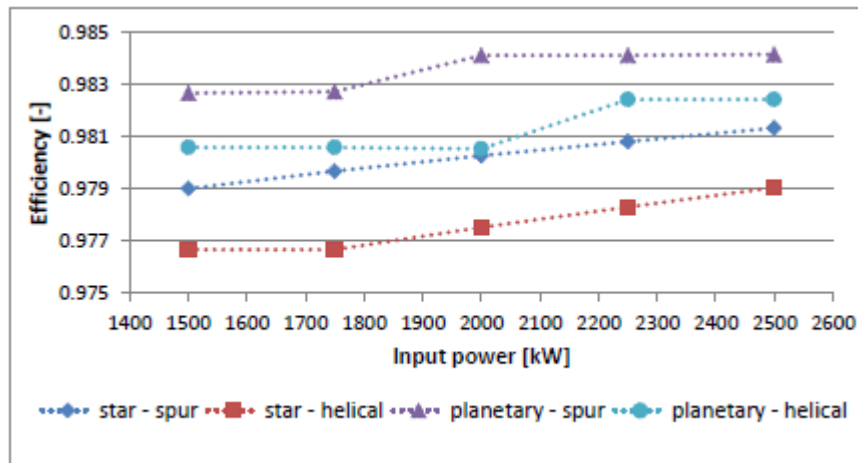


Figure 2-29: Efficiency estimation of different transmission systems, from [120]

In a combined cycle aero engine, the power produced by the bottoming cycle needs to be transferred to the LP shaft of the main engine via a power transmission system. Canovas [120] contributed to the present work by investigating different mechanical arrangements during her MSc research project and developing a preliminary design tool for a selection of these configurations. Canovas found, that a fixed gear ratio planetary gearbox transmitting approximately 2 MW of shaft power would weigh between 10 and 15 kg (see Figure 2-28) and reach an efficiency of around 98 % (see Figure 2-29). More detailed design studies would surely enable even higher gearbox efficiencies. Unfortunately, further development of the combined cycle concept determined that the power transmission should not feature a fixed gear ratio, making a purely mechanical design unlikely. Electric power transmission based on superconductivity might be available by 2050 and would free the rotational speed of the S-CO₂ power turbine from the constraints of the main engine's LP shaft.

2.7 Conclusion

This chapter outlines the limits of existing publications on combined cycle aero engines and shows that most of them used simple performance characteristics like pressure losses and heat exchanger effectiveness without a realistic component design methodology behind it. The presented state of the art in terms of S-CO₂ power cycles is beyond such simplifications, but only includes a single publication on S-CO₂ bottoming cycles for aero application. The component specific sections furthermore presented possible design choices and previous developments that could be built upon in this work.

Table 2-3 lists the highlights of a brief patent review conducted for this work on combined cycle aero engines and power cycles working with S-CO₂ and other working fluids (see Appendix D for further details). By no means could this survey replace a thorough patent review by a patent attorney, but the lack of data on S-CO₂ bottoming cycles for aero application underscores the current assessment that such a concept has been rarely, if ever, investigated.

In summary, this dissertation seeks to close the gap between existing research on S-CO₂ power cycles and combined cycle aero engines, by conceptualising a combined cycle aero engine with S-CO₂ as the working fluid of the bottoming cycle and determining its preliminary design.

Table 2-3: Comparison overview patents - current research

	Similarities	Differences
Matthew Alexander Lehar (US2014/0352306 A1)	• S-CO₂ as a working fluid • Recuperated cycle	• Recompressed cycle • Stationary power generation • Trans-critical cycle (Rankine not Brayton) • Other fluids possible
Laing & Booth (US2014/0150443 A1)	• S-CO₂ as a working fluid • Integrated cycle design	• GE main engines • Stationary power generation • More heating and cooling cycles • Secondary aft gas burner
Freund et. al. (US2012/0159923 A1)	• Bottoming Cycle	• Steam as working fluid • Stationary power generation • Focused in concept of intercooling of gas turbine through bottoming cycle
Isaac Schnaid (US5442904 (A))	• Bottoming Cycle	• Air as working fluid • Open cycle • Stationary power generation

3 Methodology

The methodology applied in this study is one of the main contributions to knowledge hereof. The common approach of assuming likely performance parameters based on experience, and designing and optimising the respective components thereafter is unfeasible when one of the working fluids shows strong real gas behaviour. The shortfall of the common modelling approach and how the methods applied in this study can help avoid excessive amounts of design iterations are the focus of section 3.1.

Following that reasoning and description of the overall project methodology, section 3.2 introduces the modelling environment of the presented models. This is necessary to establish the unique features of the software and point out why even more detailed modelling can sometimes be detrimental when working in that environment.

Subsequently, the created models are explained individually, starting with the fluid property modelling in section 3.3.

The heat exchangers are some of the most influential components for the overall concept evaluation. They are therefore the most detailed models presented and applied in the current investigation. The details of each dimension of those models are explained in section 3.4, including arguments why this methodology was chosen over other options.

Section 3.5 briefly outlines the modelling approach behind the S-CO₂ turbomachinery components, including a discussion of possible power transmission systems. Unfortunately, no model was included for the preliminary design of either component at this point, as those would have been too complex and beyond the scope of this study.

Section 3.6 explains the performance model of the secondary combustor in the engine configurations featuring inter-turbine reheat.

The integration of the main engine and the bottoming cycle models, their boundary conditions, the simplified off-design performance modelling and the component scaling approach of the main engine are subject of section 3.7. The section furthermore describes how and in what order the different tools for engine performance, general arrangement and aircraft design were used during the design space explorations and how some of the top-level design constraints were applied.

All sections highlight the limitations of the described models and their assumptions as clearly as possible, no matter if those limitations are due to mechanical, performance or

other concerns. This shall help outline the remaining shortcomings of the chosen methods in chapter 5 and give a justification for assumptions made where appropriate. The model's actual sensitivity to input parameters not included in the parametric design studies are presented in chapter 4 of this work as well. Hence, this chapter is structured in such a way, that the level of detail of the models, their limitations and how they all tie together in the combined cycle performance simulations, may be appreciated.

3.1 Overall Project Methodology

3.1.1 Requirements

The feasibility of the combined cycle aero engine concept is ultimately decided by the performance and weight of the heat exchangers involved and by the balance between these two objectives. If the heat exchangers are too large, any SFC benefit will be offset by the additional weight and drag, prohibiting any fuel burn benefit. If their effectiveness is too low or the pressure losses are too high, they won't enable any SFC benefit at all, no matter how light the heat exchangers may be. Hence, it is crucial that the heat transfer, pressure losses, weight and dimensions of the heat exchanger are appropriately accounted for even in a concept study such as this.

Common modelling approach for concept studies

Many concept studies are conducted in a two-step iterative approach. First, the engine performance potential is approximated using OD performance assumptions that are based on experience as much as possible. Data from previous projects may be extrapolated to estimate reasonably applicable ranges of component efficiencies, the effectiveness of heat exchangers or pressure losses. Afterwards, individual components may undergo a preliminary design process to match the modelling assumptions and check if the original inputs to the performance models need to be updated. Finally, the performance may be recalculated, the overall benefit of the concept established and the sensitivity to the modelling uncertainties can be quantified. This process also takes place in the industry, though in a much more integrated way and including more design iterations. This common approach is reliable and can be quick to assess the potential of a novel engine concept involving components that have been researched or even been in use for years.

Section 2.5 presented a number of studies on the detailed design of heat exchangers that operate with air or engine exhaust gases [35],[121]. Many of the studies show how

their size and performance are correlated and that for example an intercooler with an effectiveness of around 70% can likely be produced [25],[122] and help improve overall engine performance. The application of the ϵ -NTU method, best described by Kays and London [78], is a reliable method for that kind of investigation. Kays and London present test data for a host of different heat exchanger geometries and relate their performance directly to their size. This allows for the evaluation of many different designs with relatively little computational effort. Detailed design studies may reveal that an effectiveness of 75% or 65% is ideal from a fuel burn perspective, but at that point of the process the change in size and performance will be relatively small.

The need for an alternative modelling approach

However, when working with completely new components or concepts whose performance and size cannot be extrapolated from previous data in a reliable manner, this approach may become slow and unfeasible. One of the main objectives of this work was therefore to establish a set of modelling approaches and tools that integrate the preliminary design of the most crucial novel components and their performance with the performance of the whole engine. That way, the feasibility of the combined cycle engine concept can be assessed with less uncertainty and fewer design iterations. Furthermore, technology and module requirements, which would need to be reached should the further development of the concept seem lucrative, can be identified early on.

The components involved in this study are not completely new geometric designs, but are based on a number of developments from other projects. However, their application in a combined cycle aero engine using supercritical carbon dioxide in the bottoming cycle is new, and the application of S-CO₂ is also the main reason, why the integration of preliminary design and performance evaluation is crucial.

Heat exchanger effectiveness vs. temperature effectiveness in a real gas HX

The case study presented further on clearly shows how the real gas behaviour of S-CO₂ near its critical point affects the performance of the pre-cooler and why a more integrated design and performance modelling approach is necessary when evaluating S-CO₂ cycles. However, before the results can be interpreted, it is important to discuss why depicting the hot and cold-side temperature effectiveness (see equations 3-1 and 3-2) is more representative when simulating real gas heat exchangers than depicting the heat exchanger effectiveness calculated according to Kays and London [78]. The reason lies

in the way the heat exchanger effectiveness is commonly calculated (see equations 3-3 and 3-4).

$$\varepsilon_{temp,hot} = \frac{T_{t,in,S-CO_2} - T_{t,out,S-CO_2}}{T_{t,in,S-CO_2} - T_{t,in,air}} \quad 3-1$$

$$\varepsilon_{temp,cold} = \frac{T_{t,out,air} - T_{t,in,air}}{T_{t,in,S-CO_2} - T_{t,in,air}} \quad 3-2$$

$$C_{min} = \min(c_{p,S-CO_2} * W_{S-CO_2}, c_{p,air} * W_{air}) \quad 3-3$$

$$\varepsilon = \frac{c_{p,S-CO_2} * W_{S-CO_2} * (T_{t,in,S-CO_2} - T_{t,out,S-CO_2})}{C_{min} * (T_{t,in,S-CO_2} - T_{t,in,air})} \quad 3-4$$

The variation in specific heat capacity of the S-CO₂ can affect the calculated heat exchanger effectiveness significantly just by changing the numerator of equation 3-4. The change in temperature variation for the S-CO₂ flow may not even yet be accounted for therein. If the effectiveness of a fixed geometry increases when changing the S-CO₂ outlet conditions, it is not easy to determine whether that is because the average flow temperature of the S-CO₂ is slightly closer to the critical point, which would entail a larger c_p , or because the achieved temperature change has increased. Unfortunately, only the latter option is of interest from a pure engine performance perspective, as the pre-cooler needs to restore the turbine exit flow to the compressor inlet temperature and pressure.

The temperature effectiveness is a simpler metric that allows for a more direct interpretation of the results and the real gas effects. Any increase in hot-side temperature effectiveness can only be due to an increase in S-CO₂ inlet temperature or a decrease in S-CO₂ outlet temperature. An increase in cold-side temperature effectiveness on the other hand can be the result of a reduced S-CO₂ inlet temperature or an increased coolant outlet temperature. The heat exchanger's thermal performance on both sides can therefore easily be interpreted through a simultaneous evaluation of the hot and cold-side effectiveness.

Case study: Why is a different modelling approach necessary for S-CO₂ cycles?

Figure 3-1 and Figure 3-2 illustrate why such an integrated design and performance modelling approach is recommended to model S-CO₂ cycles. During the simulations, the S-CO₂ mass flow, the coolant inlet mass flow, and the coolant inlet temperature and pressure were kept constant. The heat exchanger geometry was furthermore fixed, while the pre-cooler outlet conditions of the S-CO₂ were varied. The model calculated the

temperature and pressure variation on both sides of that particular heat exchanger by calculating the S-CO₂ inlet and the coolant outlet conditions. Note that the pressure losses in the ducts connecting the pre-cooler and the compressor only have a second order effect and were neglected here.

Figure 3-1 and Figure 3-2 clearly show that the achievable temperature drop in the pre-cooler strongly depends on the inlet conditions of the S-CO₂ compressor. Close to the critical point, the specific heat capacity of S-CO₂ varies greatly for small changes in pressure and temperature. This means that the pre-cooler has to transfer a substantial amount of heat to achieve the last few degrees of cooling required. The total change in S-CO₂ fluid temperature across a pre-cooler with a fixed heat duty is therefore immensely dependent on the compressor inlet temperature. It is important to note, that the compressor inlet temperature and pressure both have a strong influence on the pre-cooler's performance, as they both significantly affect the local specific heat capacity in a heat exchanger section. Hence, the achievable S-CO₂ temperature drop in the pre-cooler is dramatically reduced, if the compressor operates close to the critical point. In the relatively small range of simulated compressor inlet temperatures, the hot side temperature effectiveness varies between 26 % and 87 % due to the variation in fluid properties near the critical point. This represents a range in achievable temperature drop between 31 K and 101 K for the same mass flows. Hence, the pre-cooler may be significantly undersized for a targeted temperature change, if preliminary design and performance are not linked directly in the models. Operating points where the compressor inlet conditions are at or extremely close to the critical point were not included in this study, as they would only be attainable with a very detailed and fast control system for the bottoming cycle, which is unlikely to be in place in an aero engine.

This study clearly shows why simply prescribing an effectiveness, instead of imposing a preliminary design and calculating the performance accordingly, is not a feasible design method for real gas heat exchangers. The achievable temperature variation of a fixed size pre-cooler strongly depends on the proximity of the compressor inlet conditions to the critical point. If the problem is viewed from a design perspective, the size of the pre-cooler that is required to achieve a certain temperature drop is strongly dependent on the compressor inlet conditions. As the pre-cooler weight (>150 kg) is crucial to the whole engine performance, accounting for these trade-offs is indispensable when assessing this concept.

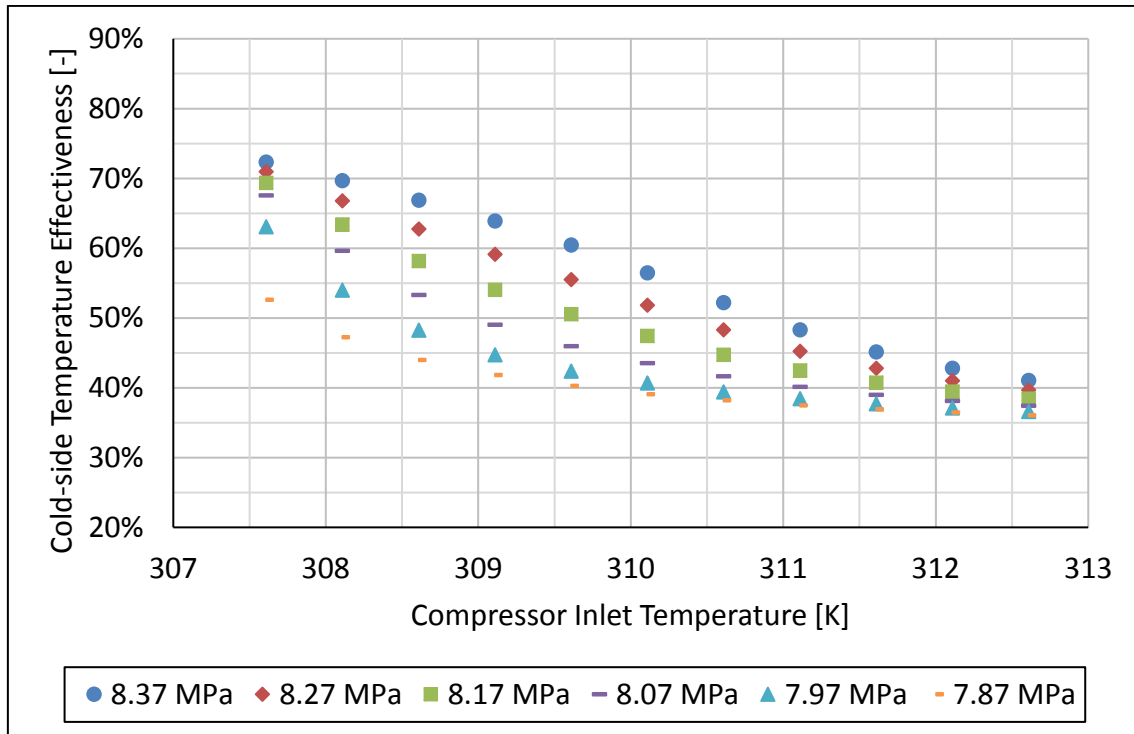


Figure 3-1: Dependency of cold-side heat exchanger effectiveness of the pre-cooler on compressor inlet conditions

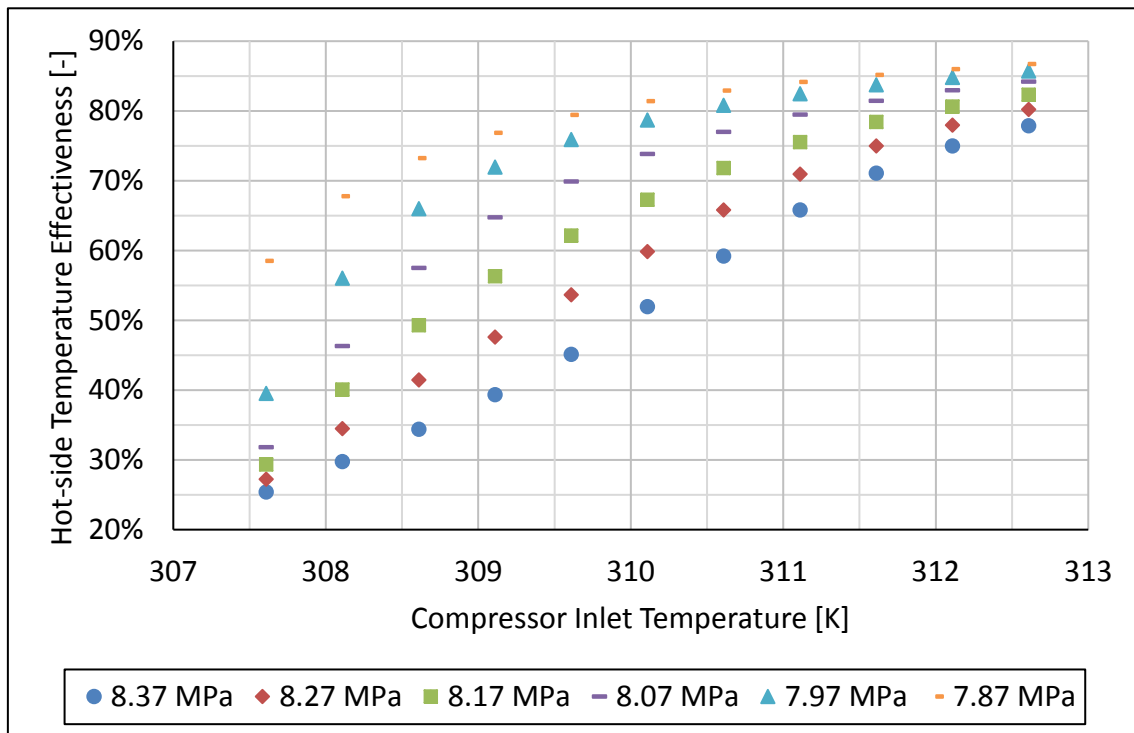


Figure 3-2: Dependency of hot-side heat exchanger effectiveness of the pre-cooler on compressor inlet conditions

The impact of the pre-cooler geometry however does not end with its effect on the bottoming cycle's performance or the resulting weight. It also affects the bypass flow that does not pass through it, by deforming the outer bypass duct. Hence, those effects also need to be modelled to get a more holistic picture of the combined cycle engine performance. Again, the coupling of these effects would be very difficult to account for with the more established simplified modelling approach. While the main heat exchanger (MHX) is not affected by variations in fluid properties as strongly as the pre-cooler, it is still a more noticeable effect than in an air-to-air heat exchanger, again necessitating the imposition of a design and the calculation of its performance thereafter.

3.1.2 Overall project methodology applied in this study

Once the component and engine models were developed to incorporate satisfactory levels of detail, the performance maturation studies presented in chapter 4 were conducted according to the following approach:

- I. The input design parameters for the engine performance and component design are chosen.
- II. The engine performance is calculated using the integrated combined cycle engine models.
- III. The engine general arrangement and total weight is calculated.
- IV. The aircraft design trade factors are used to estimate a fuel burn benefit and calculate the respective new thrust requirement.
- V. Steps II to IV are repeated until convergence is reached on fan diameter and thrust.

This way, each investigated engine configuration is matched to a different aircraft based on its mission fuel burn requirements. This approach allows for a reliable comparison of mission fuel burn values, as all aircraft are designed for the same payload. A more conventional engine and aircraft design procedure, as discussed at the start of section 3.1.1, ultimately would have arrived at very similar results. However, it would probably have taken considerably longer than with the approach presented here, because it is extremely difficult to outright estimate top-level performance parameters such as heat exchanger effectiveness for real gas components. Additionally, the applied method simplifies investigations of the engine's sensitivity to individual component design details.

Throughout the present investigations, the same approach was furthermore used to determine the answers to some general design questions like:

- Would the CCTF2050 benefit from having a PCHE recuperator in the bottoming cycle? (No)
- How many nodes does the pre-cooler have to be discretised into for the results to be realistic without impeding calculation times? (About five per pass)
- Does it make sense to have a multi-pass design for the pre-cooler and/or the MHX? (No)

Following the maturation of the concept, the potential of the engine-aircraft configurations to reduce emissions and their possible effect on direct operating costs are discussed (see section 5.3).

3.2 Simulation Environment – PROOSIS

All engine performance and preliminary design methods presented in this chapter are implemented in the commercially available “Propulsion Object Oriented Simulation Software” (PROOSIS 3.6.14, Empresarios Agrupados Internacional) unless specifically mentioned otherwise. Standard PROOSIS simulation tools include quasi-1D representations of the most influential engine components such as turbomachinery, shafts and nozzles. These 1D models primarily use component maps and top-level performance parameters such as heat exchanger effectiveness and pressure losses, but do not include geometric representations of heat exchangers by default. New component models that included the described methods therefore had to be created.

PROOSIS furthermore does not feature supercritical carbon dioxide as a possible working fluid by default. Fluid properties and the implementation of thermodynamic functions are separated as much as possible, but the default functions are based on the assumption of working with a near-ideal working fluid. In order to adequately simulate the performance of S-CO₂ components, it was therefore also necessary to adapt new versions of existing components, such as compressors and turbines, which can account for real gas behaviour. The implementation of the fluid properties and the relevant functions are subject of section 3.3. If necessary, Appendix E gives a slightly more detailed introduction to modelling in PROOSIS in general.

Perhaps the most important feature of PROOSIS is that it is non-sequential unless it is explicitly forced to follow a certain code sequence by the user. This means, that the order

in which equations are inserted is irrelevant and will most probably not be the order in which they will be solved during the simulations. However, not every combination of input parameters will be a sensible choice as a design input in the end, not only because of their physical meaning, but because choosing them might create an extremely complex iterative mathematical problem. The user defines which of the parameters that are not used as inputs are iterated upon during the simulations. These iterative parameters are called algebraics in PROOSIS and the stability of the engine model is extremely sensitive to the choice of algebraics. The initialisation of the algebraics is not an exact science by any measure and sometimes a model takes longer or is completely unable to converge, when the initial guess is too close to the final solution.

Ultimately, PROOSIS is a great simulation platform when exploring a novel engine configuration such as this, because it is not only relatively easy to code in, but it will allow for different kinds of studies to be performed with one and the same code. For example, the pre-cooler geometry can be used as an input, together with the inlet flow conditions on either side of the heat exchanger, and the model will calculate the heat transfer and all intermediate parameters. Alternatively, the same source code can be used to calculate the necessary heat exchanger geometry, by imposing the inlet flow conditions on the cold side, and the inlet and outlet flow conditions on the hot side. This is possible because the partition will rearrange the relevant equations. The down-side to this flexibility is that not all combinations of input parameters will lead to stable models. In addition, a particularly complex model, such as a combined cycle aero engine, may require modifications of the initial guesses of some of the algebraics depending the simulated operating point. This need for manual initialisation is the reason, why it was not possible to create standalone executables of the combined cycle model and integrate those in an external optimiser.

3.3 Supercritical CO₂ Fluid Properties

Capturing the real gas effects across the whole range of possible operating conditions, but especially close to the critical point is essential when simulating S-CO₂ cycles. Most S-CO₂ Brayton cycles investigated to date, aim to utilise the dramatic change in fluid properties near the critical point in the compressor to reduce compressor work and increase the cycle's thermal efficiency. The rapid changes in fluid properties are furthermore very influential on the performance and design of the heat exchangers in the system, making an accurate portrayal of their behaviour indispensable. Section 2.4.1 outlined the difficulty to represent real gas behaviour accurately and concluded that the

Span and Wagner [60] equation of state (EOS) is the most widely accepted and accurate EOS for S-CO₂. The free REFPROP mini software tool, developed by the National Institute of Standard and technology (NIST), was used to create the S-CO₂ fluid property model in PROOSIS. REFPROP calculates the fluid properties of carbon dioxide based on the Span and Wagner equation of state.

Default models of ideal gas fluid properties in PROOSIS

The implementation of ideal gas fluid properties and their use in PROOSIS was developed and presented in the doctoral thesis of Sethi [123]. In his work, Sethi explains the inner workings of the fluid model, the flow continuity modelling and which parameters are necessary to establish the flow properties at any point of the engine. By default, the fluid models of PROOSIS were created to represent ideal gas behaviour only. The default fluid property model was implemented in three stages. First, look-up-tables were created that list all necessary fluid properties such as specific heat capacity, enthalpy, Prandtl number and more for a discrete set of input variables. The variables are the fluid temperature, fuel-to-air ratio (FAR) and water-to-air ratio (WAR). Secondly, interpolation routines were created to enable the homogenous depiction of the fluid properties across the whole design range. The interpolation routines are used in what are called fluid functions in PROOSIS, to calculate the exact value of a given fluid property for any given flow condition. At the start of every PROOSIS calculation, the fluid property tables are loaded once and the fluid functions are called as necessary. The third step in the implementation of fluid properties in PROOSIS was the creation of what are called therm functions in PROOSIS. Therm functions allow for total-to-static property conversions for differing sets of input parameters based primarily on the corrected mass flow equation for ideal gases. Different functions needed to be implemented for example depending on whether the flow area or the Mach number is known for any given combination of total pressure, temperature and mass flow. All functions are solved sequentially in PROOSIS, which is why multiple implementations of the same thermodynamic equations are necessary. It would be possible to implement total-to-static property conversions directly into a component code as well, eliminating the need for multiple implementations of the same physical phenomenon depending on what inputs are used. However, this would not only make component codes less clear, it would also significantly reduce model stability.

New real gas models created for S-CO₂

Visiting researcher Mathieu Belmonte led the process of implementing a similar modelling approach for the real gas properties of S-CO₂ [61]. He created the look-up-tables for S-CO₂ using Matlab and REFPROP mini. The real gas fluid model is based on the assumption of a pure working fluid, eliminating the need for FAR and WAR as qualifiers to calculate the fluid properties. However, real gas behaviour is dependent on the working pressure as well, so the fluid property tables and the whole fluid model use the temperature and the pressure as the input variables. Belmonte conducted a sensitivity study to determine the necessary resolution of the fluid property tables to accurately simulate the real gas behaviour of S-CO₂. The resulting fluid property file of S-CO₂ is orders of magnitude larger than that required for air and air-mixtures, raising the load time at the start of any PROOSIS simulation from less than 0.1 seconds to approximately 4.5 seconds. This time increase obviously affects the number of configurations that can be evaluated in a given timeframe, but is necessary to capture real gas behaviour as accurately as possible. The CO₂ fluid property tables capture a range from 2 to 40 MPa and 225 to 2000 K with differing resolutions depending on the proximity to the critical point. Those ranges do not allow for total-to-static flow conversions beyond Mach 0.6, as those conditions would be uneconomical inside the heat exchangers anyway. Using fluid property tables and interpolation routines in PROOSIS is still considerably quicker than calling REFPROP every time a fluid property is needed, which is an approach commonly applied in other simulation environments. The routines to interpolate between the discrete points of the fluid property tables are independent of the content of the tables and therefore did not need to be implemented again. Fluid and therm functions on the other hand needed to be created specifically, as the corrected mass flow in its common form is not viable for real gases. The library of therm functions for real gases was populated as needed during this investigation, based on one example created by Belmonte (see Appendix E of [61]).

Unfortunately, validation of the therm functions was not possible, due to the lack of comparable data in the public domain. The fluid functions on the other hand could be validated against the data produced by REFPROP. The deviation between the interpolated results of the fluid functions and the explicit values calculated by REFPROP mini was less than 0.1 % for all cases investigated by Belmonte [61].

3.4 Heat Exchangers

Two types of heat exchanger models were developed for this investigation: the involute spiral multi-pass cross-counter flow tube heat exchanger for application as the pre-cooler and the main heat exchanger, and a printed circuit heat exchanger to be used as the counter flow S-CO₂-to-S-CO₂ recuperator. The following subsections explain the models, equations and modelling assumptions that led to the final models. Where appropriate, other possible modelling techniques or implications of assumptions made are discussed to clearly outline the reasoning behind the chosen approach.

Two main assumptions were made for both heat exchanger models:

- I. The total mass flow rate on either side of the heat exchanger is uniformly distributed between all flow channels and the performance can therefore be determined via numerical correlations.
- II. The temperature profile of the flow inside the tubes is calculated based on average flow conditions for the pass. The temperature profiles of individual tubes in a single pass are not calculated as their effect is accounted for in the generalised flow correlations.

All heat exchanger models work on an enthalpy basis, rather than using specific heat capacity, as the real gas behaviour of S-CO₂ can lead to large deviations from the real performance if specific heat capacity is used. Specific heat capacity is only used in instances where the introduced error is not significant and reducing the computational time is more valuable. Those instances are clearly highlighted, as with the estimation of thermal conductivity of S-CO₂ discussed in section 3.4.1.4.

3.4.1 Involute Spiral Heat Exchanger

The Pre-cooler and the Main Heat Exchanger (MHX) of the S-CO₂ bottoming cycle were modelled as multi-pass, involute spiral heat exchangers with elliptical tubes. The design was primarily based on the information provided by Zhao et.al. in [88], [95]. Several changes were made to the model, be it on the definition of the annulus space the heat exchanger is to fill or the correlations used to calculate the heat transfer coefficients for the flow inside the tubes. The subsequent sections provide a detailed description of the sub-models and the changes relative to the original model of Zhao et al.

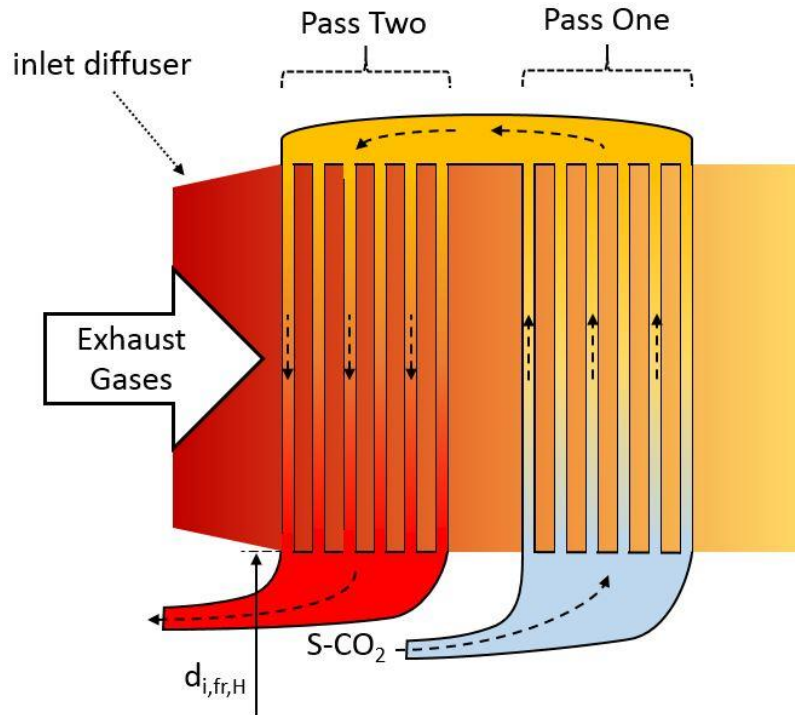


Figure 3-3: Schematic of Main Heat Exchanger (MHX)

Differences between the pre-cooler and the MHX are explicitly mentioned where necessary, otherwise the two components are modelled analogously. Both heat exchangers operate with S-CO₂ inside the tubes and air or exhaust gases around the tubes, with the obvious difference being that the S-CO₂ is the low temperature fluid in the MHX and the high temperature fluid in the pre-cooler. This results in a re-assignment of the ‘hot’ and ‘cold’ indices in the code between the outside and inside flows in the two components. If not mentioned otherwise the parameters are designated for the MHX with the cold side describing the S-CO₂ flow inside the tubes and the hot side describing the exhaust gases flowing around them. The schematic of the MHX in Figure 3-3 shows how the nomenclature of the model is chosen so that the first pass (or pass one) is the pass the S-CO₂ flows through first. The second pass is therefore upstream in the LPT exhaust gases and downstream for the S-CO₂. This is important for the discussions further on. The same principle is applied to the pre-cooler, with the pass furthest away from the fan designated as pass one.

Section 3.4.1.1 details the geometric definition of the model and what parameters are essential model inputs during the combined cycle performance simulations.

Section 3.4.1.2 outlines the modelling approach for the diffusers upstream of the heat exchangers.

Section 3.4.1.3 explains the implementation of Jon Rengel-Jimenez' findings on the internal tube structures necessary for high aspect ratio tubes containing S-CO₂ (see section 2.5.2.4).

Sub-chapter 3.4.1.4 highlights the heat transfer calculations, which numerical correlations were used and why, and how they are linked to other parts of the model.

Section 3.4.1.5 explains the pressure loss calculations, while section 3.4.1.6 outlines the modelling behind the discretisation of the pre-cooler into multiple short heat exchanger sections for the sake of accuracy.

Section 3.4.1.7 explains how the thrust produced by the heated pre-cooler exit air is accounted for in PROOSIS. The limitations of the implemented models and the applicable limits of the established parameters are explained where appropriate, as are the deviations from the original intercooler model of Zhao et. al [88].

Finally, section 3.4.1.8 gives a conclusive overview of the input values used during the performance studies.

3.4.1.1 Parametric Design and Weight Estimation

The tubes immediately downstream of a set of tubes is circumferentially offset to the middle of the gap between the first set. Such a staggered arrangement has been determined favourable in terms of heat transfer and pressure loss for elliptical tubes [122]. This way the external flow area remains relatively constant, mitigating losses induced by repeated sharp flow acceleration and diffusion. How much acceleration and diffusion still occurs along the external flow path depends largely on the relative tube spacing.

Figure 3-4 shows an idealised cross-section of the heat exchanger tubes with the external flow passing around the tubes from left to right and the S-CO₂ flowing inside the tube's internal channels. The figure also highlights some of the geometric parameters of the tubes and their arrangement. Note that in this case, the tube geometry is defined via the full major and minor axis of the elliptical or, in case of the figure, double-circular arc tube geometry. The external minor axis of the tube is dubbed $b_{tube,H}$ and the major axis is dubbed $a_{tube,H}$. The relationship of the two is defined via the tube aspect ratio (AR) as per equation 3-5. Equation 3-6 describes the circumferential tube spacing as the ratio of the gap between adjacent tubes and the minor tube axis. Equation 3-7 captures the equivalent relationship for the axial spacing.

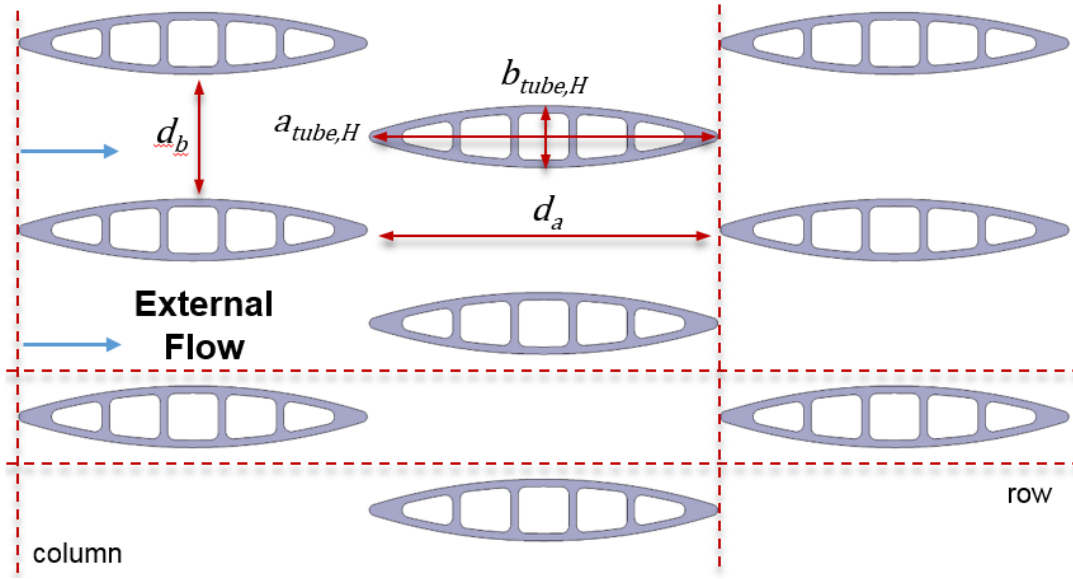


Figure 3-4: Schematic of relative tube arrangement

In this study, the circumferential and axial spacings are fixed to $m_{tube} = 2$ and $n_{tube} = 1$. Parametric studies of different spacings conducted by Zhao [88] revealed that this arrangement offered the best trade-off between heat transfer and external pressure losses and has therefore been adopted for the current investigation. The ongoing work of Giuseppe D'Antino also investigates, if this is still true for an S-CO₂ application with double-circular arc tubes.

$$AR = \frac{a_{tube,H}}{b_{tube,H}} \quad 3-5$$

$$m_{tube} = \frac{d_b}{b_{tube,H}} \quad 3-6$$

$$n_{tube} = \frac{d_a}{a_{tube,H}} \quad 3-7$$

Figure 3-4 also defines a tube column as all circumferentially aligned tubes and the offset set of tubes downstream. This means that integer and half integer values can be imposed for the number of columns as a design input, depending on where the last set of tubes should be placed. It is furthermore important to note that a tube row describes all axially aligned tubes in a pass. The number of rows is therefore dependent on the circumferential spacing and the inner annulus diameter of the duct they are placed in.

During the definition of the heat exchanger geometry, an approximate inner annulus diameter of the heat exchanger is imposed as a design boundary. The approximate

diameter is a direct input for the pre-cooler model, while it is calculated as a fraction of the LPT exit inner annulus diameter in the MHX (see equation 3-9). That initial diameter is used to estimate the number of tube rows per stack. The number of stacks and the spacing of the tubes are also design inputs. The number of tube rows per stack is rounded down (using floor function shown in equation 3-8) to full integer values and a new inner annulus diameter is calculated therewith.

$$N_{rows,stack} = \text{floor}\left(\frac{\pi * d_{i,fr,H,init}}{N_{stacks} * b_{tube,ext} * (m_{tube} + 1)}\right) \quad 3-8$$

$$x = \frac{d_{i,fr,H,init}}{d_{i,e,LPT}} \quad 3-9$$

The number of tubes per stack is simply the product of the number of tube columns and rows per stack, although a factor of two is necessary, due to the definition of a tube column to encompass the offset set of tubes as well. The total number of tubes per pass and for the whole heat exchanger are subsequently calculated according to equations 3-10 through 3-12. The axial length of the stack is calculated according to equation 3-13. The length of the inlet and outlet manifolds is estimated via equation 3-14.

$$N_{tubes,stack} = N_{rows,stack} * N_{col} * 2 \quad 3-10$$

$$N_{tubes} = N_{tubes,stack} * N_{stacks} \quad 3-11$$

$$N_{tubes,tot} = N_{tubes} * N_{pass} \quad 3-12$$

$$L_{stack} = N_{col} * (n_{tube} + 1) * a_{tube,H} * N_{pass} \quad 3-13$$

$$L_{duct} = L_{stack} * 1.15 \quad 3-14$$

The width of an individual heat exchanger stack is calculated according to equation 3-15, which in turn is used to calculate the S-CO₂ side frontal area of the heat exchanger (see equation 3-16). The flow area inside the inlet ducts of a stack just before the flow enters the tubes is seen as the frontal area of S-CO₂ side in these models. The free-flow area of the S-CO₂ is simply calculated via equation 3-17.

$$Width_{stack} = \frac{\pi * d_{i,fr,H}}{N_{stacks}} \quad 3-15$$

$$A_{fr,C} = \frac{L_{stack}}{N_{pass}} * Width_{stack} \quad 3-16$$

$$A_{ff,c,stack} = N_{tubes,stack} * A_{ff,c} \quad 3-17$$

$$W_{stack} = \frac{W_{in}}{N_{stacks}} \quad 3-18$$

The heat exchanger frontal area for the external flow is calculated using the corrected mass flow and a Mach number design input (usually between 0.1 and 0.2 [78], [122], [124]). The requirement for the total pressure at the heat exchanger inlet means an iterative procedure of the diffuser and heat exchanger design, as the total diffuser pressure losses will influence the annulus height, which in turn will affect the cone angle of the diffuser. The inlet Mach number is chosen as a design input in favour of the inlet area itself, as it makes the design more intuitive and the results more comparable, especially when the engine is scaled. The mass flow per stack on either side of the heat exchanger is calculated using equation 3-18.



Figure 3-5: Example of leaned involute tubes by Reaction Engines Ltd. [103]

The involute shape of the tubes enables a substantial reduction in necessary annulus height for a given tube length relative to a straight tube design. This benefit can be increased, if the tubes do not start perpendicular to the inner annulus wall of the heat exchanger duct. If the tubes are already slightly bent when they leave the inner annulus, the total tube length will be even greater for a fixed annulus height (see Figure 3-6 for an illustration of that effect). The tube equivalent to the red line in Figure 3-6 would be 25%

longer than tube of the blue line. Reaction Engines Ltd. has proven this feature's benefit with the design and testing of their pre-cooler [103]. The tubes in that heat exchanger depart the inner annulus at an extremely shallow angle, enabling long tubes in a relative small annulus. The benefit that those longer tubes grant in terms of heat transfer surface area will be offset by the additional pressure losses due to friction inside the tubes. Mathematically, the involute shape of the tubes is determined by the inner annulus diameter at which the involute starts, and the actual annulus height.

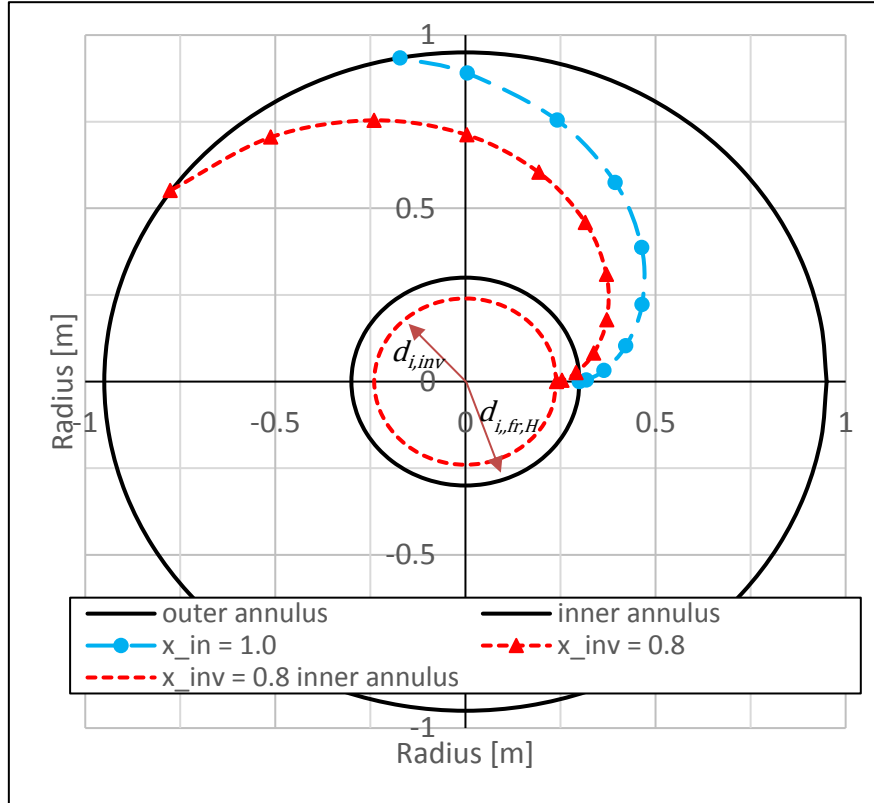


Figure 3-6: Effect of non-perpendicular start of involute tubes

This can be accounted for by moving the theoretical start of the involute inside the actual annulus and subtracting the section of the tube that would then lie inside the annulus walls from the total tube length. Equations 3-19 through 3-22 list the mathematical implementation of this approach. The ratio of the theoretical inner diameter of the involute and the actual inner diameter of the duct is called x_{inv} and is a direct design input for the heat exchanger models.

$$d_{i,inv} = d_{i,fr,H} * x_{inv} \quad 3-19$$

$$\theta_i = \sqrt{\left(\left(\frac{d_{i,fr,H}}{d_{i,inv}}\right)^2 - 1\right)} \quad 3-20$$

$$\theta_o = \sqrt{\left(\left(\frac{d_{o,fr,H}}{d_{o,inv}}\right)^2 - 1\right)} \quad 3-21$$

$$L_{tubes} = \frac{d_{i,inv}}{4} * (\theta_o^2 - \theta_i^2) \quad 3-22$$

The ratio of blocked flow area and total frontal area of outside flow is calculated according to equation 3-23 and is used to subsequently calculate the free-flow area that remains between the tubes for the outside flow (see equation 3-24).

$$\sigma_{HX,H} = \frac{m_{tube}}{m_{tube} + 1} \quad 3-23$$

$$A_{ff,H,stack} = \frac{A_{fr,H}}{N_{stacks}} * \sigma_{HX,H} \quad 3-24$$

The hydraulic diameter of a tube heat exchanger should be calculated using equation 3-25 and 3-26, as per Kays and London [78]. This is also the model implemented by Zhao [122], hence, the same has been implemented here.

$$A_{ratio,ext} = \frac{P_{tube,H} * N_{tubes,stack} * L_{tubes}}{A_{ff,stack,H}} \quad 3-25$$

$$d_{h,tube,H} = \frac{4 * L_{stack}}{A_{ratio,ext} * N_{pass}} \quad 3-26$$

The hot and cold side heat transfer surface areas per stack are calculated using only the tube dimensions (see equations 3-27 and 3-28). Secondary heat transfer via the plates the tubes are brazed to is neglected.

$$A_{h,stack,C} = P_{tube,C} * L_{tubes} * N_{tubes,stack} \quad 3-27$$

$$A_{h,stack,H} = P_{tube,H} * L_{tubes} * N_{tubes,stack} \quad 3-28$$

The S-CO₂ side inlet and outlet manifolds were specially designed by Zhao to aid flow distribution and minimise pressure losses. The same models and assumptions were introduced here. In Zhao's intercooler model, the flow is assumed to approach the inlet manifold in an annular duct (see equations 3-29 to 3-31). While the same model has

been implemented here, detailed design of the manifolds may determine a different optimal design closer to that of the pre-cooler produced by Reaction Engines Ltd. for the SABRE engine (see Figure 3-5). However, those changes likely have a second order effect on the combined cycle weight and performance. Additionally, the pressure losses in the inlet manifold are possibly overestimated by this model, as Zhao assumes that the outlet manifold needs to pass through the inlet manifold due to the positioning of the intercooler. If the components of the combined cycle engine are arranged as assumed thus far, that crossover will not be necessary, eliminating the associated pressure losses.

$$d_{o,duct,in} = \sqrt{\left(\frac{4 * A_{tot,duct,in}}{\pi} + d_{i,duct,in}^2\right)} \quad 3-29$$

$$A_{duct,in} = \frac{A_{tot,duct,in}}{N_{stacks}} \quad 3-30$$

$$P_{duct,in} = \frac{(d_{i,duct,in} + d_{o,duct,in}) * \pi}{N_{stacks}} + (d_{o,duct,in} - d_{i,duct,in}) \quad 3-31$$

The turning of the flow inside the inlet manifolds is assumed to be aided by a deflecting vane in Zhao's design, with the assumption that the flow is also diffused along that vane through the change in flow direction [88]. Detailed studies by Zhao have concluded that an area ratio of two is ideal, hence equation 3-32.

$$A_{defl} = A_{duct,in} * 2 \quad 3-32$$

The involute spiral heat exchanger modelled here, features a cross-over duct between passes, as discussed in section 2.5.2.3. The hydraulic diameter of that duct is calculated according to equation 3-33.

$$d_{h,cross} = \frac{4 * A_{fr,c}}{2 * Width_{stack} + 2 * \frac{L_{stack}}{N_{pass}}} \quad 3-33$$

An additional change made to the model relative to Zhao's original model is an option to change the number of passes of the heat exchangers. Theoretically there is no limit to the number of passes that may be simulated, but simulations with more than four passes failed to converge due to the significant increase in algebraics. This capability was added as a possible way to increase the heat transfer surface areas and counteract the significant imbalance between the internal and external heat transfer coefficients. However, initial simulations showed that the additional pressure losses and larger

component weight outweighed the benefits of the additional heat transfer, so the mission fuel burn of a multi-pass arrangement was worse than that of a two-pass arrangement. Hence, the number of passes has been kept at two during the parametric studies presented in chapter 4.

Equation 3-34 estimates the total volume of the heat exchanger material. The first term of the equation accounts for the material necessary for the tubes, while the remainder of the equation approximates the material necessary for the inlet and outlet manifolds and the cross-over duct. Assuming a material density of 4330 kg/m³ for Ti-6Al-4V [111], that volume is used to calculate the total weight of the heat exchanger.

$$V = N_{tubes,tot} * L_{tubes} * A_{sol} + \frac{\pi}{4} * L_{duct} * N_{stacks} * (N_{pass} + 1) \quad 3-34$$

$$* \left(\left(2 * tube_{th} + \sqrt{\left(\frac{4 * A_{duct,in}}{\pi} \right)^2} \right)^2 - \left(\sqrt{\left(\frac{4 * A_{duct,in}}{\pi} \right)^2} \right)^2 \right)$$

$$M = V * \rho_{mat} \quad 3-35$$

3.4.1.2 Diffuser

It is necessary to reduce the inlet velocity for the air or exhaust gases flowing around the heat exchanger tubes to avoid excessive pressure losses, as discussed in section 2.5.3. The models to calculate the performance and preliminary design of the advanced diffuser systems described in section 2.5.3 are too complex to be implemented in the current models. The dedicated models for pre-diffusion, controlled annular diffusion and a bled diffuser result in too many iterations of the heat exchanger inlet flow conditions, prohibiting convergence, particularly when coupled with the already quite detailed and highly iterative further performance models involved in the presented work.

However, conventional annular diffusers cannot deliver the necessary diffusion levels (area ratios of approximately four) in a sufficiently short duct as the findings of A'Barrow [105] and Anselmi Palma [26] have confirmed. Hence, it was necessary to implement a simplified model that enables the approximation of the diffuser performance in terms of total pressure loss depending on cone angles, area ratios and diffuser length, without sacrificing convergence or computational power. Equation 3-36 is the commonly applied Gibson correlation to estimate the dynamic head loss of a controlled diffuser as presented by White [104] extended with a technology factor (TF). That technology factor was introduced to account for the performance improvements of a bled or hybrid-diffuser

design as discussed by A'Barrow and Palma. The technology factor was calibrated, so the equation estimates a dynamic head loss of 12%, relative to the rotor exit flow for a hybrid diffuser (1.5% bleed off-take) presented by A'Barrow with a total length of 0.7 m and an area ratio of 2.13.

$$K_{diffuser} = \left(2.61 * \sin \alpha * \left(1 - \frac{A_{e,LPT}}{A_{fr,H}} \right)^2 + f_{av} * \frac{L_{diff}}{H_{av}} \right) * TF \quad 3-36$$

$$TF = 0.767 \quad 3-37$$

$$f_{av} = \left(-1.8 * \log \left(\frac{6.9}{Re_{av}} + \left(\frac{\xi_{rel,diff}}{3.7} \right)^{1.11} \right) \right)^{-2} \quad 3-38$$

$$\xi_{rel,diff} = \frac{\xi_{abs,diff}}{H_{av}} \quad 3-39$$

The surface friction in the diffuser is estimated using Haaland's correlation for Moody friction factors [125] (see equation 3-38 above). The relative surface roughness is calculated according to equation 3-39. The average height of the annular duct (see equations 3-40 to 3-42 below) is used to calculate the Reynolds number of the flow in the diffuser.

$$H_{e,LPT} = \frac{(d_{o,LPT} - d_{i,LPT})}{2} \quad 3-40$$

$$H_{stack} = \frac{d_{o,fr,H} - d_{i,fr,H}}{2} \quad 3-41$$

$$H_{av} = \frac{(H_{e,LPT} + H_{stack})}{2} \quad 3-42$$

The diffuser model is integrated directly into the heat exchanger models of the MHX and the pre-cooler. A separate implementation of the models may be more intuitive and would mean that one diffuser model may be used for different applications. However, the pre-cooler and the MHX are both designed for a given inlet Mach number, meaning the diffuser exit area and consequently the area ratio are dependent on the non-dimensional mass flow and therefore total inlet pressure. Consequently, it would have been necessary to pass a number of internal parameters between the diffusers and the heat exchangers and iterate these effects. This would have made the engine schematics and the component models unnecessarily complex and probably less stable.

The diffuser is thought to begin at the trailing edge of the fan outlet guide vane (FOGV), whether the diffusion process actually starts there or not, as this is in line with the geometric definitions of A'Barrow and Anselmi Palma (see Figure 3-7 again). It is unlikely for the diffuser to be designed in a way that even pre-diffusion would start immediately downstream of the FOGV, as pre-diffusion and the forebody geometry of the off-take have been shown to impede the efficiency of the FOGV [105] through upstream effects. Generally, the minimal distance between the FOGV trailing edge and the start of the pre-diffuser should be at least 50% of the FOGV chord length [105]. The airflow close to the FOGV hub that is ingested by the diffuser is at a lower Mach number than the rest of the bypass stream. It is commonly assumed that the inlet Mach number to an intercooler diffuser is between 0.45 and 0.4 [26],[122], with the latter being used in the current model. This is a reasonable assumption for the innermost 10 to 12% of the bypass flow [126], [127].

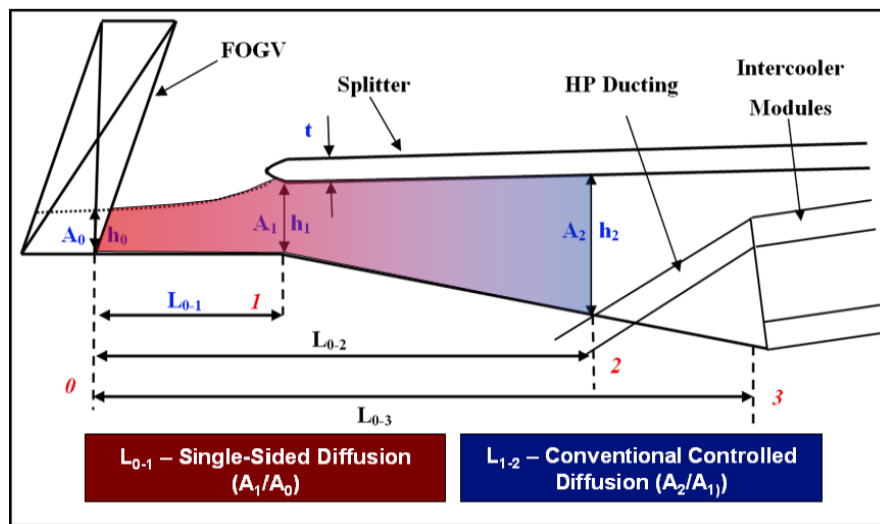


Figure 3-7: Schematic of coolant delivery system and dimensions for intercooled engine, from [105]

3.4.1.3 Internal design of elliptical Heat Exchanger tubes

The internal tube geometry is scaled from the design space explorations of Rengel [100]. Originally, the involute spiral heat exchanger tubes had an elliptical cross-section, but this was changed to a double-circular arc design by Rengel as explained in section 2.5.2.4. Rengel conducted a parametric study where the tube wall thickness, aspect ratio, web thickness, web count and the fillet radii (see Figure 3-8) were varied, creating a design space of 1350 designs for a chord length of 10 mm. An even distribution of the webs along the chord length proved to be best and the leading and trailing edge radii

were fixed to 1.5% of the chord length. The data generated by Rengel was stored in an Excel file and scaled as appropriate by a MATLAB script for each case of the current investigation.

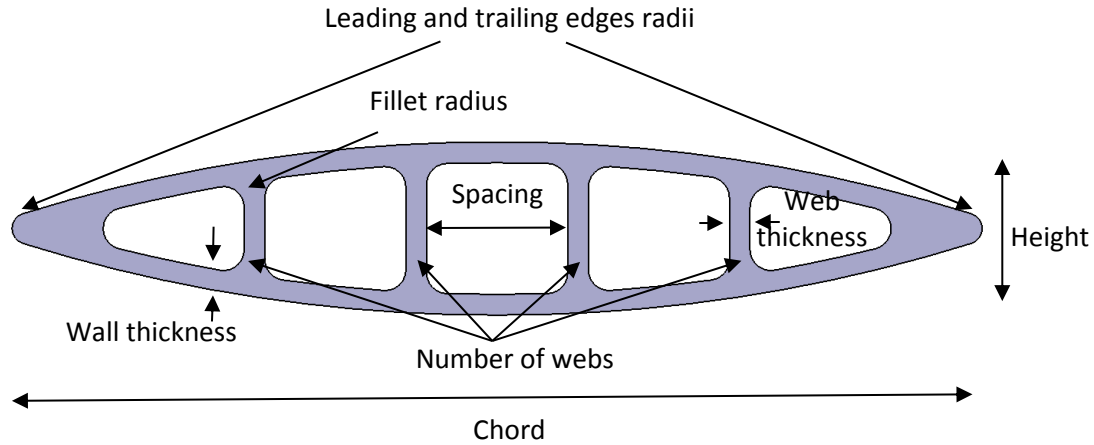


Figure 3-8: Idealised internal design of high aspect ratio double-circular arc tube

Figure 3-9 illustrates the decision-making process implemented in that script. To scale the internal tube geometry from the available data, general design boundaries, such as the aspect ratio of the tube, the maximum pressure inside the components, or the number of feasible designs per ‘filter category’, need to be imposed in the input file. Each time the script is executed, the input and output parameters of Rengel’s simulations are loaded into the cache twice, once specific to the MHX and once for the pre-cooler as both geometries are determined simultaneously but for individual inputs. Afterwards, the raw data is scaled according to the imposed boundaries. The peak stress of a design is scaled linearly with the maximum pressure inside the heat exchanger.

Distances and perimeters are scaled linearly, and all areas are scaled quadratically, with tube chord length. The solidity of each design is not scaled as it will remain constant. Minimum wall and web thickness levels of 0.1 mm were respectively established for manufacturing reasons, dismissing any designs below those thresholds. Following the scaling of the data, the maximum allowable peak stress level is determined for each heat exchanger using the inputs of average wall temperature and factor of safety on a temperature dependent yield strength calculation. Subsequently, only the designs that match the imposed tube aspect ratio are carried forward.

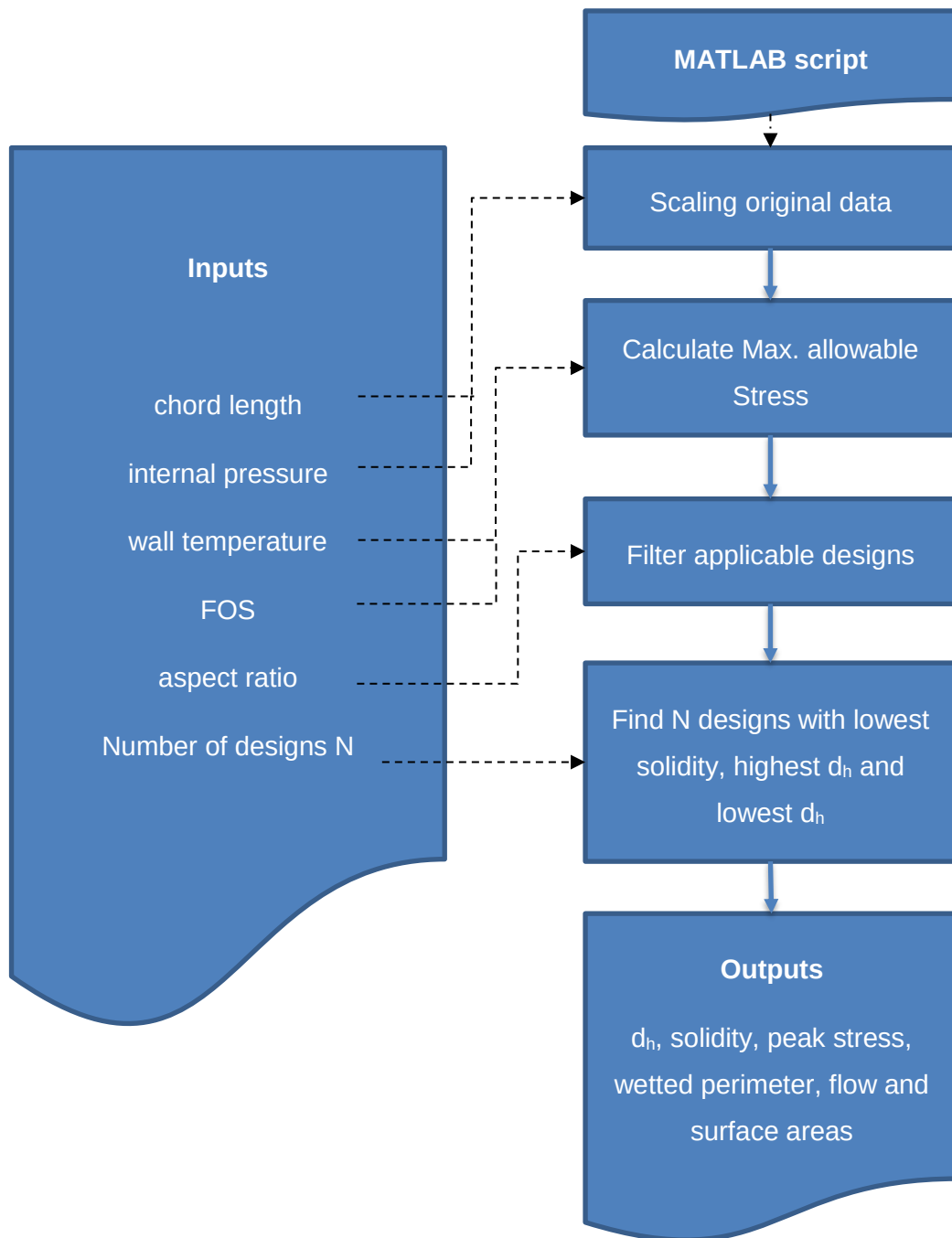


Figure 3-9: Scaling and filtering of internal tube design options

Finally, the script sorts the qualified designs according to solidity and hydraulic diameter. An output file is generated which includes the details of N designs for each of the three filter categories: the lowest solidity, the greatest hydraulic diameter and the smallest hydraulic diameter. As such, the extremes of the three design objectives low weight (solidity), high heat transfer coefficients (low d_h) and low internal pressure loss (high d_h)

are found. This approach enables the direct comparison of how all three objectives affect mission fuel burn if applied to an otherwise unchanged cycle design.

The output file includes the following parameters necessary for the performance evaluation of the pre-cooler and the MHX:

- Hydraulic diameter $d_{h,tube,C}$ of the flow channels inside a tube
- Wetted internal and external perimeter P_{cold}, P_{hot}
- Cross-sectional area of solid material A_{sol}
- Free-flow area inside the tube $A_{ff,C}$
- Tube wall thickness $tube_{th}$

Initial parametric studies concluded that the optimal internal tube geometry for any set of input parameters is always the design with the lowest solidity, hence the lowest weight. While the internal pressure losses and heat transfer coefficients are not negligible, their effect on mission fuel burn is overpowered by the weight concerns. An example of the importance of the individual effects and their trade-offs is presented in section 4.2.1. For the concept maturation scenarios presented thereafter, the lightest possible design for the given boundary conditions was always applied during the mission fuel burn calculations.

3.4.1.4 Heat Transfer

The heat transfer across the MHX and pre-cooler is calculated by calculating an overall heat transfer conductance for each pass [82], taking the flow regime and the specific heat exchanger geometry into account. When evaluating the performance of air-to-air or gas-to-air heat exchangers (intercoolers or recuperators), experience-based modelling approaches, such as the ϵ -NTU method [78], or even prescribing a heat exchanger effectiveness and pressure losses and designing the component afterward [85], may have proven acceptably accurate. However, as discussed in section 3.1, those modelling approaches are not detailed enough to assess the potential of a combined cycle engine using S-CO₂ as a working fluid, as the rapid changes in fluid properties near the critical point can disqualify the empirical data that directly correlate size and performance.

The heat transfer conductance UA consists of three main contributors if secondary effects such as longitudinal conductivity are neglected. The secondary effects can be reasonably neglected in case of cross-flow heat exchangers with large temperature differentials as in the present application.

The three contributors are:

- The convection from the hot fluid to the tube wall,
- The conduction through the tube wall,
- The convection from the tube wall to the cold fluid.

Equation 3-44 highlights how these individual terms are commonly accounted for [78], [82]. The respective heat transfer coefficients and the available heat transfer surface area dictate the hot and cold side convection. The conduction process is dependent on the tube wall thickness, the conductivity of the heat exchanger material and the average area normal to the direction of conduction. Ti-6Al-4V has been assumed as the material of choice for the heat exchangers, as it combines a relatively high yield strength, even in such high temperature environments, with approximately half the density of a comparable steel or nickel super-alloy. Unfortunately, the thermal conductivity of Ti-6Al-4V is quite low (approximately 6.7 W/(m*K) [111]). Therefore, minimising the wall thickness is not only important from a weight perspective, but also to maximise the overall heat transfer conductance of the heat exchanger. Empirical flow correlations as employed for the involute spiral heat exchanger and the recuperator are often based on Reynolds number. The Reynolds number for any duct or tube involved in these models is calculated using an adaptation of equation 3-43, where the respective hydraulic flow diameter is used as the critical dimension.

$$Re_x = \frac{\rho_x * c_x * d_{h,x}}{\mu_x} \quad 3-43$$

$$UA_{p,stack} = \left(\frac{1}{ht_{tube,m,H} * A_{h,stack,H}} + \frac{tube_{th}}{k_{mat} * \left(\frac{A_{h,stack,H} + A_{h,stack,C}}{2} \right)} + \frac{1}{ht_{tube,m,C} * A_{h,stack,C}} \right)^{-1} \quad 3-44$$

$$Q_{p,stack} = UA_{p,stack} * (T_{t,tube,m,H} - T_{t,tube,m,C}) \quad 3-45$$

$$Q_p = Q_{p,stack} * N_{stacks} \quad 3-46$$

$$h_{tube,out,H} = h_{tube,in,H} - \frac{Q_{p,stack}}{W_{stack,H}} \quad 3-47$$

$$h_{tube,out,C} = h_{tube,in,C} + \frac{Q_{p,stack}}{W_{stack,C}} \quad 3-48$$

The heat transfer in the involute spiral heat exchangers is assessed for each pass individually. The pre-cooler is further discretised into smaller tube sections to more accurately model the real gas effects of S-CO₂, as explained in section 3.4.1.6. The heat energy transferred in a single tube stack is calculated using equation 3-45, which in turn is used to calculate the total heat transferred in a pass. The specific enthalpy of the flow exiting a pass is calculated according to equation 3-47 for the hot side and equation 3-48 for the cold side. It is important to note, that equation 3-45 uses average hot and cold side temperatures across the pass for increased accuracy. Therefore, the exit temperature of the pass needs to be iterated by the model. Average flow conditions are also used to calculate the heat transfer coefficients on either side of the heat exchanger, necessitating the inclusion of those calculations in the iteration as well. Figure 3-10 outlines the iterative approach that might be implemented in a sequential code, if the inlet temperature and mass flow on either side of a pass are known. The mathematical definition of the problem in PROOSIS is too complicated to describe here, due to it using the non-sequential coding language EL and a Matrix-solver.

The iterative procedure of the model is even more complex than indicated in Figure 3-10, as the components are counter-crossflow heat exchangers. Hence, the hot side inlet conditions of pass one of the MHX are dependent on the heat transferred in pass two. However, the amount of heat transferred in pass two in turn is dependent on the cold side inlet conditions of pass two, which is dependent on the heat transfer in pass one. Moreover, all correlations to calculate heat transfer coefficients are dependent on the flow conditions, which in turn depend on the total working pressures. Consequently, the pressure loss calculations presented in section 3.4.1.5 also need to be included in the iterative processes.

The complexity of these calculations is a major contributor to the mathematical instabilities of the model. However, it is necessary to assess the performance of these heat exchangers as accurately as possible.

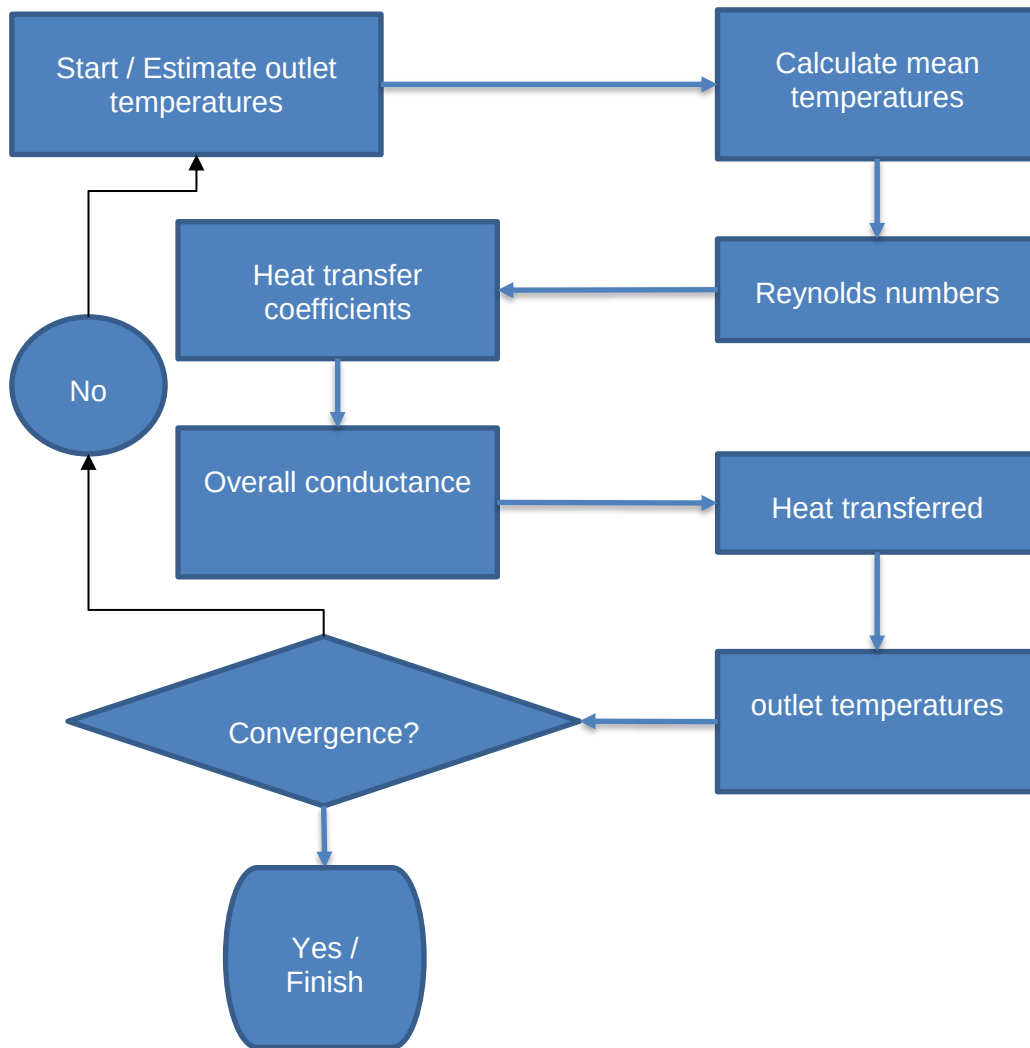


Figure 3-10: Simplified flow chart of heat transfer calculation

External Flow

The heat transfer coefficients of the external flow are calculated according to the correlations determined by Zhao using a combination CFD and component testing [88]. Zhao formulated different empirical correlations for the heat transfer and pressure losses in the external flow of the involute spiral heat exchanger. Each of the correlations is specific to a certain combination of axial and circumferential tube spacing. In this study it has been assumed that $m_{tube} = 2$ and $n_{tube} = 1$, because Zhao identified that arrangement as the best balance between external pressure losses and heat transfer. Equation 3-49 below gives the Colburn factor for the external flow [88]. The necessary Reynolds numbers are calculated using the average flow conditions and the free-flow area inside a pass. The correlation is applicable between $10,000 < Re_{tube,m,H} < 110,000$ according to Zhao et al. The maximum deviation relative to the CFD results within that

range was 0.9%. The Colburn factor is subsequently used to calculate the heat transfer coefficient of the external flow using equation 3-50.

$$j_{tube,m,H} = 0.003469 * e^{(-7.117*10^{-5}*Re_{tube,m,H})} + 0.003461 * e^{(-3.793*10^{-6}*Re_{tube,m,H})} \quad 3-49$$

$$ht_{tube,m,H} = \frac{j_{tube,m,H} * c_{p,tube,H} * W_{stack,H}}{Pr_{tube,m,C} * A_{ff,stack,H}} \quad 3-50$$

It is not necessary to include correlations for Reynolds numbers below 10,000, as the flow around the heat exchanger has led to Reynolds numbers greater than 5,000 at all times, meaning the flow is always turbulent and the deviation of the correlation from the expected results is relatively small.

Internal Flow

Zhao also derived correlations for the internal heat transfer coefficients in an involute spiral heat exchanger [128]. Those correlations were non-dimensionalised exclusively against Reynolds number, dismissing a possible change in the relationship between viscous and thermal diffusion rates (Prandtl number). It is reasonable to assume that the change in Prandtl number would be negligible in an air-to-air intercooler. However, CO₂ exhibits noticeably different Prandtl numbers, particularly close to the critical point. Figure 3-11 aims to highlight the importance of capturing the Prandtl number effects in an S-CO₂ heat exchanger.

The figure illustrates the difference in calculated heat transfer coefficients using different correlations. Zhao's correlation is used to calculate the Nusselt number of the flow directly. Afterward, the Nusselt number is used to calculate the transfer coefficient via equation 3-51.

$$ht_{tube,m,C} = \frac{Nu_{tube,m,C}}{d_{h,tube,C} * k_{tube,m,C}} \quad 3-51$$

Haaland's equation [125] is a commonly applied correlation to calculate the Moody friction factor for turbulent flow in pipes and tubes (see equation 3-52). It is also the correlation Zhao recommends in [28], as his own is project confidential. The unexpected increase in heat transfer coefficient for Reynolds numbers below 15,000 suggests that Zhao's confidential correlation is not applicable in that range.

$$f_{c,tube,m,C} = \left(-1.8 * \log \left(\frac{6.9}{Re_{tube,m,C}} + \left(\frac{\xi_{rel,tube}}{3.7} \right)^{1.11} \right) \right)^{-2} \quad 3-52$$

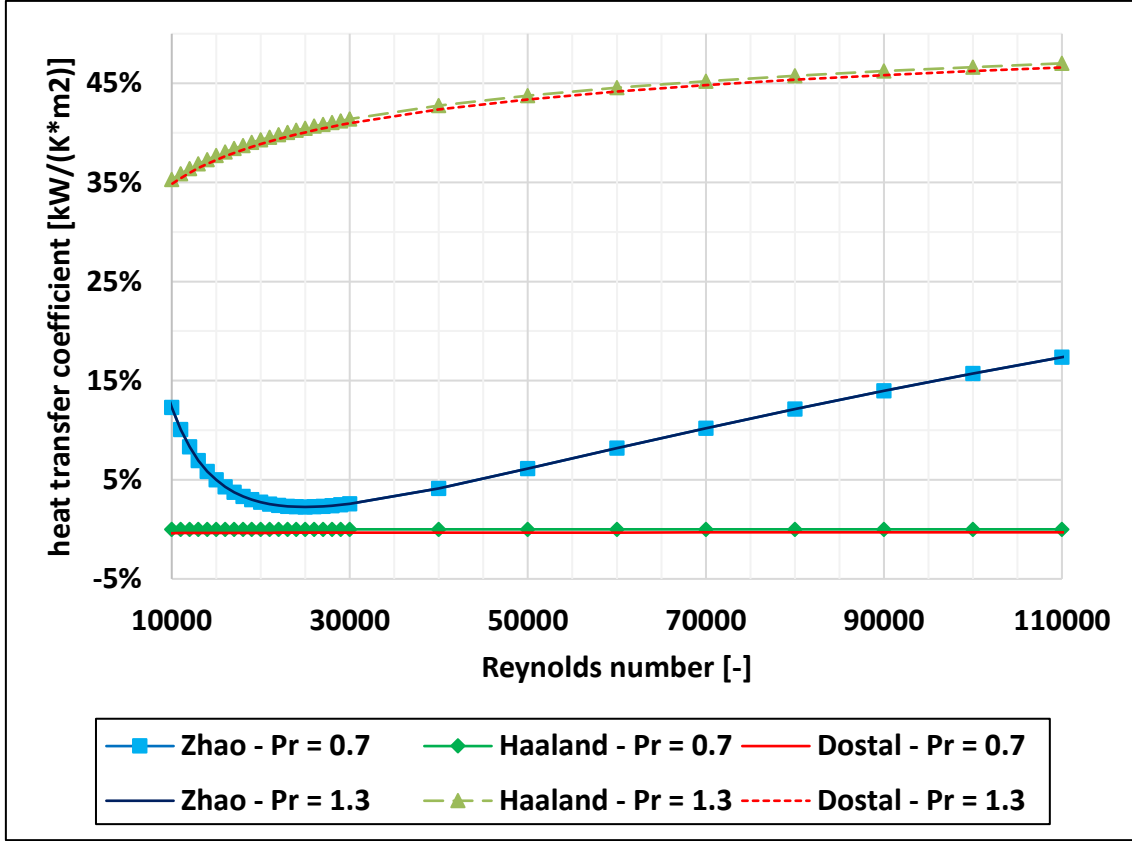


Figure 3-11: Relative difference in predicted heat transfer coefficients using different correlations

Finally, Dostál proposes the use of another correlation to estimate the Moody friction factors for fully turbulent flow inside small channels in equation 3-53. According to Dostál, this equation is valid for Prandtl numbers between 0.5 and 2,000 and Reynolds numbers up to $5 \cdot 10^6$, if used in conjunction with the simplified Gnielinski correlation [82] to calculate the Nusselt number (see equation 3-54). Gnielinski's correlation originally included another term accounting for the relationship between hydraulic diameter and tube length. That term can be neglected in this case, as $d_{h,tube,C} \ll L_{tubes}$.

$$f_{c,tube,C} = (1.8 * \log(Re_{tube,m,C}) - 1.5)^{-2} \quad 3-53$$

$$Nu_{tube,m,C} = \frac{\frac{f_{c,tube,C}}{8} (Re_{tube,m,C} - 1000) Pr_{tube,m,C}}{1 + 12.7 (Pr_{tube,m,C}^{2/3} - 1) * \sqrt{\frac{f_{c,tube,C}}{8}}} \quad 3-54$$

Additionally, Gnielinski's correlation is sometimes quoted with $\frac{f_{c,tube,C}}{2}$ terms and sometimes with $\frac{f_{c,tube,C}}{8}$ terms. Which of them is correct for a given model depends on the friction factor. If Moody friction factors are calculated as in this case, then the term $\frac{f_{c,tube,C}}{8}$ is correct. If Darcy friction factors are used, then $\frac{f_{c,tube,C}}{2}$ is correct. Gnielinski's equation is also used in conjunction with Haaland's correlation for Moody friction factors. The heat transfer coefficients are calculated using equation 3-51 in both cases, just as they would be when using Zhao's confidential correlation. One significant difference between Haaland's correlation and the other two, is the ability to take the effects of the surface finishing of the tubes into account. This can have a significant effect on the heat exchanger performance, especially with hydraulic diameters as small as those relevant here.

Figure 3-11 shows the results of six sets of calculations. All three correlations were applied to the same flow conditions twice, once with a Prandtl number of 0.7 (close to air) and once with a Prandtl number of approximately 1.3 (closer to MHX working conditions). The relative surface roughness of the tubes was set to zero in order to dismiss the surface finishing effect from Haaland's correlation. To better visualise the relative differences of the three approaches, the resulting heat transfer coefficients were normalised against the equivalent value from Haaland's equation for $Pr_{tube,m,C} = 0.7$ and a given Reynolds number (red line). All three correlations were applied assuming a tube hydraulic diameter of 0.7 mm and a thermal conductivity of 0.064, as those are representative of possible flow conditions inside the MHX. The graphs clearly show the effect the Prandtl number has on Haaland's (upper green line) and Dostál's correlation (upper red line), as the calculated heat transfer coefficients were between 35 and 47% greater in case of the higher Prandtl number. Zhao's correlation related Reynolds and Nusselt numbers directly without taking the Prandtl number into account at all, resulting in identical results. The two lines are therefore indistinguishable in the figure (two blue lines in the centre).

As Zhao's correlation was derived for application in an air-to-air heat exchanger, the relative difference to the $Pr_{tube,m,C} = 0.7$ cases of Haaland (bottom red baseline) and Dostál (bottom green line) are more representative of its feasibility. Zhao's approach predicted consistently higher heat transfer coefficients than the other two, with the biggest deviation being approximately 18%. It is fathomable that the involute shape of the tubes would lead to increased turbulence and boundary layer separation inside the

tubes and therefore enhanced heat transfer coefficients. However, the inability to take changes in Prandtl number into account disqualifies the method from application in an S-CO₂ heat exchanger.

In the end, Haaland's approach was chosen for the current models over Dostál's approach, as it can account for the surface roughness in the performance models, increasing the overall confidence in the models. Note that the difference in results between Haaland's equation and the commonly used Colebrook-White correlation [129] is minimal in the investigated range. However, the Colebrook-White equation is implicit, requiring additional iterations of the model that would have led to further instabilities.

All of the discussed models were derived to describe turbulent flow conditions. However, the S-CO₂ will likely flow with a low velocity inside the tubes, due to the extreme density of the fluid and the high number of tubes. As a result, it is possible that the flow inside those tubes may be in the transitional region between laminar and turbulent flow. The friction factor for laminar flow is commonly estimated using the Hagen-Poiseuille law (see equation 3-55). The Reynolds number at which the transition between laminar and fully developed turbulent flow begins is calculated using equation 3-56. Neither Haaland's equation nor the Hagen-Poiseuille law are applicable in the transition region and the friction factors calculated by the two differ greatly. An interpolation routine between the two equations is therefore necessary to avoid discrete jumps in predicted friction factors around Re_0 and risk convergence issues. Consequently, a third order polynomial is imposed in the models to automatically bridge the region between Re_0 and $Re = 5000$ (see Figure 3-12).

$$f_{c,tube,C} = \frac{64}{Re_{tube,m,C}} \quad 3-55$$

$$Re_0 = 754 * e^{\left(\frac{0.0065}{\xi_{rel,tube}}\right)} \quad 3-56$$

Note that the used form of the Gnielinski equation is only valid if $Re_{tube,m,C} > 2300$. Hence, a certain amount of modelling inaccuracy is introduced if the flow were to be fully laminar. This is not a problem for the presented models and results however, as the Reynolds number of the internal flow rarely drops below even 10,000. Gnielinski's equation is furthermore only intended for smooth duct simulations. The inaccuracy of the predictions is likely to be minimal in this case however, as the friction factor correlation takes the surface roughness into account, a method that was not originally anticipated

by Gnielinski. Furthermore, more elaborate Nusselt number correlations that are capable of accounting for the surface roughness, even when used with smooth-duct friction factor correlations, proved too complex for implementation in the current models if the necessary interpolation routines were to be employed as well.

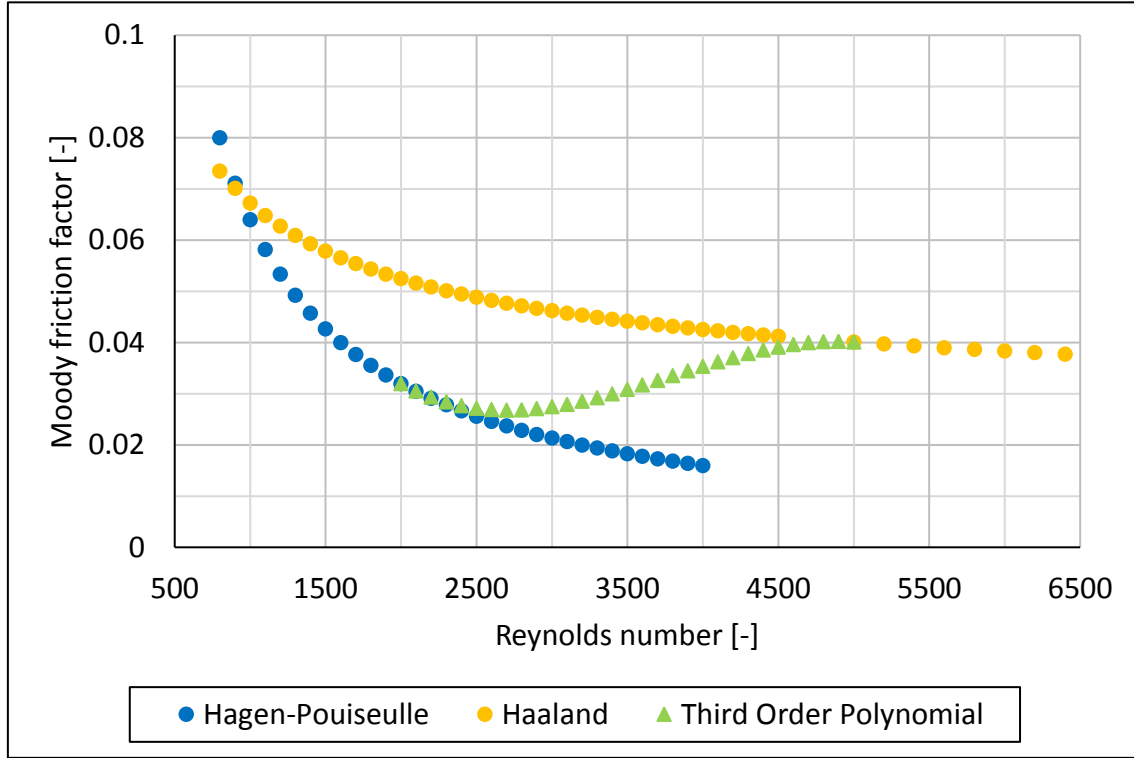


Figure 3-12: Example of interpolation between Haaland and Hagen-Poiseuille

The thermal conductivity of air is calculated using an empirical correlation (see equation 3-57) in all heat exchanger models, while the thermal conductivity of CO₂ is calculated using the specific heat capacity (see equation 3-58). The thermal conductivity of the S-CO₂ inside the heat exchangers is calculated based on the average flow conditions inside a node or pass, introducing a small additional uncertainty to the calculations of the heat transfer coefficients. The thermal conductivity of the S-CO₂ is calculated using equation 3-58 rather than with REFPROP, due to the structure of the fluid property model in PROOSIS.

$$k_{air} = (3E - 4 * T_{t,air}^{0.78}) \quad 3-57$$

$$k_{S-CO2} = \frac{\mu_{S-CO2} * c_{p,S-CO2}}{Pr_{S-CO2}} \quad 3-58$$

3.4.1.5 Pressure Drop

The pressure loss calculations for the involute spiral heat exchangers are implemented almost identically to Zhao's recommendations [88], except for the internal tube pressure losses and the inlet diffuser.

External Flow

The pressure loss and geometry estimation for the inlet diffuser containing the external flow has been explained in detail in section 3.4.1.2. Equation 3-59 below was derived by Zhao to approximate the friction factor across a stack of tubes. The friction factor is correlated against the inlet Reynolds number to the tube stack in this case, rather than the average value, which reduces the need for iteration. Afterward the friction factor is used to calculate the total pressure at the outlet of the pass according to equation 3-60.

$$f_{d,tube,H} = 0.01044 * e^{(-6.806*10^{-5}*Re_{tube,in,H})} + 0.008109 \quad 3-59$$

$$* e^{(-2.908*10^{-6}*Re_{tube,in,H})}$$

$$p_{t,tube,out,H} = p_{t,tube,in,H} - \frac{W_{stack,H}^2}{2 * A_{ff,stack,H}^2 * \rho_{tube,in,H}} \quad 3-60$$

$$* \left(f_{d,tube,H} * \frac{A_{ratio,ext} * \rho_{tube,in,H}}{\rho_{tube,m,H}} + (1 + \sigma_{HX,H}^2) \right)$$

$$* \left(\frac{\rho_{tube,in,H}}{\rho_{tube,out,H}} - 1 \right)$$

Internal Flow

The flow path of the S-CO₂ is split into several stages, with individual pressure loss characteristics for each sub-component of the heat exchanger. The inlet flow path of an S-CO₂ heat exchanger will likely be different to that presented by Zhao, but the sub-models were implemented nonetheless to compute a more realistic result. Whether the inlet and outlet manifolds will resemble Zhao's design or not, any flow disturbance and diffusion process will result in a total pressure loss. It is therefore deemed better to incorporate slightly inaccurate models and attain possibly pessimistic results, than to underestimate the pressure losses and over-predict whole engine performance.

The pressure loss factor for the initial part of the inflow duct is determined via equation 3-61. The equation is valid in the Reynolds number range of 250,000 to 1,400,000. The pressure loss factor is then used to estimate the static pressure in front of the tube entry according to equation 3-62.

$$K_{duct,in,C} = -1.626 * 10^9 * Re_{duct,in,C}^{-1.837} + 0.5513 \quad 3-61$$

$$p_{s,defl,in} = p_{s,duct,in,C} - (p_{t,duct,in,C} - p_{s,duct,in,C}) * K_{duct,in,C} \quad 3-62$$

The pressure loss due to the sudden flow contraction at the tube inlet is calculated using the approach presented by White [104]. Depending on the contraction rate (equation 3-63), the pressure loss factor is estimated according to equation 3-64 or equation 3-65. Either way, the pressure loss is subsequently estimated using equation 3-66.

$$A_{ratio,tube,in} = \frac{A_{ff,stack,C}}{A_{fr,C}} \quad 3-63$$

If $A_{ratio,tube,in} < 0.5776$:

$$K_{SC} = 0.42 * (1 - A_{ratio,tube,in}) \quad 3-64$$

Else:

$$K_{SC} = (1 - A_{ratio,tube,in})^2 \quad 3-65$$

$$p_{t,tube,in,C,p1} = p_{t,defl,out} - K_{SC} * \frac{\rho_{defl,out}}{2} * c_{defl,out}^2 \quad 3-66$$

The equations above are re-applied any time the flow enters another set of tubes, hence following any cross-over duct. The friction factor inside the tubes themselves is once again calculated using Haaland's correlation as recommended by Zhao [28], [88]. Equation 3-68 shows how the friction factor is then used to estimate the total pressure loss using the Darcy-Weisbach equation [104].

$$f_{d,tube,C} = \left(-1.8 * \log \left(\frac{6.9}{Re_{tube,in,C}} + \left(\frac{\xi_{rel,tube}}{3.7} \right)^{1.11} \right) \right)^{-2} \quad 3-67$$

$$p_{t,tube,out,C} = p_{t,tube,in,C} - f_{d,tube,C} * \frac{L_{tubes}}{d_{h,tube,C}} * \frac{\rho_{tube,in,C}}{2} * c_{tube,in,C}^2 \quad 3-68$$

The sudden flow expansion at the tube outlet into the cross-over duct, or the outlet manifold depending on the pass, provokes a further reduction in total pressure. The corresponding loss factor is calculated according to equation 3-69, with the total pressure loss being calculated using equation 3-70.

$$K_{SE} = (1 - A_{ratio,tube,in})^2 \quad 3-69$$

$$p_{t,cross,in} = p_{t,tube,out,C} - K_{SC} * \frac{\rho_{tube,out,C}}{2} * c_{tube,out,C}^2 \quad 3-70$$

The loss factor due to the cross-over duct geometry was a further research result of Zhao and is characterised by equation 3-71.

$$K_{cross} = -1.17 * 10^{-6} * Re_{cross} + 4.227 \quad 3-71$$

$$p_{s,cross,out} = p_{s,cross,in} - (p_{t,cross,in} - p_{s,cross,in}) * K_{cross} \quad 3-72$$

The outlet manifold was optimised by Zhao, resulting in the loss factor correlation of equation 3-73. The correlation is deemed reliable in a Reynolds number range of 1,500,000 to 5,500,000, with the outlet Reynolds number being used. The incurred pressure losses are calculated using equation 3-74 thereafter.

$$K_{duct,out,C,out} = 1.939 * 10^{11} * Re_{duct,out,C,out}^{-2.1} + 0.08107 \quad 3-73$$

$$p_{t,duct,out,C,out} = p_{t,duct,out,C,in} - K_{duct,out,C,out} * (p_{t,duct,out,C,in} - p_{s,duct,out,C,in}) \quad 3-74$$

It is important to note, that some of the pressure loss calculations use simplified total-to-static pressure conversions for incompressible flow. While this is not a problem in the current application, due to the very low flow velocities, this is an important concern if the models were adapted for air-to-air heat exchangers ever again. In that case a radical design may push the flow beyond Mach 0.3, the commonly accepted threshold for incompressible flow simulations, reducing the confidence in the obtained results.

3.4.1.6 Discretisation of Pre-cooler

The closer the compressor inlet conditions are to the critical point of the S-CO₂, the lower the power consumption of the compressor and therefore the higher the net power output and the thermal efficiency of the cycle. Unfortunately, the proximity to the critical point also means, that the S-CO₂ will exhibit dramatic variations in fluid properties while passing through the pre-cooler. Beside the implications this has on the achievable temperature change in a fixed size pre-cooler, as discussed in section 3.1, this also means that estimating the pre-cooler's performance as accurately as possible is more difficult. The variation in specific heat capacity inside a pass of the MHX is relatively small. It is therefore possible to calculate the heat transfer using average flow conditions for the hot and cold sides of a pass without introducing a considerable error. However, depending on the proximity of the pre-cooler outlet conditions to the critical point, the

specific heat capacity can vary by a factor of 10 or more within a few centimetres of tube length. As a result, any attempt to predict the heat transfer in one pass of the pre-cooler using average flow conditions will lead to a significant overestimation of the performance.

This phenomenon is similar to the pinch point problem in an S-CO₂-to-S-CO₂ recuperator as described by Dostál. The solution lies in adapting a similar modelling approach as for the recuperator. The tubes of each pass of the pre-cooler need to be split along their length into multiple discrete sections and the performance of each of those sections needs to be determined individually.

The pre-cooler model therefore has an additional input parameter: the number of nodes each pass is discretised into. For high effectiveness (>90%) recuperators, 40 nodes has become a commonly applied figure. However, the pre-cooler is neither a high effectiveness heat exchanger, due to the size constraints, nor does the current model successfully converge with 40 nodes.

The heat transfer and pressure loss correlations of each individual node are calculated using generally the same set of equations presented in sections 3.4.1.4 and 3.4.1.5. The tubes are discretised so that each tube section spans the same portion of the total annulus height. This means that the outer tube sections are longer than those toward the hub of the heat exchanger. It would have also been possible to split the tubes into equally long tube segments. However, each tube segment will interact with a different amount of coolant mass flow. The calculation of the mass flows for equally long tube sections would have been more intricate and the model therefore less stable than the calculations for different tube lengths covering equal annulus height segments.

Table 3-1: Pre-cooler performance sensitivity to number of nodes

N_{nodes} [-]	S-CO ₂ inlet temperature [K]	S-CO ₂ inlet pressure [MPa]	Number of algebraics [-]
2	463.5	8.416	18
5	443.5	8.410	35
8	441.5	8.409	53
10	441.1	8.409	64

The performance of all nodes is obviously dependent on each other, resulting in a dramatic increase in the amount of iterations necessary to reach convergence. The

number of algebraics necessary to solve the mathematical problem furthermore increases drastically as can be seen in Table 3-1.

Consequently, the model's stability decreases considerably when the number of nodes is increased and a trade-off between modelling accuracy and computational resources has to be accepted. A sensitivity study with fixed S-CO₂ outlet conditions, coolant inlet conditions and a fixed geometry was conducted. Only the pre-cooler was simulated and not the whole engine, in order to reduce model instabilities. The calculated inlet temperature reduced by 20 K, when increasing the number of nodes from two to five, which would have a dramatic effect on the estimated cycle performance. The calculated inlet pressure also slightly decreased, though that variation is considerably smaller and less influential on whole engine performance. Discretising the pre-cooler into more than eight nodes per pass affected the predicted S-CO₂ inlet temperature by less than 0.5 K (see Table 3-1) and led to no significant change in inlet pressure. Such a temperature variation is likely within the error margin of the correlation-based models. Hence, any further discretisation will only lead to increased computational effort without improving the accuracy of the results. In addition, the model no longer converged fully, but always returned a warning that at least one residual still exceeded the convergence criterion. The number of nodes was kept at five during the combined cycle studies presented in section 4. The increased inaccuracy of the performance prediction of the pre-cooler had to be accepted in this case, as the combined cycle model would otherwise not converge for over 90% of the intended designs, making it impossible to adequately assess and improve the whole engine performance.

3.4.1.7 Pre-cooler duct nozzle

After exiting the pre-cooler, the heated bypass air can either be mixed with the remainder of the bypass flow or it can be discharged via a third nozzle. Previous research projects such as NEWAC investigated the potential SFC benefit that could be achieved if the outer bypass and the intercooler exit streams are ideally mixed. Most of the time the maximum benefit of that mixing process was of the order of only 1 % SFC [126], [27], as the intercooler (or pre-cooler) exit air temperature is only slightly warmer than the outer bypass flow. On the other hand, having a mixed bypass nozzle would likely require a significantly longer nacelle and a mixer, both of which will add weight that may completely offset the fuel improvements of the mixing process itself. Additionally, designing the engine for the right pre-cooler exit conditions to enable mixing, would put further strain on an already complex design process, which would make model convergence even

harder to achieve. Due to these reasons, the engines simulated in this work all feature separate bypass nozzles for the inner and outer bypass flows. The gross thrust produced by the inner bypass is simply added to the total thrust balance. The nozzle itself is assumed to share the characteristics and technology of the outer bypass nozzle. Hence, the same thrust and discharge coefficients are imposed.

3.4.1.8 Input parameters for performance calculations

The pre-cooler and the MHX both require a large number of geometric input parameters in order to fully define their preliminary design and subsequently enable the calculation of their performance. Table 3-2 lists them all for the MHX, together with their respective theoretical and applied limits for the parametric studies conducted to mature the concept and estimate the feasibility. For many inputs, the range of actually applied inputs is considerably smaller than the theoretically possible. For some this is due to convergence issues and for some, the practical limits were derived from the results of previous studies or manufacturing concerns.

Table 3-2: Inputs and their limits of involute spiral heat exchanger (MHX example)

Parameter	Theoretical limits	Applied limits	Comment
$a_{tube,H}$ [m]	N/A	0.005 – 0.02	5 mm chord length thought to be manufacturing limit
AR [-]	6 – 8	8	Flow correlations for other aspect ratios not yet determined
A_{sol} [m ²]	N/A	N/A	Directly scaled from Rengel's data and dependent on chord length
$A_{tube,C}$ [m ²]	N/A	N/A	Directly scaled from Rengel's data and dependent on chord length
$d_{h,tube,C}$ [m]	N/A	N/A	Directly scaled from Rengel's data and dependent on chord length
k_{mat} [W/(m*K)]	0 – 1,000	6.7	Ti-6Al-4V assumed [111]
L_{diff} [-]	0 – 5	1.2	Approximately possible length estimated from TF2050 cross-section
$M_{fr,H}$ [-]	0 – 0.99	0.08 – 0.15	Mach numbers below minimum lead to excessive size, above maximum to excessive pressure loss

Parameter	Theoretical limits	Applied limits	Comment
N_{col} [-]	0.5 – 30	5 – 8	Convergence was impossible to reach outside the applied limits
$P_{tube,C}$ [m]	N/A	N/A	Directly scaled from Rengel's data and dependent on chord length
$P_{tube,H}$ [m]	N/A	N/A	Directly scaled from Rengel's data and dependent on chord length
$tube_{th}$ [m]	N/A	N/A	Directly scaled from Rengel's data and dependent on chord length
x [-]	0 – 2.0	0.75 – 1.0	Minimum of 0.75 applied to leave room for manifolds, values above maximum uneconomical
x_{inv} [-]	0 – 1.0	0.9 – 1.0	If tubes start at too great an angle, the cross-section at hub is no longer close to elliptical and the calculation of N_{rows} is wrong
$\xi_{abs,diff}$ [m]	0 – 0.001	1E-5	Achievable value for drawn tubes. Values for 3D-printing need development
$\xi_{abs,tube}$ [m]	0 – 0.001	1E-5	Achievable for sheet metal today
ρ_{mat} [kg/m ³]	0 – 20,000	4,330	Ti-6Al-4V assumed [111]

As discussed in section 3.4.1.1, the x parameter of the MHX does not exist for the pre-cooler. Instead, the initial inner annulus diameter at which the pre-cooler tubes start is imposed directly. A fixed value of 0.83 m has been applied during the simulations for $d_{i,fr,C,init}$ in the pre-cooler. That value has been extrapolated from the cross-section of the reference engine presented in section 2.1 and confirmed as feasible in personal communication in the ULTIMATE project [127]. A sensitivity study of the current models to this input is not necessary, as any reduction in that parameter will enable a greater amount of bypass air to be used as coolant for the bottoming cycle without affecting the remaining bypass stream. The performance of the engine will therefore always improve if the diameter is lowered. Whether or not such a reduction is feasible would be the outcome of a detailed component and whole engine design, should the concept prove to be feasible. The same inner diameter boundary has been applied for the CCTF2050 and

the CCTF2050_ITR engines, although the reduced core size of the CCTF2050_ITR should enable a further reduction of that dimension. This would lead to an improvement of the mission fuel burn performance of the concept.

3.4.2 Printed Circuit Recuperator

The S-CO₂-to-S-CO₂ recuperator model is primarily based on the printed circuit heat exchanger (PCHE) model presented by Dostál [55] and manufactured by Heatric [130], [131]. Figure 3-13 shows a cut-away of the PCHE recuperator as it may be determined by the performance and preliminary design models presented in the following sections. A number of modelling changes and additions have been made to Dostál's model and will be highlighted where appropriate. Section 3.4.2.1 explains the geometric definition of the recuperator model. Section 3.4.2.2 explains the heat transfer calculations, while section 3.4.2.3 outlines the pressure drop modelling. Finally, section 3.4.2.4 highlights the modelling approach behind axially discretising the recuperator into multiple sections for increased accuracy.

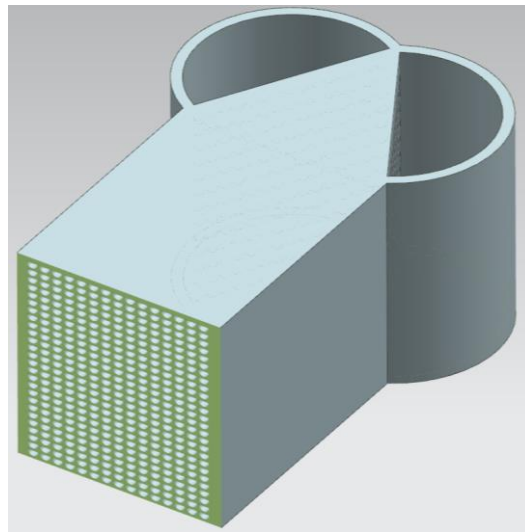


Figure 3-13: CAD model of symmetric PCHE cut-away with manifolds

The models for the hot and cold fluids are identical, as both are supercritical carbon dioxide. Indices are therefore only introduced to the equations and symbols of the following sections, if there is a mathematical or physical reason, for example for the heat balance. Otherwise, the indices for the fluid stream are set to X and it may be assumed that the equivalent equations exist for either side.

3.4.2.1 Parametric Design and Weight Estimation

The recuperator is modelled as a counter-flow PCHE with alternating hot and cold flow plates stacked on top of each other and cross-flow inlet and outlet manifolds. The inlet and outlet manifolds are based on the shape of an isosceles right triangle. Their design is not optimised for the application, but only added to improve the weight estimation of the models. Figure 3-14 a) illustrates an approximate flow path of the hot and cold S-CO₂ streams passing through the recuperator. The high-temperature, low-pressure turbine exit stream enters the component at the top and exits at the bottom of the schematic. The low-temperature, high-pressure compressor exit flow enters the manifold on the left and exits on the right-hand side of the illustration. The figure furthermore explains the three main dimensions of the heat exchanger matrix in the middle of the component, the main body: length, width and height. Those are three of the main input parameters to the PCHE recuperator model. The recuperator model assumes straight flow channels. Wavy or zig-zag channels would also be possible and may improve heat transfer, but increase pressure losses [132], [133], which is why they are not included here. A trade-off study of the ideal channel shape was beyond the scope of this work.

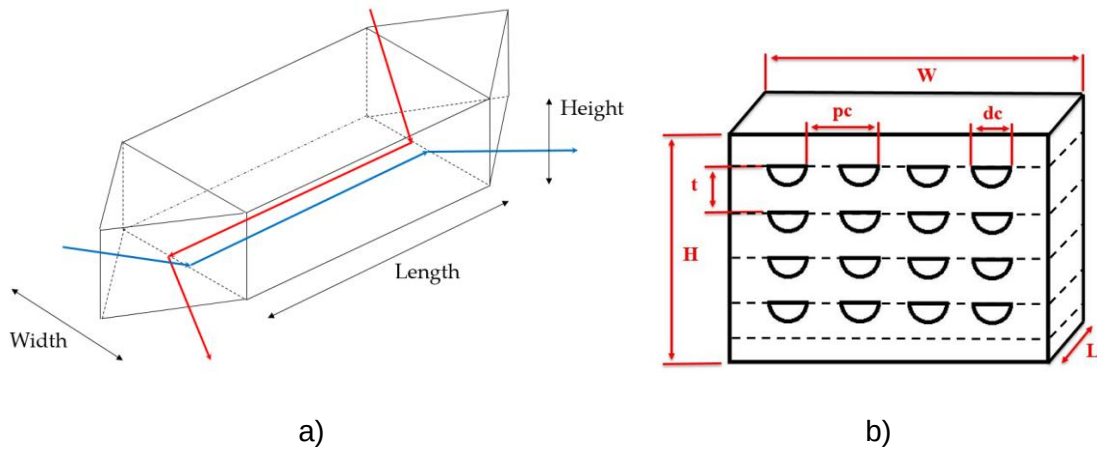


Figure 3-14: Schematic of PCHE structure

Figure 3-14 b) illustrates the remaining design inputs of the model: the diameter of the etched channels d_c , the pitch between channels on one flow plate p_c and the plate thickness t . Theoretically, each of these three must be imposed twice during the performance simulations, once for the hot and once for the cold side. In practice, both the channel pitch and the plate thickness are directly related to the channel diameter via equations 3-75 and 3-76, reducing the number of design parameters. The two equations imply constant wall thicknesses to be applicable for all investigated designs. This limits

the range of feasible channel diameters, if the mechanical integrity of the recuperator should not be compromised.

Heatric presented a simplified mechanical integrity model that may be implemented during preliminary design studies of PCHEs [131]. Unfortunately, implementing that approach in the current models proved too complex for the combined cycle studies, but may be pursued in further work. The working conditions of the two streams are very different, resulting in different flow areas for a given Mach number and different specific heat capacities. This is why the channel dimensions of the two flows need to be independent in the model, to allow for separate flow path optimisation for the two streams.

$$p_{c,X} = d_{c,X} + 0.0004 \quad 3-75$$

$$t_X = \frac{d_{c,X}}{2} + 0.0005 \quad 3-76$$

The hydraulic diameter of the channels is calculated according to equation 3-77.

$$d_{h,X} = \frac{\pi * d_{c,X}^2}{2 * \left(\frac{\pi * d_{c,X}}{2} + d_{c,X} \right)} \quad 3-77$$

The height and width design inputs are only initial guesses. The real width and height of the main body is calculated, so the number of plates n_{pl} and the number of channels per plate n_c are integers. The final design will always be slightly smaller than the available space imposed by the input height and width, because the plate and channel numbers are rounded down. Equations 3-78 to 3-82 outline the approach.

$$n_{c,C} = \text{floor} \left(\frac{\text{Width}}{p_{c,C}} - 2 \right) \quad 3-78$$

$$\text{Width}_C = (n_{c,C} + 2) * p_{c,C} \quad 3-79$$

$$n_{c,H} = \text{floor} \left(\frac{\text{Width}}{p_{c,H}} - 2 \right) \quad 3-80$$

$$\text{Width}_H = (n_{c,H} + 2) * p_{c,H} \quad 3-81$$

$$\text{Width}_{new} = \max(\text{Width}_C, \text{Width}_H) \quad 3-82$$

The number of channels that fit on one plate is reduced by two from the theoretically possible value, to allow for thicker bezels at the sides of the plates. Those bezels are necessary for the diffusion bonding process and improve the mechanical integrity. The width necessary to fit the flow channels is calculated separately for the hot and cold side and the greater of the two values is chosen as the final module width as per equation 3-82. The number of flow plates is calculated according to equation 3-83 and describes the number of hot and cold plates respectively. The total number of flow plates is therefore twice as large. The number of plates is subsequently used to calculate the new module height and the total number of hot and cold flow channels according to equations 3-84 through 3-85.

$$n_{pl} = \text{floor}\left(\frac{H}{t_c + t_H}\right) \quad 3-83$$

$$H_{new} = n_{pl} * (t_c + t_H) \quad 3-84$$

$$n_{c,X,ov} = n_{c,X} * n_{pl} \quad 3-85$$

The heat transfer surface area per channel for either fluid is calculated according to equation 3-86. The total heat transfer surface area of all channels for either flow is calculated according to equation 3-87. The free-flow area per channel and for the whole flow is calculated according to equations 3-88 and 3-89 respectively.

$$A_{h,X} = \text{Length} * \left(\frac{\pi * d_{c,X}}{2} + d_{c,X}\right) \quad 3-86$$

$$A_{h,X,total} = A_{h,X} * n_{c,X,ov} \quad 3-87$$

$$A_{ff,c,X} = \frac{\pi * d_{c,X}^2}{8} \quad 3-88$$

$$A_{ff,X} = A_{ff,c,X} * n_{c,X,ov} \quad 3-89$$

The volume of solid material in the main body of the PCHE is calculated according to equation 3-90. The ratio of free flow area to solid material is calculated as shown in equation 3-91.

$$V_{body} = \text{Length} * \left(\text{Width}_{new} * H_{new} - (A_{ff,c} + A_{ff,H})\right) \quad 3-90$$

$$A_{ratio,head} = 1 - \frac{A_{ff,c} + A_{ff,H}}{\text{Width}_{new} * \text{Height}_{new}} \quad 3-91$$

The prism of the manifolds is based on an isosceles right triangle. Hence, the width of the frontal areas of the inlet and outlet manifolds is calculated according to equation 3-92. The wall thickness of the manifolds is calculated according to equation 3-93, assuming a factor of safety (FOS) of two and a yield strength of $R_{0,2} = 900 \text{ MPa}$ for the material.

$$Width_{head} = \frac{Width_{new}}{\sqrt{2}} \quad 3-92$$

$$head_{th,X} = \frac{p_{t,in,X} * Width_{head} * FOS}{2 * R_{0,2}} \quad 3-93$$

$$V_{head} = \frac{\sqrt{3}}{2} * Height_{new} * A_{ratio,head} * Width_{head}^2 + \frac{\pi}{4} * Height_{new} * \left((Width_{head} + 2 * head_{th,H})^2 - Width_{head}^2 \right) + \left((Width_{head} + 2 * head_{th,C})^2 - Width_{head}^2 \right) \quad 3-94$$

$$M_{rec} = \rho_{mat} * (V_{body} + V_{head}) \quad 3-95$$

The volume of solid material in the inlet and outlet manifolds including the crossover sections, the semi-circular manifolds and a solid base plate at the bottom of the semi-circular ducts is calculated according to equation 3-94. Finally, the weight of the recuperator is calculated according to equation 3-95.

It is important to note, that the heat transfer and the pressure losses occurring in the crossover flow section of the manifolds are not accounted for in the recuperator model. The discretised recuperator model already introduced a large number of additional algebraics to the whole engine model, leading to severe convergence problems. Implementing the heat transfer and pressure loss calculations in the inlet and outlet manifolds added too much additional complexity, making it impossible to reach convergence of the combined cycle engine. Further pressure losses arise in the manifolds when the flow enters or exits the heat exchanger matrix, due to sudden contraction or expansion of the flow area. Those pressure losses are accounted for in the model and the calculations are explained in section 3.4.2.3.

Neglecting the heat transfer and pressure losses in the cross-flow section of the inlet and outlet manifolds will lead to a small miscalculation of the recuperator's performance, though the effects cancel each other out to a large part. The additional heat transfer

occurring in the cross-flow sections means that the main body of the heat exchanger can be shorter for a targeted heat duty. This length reduction in the main body in turn will lead to a reduction in total pressure losses, offsetting a considerable portion of the additional pressure losses from the cross-flow section. The pressure losses and the length of the flow path are approximately linearly correlated. The temperature on the other hand, changes asymptotically with channel length, giving diminishing returns for any additional channel length. If the performance of the inlet manifolds is therefore fully accounted for and the heat exchanger is designed for the same heat duty, it may be expected that the pressure drop will be increased relative to that calculated by the simpler model neglecting the manifold effects.

However, this relation is dependent on the heat exchanger effectiveness. The higher the effectiveness, the lower the additional heat that is transferred via the additional heat transfer surface area of the cross-flow sections. The additional pressure losses will increase noticeably. In case of a low heat exchanger effectiveness ($<75\%$), the heat transfer will also improve significantly due to the manifolds, meaning a larger part of the main body can be dropped, significantly reducing pressure losses as well. In that case the pressure drop across the recuperator will only be slightly higher, or maybe even lower, than the simplified model may predict without taking the manifolds into account. The weight reduction that may be possible due to the manifold performance is not accounted for either way. The simple model therefore delivers a conservative estimate of the recuperator's weight. Dostál does not include any manifold calculations, weight or otherwise, in his research [55], as he is looking at very large, high effectiveness heat exchangers for stationary application, where weight and size are secondary concerns.

A case study of an isolated high effectiveness recuperator ($\sim 96\%$) was conducted to illustrate the implications of the model simplifications and show that they do not lead to an overestimation of the recuperator performance. The main body of the recuperator that does not account for the manifold effects had the following specifications:

- Length = 0.7 m
- Width = 0.3 m
- Height = 0.2 m
- $d_{c,C} = 0.001$ m
- $d_{c,H} = 0.0015$ m

The results (see Table 3-3) show that the main body of the recuperator could be shortened by approximately 31%, if the heat transfer in the manifolds was taken into account and the overall temperature differential on both sides was kept constant. The pressure losses on either side of the heat exchanger increased by approximately 10 kPa, signifying a relative increase of around 8% in both cases. However, a pressure drop of 10 kPa only has a second order effect on the cycle performance, due to the extreme working pressures of the S-CO₂ cycle.

Table 3-3: Approximation of manifold's effect on recuperator performance

	Neglecting manifold effects	Including Manifold effects	Relative change
ΔT_{hot} [K]	42	42	-
ΔT_{cold} [K]	27	27	-
ΔP_{hot} [kPa]	124	134	+8.4%
ΔP_{cold} [kPa]	123	133	+7.8%
ΔP_{hot} [%]	0.37	0.40	+0.03
ΔP_{cold} [%]	1.44	1.56	+0.12
M_{rec} [kg]	149	112	-25.1%

Overall, the hot-side total pressure loss increased by only 0.03 percentage points, while the cold-side pressure loss increased by 0.12 percentage points. Both signify small changes that may be within the overall uncertainty of the model. The 25% reduction in recuperator weight would likely have a bigger effect on mission fuel burn than the pressure losses. However, the simplistic design of the manifolds and the calculation of the wall thickness of those manifolds may underestimate the component weight. Overall, the omission of the thermodynamic modelling in the inlet and outlet manifolds is likely to lead to a conservative evaluation of the performance of the recuperator. Given the preliminary status of the overall cycle design, it is always advisable to include a more conservative approach to leave room for detailed design implications.

3.4.2.2 Heat Transfer

The heat transfer model implemented for the main heat exchanger matrix is similar to that described by Dostál [55], with some slight changes particularly regarding the transitional flow regime. The hydraulic diameter for either fluid is calculated according to equation 3-77 in section 3.4.2.1. The hot and cold fluid is supercritical CO₂ in both cases.

Hence, the difference between the hot and cold side equations are only pointed out where necessary, otherwise the index X highlights the validity for both fluids.

The Moody friction factor for the laminar flow regime ($Re < 2000$) is calculated according to the Hagen-Poiseuille law shown in equation 3-97. The Moody friction factor of a fully developed turbulent flow is calculated according to equation 3-98, as is recommended by Dostál. The friction factors calculated for transitional flow however are not calculated following Dostál's model, as that approach potentially results in sudden and significant changes in friction factors for minor changes in Reynolds number. Instead, a modelling approach similar to that presented in section 3.4.1.4 is implemented, where a third order polynomial interpolation between the Hagen-Poiseuille law and equation 3-98 is applied. The third order interpolation is applied in the range of $2000 < Re < 5000$. Again, the Reynolds number is determined using the channel's hydraulic diameter as the critical dimension (see equation 3-96).

$$Re_{c,X} = \frac{\rho_X * c_X * d_{h,X}}{\mu_X} \quad 3-96$$

$$f_{c,c,X} = \frac{64}{Re_{c,m,X}} \quad 3-97$$

$$f_{c,c,X} = (1.8 * \log(Re_{c,m,X}) - 1.5)^{-2} \quad 3-98$$

The Reynolds numbers used in these correlations are once again determined using the average flow conditions in the investigated section of the channel. The calculated Moody friction factors are subsequently applied to the simplified Gnielinski equation (equation 3-99) to determine the corresponding Nusselt number. The Gnielinski equation can be simplified relative to its original definition [82], because the channel length is much greater than its hydraulic diameter, effectively eliminating the effect of the additional term. Equation 3-100 then allows for the calculation of the heat transfer coefficient.

$$Nu_{c,m,X} = \frac{\frac{f_{c,c,X}}{8} (Re_{c,m,X} - 1000) Pr_{c,m,X}}{1 + 12.7 \left(Pr_{c,m,X}^{\frac{2}{3}} - 1 \right) * \sqrt{\frac{f_{c,c,X}}{8}}} \quad 3-99$$

$$ht_{c,m,X} = \frac{Nu_{c,m,X}}{d_{h,X} * k_{c,m,X}} \quad 3-100$$

The overall heat transfer conductance is subsequently determined via equation 3-101. The effect of thermal conductance through the material is approximated using the

average hot and cold side heat transfer surface areas and the average wall thickness between the hot and cold flow channels. The overall heat transfer conductance is used to determine the heat transfer via equation 3-102.

$$UA = \left(\frac{1}{ht_{c,m,C} * A_{h,C,total}} + \frac{(t_C - d_{c,C}) + (t_H - d_{c,H})}{k_{mat} * (A_{h,C,total} + A_{h,C,total})} + \frac{1}{ht_{c,m,H} * A_{h,H,total}} \right) \quad 3-101$$

$$Q = UA * (T_{t,m,H} - T_{t,m,C}) \quad 3-102$$

$$h_{ce,X} = h_{he,X} - \frac{Q}{W_X} \quad 3-103$$

The outlet enthalpy of a node (see section 3.4.2.4), or the whole recuperator, is calculated using equation 3-103 for the either side of the heat exchanger. The same equation can be applied, with different indices of course, because the nomenclature of the PROOSIS model is defined in terms of the hot and cold end of the component (or the node), rather than inlet and outlet flow.

The heat transfer in the crossover sections of the inlet and outlet manifolds is not accounted for, as discussed in section 3.4.2.1.

3.4.2.3 Pressure Drop

The pressure loss model of the recuperator is identical for both flows and accounts for three effects:

- The sudden flow path contraction from the inlet manifold into the heat exchanger matrix
- The friction inside the channels of the main body of the recuperator
- The sudden flow path expansion from the heat exchanger matrix into the outlet manifold

The pressure losses inside the crossover flow section of the manifolds are not accounted for, as discussed in section 3.4.2.1.

The pressure losses induced by the sudden flow expansion and contraction of the flow path in the manifolds are determined using equation 3-104. The pressure loss factors for the two cases are set to $K_{SC} = 0.5$ and $K_{SE} = 1$. These values were implemented by

Dostál and are consistent with the worst case scenarios for flow contraction and expansion with sharp, unrefined edges presented by White [104].

$$\Delta p_t = K_{SC/SE} * \frac{\rho_{in} * c_{in}^2}{2} \quad 3-104$$

Rounding the edges of the channels can significantly improve the pressure losses due to the sudden flow contraction and expansion, but may be difficult to manufacture reliably, given the small diameter and large number of channels in the recuperator. Furthermore, these pressure losses are at least an order of magnitude smaller than the friction induced losses inside the flow channels in this application, making their minimisation a second order priority.

The model to calculate the friction factors of the flow inside the recuperator channels is changed relative to that presented by Dostál to improve mathematical stability and convergence. The laminar flow region is represented by the Hagen-Poiseuille law again (see equation 3-105), while the fully developed turbulent flow is characterised by Haaland's friction factor correlation (see equation 3-106). Dostál proposes the use of the Colebrook-White equation to estimate the Moody friction factors of the fully developed turbulent flow. However, the Colebrook-White equation is an implicit function that causes additional iterations of the model, reducing stability. The explicit formula of Haaland (see equation 3-106) has been shown to result in only minor deviations from the results of Colebrook-White when applied to S-CO₂ in the relevant region [134], making it an adequate replacement. The friction factors for transitional flow are estimated according to a third order polynomial fit between the Hagen-Poiseuille law and Haaland's equation. Haaland's equation is employed for Reynolds numbers greater than 5,000. The Reynolds number Re_0 as the departure point from the Hagen-Poiseuille law depends on the relative surface roughness of the channel and is calculated according to equation 3-107.

$$f_{d,c,X} = \frac{64}{Re_{c,m,X}} \quad 3-105$$

$$f_{d,c,X} = \left(-1.8 * \log \left(\frac{6.9}{Re_{c,m,X}} + \left(\frac{\xi_{rel,c,X}}{3.7} \right)^{1.11} \right) \right)^{-2} \quad 3-106$$

$$Re_0 = 754 * e^{\left(\frac{0.0065}{\xi_{rel,c}} \right)} \quad 3-107$$

An absolute surface roughness of 1E-5 m is assumed for all simulations. According to Cakir [135] this is a conservative assumption, as the results of his experiments showed

that the absolute surface roughness of chemically etched channels in stainless steels never exceeded 1.6 μm . The 10 μm assumption was nonetheless upheld in the current work, as this fits the guidelines of commercial PCHE manufacturer Heatric [90].

3.4.2.4 Discretisation of Recuperator

The rapid change in fluid properties of supercritical carbon dioxide near its critical point necessitates a discretisation of the recuperator to avoid overestimation of its performance. The discretisation of the recuperator is indispensable when designing very high effectiveness (>95%) recuperators for stationary power applications [65], to avoid having a pinch point inside the heat exchanger, rather than at an end. Another reason why discretisation is so important for stationary applications is the high overall degree of recuperation inside the cycle. It is not uncommon that the recuperator(s) are responsible for over 50% of the heat addition to the compressor exit flow. In this investigation, the effectiveness of the recuperator may also be high (depending on the design), but the proportion of recuperation to heat input in the MHX will be low. This is due to the modest turbine inlet temperatures and the departure of the compressor inlet temperatures from the critical point, which leads to increased compressor exit temperatures.

It is necessary for the recuperator to be discretised into shorter axial sections to improve the reliability of the performance calculations. However, discretising the relatively short recuperator (length < 1 m) into 40 nodes, as is a commonly accepted approach [55], [136], [137], rather than 10 nodes is excessive. It will dramatically reduce the chances of the model to converge, while leading to virtually no change in predicted performance. A similar sensitivity study to that presented in section 3.4.1.6 was conducted and showed that anything more than 10 nodes did not bear any benefits for the current application.

$$Length_{node} = \frac{Length}{N_{nodes}} \quad 3-108$$

$$A_{h.node} = \frac{A_h}{N_{nodes}} \quad 3-109$$

The length of the recuperator was therefore split into ten equally long sections, with the individual heat transfer surface areas being determined using equation 3-108. The heat transfer and pressure loss calculations described in sections 3.4.2.2 and 3.4.2.3 are therefore performed separately for each node, with the outlet conditions of one node being passed on as the inlet condition to the next. Due to the non-sequential nature of EL and PROOSIS, it is impossible to definitively state the order of the iterations that are

involved in solving the resulting set of equations, as that order depends on what boundary conditions are chosen.

Figure 3-15 helps to visualise the overall calculation procedure of an equivalent sequential solver however. If for example, the hot and cold flow inlet conditions were known and the heat exchanger geometry is imposed, a sequential code may guess the recuperator's hot flow outlet conditions. Afterward, the code would proceed with the heat transfer and pressure loss calculations for each node individually from left to right (node 1 to node N). Following the nodes in a sequential manner, the calculations would finally calculate a hot-flow inlet temperature and pressure, which would be compared against the known boundary. If the difference between the two sets of parameters is below the convergence criterion, the calculations are finished, otherwise, the initial cold-end conditions of the hot flow would be iterated again. PROOSIS does not iterate in such a sequential manner, but uses its matrix solver to iterate on the inlet conditions for multiple nodes at once, to accelerate the calculations. The non-sequential nature of PROOSIS will also allow for a completely different definitions of the problem without any additional coding. The same recuperator model would for example also be able to calculate the necessary recuperator length, if both inlet flow conditions and one outlet flow condition were known.

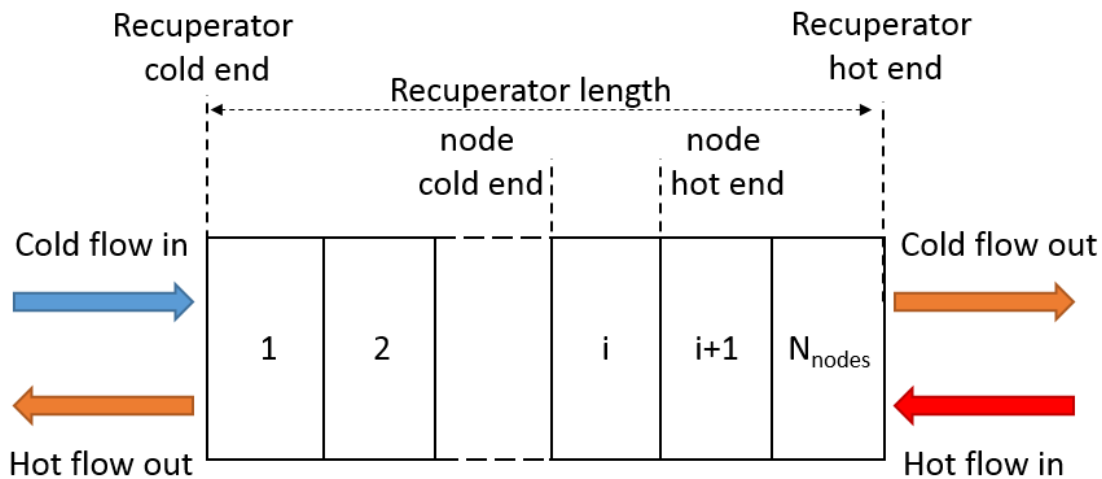


Figure 3-15: Schematic of recuperator discretisation

A discretisation in the direction of the other two main dimensions of the recuperator main body is not implemented or necessary, due to the baseline assumption of uniform flow distribution to all flow channels. In practice, the outer-most flow channels in either direction would likely transfer less heat than those toward the core of the heat exchanger,

but that should only have a minor effect on the overall performance and is therefore neglected here.

3.4.2.5 Input parameters for performance calculations

The recuperator model requires a number of geometric design inputs, in order to fully define the component and allow for the performance evaluation using the flow conditions at only two points of the heat exchanger. Table 3-4 lists all the design inputs required for the model.

Table 3-4: Input and their limits for the recuperator model

Parameter	Theoretical limits	Applied limits	Comment
$d_{c,C}, d_{c,H}$ [m]	0.0001 – 0.01	0.0005 – 0.002	Likely region of optimal design based on initial studies
FOS_{head} [-]	1 – 10	2	FOS of 1.5 or even below is commonly applied in aerospace. Higher FOS here incorporated due to preliminary models
$Height$ [m]	0.01 – 0.5	0.01 – 0.5	Likely region of optimal design based on initial studies
k_{mat} [W/(m*K)]	0 – 1,000	6.7	Ti-6Al-4V assumed [111]
$Length$ [m]	0.01 – 10	0.01 – 1.0	Likely region of optimal design based on initial studies
N_{nodes} [-]	2 – 100	10	Maximum possible discretisation while retaining convergence
$R_{0,2}$ [MPa]	100 – 5,000	900	Approximate yield strength of Ti-6Al-4V in relevant temperature region [101]
$Width$ [m]	0.01 – 0.5	0.01 – 0.5	Likely region of optimal design based on initial studies
$\xi_{abs,c,X}$ [m]	0 – 0.001	1E-5	Conservative estimate for chemical etching process [135]
ρ_{mat} [kg/m ³]	0 – 20,000	4,330	Ti-6Al-4V assumed [111]

Most of the parameters need to be specified for the hot and cold side of the heat exchanger separately. The theoretical and practical limits of the ranges that can be assessed using the model are also indicated. The reasoning behind the differences

between the technically applied and theoretical limits is explained in the comment section where appropriate.

The channel pitch and plate thicknesses may also be imposed in the model directly, but have been inferred from the channel diameter as discussed in section 3.4.2.1. Many of the PCHE's produced by Heatric feature a channel diameter of 2 mm and this is also the advertised 'standard' on the company website [90]. However, Cakir [135] and Heatric [131] themselves have shown that smaller channels are possible, though this may result in higher manufacturing costs due to the necessarily higher precision of the chemical etching process.

3.4.3 Bypass Flow Pressure Losses

The pressure losses in the bypass duct of a modern turbofan engine depend on numerous parameters such as the outlet swirl and turbulence from the fan outlet guide vane, the surface finishing of the duct, the length of the nacelle or the number and position of the fairings at the back of the engine. In the present simulations the effects of the FOGV are accounted for in the component efficiency of the fan and are therefore of no immediate concern when modelling the pressure losses in the bypass duct. Most of the losses that arise from the remaining loss sources are dependent on the Mach number of the bypass flow.

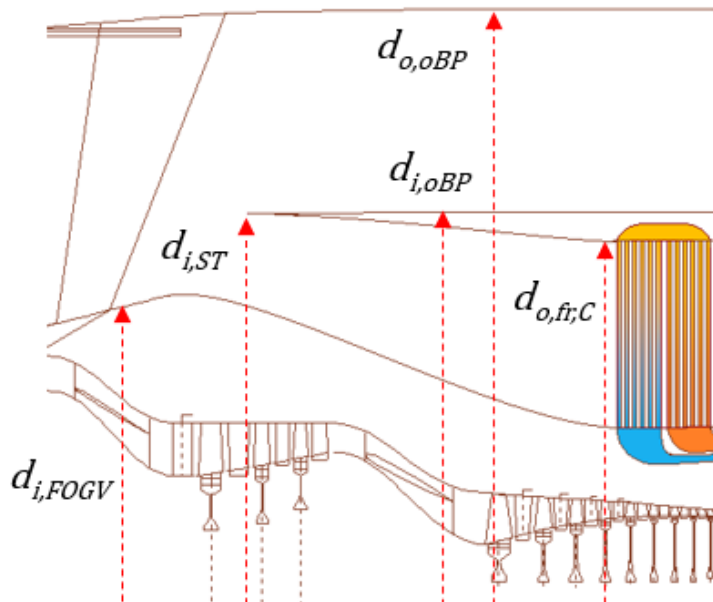


Figure 3-16: Excerpt of a CCTF2050 cross-section with pre-cooler dimensions

Introducing additional components to part of the bypass, such as an intercooler or the pre-cooler in the current investigation, will affect the bypass geometry and therefore the pressure losses. The inner bypass flow that is diverted from the outer bypass in the inflow duct of the pre-cooler is subsequently diffused to manage the pressure losses across the heat exchanger. This diffusion process will lead to a large outer diameter of the pre-cooler ($d_{o,fr,C}$) and hence of the intermediate casing around the inner bypass duct ($d_{i,oBP}$). If the inner diameter of the outer bypass duct ($d_{i,oBP}$) is greater than the inner diameter of the imaginary stream tube that is ingested by it ($d_{i,ST}$), two possible scenarios (or a combination thereof) are possible:

- I. The outer bypass can be redesigned to keep the Mach number in the outer bypass constant. This will lead to an increased nacelle diameter and hence engine drag
- II. The nacelle diameter can be kept constant. This will lead to a contraction of the flow path and hence acceleration of the outer bypass flow, resulting in higher bypass pressure losses.

In practice, a combination of the two scenarios would likely be employed, as any change in $d_{i,oBP}$ would require only a fraction of that change to be applied to $d_{o,oBP}$ for a constant flow area. Small changes in nacelle diameter may therefore help avoid excessive bypass losses. In order to take this trade-off into account and find the optimum engine installation, it would be necessary to quantify the nacelle drag independently from the remaining engine and aircraft parameters. However, the aircraft model employed in this investigation directly links fan diameter and nacelle diameter, making a separate increase in nacelle diameter impossible. The aircraft trade factors employed here unify the different influences of increased fan diameter on the aircraft performance by taking changes to the load bearing structures in the wing, longer landing gears, higher momentum drag and increased nacelle drag into account all at once. Hence, it is not possible to explore option I and it is necessary to account for the changes in bypass pressure losses due to flow acceleration.

Before the increase in pressure loss across the outer bypass can be calculated, the flow acceleration needs to be determined. First, the mass flow average Mach number of the whole FOGV exit flow was determined as $M_{ref} = 0.575$, using the cruise operating point of the performance model and the TF2050 cross-section. It is important to note, that the FOGV outlet flow will show a large variation in Mach number with annulus diameter. In addition, the combined cycle engine model uses the coolant mass flow in the pre-cooler

$W_{diff,in}$ as a design boundary and the pre-cooler diffuser model is based on the assumption that the FOGV outlet flow at the hub has a fixed average flow Mach number of $M_{diff,in} = 0.4$ (see section 3.4.1.2). Hence, the flow Mach number of the remaining bypass flow has to be higher than the mass flow average of the whole of the FOGV exit flow. The average flow Mach number of the outer bypass flow needs to be determined for every simulated case depending on the pre-cooler air mass flow. Only thereafter can the pressure losses in the bypass can be calculated. The inlet Mach number $M_{oBP,in}$ to the outer bypass is calculated according to equation 3-110.

$$M_{oBP,in} = \frac{M_{ref} * \left(\frac{W_1}{BPR + 1} \right) * BPR - M_{diff,in} * W_{diff,in}}{BPR * \left(\frac{W_1}{BPR + 1} \right) - W_{diff,in}} \quad 3-110$$

$$d_{i,oBP} = d_{o,fr,C} + 0.1 \quad 3-111$$

The variation in outer bypass inlet Mach number due this approach is not very large, but nonetheless important for a consistent modelling approach. The peak Mach number in the outer bypass is calculated using the corrected mass flow and the change in inner annulus diameter of the outer bypass due to the pre-cooler duct. The casing around the pre-cooler is assumed to be approximately 5 cm thick, leading to equation 3-111. The change in flow condition from the outer bypass to the peak velocity is used to estimate the pressure loss increase in the outer bypass duct. The Mach number inside the outer bypass is dubbed M_{oBP} .

No publicly available and established modelling approach for this particular problem could be identified. Instead, multiple approaches that correlate pressure losses in ducts and pipes with Mach number were investigated, of which the following three were compared in detail:

- Darcy-Weisbach's [104] pressure loss correlation for flow in a pipe

$$\Delta p_{BP} = f_{d,BP} * \frac{L_{BP}}{d_{h,BP}} * \frac{\rho_{BP} * c_{in,BP}^2}{2} \quad 3-112$$

- Walsh and Fletcher's [14] pressure loss correlation for flow in a pipe

$$\Delta p_{BP} = f_{d,BP} * \frac{L_{BP}}{d_{h,BP}} \quad 3-113$$

- Correlating bypass pressure loss directly to Mach number squared

$$\Delta p_{BP} = \Delta p_{BP,ref} * \left(\frac{M_{oBP}}{M_{oBP,in}} \right)^2 \quad 3-114$$

The FOGV outlet Mach number at cruise and the respective bypass pressure loss at cruise were used as the reference values, from which all three approaches started. The values were derived from the TF2050 model and the respective engine cross-section. The FOGV outlet Mach number was assumed to be constant in axial direction for the whole of bypass duct length.

The former two approaches can be applied in two different ways. One possible implementation could use the FOGV outlet Mach number and pressure loss to calculate the friction factor for the reference engine. That friction factor could then be kept constant and the pressure loss would be scaled only with the variation of bypass hydraulic diameter, and flow conditions in the case of the Darcy-Weisbach approach. However, the friction factor of the flow should vary with Mach number, which leads to the second possible implementation of equations 3-112 and 3-113. The known Mach number and pressure loss without the pre-cooler duct can be used to calculate a pressure loss factor K_{BP} (dynamic head loss), which is in turn used to calculate the friction factor. When the pre-cooler duct is added, the friction factor is recalculated depending on how the hydraulic diameter of the outer bypass changed. That new friction factor is subsequently used to calculate the new pressure loss of the outer bypass.

Beside the fundamental difference of variable or constant friction factors, the definition of the bypass hydraulic diameter needs to be declared. For a 'clean' bypass with no pre-cooler duct, the hydraulic diameter is simply calculated according to equation 3-115. For the combined cycle engine the calculation is less straight-forward. In this investigation, the hydraulic diameter of the outer bypass section is calculated using the outer bypass wall and an assumed inner wall at the annulus diameter where the stream tubes of the outer and inner bypass flows are split (see equation 3-116).

$$d_{h,BP} = \frac{4 * A_{oBP}}{\pi * (d_{i,FOGV} + d_{o,oBP})} \quad 3-115$$

$$d_{h,BP} = \frac{4 * A_{oBP}}{\pi * (d_{i,ST} + d_{o,oBP})} \quad 3-116$$

$$d_{h,BP} = \frac{4 * A_{oBP}}{\pi * (d_{i,oBP} + d_{o,oBP})} \quad 3-117$$

The hydraulic diameter of the contracted outer bypass flow path is subsequently calculated according to equation 3-117. Hence, the difference in hydraulic diameter calculated with equation 3-116 and 3-117 represents the flow path contraction and leads to the change in friction factor and pressure losses.

Figure 3-17 shows the results of the five modelling approaches across a relatively wide range of outer bypass Mach numbers. The models using the Darcy-Weisbach equation and Walsh & Fletcher's equation are presented twice, once with constant and once with variable friction factors. Only one way of correlating the square of the flow Mach number and the total pressure loss existed, leading to only one curve. Overall, it can be seen that all five approaches predict similar variations of the bypass pressure losses with Mach number, though the methods that assume a fixed bypass friction factor (carrying the index 'fric_const') predict slightly greater changes than the others.

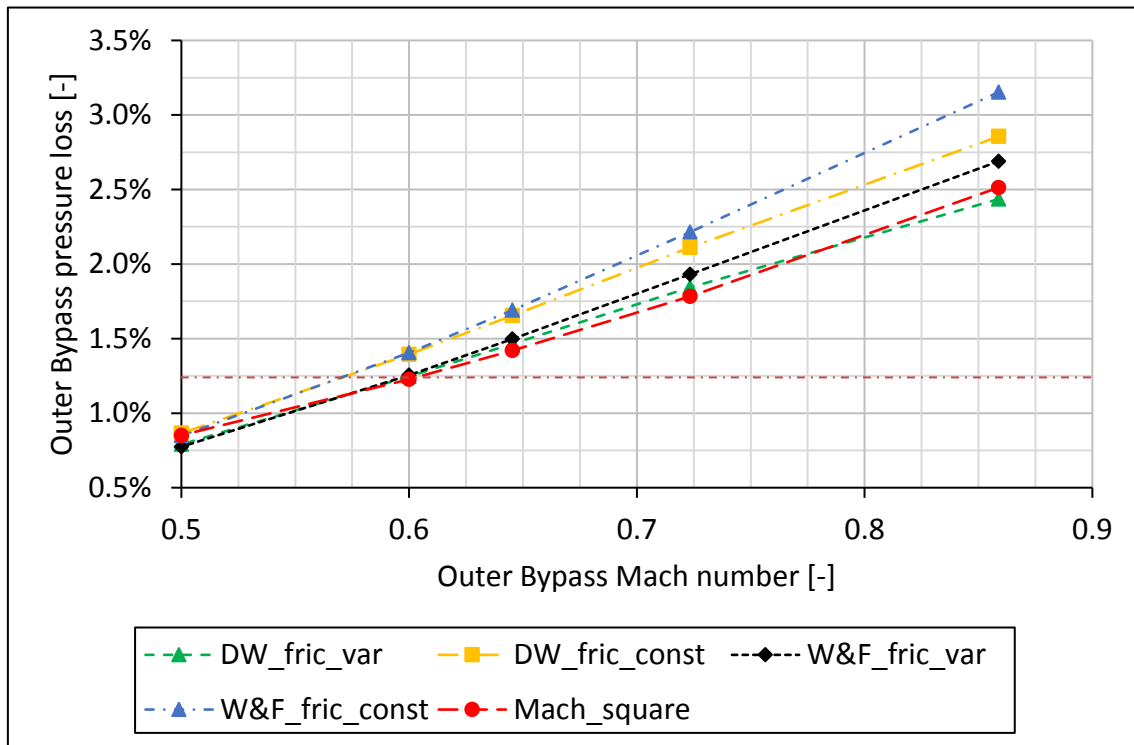


Figure 3-17: Modelling of bypass pressure loss variation with Mach number

All of the presented approaches appear to predict reasonable variation for Mach numbers up to approximately 0.7, when starting from a reference Mach number of 0.575.

However, the pressure losses are probably underestimated for greater accelerations [127]. The application of the presented models is therefore only recommended for Mach numbers below 0.7.

Additionally, the Darcy-Weisbach correlation was originally derived to describe incompressible flow, as the simplification of dynamic head in the last term of the equation indicates. Those modelling approaches were therefore dismissed from the current study. Scaling the pressure losses with Mach number squared was also determined to be sub-optimal, as that approach does not account for the influence of different annulus geometries (e.g. via hydraulic diameter). Hence, Walsh and Fletcher's equation with variable friction factors was implemented in the final model. Note, that the implemented model does not allow the pressure loss in the outer bypass to drop below the value of reference engine (represented by red horizontal line in Figure 3-17), even if the outer flow Mach number would be reduced to less than 0.575. This constraint is imposed to respect the technology limits of the bypass duct [127].

It can be expected that the optimal combined cycle engine configuration will feature as small a bypass flow disturbance as possible. The pressure loss increase calculated with the applied model should be a strong incentive to avoid considerable acceleration of the outer bypass flow. Especially the combined cycle simulations without inter-turbine reheat should show that an engine configuration with little to no bypass flow acceleration would give the greatest fuel burn benefit.

Any further increase in coolant and therefore S-CO₂ flow will lead to a relatively small increase in power output from the bottoming cycle, which is probably not enough to offset a considerable change in bypass pressure loss. The trend may be different in case of the combined cycle with inter-turbine reheat. The greater LPT exhaust gas temperature will increase the amount of additional power that may be extracted from the bottoming cycle for an increased S-CO₂ flow. Consequently, the balance between bypass pressure losses, useful power and fuel burn will be different to that in the CCTF2050, likely leading to a larger pre-cooler and increased bypass pressure losses in the optimal design.

3.5 S-CO₂ Turbomachinery

The component models of the S-CO₂ turbine and compressor are not detailed preliminary design models as the heat exchanger models described before. Instead, they scale performance maps of existing S-CO₂ turbomachinery components to represent a geometry and simulate off-design behaviour. The design point performance of the

turbomachinery components is calculated on an enthalpy and entropy basis, where equation 3-118 is used to calculate the isentropic pressure ratio of the compressor.

Note that φ_t is the entropy function and not the actual entropy at a state. Sethi defines the entropy function for PROOSIS as follows: “The change in specific entropy at any condition (where $P = P_2$ and $T = T_2$) and a stipulated reference datum point (where $P = P_{Ref}$ and $T = T_{Ref}$) can be represented as shown in Equation 3-119. The term $\int_{T_{Ref}}^{T_2} \frac{C_p}{T} dT$ is defined as the entropy function.”

The default ideal gas components of PROOSIS are capable of taking heat leakage from the turbomachinery to their casings and the environment into account when calculating transient performance. However, transient performance simulations were not part of the current investigation, so the S-CO₂ components do not take these effects into account.

$$PR_{cmp} = e^{\frac{(\varphi_{t,out,is} - \varphi_{t,in})}{R}} \quad 3-118$$

$$S(T_2, P_2) - S(T_{Ref}, P_{Ref}) = \int_{T_{Ref}}^{T_2} \frac{C_p}{T} dT - R * \ln\left(\frac{P_2}{P_{Ref}}\right) \quad 3-119$$

The S-CO₂ bottoming cycle is assumed to include one compressor and two turbines. The high pressure turbine drives the compressor via a shaft, while the free power turbine provides useful power to the low pressure shaft of the main engine via an electric or electro-mechanical transmission system. Olatz Canovas [120] developed a Matlab tool to size a mechanical transmission system with a fixed gear ratio to transfer the shaft power from the free power turbine to the main engine. However, it has since been determined that it would be beneficial for the off-design and especially transient performance of the bottoming cycle, if the rotational speeds of the two cycles were independent. Hence, the preliminary design tool was not included in the current investigation. Canovas’ work determined that a gearbox with a fixed speed ratio that transfers approximately 2 MW of power would weigh approximately 15 kg. Given the additional components necessary for an electric transmission system, a provisional 25 kg was assumed for the transmission system in the combined cycle performance studies. The efficiency of the transmission system was fixed at 99.5%.

Validation of any of the S-CO₂ models is extremely difficult, due to the little amount of data available. While the number of studies regarding S-CO₂ power cycles has risen considerably in recent years, much of the raw data published is non-dimensionalised or

not published at all. In case of the turbomachinery components, the models are compared with the design point data presented by Sandia [138]. The boundary conditions for the main compressor are:

- $T_{t,cmp,in} = 305 \text{ K}$
- $P_{t,cmp,in} = 7.69 \text{ MPa}$
- $W_{cmp,in} = 3.46 \frac{\text{kg}}{\text{s}}$
- $\eta_{is} = 0.67$
- $PR = 1.8$

The deviation of the results strongly depends on the number of significant figures published. If the exact design inputs mentioned above are used in PROOSIS, the compressor work is predicted to be about 4.3% lower than expected (see Table 3-5). However, the PROOSIS results can match the Sandia data perfectly, if some of the design inputs are tweaked, so that their rounded values remain unchanged and the results are rounded to the same number of digits as in the report (see latter two columns). Hence, it can be assumed that the PROOSIS model is able to predict the compressor performance as accurately as the tools at Sandia National Labs.

Table 3-5: PROOSIS compressor performance vs Sandia data

	Units	Sandia data	PROOSIS results	Deviation	'matched' PROOSIS	Deviation
$T_{t,cmp,in}$	K	305	305		305.3	
η_{is}	-	0.67	0.67		0.665	
$T_{t,cmp,out}$	K	324	322.75	-0.4 %	324	0 %
$P_{t,cmp,out}$	MPa	13.842	13.842	0 %	13.842	0 %
Pwr	kW	51	48.8	-4.3%	51	0 %

A similar study of the turbine model was conducted. The boundary conditions given by Sandia are:

- $T_{t,trb,in} = 810 \text{ K}$
- $P_{t,trb,in} = 13.499 \text{ MPa}$
- $W_{trb,in} = 2.58 \frac{\text{kg}}{\text{s}}$
- $\eta_{is} = 0.86$
- $P_{t,trb,out} = 7.885 \text{ MPa}$

The PROOSIS turbine performance results matched the original data to within 0.9% without applying any changes to the input variables. If the input parameters are changed slightly and the results rounded to the stated number of decimals, the original data can be matched almost perfectly once again. The model is therefore assumed to be able to accurately predict the design point performance of an S-CO₂ turbine. The two presented test cases are no full-scale validation studies, but they increase the confidence in the PROOSIS models nonetheless. More detailed studies should be conducted when greater data sets are publicly available.

Table 3-6: PROOSIS turbine performance vs Sandia data

	Units	Sandia data	PROOSIS results	Deviation	'matched' PROOSIS	Deviation
$T_{t,trb,in}$	K	810	810		810.4	
$P_{t,trb,in}$	MPa	13.499	13.499		13.4985	
$W_{trb,in}$	kg/s	2.58	2.58		2.584	
η_{is}	-	0.86	0.86		0.864	
$T_{t,trb,out}$	K	750	748.4	-0.2 %	749	-0.1 %
$P_{t,trb,out}$	MPa	7.885	7.885	0 %	7.885	0 %
Pwr	kW	178	176.4	-0.9 %	178	0 %

The following sections explain further details of the modelling approach and assumptions made with regard to the S-CO₂ turbomachinery components. Section 3.5.1 explains how the component maps were created, implemented and used in PROOSIS. Section 3.5.2 discusses the imposed isentropic efficiencies of the S-CO₂ turbomachinery components and the dependency of the final fuel burn estimates on their accuracy. Section 3.5.3 explains how the turbomachinery weight was estimated in a simple manner. Finally, section 3.5.4 summarises the design inputs during the combined cycle simulations presented further on.

3.5.1 Component maps

The component characteristics used for the S-CO₂ compressor and turbines during the combined cycle simulations were derived from the partial component maps published by Sandia National Laboratories (SNL) [138]. Those maps were provided to Sandia by Barber-Nichols Inc. (BNI), the manufacturer of the test rig turbomachinery. The assembly report of the test rig does not include a tabulated form of the component maps, as would

be necessary to implement them in PROOSIS. Hence, the pictures (see Figure 3-18 and Figure 3-19) included in the report were digitized using the WebPlotDigitizer [139]. The resulting data tables were converted to SI units in Excel. The data was subsequently introduced to Matlab, where polynomial fits were determined for each iso-efficiency line and non-dimensional speed line. BETA, or in the case of the turbines ZETA, lines were superimposed onto the map in parallel to the surge line of the map.

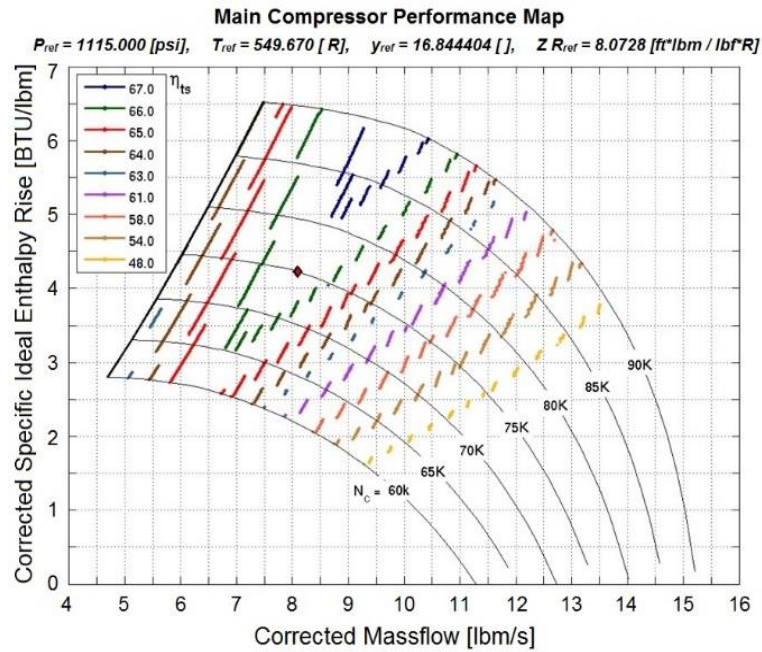


Figure 3-18: Partial component map of BNI radial compressor [138]

The intersections of all these lines with each other were then used to generate the tables in the correct format for PROOSIS. Figure 3-20 shows the compressor and Appendix G the turbine map, both scaled to the test case described by SNL [138]. The turbine map is depicted differently in PROOSIS than it is in the Sandia report, making a visual comparison of the plots almost impossible. Unfortunately, a validation of these maps has not been possible, due to a lack of publicly available off-design performance data.

Without a preliminary design tool, it is not possible to definitively say whether the compressor is going to be a single stage radial compressor, a two-stage radial compressor or an axi-centrifugal compressor. Brun et al. suggest that single stage designs may be best for compressors with a power requirement of less than 1 MW [65]. However, the pressure ratio assumed in that simplified classification was not mentioned and neither was the compressor's isentropic efficiency. It is therefore possible that a multi-stage design may bear better overall performance. No matter how many stages the

compressor ultimately has, it is modelled as a single component in PROOSIS, using a scaled component map such as that plotted in Figure 3-20.

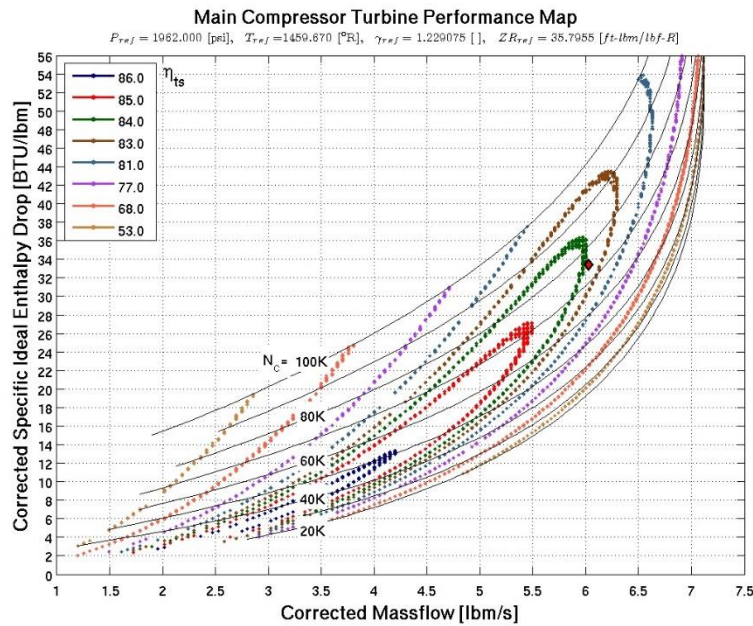


Figure 3-19: Partial component map of BNI radial turbine [138]

The original component maps from BNI are specific to the test rig setup at SNL [68]. In order to use them for different engine designs and inlet conditions, the maps are scaled in PROOSIS. The scaling routines are not specific to real or ideal gas applications and could therefore be applied from the standard libraries. The scaling of the component maps is influenced by the user through the definition of a range of design point parameters. Different input parameters may be chosen, due to the flexibility of the PROOSIS models, but typical design inputs are:

- Inlet mass flow, temperature and pressure
- Pressure ratio, component isentropic or polytropic efficiency and absolute rotational speed
- BETA / ZETA value of design point after scaling
- Non-dimensional rotational speed before and after scaling

These parameters allow the model to scale the map to the intended application while retaining its shape. The axis and iso-lines of the map are scaled linearly, the scaling factors and adders are automatically calculated by the model. The definition of the non-dimensional rotational speed before and after scaling moves the design point of the component relative to the iso-efficiency lines, which enables a shift in off-design

behaviour of the component. This is a useful handle when matching an existing design, but is unnecessary for a feasibility study such as this. Hence, the non-dimensional rotational speeds were both set to unity in the current investigations and the relative position of the design point remained unchanged. The number and kind of necessary design inputs depends on the model, as the rotational speed of the HPT would for example be the same as for the compressor, eliminating the need to impose the same value twice.

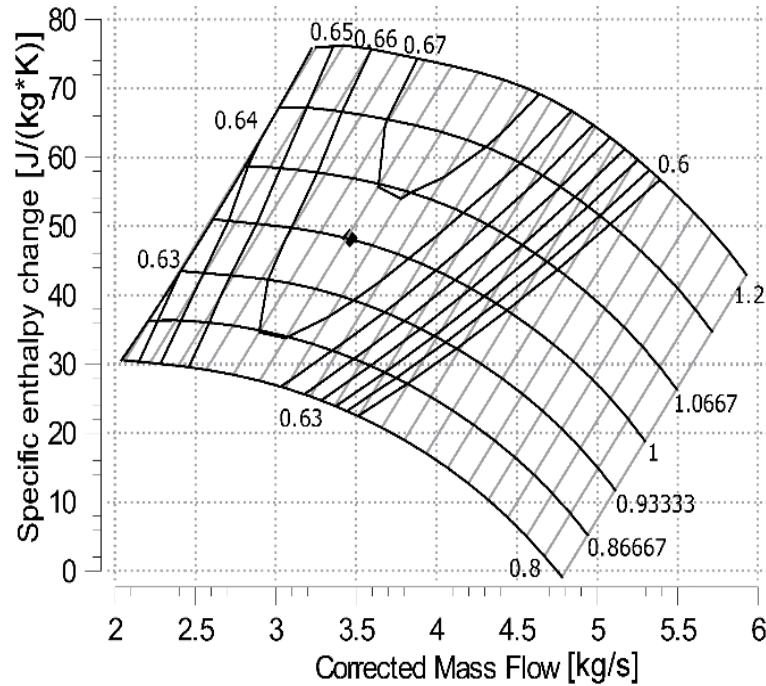


Figure 3-20: Radial compressor map in PROOSIS scaled to SNL test case [138]

The digitisation and subsequent scaling of the BNI maps is an imprecise process, which inherently introduces uncertainty to the off-design modelling of the cycle. The design point performance will be unaffected by that uncertainty as long as the design inputs are realistic. Furthermore, the component maps delivered by BNI are extrapolations of BNI in-house simulation data and include further uncertainties. Bechtel Marine Propulsion Corporation operates another micro-scale S-CO₂ test rig (about 100 kWe maximum power output) whose turbomachinery components and maps were supplied by BNI. Initial off-design performance tests at BMPC identified significant disparities between the performance predicted by the simulation models using BNI's component maps and the measurement data [69]. Due to a number of unrelated issues at that point of the test campaign, the test rig could only be run up to approximately 50% of the design power output.

In that range, the following differences between the measured and predicted component performance were observed:

Table 3-7: Reported differences between model and test data at BMPC [69]

	η_{is} predicted	η_{is} measured	Further measured benefits
Compressor	58 %	72.4 %	+72 % enthalpy rise
Compressor turbine (HPT)	80.7 %	85.2 %	+6 % turbine flow
Generator turbine (FPT)	81.3 %	83.6 %	+2 % turbine flow

The turbines and the compressor therefore performed well above the predictions of the performance model. The compressor actually operated at a higher isentropic efficiency than the stated design point efficiency of 61 %. A large part of this uncertainty is likely due to the low power conditions of the test run, but the data highlights the need for continued research of S-CO₂ turbomachinery.

Any modelling uncertainty inherent to the newly derived maps for the S-CO₂ turbomachinery components are not consequential to the presented mission performance data of the combined cycle engine. The off-design performance of the combined cycle engine had to be scaled rather than calculated using the PROOSIS models, due to instabilities of the combined cycle models when operated at off-design. Section 3.7.3 explains the scaling methods that were employed during the parametric studies.

3.5.2 Component efficiencies

The isentropic efficiencies of the compressor and the turbines are prescribed at design point and varied according to the scaled component maps at off-design. An automated preliminary design and performance estimation tool, such as those developed by Qi et al. [114] or Monge [113], would have increased the fidelity of the turbomachinery models and reduced uncertainty introduced during the digitisation of the BNI maps. However, these processes were too complex and iterative to be implemented in the present models. Should the concept development be continued, the preliminary design tools may be developed externally and executed iteratively with the PROOSIS model or integrated directly into the PROOSIS component codes.

The isentropic efficiency of the compressor was fixed at 89 % during the current investigations, while the turbine efficiencies were set to 90 %. These values are consistent with the assumptions made by Dostál [55] for nuclear applications with an EIS in 2025. Further studies into S-CO₂ cycles for the close- to mid-term future [115], [140] assume or calculate isentropic efficiencies of 85% and 90% for the compressor and turbine respectively, though for larger mass flows. Li et al. identified more moderate values of 72 % and 82 % for their compressors and turbines, though their design space was limited through a fixed rotational speed of 30,000 RPM. Overall, the turbomachinery efficiencies assumed herein are ambitious, but may be tenable considering the 35-year prelude to the estimated entry into service of the engines.

The results of a sensitivity study conducted for the matured combined cycle engine presented in section 4.1 are shown in Figure 3-21. Note, that the design point component efficiencies are the only parameters that were changed for each of these cases. The blue line represents the change in delta fuel burn relative to the TF2050, when only the compressor isentropic efficiency is changed. The fuel burn benefit is decreased by approximately 0.3 percentage points, if the compressor isentropic efficiency were 85 % instead of the assumed 89 %. The red line shows the effect the turbine efficiencies have on the fuel burn benefit. The isentropic efficiencies of the S-CO₂ HPT and FPT are changed in unison in this case. The turbine efficiencies affect the overall performance of the cycle much more than the compressor efficiencies. This was to be expected, as the combined turbine work is considerably larger than the compressor work. The fuel burn benefit of the combined cycle is reduced by approximately 0.7 percentage points if the isentropic turbine efficiencies were 85 % instead of 90 %. This represents a loss of almost 22 % of the benefit. Even though the turbine efficiencies are more influential, they are also traditionally easier to design. If the turbine and compressor efficiencies were both significantly lower than the assumed 89 % and 90%, it can be deduced from this graph that the combined cycle benefit can easily vanish completely. High turbomachinery efficiencies are therefore crucial to the cycle's feasibility. Again, the design point component efficiencies are the only changed inputs during these simulations. As a result, the heat exchangers and compressor inlet conditions were no longer optimal for those cycles. Hence, the inclination of the graphs is therefore likely an exaggeration, as parametric design studies of the combined cycle would probably be able to improve upon the presented fuel burn values.

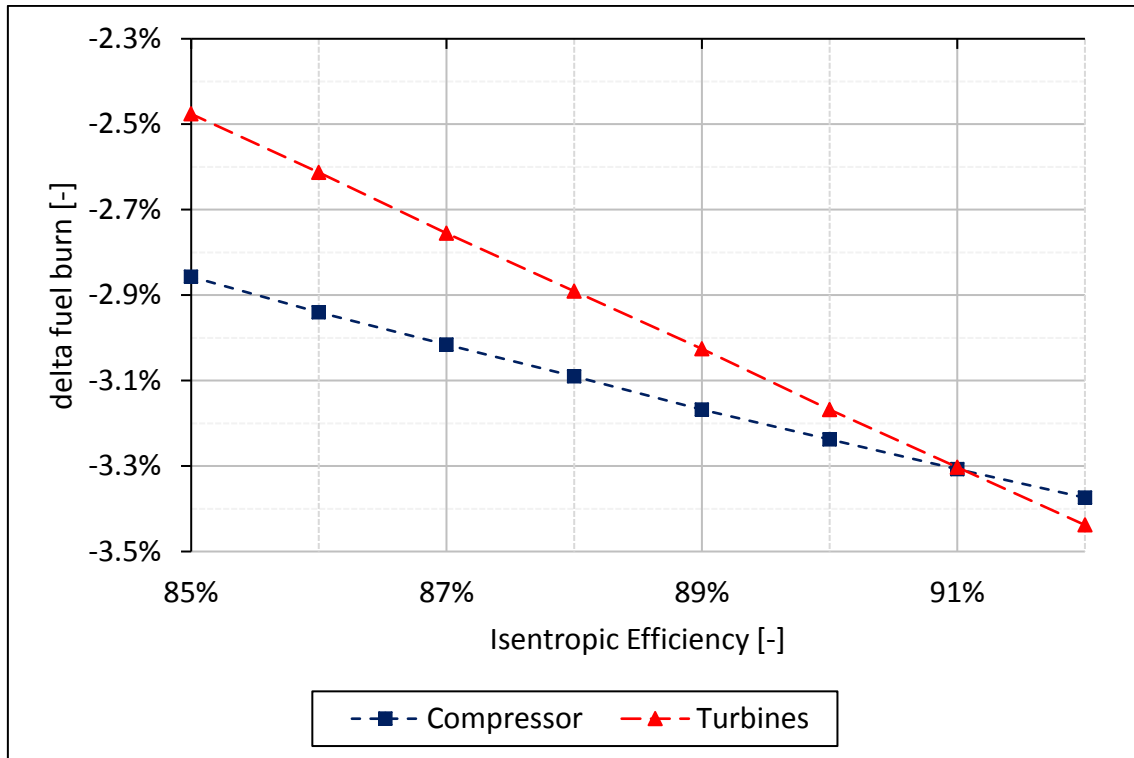


Figure 3-21: Sensitivity of combined cycle to turbomachinery efficiencies

Equally as important to the overall performance of the bottoming cycle as the turbomachinery efficiencies themselves are the design and performance of the bearings and seals involved. Section 2.6 discussed some recent developments in that area and showed that especially windage losses due to gas foil bearings in micro-scale test rigs can have a significant impact on the overall performance. This concept study does not include preliminary design or performance models for the bearings or shafts of the S-CO₂ turbomachinery, but it is obvious that any further development of the concept would require detailed studies of those technologies. The power rating of the bottoming cycle in the CCTF2050_ITR engine is approximately an order of magnitude greater than that at Bechtel Marine Propulsion Corporation (BMPC) [69] or SNL [68], which means that the percentage of turbine power lost to windage and other effects will be significantly smaller than the up to 10% observed in those examples. Nonetheless, these losses will be significant and further development of sealing and bearing solutions will be required to meet the stated performance figures.

3.5.3 Weight Estimation

The S-CO₂ turbomachinery component models do not include a weight estimation procedure. Instead, the combined weight of the compressor, the two turbines and the

shaft is scaled from test rig data. The SunShot Program was a research program funded by the U.S. Department of Energy [73]. Under the umbrella of that research program, different test rigs were conceptualised, one of which was originally supposed to include a 10 MW S-CO₂ turbine. The test rig was redesigned and the turbine's power reduced to approximately 2 MW. However, the original 10 MW turbine would have weighed approximately 200 lbs (~91 kg) [141], including its casing, bearings and shaft. The net sum of the power consumed by the compressor and produced by the two turbines is approximately 2.6 MW in the combined cycle aero engine with inter-turbine reheat. The combined weight of these components was linearly scaled based on the SunShot information, which leads to an approximate weight of 24 kg. An additional 25% was added to that estimate, due to the crude method used. In practice, the turbomachinery weight would likely scale quadratically with power, leading to a lower component weight. However, as the scaling was based on only one data set, this would appear as too optimistic an approach.

Due to the absence of a dedicated sizing and weight estimation tool for these components, it was decided to assume the same component weight during all combined cycle simulations. 30 kg may be an overestimation of the combined compressor and turbine weight, considering the very high working pressures and hence very small scale of the components. This is especially true, as the weight was estimated based on the increased power output of the combined cycle engine with inter-turbine reheat rather than the non-reheat configuration. Assuming the same weight for all configurations also leads to a slightly disproportional penalisation of configurations whose turbomachinery components produce and consume less power, but the effect is negligible. An additional mass of 30 kg per engine raises the mission fuel burn of this type of aircraft by only around 0.06%. Any uncertainty introduced here will therefore not have a significant effect on the feasibility of the concept.

3.5.4 Input parameters for performance calculations

The isentropic efficiency of the compressor was fixed to 89% during the parametric design studies presented in section 4. The isentropic efficiencies of the two turbines was furthermore fixed to 90%. The weight of the whole turbomachinery assembly including bearings, seals, shaft and casings was estimated to 30 kg. An additional weight of 25 kg was allocated for the power transmission between the free power turbine and the low-pressure shaft of the main engine. The efficiency of that power transmission system was furthermore assumed to reach 99.5%. Most of these values are rather ambitious top-

level module requirements, which will necessitate considerable research and development efforts to achieve, should the concept be explored in further detail. However, given the 35 years of possible development time and the possible transfer of knowledge from research efforts of related issues for more conventional engines, these targets to do seem tenable.

3.6 Inter-turbine Reheat

The secondary combustor is modelled analogously to the primary combustor, using pressure losses, fuel flows and the combustor efficiency as the main input parameters beside the inlet flow. The high pressure turbine can no longer be simulated as an equivalent single stage (EQSS) turbine as in the TF2050 or the CCTF2050, because the secondary combustor's optimal position will be between the HPT stages rather than between turbines. Previous studies determined that the achievable performance benefit will be greater the earlier the additional heat is introduced in the expansion process [38], [39]. This trend has been confirmed in the current investigation as can be seen in section 4.1. Introducing the reheat combustor between the first stage (HPT1) and second stage (HPT2) of the HPT, means that the two stages are modelled separately, but not independently, as the work split at design point has to be specified. The work split is implemented as a design parameter in the new turbine components and is defined according to equation 3-120.

$$WS_{HPT} = \frac{Pwr_{HPT1}}{Pwr_{HPT1} + Pwr_{HPT2}} \quad 3-120$$

The efficiency penalty that may be passed on from the HPC due to excessively small blade heights is only applied to HPT1, as it is assumed that the blades of HPT2 will be large enough to not be significantly affected by this anymore.

The detailed design of the reheat combustor is not of concern in this investigation and would be investigated at a later stage of a possible engine development program. The design will not only be affected by design point and high-power operating point conditions, but also by altitude relight ability, emissions and aeroacoustics concerns. From an emissions perspective it might be desirable to design the reheat combustor as a Rich-burn, Quick-Quench, Lean-Burn (RQL) combustor, as this could help reduce NO_x emissions when coupled with an LDI-design primary combustor. Alternatively, a combustor design similar to the 'flameless combustor' developed in the AHEAD project [42] might be an attractive solution (see Figure 3-22), if the claimed NO_x reductions

potential can be proven in operation. The weight of the engine will be affected by the addition of a reheat combustor in a number of ways. First, the specific power of the core engine will be increased, which will result in a reduced core mass flow and weight. This effect is captured in this study by the weight estimation tool WeiCo. On the other hand, the weight of the combustor itself, and the necessarily longer engine shafts and casings will increase the core weight again. This effect is not accounted for in the present investigation, introducing some further uncertainty to the final fuel burn estimates.

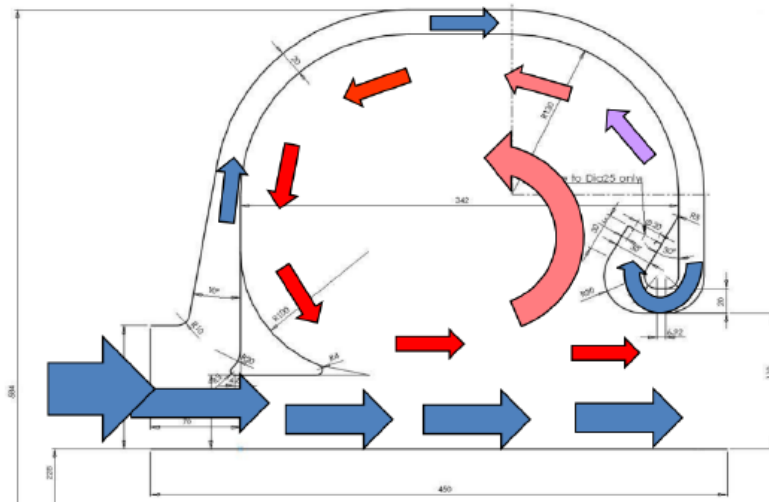


Figure 3-22: design sketch and schematic flow pattern of oxy-poor flame combustor, from AHEAD project [42]

3.6.1 Model assumptions

Numerous modelling assumptions had to be made when modelling the secondary combustor using these OD models. One of the most detailed studies of inter-turbine reheat on a geared turbofan engine in recent years was conducted by Köhler [39],[142]. Hence, a number of modelling assumptions were adapted from or can be compared against those made in his work. In his study, the combustor outlet temperature of the secondary combustor (TET_2) was allowed to exceed even the outlet temperature limit of the primary combustor. This seems like an unreasonable assumption, as the cooling and material requirements of the downstream stages, particularly the LPT stages, would be dramatically different to the reference engine. Consequently, the technical limit for TET_2 in the present study is the same as for the primary combustor (TET_1) at any given operating point (see section 2.1 for those limits). This does not necessarily mean that $TET_2 < TET_1$ during operation, though it is very likely that that will be the outcome of the parametric and optimisation studies of section 4.

Köhler's model of a geared turbofan engine with inter-turbine reheat did not include a method to account for the depreciation in component efficiencies of the high-pressure components of the core due to extremely small blade heights. This may lead to a noticeable overestimation of the performance potential of the concept, if the increased specific power of the core is used to shrink the core of the engine. In the past, inter-turbine reheat was most commonly researched as a way to increase the specific thrust of military engines. For civil applications, specific thrust reductions are often pursued, as they often go hand-in-hand with improvements in propulsive efficiency. Nonetheless, a slight increase in engine specific thrust may actually be a beneficial outcome of adding ITR to a civil turbofan. The reduction in core engine weight might affect the mission fuel burn more than the possibly rather slim SFC benefits that may be claimed by adding ITR to a civil turbofan. The inclusion of the non-linear aircraft trade factors added further depth to the analyses possible in this investigation relative to what Köhler presented.

The total pressure loss across the reheat combustor is assumed to be constant for all investigated cycles and has been set to 2 %. This may be an optimistic assumption, even though the downstream CMC NGV does not need film cooling. The combustor efficiency was set to 99.95%, which is a reasonable assumption considering that only high-power flight conditions are simulated.

3.6.2 Effect on cooling flow and reference model calibration

Köhler's model of a geared turbofan engine with inter-turbine reheat included a considerably more detailed model of the secondary air system than the present model. The implementation of that approach would have significantly reduced the stability of the models once more, prohibiting optimisation studies using Isight. The cooling model of the current engines is therefore purely based on the default capabilities of PROOSIS. The user imposes the cooling technology level via:

- The cooling air coefficient (CA_c) (see equation 3-121)
- The total temperature fraction of the NGV exit temperature, which the rotor sees ($T_{rel,C}$) (see equation 3-122)
- The percentage of the total cooling flow entering the turbine directed to the NGV (W_{NGV})
- The percentage of the total cooling flow entering the turbine pumped up in the rotor (W_{pump})

- The percentage of the total cooling flow entering the turbine that does work in the turbine (W_{tw})

$$CA_c = \frac{W_{cool}}{W_{rotor,in}} * \frac{(1 - eff_{cool})}{eff_{cool}} \quad 3-121$$

$$T_{t,rel} = T_{t,NGV_{exit}} * T_{rel,C} \quad 3-122$$

These model inputs are used in conjunction with the cooling mass flow to calculate a blade metal temperature of an equivalent single stage turbine. Alternatively, the flexibility of the models allows for the blade metal temperature to be used as an input and the necessary cooling flow is calculated. The latter approach was implemented at design point in this case, with the cooling flow for both stages of the HPT being extracted from the HPC exit flow. The values for the cooling flow technology indicators mentioned above for HPT1 and HPT2 were kept the same as in the TF2050 (see Table 3-8).

Table 3-8: Cooling flow model assumptions of TF2050

Variable	Value
CA_c [-]	0.041
$T_{blade,cruise}$ [K]	1140
$T_{rel,C}$ [-]	0.895
W_{NGV} [-]	0.313
W_{pump} [-]	0.382
W_{tw} [-]	0.358

Prior to the definition and calibration of the cooling model for HPT1 and HPT2, the performance of the engine had to be calibrated against the performance of the TF2050 to enable a like-for-like comparison. Splitting the EQSS into two separate turbine components theoretically should not affect the performance of the engine, as long as the pressure loss and heat input in the secondary combustor are set to zero. However, two effects led to a 1.5 % SFC improvement of the engine performance after all. First, the flow percentage of the cooling air introduced to HPT1 that does work in HPT1 was kept constant from the TF2050 model. As a result, that cooling flow does work in part of HPT1 and the whole of HPT2, resulting in a greater net total of produced power. This only improves the SFC by less than 0.3 %. The second effect was more significant. The variation in fluid properties, particularly the ratio of specific heats, from the inlet of HPT1 to the inlet of HPT2 led to a more efficient high pressure turbine assembly, even though

the same polytropic component efficiency was applied to each turbine as in the TF2050. The design point polytropic efficiencies of HPT1 and HPT2 were consequently lowered by about 1.4 % each to mitigate the two effects collectively and match the engine performance of the TF2050_ITR to that of the TF2050, when the secondary combustor is turned off.

The cooling flow of the TF2050 with EQSS HPT was determined by enforcing a maximum blade metal temperature of 1400 K at max power EOR conditions. The HPC mass flow percentage spent on cooling the HPT is kept constant throughout the flight envelope, so it is determined at cruise by imposing a blade metal temperature and checking the compliance of the EOR limit at off-design. The 1400 K limit was identified as the probable evolutionary limit for CMC blades by 2050. In the TF2050_ITR, the blade metal temperature of HPT2 is calculated to be 1427 K, if no cooling flow is introduced and the reheat combustor is unlit. Even though that temperature is beyond the assumed temperature limit for CMCs, it is kept constant as the input value to calculate the necessary cooling flow during the full design and off-design optimisation study of the TF2050_ITR. The necessary iterations between assumed blade metal temperature at cruise and maximum blade metal temperature at EOR are conducted automatically by Isight. Section 3.7.2 explains the setup of the optimisation environment. For the parametric studies of the CCTF2050_ITR, which use the simplified off-design performance estimation approach, this means that the blade metal temperatures are imposed at about 1140 K and 1185 K at cruise for HPT1 and HPT2 respectively. The inability to use the present models to reliably simulate off-design conditions of the combined cycle means that the maximum occurring blade metal temperatures cannot be predicted reliably. However, it is unlikely that the blade metal temperature will differ by more than 3-5 K from the desired levels at EOR, introducing only a second order uncertainty to the cooling flow calculations and the mission fuel burn estimates.

3.7 Concept Design and Performance Simulations

The feasibility of the combined cycle engine concept and the design of the most efficient engine configuration is determined using parametric studies. The biggest challenge when trying to mature the combined cycle engines originates from the amount of influential design parameters. The CCTF2050 has over 20 influential design parameters, 13 of which have to be investigated to be able to realistically assess the concept's potential. 13 influential design parameters and five values that needed to be assessed per parameter would result in a design space of over 1.2 billion possible combinations.

Even if every combination could be evaluated within 0.1 s, which they cannot given the manual process and the speed of the tools, the calculations would take almost 4 years. Automated optimisation techniques could identify the optimum parameter combination within days or even hours, but the instability of the models prohibits their use. Nonetheless, those cycles that offer the greatest fuel burn benefit and its parameters will be called optimal from here on, even if no automated optimisation study was conducted.

Multi-layered parametric studies were performed, during which design trends for individual parameters were identified to reduce the number of configurations that had to be assessed and therefore reduce the overall calculation time. During these studies, each parameter was investigated again for each additionally introduced parameter. The parametric studies of the CCTF2050 for example started with fixed design inputs for everything but the coolant mass flow of the pre-cooler. Once the coolant mass flow was identified that resulted in the lowest mission fuel burn of that particular design, the inlet temperature of the S-CO₂ compressor was added to the investigation. Due to the interdependency of all the design parameters, it was necessary to re-investigate the optimal coolant mass flow for every compressor inlet temperature. The fluid properties of S-CO₂ are strongly dependent on temperature and pressure, resulting in the need to find the optimal combination of coolant mass flow, compressor inlet temperature and pressure. Afterward, the geometric features such as the number of rows in the pre-cooler were investigated and again, the optimal combination of the previously investigated parameters needs to be identified. The optimal values of some parameters are unaffected by changes in other design inputs, reducing the number of calculations further. However, even with the extrapolation of design trends that can be observed during these studies this process is still highly iterative and time-consuming and over 2000 cases were evaluated as part this investigation. The performance potential of a geared turbofan engine with only the inter-turbine reheat as a novel component (TF2050_ITR) was quantified by integrating the relevant tools in Isight. Isight is a commercial optimisation environment that performs the same overall modelling approach, but more quickly and efficiently.

The following sections explain the details of the parametric studies. Section 3.7.1 describes the simulation environment, the tools used and what inputs were iterated upon outside of PROOSIS. Section 3.7.2 explains why the fan diameter is simultaneously a design input and output in the current setup and which ranges of each parameter were investigated in the individual engines. Section 3.7.3 explains how and why the off-design

performance of the combined cycle engine was estimated rather than calculated within PROOSIS. Finally, the engine weight estimation method is briefly presented again in section 3.7.4.

3.7.1 Simulation environment

The combined cycle models calculate the design and off-design performance of the engine and estimate the component weight for the S-CO₂ heat exchangers. However, the models do not estimate the weight of the remaining engine components or scale the aircraft to match the mission fuel burn of the engine. Instead, WeiCo (see section 3.7.4) is used to estimate the total engine weight and calculate its fan diameter during the parametric studies and the weight of the remaining S-CO₂ components is estimated as described before. The non-linear trade factors (NLTFs), provided by ULTIMATE partner BHL, are used to estimate aircraft mission performance. The different tools would ideally be integrated in an automated environment that is capable of single- or multi-objective optimisation. Unfortunately, this is presently not feasible. A manual procedure was created instead to iteratively match engine and aircraft performance and perform the design space studies necessary to mature the combined cycle engine concept. This approach is not equivalent to a full-fledged optimisation study but should suffice to determine the approximate potential of the concept.

Figure 3-23 illustrates the workflow of the parametric studies. Following the definition of the necessary design parameters of the engine (see section 3.7.2.1), the necessary engine inlet mass flow $\dot{W}_{in}$ is estimated, along with the fan diameter and the pressure ratio of the high-pressure compressor. Thereafter, the estimated inlet mass flow and the coolant mass flow of the pre-cooler (one of the design parameters) are used to calculate the inlet Mach number to the outer bypass according to equation 3-110 in section 3.4.3.

All design inputs are then imposed in PROOSIS and the design and off-design performance of the engine is calculated.

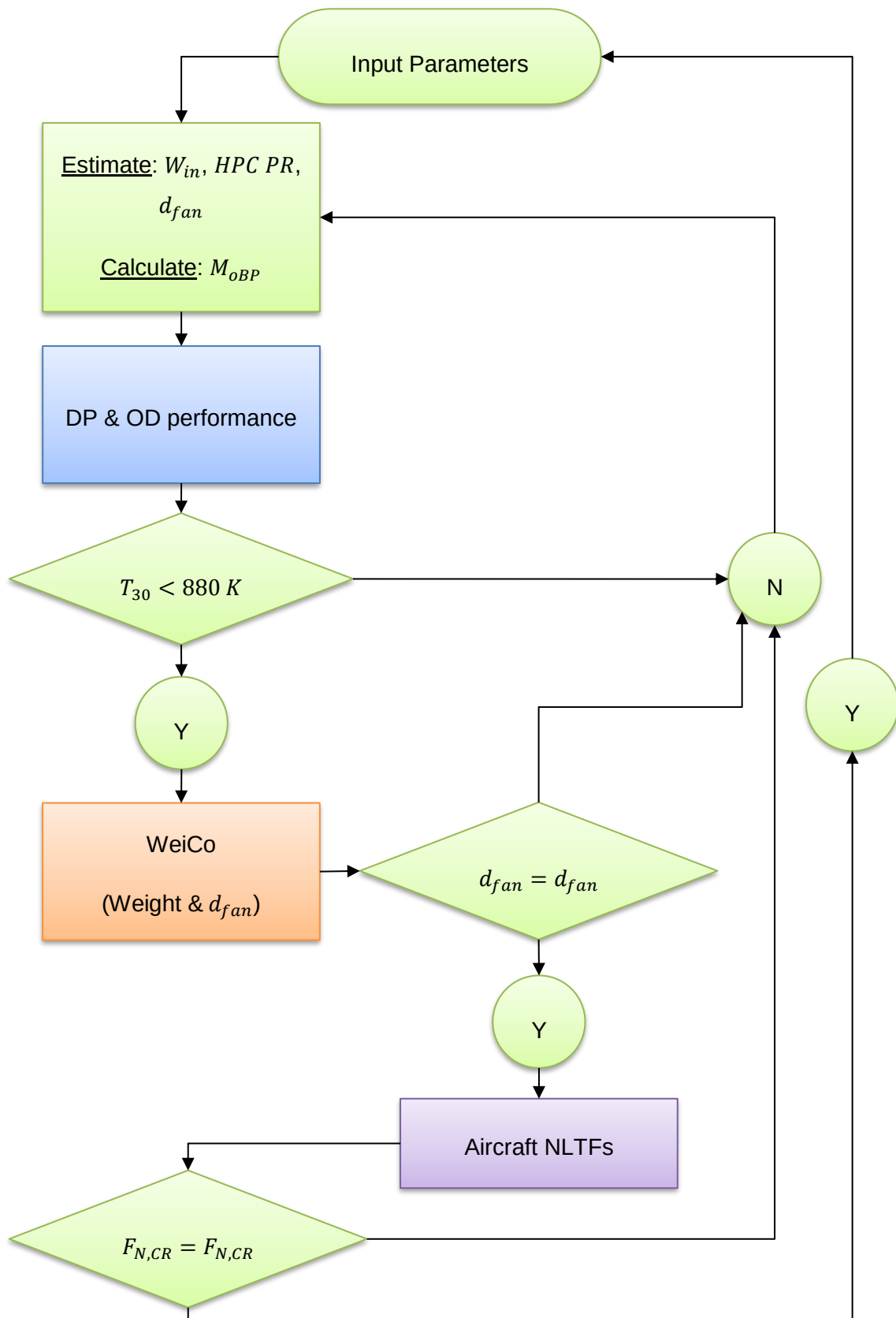


Figure 3-23: Flow chart of manual process for parametric concept studies

Subsequently, it is checked whether the HPC delivery temperature exceeds the threshold of 880 K at cruise. If so, the pressure ratio of the HPC is lowered and the performance is recalculated. The HPC pressure ratio is increased however, if the HPC delivery temperature is noticeably below the threshold. This way, the OPR of the engine is maximized for every design, as the greatest possible OPR proved to always give the best mission fuel burn for the CCTF2050. For the CCTF2050_ITR the same was done during most of the simulations, though the results showed that an OPR reduction may be beneficial, if the optimal TET2 cannot be reached (see section 4.3.2.4 for more details). Following the check of the HPC delivery temperature, WeiCo is executed. The fan diameter calculated by WeiCo is compared to the PROOSIS input value for fan diameter. If a deviation exists, the PROOSIS input will be replaced and the engine is recalculated. The aircraft non-linear trade factors are used to estimate the mission fuel burn value of the engine configuration, once the difference in fan diameters is zero. The NLTFs not only estimate the mission fuel burn, but also calculate the design and off-design thrust requirements of the aircraft for that particular engine, leading to a further design iteration using the engine inlet mass flow.

The whole process is repeated until the thrust produced by the engine at mid-cruise matches the aircraft thrust requirements to within $\pm 0.05\%$. During this process, the most influential performance and design data is recorded in Excel, including the design inputs of the investigated cycles. Excel was also used to check that the thermodynamic and mechanical design constraints explained in section 3.7.2.1 were adhered to.

The model of the geared turbofan engine with inter-turbine reheat (TF2050_ITR) is stable enough, to be integrated in an external optimiser. The optimisation environment chosen for this exercise is version 5.9-4 of the commercially available Simulia Isight Suite, developed by Dassault Systemes [143]. The iterative design procedure described above and depicted in Figure 3-23 was implemented in Isight, with the exception that the execution of the models, the iterations and the choice of design inputs is completely automated. Standalone engine performance decks were created and the same design constraints were applied to the optimisation, though the convergence criteria were tighter ($\pm 0.001\%$) than during the manual studies.

Isight includes a number of optimisation techniques [143], among which the Non-dominated Sorting Genetic Algorithm II (NSGA-II) and the Pointer technique are of greatest interest for the non-linear design space at hand here. NSGA-II is a well-established multi-objective exploratory optimisation technique that is well-suited for

highly non-linear and potentially discontinuous design spaces. The technique's ability to cope with discontinuous problems is particularly important, as the integrated tools themselves should not produce any discontinuous results, but convergence issues with one or multiple tools may seem like discontinuities to the solver. *"In the Non-dominated Sorting Genetic Algorithm (NSGA-II), each objective parameter is treated separately. Standard genetic operation of mutation and crossover are performed on the designs. The selection process is based on two main mechanisms, 'non-dominated sorting' and 'crowding distance sorting'. By the end of the optimization run, a pareto set is constructed where each design has the 'best' combination of objective values and improving one objective is impossible without sacrificing one or more of the other objectives."* [143]. The solver still determines an 'optimum' solution in the end, based on a set of weight factors the user can prescribe for each objective. The Pointer algorithm is also well suited for non-linear and potentially discontinuous design spaces, largely due to it being a combination of four different optimisation techniques. *"Pointer is an automatic optimization engine that controls a set of standard optimization techniques. It currently controls four optimization methods: a genetic algorithm, Nelder and Mead downhill simplex, sequential quadratic programming (NLPQL), and a linear solver. This complementary set of algorithms was selected because each succeeds and fails for different topography features. It has been found that a hybrid combination of these methods solves a broad range of design optimization problems. The Pointer Automatic Optimizer can control one algorithm at a time or all four at once. As the optimization is proceeding, the technique determines which algorithms are most successful as well as optimal internal control parameters (step sizes, numbers of iterations, number of restarts, etc.). This procedure is hidden from the user."* [143].

There are a few considerable differences between the NSGA-II and the Pointer algorithm. First, the Pointer algorithm is partially gradient based, which should help separate local from global optima faster, should there be any. The partially gradient-based approach of the Pointer technique should also be quicker to approach the global optimum. Another big difference lies in the setup of each technique. While NSGA-II investigates a given number of individuals in a pre-defined number of generations with a fixed mutation and crossover index, the Pointer algorithm automatically tries to find the optimum balance of all these parameters without the need for user input. The Pointer is therefore set to execute for a given time, while the execution time of an NSGA-II optimisation is dependent on the number of calculations and their individual calculation times. This can be particularly time-consuming for design spaces with a large number of

input variables, for which relatively large ranges of values need to be considered. Convergence rates will also have a substantial influence on the total calculation time of an NSGA-II optimisation, as non-converged engine decks for example can take ten times longer to close than converged cases. The Pointer technique is furthermore capable of single- and multi-objective optimisation, whereas the NSGA-II always needs to solve for at least two objectives. The NSGA-II can be turned into a de-facto single-objective solver through the weight factors, which can be used to immensely prioritise one of the two objectives, but that is not an ideal setup, as the solver will still investigate the improvement of the secondary objective as well. Overall then, the Pointer algorithm was deemed the more suitable technique and was employed for the fuel burn optimisation of the TF2050_ITR.

Optimising the TF2050_ITR with Isight, rather than investigating its potential through sequential parametric studies, inherently leads to a more precise exploitation of the cycle's performance potential. This should be taken into consideration when comparing the fuel burn values of the optimal TF2050_ITR with the combined cycle engines. However, the tools and the setup used during the Isight optimisation are essentially the same as during the parametric studies. The TF2050_ITR was also optimised using Isight, because it allowed for a comparison of the two possible methods to determine the engine's off-design performance (see section 3.7.3).

The two different off-design modelling approaches also lead to different Isight setups. The overall approach (see Figure 3-24) is similar for both setups and also to that depicted in Figure 3-23. Note, that the TF2050_ITR model uses the net thrust at design point as a handle rather than the inlet mass flow, simplifying the iterations. In both Isight setups, the optimiser chooses the necessary design inputs that are passed to the target solver, though the number and range of the inputs is different for the two setups (see section 3.7.2.1). The target solver is essentially an automated iteration loop that executes the standalone PROOSIS deck, WeiCo and the non-linear trade factors over and over again until a given convergence criterion is met or the maximum number of iterations is exceeded.

The target solver is calibrated differently for the two approaches. In the simplified model, only the net thrust at the design point is a variable of the target solver. The target solver matches that design thrust to the thrust requirements calculated by the non-linear aircraft trade factors. The target solver therefore only has one variable and target in the simplified approach. The convergence criterion is set to 1E-6, while the maximum number of

iterations is set to 10. The optimiser was allowed to run for 10 hours using the pointer algorithm.

The definition of the target solver for the second Isight setup using specific design and off-design performance decks is considerably more complicated. The solver is given six handles to meet five targets. The net thrust at design point, the HPT1 turbine entry temperature at cruise, TOC and EOR, and the blade metal temperatures of HPT1 and HPT2 at cruise are the variables of the target solver. The net thrust at cruise, TOC and EOR and the blade metal temperatures of HPT1 and HPT2 at EOR are the targets. This more elaborate setup ensures that the new engine not only matches the thrust requirements at cruise, but also at TOC and EOR. All the while, it ensures that the cooling flow is sized appropriately to ensure that the temperature limits of the HPT are not exceeded at EOR, the hottest operating point.

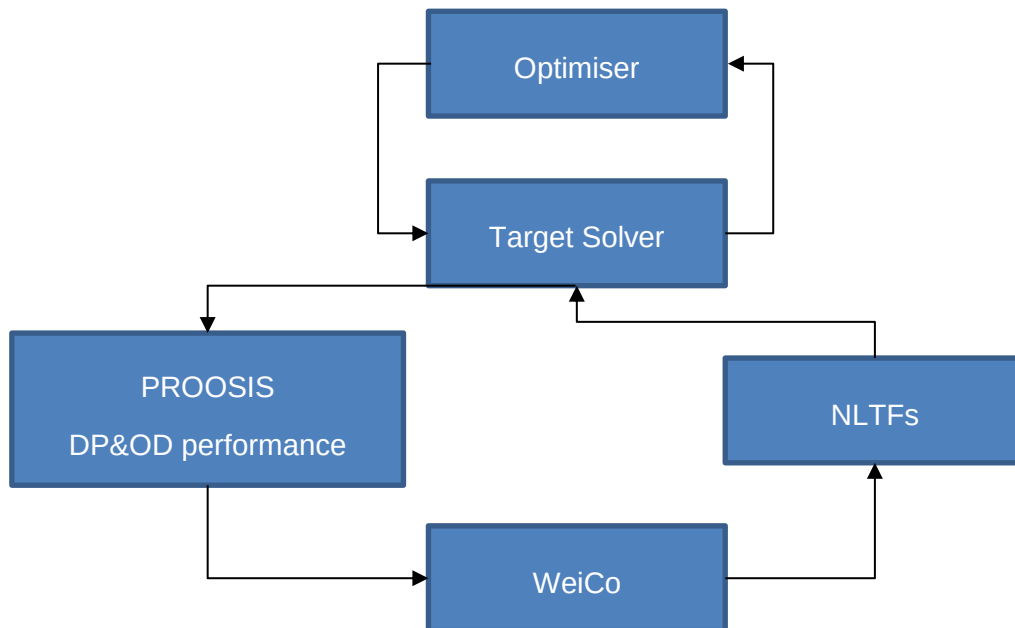


Figure 3-24: Schematic of Isight setup for TF2050_ITR optimisation

The higher number of variables and targets set in the target solver leads to a larger number of iterations that are necessary to meet the convergence criterion of 1E-6 for all targets. The maximum number of iterations per configuration was therefore increased to 35 in the latter Isight models. The optimiser was instructed to run for 30 hours using the pointer algorithm. The more detailed Isight setup leads to a more reliable assessment of the performance potential of the TF2050_ITR. Unfortunately, it requires too many iterations to converge to make a similar approach feasible for the manual parametric

studies of the CCTF2050_ITR. However, comparing the results of the two Isight setups allows for conclusions to be drawn regarding the fuel burn error that is introduced when the simplified off-design modelling technique is adapted for the combined cycle studies. The difference between the results of the two methods will vary with the engine configuration, but the order of magnitude will likely be the same.

3.7.2 Design point performance simulations

The design point performance of the three investigated engine configurations is calculated using PROOSIS. Each of the models has a specific set of design parameters used during the parametric studies. The investigated range of each of these parameters and the design constraints applied during the investigations are listed in section 3.7.2.1. Individual cases outside the stipulated ranges were simulated throughout, to ensure that the ranges encapsulated the ideal design space. Section 3.7.2.2 outlines how the polytropic efficiency of the HPC is penalised, if the blade height of the last compressor stage is too small.

Neither of the combined cycle engine configurations investigated here include a recuperator after all. The created model was used during earlier design studies and it was determined, that the power-to-weight ratio of the recuperator was too low to warrant its inclusion in the system. The final pre-cooler and MHX designs transfer between 12 and 15 kW of heat for every kilogram of material, whereas the very compact, but heavy PCHE recuperator only transfers between 3 and 4 kW per kilogram. The high sensitivity of the fuel burn to the weight of the engine therefore led to the conclusion that it is better not to include a recuperator.

3.7.2.1 Input parameters and general design constraints

The fan diameter is usually not a design input during 0D or 1D engine performance simulations. It is a necessary design input for the combined cycle engines however, due to the pre-cooler. The pre-cooler diffuser model requires the hub diameter of the fan outlet guide vane (FOGV) as an input as described in section 3.4.1.2. However, the hub diameter of the FOGV is dependent on the fan diameter and bypass ratio, and therefore needs to be calculated anew for each investigated engine configuration. The fan component model calculates the FOGV hub diameter based on the outlet Mach number of the FOGV, the fan diameter and the flow conditions at the fan outlet. The FOGV outlet Mach number is representative of the assumed technology level for the year 2050 and is therefore kept constant (approximately 0.575) during all simulations. The fan diameter

therefore directly affects the cold-side pressure losses in the pre-cooler. This in turn affects whole engine performance and consequently the fan diameter calculated by WeiCo. Hence, the fan diameter input parameter needs to be iterated as shown in Figure 3-23.

Additionally, the fan pressure ratio is one of the design inputs for all three investigated engine configurations, as its optimal value depends on the specific thrust design of the engine. Only the pressure ratio of the outer bypass section is used as a direct design input however. The pressure ratio of the inner bypass section is usually significantly lower than that of the outer bypass section due to the lower blade speed and subsequent design problems. The pressure ratio of the core bypass section (FPR_{CO}) is calculated according to equation 3-123 for the purposes of this study. That way, the relation between inner and outer FPR is kept the same as in the reference TF2050.

$$FPR_{CO} = \left((FPR_{BP} - 1) * \frac{0.24}{0.33} \right) + 1 \quad 3-123$$

The inlet Mach number of the MHX was kept constant at about 0.1 during the combined cycle simulations, as no direct coupling of the outer bypass to the MHX was in place that would have allowed for trade-off studies between the MHX outer diameter and the bypass pressure losses for example. Instead, another design constraint was implemented that forced the outer diameter of the MHX to be at least 10 cm lower than then outer diameter of the stream tube ingested by the pre-cooler diffuser. This way, a practical limit is applied, but no quantification of any variation in Mach number is captured. Hence, not all effects of a reduction of the MHX inlet Mach number are accounted for and it was dismissed as a credible design input during the parametric studies.

A general rule of thumb when integrating novel components that may affect the bypass stream is to keep the outer diameter of the added component to within 60% of the fan diameter [126]. 65 % may be seen as the critical limit beyond which all manner of difficulties arise, from the bypass flow accelerations or nacelle drag increases discussed in section 3.4.3 to general installation problems. That rule was not implemented directly within the presented models, but the implemented bypass pressure loss model has had a similar effect. The parametric studies of the combined cycle engine without inter-turbine reheat have identified the optimal outer diameter of the MHX and the pre-cooler to respect the 60 and 65% limits respectively. The outer diameter of the MHX was indirectly

affected by the bypass pressure loss model, due to its aforementioned forced relationship with the outer diameter of the stream tube ingested by the pre-cooler diffuser. The CCTF2050_ITR features a slightly greater outer radius of for the pre-cooler, with a maximum dimension of about 69% of the fan diameter. Nonetheless, these values seem acceptable, given that the increase in bypass pressure loss is directly accounted for in the engine performance. Minor redesigns of the heat exchangers to higher inlet Mach numbers or slightly lower mass flows would also help reduce the radial dimensions. However, more detailed studies of the rest of the engine installation would be required to quantify these incentives, so they may be integrated in the current engine models.

CCTF2050

The new bottoming cycle component models introduce a host of new input parameters that may be adjusted to optimise engine performance and general arrangement. Not all of those parameters however, are very influential on the mission fuel burn value in their current state, as most installation limitations for example are only applied via external constraints on the results. Hence, the number of design inputs included in the parametric studies performed to assess the feasibility of a combined cycle aero engine with supercritical carbon dioxide in the bottoming cycle was reduced to those listed in Table 3-9. The remaining input parameters of the respective models were kept at the constant values stated in their respective sections in this chapter. The number of involved design parameters of the main engine was furthermore kept as low as reasonably possible, to reduce the total amount of engine configurations that needed to be evaluated.

The pressure ratio of the IPC was fixed to 2.64 and the pressure ratio of the HPC was maximised for every configuration until the temperature limit of the HPC delivery air was reached. The component efficiencies of the turbomachinery components of the main engine were furthermore kept at the levels described in section 2.1. The only exceptions to that were the polytropic efficiencies of the HPC and HPT, whose baseline efficiency was kept constant, but their actual value was scaled with the blade height of the last stage of the HPC, as describe in section 3.7.2.2.

The lengths of the pre-cooler and the MHX diffusers were fixed to 1.2 m. The cross-section of the TF2050 revealed that a total length of 1.5 m for the pre-cooler diffuser might be eligible, while the MHX diffuser would probably need to be shorter than the 1.2 m. Assuming fixed values therefore introduces additional uncertainty to the results, but improving upon those assumptions is not reasonably possible without more detailed diffuser and installation models.

Table 3-9: Inputs and their investigated range in parametric CCTF2050 studies

Component	Parameter	Range
Main Engine	BPR [-]	16 – 18
	FPR_{BP} [-]	1.35 – 1.5
	W_{in} [kg/s]	410 – 440
S-CO₂ Compressor	T_{in} [K]	307.15 – 314.15
	P_{in} [MPa]	7.87 – 8.47
	PR_{BC} [-]	2 – 4
Pre-cooler	$M_{fr,C}$ [-]	0.08 – 0.15
	N_{col} [-]	6 – 8
	$a_{tube,C}$ [m]	0.005 – 0.01
	x_{inv} [-]	0.9 – 1.0
	$W_{coolant}$ [kg/s]	35 – 45
MHX	N_{col} [-]	6 – 8
	$a_{tube,H}$ [m]	0.005 – 0.01
	x_{inv} [-]	0.9 – 1.0

The cooling flow percentage for the HPT is calculated by the model, by imposing a blade metal temperature of 1140 K at cruise during all simulations (see section 3.6.2) and keeping that mass flow percentage constant throughout the flight envelope. This should keep the HPT blade metal temperature around the temperature limit of 1400 K at End-of-Runway conditions as reported in sections 2.1 and 3.6.2. Ideally, the HPT cooling flow would be set to a fixed value at cruise that ensures that the EOR limit is not exceeded. However, this was not possible due to the instability of the combined cycle model, so a simplification had to be applied.

TF2050_ITR

The TF2050_ITR engine does not include a bottoming cycle and consequently features significantly fewer design parameters. The TF2050_ITR is included here in order to identify the isolated potential to reduce the mission fuel burn of the reference TF2050 by adding a secondary combustor between HPT1 and HPT2. The performance of the TF2050_ITR was optimised twice using the Isight simulation environment. First, the optimisation was performed using a simplified off-design scaling approach (see section 3.7.3). Afterward, the dedicated optimisation of the fuel flow to both combustors at cruise, Top-Of-Climb (TOC) and End-Of-Runway conditions was performed. This way, the

simulation error introduced through the simplified off-design performance estimation procedure can be approximated.

Table 3-10: Inputs and their investigated range in simplified TF2050_ITR study

Component	Parameter	Range
Fan	BPR [-]	16 – 23
	FPR_{BP} [-]	1.35 – 1.5
	W_{in} [kg/s]	360 – 420
IPC	PR_{IPC} [-]	2 – 2.64
HPC	PR_{HPC} [-]	17 – 20
Reheat Combustor	TET_2 [K]	1340 – 1580
	WS_{HPT} [-]	0.45 – 0.55

The variables of the simplified optimisation setup are listed in Table 3-10. The work split between the first and second stage of the HPT at design point is only moderately varied, because the efficiency and design of the turbines would otherwise be very different. The pressure loss in the second combustor was fixed to 2 % for all simulations. The outlet temperature of the first combustor was kept constant at 1580 K during the simplified optimisation process and the cruise blade metal temperature is fixed during the simplified optimisation study, as in the combined cycle studies.

Table 3-11: Inputs and their ranges in detailed DP&OD study of TF2050_ITR

Component	Parameter	Range
Fan	BPR [-]	16 – 23
	FPR_{BP} [-]	1.35 – 1.5
	W_{in} [kg/s]	360 – 420
IPC	PR_{IPC} [-]	2 – 2.64
HPC	PR_{HPC} [-]	17 – 20
Secondary Combustor	$TET_{2,cruise}$ [K]	1200 – 1400
	$TET_{2,TOC}$ [K]	1400 – 1890
	$TET_{2,EOR}$ [K]	1400 – 1950
	WS_{HPT} [-]	0.45 – 0.55

The second study of the TF2050_ITR included a dedicated off-design performance deck, which allowed for the optimisation of the fuel flows to the two combustors at all three

simulated operating points (cruise, TOC and EOR). Table 3-11 includes the variables of the detailed optimisation study. Any parameter that does not include an index for the operating point was defined at cruise. The turbine entry temperature to HPT1 is not included in this list. The target solver, while matching the thrust requirements at the three operating points, determined the respective TET_1 values.

During the full design and off-design optimisation, the mass flow percentage that is used to cool the HPT is adjusted as outlined in section 3.6.2. The pressure losses in the secondary combustor were once again assumed constant at 2 %.

CCTF2050_ITR

The parametric studies of the combined cycle engine including inter-turbine reheat were performed using the same inputs as the CCTF2050, but with the addition of the outlet temperature of the second combustor (TET_2) and the work split of the two HPT stages. Table 3-12 presents all design inputs and their ranges applied during the simulations.

Table 3-12: Inputs and investigated ranges in parametric CCTF2050_ITR studies

Component	Parameter	Range
Main Engine	BPR [-]	16 – 23
	FPR_{BP} [-]	1.35 – 1.5
	W_{in} [kg/s]	390 – 420
	TET_2 [K]	1340 – 1580
	WS_{HPT} [-]	0.45 – 0.55
S-CO₂ Compressor	T_{in} [K]	309.15 – 315.15
	P_{in} [MPa]	8.07 – 8.57
	PR_{BC} [-]	2 – 4
Pre-cooler	$M_{fr,C}$ [-]	0.08 – 0.15
	N_{col} [-]	7 – 9
	$a_{tube,C}$ [m]	0.005 – 0.01
	x_{inv} [-]	0.9 – 1.0
	$W_{coolant}$ [kg/s]	35 – 50
MHX	N_{col} [-]	7 – 9
	$a_{tube,H}$ [m]	0.005 – 0.01
	x_{inv} [-]	0.9 – 1.0

The ranges of several design parameters have changed relative to the CCTF2050 engine. For example, the number of columns in the pre-cooler and the MHX has increased, because the higher LPT exhaust gas temperature significantly increases the heat input and subsequently the shaft power produced in the bottoming cycle.

3.7.2.2 Main Engine Turbomachinery Scaling

The HPC last stage blade height is very influential on the HPC component efficiency, as very small blades in the last stage will lead to large over-tip leakage (OTL) losses. The final compressor blade height of the TF2050 reference engine is approaching what is believed to be a sensible design limit, whilst avoiding significant losses due to manufacturing and assembly limitations. Assuming a typical HPC exit hub-to-tip ratio (H/T) of 0.92 and an axial exit Mach number of approximately 0.27, a last stage blade height of 17 mm was determined for the initial design of the reference engine.

The additional shaft power of the bottoming cycle, especially in the reheated configuration, will increase the engine's core specific power. The core mass flow can therefore be reduced, increasing the bypass ratio and aggravating the OTL problems in the HPC. Hence, it is necessary to estimate how the HPC efficiency would change when the engine core is scaled down, so as to adequately assess the performance of engine. An efficiency penalty function has been implemented based on the last stage blade height h_{ls} (mm), to estimate the effect engine scaling has on the HPC efficiency. The hub-to-tip ratio and exit Mach number of the HPC have been kept constant during the parametric studies to represent comparable technology levels, even though a slight redesign of the compressor might improve the performance overall. The penalty function (see equation 3-124) is a new correlation to fit the efficiency deltas presented by Kyprianidis and Rolt [144]. The method correlates the polytropic efficiency penalty directly to the last stage blade height based on the work of Lakshminarayana [145]. Half the calculated compressor penalty is also applied to the HPT polytropic efficiency, as small engine cores and resulting OTL also affect the performance of the HPT, but not quite as severely as the HPC, because the pressure gradient is working for the flow in a turbine.

$$\Delta\eta_{poly,HPC} = 0.038721 - 0.554 * h_{ls}^{-1} - 1.7724 * h_{ls}^{-2} \quad 3-124$$

This relationship was originally derived for a compressor design without active tip clearance control technology on the HPC (or the HPT) and might therefore be seen as a conservative approach to engine scaling for OTL in this instance. However, there are

further causes of efficiency loss when scaling-down the blading, which are not otherwise accounted for, so these effects are assumed to cancel out.

3.7.3 Off-design performance estimation

The combined cycle engine models are presently not stable enough to simulate off-design conditions even at high power conditions such as TOC or EOR. However, WeiCo needs a number of performance parameters at cruise, TOC and EOR in order to size the geared turbofan geometry of the main engine correctly. Additionally, the non-linear trade factors require the SFC of the engine at cruise, TOC and EOR, as the three are weighted to calculate a mission average SFC value (see equation 2-1 in section 2.1.2). Hence, a workaround had to be implemented to extrapolate the off-design performance of the combined cycle engines without running the current models at those points and to therefore enable mission level fuel burn analyses.

The implemented simplification of the off-design modelling is based on the assumption, that the relative change of most performance parameters from cruise to TOC or to EOR is constant, no matter what engine configuration is assessed. To implement this, the ratio of each WeiCo relevant parameter between TOC and cruise, and between EOR and cruise was calculated for the TF2050. The relative change in SFC was also captured in the same manner for the NLTFs. This means, that the relative SFC delta at cruise is almost equal to the deltas at TOC and EOR. These parameter ratios were subsequently included in the design point PROOSIS models of the advanced cycles, so the off-design data can be extrapolated based on the cruise data only. The method therefore enables the use of WeiCo and the NLTFs without dedicated off-design performance simulations.

This approach likely represents an oversimplification of the problem, as the considerable change in LPT and fan outlet conditions will directly affect the performance of the bottoming cycle in a way that could not be accounted for at this point. Theoretically, the higher fan outlet temperatures at TOC and EOR will mean, that the S-CO₂ compressor will not operate as close to the critical point as intended anymore, reducing the thermal efficiency of the bottoming cycle. However, the parametric studies in sections 4.2 and 4.3 show that the compressor inlet conditions of the most fuel burn efficient combined cycle engine configurations are no longer overly close to the critical point anyway, due to the performance and size limitations of the pre-cooler. The further departure from the critical point at off-design will therefore have a smaller adverse effect on whole engine performance. Additionally, the LPT exhaust gas temperatures (EGT) are also

considerably higher at TOC and EOR than at cruise, which should raise the thermal efficiency of the bottoming cycle again. The ratio of LPT EGT to fan outlet temperature actually increases from cruise to TOC and EOR in a modern turbofan. In case of the TF2050, the ratio changes from 2.25 at cruise to 2.58 at TOC and 2.46 at EOR. Hence, it is not unreasonable to assume that the useful power produced by the S-CO₂ bottoming cycle increases and the resulting SFC benefit stays approximately constant when moving from cruise to the high-power conditions. Another reason why the applied simplification of the off-design performance will only introduce a relatively small error to the fuel burn estimates originates from the mission profile of the aircraft. The advanced tube-and-wing aircraft is designed for long-range applications and consumes approximately 90% of its fuel during cruise. Any error in estimated climb or take-off SFC therefore only has a minor effect on the mission average SFC value that is used by the non-linear trade factors to determine the total mission fuel burn.

The off-design values of the individual engine parameters of the combined cycles calculated this way will not represent the values of a real combined cycle engine very well, but the calculated mission fuel burn benefit should be relatively reliable. The comparison of the simplified off-design approach when applied to the TF2050_ITR with the full design and off-design operation optimisation of the TF2050_ITR engine is presented in section 4.1 and confirms these assumptions.

3.7.4 Weight estimation

The total weight of any of the simulated engine configurations was estimated using a combination of tools and equation 3-124. The built-in design procedures of the S-CO₂ heat exchangers were used to determine their weight including the respective inlet and outlet manifolds. The weight of the power transmission system between the bottoming cycle and the main engine was assumed constant at approximately 25 kg as per section 3.5.3. That section also explains how and why the weight of the S-CO₂ turbomachinery components was fixed to 30 kg. An additional 45 kg was added to these estimates to account for the weight of the connecting ducts and pipes between all of the S-CO₂ components and the mass of the S-CO₂ itself, taking the 'fixed weight total' to 100 kg.

The weight of the more conventional engine components, such as the main engine turbomachinery and shafts, the power gearbox, the nacelle, accessories and more is estimated using WeiCo 9.6. WeiCo is an engine weight estimation and general arrangement prediction tool developed by Chalmers University. WeiCo is based on a

component based modelling approach similar to that of the Cranfield in-house weight estimation tool Atlas [146]. WeiCo has been shown to match the total engine weight of existing turbofan engines to within $\pm 5\%$ and was calibrated to account for the material and technology improvements of the reference engine TF2050 for the ULTIMATE project. WeiCo was hence chosen over Atlas as the weight estimation tool for this study.

$$Weight_{total} = Weight_{WeiCo} + 100 \text{ kg} + Weight_{S-CO2, HeatExchangers} \quad 3-125$$

Unfortunately, WeiCo showed some instabilities calculating the LPT weight, inexplicably jumping between around 95 and 400 kg estimates for virtually equal combined cycle engine designs. The $Weight_{WeiCo}$ parameter in equation 3-125 is therefore not the actual total engine weight calculated by WeiCo. Instead, the total engine weight estimate of WeiCo is reduced by the component weight of the LPT. Afterward, the LPT weight of the reference TF2050 is scaled with core mass flow and added to that difference, giving $Weight_{WeiCo}$.

The introduction of inter-turbine reheat would increase the component weight in a number of ways. The secondary combustor itself would need to be designed. The shafts and casings of the main engine would be elongated, which will result in problems with the rotor dynamics. The first stages of the low pressure turbine may have to be made from Nickel alloys rather than Titanium alloys as assumed for the TF2050. None of these effects are taken into consideration when estimating the total engine weight of the reheated engine configurations, adding to the uncertainty of the claimed fuel burn values. The dedicated design studies that would have to be carried out for the aforementioned components are outside the scope of this study. The sensitivity of the mission fuel burn values to incorrect weight predictions is easily estimated using the aircraft trade factors. Every 100 kg of additional engine weight adds approximately 0.2 % of mission fuel burn. This is only an approximation around the designs presented in section 4 though, as the trade factors are non-linear across the whole design space.

4 Results

The performance of the bottoming cycle is intrinsically tied to the design and performance of the heat exchangers, which in turn are dictated by the performance and structure of the main engine. This chapter presents the performance and preliminary design data gathered for all three investigated engine configurations using the modelling approach presented in chapter 3:

- the geared turbofan engine with inter-turbine reheat **TF2050_ITR**
- the geared turbofan engine with a supercritical CO₂ bottoming cycle **CCTF2050**
- the geared turbofan engine with a supercritical CO₂ bottoming cycle and inter-turbine reheat **CCTF2050_ITR**

The TF2050_ITR studies are presented first in section 4.1, to assess the potential of ITR to improve mission fuel burn without a waste heat recovery system. Section 4.1.1 describes the performance of the optimal TF2050_ITR as identified using the simplified modelling approach. Afterward, section 4.1.2 illustrates the engine's sensitivity to a selection of the three main design parameters: Bypass Ratio, HPT Turbine Work Split and TET₂. Finally, section 4.1.3 reports the optimal cycle parameters determined by using the more complex multi-point off-design performance methodology.

Section 4.2 follows a similar structure for the combined cycle engine without inter-turbine reheat. Section 4.2.1 specifies the detailed design inputs and performance parameters of the CCTF2050. The approximate and isolated sensitivities to the most influential parameters of the bottoming cycle are also specified around the identified optimal cycle to enable a broader appreciation of the final fuel burn estimate. Section 4.2.2 presents selected parametric and sensitivity studies. These parametric studies were used to find the fuel burn 'optimal' optimal CCTF2050 configuration. The data illustrates that the trends observed during these studies were highly non-linear even when close to the optimal design. The sensitivity studies also highlight the influence of selected modelling assumptions that were kept constant during the parametric studies, such as the surface finishing of the diffuser walls or inside the heat exchanger tubes.

Finally, the performance and general arrangement of the combined cycle engine with inter-turbine reheat (CCTF2050_ITR) is presented in section 4.3, including a selection of similar parametric and sensitivity studies as the CCTF2050 and the TF2050_ITR. Not all studies that were conducted are presented however, due to the qualitatively similar behaviour of the CCTF2050_ITR to the data presented for the CCTF2050.

4.1 TF2050 with ITR

Contrary to previous investigations into the introduction of inter-turbine reheat to modern turbofan engines [38], [39], it is unlikely that the TF2050_ITR will offer a fuel burn benefit over the reference TF2050 engine. Previous studies neglected the effects of core engine turbomachinery scaling and aimed at OPR increases during cruise, which are not possible in this case, because the same compressor delivery temperature limits were assumed for all engine configurations. Additionally, the TETs are limited to the same values as in the TF2050 and the exit temperature of the second combustor is not allowed to exceed that of the first. Few previous studies pursued the minimisation of mission fuel burn, with SFC reductions usually being the main objective instead. The TF2050_ITR was optimised for minimum fuel burn, despite the fact that it may lack some detail in terms of modelling the secondary air system or sizing the reheat combustor.

Overall, this section serves two main purposes. The primary objective is to establish the approximate fuel burn reduction potential of the technology using the same set of tools as during the assessment of the combined cycle engines. This enables the comparison of the individual contributions of the bottoming cycle and inter-turbine reheat in the TF2050_ITR and the CCTF2050, as well as the synergies of the two concepts when combined in the CCTF2050_ITR. Secondly, this section presents the difference in mission fuel burn estimates predicted using the simplified and the detailed off-design methodologies discussed in chapter 3. Both methods predict that the mission fuel burn will increase when ITR is introduced, because the additional pressure losses and cooling flow requirements cannot be offset by higher OPRs or smaller core engines. The comparison of the results of the simplified and detailed off-design performance methods indicates the approximate error introduced. The same error is assumed to be introduced when the simplified method is applied for the combined cycle engines.

4.1.1 Single-point Optimised Cycle with simplified OD model

The Isight optimisation using the simplified off-design method predicted a mission fuel burn penalty of 0.95% relative to the TF2050. Table 4-1 contains the main performance parameters that were determined optimal for the simple_TF2050_ITR engine model. The table lists two sets of net thrust and SFC for the engine. The design point at max cruise is the first of the two sets. That data is used to linearly scale the climb and take-off performance and subsequently size the engine using WeiCo (see section 3.7.3). The second set describes the mid-cruise (MC) condition, which was also scaled from the

design point performance. The mid-cruise data describes the usual operation of the engine and is used, in conjunction with the scaled climb and take-off SFCs, the engine weight and the fan diameter to estimate the mission fuel burn.

The OPR of the TF2050 was limited by the compressor delivery temperature of the HPC at cruise. The limit has not changed for the TF2050_ITR, because the same technology level is assumed for the HPC. The engine also does not feature intercooling, which would help to reduce the flow temperature in the HPC and therefore enable an OPR increase. Furthermore, the HPC and HPT polytropic efficiencies are reduced if the last stage blade height of the HPC is getting too small, mitigating much of the potential benefit gained from increasing the OPR of the engine.

Table 4-1: Performance of simple_TF2050_ITR

Parameter	Units	Max Cruise (DP)	Mid-Cruise
F_N	kN	37.4	32.6
SFC	g/(kN*s)	12.98	12.71
OPR	-	63.3	62.1
sF_N	m/s	87.4	85.6
BPR	-	16.36	16.46
BP $\Delta P/P$	%	1.12	1.12
FPR_{BP}	-	1.41	1.39
IPC PR	-	2.64	2.67
HPC PR	-	18.82	18.31
HPC η_{poly}	%	90.8	90.7
T₃₀	K	880	860
TET₁	K	1580	1540
TET₂	K	1347	1311
$\Delta P/P$ Comb₂	%	2	2
WS_{HPT}	-	0.45	0.45
d_{Fan}	m	2.83	
Engine Weight	kg	5125	
Δ FB_{y2050}	%	+0.95	
Δ FB_{y2000}	%	-45.34	

TET_2 and the HPT turbine work split (WS_{HPT}) define the TF2050_ITR. As expected, the fuel burn is at its minimum, if the reheat combustor is introduced as early in the expansion process as possible, leading to the minimum allowable HPT work split (45%). The second combustor is essentially turned off in the present configuration. This was also to be expected, as significant additional heat input in the second combustor would require a significant increase in downstream cooling flows. Additionally, the EGT would increase and the thermal efficiency drop, because the OPR of the engine could not be raised. Section 4.1.2 illustrates how the placement and operation of the second combustor affect the mission fuel burn estimate for the simple_TF2050_ITR.

It might have been expected that the optimal bypass ratio of the TF2050_ITR would exceed that of the TF2050. The specific core specific power describes the amount of shaft power the turbines can generate from a given core flow. Increasing the average flow temperature in the expansion process should reduce the core mass flow necessary to drive a given fan and increase the specific power. However, as the reheat combustor is effectively unlit at maximum cruise design conditions, it does not raise the specific power of the engine, as would have been necessary to facilitate an increase in BPR and a reduction in specific thrust. Furthermore, the additional pressure losses in the second combustor reduce the thermal efficiency and specific power of the core engine, favouring a slightly larger core mass flow and reduced BPR. Hence, the bypass ratio of the simple_TF2050_ITR is actually slightly lower than that of the TF2050. The larger core mass flow in turn leads to a small reduction in HPC efficiency penalty due to excessively small blades. Still, the SFC and hence the mission fuel burn are noticeably increased relative to the TF2050, because of the reduced thermal and propulsive efficiencies of the engine.

The total weight of the engine is reduced by approximately 36 kg relative to the TF2050, which is primarily due to the slightly smaller fan diameter of the engine. However, the weight increase of the shafts and the HPT casing due to the reheat combustor is not accounted for in the present models, which may reasonably offset this reduction. If so, the fuel burn penalty of the simple_TF2050_ITR relative to the TF2050 may rise by another 0.07 percentage points to 1.02 %.

Figure 4-1 illustrates the thermodynamic conditions inside core of the simple_TF2050_ITR at maximum cruise conditions (DP). Note that the T-S-diagram tentatively connects the operating conditions at the different stations of the engine linearly. Obviously, this does not represent the intermittent flow conditions realistically,

but is only done for illustrative purposes. This way, the turbine work split is highlighted more clearly, as the line to signify the expansion in HPT1 is noticeably shorter than that of HPT2. The reheat combustor has almost no effect on the flow conditions, aside from introducing a pressure loss.

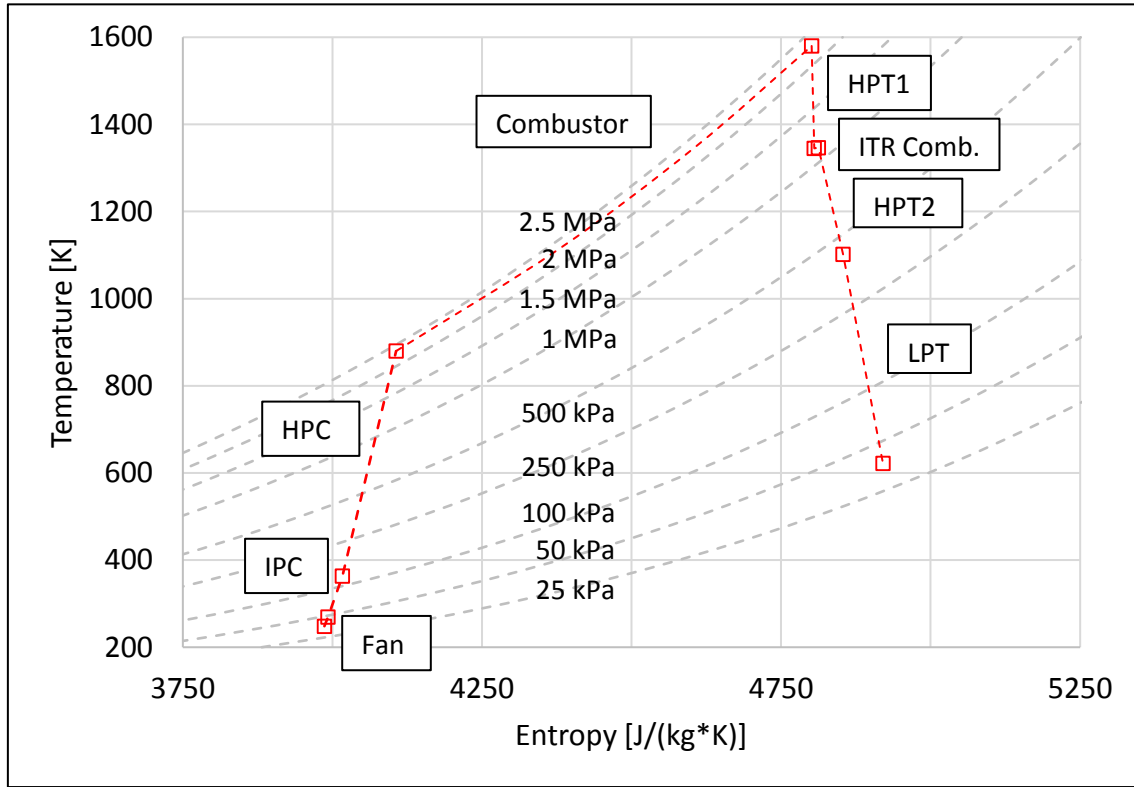


Figure 4-1: T-S-diagram of simple_TF2050_ITR at DP

4.1.2 Parametric and Sensitivity Studies

The current models determined that introducing inter-turbine reheat to the TF2050 does not yield any fuel burn improvements, due to the limited opportunities for the engine to harness the additional heat energy efficiently. Reheated engines would theoretically benefit from increased OPRs, but this is not possible here due to the temperature limitation for the HPC exit flow. Nonetheless, the engine configurations estimated to feature the lowest mission fuel burn are those that have a compressor delivery temperature (T_3) very close to the allowable limit in order to minimise the fuel burn.

This effectively eliminates OPR as one of the main design parameters of the simple_TF2050_ITR. Isight furthermore determined that the optimal IPC pressure ratio is always 2.64. The fan pressure ratio is de facto not an independent design parameter either, as the optimiser always finds the fuel-burn-optimal FPR for a given BPR. Hence,

the optimal HPC pressure ratio is dictated by FPR and T_3 and no longer an independent design variable.

Therefore, of the six design inputs given to the optimiser, only three are largely independent. This section outlines the influence of those three most influential design inputs: the bypass ratio, the turbine entry temperature to HPT2 (TET_2) and the work split of the HPT at design point (WS_{HPT}). The turbine entry temperature to HPT1 has been kept constant, as it would have been determined optimal by the optimiser anyway.

4.1.2.1 Bypass Ratio

Figure 4-2 shows a selection of the feasible individuals (blue points) evaluated by the optimiser. Only the cases closest to the fuel burn optimum are shown, to improve the readability of the graph. The red line represents a quadratic fit to the most fuel efficient cases and may be seen as an approximation of the design trend. The parabola actually has its minimum closer to BPR = 16.5 rather than BPR = 16.37 as for the optimal simple_TF2050_ITR, but such a deviation from the real optimum is within the expected range for a second order polynomial fit.

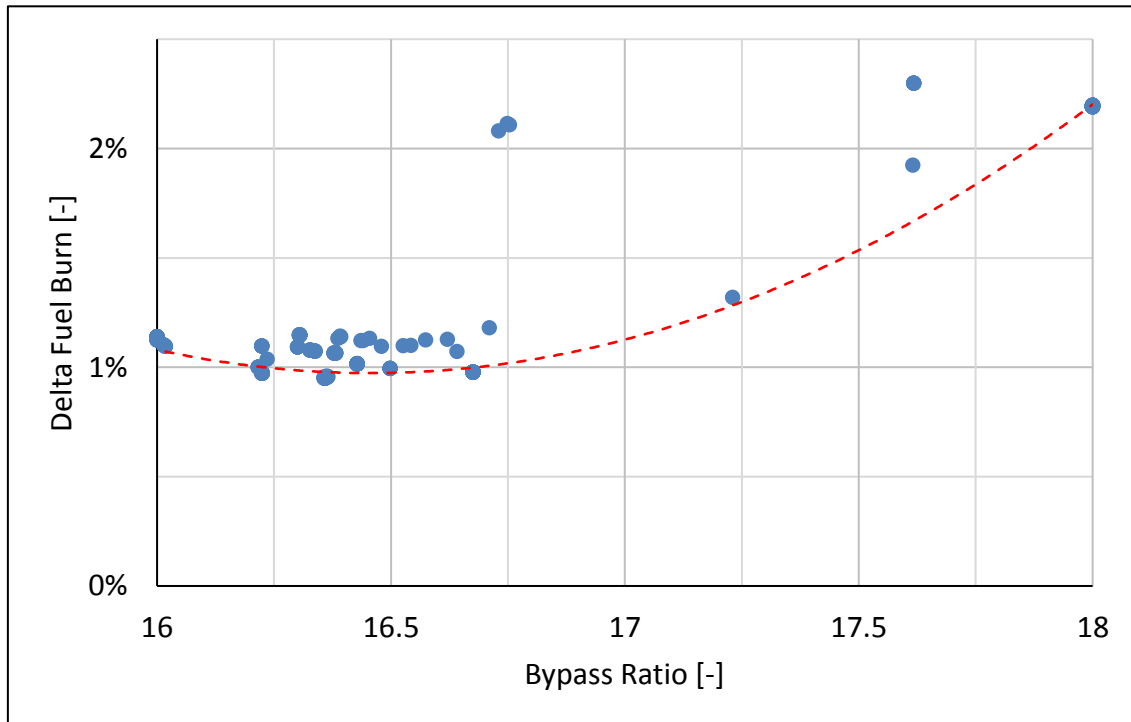


Figure 4-2: Influence of Bypass Ratio on simple_TF2050_ITR

Either way it is clear that the optimal BPR of the simple_TF2050_ITR is slightly lower than that of the TF2050. This is due to the unlit reheat combustor, which still entails a

pressure reduction between HPT1 and HPT2 as discussed before. The figure shows that the fuel burn penalty more than doubles if the BPR rises to 18, the maximum investigated in this study. If the BPR is reduced from the optimal 16.36 to 16 however, the fuel burn benefit only rises by about 0.15 percentage points. Overall, the fuel burn is only slightly affected by BPRs between 16 and 17, suggesting that other objectives such as noise or emissions might determine the final choice.

4.1.2.2 HPT Turbine Work Split

Figure 4-3 illustrates the effect the HPT work split has on mission fuel burn. Again, only configurations relatively close to the fuel burn minimum are depicted. The fuel burn penalty of the simple_TF2050_ITR over the TF2050 reduces continuously with the work split along the quadratic fit to the most fuel burn effective cases. The total variation within the investigated range is surprisingly large, as the fuel burn penalty of the most efficient configurations rises from 0.95% at $WS_{HPT} = 45\%$ to about 7% at $WS_{HPT} = 55\%$. It is important to note however, that these are the results of the optimiser, not of a parametric study. Hence, the optimiser pursued the global fuel burn minimum and did not evaluate the full range of possible configurations for every turbine work split. The design space is therefore not fully populated, which explains the distance of the evaluated cases from the quadratic fit at work splits above 48%.

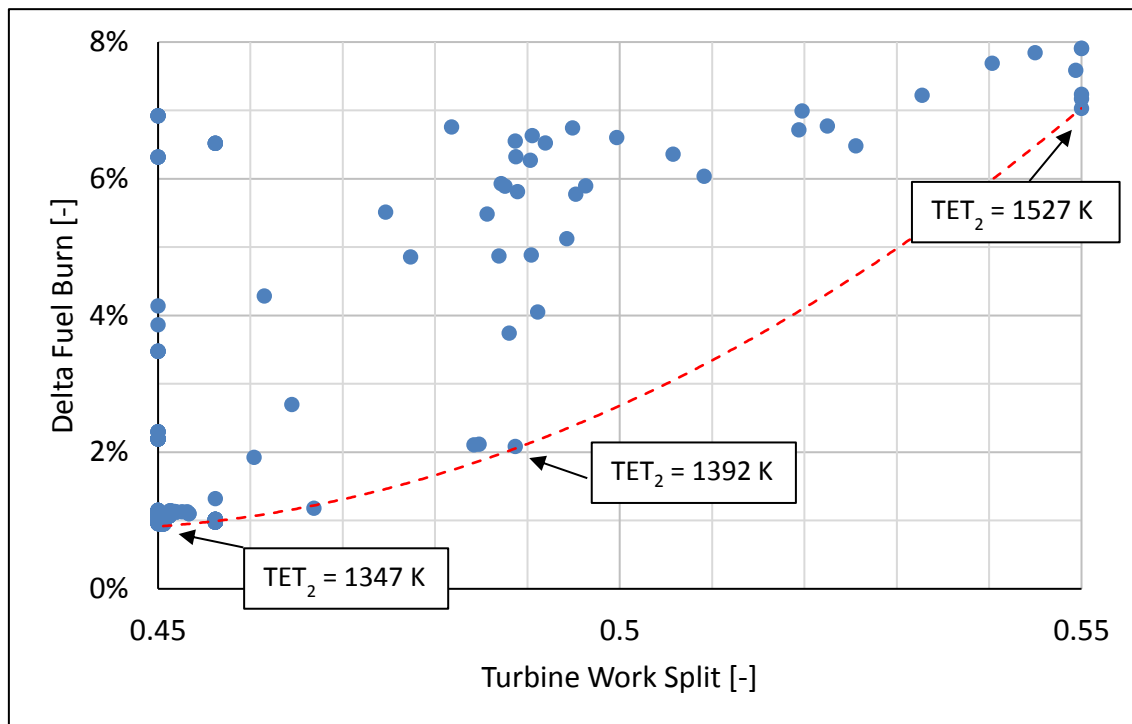


Figure 4-3: Influence of HPT Turbine Work Split on simple_TF2050_ITR

Furthermore, the fuel burn optimal configuration and any cycle close to the fitted optimal design line with work splits between 45 % and 47 % include an essentially unlit reheat combustor. If the work split is greater than 47 %, the fuel flow to the reheat combustor of the most fuel burn efficient configurations increases to offset the pressure losses in the reheat combustor this late in the expansion process and raise the specific power of the core engine. TET_2 rises from about 1347 K at $WS_{HPT} = 45 \%$ to about 1527 K at $WS_{HPT} = 55 \%$. The fuel burn optimal simple_TF2050_ITR features the lowest possible WS_{HPT} .

4.1.2.3 Turbine Entry Temperature to HPT2

The interdependency of the optimal WS_{HPT} and the optimal TET_2 complicates the assessment of their respective influences. Figure 4-4 depicts the observed influence of TET_2 on the mission fuel burn penalty suffered by the simple_TF2050_ITR. The fuel burn penalty is clearly reduced if TET_2 is reduced, with the most fuel burn efficient engines featuring essentially no temperature rise in the reheat combustor at all. The trend is close to linear as illustrated by the fit to the most fuel burn effective cycles in the investigated range.

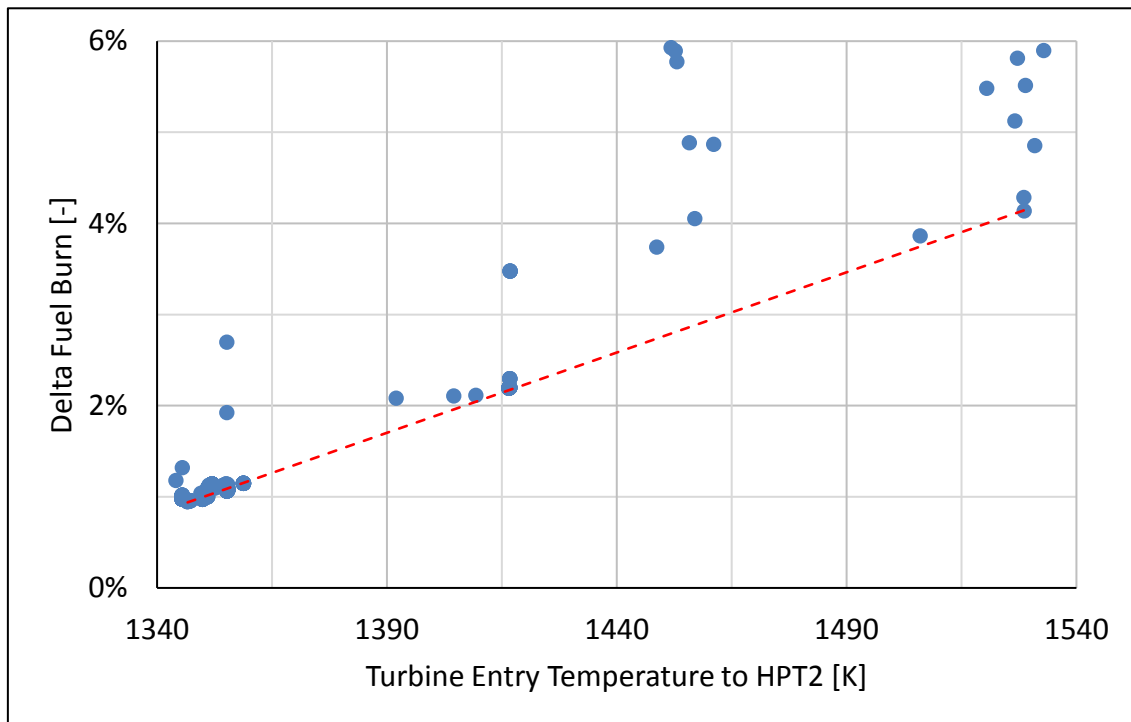


Figure 4-4: Influence of TET of HPT2 on TF2050_ITR

The graph does not include the temperature range between 1540 K and 1580 K even though it was part of the simulation. This is due to the fact that no cycle with such an

input was investigated by the optimiser, as the fuel burn trends clearly showed that little or no temperature rise would be more beneficial. The fuel burn penalty rises from the minimal 0.95 % to at least 4.1 % if the TET of HPT2 is raised to approximately 1530 K.

4.1.3 Multi-point Optimised Cycle

The TF2050_ITR was optimised a second time using the more detailed off-design modelling approach described in section 3.7.3. The resulting cycle was dubbed detailed_TF2050_ITR. Beside the maximum thrust cruise condition, the performance of the engine was specifically simulated at three off-design points: Mid-Cruise, Top-of-Climb and End-of-Runway. The performance at these three points was used to estimate the mission average SFC (see equation 2-1 in section 2.1.2), which in turn was used in combination with the non-linear aircraft trade factors to estimate the fuel burn. The engine performance at max cruise was used in conjunction with high-power Top-of-Climb and End-of-Runway conditions to determine the engine size and weight using WeiCo.

Table 4-2 displays the most important performance parameters of the engine at the design point and the three operating points used to determine the mission average SFC. The detailed optimisation determined that the bypass ratio of the engine should be lower than that of the TF2050 in order to increase the specific thrust and reduce engine weight and drag. The fan diameter is about 12 cm smaller than that of the simple_TF2050_ITR. The engine is furthermore about 403 kg lighter than the previous TF2050_ITR.

However, the HPC efficiency penalty is greater in this configuration than in the simple_TF2050_ITR presented in section 4.1.1, because the core mass flow is even smaller. The OPR of the engine is no longer dictated by the compressor delivery temperature alone. The exponential HPC penalty function means that the fuel burn is minimised if the OPR is slightly reduced. The OPR reduction is split between the IPC and HPC pressure ratios in the present configuration.

The more detailed off-design methodology has not changed the optimal turbine work split. The optimal temperature rise at max cruise conditions has been changed however, which results in a considerable increase in maximum cruise thrust available to the engine. On the other hand, the temperature rise in the reheat combustor of this engine is virtually non-existent during normal operation at mid-cruise conditions, same as the engine reported upon in section 4.1.1.

Table 4-2: Performance of detailed_TF2050_ITR

	Units	Max Cruise (DP)	Mid- Cruise	Top-of- Climb	End-of- Runway
F_N	kN	39.9	32.4	49.8	181.7
SFC	g/(kN*s)	13.20	12.83	13.88	8.55
OPR	-	61.4	59.3	71.5	57.0
sF_N	m/s	99.7	92.3	120.7	194.5
BPR	-	16.0	16.3	15.6	15.9
BP $\Delta P/P$	%	1.12	1.12	1.12	1.12
FPR_{BP}	-	1.48	1.44	1.57	1.44
IPC PR	-	2.6	2.4	2.3	2.2
HPC PR	-	18.1	18.8	22.3	20.0
HPC η_{poly}	%	90.7	90.7	89.5	90.6
T₃₀	K	873	851	992	1034
TET₁	K	1580	1569	1890	1914
TET₂	K	1432	1347	1655	1699
WS_{HPT}	-	0.45	0.47	0.46	0.46
d_{Fan}	m		2.71		
Total Weight	kg		4722		
ΔFB_{y2050}	%		+1.17		
ΔFB_{y2000}	%		-45.22		

Table 4-3 compares the main engine parameters of the simple_TF2050_ITR and the detailed_TF2050_ITR. Only the mission average SFC, the fan diameter and the total engine weight are listed in the table, as they are the direct inputs to the non-linear aircraft trade factors. The differences with regard to more detailed performance parameters such as temperatures at certain stages will be even less comparable due to the difference in modelling approach and are therefore not shown. The fuel burn penalty of the TF2050_ITR over the TF2050 has increased from 0.95 % to 1.17 %. The increased penalty is primarily due to the increased SFCs at the three off-design points. The SFC at mid-cruise is approximately 1% higher than that predicted by the simplified off-design method, despite the fact that TET_1 and TET_2 are individually optimised in the more detailed setup. The reduced drag due to the smaller fan and the reduced engine weight offset much of the SFC penalty and are the reason why this engine's fuel burn benefit does not differ too much from the result of the simplified approach.

Table 4-3: Comparison of main parameters of simple and detailed TF2050_ITR

	Units	simple_TF2050_ITR	Detailed_TF2050_ITR
SFC_{ave}	g/(kN*s)	12.74	12.85
d_{Fan}	m	2.83	2.71
Total Weight	kg	5125	4722
Δ FB_{y2050}	%	+0.95	+1.17
Δ FB_{y2000}	%	-45.34	-45.22

Some of the component and performance parameters presented in Table 4-2 at off-design differ significantly from those predicted by the simplified approach. As a result, only the design point data is presented for the combined cycle engines, so as to minimise the introduced error. Nonetheless, the fuel burn penalty estimated by the different approaches differs only slightly and it is assumed that the same margin of error is sustained for the combined cycle engines.

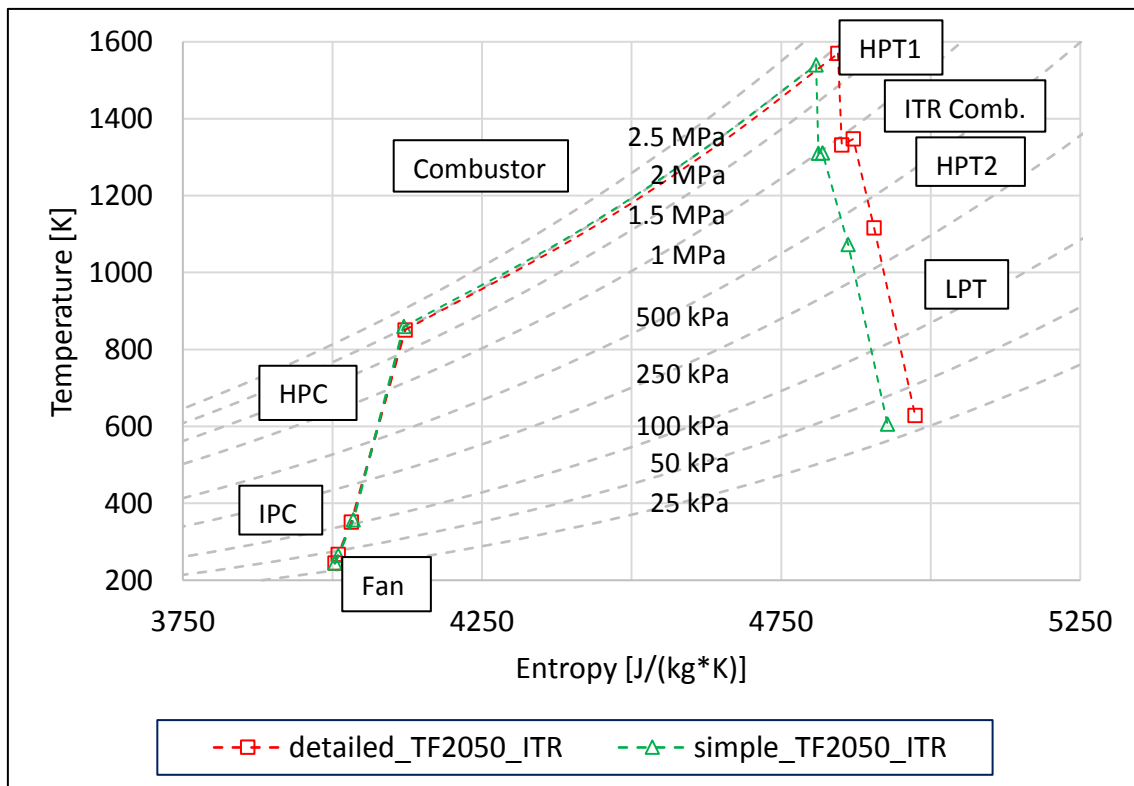


Figure 4-5: T-S-Diagram of the TF2050_ITR engines at mid-cruise, identified by the simplified (green) and the detailed (red) off-design method

The T-S-diagram in Figure 4-5 illustrates the mid-cruise operation of the two TF2050_ITR configurations rather than max cruise conditions, because of the significantly different maximum thrust ratings of the simple_TF2050_ITR and the detailed_TF2050_ITR. The only noticeable difference in the two cases arises from the higher TET_1 in the engine found by the detailed off-design method (red line). The whole expansion process occurs at a higher mean temperature. Unfortunately, the engine cannot capitalise on the additional heat energy properly, because the OPR of the engine could not be increased. Consequently, the LPT EGT of the detailed_TF2050_ITR is higher, worsening the thermal efficiency of the cycle and increasing the specific fuel consumption.

4.2 CCTF2050

This section outlines the performance and preliminary design details of the CCTF2050 combined cycle engine without inter-turbine reheat. The final configuration was determined using multiple rounds of parametric design studies as described in chapter 3. The final cycle will be referred to as the fuel-burn optimum throughout this section despite the fact that no automated optimisation algorithm has been employed during its identification. The details of the optimal CCTF2050 are presented in section 4.2.1. A selection of parametric and sensitivity studies is presented in section 4.2.2.

Distinction between sensitivity studies and parametric studies

Sensitivity studies are commonly used to illustrate how minor changes in modelling assumptions, often related to manufacturing techniques, may affect the overall performance of an engine. These dependencies are frequently evaluated through minor changes (usually $\pm 1\%$) and the assumption that the trends would be linear in the applicable range. The sensitivities of this engine are depicted over a wider range due to the strong non-linear and sometimes discontinuous behaviour of the trends. Additionally, the investigated range is larger, because the uncertainty with regard to realistic manufacturing limits is significantly greater, considering the 30 year prelude to the engine possibly going into production. The larger range makes the sensitivity studies more akin to common parametric studies, but parametric studies investigate the influence of direct design parameters such as the number of tubes in a heat exchanger for example.

Parametric studies in this work may investigate one or multiple parameters at a time. Often they depict the optimal result for each investigated point (e.g. Pre-cooler inlet Mach number) of the parameter in terms of compressor inlet conditions and coolant mass flow as well. That way, a more complete description of the parameter's influence on the

optimal fuel burn is provided rather than just showing isolated trends. The specific conditions and varied parameters will be clearly outlined in the respective sections.

The bottoming cycle: Useful Power versus Thermal Efficiency

The trends depicted in the following sections are very specific to aero application and even to this specific engine. Previous studies on S-CO₂ power cycles for nuclear or CSP applications focused on maximising the cycle's thermal efficiency within the physical boundaries of their respective construction sites. The compressor inlet conditions of these cycles are therefore often very close to the critical point (if they are envisioned as Brayton and not Rankine cycles). In the present work, the space available for the bottoming cycle components is significantly smaller and hence more influential on the performance. This is why the combined cycle engine's performance had to be determined using preliminary design tools rather than component level estimates. The primary drivers of the bottoming cycle's potential to enhance the performance of the main engine are therefore the pressure losses across the heat exchangers and the bypass duct, and the generated shaft power. The thermal efficiency of the bottoming cycle is only a secondary priority.

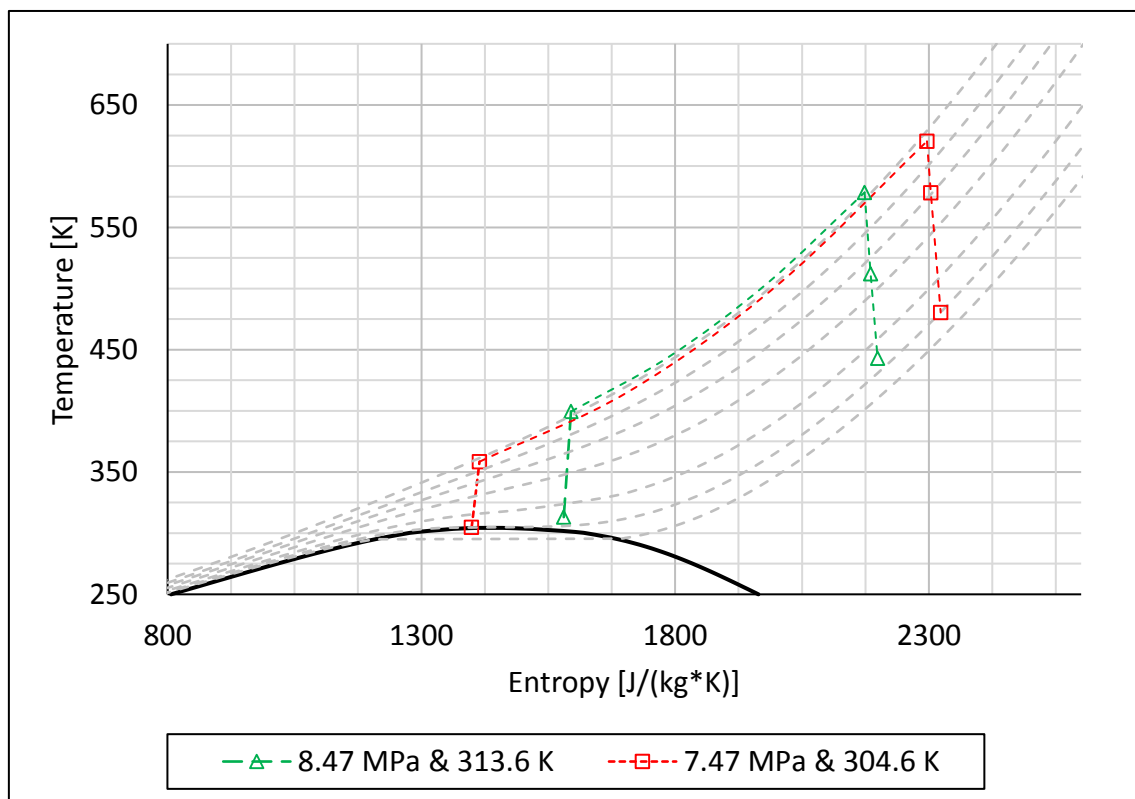


Figure 4-6: Comparison of fuel-burn optimum and thermal efficiency optimal operating conditions of the same bottoming cycle

Figure 4-6 illustrates the implications of this very clearly. The figure shows the T-S diagrams of the same combined cycle engine with the sole distinction of different inlet conditions to the S-CO₂ compressor. The green lines signify the operating conditions of the bottoming cycle of the fuel-burn optimal CCTF2050 presented in section 4.2.1. The compressor inlet conditions of that cycle (8.47 MPa and 313.1 K) differ noticeably from the critical point (7.38 MPa and 304.13 K), the theoretically optimal inlet conditions for an S-CO₂ Brayton cycle in terms of thermal efficiency. The red line illustrates the operating conditions of the same bottoming cycle (in terms of heat exchangers and turbomachinery design parameters) operating with compressor inlet conditions (7.47 MPa and 304.6 K) considerably closer to the critical point.

The 'red cycle' leads to a considerably higher thermal efficiency of the bottoming cycle (20.5 % vs. 19.3 %). The pressure losses across the heat exchangers are almost identical due to the same physical dimensions of the components. For this and all subsequent simulations, the coolant mass flow is a design input and the corresponding S-CO₂ mass flow is calculated. In this case, the coolant mass flow is fixed to 38 kg/s for both cycles. The reduced variation in specific heat capacity of the S-CO₂ in the pre-cooler, due to the departure from the critical point, means that a greater mass flow can be cooled to the 'green cycle's' inlet conditions. Hence, the shaft power generated by the green cycle is approximately 12.4 % higher than that of the red cycle (634 kW vs. 564 kW) despite it not utilising the real gas behaviour of the S-CO₂ in the compressor as much. The 'green cycle' therefore enables a fuel burn reduction of about 3.17 % relative to the TF2050, while the 'red cycle' only allows for a 2.08 % reduction.

Operating the bottoming cycle even further from the critical point would not be beneficial however, as the tubes of the main heat exchanger would require a different, heavier internal support structure to abide by the required factor of safety. The optimal compressor inlet conditions of the CCTF2050 are therefore dependent on a range of design inputs. Overall, the ideal inlet conditions for the S-CO₂ compressor will be a compromise of multiple effects. The power consumption of the compressor is reduced the closer the inlet conditions are to the critical point, but the S-CO₂ mass flow and therefore the power output are reduced simultaneously, because the variation of the specific heat capacity of the S-CO₂ in the pre-cooler increases. Section 4.2.2 will illustrate a number of such trade-offs and explain how the optimal compromise was determined.

4.2.1 Optimal Cycle Performance and Design

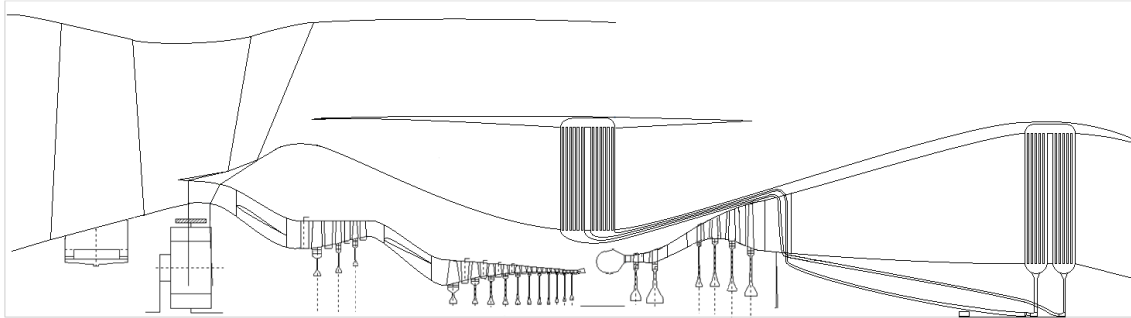


Figure 4-7: Scaled engine cross-section of CCTF2050

Figure 4-7 depicts the engine cross-section of the CCTF2050 with all components to scale. Several points need to be noted with regard to the cross-section. The extreme pressures and relatively small S-CO₂ mass flows in the bottoming cycle lead to very small turbomachinery components, even if the inlet and outlet Mach numbers are as low as, or below, 0.2. The outer diameter of the centrifugal compressor for example is only approximately 4.2 cm, if the outlet blade height is 7 mm and the outlet Mach number is 0.05. The relative size when shown together with the main engine is why the S-CO₂ compressor (on the far right) and the two S-CO₂ turbines (to the left of the compressor) are barely visible.

Furthermore, the connecting ducts between the heat exchangers and the turbomachinery components were not designed in the present investigation. The depicted ducts and pipes are therefore only representative and do not reflect actual component designs. The S-CO₂ ducts and pipes will also have a very small diameter, which should help to simplify the installation. For example, the pipes to and from the pre-cooler could pass through different support struts behind the LPT. That way they do not have to cross-over as suggested in the illustration and any heat transfer between them could also be avoided.

Finally, the shape of the inlet to the pre-cooler duct is obviously not optimised. The dimensions arise from the implemented methodology that differentiates only between the Mach number at the hub of the FOGV and at the tip. A more differential analysis of the flow field out of the FOGV would undoubtedly lead to a more aerodynamic design.

Table 4-4 contains the most important performance and preliminary component design parameters of the optimal CCTF2050 identified during the parametric design studies. If not specified otherwise, all data is shown for maximum cruise thrust conditions. The

specific fuel consumption of the identified engine is approximately 2.7% lower than the SFC of the TF2050. The CCTF2050 also features a 5 cm smaller fan diameter and weighs approximately 344 kg more than the reference engine. The CCTF2050 therefore uses about 1.5 t or 3.17 % less fuel on a 7000 nm mission. The mission fuel burn is reduced by about 47.6% relative to the baseline LRTF2000.

Table 4-4: Performance and component parameters of optimal CCTF2050

Parameter	Units	Value	Parameter	Units	Value
F_N	kN	37.1	BC Weight	kg	526
F_N Mid Cruise	kN	32.4	MHX N_{col}	-	6.5
SFC	g/(kN*s)	12.53	MHX M_{fr}	-	0.1
SFC Mid Cruise	g/(kN*s)	12.27	MHX a_{tube}	mm	7.5
OPR	-	63.1	MHX x	-	0.775
sF_N	m/s	88.8	MHX x_{inv}	-	1
BPR	-	16.8	MHX $\Delta P/P_{air}$	%	6.4
BP $\Delta P/P$	%	1.15	MHX $\Delta P/P_{S-CO2}$	%	0.5
FPR_{BP}	-	1.41	MHX ΔT_{air}	K	127
IPC PR	-	2.64	MHX M	kg	216
HPC PR	-	18.7	Pre N_{col}	-	7
HPC η_{poly}	%	90.7	Pre M_{fr}	-	0.089
T_{30}	K	880	Pre a_{tube}	mm	7.5
TET_1	K	1580	Pre x_{inv}	-	0.94
BC $T_{in,Cmp}$	K	313	Pre $\Delta P/P_{air}$	%	7.6
BC $P_{in,Cmp}$	MPa	8.47	Pre $\Delta P/P_{S-CO2}$	%	0.5
BC PR	-	4	Pre ΔT_{air}	K	69
BC W_{S-CO2}	kg/s	11.9	Pre M	kg	211
BC Pwr	kW	634	Pre W_{cool}	kg/s	38
d_{Fan}	m				2.79
Weight	kg				5505
ΔFB_{y2050}	%				-3.17
ΔFB_{y2000}	%				-47.57

The thrust requirements of the engine were slightly reduced relative to the TF2050 due to the reduced aircraft weight and drag. The overall pressure ratio of the engine is similar to that of the TF2050, as it is constrained by the same compressor delivery temperature

limit. Lower OPRs have delivered an increase in mission fuel burn in the current models. The specific thrust of the CCTF2050 is slightly increased relative to the TF2050 (from 86.5 to 88.8 m/s), despite the reduction in core exhaust gas temperature. The higher fan pressure ratio and the increased temperature of the air exiting the pre-cooler are the main reasons for this increase.

The bottoming cycle provides about 634 kW of useful power to the LP shaft of the main engine, which represents about 4.7 % of the total power produced by the low pressure system. Using the methods described in chapter 3, the bottoming cycle is estimated to weigh approximately 526 kg in total, of which the pre-cooler and MHX account for about 40 and 41% respectively.

The heat exchanger pressure losses stated in Table 4-4 include the losses in the diffusers to the heat exchanger matrices. The pressure losses of about 6.4% for the pre-cooler and 7.5 % for the MHX match the losses calculated by Zhao et al. for the involute spiral intercooler [28], [88], [122] well. The internal pressure losses appear relatively low if listed in relative terms, but the absolute figures match Zhao's data well again. The relative pressure losses appear small due to the very high pressure levels in the system.

Overall, the design parameters of the MHX and the pre-cooler are within the expected range, with the possible exception of the tube chord lengths and the inlet Mach numbers.

The simulations determined that a chord length of 7.5 mm would be optimal for the elliptical tubes. However, this may be close to or just below what can be manufactured in a reliable and cost-effective way, especially given the complex internal structure of the tube (see section 2.5.2.4 or [92], [101]). The elliptical tubes are reinforced by five internal support struts, resulting in tiny channels inside the tubes. It is unlikely that these tubes could be manufactured at a competitive cost today. However, further advances in additive manufacturing for example may enable such designs by 2040 – 2045, when these parts would have to be produced and tested, if they were to enter into service by 2050. Section 4.2.2.8 investigates the implications of different chord lengths in more detail.

The inlet Mach number of the pre-cooler coolant flow is furthermore relatively low at 0.089. Many intercooler studies assumed inlet Mach numbers between 0.1 and 0.15 [122], though designs with $M = 0.05$ [124] have also been reported. For the purposes of this study, the identified value of 0.089 seems reasonable, given that weight and bypass pressure loss implications have been accounted for in the models.

Figure 4-8 shows the T-S-diagram of the core of the optimal CCTF2050. The most notable differentiator of the graph from that of a conventional turbofan are the temperature and entropy reductions after the LPT due to the heat transfer and pressure loss in the MHX. Part of the respective energy has been used to generate the 634 kW of further useful power in the bottoming cycle and the remainder has been rejected to part of the bypass flow by the pre-cooler.

The mission fuel burn benefit of the CCTF2050 depends strongly on the modelling assumptions and the design inputs chosen. Table 4-5 indicates the sensitivity of the CCTF2050's performance on isolated changes of most technology assumptions and design inputs. The relative changes of the individual design parameters are not as small as in many other sensitivity studies, where the influence of changes between 1 – 2 % are reported. Instead, larger changes are applied that more adequately represent a large part of the design space or possible manufacturing limitations. The effect of each of the parameters was investigated in isolation, with all other parameters being equal to the CCTF2050 data shown in Table 4-4. The bypass ratio is the only exception to that, as fan pressure ratio and bypass ratio are intrinsically linked and always designed together. The fuel burn optimal fan pressure ratio is therefore specified for each bypass ratio.

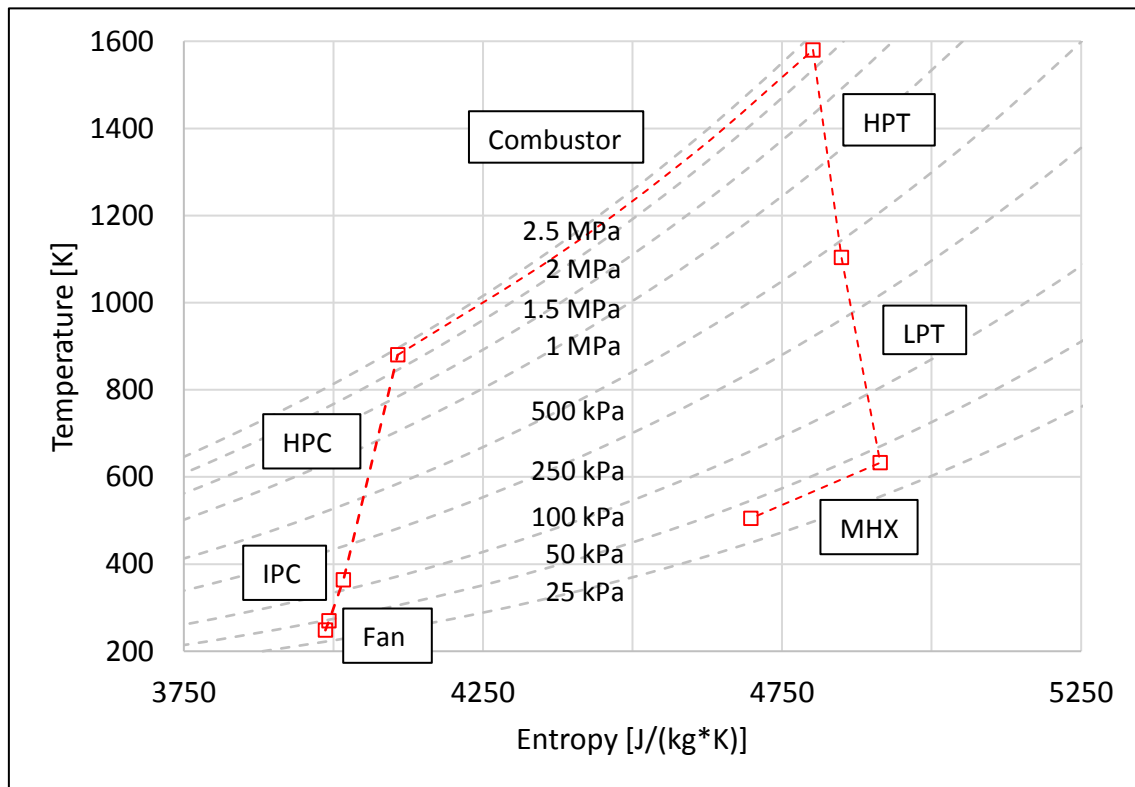


Figure 4-8: T-S-diagram of optimal CCTF2050 at DP

It is important to note that the presented values are specific to the changes of the individual design parameters specified in the table and they should not be extrapolated. The relative deviations should also not be used to linearly interpolate smaller changes, as the trends are rarely linear. Furthermore, the individual deviations of two or more parameters should not simply be added to assess their combined effect.

The non-linear trends observed during the investigations, some of which are outlined in section 4.2.2, mean that simple addition of deviations will introduce considerable uncertainty. For example, the pre-cooler inlet Mach number may be 0.1 instead of 0.089 and the coolant mass flow might be limited to 34 kg/s instead of 38 kg/s for some reason. Simply adding the two individual differences listed in Table 4-5 would suggest that the fuel burn benefit would be reduced by 0.58 percentage points. However, when applied to the CCTF2050 model, the fuel burn benefit is actually reduced by 0.64 percentage points. Hence, the simple addition of effects underestimated the real effect by about 10%. The fuel burn benefit may be improved again if other design inputs such as the number of pre-cooler tube columns or the compressor inlet conditions were changed, but those effects are not captured by the sensitivity data in Table 4-5.

Table 4-5: Selection of sensitivities of the CCTF2050

Component	Parameter	Units	Value	Δ FB [%]
Main Engine	BPR	-	16.25 / 17.75	+0.01 / +0.12
S-CO₂ Compressor	T_{in}	K	311 / 314	+0.03 / +0.04
	P_{in}	MPa	8.27 – 8.57	+0.05 / +0.02
	PR_{BC}	-	3 / 4	+0.4 / +0.0
Pre-cooler	$M_{fr,C}$	-	0.08 / 0.1	+0.06 / +0.35
	N_{col}	-	6 / 8	+0.07 / +0.05
	$a_{tube,C}$	m	0.006 / 0.01	+0.18 / +0.18
	x_{inv}	-	0.9 / 1.0	+0.0 / +0.04
	$W_{coolant}$	kg/s	34 / 42	+0.23 / +0.08
	L_{diff}	m	0.6 / 1.5	+0.67 / -0.1
	N_{col}	-	5.5 / 7.5	+0.13 / +0.01
MHX	$a_{tube,H}$	m	0.006 / 0.01	+0.18 / +0.18
	x_{inv}	-	0.9 / 1.0	+0.05 / +0.0
	L_{diff}	m	0.6 / 1.5	+0.58 / -0.1

The ranges that were applied in these cases differ for every parameter, making a direct comparison of their effects difficult, though for most variations between 10 and 20 % were investigated. Smaller ranges were investigated if a parameter showed particularly strong non-linear behaviour. For example, the effects of varying the compressor inlet temperature and pressure seem relatively small, but this is only the case when close to the optimal point. For greater deviations, the gradient becomes considerably stronger.

In most cases, the investigated range was chosen to cover a more significant portion of the total design space of the parametric studies. The ideal numbers of columns in the heat exchangers were reduced by a whole column around their respective optima to illustrate a significant change in heat duty. The coolant mass flow was varied by approximately $\pm 10\%$ as a variation of $\pm 1\%$ would only signify a change of about 0.4 kg/s, a negligible change that would be difficult to ascertain with the available engine control systems anyway. The chord length of the heat exchanger tubes is very influential on overall performance. 6 mm was chosen as the minimally investigated chord length for the sensitivity table, because it is unlikely that smaller tubes with such a complex internal structure could be manufactured in a reliable and cost-effective way. On the other hand, the parametric studies showed that smaller chord lengths lead to fuel burn reductions, as long as the internal structure does not get too complex. It would therefore be unreasonable to depict the influence of a chord length greater than about 10 mm, given that the ideal chord length was determined to be 7.5 mm.

Overall it must be said that most of the listed parameters have a significant influence, but the coolant mass flow, the compressor inlet temperature and the compressor inlet pressure are three of the most important design inputs. Ultimately, the pre-cooler is the component that dictates the performance of the bottoming cycle the most, outside the turbomachinery efficiencies of course, as it directly dictates the S-CO₂ mass flow, the bypass pressure losses and the total engine weight.

4.2.2 Parametric and Sensitivity Studies

The parametric design and sensitivity studies presented in this section were conducted around the optimal CCTF2050 configuration presented in Table 4-4. The gradients of the depicted trends differ depending on how far away from the optimal configuration the individual study may be conducted. Hence, some of the fuel burn trade-offs may seem rather small, but the trends are stronger and more definitive when identifying the optimal design from a wider design space. Depicting the effects across the whole design space

rather than just around the optimal combination would have involved too many interdependent influences too depict in a comprehensive way in this section.

For example, an increase in the number of tube columns in the pre-cooler means that the heat duty of that heat exchanger increases. Depending on how close to the optimal compressor inlet conditions the cycle is operated, this may lead to different further design changes. The minimum fuel burn objective may push the design to a lower compressor inlet temperature and pressure, increasing the power output and thermal efficiency of the bottoming cycle. Alternatively, the S-CO₂ mass flow may just be increased and the compressor inlet conditions might be kept constant, leading to a higher power output but a lower thermal efficiency of the BC. The non-linear trade factors allowed the identification of the fuel-burn optimal option in any such case.

4.2.2.1 Internal Tube Geometry

Rengel [100] contributed to this work through the identification of feasible internal tube geometries for the high aspect ratio elliptical tubes of the pre-cooler and the main heat exchanger (MHX). His results were integrated in the combined cycle evaluation process as described in section 3.4.1.3. Due to the large number of feasible designs identified by Rengel, it is uneconomical to evaluate every feasible design for every cycle configuration. It was therefore necessary to determine the relative importance of heat transfer, pressure drop and component weight when pursuing minimal mission fuel burn. It can be expected that the weight of the tubes is the most important of the three objectives. The heat transfer of the S-CO₂-to-air heat exchangers is primarily limited by the low air-side heat transfer coefficients and the total pressure losses vary very little if the design is approximately in the right range, given the high overall working pressures.

Table 4-6: Common bottoming cycle parameters

Parameter	Units	Values	Parameter	Units	Values
BC $T_{in,Cmp}$	K	307.6	Pre x_{inv}	-	1.0
BC $P_{in,Cmp}$	MPa	7.87	MHX N_{col}	-	6
BC PR	-	4	MHX M_{fr}	-	0.1
Pre W_{cool}	kg/s	26	MHX a_{tube}	mm	6.3
Pre N_{col}	-	7	MHX x	-	0.775
Pre M_{fr}	-	0.1	MHX x_{inv}	-	1
Pre a_{tube}	mm	6.2			

A case study was conducted to confirm these expectations, where a wider selection of feasible designs was applied to a combined cycle with the bottoming cycle parameters listed in Table 4-6. The corresponding mission fuel burn was estimated using the engine and aircraft models described in chapter 3. The four designs with the lowest solidity (hence lowest weight), the lowest internal hydraulic diameter (hence highest heat transfer coefficients) and the highest internal hydraulic diameter (hence lowest pressure loss) were each filtered from the design space, giving 12 designs per heat exchanger. Hence, 144 different heat exchanger combinations were evaluated in total (see Figure 4-9 for simplified illustration of how designs were combined).

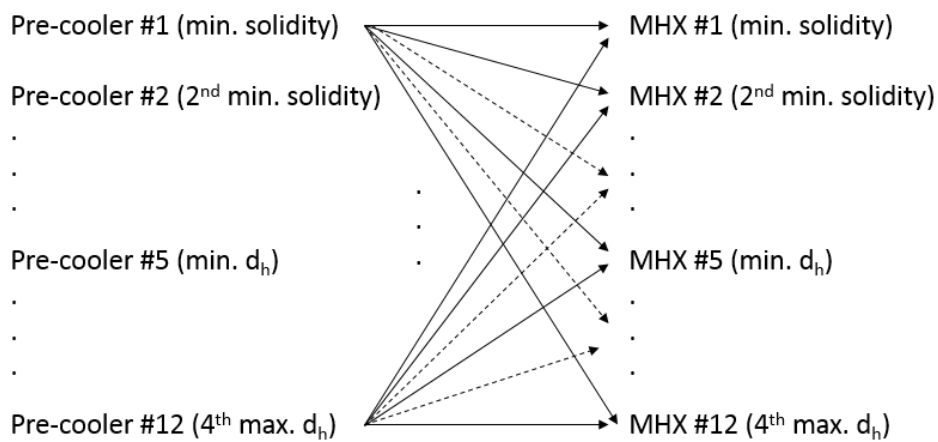


Figure 4-9: schematic of how different tube geometries were combined

Figure 4-10 and Figure 4-11 depict the dependencies of the mission fuel burn estimates on the pre-cooler and MHX internal tube geometries. Figure 4-10 illustrates the influence of the internal hydraulic diameter and the solidity of the pre-cooler tubes (fraction of solid material of the total cross-sectional area) on mission fuel burn. Twelve different fuel burn estimates are calculated for each investigated combination of solidity and hydraulic diameter in the pre-cooler tubes, because twelve different MHX configurations were combined with each of the pre-cooler cases. Hence, there are 12 vertically aligned blue points per pre-cooler configuration in the plot.

The lowest point of each of these columns was used to depict an optimal response surface. These twelve most fuel burn efficient combinations correspond to the lightest possible tube geometry of the MHX tubes.

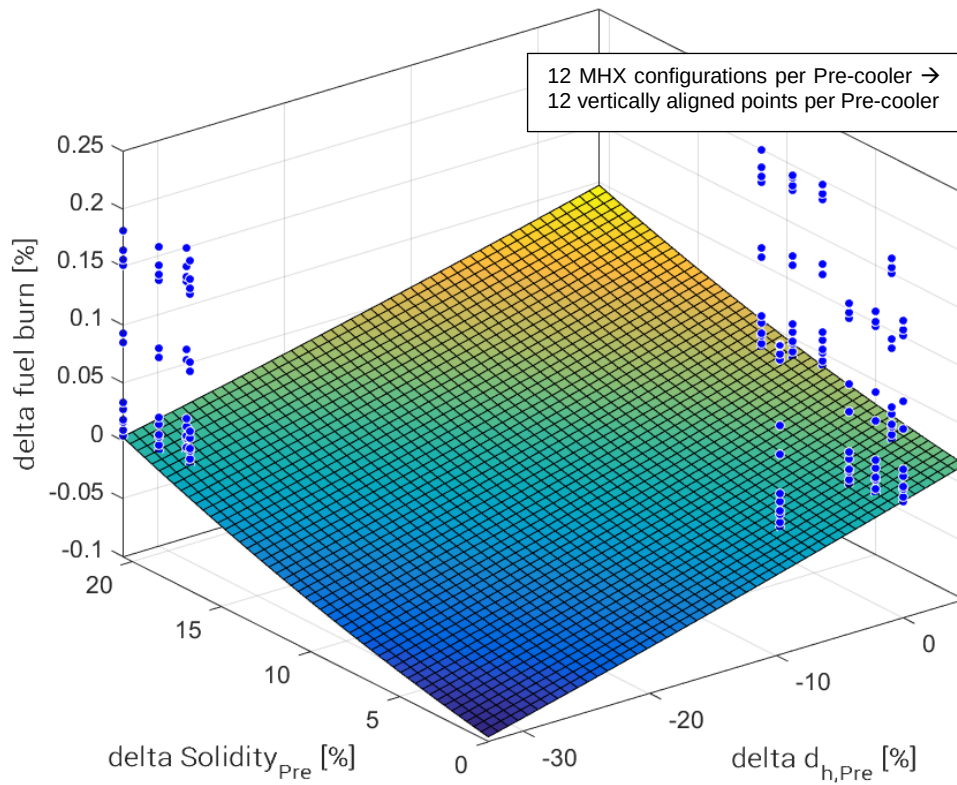


Figure 4-10: Influence of Pre-cooler weight and hydraulic diameter on fuel burn

For the pre-cooler, the internal hydraulic diameter of the tubes seems to have a similar influence as the solidity. However, the coolant mass flow was significantly lower than optimal during these cases. Thus, the outer diameter of the pre-cooler annulus was significantly lower than the inner diameter of outer bypass flow, resulting in uncharacteristically short tubes whose weight was less influential on whole engine performance. The coolant mass flow could have been increased without increasing the outer bypass pressure losses, increasing the S-CO₂ mass flow and hence the power output of the bottoming cycle. Hence, a significant portion of the fuel burn reduction potential was not exploited in the original case study. A repetition of the study with a higher coolant mass flow led to longer and hence heavier tubes, increasing the importance of a low solidity of the internal tube geometry. The resulting fuel burn figures bear closer resemblance to the trends depicted for the MHX in Figure 4-11. The sub-optimal data of the original pre-cooler study in Figure 4-10 was depicted on purpose to illustrate the fact that the relative importance of tube weight, heat transfer and pressure loss can change, depending on other parameters such as the overall dimensions of the heat exchanger.

Figure 4-11 shows that the solidity of the MHX tube, and hence its weight, have a larger influence on mission fuel burn than the internal hydraulic diameter. The hydraulic diameter exhibits a non-linear influence on mission fuel burn, due to the competing influences of pressure losses and heat transfer. It appears that the hydraulic diameter should either be small to maximise heat transfer or high to minimise pressure losses. A compromise of the two influences does not appear to prove the best course of action based on the gathered data. Note that any excessive pressure loss due to too small flow channels could potentially be offset by adding another column of tubes to the MHX. However, this will also increase the weight of the cycle and will not necessarily improve fuel burn.

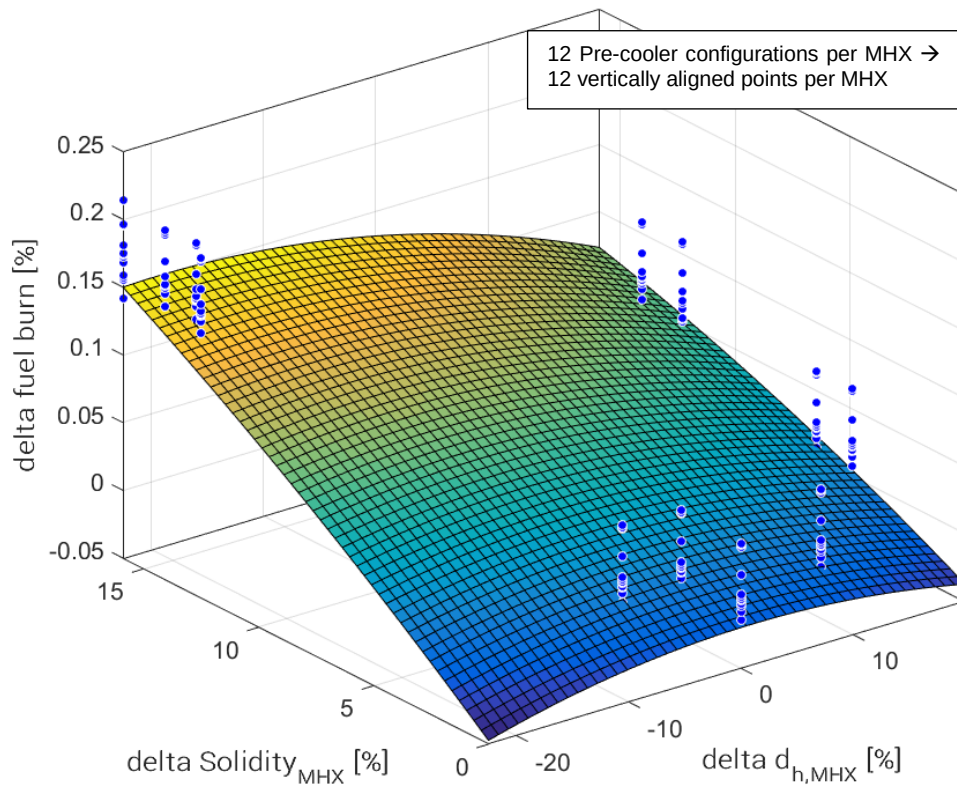


Figure 4-11: Influence of MHX weight and hydraulic diameter on fuel burn

This study determined that the weight of the heat exchangers is more influential on mission fuel burn, within the investigated range, than internal pressure losses and heat transfer. This matches expectations well. The higher overall pressure levels of the S-CO₂ mitigate the influence of the internal pressure losses and any increase in internal heat transfer coefficient has little effect on the total heat transfer due to the low external heat transfer coefficients.

Based on these observations, it was decided to apply the internal tube design with the lowest solidity that was feasible for the pre-cooler and the MHX respectively at the individual operating conditions during all subsequent studies.

4.2.2.2 Compressor Inlet Conditions and Coolant Mass Flow

The inlet temperature and pressure to the S-CO₂ compressor strongly affect the shaft power consumed by the compressor for a given pressure rise due to the real gas behaviour of the working fluid. The T-S-diagrams at the beginning of section 4.2 illustrate that effect clearly. Additionally, the coolant mass flow in the pre-cooler directly affects the combined cycle performance, as the S-CO₂ mass flow is calculated based on the geometry of the heat exchangers and the inlet and outlet temperatures and pressures of the pre-cooler. The coolant mass flow also determines the outer diameter of the pre-cooler arrangement and therefore its weight at a given inlet Mach number, as the inlet temperature and pressure are largely constant at a given fan pressure ratio and diffuser length. Hence, the optimal combination of all three design inputs has to be identified repeatedly for a number of different cycles during many of the parametric studies presented further on, in order to isolate the minimal effect of any design parameter investigated in those studies.

The individual effects of these design parameters need to be understood first. Figure 4-12 illustrates multiple trends at once. The bottom three lines show how the fuel burn benefit relative to the TF2050 (vertical axis on the left) changes with varying coolant mass flow at a fixed compressor inlet temperature. The compressor inlet pressure is kept constant at 8.47 MPa for all three lines. All three lines exhibit similar trends and have a minimum around 38 kg/s. However, the minimum seems to move continuously to lower mass flows as the temperature is increased. It seems that if the coolant mass flow were changed in smaller increments, the precise coolant mass flow offering the lowest fuel burn would change from around 38.5 kg/s for 312.6 K to about 38 kg/s for 313.6 K. This may seem like an unimportant trend in the depicted range, but is very influential when considering a wider design space.

Furthermore, the configurations with compressor inlet temperatures of 312.6 K and 313.1 K offer almost exactly the same maximum fuel burn benefit, while the cycle with 313.6 K as its compressor inlet temperature suffers a small penalty. A similar fuel burn minimum, but at a higher coolant mass flow, could be observed, if a compressor inlet temperature below 312.6 K were depicted. The optimal coolant mass flow reduces with increasing compressor inlet temperature, because of how the useful power output

changes with coolant mass flow (see Figure 4-13). The useful power output increases if the coolant mass flow is increased, due to the higher thermal capacity of the heat exchanger and hence a larger S-CO₂ mass flow. However, the gradient of this increase is dependent on the compressor inlet temperature. The higher the compressor inlet temperature, the further away are the compressor inlet conditions from the critical point. As a result, the variation in specific heat capacity inside the pre-cooler is reduced and the same pre-cooler can cool a larger S-CO₂ mass flow. However, the departure from the critical point also means that the compressor irreversibilities are increased, increasing the power consumption of the compressor. Hence, the useful power increases less if the compressor inlet temperature is further away from the critical point. The combination of these effects leads to the observed trend, in which the optimal coolant mass flow is reduced if the compressor inlet temperature is increased, even if the effect is small in the investigated range.

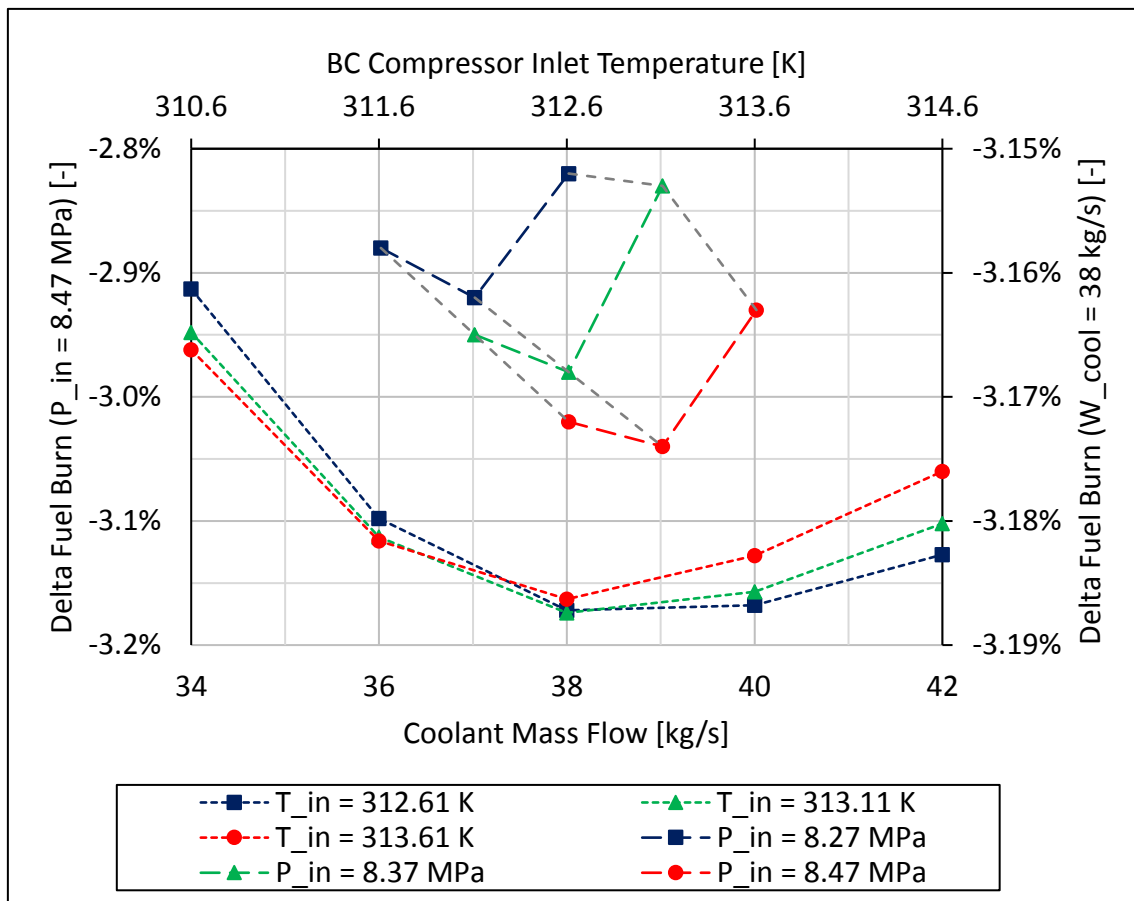


Figure 4-12: Influence of compressor inlet conditions and coolant mass flow on fuel burn reduction potential of combined cycle engine

The upper set of lines depicted in Figure 4-12 corresponds to the vertical axis on the right hand side of the graph. They illustrate the fuel burn minima determined by studies as represented by the bottom three lines, but at three different pressure levels. The red line represents the fuel burn value corresponding to the optimal coolant mass flows for a range of compressor inlet temperatures and at a compressor inlet pressure of 8.47 MPa. Hence, the red line is a different illustration of the minima of the lower three lines. The two remaining data sets show how the fuel burn minimum changes with compressor inlet temperature, if the compressor inlet pressure is 8.27 MPa or 8.37 MPa respectively.

The fuel burn differences are relatively small in the investigated range, but clear trends can be identified nonetheless. The optimal compressor inlet temperature increases by about 0.5 K for every 100 kPa increase in compressor inlet pressure. The optimal coolant mass flow is largely constant in the investigated range, but can be expected to decrease with increasing compressor inlet temperature. Additionally, the depicted results suggest that the fuel burn benefit increases if the compressor inlet conditions increase. The improvement is relatively small between the cases simulated here, but it is much more substantial when simulating configurations further away from the optimum. The fuel burn improves with higher compressor inlet conditions primarily due to slightly larger S-CO₂ mass flows and hence reduced SFCs of the combined cycle engines.

However, this trend should not be extrapolated beyond the depicted range. The internal geometry of the MHX and the pre-cooler tubes are the same for all cases in the depicted range, because they are the lightest feasible designs at these pressures. If the compressor inlet pressure is increased to 8.57 MPa, the internal design of the MHX tubes has to be changed in order to respect the mechanical integrity limits described in section 3.4.1.3. As mentioned before, Rengel's results were not used to generate a surrogate model with interpolation routines, but the discrete results were scaled with chord length and maximum internal pressure. This means, that the solidity of the lightest feasible MHX tubes increases from about 55 % for the cycle with 8.47 MPa, to close to 69 % for the cycle with 8.57 MPa as compressor inlet pressure. As a result, the fuel burn benefit of the cycles with a compressor inlet pressure of 8.57 MPa is actually smaller than that with an inlet pressure of 8.47 MPa due to the significantly higher weight of the MHX.

The non-linear nature of the fuel burn benefit can be explained by looking at the competing effects of useful power, bypass pressure loss and total engine weight. Figure 4-13 shows how the useful power and the bypass pressure losses change with coolant

mass flow throughout the three studies featuring compressor inlet pressures of 8.47 MPa. All values are normalised against the values of the optimal CCTF2050 configuration presented in Table 4-4. The useful power increases continuously with coolant mass flow, because the S-CO₂ mass flow in the bottoming cycle increases. The gradient of the useful power increase depends on the proximity of the compressor inlet temperature of the S-CO₂ to the critical point. The closer the compressor inlet temperature to the critical point, the more can the compressor exploit the real gas behaviour of the fluid, hence the greater the useful power increase when the coolant flow increases.

The bypass pressure losses however also increase with coolant mass flow. This is due to the resulting increase in pre-cooler outer diameter and subsequent contraction of the flow area of the outer bypass. The only deviation from this trend is observed between 34 and 36 kg/s in the figure, which is due to the fact that the bypass pressure losses are not allowed to drop below the reference value of the TF2050 as explained in section 3.4.3. The bypass pressure loss changes independently of the compressor inlet conditions in the bottoming cycle, because the cold side inlet conditions to the pre-cooler define the outer diameter and they are unaffected by the bottoming cycle's performance.

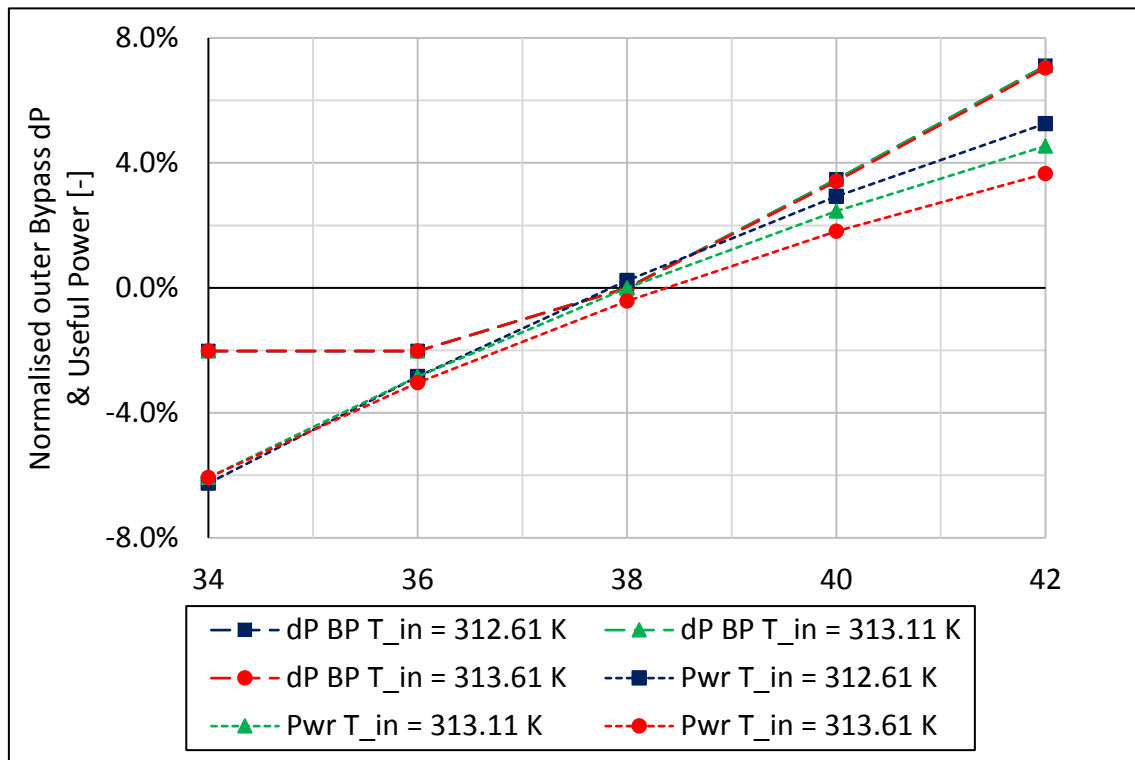


Figure 4-13: Influence of compressor inlet conditions on generated shaft power and the bypass pressure losses

The cold-side pressure losses across the pre-cooler increase very slightly with increasing coolant mass flows. This is likely due to the additional surface area of the pre-cooler tubes and the subsequent increase in frictional losses. On the other hand, the higher coolant mass flow leads to slightly reduced flow velocities through the heat exchanger matrix, offsetting some of these pressure losses. Overall, the variation in cold-side pressure loss is so small for the presented cases that it does not warrant to be plotted. However, the cold-side pressure losses become increasingly influential with higher coolant mass flows, because the relative amount of thrust that has to be provided by that duct increases as a larger portion of the bypass flow is diverted into the duct.

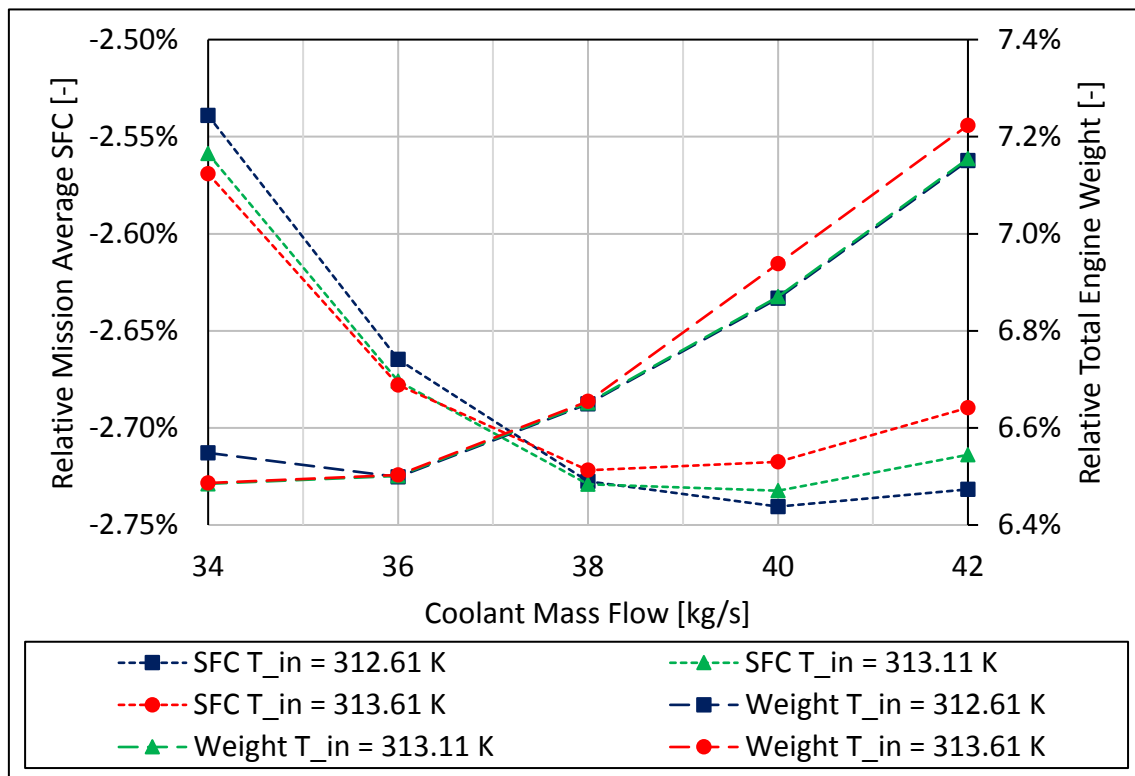


Figure 4-14: Influence of compressor inlet conditions on mission average SFC and total engine weight

The variation in bypass pressure loss and shaft power generated by the bottoming cycle lead to the coolant mass flow dependency of the engine's SFC depicted in Figure 4-14. As seen in Figure 4-13, the useful power does not increase linearly with coolant mass flow, but its gradient reduces continuously. Hence, the beneficial effect of the increased useful power output on SFC reduces, while the pressure losses in the outer bypass flow continuously increase. Consequently, the SFC reduces until the pressure losses become the driving factor, at which point the SFC increases again.

The coolant mass flow that offers the lowest SFC does not exactly coincide with the fuel burn minima shown in Figure 4-12. The weight increase of the pre-cooler is the reason for this discrepancy. The total engine weight of the CCTF2050 is normalised against the weight of the TF2050 and is shown on the vertical axis on the right side of Figure 4-14. Even though the variation in added weight with coolant mass flow is relatively small in this range, it is enough to push the optimal coolant mass flow in terms of mission fuel burn to slightly lower values than the optimal coolant mass flow in terms of SFC.

This section illustrates the intricacies of the weight of the heat exchangers, the pressure losses in the flows passing through them and the pressure losses of the outer bypass flow. Many parametric studies of other design inputs therefore require the establishment of the optimal combination of coolant mass flow and compressor inlet conditions for every investigated configuration, because most of these effects are so delicately linked and because the performance of the S-CO₂ compressor is strongly dependent on the compressor inlet conditions. Pure sensitivity studies that only change one design parameter may not show the whole picture of the combined cycle engine's performance. This is also a reason why the parametric studies conducted to identify the optimal CCTF2050 and CCTF2050_ITR engines were so time consuming.

For some parameters, such as the diffuser length or the absolute surface roughness of the channels in the heat exchanger tubes, not all design drivers could be incorporated in the present models. In these cases, sensitivity studies will at least give an indication of the implications, should the assumptions made during the parametric studies, and for the final combined cycles, be wrong.

4.2.2.3 Bottoming Cycle Pressure Ratio

As with any Brayton cycle, the pressure ratio of the S-CO₂ bottoming cycle greatly affects its performance. Figure 4-15 shows how the fuel burn benefit of the combined cycle over the TF2050 changes when the design pressure ratio of the bottoming cycle is varied. It also illustrates the pressure ratio's effect on the useful power generated by the bottoming cycle and on the total engine weight, the two main drivers of the observed fuel burn trends. The change in total engine weight is shown relative to the TF2050, while the change in useful power output of the bottoming cycle is normalised against the useful power produced by the optimal CCTF2050 configuration with a pressure ratio of 4.0.

The combined cycle offers a fuel burn benefit of about 3.17 % if the pressure ratio of the bottoming cycle is 4.0. If the pressure ratio is reduced to 3.0, the fuel burn benefit reduces

to about 2.75 %. The gradient with which the fuel burn benefit rises, decreases with increasing pressure ratio. This is in unison with the reducing useful power gradients shown in Figure 4-15. Based on the gradients, it could have been expected for the optimal pressure ratio to be slightly higher. However, increasing the pressure ratio to 4.1 necessitates a higher solidity in the MHX tubes, which leads to an increased total engine weight. The slight reduction in SFC, due to the higher increased useful power output, is not enough to offset that weight increase, leading to a higher mission fuel burn. Higher pressure ratios than 4.1 were not able to recover that fuel burn penalty, suggesting 4.0 to be the ideal pressure ratio for this cycle. The internal tube geometries did not change between pressure ratios 3.1 to 4.0. At 3.0 an even lighter design was available and therefore applied for the MHX tubes, but the fuel burn effect is relatively small.

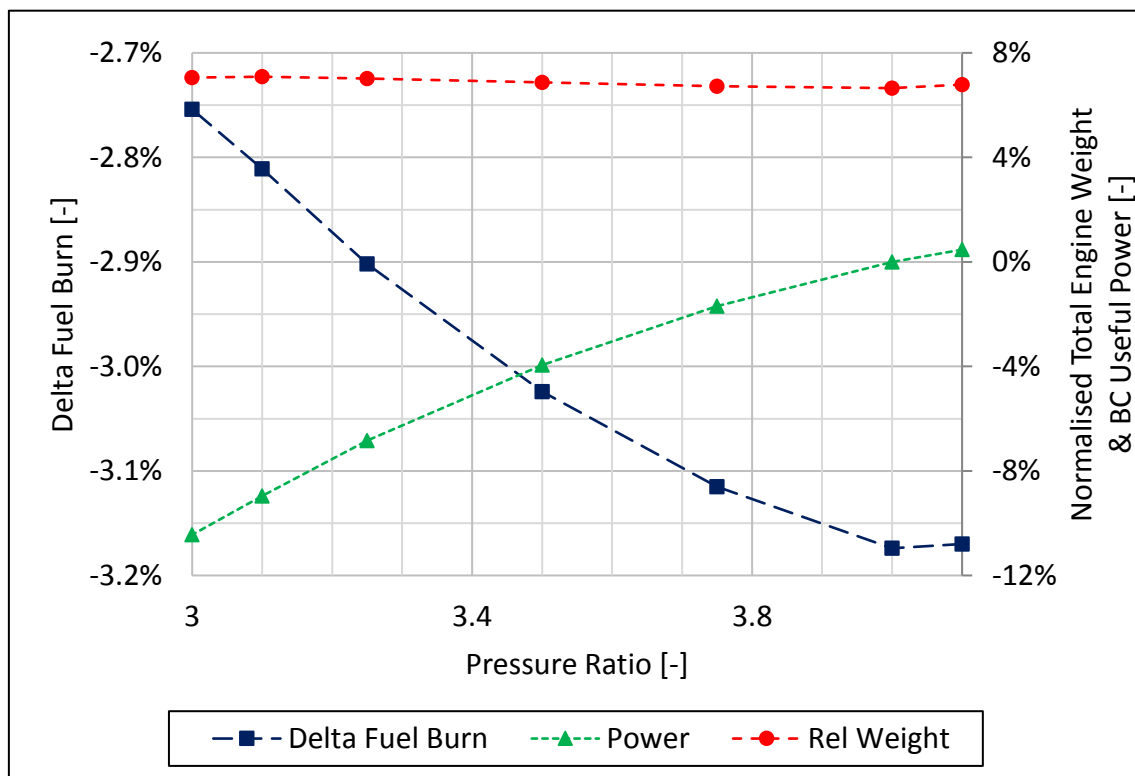


Figure 4-15: Influence of S-CO₂ compressor pressure ratio on CCTF2050

These trends were observed for multiple parametric studies and they are largely independent of most of the other design parameters of the bottoming cycle components. Hence, the pressure ratio was kept constant at 4.0 for all subsequent studies.

It must be mentioned, that a reduction of the bottoming cycle's pressure ratio would allow for thinner walls in the ducts and pipes between the major components of the bottoming cycle. The resulting weight reduction has not been accounted for in the present study

and may reduce the gradients of the fuel burn benefit. Consequently, the optimal pressure ratio may be lower than anticipated here. However, the difference is likely small, as the tubes and ducts will have a circular cross-section and are presently assumed to account for a relatively small portion of the additional weight of the bottoming cycle.

4.2.2.4 Bypass Ratio

The bypass ratio is one of the defining characteristics of a modern turbofan. Over the years, the bypass ratio of civil aero engines has increased continuously, as they usually lead to a reduction in engine specific thrust and hence an increase in propulsive efficiency of the engine. Fundamentally, the bypass ratio can be changed by either incorporating a larger fan or by shrinking the core. Scaling effects on the core turbomachinery, design challenges for the LP shaft, increased nacelle and afterbody drag and limited space underneath the wing are only a few of the design drivers to compromise those benefits, but not all of them are routinely included in performance studies. The implemented efficiency scaling for the HPC and HPT and the non-linear trade factors allow the inclusion of most of these effects in the present study and illustrates how improving the propulsive efficiency can lower the thermal efficiency of the engine. The LP shaft design challenge is not directly addressed, but it is assumed to be resolved through the inclusion of a power gearbox. The present models can therefore also be used to determine which of the two approaches to increasing the bypass ratio is more fuel efficient, or if a compromise should be applied.

The CCTF2050 benefits from an increased core specific power due to the additional shaft power provided by the bottoming cycle. Hence, it might be expected that the additional shaft power may be best used to increase the bypass ratio by shrinking the core engine. However, shrinking the core too much will impede the polytropic efficiency of the high-pressure turbomachinery. Additionally, any reduction in core mass flow will reduce the heat input to the bottoming cycle and therefore also its useful power output. Hence, the higher specific power is primarily used to increase the fan pressure ratio, which slightly raises the specific thrust and reduces propulsive efficiency, but improves the heat transfer in the pre-cooler.

Figure 4-16 shows the results of a parametric study of bypass ratio for the CCTF2050. The compressor inlet conditions and the coolant mass flow were not optimised for each investigated configuration, which means that the fuel burn figures of the engines with bypass ratios other than 16.75 may be slightly pessimistic. The fan pressure ratio however was varied to give the minimum fuel burn for each bypass ratio. The optimal

fan pressure ratio of the bypass section reduced from 1.423 at a BPR of 16.25 to 1.389 at a BPR of 17.75. The pressure ratio of the core fan section was continuously altered as appropriate, as described in section 3.7. In commercial engines, the fan pressure ratio is often slightly lower than the SFC optimum would call for due to a number of reasons, such as to improve engine noise. Engines with very low fan pressure ratios and specific thrust ratings may require variable area nozzles to avoid flutter and improve off-design performance (see Appendix C).

Beside the fuel burn benefit, Figure 4-16 also shows the underlying effects the bypass ratio has on the engine weight increase and SFC reduction, both normalised against the TF2050 values. Additionally, the variation in fan diameter and bypass pressure loss are both shown relative to the values of the optimal CCTF2050.

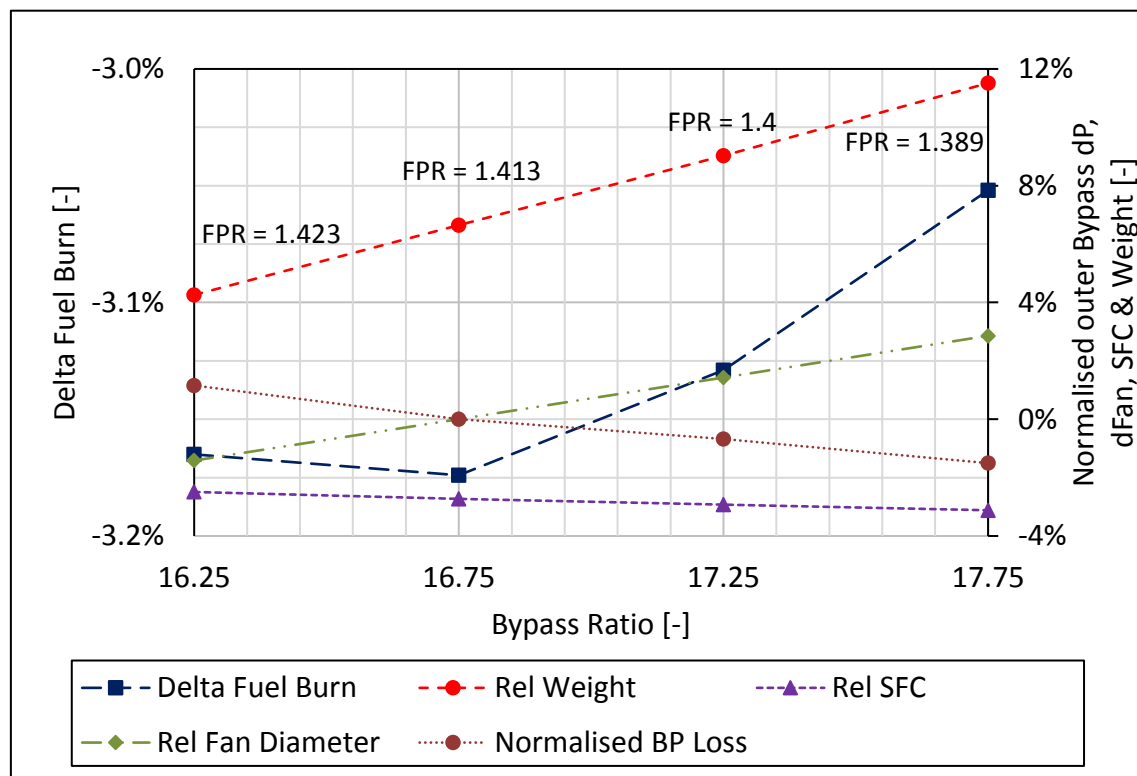


Figure 4-16: Influence of Bypass ratio on CCTF2050 performance

The graph shows that the fuel burn benefit is at its greatest when the bypass ratio is 16.75. At a bypass ratio of 16.25, the fuel burn benefit is very similar, while an increase in bypass ratio leads to definite fuel burn increases. The fuel burn increases with BPR, because the engine weight and the fan diameter both increase, which offsets the slight SFC improvement that can be observed. The coolant mass flow through the pre-cooler is constant for all simulated cases, but the nacelle diameter increases with bypass ratio

because the engine inlet mass flow is increased. Alternatively, the engine core could have been shrunk, but the increasing the inlet mass flow and incorporating a larger fan proved to be more fuel efficient. The flow area of the outer bypass flow is therefore increased and the effect of the pre-cooler outer diameter on the outer bypass flow is reduced. Hence, the bypass flow is not accelerated as much and the pressure losses are reduced. As a result of this and the reduced specific thrust of the engine, the SFC reduces slightly with BPR.

4.2.2.5 Pre-cooler Inlet Mach number

The Mach number at the inlet of the pre-cooler affects the performance of the combined cycle engine in a number of ways. A reduced pre-cooler inlet Mach number means that the area ratio of the inlet diffuser will increase, increasing the diffuser pressure loss and worsening flow stability within. The pressure losses across the heat exchanger matrix however will likely be reduced, because the average flow velocity inside the heat exchanger and the friction losses are reduced. The frontal area of the heat exchanger will be increased due to a lower inlet Mach number, which leads to longer heat exchanger tubes and a greater heat transfer surface area improving the performance of the CCTF2050. However, the greater frontal area also leads to a larger outer diameter of the pre-cooler duct, because the inner annulus diameter is fixed, resulting in increased pressure losses in the outer bypass flow (see Figure 4-17 for pre-cooler dimensions again). Additionally, a reduction in flow velocity inside the pre-cooler leads to reduced heat transfer coefficients, though the higher total pressure may offset part of that effect.

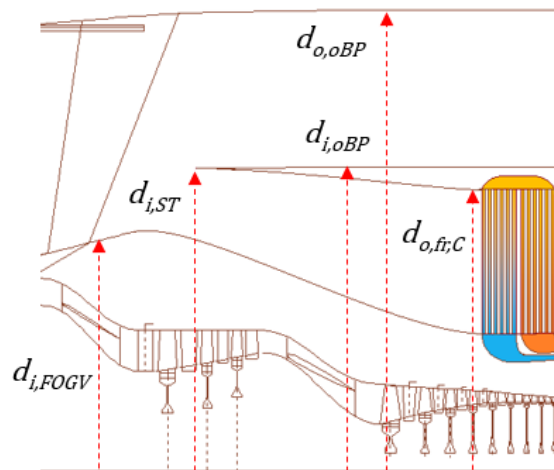


Figure 4-17: Excerpt of a CCTF2050 cross-section with pre-cooler dimensions

The number of competing trends due to a change in inlet Mach number, make the identification of the optimal value particularly important and complex. This makes the inlet Mach number a prime example to illustrate the difference between the results of a parametric study that only varies the inlet Mach number and those of a parametric study that identifies the optimal combination of compressor inlet conditions and coolant mass flow for every investigated pre-cooler inlet Mach number.

Figure 4-18 illustrates the CCTF's dependency on the pre-cooler inlet Mach number alone. The cold-side losses are plotted directly in the graph, while the variation in outer bypass pressure loss and the useful power generated by the bottoming cycle are normalised against the data from the optimal CCTF2050 for better visibility.

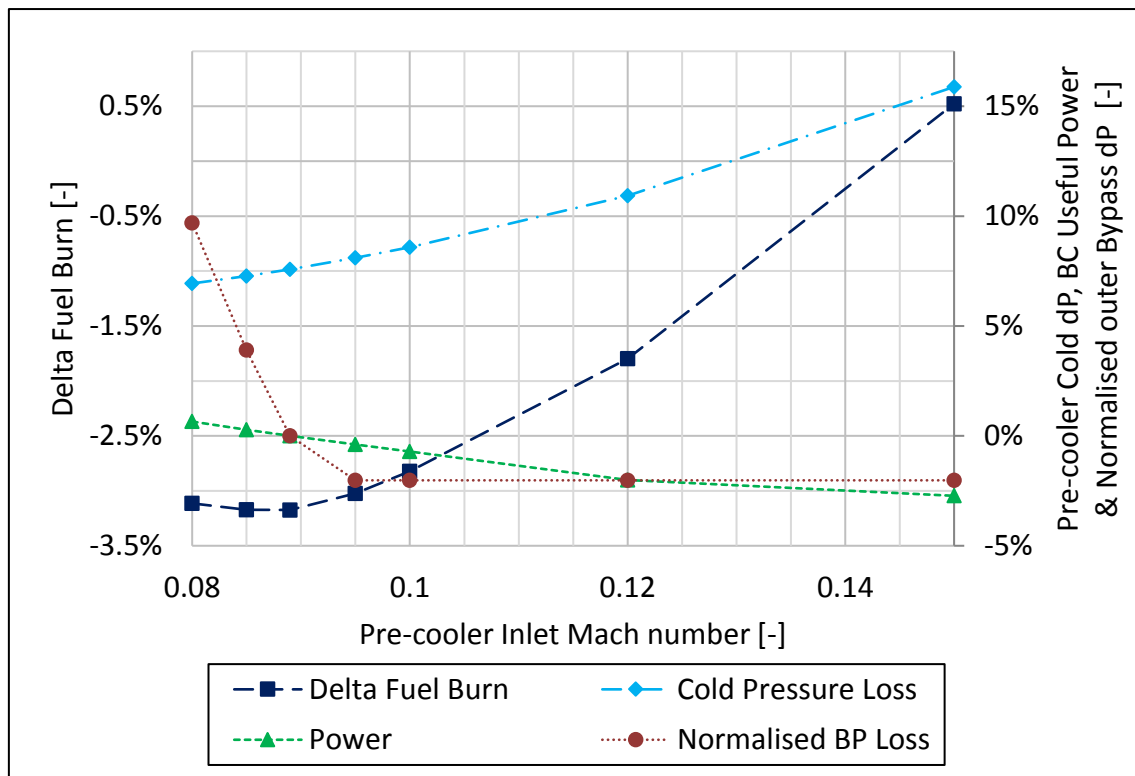


Figure 4-18: Sensitivity of CCTF2050 to pre-cooler inlet Mach number

The cold pressure losses across the pre-cooler duct increase continuously with Mach number, because the pressure losses across the heat exchanger matrix account for a larger portion of the cold losses than the diffuser losses. The reduced heat transfer surface area and subsequently reduced heat duty of the pre-cooler is the main reason why the useful power generated by the bottoming cycle reduces with increasing Mach numbers. The bypass pressure loss is significantly affected by the pre-cooler inlet Mach number, because the outer diameter of the heat exchanger increases if the Mach number

is reduced. In this simulation however, increasing the inlet Mach number beyond 0.095 does not affect the outer bypass pressure loss anymore, because the model does not allow for the losses to be reduced below the reference values of the TF2050. The change in fuel burn benefit depicted in Figure 4-18 is the culmination of all these effects. Starting at an inlet Mach number of 0.08, the fuel burn benefit increases due to the reduced bypass pressure losses until it reaches its maximum at an inlet Mach number around 0.089. If the inlet Mach number is increased beyond 0.089, the bypass pressure loss reduction can no longer offset the increase in cold-side pressure losses and the loss in useful power, and the fuel burn benefit reduces continuously. The sensitivity of the combined cycle model to the pre-cooler inlet Mach number is so strong, that an inlet Mach number greater than 0.142 no longer leads to any fuel burn benefit over the TF2050 at all.

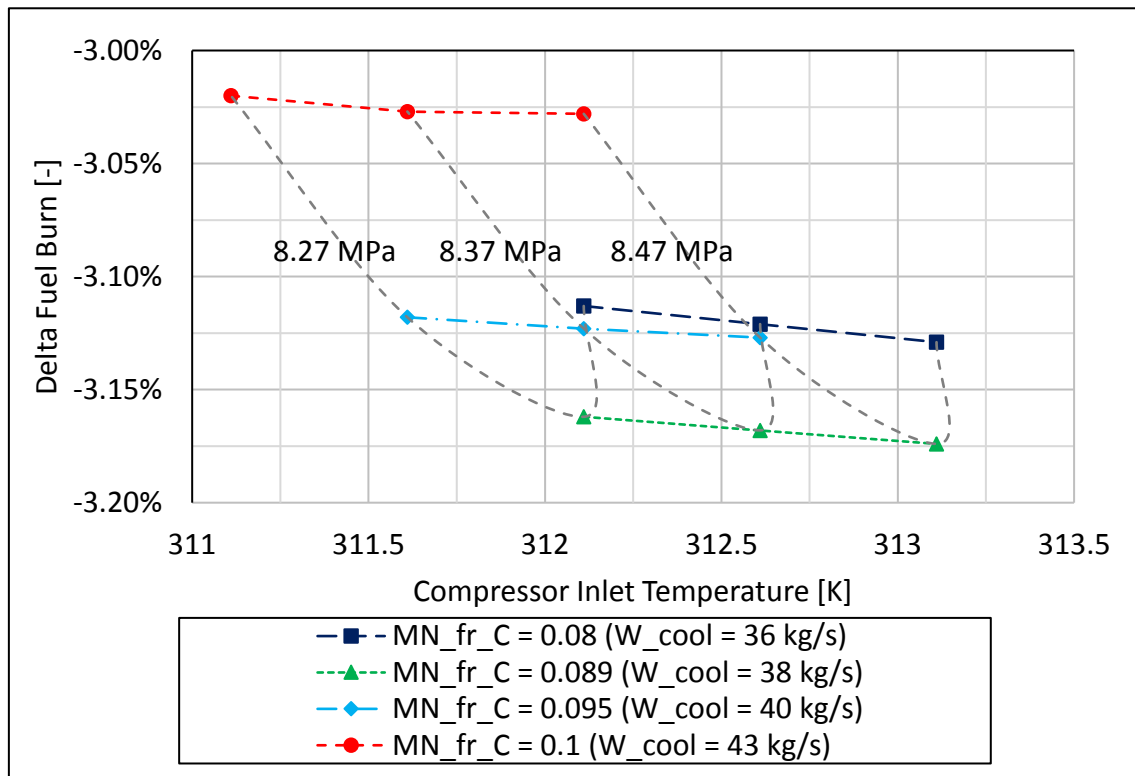


Figure 4-19: Influence of compressor inlet conditions and pre-cooler inlet Mach number

Figure 4-19 and Figure 4-20 illustrate how this sensitivity can be reduced by matching the right coolant mass flow and compressor inlet conditions to each investigated pre-cooler inlet Mach number. So, if another constraint not included in the current models

were to necessitate an inlet Mach number greater than 0.089, the identified design changes could be applied to reduce performance penalties.

Figure 4-19 illustrates how the compressor inlet temperature affects the fuel burn benefit. Note, that the optimal coolant mass flow has been identified for each inlet Mach number. The optimal coolant mass flow is usually the one that leads to the least acceleration of the bypass flow. Larger coolant mass flows, and hence pre-cooler outer diameters and bypass flow accelerations, increase the bypass pressure loss too much. Smaller coolant mass flows would mean that the pre-cooler is not as big as it could be without impeding the bypass flow. The compressor inlet pressure is varied with compressor inlet temperature as discussed before. So for every 0.5 K change in inlet temperature, the inlet pressure is also changed by 100 kPa.

The graphs clearly show that the engine with an inlet Mach number of 0.089 achieves the greatest fuel burn benefit. Furthermore, the iso-Mach number lines indicate that the optimal compressor inlet conditions are those furthest away from the critical point, as discussed before. Further increases would not yield improvements however, due to the necessary changes of the internal tube geometry of the MHX.

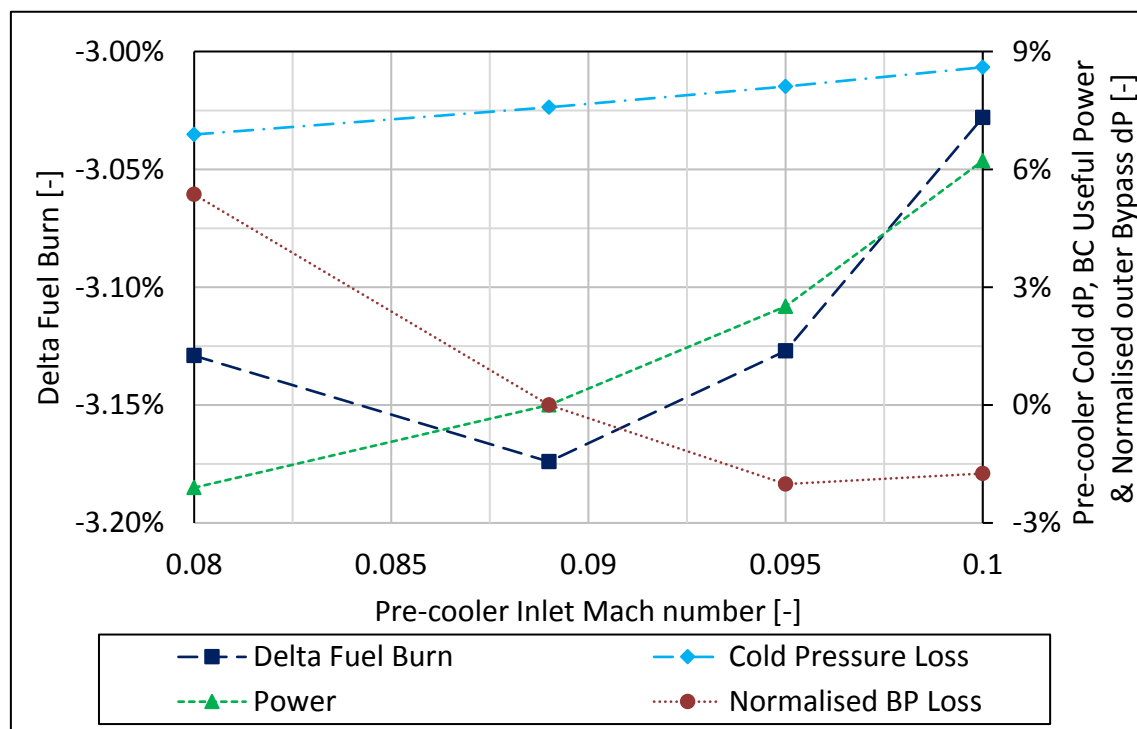


Figure 4-20: Influence of pre-cooler inlet Mach number

The fuel burn data shown in Figure 4-20 illustrates the respective minimum of each of the iso-Mach number lines in Figure 4-19, representing the optimal combination of all four design parameters. Additionally, the plot shows the variation of the cold losses (light blue), and how the useful power generated by the bottoming cycle (green) and the outer bypass pressure losses (brown) change. The latter two parameters were normalised against the optimal CCTF2050 configuration. It is important to note, that the variation in useful power depicted contradicts the trends shown in Figure 4-18. This is due to the variation in coolant mass flow and compressor inlet conditions for each simulated pre-cooler inlet Mach number. The inlet Mach number obviously affects further parameters of the combined cycle engine, the total engine weight continuously decreases with Mach number for example, but those have a secondary effect on the fuel burn estimate.

Overall, the combined cycle still offers the greatest fuel burn benefit over the TF2050, if the Mach number in front of the pre-cooler is about 0.089. Lower Mach numbers lead to reductions in useful power and increased bypass losses, while higher Mach number designs suffer due to the increased pressure losses in the airflow passing through the pre-cooler. The initial sensitivity study predicted that the fuel burn benefit would drop by 0.35 percentage points from 3.17 % to about 2.82 %, if the inlet Mach number had to be increased to 0.1. However, that drop can be mitigated and the fuel burn benefit at $M = 0.1$ can be raised to 3.03 % again, if the coolant mass flow and compressor inlet conditions are adapted. This clearly illustrates the benefits that may be obtained from conducting multi-parameter studies over simpler single-parameter studies for the CCTF.

4.2.2.6 Number of Tube Columns per Pass in Pre-cooler

The inner annulus diameter of the pre-cooler determines the number of tube rows in the arrangement. The number of pairs of tube columns however is a design input for the present model. Hence, the heat duty of the pre-cooler can be directly influenced through this design parameter. The CCTF2050 offering the greatest mission fuel burn benefit features seven tube columns per pre-cooler pass (see Figure 4-21), so 14 tubes arranged in a zig-zag pattern. The fuel burn benefits displayed in the graphs are once again a result of the individual identification of the optimal combination of compressor inlet conditions and coolant mass flow. The cycles with a compressor inlet pressure of 8.47 MPa once again offer the greatest benefits, but the fuel burn data for 8.27 MPa and 8.37 MPa are also shown to corroborate this trend. The optimal compressor inlet temperature rises by 0.5 K for every 100 kPa pressure increase at the compressor inlet, as observed before. However, the useful power, outer bypass pressure loss and cold-

side pressure loss data shown in Figure 4-22 all correspond to the 8.47 MPa cases. During this study, the column count could also have half integer values, because one column accounts for two tubes that are circumferentially offset. See section 3.4.1.1 for an illustrated explanation of this.

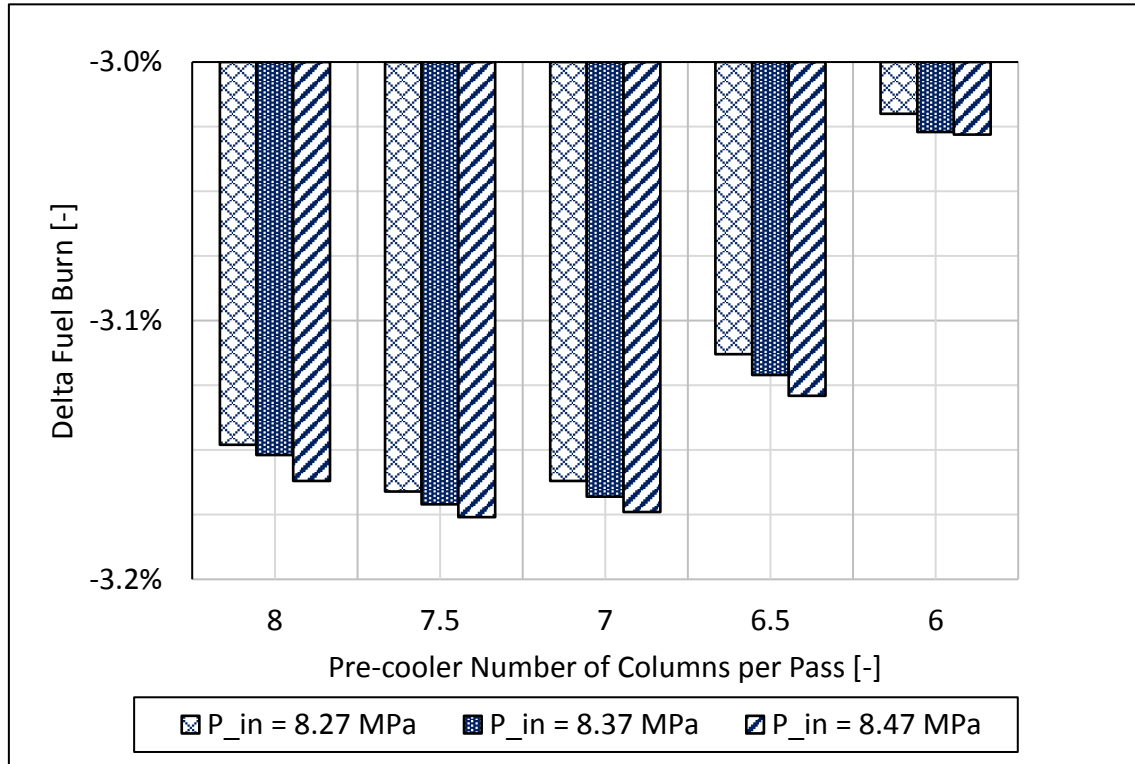


Figure 4-21: Fuel burn influence of number of tube columns per pre-cooler pass

All results of this parametric study are depicted in bar charts rather than graphs, because intermediate numbers of tube columns are very unlikely to be produced and the data should therefore not be connected via trend lines. The individual modules of the pre-cooler could theoretically feature different column counts, but this would make manufacturing and assembly overly complicated.

Once again, the data seems to suggest that higher compressor inlet conditions would further improve the fuel burn benefit, but any such benefit would be negated by the necessary increase in MHX tube solidity. Figure 4-22 depicts the main drivers behind the fuel burn benefit trends: the cold pressure losses across the pre-cooler duct (light blue), the useful power (green) and the pressure losses in the outer bypass (orange). The bypass pressure losses are once again normalised against the optimal CCTF2050, as is the useful power output of the bottoming cycle. The cold pressure losses across the pre-cooler duct are displayed directly.

In most cases, the optimal coolant mass flow reduces if the number of pre-cooler tube columns is increased. Consequently, the pre-cooler outer annulus diameter decreases, reducing the bypass pressure losses as seen. The useful power increases noticeably with column count, as the greater heat duty of the pre-cooler is used to cool a larger S-CO₂ mass flow. Unfortunately, the cold pressure losses across the pre-cooler duct also increase noticeably with increasing tube counts, due to the additional friction losses.

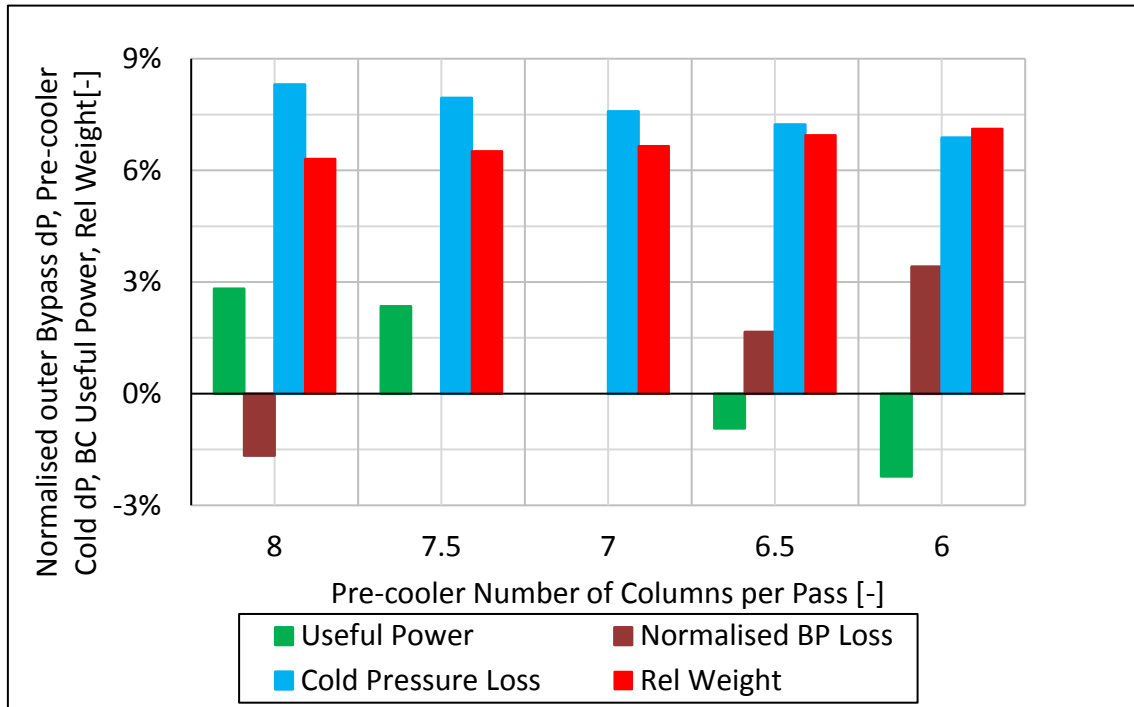


Figure 4-22: Influence of number of tube columns per pre-cooler pass

The otherwise almost linear trends of the bypass pressure loss and useful power show a different behaviour when the number of columns is changed from 7 to 7.5. The change from 7 to 7.5 tube columns per pass increases the heat duty of the pre-cooler to a degree that the optimal compressor inlet temperature is reduced by 0.5 K (still at 8.47 MPa) and the optimal coolant mass flow stays the same. The outer bypass losses therefore remain constant because the pre-cooler's outer diameter remains constant. Additionally, a step change in S-CO₂ mass flow and therefore in useful power occurs, because the optimal coolant mass flow remains the same. The lower compressor inlet temperature furthermore reduces the necessary compressor work, further improving the useful power output. The trends would be more linear, if the models allowed for a differential variation of the heat duty or if the compressor inlet conditions were changed in finer steps during the study. While smaller temperature and pressure steps could have been investigated, the benefit would have been minimal during the pursuit of minimum mission fuel burn.

Changing the heat duty of the pre-cooler in smaller discrete steps on the other hand would not make sense, as the observed trends are the logical conclusion of accounting for preliminary design effects this early in the development of the concept. It would of course be possible to include differing tube counts in the individual pre-cooler modules, but such individuality would be less than ideal during mass production.

The effects of the number of tube columns of the MHX are not argued specifically in this section, as the general trends follow a similar logic as presented for the pre-cooler tube columns. A larger number of heat exchanger tubes obviously leads to an increased engine weight, but it also leads to an increase in peak temperature of the bottoming cycle. While this raises the thermal efficiency of the cycle, the pre-cooler inlet temperature will still be increased as well, which means that the S-CO₂ mass flow has to be decreased in order to meet the same compressor inlet conditions. Alternatively, the optimal compressor inlet conditions could be increased with the number of MHX tube columns, in order to facilitate a larger S-CO₂ mass flow and therefore generate more power, even if the thermal efficiency of the cycle does not increase. The optimal number of MHX tube columns is therefore dependent on the heat duty of the pre-cooler and therefore on the number of pre-cooler tube columns, as increasing the number MHX tubes alone does not necessarily improve the fuel burn benefit. For the CCTF2050 it could be observed, that the MHX and the pre-cooler should have similar numbers of columns in order to minimise fuel burn. The necessarily greater heat duty of the MHX will be realised through the longer heat exchanger tubes.

4.2.2.7 Leaned involute pre-cooler tubes

Leaning the tubes at the hub is one way to increase the length of the tubes without increasing the annulus height or the outer diameter of the pre-cooler. For involute tubes, this can be modelled, by moving the theoretical start of the involute shape to a lower diameter than the actual inner annulus diameter (see Figure 4-23 again). The ratio of the two diameters is dubbed x_{inv} in the present models.

Figure 4-24 shows the sensitivity of the CCTF2050 to that ratio. The coolant mass flow and the compressor inlet conditions remained constant during this investigation. The fuel burn benefit of the CCTF2050 over the TF2050 increases only slightly if x_{inv} is reduced and the tubes start of at a shallower angle to the inner annulus wall within the investigated range. The additional tube length resulting from reducing x_{inv} leads to an increase in useful power, which is once again normalised against the optimal CCTF2050 in the figure.

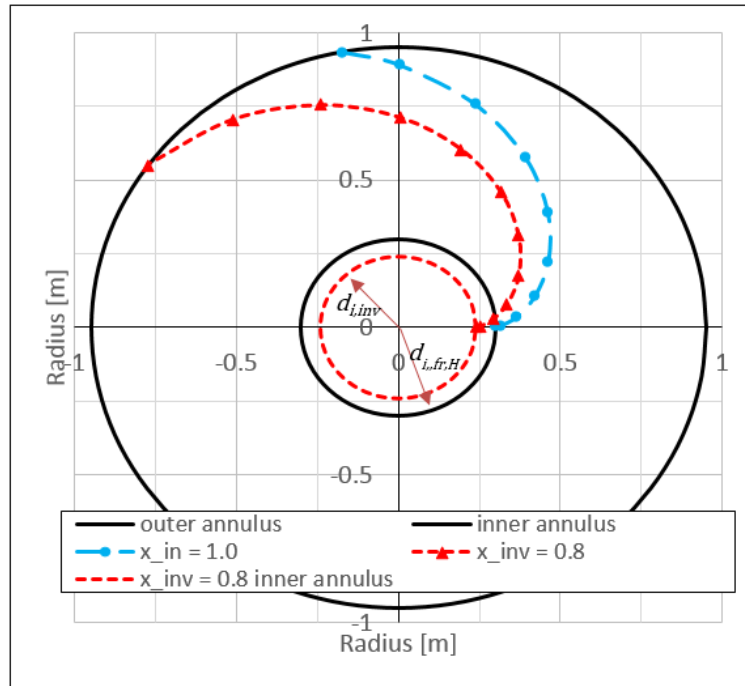


Figure 4-23: Definition of x_{inv} (ratio of diameters)

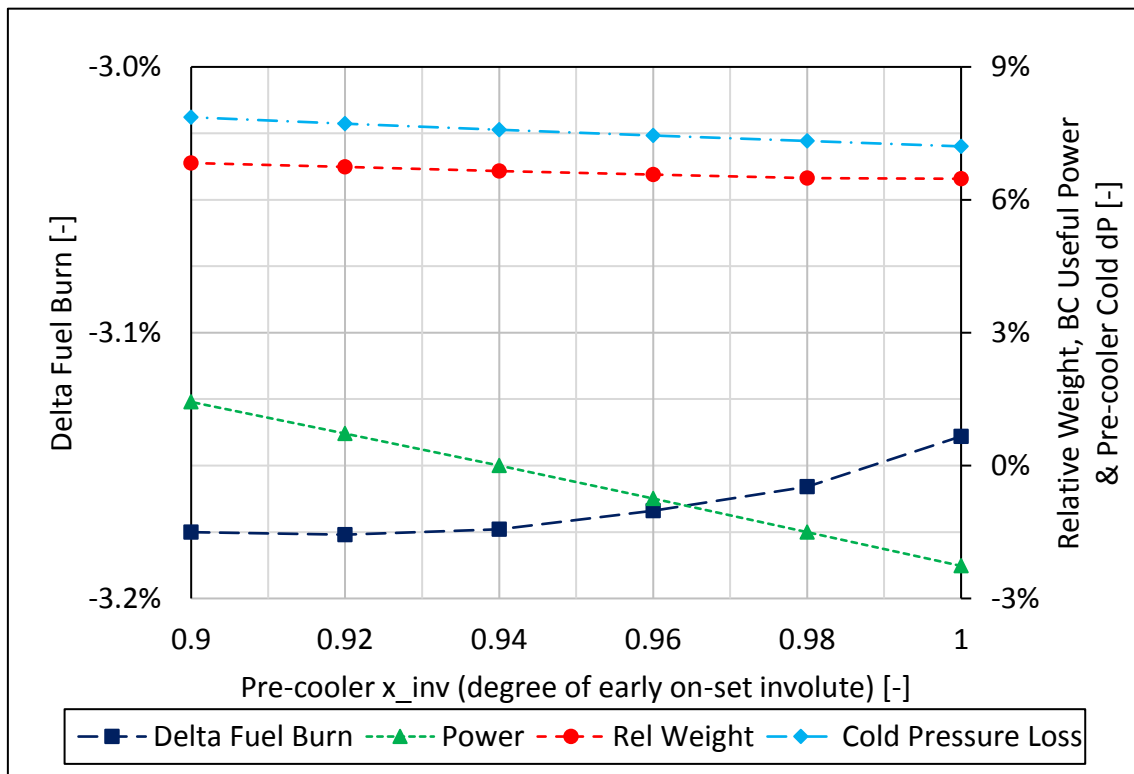


Figure 4-24: Sensitivity of CCTF2050 to leaning the involute tubes at the offset

However, the combined detrimental effects of a higher engine weight (red), higher cold pressure losses across the pre-cooler duct (light blue) and higher pressure losses on the

CO₂ side offset a large part of that benefit and the fuel burn benefit actually decreases again for x_{inv} values below 0.92. The weight increase is once again shown relative to the TF2050 and the cold pressure losses are depicted in directly.

The benefit from starting the tubes at a shallower angle to the inner annulus wall was expected to be larger than realised in this present study, based on the design choices made by Reaction Engines for the SABRE engine [103]. The fuel burn benefit of the CCTF2050 is only increased from 3.14 % to about 3.17 %. Such a change is within the margin of errors of the models anyway. The qualitative trends should nonetheless be valid and it is important to know that such a design change would not yield considerable benefits. Furthermore, the influence proved to be larger during the larger multi-layered parametric studies that led to the optimal CCTF2050 configuration, because it was traded against changes for the design parameters such as the number of columns and the pre-cooler inlet Mach number. Unfortunately, the trends are less discernible during these multi-parameter studies and hence not easy to display in 2D plots such as these.

The fact that the optimal x_{inv} is not lower than 0.9 also means that the manufacturing of the heat exchanger modules is not complicated further, avoiding unnecessary costs and potential failure scenarios.

4.2.2.8 Pre-cooler Tube Chord Length

The chord length of the heat exchanger tubes not only affects the heat transfer surface area available per tube and the friction losses occurring around and through them, it also affects the internal tube geometry. The integration and scaling of Rengel's internal tube geometries may lead to discontinuities in the performance data of the CCTF2050 as has been shown in previous examples. This becomes particularly apparent when the CCTF's sensitivity to tube chord length is investigated (see Figure 4-26 and Figure 4-26). The compressor inlet conditions, the coolant mass flow and all other design inputs were kept constant again during this parametric study. Only the chord length and the internal geometry of the tubes was changed.

A reduction in chord length of the elliptical tubes for a given number of tubes should lead to a continuous decrease in heat exchanger and hence total engine weight. However, the discrete changes of the internal tube geometry make this impossible. A number of restrictions were applied to the internal tube design data. For example, the wall thickness and the internal web thickness cannot drop below 0.15 and 0.1 mm respectively, due to assumed manufacturing limits. Also, the fillet radii of the internal flow channels have to

be chosen so the fillets never intersect, always leaving at least circular channels between them.

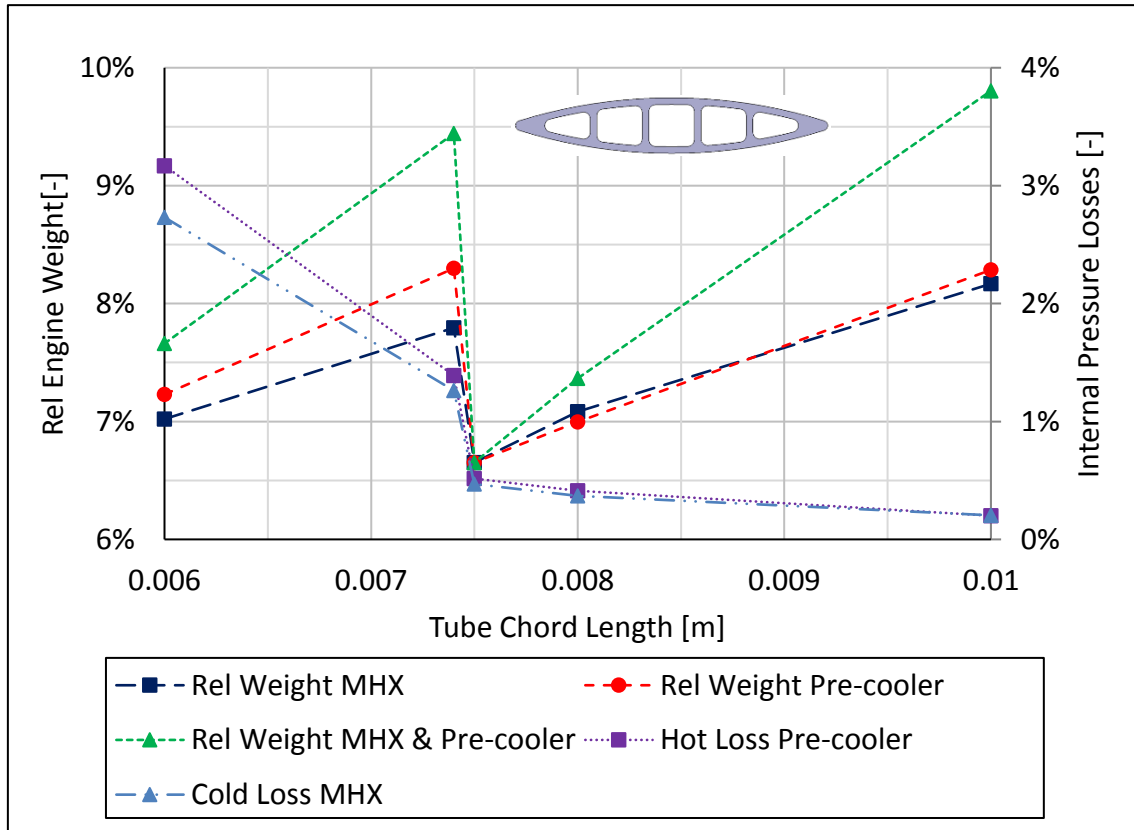


Figure 4-25: Sensitivity of heat exchanger weight and pressure losses to chord length

Due to these constraints, the number of webs and the fillet radii (defined as percentages of the chord length) change in discrete steps when the chord length is reduced from 7.5 to 7.4 mm. The reduced number of webs that can be placed in the tubes with short chord lengths requires for the outer walls of the tube to be thicker in order to withstand the internal pressures. This discrete increase in outer wall thickness (from 2 to 3 % chord length) significantly affects the internal design and the tube solidity. When the chord length is reduced to 7.4 mm, the solidity of the lightest feasible pre-cooler tubes increases from about 48 % to 66 %. The same change in chord length leads to an increase from about 55 % to 69 % solidity of the lightest possible MHX tubes. These are the only significant discrete changes in internal tube geometry in the investigated range, but the effect is quite severe as Figure 4-25 and Figure 4-26 clearly show.

Figure 4-25 also shows how the total weight increase of the CCTF2050 over the TF2050 changes when the chord length of only the MHX (blue), only the pre-cooler (red) or of

both heat exchangers is varied simultaneously. The individual effects are similar if the chord length of the tubes of the pre-cooler or the MHX are increased beyond the 7.5 mm of the optimal CCTF2050. A reduction of the chord length to 7.4 mm on the other hand has a noticeably stronger effect on the total engine weight if applied to the pre-cooler, due to the significantly larger increase in solidity of the tubes.

The weight of the heat exchangers is not the only parameter that changes significantly when the tube chord length is varied. The total available free-flow area inside the tubes also changes, because the number of tubes is kept constant. Hence, the average flow velocity will decrease with increasing chord length, reducing the heat transfer coefficients. The additional heat transfer surface area and the fact that the external heat transfer coefficients are the limiting factor for both heat exchangers mitigate that penalty. Again, these almost linear trends are changed by the significant change in internal tube geometry when the chord length drops to 7.4 mm. The significant increase in tube solidity at that point also means, that the hydraulic diameter and the available flow area significantly decrease. Hence, the internal pressure losses increase significantly. The even smaller flow channels in 6 mm chord length tubes further aggravate the problem due to yet another internal tube geometry, leading to further significant internal pressure loss increases. The internal pressure losses gradually decrease if the chord length of the tubes is increased beyond 7.5 mm.

Figure 4-26 illustrates how the mission fuel burn benefit of the CCTF2050 over the TF2050 changes, if only pre-cooler tubes (red), only the MHX tubes (blue) or both sets of tubes are changed simultaneously. In any case, increasing the chord length beyond 7.5 mm reduces the fuel burn benefit. Reducing the chord length however also worsens the mission fuel burn due to the change in internal flow geometry. The individual effects of the two heat exchangers add to another, leading to more significant fuel burn increases if the chord lengths vary from the optimal 7.5 mm. The fuel burn benefit is reduced by about 0.2 percentage points if the heat exchanger tubes of either heat exchanger have a design chord length of 7.4 mm instead of 7.5 mm. If both heat exchangers were to be designed that way, the fuel burn benefit would be reduced by almost 0.4 percentage points. It is important to emphasise that these effects only occur if the internal tube geometry is changed in unison. Uncertainties in the manufacturing processes will not have such an effect as the internal tube geometry will have the same approximate design. Manufacturing uncertainties may however increase the risk of tube failures, though the applied factor of safety should account for typical tolerances.

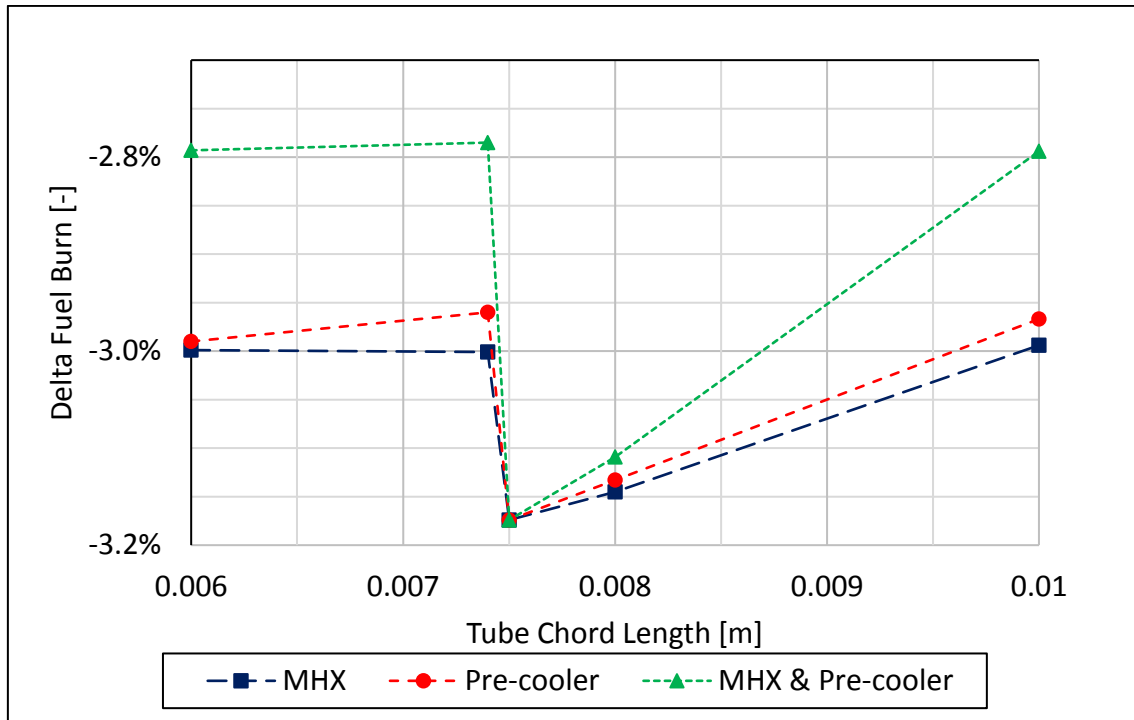


Figure 4-26: Sensitivity of CCTF2050 to the chord length of the pre-cooler tubes

Part of the illustrated effects, for example the internal pressure losses due to excessive flow acceleration, could be remedied if the number of tube columns of the heat exchangers were changed with chord length. This study clearly shows that a large number of smaller heat exchanger tubes can be more efficient than a small number of larger tubes, but mechanical integrity and manufacturing constraints can noticeably alter these trends.

4.2.2.9 Absolute Surface Roughness

The absolute surface roughness of the diffuser walls and the channels inside the heat exchanger tubes has been assumed fixed at $1\text{E-}5$ m. However, the actual surface finish strongly depends on the manufacturing technique, which in turn may be chosen based on the respective costs. Hence, a sensitivity study was conducted (see Figure 4-27) in which the absolute surface roughness was varied between $1.6\text{E-}6$ m and $3\text{E-}5$ m. The fuel burn benefit of the CCTF2050 is depicted in the plot to illustrate the sensitivity to surface roughness inside the heat exchanger tubes (blue), the surface roughness of the diffuser annulus walls (green) and of the combined effects of the two. Both component models account for surface roughness effects by using the relative surface roughness during the estimation of the respective friction factors. The orange and purple lines illustrate how the internal pressure losses and the heat transfer in the MHX change with

the surface finish inside the MHX tubes. Both parameters are normalised against the optimal CCT2050. The total variation in mission fuel burn in this study is small, because the individual contributions of the pressure losses and the heat transfer are small.

It might be expected that a smooth wall would give the best performance for any of these cases, because the incurring pressure losses would be reduced. However, this is not the case when the absolute surface roughness of the internal heat exchanger walls is investigated. The internal pressure loss inside the heat exchangers is reduced, but that change has only a small effect on the mission fuel burn, given the low relative pressure loss of about 0.5% per heat exchanger. However, the heat transfer coefficients inside the heat exchangers are also reduced due to the lower friction factors and that effect is much more influential on the estimated mission fuel burn. Hence, the fuel burn benefit is actually bigger for larger absolute surface roughnesses inside the heat exchanger channels. A similar effect could theoretically be achieved in smooth channels by including secondary devices such as ribs or dimples, same as in the cooling channels of modern turbine blades. However, the channels inside these tubes are probably too small to manufacture such vortex inducing design features reliably.

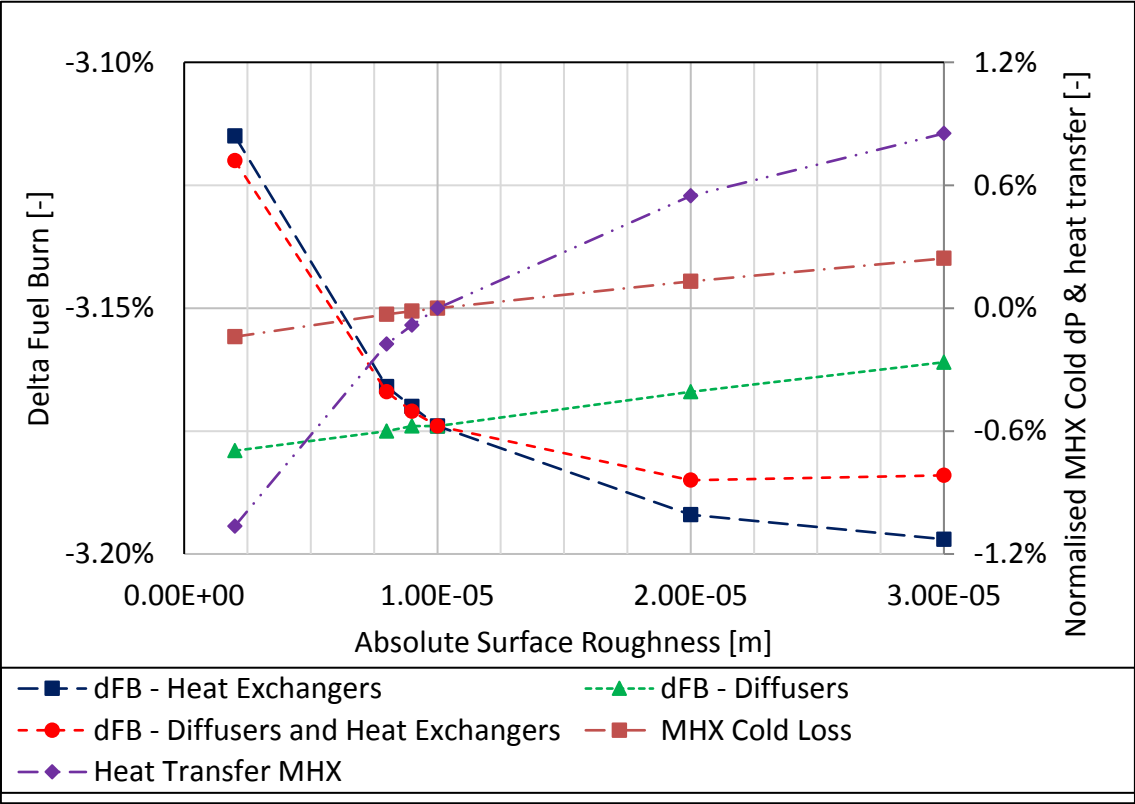


Figure 4-27: Sensitivity of CCTF2050 to absolute surface roughness in heat exchanger tubes and diffusers

Increasing the surface roughness of the diffuser walls leads to the expected linear increase in mission fuel burn. Overall, the fuel burn benefit is dominated by the effect of the absolute surface roughness inside the heat exchangers. The reduced heat duty of the heat exchangers due to the lower surface roughness outweighs the lower pressure losses in the heat exchanger and diffuser. However, it is unlikely that the surface roughness inside the heat exchanger tubes could be reduced as much or as easily as that of the diffuser walls, given the size of the channels.

4.2.2.10 Diffuser Length

The diffusers leading up to the pre-cooler and the MHX were modelled using the same simplified methodology described in 3.4.1.2. The technology factor that was introduced to the basic equation presented by White [104] was furthermore kept constant for both diffusers, even though the MHX diffuser is unlikely to include a substantial amount of boundary layer bleeding or any pre-diffusion. The length of the diffuser, together with the area ratio, determines whether surface friction or flow separation is the dominant loss mechanism inside the diffuser as discussed in section 2.5.3.

Figure 4-28 highlights the significant influence of the diffuser length, and hence pressure loss, on the fuel burn benefit the CCTF2050 may achieve relative to the TF2050. The length of each diffuser was varied individually (red and blue data) and simultaneously (green) between 0.5 and 1.5 m, with both having to provide similar diffusion levels. No interaction between the diffuser length and any other component is captured by the current models beyond the pressure losses. Hence, only mission fuel burn is shown in Figure 4-28. The fuel burn benefit is reduced by about 0.6 percentage points, if the diffuser length of either heat exchanger is reduced from the assumed 1.2 m of the optimal CCTF2050 to 0.5 m. The effect is slightly more significant for the diffuser ahead of the pre-cooler (red), because its area ratio is slightly larger. The influence of the diffuser length is clearly non-linear, as the relative importance of the friction losses is reduced considerably as the diffuser is shortened. If the length of both diffusers were changed simultaneously, their respective effects would be added, meaning the fuel burn benefit of the CCTF2050 would be reduced to about 1.93 % if both diffusers could only be 0.5 m long. However, it is unlikely that the diffusers of the two heat exchangers would be changed in unison. In fact, it may be possible to increase the length of the pre-cooler diffuser to about 1.5 m in the current designs. The actually available space would have to be determined by more detailed models that can assess whether or not it would be possible to place the pre-cooler around the HPT rather than at the back of the HPC.

However, the present models predict the resulting fuel burn improvement to be relatively small, as boundary layer separation is less important and the friction losses begin to dominate the total pressure loss.

It may not be feasible on the other hand to position the MHX 1.2 m downstream of the LPT, necessitating a shorter diffuser design. The fuel burn benefit of the CCTF2050 would be reduced from about 3.17 % to about 3 % if the pre-cooler diffuser were 1.5 m long and the MHX diffuser were 0.9 m long.

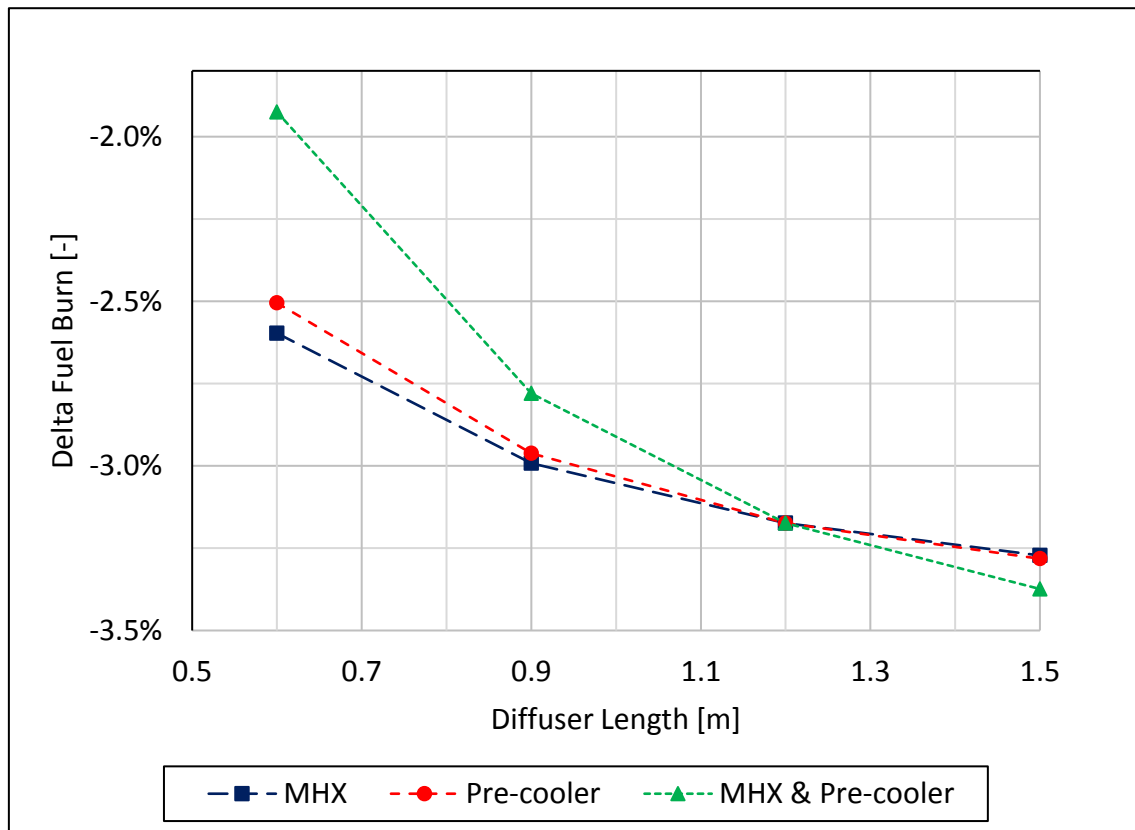


Figure 4-28: Sensitivity of CCTF2050 to diffuser length for MHX and pre-cooler

The present diffuser models are not detailed enough to fully appreciate the influence of their design, but this study illustrates the importance of designing low-pressure loss diffusers for these heat exchangers. The more detailed models of A'Barrow would have allowed for a more precise evaluation of these effects, but their added complexity and computational effort bears little reason if the maximum available space cannot be specified properly first.

4.2.2.11 Further influences

This section not only illustrated the CCTF's sensitivity to some of the fixed modelling assumptions such as the absolute surface roughness of the heat exchanger walls, it also helps to understand how the optimal CCTF2050 has been identified. A number of additional design parameters could be changed and their possible implications are discussed in chapter 5, but the quantification of their effects would go beyond the scope of this study.

4.3 CCTF2050 with ITR

This section describes the results of multiple rounds of parametric design studies of the combined cycle engine with inter-turbine reheat CCTF2050_ITR. The structure is similar to the section describing the CCTF2050 without inter-turbine reheat. First, the identified optimal configuration is presented in section 4.3.1 together with a selection of sensitivities. Afterwards, parametric and sensitivity studies are presented in detail, though the number of studies presented in 4.3.2 is significantly lower than for the CCTF2050. Many of the trends observed during the maturation of the CCTF2050_ITR are qualitatively very similar to those presented for the CCTF2050. The driving forces behind those trends are furthermore the same, even if their quantitative effects may differ due to the different thermal conditions of the cycle. A repetition of the resulting discussions based on similar plots therefore does not add much to the understanding of the concept and is avoided as much as possible.

Only two parameter groups are discussed anew for the CCTF2050_ITR, due to their large influence on mission fuel burn and because their individual optima substantially changed relative to the CCTF2050.

First, the compressor inlet conditions of the bottoming cycle and the coolant mass flow are reinvestigated. The higher LPT exhaust gas temperature of the main engine results in potentially higher peak temperatures in the bottoming cycle. The optimal S-CO₂ compressor inlet temperature is therefore increased for a given pre-cooler design, even though the thermal efficiency of the cycle is increased. A higher S-CO₂ mass flow and useful power output are direct consequences. The higher thermal efficiency and larger power contribution of the bottoming cycle also affect the balance of the bypass pressure losses and the pre-cooler size.

The second revisited set of design inputs is the combination of bypass ratio and fan pressure ratio. The increased mean temperature in the turbines of the main engine increases the core specific power significantly. The bypass ratio of the optimal CCTF2050_ITR is therefore considerably higher than that of the CCTF2050 even at slightly smaller fan diameters. The optimal fan pressure ratio will therefore also differ. However, in conventional turbofan engines the optimal fan pressure ratio usually decreases with BPR, leading to lower specific thrusts. Larger BPRs often mean larger fan diameters for conventional turbofan engines, which also necessitate reduced FPRs in order to retain high isentropic efficiencies. A combined cycle engine however benefits from higher pressures in the pre-cooler, because the heat transfer coefficients are increased. Being able to shrink the core due to the higher specific power and therefore increasing the BPR without increasing the fan diameter means that the FPR can actually be increased without penalising the fan efficiency.

The reheat combustor of the CCTF2050_ITR introduces two new design parameters: the turbine work split of the HPT at cruise (WS_{HPT}) and the TET to the second stage of the HPT. Parametric studies are presented for both parameters to illustrate their effect on the mission fuel burn reduction of the CCTF2050_ITR.

4.3.1 Optimal Cycle Performance and Design

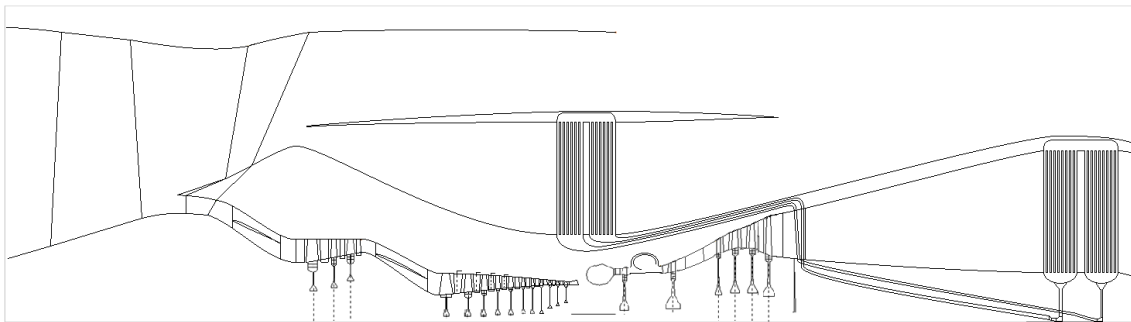


Figure 4-29: Scaled engine cross-section of CCTF2050_ITR

Figure 4-29 depicts the engine cross-section of the CCTF2050_ITR with all components to scale. Many of the same issues that were already identified when examining the CCTF2050 cross-section in Figure 4-7 apply for the CCTF2050_ITR cross-section as well. The extreme pressures and relatively low mass flows of the bottoming cycle still lead to very small turbomachinery components, even if the S-CO₂ mass flow is approximately 15 % larger than in the CCTF2050. Additionally, the connecting ducts between the heat exchangers and the turbomachinery components were not specifically

designed for the CCTF2050_ITR either. The depicted ducts and pipes are only representative and do not reflect actual component designs. Finally, the shape of the inlet to the pre-cooler duct is also sub-optimal. The dimensions arise from the implemented methodology that only differentiates between the Mach number at the hub of the FOGV and another Mach number at the tip. A more differential analysis of the flow field out of the FOGV would undoubtedly lead to a more aerodynamic design.

Table 4-7 contains the most important performance and preliminary component design parameters of the optimal CCTF2050_ITR as identified through the parametric design studies. Again the cycle is referred to as 'optimal' despite the fact that no automatic optimiser was used. The specific fuel consumption of the optimal combined cycle engine with inter-turbine reheat is approximately 2.5 % lower than that of the TF2050. Furthermore, the fan of the CCTF2050_ITR is almost 10 cm smaller and the engine weighs only about 144 kg more than the TF2050. The CCTF2050_ITR therefore uses about 3.25 % less fuel on a 7000 nm mission than the TF2050. The mission fuel burn is reduced by about 47.6 % relative to the baseline LRTF2000.

The aircraft weight and drag are reduced relative to the TF2050 configuration, leading to lower thrust requirements for the engine as seen in the table. The OPR is noticeably reduced relative to the TF2050 and the CCTF2050, because the BPR is significantly larger, the core is smaller and the core turbomachinery therefore less efficient. The polytropic efficiency of the HPC is reduced by almost 1.4% due to excessively small blades. The HPC delivery temperature is actually the same for all three cycles. The specific thrust of the CCTF2050_ITR is increased relative to the CCTF2050 and the TF2050, due to the higher fan pressure ratio. The exhaust gas temperature of the CCTF2050_ITR is slightly lower than that of the CCTF2050, despite the increased LPT EGT, because the number of tube columns per MHX pass has increased from 6.5 to 9.5.

The bottoming cycle provides about 1.02 MW of useful power to the LP shaft of the main engine, representing about 7.8 % of the total power delivered to the LP system. The increase in useful power relative to the CCTF2050 illustrates the synergy between the bottoming cycle and inter-turbine reheat. Unfortunately, the combined cycle performance is still largely determined by the pre-cooler size and performance, which is why the fuel burn benefit has not substantially improved. The total weight of the bottoming cycle is estimated to about 683 kg, an increase of almost 160 kg relative to the bottoming cycle of the CCTF2050. However, the total engine weight of the CCTF2050_ITR is actually lower than that of the CCTF2050, due to the small core and fan. The pre-cooler accounts

for approximately 38 % of the bottoming cycle's weight, while the MHX accounts for just over 47 %. This illustrates the shifted balance in heat exchanger designs of the CCTF2050_ITR due to the higher working temperatures.

Table 4-7: Performance and component parameters of optimal CCTF2050_ITR

Parameter	Units	Value	Parameter	Units	Value
F_N	kN	37.0	BC W_{S-CO2}	kg/s	13.7
F_N Mid Cruise	kN	32.3	BC Pwr	kW	1022
SFC	g/(kN*s)	12.56	BC Weight	kg	683
SFC Mid Cruise	g/(kN*s)	12.31	MHX N_{col}	-	9.5
OPR	-	61.6	MHX M_{fr}	-	0.1
sF_N	m/s	91.6	MHX a_{tube}	mm	7.5
BPR	-	21.8	MHX x	-	0.775
BP $\Delta P/P$	%	1.32	MHX x_{inv}	-	0.9
FPR_{BP}	-	1.43	MHX $\Delta P/P$	%	8.5
IPC PR	-	2.64	MHX ΔT_{air}	K	256
HPC PR	-	18.1	MHX M	kg	323
HPC η_{poly}	%	89.9	Pre N_{col}	-	7.5
T_{30}	K	880	Pre M_{fr}	-	0.089
TET₁	K	1580	Pre a_{tube}	mm	7.5
TET₂	K	1580	Pre x_{inv}	-	0.93
WS_{HPT}	-	0.45	Pre $\Delta P/P$	%	8.4
BC $T_{in,Cmp}$	K	314	Pre ΔT_{air}	K	92
BC $P_{in,Cmp}$	MPa	8.47	Pre M	kg	261
BC PR	-	4	Pre W_{cool}	kg/s	44
d_{Fan}	m				2.75
Weight	kg				5306
ΔFB_{y2050}	%				-3.25
ΔFB_{y2000}	%				-47.61

The detailed design of the compressor casings and auxiliary systems therein are beyond this project. Hence, the inner annulus diameter of the pre-cooler was not changed from the CCTF2050 despite the smaller engine core to avoid overly optimistic performance estimates.

The listed pressure losses for the external flows passing through the MHX and the pre-cooler are slightly higher than in the CCTF2050. For the pre-cooler this increase arises from a combination of reasons. The heat exchanger tubes are longer and the heat transfer surface area greater, leading to higher friction losses. Additionally, the number of columns per pass has increased slightly. Finally, the temperature increase of the coolant mass flow is greater, leading to a stronger flow acceleration of the air inside the heat exchanger. The pressure losses of the exhaust gases passing through the MHX are primarily increased, because the number of heat exchanger tube columns and the exhaust gas inlet temperature are considerably higher. Nevertheless, both pressure losses are within a reasonable range, though they are likely more significant at climb or take-off conditions.

The optimal tube chord length is the same as for the CCTF2050, even though the higher working temperatures in the cycle lead to higher wall temperatures and therefore to different internal tube geometries for the heat exchanger tubes. It proved beneficial for the CCTF2050_ITR to have leaned MHX tubes at the hub ($x_{inv} = 0.9$).

As expected, the optimal compressor inlet temperature is slightly higher in the engine featuring inter-turbine reheat. The increased peak temperature in the bottoming cycle means that the efficiency of the cycle rises, but the inlet temperature of the pre-cooler rises as well. A slight increase in pre-cooler exit temperature allows for a larger S-CO₂ mass flow to be cooled down, because the specific heat capacity varies less inside the pre-cooler. The larger S-CO₂ mass flow in turn leads to an increased useful power output, ultimately improving the mission fuel burn. The compressor inlet pressure on the other hand was not increased, as the resulting change in internal tube geometry in the MHX eliminated any possible benefit. The coolant mass flow is about 15 % greater in the CCTF2050_ITR than in the CCTF2050. The larger coolant mass flow also enables a greater S-CO₂ mass flow to be cooled, raising the useful power output. The 61 % increase in useful power output is furthermore large enough to offset the increased bypass pressure losses due to the larger pre-cooler outer diameter. Section 4.3.2.1 discusses the intricacies of these three design inputs in further detail.

The reheat combustor is operated differently in a combined cycle engine than it would be if added to a conventional turbofan. The parametric studies determined that the additional heat introduced in the second combustor is crucial to the performance improvement of the bottoming cycle, so the outlet temperatures of the two combustors are equal and primarily limited by lifing considerations for the respective downstream

turbine stages. The studies also concluded that it is important to introduce the reheat combustor as early in the expansion as possible. Within the investigated range, the first stage of the HPT ideally produces only 45 % of the total power of the HPT. The present models would have predicted a greater fuel burn benefit if the reheat combustor were introduced even earlier. However, this would probably require a significant design change of the HPT stages including different component efficiencies, neither of which the current models are capable of. The heat input in the reheat combustor also means however, that an additional 5.7 % of the compressor delivery air are needed to cool the second stage of the HPT.

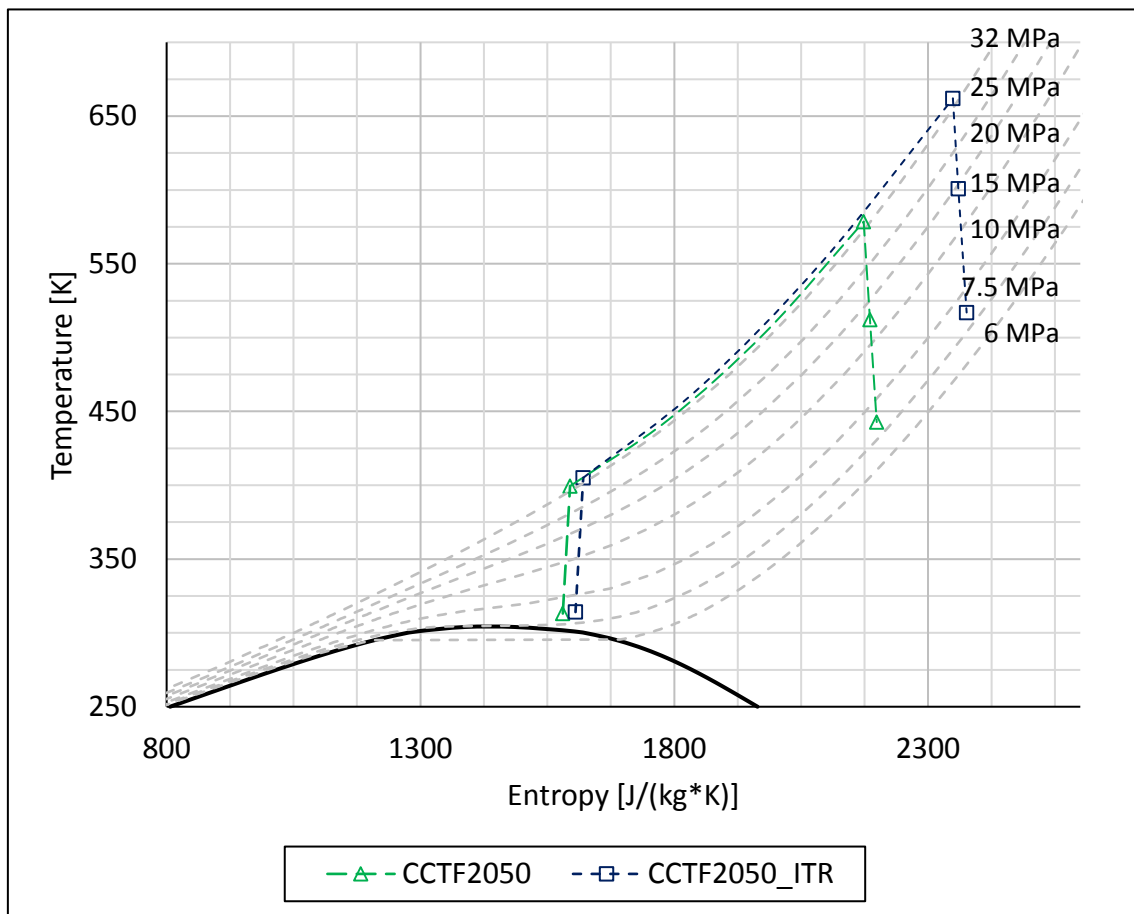


Figure 4-30: Comparison of bottoming cycle conditions of CCTF2050 and CCTF2050_ITR

Figure 4-30 illustrates the different operating conditions of the bottoming cycles in the CCTF2050 (green) and the CCTF2050_ITR (blue). The reheat combustor leads to a notable increase in peak temperature in the bottoming cycle. Hence, the power output and the efficiency of the cycle are increased. The T-S-diagram also shows that the inlet conditions to the S-CO₂ compressor differ between the two cycles. The overall working

pressures in the system are largely the same however, in order to retain light tube designs for the pre-cooler and the MHX.

The T-S-diagram of the combined cycle (see Figure 4-31), illustrates the significant heat input in the reheat combustor and the noticeably larger effect of the MHX when compared to the CCTF2050 (see Figure 4-8 in section 4.2.1).

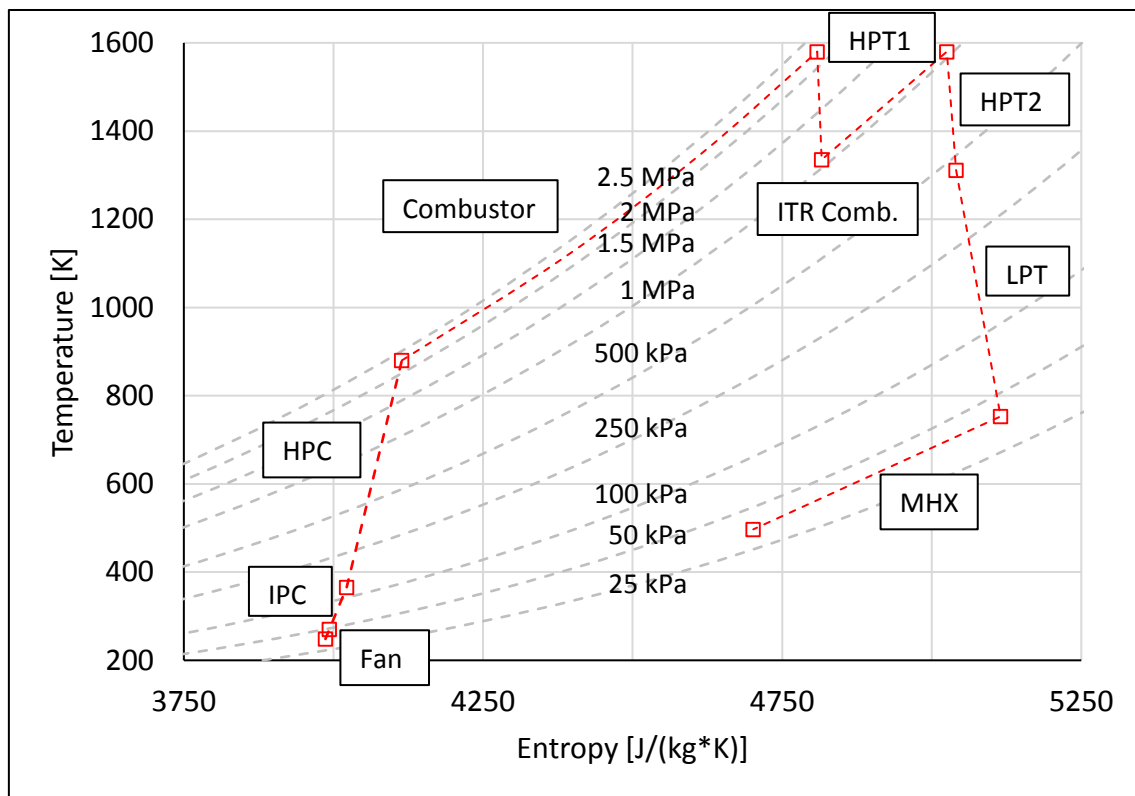


Figure 4-31: T-S-diagram of optimal CCTF2050_ITR

Similarly as CCTF2050, the mission fuel burn benefit of the CCTF2050_ITR depends strongly on the modelling assumptions and the design inputs chosen. Table 4-8 indicates the sensitivity of the performance of the CCTF2050_ITR to isolated changes of most technology assumptions and design inputs. The relative changes of the individual design parameters are similarly large as during the investigation of the CCTF2050. The effect of each of the parameters was investigated in isolation, with all other parameters being equal to the CCTF2050_ITR data shown in Table 4-7.

Once again, it is important to note that the presented sensitivities are specific to the changes of the individual design parameters in the table and they should not be extrapolated. The relative deviations should also not be used to linearly interpolate

smaller changes, as the trends are rarely linear. Furthermore, the individual deviations of two or more parameters should not simply be added to assess their combined effect.

Table 4-8: Selection of sensitivities of the CCTF2050_ITR

Component	Parameter	Units	Value	Δ FB [%]
Main Engine	BPR	-	21.25 / 22.75	+0.0 / +0.02
	TET_2	K	1555 / 1580	+0.23 / +0.0
	WS_{HPT}	-	0.45 / 0.5	+0.0 / +0.62
S-CO₂ Compressor	T_{in}	K	312 / 315	+0.2 / +0.0
	P_{in}	MPa	8.27 – 8.57	+0.01 / +0.02
	PR_{BC}	-	3 / 4	+1.05 / +0.0
Pre-cooler	$M_{fr,C}$	-	0.08 / 0.1	+0.12 / +0.17
	N_{col}	-	6.5 / 8.5	+0.19 / +0.0
	$a_{tube,C}$	m	0.006 / 0.01	+0.37 / +0.27
	x_{inv}	-	0.9 / 1.0	+0.0 / +0.06
	$W_{coolant}$	kg/s	40 / 48	+0.17 / +0.03
	L_{diff}	m	0.6 / 1.5	+0.89 / -0.15
MHX	N_{col}	-	8.5 / 10.5	+0.09 / +0.0
	$a_{tube,H}$	m	0.006 / 0.01	+0.14 / +0.27
	x_{inv}	-	0.9 / 1.0	+0.0 / +0.05
	L_{diff}	m	0.6 / 1.5	+0.45 / -0.08

The individual ranges of the investigated parameters were furthermore kept similar to the studies discussed for the CCTF2050 to make the results more easily comparable. The engine's sensitivity to the HPT turbine work split was only shown for half the total range, because the effect is already quite severe therein. TET_2 was only varied in a relatively small range as well, because its effect on mission fuel burn is highly non-linear and even a 25 K variation already leads to a substantial change in mission fuel burn. Sections 4.3.2.3 and 4.3.2.4 explain the individual effects of the turbine work split and TET_2 in greater detail.

Overall it must be said that most of the listed parameters have a significant influence, but the coolant mass flow, the compressor inlet temperature and the compressor inlet pressure are once again three of the most important of the design inputs of the bottoming cycle, even if Table 4-7 does not emphasize that in the depicted range. The component

efficiencies of the S-CO₂ turbomachinery components obviously have a substantial effect on the combined cycle's performance as discussed in section 3.5.2. Ultimately, the pre-cooler is still the component that dictates the performance of the bottoming cycle the most, but the increased heat input and hence the power generated by the bottoming cycle mean that an increase in outer bypass pressure losses can be accepted.

4.3.2 Parametric and Sensitivity Studies

The parametric and sensitivity studies presented in this section were not performed around the optimal CCTF2050_ITR configuration presented above. The x_{inv} parameter, which determines the degree to which the tubes are already leaned at the hub of an involute heat exchanger, was increased to 96 % respectively in order to improve convergence. This change altered the fuel burn minimum by about 0.05 percentage points to 3.2 %. The change also increased the optimal coolant mass flow a little, due to the changed heat duty balance between the MHX and the pre-cooler, but again the effect is relatively small (about 1 kg/s change in optimal coolant mass flow). These penalties and variations were accepted to more adequately show the relative effects of the presented design inputs, as the amount of cases that could be evaluated would otherwise have been limited by convergence issues.

This is the only change from the optimal CCTF2050_ITR in the presented studies however, aside from the investigated parameters in the respective sections obviously. Similar to the studies of the CCTF2050, not the whole of the investigated design space is presented in the individual studies, in order to better isolate the performance around the optimal CCTF2050_ITR. The individual parameters affect the fuel burn estimates more strongly when further away from the optimal cycle. Each section clearly explains whether only one parameter was investigated in a parametric study or whether multiple parameters were changed at the same time in order to identify the minimally possible effect of changing the investigated input.

4.3.2.1 Compressor Inlet Conditions and Coolant Mass Flow

The compressor inlet temperature and pressure determine how strong the real gas behaviour is in the compressor and the pre-cooler. Strong real gas behaviour reduces the power consumption of the compressor and therefore improves the power balance between the turbines and the compressor. Unfortunately, the variation in specific heat capacity for the flow inside the pre-cooler also means that the same pre-cooler will only be able to cool a smaller mass flow from the turbine exit to the compressor inlet

conditions. Hence, the useful power output is actually reduced. The higher LPT exhaust gas temperature of the main engine begets an increased peak temperature in the bottoming cycle. While this leads to an increased thermal efficiency in the bottoming cycle, it still raises the S-CO₂ inlet temperature for the pre-cooler, requiring either a larger pre-cooler or an increased pre-cooler exit temperature to be accepted. A combination of the two proved most efficient overall and the isolated influence of varying the inlet conditions is presented here.

The trade-off becomes even more complex through the incorporated bypass-pressure loss model, which varies the pressure loss according to the acceleration of the outer bypass flow. The flow acceleration is due to a flow path contraction caused by the outer diameter of the pre-cooler. The outer diameter of the pre-cooler is largely dictated by the coolant mass flow, making this a further crucial design input to be investigated here. The nature of this problem has not changed from the CCTF2050 to the CCTF2050_ITR, but the higher peak temperatures in the bottoming cycle and the subsequently larger useful power outputs are able to offset greater outer bypass flow pressure losses, which means that higher coolant mass flows prove to be ideal.

Figure 4-32 illustrate these effects clearly. The bottom three lines show that increasing the coolant mass flow increases the mission fuel burn benefit until an optimum is reached between 44 and 46 kg/s, depending on the compressor inlet temperature. This is significantly higher than the optimal coolant mass flow of the CCTF2050 (around 38 kg/s). The useful power increase due to the larger heat input into the bottoming cycle outweighs the incurring greater outer bypass pressure losses and component weight. All cycles involved in this part of the study featured a compressor inlet pressure of 8.47 MPa, because that pressure offered the greatest fuel savings. The optimal coolant mass flow reduces with increasing compressor inlet temperature, because the pre-cooler requires less coolant to restore a given amount of S-CO₂ to the compressor inlet conditions. However, the S-CO₂ mass flow also reduces when the coolant mass flow is reduced, because the pre-cooler gets smaller. The outer bypass pressure losses also reduce if the pre-cooler outer diameter reduces, but this effect is less influential in the CCTF2050_ITR than it is in the CCTF2050. The combination of all these effects is the reason why the intermediate compressor inlet temperature of 314.11 K gives the highest fuel burn benefit in the illustrated range.

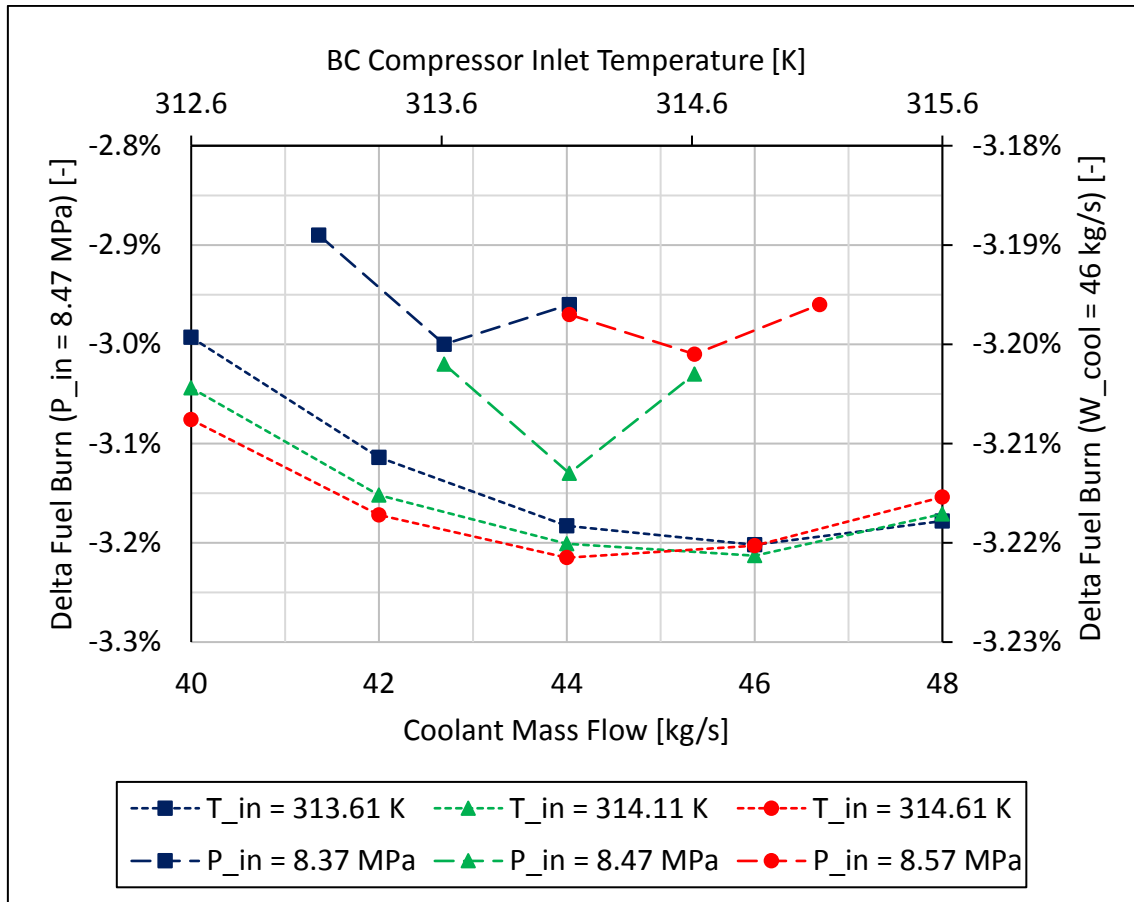


Figure 4-32: Influence of compressor inlet conditions and coolant mass flow on CCTF2050_ITR

The upper three lines of the plot show the optimal combinations of compressor inlet conditions and coolant mass flow. Each line depicts the fuel burn benefit of the CCTF2050_ITR at a constant compressor inlet pressure over a range of inlet temperatures. The coolant mass flow is fixed to 46 kg/s in all cases as indicated by the relevant vertical axis on the right hand side of the plot. The optimal compressor inlet temperature clearly depends on the inlet pressure, similar to the CCTF2050. The optimal compressor inlet temperature is raised by 1 K for each of the investigated inlet pressure levels relative to the CCTF2050. Increasing the compressor inlet pressure initially improves the fuel burn benefit, because the limited heat duty of the pre-cooler is more influential than fact that the S-CO₂ compressor does not benefit from very strong real gas behaviour of the flow. However, compressor inlet pressures greater than 8.47 MPa did not yield greater fuel savings due to the necessarily heavier tube designs in the MHX, as can be seen from the 8.57 MPa cases in the plot.

Overall, the trends are qualitatively similar to those observed during the investigation of the CCT2050, but the optimal coolant mass flow and compressor inlet temperature are noticeably higher. It cannot be dismissed that smaller discrete variations in compressor inlet conditions and coolant mass flow would improve the mission fuel burn estimate. However, it is unlikely that this would change the fuel burn benefit by more than 0.1 percentage points, making such design explorations unreasonable in this case. Full-fledged optimisation studies would help to quantify the additional benefit.

4.3.2.2 Bypass Ratio

The bypass ratio of the CCTF2050_ITR exhibits a similar qualitative behaviour as for the CCTF2050, though the absolute variation in fuel burn benefit is noticeably smaller. Figure 4-33 illustrates how the bypass ratio affects the performance and design of the combined cycle engine with inter-turbine reheat. The fuel-burn-optimal fan pressure ratio was identified and is presented for each bypass ratio. The compressor inlet conditions and coolant mass flow however were not varied in this study to find the optimal combination for each BPR.

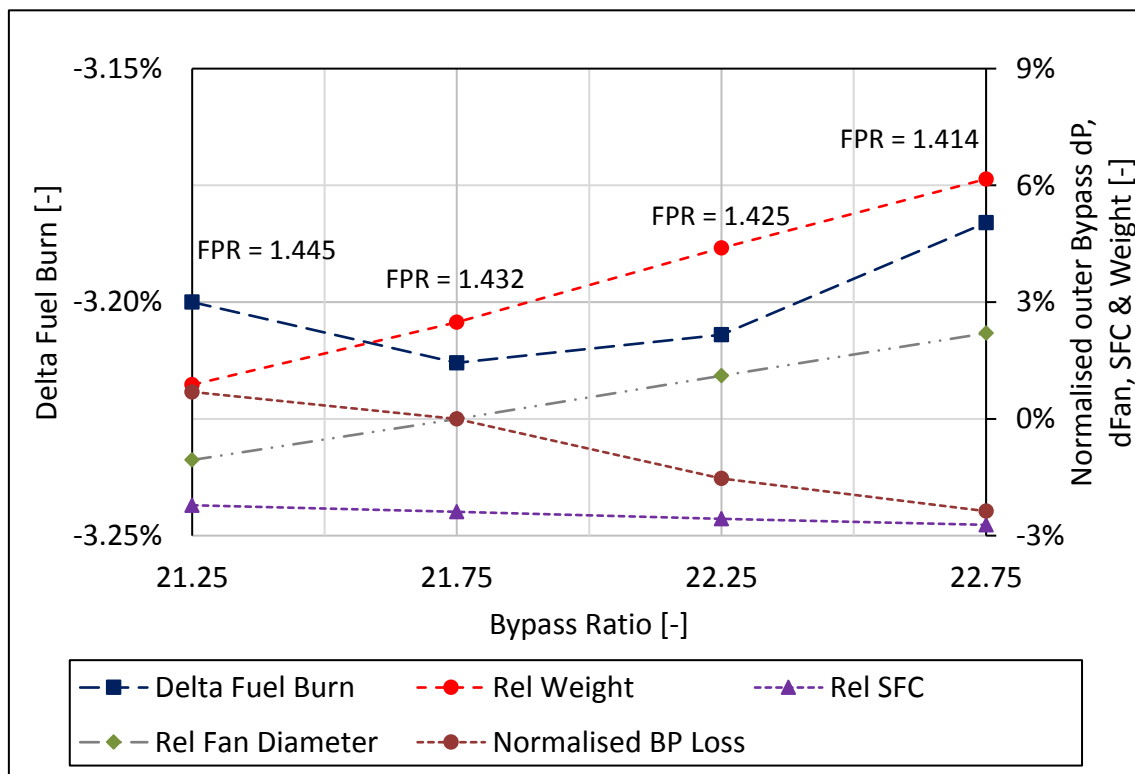


Figure 4-33: Influence of bypass ratio on CCTF2050_ITR

The fuel burn benefit reaches its maximum at a significantly higher bypass ratio than it did for the CCTF2050. The optimal specific thrust and the necessary total net thrust of

the engines remains largely constant however, so the increased bypass ratio primarily signifies a change in core mass flow. The additional heat input in the second combustor raises the exit temperatures of HPT2 and the LPT, leading to an increase in core specific power. Consequently, the core mass flow was reduced by about 24 % from the CCTF2050 to the CCTF2050_ITR. The smaller but hotter LPT exhaust gas flow helps to increase the thermal efficiency and the useful power output of the bottoming cycle, further improving the specific power of the core. The reduced core size is the main reason why the change in total engine weight relative to the TF2050 is smaller for the CCTF2050_ITR than it is for the CCTF2050.

In the investigated range however, any BPR variation primarily affects the fan diameter, increasing the total engine weight with BPR. The SFC of the engine improves with bypass ratio, as the specific thrust of the engine is reduced. Furthermore, the relative portion of bypass flow passing through the pre-cooler reduces with BPR, because the coolant mass flow was kept constant while the engine inlet mass flow increased. Hence, the outer diameter of the outer bypass duct increases, and the acceleration and the pressure losses in the outer bypass flow decrease. The bypass pressure losses are normalised against the optimal CCTF2050_ITR with a BPR of 21.75. The changes in SFC, total engine weight and fuel burn are shown relative to the TF2050.

Overall, the SFC reductions are not enough to indefinitely offset the increased engine weight and drag due to the larger fan when BPR increases, and the fuel burn benefit is reduced. The total variation in fuel burn in the investigated range may be reduced if the compressor inlet conditions and coolant mass flow were improved upon for each investigated BPR (except 21.75), but the qualitative trends are not likely to change.

4.3.2.3 High Pressure Turbine Work Split

The position of the reheat combustor is very influential on the performance of the combined cycle engine, as has been shown by multiple theoretical studies [38], [39] and the industrial GT24/GT26 engine originally developed by Alstom. Given the constraints on the main engine in terms of compressor delivery temperature, turbomachinery scaling and cooling requirements, introducing inter-turbine reheat to a conventional turbofan worsens the mission fuel burn, as shown for the TF2050 in section 4.1. The fuel burn penalty was smallest, if the reheat combustor remained unlit and the work split of the HPT (see equation 3-120 in section 3.6.1) was as small as possible.

A parametric study investigating the effect the position of the reheat combustor has on mission fuel burn concluded that the HPT work split should also be as small as possible on the CCTF2050_ITR. The compressor inlet conditions and the coolant mass flow were also varied for every investigated HPT work split, to identify the minimal effect the work split has on mission fuel burn. Increasing the HPT work split at design point from 45 to 50 % reduces the fuel burn benefit by about 0.6 percentage points, or almost 20% of the total benefit. The TET of HPT2 is equal to the TET of HPT1 (section 4.3.2.4) in either case. Additionally, the optimal compressor inlet temperature increases and the optimal coolant mass flow decreases if the work split is increased, as shown in Figure 4-34.

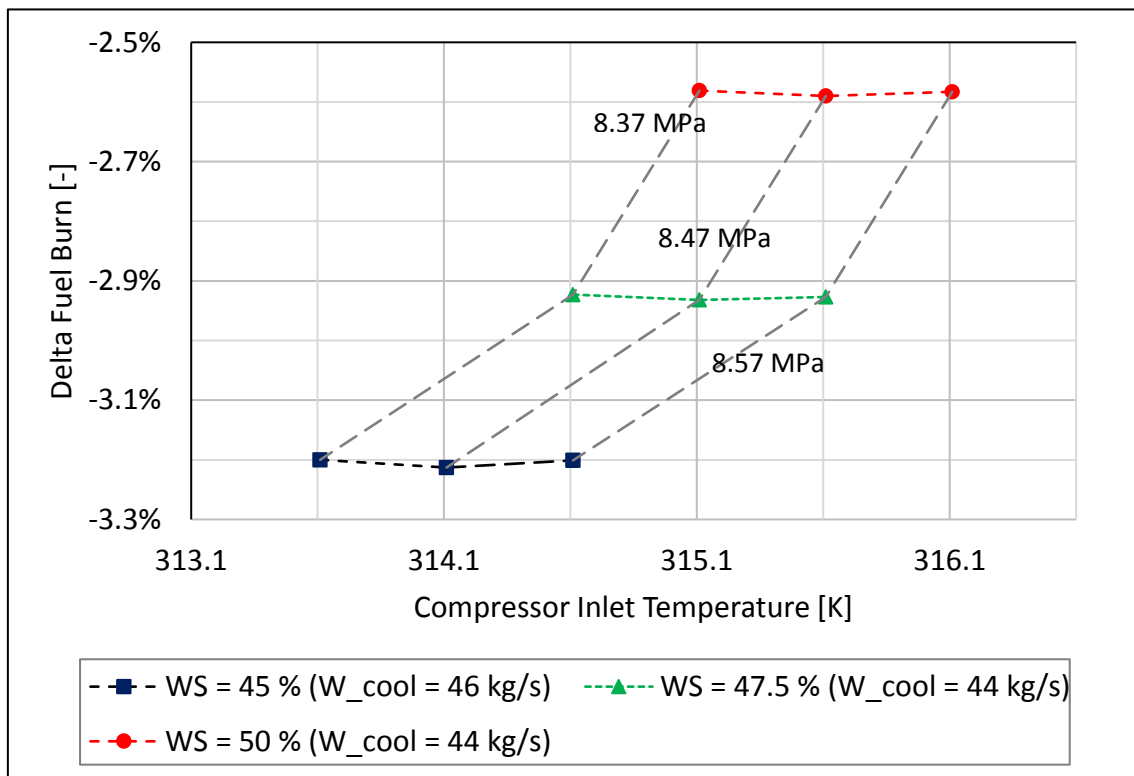


Figure 4-34: Influence of HPT work split at design point on CCTF2050_ITR

A larger HPT work split leads to an increased LPT EGT and hence a potentially higher peak temperature in the bottoming cycle. This begets an increase in optimal compressor inlet temperature as can be seen in Figure 4-34, if the heat duty of the pre-cooler is not increased as well. Increasing the heat duty of the pre-cooler would entail either further tubes and air-side pressure losses or a greater coolant mass flow and hence greater bypass pressure losses. The number of tube columns was not varied during the parametric study presented in this section to simplify the results, but the effect of the coolant mass flow was investigated again. Contrary to initial expectations, the optimal

pre-cooler coolant mass flow actually reduces with increasing work splits. The increased optimal compressor inlet temperature means that a smaller S-CO₂ temperature difference has to be achieved by the pre-cooler, lowering the cooling flow requirements. At the same time, the effectiveness of the pre-cooler increases because the S-CO₂ inside the pre-cooler undergoes smaller variations in specific heat capacity, meaning a larger S-CO₂ mass flow can be cooled. Additionally, a smaller coolant mass flow results in a smaller outer diameter of the heat exchanger, lowering the bypass pressure losses. The combination of all these effects leads to the observed reduction in optimal coolant mass flow with increasing turbine work split. Unfortunately, the lower coolant mass flow leads to a larger air-side temperature variation in the pre-cooler, which accelerates the airflow and increases cold-side pressure losses and the SFC.

Initially it might be expected, that increasing the HPT work split may help to reduce the mission fuel burn, because the LPT exhaust gas temperature is increased. The peak temperature of the bottoming cycle therefore increases, which should improve the performance of the bottoming cycle and the whole engine overall. However, in order to utilise the additional heat input, the MHX and the pre-cooler would have to be significantly bigger as well, which did not prove beneficial in this investigation. The size of the MHX could still be increased, even though it is quite large already when compared to the MHX in the CCTF2050, because it does not affect the rest of the engine outside of a higher component weight. However, the size of the pre-cooler is a compromise between the pressure losses in the outer bypass, the air-side pressure losses, the component weight and the inlet conditions to the S-CO₂ compressor. Consequently, increasing the size and heat duty of the pre-cooler does not yield a fuel burn benefit.

Therefore, the higher LPT exhaust gas temperature remains largely unexploited by the MHX if the HPT work split is increased and the amount of waste heat in the core exhaust gases actually increases. As a result, the specific thrust of the engine increases with increasing work split, which leads to a lower propulsive efficiency of the engine and ultimately a larger SFC. The higher power output of the bottoming cycle due to the higher peak temperature also leads to a more power dense core engine and hence a reduced fan diameter at a given bypass ratio. Unfortunately, the smaller core mass flow means that the main engine's turbomachinery and the MHX are smaller and lighter, which improves the total engine weight, but reduces the efficiencies of HPC and HPT.

In the end, the weight and fan diameter reductions are not enough to offset the increase in SFC and the fuel burn benefit is diminished if the HPT work split is increased.

Decreasing the HPT work split below 45 % was deemed not feasible using the present models, as this would require a considerable design and efficiency change for the HPT stages.

4.3.2.4 Turbine Entry Temperature to HPT2

The inlet temperature to the second stage of the HPT is one of the design handles of the present models and dictates the fuel flow in the reheat combustor. TET_2 is limited to 1580 K, same as the main combustor, and is extremely influential on the combined cycle performance as illustrated in Figure 4-35.

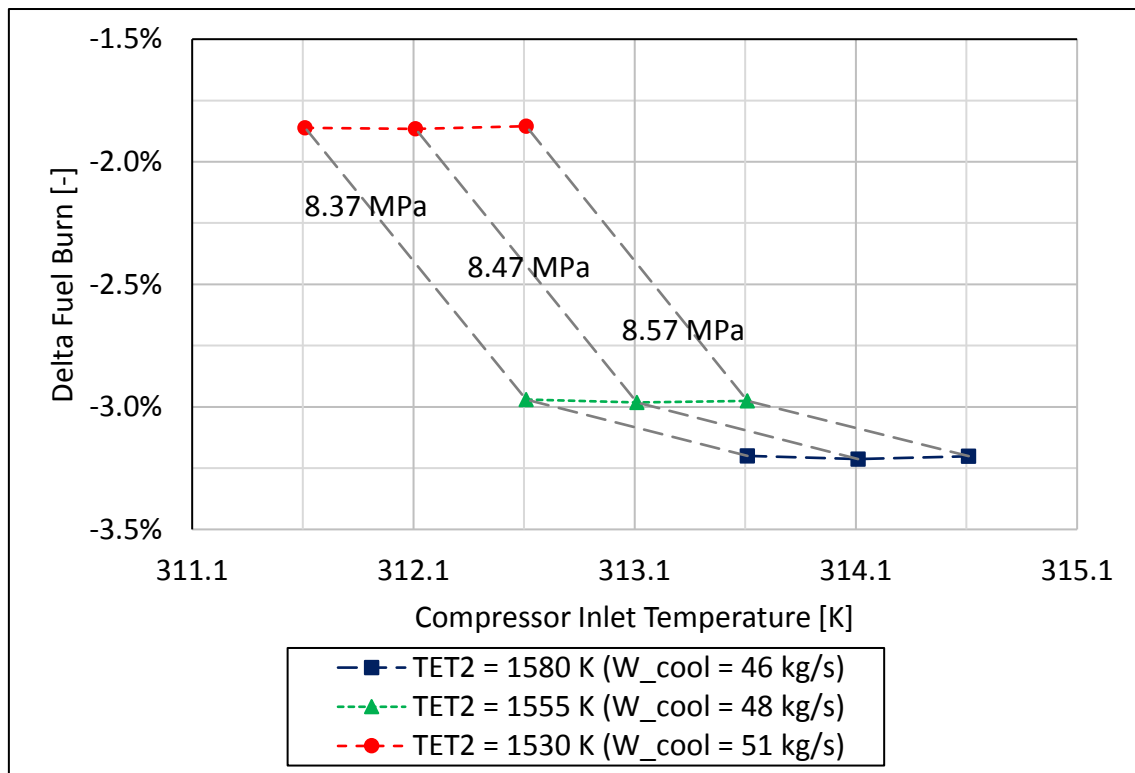


Figure 4-35: Influence of TET_2 on CCTF2050_ITR

Three TET_2 levels relatively close to the temperature limit were investigated in this parametric study and the optimal compressor inlet conditions and coolant mass flows were determined for each case. The fuel burn benefit appears to hardly change with compressor inlet pressure when TET_2 is kept constant, but that is primarily due to the relatively large scale of the vertical axis of the plot when compared with other graphs presented in this chapter. Finding the optimal combination of inlet temperature and pressure for the S-CO₂ compressor is still crucial to maximising the fuel burn benefit of the CCTF2050_ITR.

By reducing TET_2 by 25 K from the 1580 K of the optimal CCTF2050_ITR to 1555 K, the fuel burn benefit is reduced by about 0.2 percentage points. However, the influence of TET_2 is very non-linear and a further 25 K reduction to 1530 K reduces the fuel burn benefit further by about 1.1 percentage points. A combination of a number of effects lead to this highly non-linear behaviour.

Reducing TET_2 reduces the mean temperature in the downstream turbine stages and therefore the specific power of the core engine. Hence, the core mass flow and the fan diameter increase at a given bypass ratio, increasing the total engine weight and drag. The LPT EGT is also reduced by about 30 K as a result of a 25 K drop in TET_2 . Hence, the MHX is exposed to a larger, but colder exhaust gas flow. Additionally, the increase in core mass flow has a bigger influence on the inlet flow area than the reduction in flow temperature, which means that the inlet area to the MHX and hence the heat exchanger get bigger.

The optimal inlet temperature to the S-CO₂ compressor is also reduced by about 1 K for every 25 K of TET_2 , while the optimal coolant mass flow increases by 2 to 3 kg/s. This is in spite of the fact that the increased coolant mass flow leads to a greater pre-cooler outer diameter, which in turn increases the pre-cooler's weight and the pressure loss in the outer bypass flow. The bypass pressure loss increases exponentially with coolant mass flow, which is why the relative change is much greater when TET_2 is changed from 1555 K to 1530 K than it is when TET_2 is changed from 1580 K to 1555 K. However, the increased heat duty of the pre-cooler and the lowered compressor inlet conditions still make for the most fuel burn effective combination.

Unfortunately, the optimal combination of the coolant mass flow and compressor inlet conditions is not enough to mitigate the useful power loss due to the lower LPT EGT. The useful power generated by the bottoming cycle drops by about 4.5 % for every 25 K TET_2 is reduced by. The total engine weight furthermore increases by more than 200 kg per 25 K reduction in TET_2 , about 70 kg of which are due to the larger MHX and pre-cooler. These effects, combined with the increased drag due to the larger fan diameter, are the main reason why reducing TET_2 diminishes the potential fuel burn benefit so strongly.

The fuel burn benefit is slightly helped by the reduction in cooling flow necessary for HPT2 in order to keep the blade metal temperature constant irrespective of TET_2 . The necessary cooling flow is extracted from the HPC delivery air flow and is reduced by

about 0.54 percentage points for every 25 K TET_2 is reduced. However, the effect is not strong enough to avoid the reduction in fuel burn benefit.

It is important to note however that the strong non-linear behaviour depicted in Figure 4-35 is partially due to non-optimal OPRs. The optimal OPR decreases significantly if TET_2 is reduced, due to the reduction in core specific power and LPT EGT. At $TET_2 = 1555\text{ K}$, the effect is very small, but at $TET_2 = 1530\text{ K}$ decreasing the pressure ratio of the HPC can raise the fuel burn benefit by more than 0.6 percentage points. Figure 4-36 illustrates a parametric study in which the pressure ratio of the HPC is varied. The compressor inlet conditions, the coolant mass flow and TET_2 are kept constant at the values determined optimal for the case with $TET_2 = 1530\text{ K}$ presented in Figure 4-35. The lowest HPC pressure ratio investigated in this study is 17, because any further reduction would probably require a significant redesign of the whole HP compressor. The HPC would probably require fewer stages and feature larger blades in the last stage, both of which can improve the efficiency of the component. The fuel burn benefit, the change in SFC and the change in total engine weight are all shown relative to the TF2050.

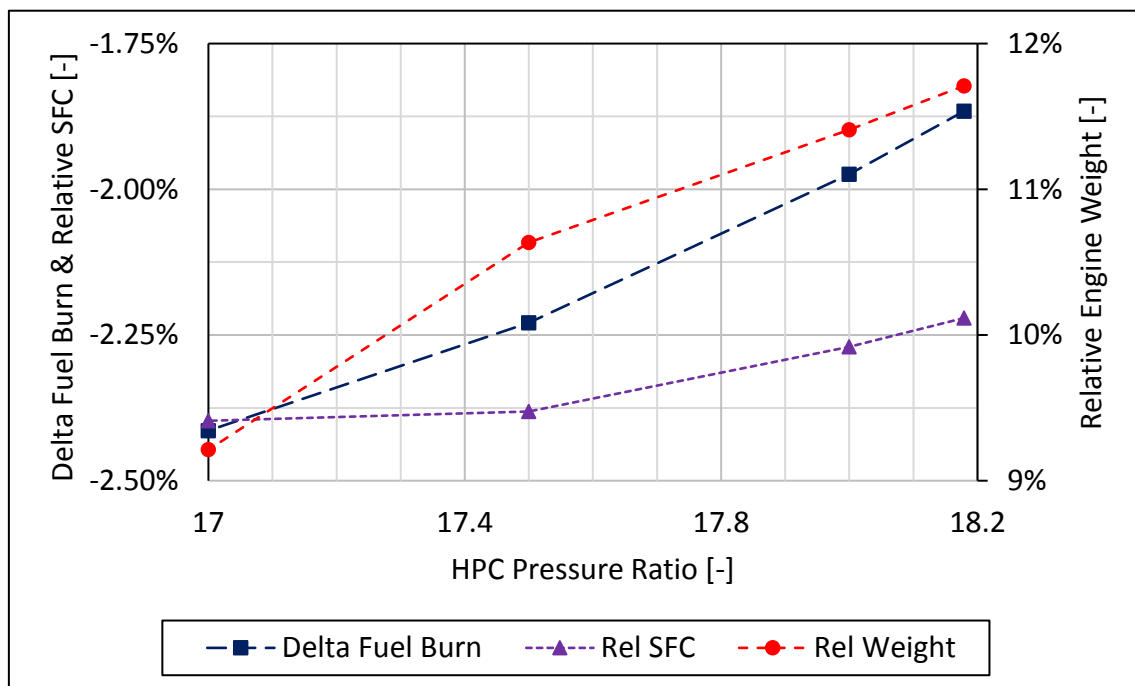


Figure 4-36: Sensitivity of CCTF2050_ITR to HPC PR if $TET_2 = 1530\text{ K}$

Reducing the OPR of the engine raises the LPT EGT, improving the heat transfer to the bottoming cycle but lowering the thermal efficiency of the engine. The same would be

true if $TET_2 = 1580\text{ K}$, but in that case a higher OPR is favourable to a greater heat input to the bottoming cycle. In the case of this study, the increased EGT raises the useful power output of the bottoming cycle, which partially offsets the reduction in thermal efficiency due to the lower OPR. The temperature and pressure of the exhaust gases exiting the MHX are also increased, raising the nozzle pressure ratio and the net thrust of the core. This in turn raises the specific thrust of the engine, leading to a smaller engine inlet mass flow and fan diameter. While the fan diameter reduction results in less drag and hence a fuel burn improvement, it also means, that the pre-cooler outer diameter becomes more influential. The pre-cooler outer diameter is essentially constant during this study, because the coolant mass flow has been kept constant. Consequently, the pressure losses in the outer bypass flow increase if the HPC PR is decreased, because the outer bypass flow is accelerated more. Nonetheless, the propulsive efficiency of the engine increases if the HPC pressure ratio and hence the OPR is reduced, despite the increase in specific thrust.

Overall, the SFC only changes by about 0.2 percentage points in the investigated range. The SFC is therefore not the main driver behind the fuel burn improvement. The variation in total engine weight has a considerably larger influence. The increase in specific thrust while keeping the BPR constant results in a smaller core mass flow and hence smaller and lighter core turbomachinery components. The reduction in HPC PR further reduces the component weight, as fewer stages suffice. The lower core mass flow furthermore reduces the inlet flow area to the MHX and hence its weight, albeit that the higher inlet temperature offsets a considerable part of that. Finally, the total engine weight is reduced because the fan diameter reduces with HPC pressure ratio in the investigated range, leading to a lighter fan and nacelle and less drag.

Overall, the total engine weight varies by about 130 kg in the investigated range, which has a significant effect on mission fuel burn. The optimal HPC pressure ratio for an engine with a TET_2 of 1530 K, is likely lower than 17 and therefore outside the limits of this study. However, even if the HPC pressure ratio were further reduced, the fuel burn benefit of an engine with $TET_2 = 1530\text{ K}$ is not going to match the fuel burn benefit of an engine with $TET_2 = 1580\text{ K}$. The optimal CCTF2050_ITR is still about 0.8 percentage points more fuel efficient than the most fuel efficient case shown in Figure 4-35.

In conclusion of the past two studies, if a reheat combustor is added to a combined cycle aero engine, the combustor should be sit as early in the expansion process as possible and it the outlet temperature should be as high as possible.

5 Discussion

The combined cycle engine promises only a moderate fuel burn benefit over a projected year 2050 reference engine, even if the individual components can meet the specifications of the models used in this work. The current models further suggest that the concept will not significantly benefit from the addition of inter-turbine reheat. This chapter interprets the data shown in chapter 4 and what they mean with respect to engine performance, component design and overall concept feasibility.

First, the performance of the ‘optimal’ cycles is discussed individually in section 5.1. To that end, the respective shortcomings of the models and the possible implications of design choices that were not investigated in this study are discussed. Special emphasis is given to the possible influence of the off-design performance control and simulation of the bottoming cycle. The performance of the bottoming cycle might be regulated using a few different methods and the theoretical implications of each of them when applied to an aero engine are discussed. The discussions culminate in an approximation of the uncertainty for the predicted fuel burn estimates of each configuration.

Once the individual uncertainties are established, the absolute performance predictions of chapter 4 are used in section 5.2 to argue how they compare against different targets. First, the data of the three cycles are compared to each other in section 5.2.1. Section 5.2.2 evaluates the ability of each configuration to help realise the fuel-burn reduction targets of Flightpath 2050. Finally, the three configurations are compared to alternative novel engine concepts in section 5.2.3.

Section 5.3 contains a brief qualitative discussion of the noise and gaseous emissions of the combined cycle engine. The section furthermore includes an approximation of the economic viability of the combined cycle engine concept.

5.1 Cycle performance and confidence in the results

One of the main contributions to knowledge of this work is the level of detail that has been incorporated in the S-CO₂ component models and their application to an aero engine. The results presented in chapter 4 have illustrated how influential the different design parameters of these models are, but there is still further improvement potential for the presented methods and tools. Before the three cycles are discussed individually, a number of phenomena concerning multiple engine configurations are presented.

5.1.1 Sources of uncertainty common to multiple configurations

Weight Estimation

The WeiCo weight estimation tool was used to predict the total engine weight of the main engine. WeiCo may be seen as a 'black-box' application for which it is not possible to investigate the root-causes of possible convergence problems or irregularities in the results for anyone outside of Chalmers University. As a result, the limitations of WeiCo's LPT weight prediction method for the CCTF2050 and the CCTF2050_ITR meant that the LPT weight had to be scaled simplistically from the weight of the LPT of the TF2050 using the core mass flow. This is unlikely to have introduced a substantial error, but it cannot be dismissed and other, possibly open-source, solutions might have to be explored in the future.

Off-design modelling

The uncertainty introduced by the simplified off-design performance modelling approach is difficult to quantify. The comparison of the results attained from the detailed and the simplified methods for the TF2050_ITR showed that the fuel burn benefit may be estimated to within 0.2 % for that cycle. The variation in SFC, engine weight and fan diameter was greater however and given that the SFC is the most influential of the three, it is not guaranteed that the simplified method will predict the mission fuel burn of the combined cycle engine with the same degree of accuracy. Improving the stability of the combined cycle engine models at off-design should therefore be a prime objective of potential follow-on studies to increase the confidence in the obtained results.

Model validation

The evaluation of a completely new or very rarely researched concept such as a combined cycle aero engine with a supercritical CO₂ bottoming cycle requires the creation of a large number of new component models built on an even larger number of assumptions. Unfortunately, it is almost impossible to validate those models, because little to no relevant data is available in the public domain. It was ascertained that as many validated correlations were incorporated as possible during the creation of the present models. Additionally, advice from industrial experts within ULTIMATE and beyond was sought as much as possible to increase the reliability of the models and their results. The effects of the most influential modelling assumptions were presented where appropriate in this work and the assumed values were frequently based on public domain data. Despite these efforts, the final margin of error determined for each engine in sections

5.1.2 through 5.1.4, cannot be definitively corroborated until component and/or engine tests are conducted. The ongoing developments and operation of the S-CO₂ test rig at Cranfield University could significantly contribute to these efforts.

Balancing bypass losses and engine drag

The non-linear aircraft trade factors (NLTFs) represent a range of configurations, where the engine diameter, weight and efficiency is changed and the aircraft is subsequently redesigned. The NLTFs include ground clearance effects for the engine, wing box and landing gear changes, estimates for the nacelle drag and more. While this makes the NLTFs a simple to use and universally powerful tool, the incorporation of the nacelle drag limits the engine design choices somewhat. Introducing an intercooler or a pre-cooler to part of the bypass duct will also affect the bypass flow that does not pass through the heat exchanger. Ideally, it would be possible to investigate the fuel burn implications of either accelerating the bypass flow and accepting higher pressure losses, or of increasing the nacelle diameter and accepting a greater nacelle drag. Such a trade-off was not possible in the present work because the NLTFs are essentially a surrogate model for a range of aircraft designs rather than a dedicated point mass based aircraft model that allows for changes to the aircraft aerodynamics and structure. Employing a more detailed and flexible aircraft model, would increase the credibility of the overall performance assessments in the future. It is likely, that the fuel burn benefit would rise, if the nacelle diameter were increased a little, but the effect is likely small.

More vigorous integration of internal design of heat exchanger tubes

The results of Rengel's detailed study of feasible internal tube structural designs for the elliptical tubes of the involute spiral heat exchanger were used in a reduced form in this work. Only the lightest design that abided by a factor of safety of two for a given internal pressure and chord length was applied during most of the parametric studies. The average wall temperature in the tube walls was estimated using a simplified approach and used to determine the material's yield strength at a given temperature. However, the temperature estimation method was so simplistic, that it did not warrant repeated iterations of the calculations for a given configuration, because the variation in wall temperature would have been minimal. This may mean, that some of the designs investigated in this work did not uphold a factor of safety of two completely. The FOS was chosen high enough for such a variation not to matter, but a more integrated application of Rengel's designs into the design process, which predicts the wall temperature more accurately, may increase confidence in the results nonetheless.

Neglected component performance variations

The component and engine models were created with as much detail as possible for the intended studies, but a number of design aspects are still not accounted for. The optimum LPT stage count for example may change if the bottoming cycle is added to the geared turbofan, because the necessary power output is reduced. This would reduce the engine's weight and length. However, this is more likely to apply to the CCTF2050_ITR than to the CCTF2050. Additionally, the LPT stage count and the gear ratio of the reduction gearbox were kept constant irrespective of the bypass ratio. If preliminary component design methods for the core turbomachinery were integrated into the models, which balance mechanical and aerodynamic performance, these and further effects could be accounted for. Unfortunately, it is almost impossible to quantify the effect this would have on the fuel burn estimates.

5.1.2 TF2050_ITR

The geared turbofan engine model with inter-turbine reheat between the two HP turbine stages does not offer a fuel burn benefit over the year 2050 reference engine. On the contrary, the fuel burned on a 7000 nm intercontinental mission is actually increased by about 1 %. The limitations on core turbomachinery size and the temperature limits for the combustor and compressor delivery temperatures mean that the core specific power cannot be raised enough to offset the additional pressure loss in the reheat combustor or the additional cooling flows to HPT2 if the reheat combustor is lit. In fact, the reheat combustor remains essentially unlit in the optimal TF2050_ITR. The studies furthermore revealed that the reheat combustor should be positioned as early in the expansion process as possible. Previous studies reported marginal fuel burn improvements to be possible [38], [39], but those studies set out from a less mature reference engine and did not account for core turbomachinery scaling effects.

Suggested model and concept improvements

The current models do not include a very detailed secondary air system model, nor do they automatically assess the effect of the second combustor on the flow field and hence the efficiency of the second stage of the HPT. Incorporating either of these models will likely worsen the mission fuel burn estimates further.

Inter-turbine reheat is only going to help improve the mission fuel burn, if coupled with another concept. Intercooling circumvents the temperature limitation of the compressor delivery air, allowing for higher OPRs and hence a reduction of the LPT EGT. Combining

the potential of intercooling to increase the core specific power with a similar potential from ITR, might result in a significantly more efficient engine configuration, though turbomachinery scaling would become a major design problem. Moving to an axial-centrifugal compressor design or a reverse flow core concept might help alleviate these problems. Alternatively, an engine that can harness the higher heat energy in the LPT exhaust gases such as a bottoming cycle or recuperation may find ITR to be beneficial. The combination of ITR with a bottoming cycle has been investigated in this work, and ULTIMATE project partners AUTH currently investigate the combination of ITR and intercooled-recuperation [37].

The different off-design performance methods

The investigation of the fuel burn reduction potential of the TF2050_ITR was not the main objective of this work. Primarily, the configuration was assessed, to enable a better interpretation of the performance of the combined cycle engine with inter-turbine reheat CCTF2050_ITR. Additionally, the less complex TF2050_ITR model was used to investigate how the simplified off-design performance estimation method (see section 3.7.3), which had to be applied to the combined cycle engines, might affect the fuel burn estimates.

The net thrust rating of the detailed_TF2050_ITR at max cruise conditions is noticeably higher than the net thrust rating of the simple_TF2050_ITR. As a result, the engine weight may be slightly higher than necessary. This is likely due to the fact that the optimiser was only able to evaluate 630 individuals during a 30 hour optimisation, because of the large number of iterations necessary to match the thrust requirements of the non-linear aircraft trade factors at three operating points and the blade temperature limits for the HPT stages. Longer optimisation runs may be able to identify a smaller and lighter engine configuration, but the difference is unlikely change the fuel burn estimates enough to make the concept feasible.

The simplified off-design performance method appears to underestimate the mission average SFC of an engine, but it may overestimate the engine's weight and fan diameter. The deviation in fuel burn penalty predicted by the two methods is relatively small (+0.2 percentage points) and it is assumed that a similar margin of error can be maintained for the combined cycle engines, despite their increased complexity.

Confidence in the results

The fuel burn penalty predicted by the simplified off-design method will be used to evaluate the performance improvement of the CCTF2050_ITR relative to the CCTF2050 and to assess the synergy potential of the two technologies. The detailed_TF2050_ITR may be more realistic, but the same method was not applicable to the combined cycle engines, which makes a like-for-like comparison impossible.

The deviation between the predicted fuel burn penalties of the two methods is used to estimate the uncertainty of the results. This deviation appears to be the only major contributor to the model's uncertainty quantified so far. The uncertainty stemming from WeiCo's LPT weight estimates does not apply in this case, as WeiCo did not exhibit such behaviour for this engine configuration. The effects of more eloquent turbomachinery and secondary air system models are not easily quantifiable.

Hence, the TF2050_ITR suffers a $0.95 \% \pm 0.2 \%$ fuel burn penalty relative to the TF2050.

5.1.3 CCTF2050

The model of a geared turbofan engine with a supercritical CO₂ bottoming cycle promises to lower the mission fuel burn of a year 2050 aircraft on an intercontinental mission by about 3.17 %. The improvements are solely due to the novel core components, as the reference engine and aircraft have been projected to what is assumed to be the evolutionary limit of the conventional design approach. The fuel burn benefit is primarily limited by the size and design of the pre-cooler. The pre-cooler effectively dictates the useful power output of the bottoming cycle, because it determines the S-CO₂ mass flow for a given coolant mass flow and S-CO₂ compressor inlet conditions. The pre-cooler also affects not only the part of the bypass flow used as a coolant, but the outer bypass flow as well, because its outer radius may constrict the flow area of that outer bypass. The combination of all these effects has been discussed in great length in section 4.2.2.2. The component efficiencies of the S-CO₂ turbomachinery are similarly important to the feasibility of the concept overall. The following sections argue the remaining sources of uncertainty due to the novel component models and the related assumptions.

Operation of the combined cycle engine at off-design

The current investigation is primarily focused on establishing the design point performance of a combined cycle engine with an S-CO₂ bottoming cycle and estimating the mission fuel burn benefit through a simplistic off-design method. Fortunately, most of

the off-design performance parameters are secondary, because the concept is applied to a long-range intercontinental aircraft that spends over 90 % of its time at cruise conditions. Some performance parameters had to be scaled from the design point data, in order to use WeiCo as the weight estimation tool of the main engine and so the non-linear aircraft trade factors could be applied. However, the actual off-design operation of the engine is yet to be investigated and may affect the mission fuel burn in a number of ways. Depending on how the main engine and especially the bottoming cycle are operated, some components may require design changes to mitigate higher stress levels or to avoid surge or overspeed events.

The LPT EGT will rise significantly if the engine operates at higher power settings than at cruise, for example during climb and take-off. As a result, the heat transfer in the MHX might increase. Additionally, the temperature and pressure of the pre-cooler coolant mass flow are greater at lower altitudes and higher power settings, which means the design inlet conditions of the S-CO₂ compressor cannot be reached anymore. The loss in useful power due to the departure of the compressor inlet conditions from the critical point will be noticeable but not overwhelming at off-design, because even the design point conditions of the optimum CCTF2050 are not very close to the critical point. The size and weight constraints of the pre-cooler necessitated a compromise to be reached with regard to the compressor inlet conditions (see section 4.2.2.2). The increase in power consumed by the S-CO₂ compressor for a given pressure ratio at off-design is therefore not as substantial as it would be if the design point were closer to the critical point. The higher LPT EGT and subsequently higher peak temperature in the bottoming cycle however, will improve the power output for a given S-CO₂ mass flow. How these effects are balanced and what that balance means for the efficiency of the combined cycle engine as a whole is yet to be determined. A number of components and parameters could potentially be used to influence the bottoming cycle's performance directly and indirectly. For the purposes of this investigation it was assumed that the increased temperature difference between LPT EGT and bypass air at Top-of-Climb and End-of-Runway would be enough to offset the loss in bottoming cycle efficiency due to a higher fan outlet and hence S-CO₂ compressor inlet temperature. The SFC benefit due to the combined cycle arrangement is therefore assumed to be constant throughout the flight envelope (see section 3.7.3).

The LPT EGT can be manipulated through the introduction of variable stator vanes (VSV) in the LPT and/or a variable area nozzle for the core engine. A variable area nozzle in

the pre-cooler bypass duct and/or a variable pitch fan could be used to control the cooling flow conditions. These mechanisms can be seen as indirect control mechanisms for the bottoming cycle, because they alter the operating point of the main engine and therefore only affect the bottoming cycle indirectly. VSVs and VANs only enable small variations of the LPT EGT however and are therefore not likely to suffice as the only control mechanisms for the bottoming cycle.

Different operating schemes have been discussed for stationary closed S-CO₂ power cycles, for example by Dostál [55] or Brun et al. [65]. The larger range of operating conditions the S-CO₂ cycle will encounter during a standard mission means that their application might be more difficult on an aero engine. These mechanisms are seen as direct control mechanisms for the purposes of this discussion. Which of the mechanisms is most feasible for combined cycle engines is yet to be determined, as none of them are implemented in the current models. Dostál distinguishes between three basic control concepts: inventory control (pressure control), bypass control and temperature control.

Inventory Control

Inventory control entails the introduction of a storage tank (or control vessel) that extracts S-CO₂ from the cycle at the outlet of the compressor. The power output of a closed cycle is directly dependent on the mass flow through the turbines, although the working pressures and temperatures obviously also have a significant effect in a real gas cycle as discussed at the start of section 4.2. If higher power settings and larger S-CO₂ mass flows are required, the S-CO₂ can either be reintroduced to the cycle at the exit of the compressor or in front of the pre-cooler. The first option means that the pressure ratio of the cycle will be lower than usual when operating at low power conditions. As a result, the ramp up of the power output can be slower. With the second approach however, the pressure ratio is kept high at low power conditions, which enables a faster load increase for the bottoming cycle. The second approach does entail more difficult piping and control systems and an additional small compressor to raise the excess mass flow to the necessary pressures. The first and simpler approach is therefore preferred for most stationary applications, despite the limited response times, but on an aero engine the second approach may become more attractive. A general drawback of inventory control is the requirement for a possibly large and heavy storage tank. The problem would be aggravated in an aero engine due to the large range of operating conditions that need to be managed. However, the high pressure levels and relatively low design point mass flow of the S-CO₂ in the CCTF2050 may help keep the weight of the tank down.

Bypass Control

Bypass control describes a very common system where part of the high-pressure recuperator exit flow, in this case the compressor exit flow, is diverted to the pre-cooler inlet flow without a storage tank in the middle. This leads to a mass flow and pressure ratio reduction in the turbines. The power output of the cycle as a whole is reduced as well. Hence, instead of altering the mass flow in the whole system, as in the inventory control case, an imbalance between the individual components of the bottoming cycle is created.

The efficiency of the turbines would also change, because the velocity triangles would change if the mass flow is reduced. How and by how much the efficiency changes depends on the design point of the turbines and the relative change in operating conditions. For example, the efficiency may be reduced at climb if the turbines are designed for a reduced mass flow for optimal cruise performance. If the turbines are designed to deliver peak efficiency at maximum power conditions however, the cruise efficiency would be lower than the climb efficiency. The former is probably advisable for the current application, which signifies a departure from stationary applications. The system would be controlled via bypass valves that regulate the mass flow as appropriate. Bypass control systems are able to facilitate rapid power changes, which is why they are very common in closed power cycles. In small power cycles, bypass control is furthermore used as an emergency response tool for very low power operation, because 10% step changes in power output can be achieved.

Temperature Control

Temperature control in a closed stationary power cycle usually entails powering down the reactor to reduce the heat input. This enables load rate changes of about 5 % per minute, which suffices for most power plants, but would be inadequate for an aero engine. Additionally, it is virtually impossible to vary the core engine's EGT without changing the operating point of the main engine, unless an afterburner would be installed between the LPT and the MHX. In fact, the bottoming cycle's control system has to be designed to react to the temperature variations of the LPT exhaust rather than influencing them directly.

Final discussion

Before designing and tuning a complex control system for the bottoming cycle, off-design simulations need to prove that the direct manipulation of the off-design performance through all kinds of devices actually brings a performance benefit over a self-regulated cycle. It might be that the performance penalty, which arises when the bottoming cycle inadvertently converges to an off-design point with higher compressor inlet conditions, is relatively small and isn't worth adding further complexity to the engine over. The power transmission system between the FPT of the bottoming cycle and the LPT of the main engine may not be able to harmonize the different transient behaviours of the two components. In that case it might be worth considering to disengage the bottoming cycle for transient or even for most of the climb phase completely, so the cycle can self-regulate. This would require the main engine to be slightly oversized for cruise operation in order to still meet the thrust requirements of the engine at climb and take-off.

If the necessary future investigations on the subject show that a control system is beneficial, a combination of a small storage tank for inventory control coupled with bypass valves for bypass control seems to be the most likely arrangement for the bottoming cycle. Variable stator vanes in the LPT and a variable area nozzle for the core engine might be used to manipulate the LPT EGT, though both will also have a significant effect on the performance of the rest of the engine. A variable area nozzle may be necessary for the pre-cooler bypass duct, in order to manage the pre-diffusion levels of the inlet diffuser at off-design. The air mass flow through the pre-cooler does not necessarily need to be controlled this way, as the bottoming cycle is primarily introduced to improve the engine's cruise performance, whereas an intercooler's biggest benefit might stem from its operation at climb and take-off. When modelling the off-design behaviour of the CCTF2050, special attention should be given to accurately portraying the changes in pressure losses in the bypass and pre-cooler ducts, because the bypass section provides over 90 % of the net thrust of the engine.

The sheer range of possible control mechanisms and the different effects each of them would have on the performance of the engine, make it impossible to say if the simplified off-design estimation method that was used in this work has determined realistic parameters or not. A separate detailed research project would have to assess the optimal design of the engine control system. It is therefore also not possible to quantify the uncertainty that is introduced to the fuel burn estimates because of the simplified off-design method. As a simplistic estimate, the ± 0.2 % uncertainty determined for the

TF2050_ITR is assumed to also be applicable to the CCTF2050, though that may be optimistic.

S-CO₂ Turbomachinery

The performance models of the S-CO₂ turbomachinery components are relatively simple, using prescribed rotational speeds, pressure ratios and isentropic or polytropic efficiencies at design point at scaled component maps at off-design. As a result, the assumed values may differ considerably from what might be technically achievable. Furthermore, the engine is proposed to enter into service by 2050, meaning the respective technology can be developed further in the coming 30 years.

It is virtually impossible to ascertain what kind of advancements can be made in that time, because S-CO₂ turbomachinery development is still in its early development stages in spite of some of the sophisticated design tools that are already commercially available (see NREC's AXIAL [116] or Softinway's AxStream [147] software suites). Sensitivity studies have been performed as part of this work to establish the dependency of the claimed fuel burn benefits on the assumed component efficiencies. Assuming all component models are otherwise realistic, the isentropic efficiencies of the S-CO₂ turbines and compressor would have to be greater than 72 %, for the concept to warrant any fuel burn benefit at all. 72 % isentropic efficiency has been shown to be attainable at an even smaller scale already [114], [138], but the strong linear relationship between component efficiencies and fuel burn is still a critical concern for the future development of the concept. Of course, any improvement beyond the assumed levels of 89 % isentropic efficiency for the compressor and 90 % for the turbines would increase the fuel burn benefit of the concept, but the assumed values are rather ambitious already. In the end, a detailed investigation of the compression system of the bottoming cycle might reveal that it would be better to employ positive displacement machines than turbomachinery components at such a small scale.

The current models also scale a digitised version of the maps published by Sandia [138] according to the design specifications, to enable off-design performance simulations. However, predicting the off-design performance of real gas turbomachinery components can be difficult, as the discrepancies between the predicted and observed values of BMPC's test rig clearly show [70]. Hence, considerable time and resources still have to be spent on developing adequately precise tools that are simultaneously fast enough to be integrated into a full-fledged concept optimisation setup. This is particularly important for aero applications, because the range of commonly encountered operating conditions

is quite large and in an extreme case a large variation of the overall pressure levels in the system could endanger the safety of the aircraft if left unchecked. A less dramatic result of faulty off-design performance predictions would be that the bottoming cycle does not deliver enough shaft power to the main engine, which would require the engine to run hotter and burn more fuel. Increased operating temperatures in the core engine would significantly reduce the component life however, which would diminish or possibly even dismiss the economic benefit of the combined cycle benefit.

Further uncertainty around the fuel burn estimates stems from the way the weight of the turbomachinery components is estimated. Even though these estimates are based on approximate parameters of existing S-CO₂ components [141], the final designs may well be heavier than assumed. Especially the casings of the components will have to have thick walls and may therefore be heavier than anticipated. If the turbomachinery components would weigh 130 kg instead of the assumed 30 kg, the fuel burn benefit would be reduced by about 0.2 percentage points. Consequently, the isentropic efficiency of the turbomachinery components would have to be at least 75 % instead of the previously argued 72 % in order for the concept to offer a benefit at all. Given that the diameter of the S-CO₂ compressor wheel is currently estimated to about 4.2 cm however, such a dramatic weight increase seems unlikely.

Weight of the S-CO₂ components

The weight estimation of the S-CO₂ turbomachinery components is not the only source of uncertainty regarding the weight however. The weight of the S-CO₂ heat exchangers is estimated to a reasonable degree of accuracy, given the detail of the models. The weights of the remaining parts of the S-CO₂ bottoming cycle on the other hand are largely based on a very limited amount of data, best engineering judgment and personal communication with industry experts. Unfortunately, it is difficult to quantify the uncertainty in terms of mission fuel burn, as the discrepancy will be strongly affected by the amount of redundancies and bypass ducts that might be necessary for off-design operation or safety purposes. One contribution that has not been taken into account sufficiently is the weight of the working fluid itself, as it is difficult to determine without designing fluid tanks, all the major ducts and pipes and optimise the flow velocities in all these components. Again, the aircraft trade factors would be used to address any neglected weights, raising the mission fuel burn by about 0.2 % for every 100 kg of additional engine weight.

Mechanical integrity of involute elliptical tubes in heat exchangers

The high internal pressures in the MHX and the pre-cooler tubes necessitated a dedicated investigation of feasible internal tube geometries. To contribute to this work, Rengel conducted a parametric study to that end, in which he performed over 1300 FEM simulations with variable internal design features (see section 2.5.2.4). The resulting data was used in the present investigation to inform the performance and weight estimation calculations, but some uncertainty still remains.

The factor of safety that was applied to the predicted peak stress of a given tube design was generously set to two, which is more than many aero engine design features usually allow for. However, the peak stress level in the cycle is estimated using the pressures inside the tubes at cruise conditions. Depending on the way the bottoming cycle is operated at off-design, the pressure levels and hence the stresses in the tube walls may rise significantly during climb or take-off. The exhaust gas temperatures will also increase at those higher power conditions, raising the average wall temperatures in the tubes and lowering the yield strength of the material. The combination of possibly higher internal pressures and higher wall temperatures at climb or take-off conditions means that the factor of safety will probably be significantly smaller at off-design. Ideally the internal tube geometry would have been chosen based on End-of-Runway conditions, but that was not possible in this case. The simple off-design method employed in this investigation was primarily used to estimate the weight of the main engine with WeiCo and to scale the SFC of the engine at off-design. Beyond that, temperatures and pressures at specific stations in the engine were not needed and couldn't have been realistically assessed with the method anyway. Using such estimates to determine the optimal internal design of the heat exchanger tubes would have introduced an intangibly large uncertainty to the results.

Whether or not the solidity of the tubes would have to be increased in order to withstand the off-design parameters of the engine will depend on how large the variation is and by how much the factor of safety could be reasonably reduced for a production engine. Additionally, the internal design of the tubes and their flow channels does not only affect the weight of the heat exchanger, but the heat transfer and pressure loss as well, as discussed in section 4.2.2.1. Again, these parameters also depend on the undetermined operating scheme of the bottoming cycle. Hence, it is next to impossible to reliably quantify the resulting uncertainty in the fuel burn estimate. However, the step changes in engine weight and fuel burn observed in section 4.2.2.8 when altering the chord length

of the tube give an indication and might estimate the uncertainty to about 0.4 % fuel burn. The penalty could be traded against a reduced compressor pressure ratio in the S-CO₂ cycle, which will lower the power output but may allow for lighter tube designs to be used. Finally, it should be noted that Rengel only accounted for linear elastic deformations of the tubes during his FEM simulations. Plastic deformation of the tubes would lead to a redistribution of the stresses in the tube walls and might reduce the peak stress levels overall. Plastic deformation could therefore improve the mechanical integrity of the tubes, but it would probably also affect the aerodynamics of the external flow in a presently unquantifiable way.

Aerodynamics of involute elliptical tubes in heat exchangers

Rengel simulated the tubes of the MHX not as perfect ellipses, but as double-circular arc designs. Aerodynamically, an elliptical cross-section is likely to produce less drag, which is why Zhao et al. investigated that shape in the first place [34], [122]. In order to model the heat exchanger tubes consistently across all disciplines and reduce the uncertainty in the obtained results, it is therefore necessary to conduct similar CFD simulations as Zhao et al. did for the involute elliptical tubes and derive new non-dimensional friction and Colburn factor correlations for the double-circular arc tubes. D'Antino has taken on that task and is currently conducting the simulations as part of his MSc research project. His investigations are expected to conclude in August 2018.

Overall, changing the design from an elliptical outer tube cross-section to a double-circular arc design is unlikely to have a very significant effect on the current fuel burn estimates, as the aspect ratio of the tubes was retained and the leading edges of the tubes can still be elliptical.

A further objective of D'Antino's efforts is to derive additional correlations that would allow for the inclusion of even more design options for the heat exchangers, such as different tube spacings, different aspect ratios and possibly even finned tubes. Rengel already included different aspect ratios of the tubes in his simulations, so the same fidelity of the heat exchanger models could be assumed once D'Antino has concluded his work.

Diffuser design boundaries

Section 4.2.2.10 showed how sensitive the fuel burn estimates can be, if the diffuser of the MHX and pre-cooler had to be shorter than assumed. The exact length available for either diffuser is difficult to ascertain at the present stage of the work. The influence of a longer MHX diffuser, nacelle and nozzle cone on the afterbody drag and the centre of

gravity of the engine would have to be assessed. It would also have to be ascertained that the aircraft can still roll at the end of runway with reasonable ground clearance.

The radial dimensions of auxiliary systems such as the active clearance control mechanism of the HPT would have to be finalised in order to more definitively determine the length available for the pre-cooler duct. Neither of these features are routinely included in preliminary performance studies such as this.

If the length of either diffuser had to be reduced, the increase in pressure losses across the diffuser could be mitigated by increasing the inlet Mach number to the heat exchanger downstream. However, while the reduced area ratio would lead to reduced diffuser losses, the pressure loss across the subsequent heat exchanger will likely increase due to the higher flow velocities. The ideal trade-off of these effects would have to be determined again. Based on the figures presented in section 4.2.2.10, which assume the same heat exchanger inlet temperature for all cases, the uncertainty in terms of mission fuel burn may be about $\pm 0.2\%$.

The pressure losses occurring in the diffusers were furthermore calculated using a rather simple correlation. Detailed design investigations of the two diffusers would need to be conducted in cooperation with updated engine models in order to realistically assess the whole engine performance implications of pre-diffusion and bleeding off part of the air or exhaust gas flows. The correlation presented in section 3.4.1.2 is calibrated to match the performance estimates of A'Barrow [105] for a bled diffuser, which ought to reduce the uncertainty of using this simplified method. Unfortunately, quantifying the remaining uncertainty in terms of mission fuel burn due to the used correlation is not possible at this point.

LPT component efficiency

Finally, the bottoming cycle reduces the amount of power the LPT has to produce to drive the fan and the IPC. This can either be used to lower the stage count of the LPT or to lower the stage loading and increase the efficiency of the LPT. Neither of the two effects are quantified in the present models, as it is unlikely that the power reduction is large enough to drop a whole stage and the efficiency of the LPT is already assumed to be very high. Either design change would improve the fuel burn benefit of the engine, but the effect cannot be quantified with reasonable confidence to be taken into account here. The weight change of the LPT due to a variation in core mass flow is of course accounted for as discussed in section 3.7.4.

Confidence in the results

A large number of possible sources of uncertainty have been discussed in this section and their effects have been tentatively quantified where possible. Overall, the mission fuel burn benefit of the 'optimum' CCTF2050 is estimated to about 3.17 % with a cumulative uncertainty between 0.8 and 4.0 %, depending on what turbomachinery efficiencies are assumed to be realistic by 2050. Theoretically, at least some of these uncertainties partially apply in both directions, but it is unlikely that the possible fuel burn benefit is underestimated that much as many assumptions are optimistic already. It is however possible that the conducted parametric studies have not identified the global design optimum as an optimisation algorithm would have.

Hence, the CCTF2050 is estimated to offer a fuel burn benefit relative to the TF2050 of about 3.17 % with an approximate uncertainty of +0.3 / -4.0 %. This is obviously a rather large range and could potentially mean that the concept does not actually offer a fuel burn benefit at all. However, 30 years of further research and development are likely to enable at least a small fuel burn benefit even if it is not the estimated 3.17 %.

5.1.4 CCTF2050_ITR

The model of a geared turbofan engine with a supercritical CO₂ bottoming cycle and inter-turbine reheat promises to lower the mission fuel burn of a year 2050 aircraft on an intercontinental mission by about 3.25 %. Adding a reheat combustor to the CCTF2050 unfortunately did not result in a significant fuel burn improvement. Scaling effects of the core turbomachinery and the pressure losses in the reheat combustor mitigated much of the potential benefit that might have been expected. The CCTF2050_ITR incorporates the same component models as the CCTF2050, with the addition of a second combustor component between HPT1 and HPT2. As a result, the same uncertainties apply to the CCTF2050_ITR as they did to the CCTF2050 and this section only covers additional phenomena that may affect the fuel burn estimates.

Reheat combustor modelling

The reheat combustor is modelled by prescribing a combustion efficiency, a total pressure loss and the outlet temperature. The same simplified model is used for the main combustor, but the design and the characteristics of a reheat combustor are much less established in the public domain than those of the main combustor. The total pressure loss across neither of the two combustors is increased through the presence of film cooling in the NGVs of the downstream HPT stage. The reheat combustor will

furthermore not be required to raise the gas temperature as much as the main combustor, which may help to reduce its size. However, the flow entering the reheat combustor is at a lower pressure and temperature than the flow entering the main combustor, which will increase the size of the reheat combustor. Additionally, the flow entering the second combustor contains a number of combustion products from the main combustor, while the main combustor introduces the fuel to a clean air. Consequently, the combustion process in the second combustor is more difficult to predict and the emission-optimal equivalence ratio is affected by the existing combustion products. Martinez-Bueno [43] investigated possible design solutions for a reheat combustor for a similar engine and determined that the pressure loss across an 'oxy-poor flame combustor' may be less than 2 %. Hence, 2 % has been used as a constant design input in the present simulations. However, some errors and necessary further work were established in Martinez's work and the pressure loss might be closer to 3 %. In that case, the fuel burn benefit of the CCTF2050_ITR would be reduced by just over 0.3 percentage points, which is why this uncertainty has to be assumed for the fuel burn estimate of the CCTF2050_ITR.

Further uncertainty arises from neglecting the additional engine length and weight due to the reheat combustor. Both shafts need to be longer because the reheat combustor is introduced between HPT stages. The disks would furthermore have to be redesigned, because the average flow and hence metal temperature in HPT2 are significantly higher than before. The casings around the engine core would furthermore have to be elongated. In addition, extending the shafts has implications beyond the weight increase. The bearings at either end of the shaft would either have to sit further apart or one of them would work in a higher temperature environment. If the bearings sat further apart, the rotor dynamics get more problematic, which may reduce the life of the bearings or, in a worst case, endanger the safety of the aircraft. Keeping the distance between the bearings and operating one of them in a higher temperature environment is also not a good idea, because the cooling requirements and the component life would be adversely affected. Unfortunately, none of these concerns are addressed in the present models. The effect of underestimating the engine weight is remedied easily through the aircraft trade factors (about 0.2 % fuel burn per 100 kg of additional engine weight), but the rotor dynamics discussions cannot be reasonably quantified here.

HPT2 cooling flow estimation

The TF2050 originally did not use separate turbine components to model the two-stage of the HPT. Instead it was modelled as an equivalent single-stage (EQSS) turbine, whose cooling flow requirements were established based on the calculated blade metal temperature of that EQSS blade. The introduction of the reheat combustor in the CCTF2050_ITR meant that the HPT had to be modelled with separate turbine components, which necessitated a number of changes regarding the modelling assumptions as presented in section 3.6. Among them, the blade metal temperature of HPT2 was frozen to 1185 K at max cruise. The CCTF2050_ITR model calculates the necessary cooling flow to enable that temperature at cruise. The percentage of compressor delivery air diverted to the HPT2 for cooling is subsequently kept constant throughout the flight envelope.

This approach introduces a greater uncertainty when applied to the reheat combustor than it does for the main combustor. How much cooling is necessary at each off-design point will vary depending on if and how the reheat combustor is operated at each point. Unfortunately, it is impossible to determine if the HPT2 cooling flow is sufficient to keep the metal temperature of HPT2 in the feasible range, unless the optimum fuel flow in the reheat combustor is established through full mission optimisation studies.

Unfortunately, it is also not realistically possible to estimate the uncertainty introduced by the simplified cooling model in terms of mission fuel burn. If it is furthermore assumed that the same percentage of compressor delivery air is used to cool HPT2 at all operating points, which does not reflect realistic engine operation. However, a simple trade-off can be established. If the blade metal temperature of HPT2 has to be lowered by 10 K at cruise to respect the metal temperature limits at off-design, the mission fuel burn benefit of the CCTF2050_ITR will be reduced by approximately 0.17 percentage points.

LPT component weight

The additional heat input in the reheat combustor significantly raises the HPT2 outlet temperature. The LPT is uncooled in the TF2050 and the blades are assumed to be manufactured from Ti-6Al-4V. The same assumptions have been kept for the CCTF2050_ITR, but it is unlikely that uncooled titanium alloy blades would offer a reasonable component life given the elevated temperatures. Hence, the first one or two stages of the LPT would have to be manufactured from nickel-alloys, which would almost

double their weight, and/or the blades would require additional cooling. Either solution will reduce the fuel burn benefit of the CCTF2050_ITR significantly.

Assuming the first two stages and their discs account for about 35 % of the total weight of the LPT, the shift from Ti-6Al-4V to a nickel alloy would increase the total engine weight by about 100 kg and hence reduce the fuel burn benefit by 0.2 percentage points. Quantifying the fuel burn increase if the LPT were to be cooled is beyond the scope of this work, as it would entail the identification of the ideal extraction point in the compression system and the definition of a maximum allowable blade metal temperature for the LPT. It is likely however, that it is more efficient to change the material of the first one or two stages of the LPT. Cooling the LPT blades may result in lower direct operating cost for the engine however, creating another opportunity for multi-disciplinary design optimisation.

Bypass loss model

The outer diameter of the pre-cooler can be so large that it leads to a contraction of the outer bypass flow area. The CCTF2050_ITR uses the same loss model to scale the outer bypass pressure losses as the CCTF2050. However, the accuracy of the method was not of great concern for the CCTF2050, because the parametric studies showed that the fuel burn benefit of the CCTF2050 would be greatest, if the outer bypass flow is virtually unaffected. The results of the CCTF2050_ITR on the other hand show that a certain increase in outer bypass pressure losses due to a larger pre-cooler coolant mass flow can be accepted, because the useful power output of the bottoming cycle increases strongly enough at the same time.

In this case, the gradients of the pressure loss calculation method (see Figure 3-17 in section 3.4.3) dictate the trade-off between useful power and outer bypass pressure losses. The implemented method has helped identify reasonable qualitative design trends for the CCTF2050 and the CCTF2050_ITR, but the accuracy of the results ultimately depends on the accuracy of the bypass pressure loss calculations. Unfortunately, the related uncertainty cannot be estimated via a simple variation of the outer bypass pressure losses, because a different scaling method would have led to different trade-offs and not a constant offset of the results. It is therefore not possible to quantify the effect of the present assumptions, but it is clear that a more detailed design study should be conducted. The results of that study could subsequently be used to validate the method used in this work or to formulate a new one.

Additionally, the initial design input for the inner diameter of the pre-cooler annulus was kept the same as for the CCTF2050. A smaller radius would probably have been feasible however, because the core mass flow and hence the radial dimensions of the turbomachinery and its auxiliary systems have been reduced. This is likely to slightly improve the mission fuel burn benefit of the CCTF2050_ITR configuration, but a reliable quantification of this uncertainty would have required a redesign of the main engine and was therefore outside the scope of this work.

Further off-design performance implications

The off-design performance of the CCTF2050_ITR is even less predictable than that of the CCTF2050, because the operation of the reheat combustor adds further complexity.

The bottoming cycle benefits from an increased TET_2 , because the temperature difference between the LPT exhaust gases and the bypass air increases. This temperature difference defines the maximum temperature range in the bottoming cycle and hence its efficiency. However, higher working temperatures through the MHX will also require stronger, more solid heat exchanger tubes, which will affect the pressure losses, heat transfer and especially the weight of the component significantly. Therefore, the operation of the combustors of the CCTF2050_ITR at off-design will probably be different to that of the TF2050_ITR and will definitely require multi-point and multi-disciplinary optimisation.

Furthermore, controlling the off-design performance of the bottoming cycle and the main engine is more difficult and therefore even more important in the CCTF2050_ITR. The storage tanks of the inventory control system have to be larger, because the temperature range and the design point mass flow of the bottoming cycle are larger. The bottoming cycle also contributes a larger share of the total LP shaft power, making the performance of the combined cycle more dependent on the optimal operation of the bottoming cycle.

Ultimately, the uncertainty that was found to arise from the simplified off-design method for the TF2050_ITR is doubled to 0.4 percentage points for the CCTF2050_ITR in order to reflect the increased complexity of the engine and its operation.

Identification of global optimum through parametric studies

The parametric studies performed during this investigation do not guarantee that the global fuel burn optimum has been identified. This is also true for the CCTF2050, but the uncertainty is larger for the more complex CCTF2050_ITR.

The relative placement and operation of the reheat combustor leads to greater variation in core specific power. Thus, the blade height of the last stage of the HPC varies more, leading to a greater range of polytropic efficiencies for HPC and HPT. Section 4.3.2.4 illustrated how the optimal OPR of the engine depends on TET_2 as an example of the additional trade-offs that influence the performance of the CCTF2050_ITR. It is not certain that all of these trade-offs were captured by the parametric studies, although at least the OPR- TET_2 dependency was investigated.

Confidence in the results

The number of assumptions that had to be made when modelling the CCTF2050_ITR is even greater than the number of assumptions made for the CCTF2050. The confidence in the predicted mission fuel burn figures is therefore even lower. Consequently, the approximate uncertainties mentioned in this section have to be added to those identified in section 5.1.3.

The larger number of design inputs also means that it is more likely that the parametric studies did not find the global optimum for the CCTF2050_ITR. This is why it is assumed that the fuel burn benefit of the CCTF2050_ITR may increase by up to 0.5 percentage points if the same models are handed to an optimiser such as Isight.

Thus, the CCTF2050_ITR is estimated to deliver a fuel burn reduction of about 3.25 % relative to the TF2050, with an approximate uncertainty of +0.5 / -4.8 %. Once again, the range of possible fuel burn benefits is larger than the actual benefit, meaning the CCTF2050_ITR may not represent an improvement at all. Further detailed design investigations would be necessary to increase the confidence in the obtained results, though whether or not it is advisable to spend further time and resources on the combined cycle concept using an S-CO₂ bottoming cycle is questionable at this point.

5.2 Cycle Comparison

The previous discussions explained that the uncertainty regarding the claimed fuel burn benefit differs for each of the three engine configurations. These uncertainties are largely neglected in this section, in order to simplify the comparison of the three engines against each other and other engines. Initially, the 'optimum' fuel burn and general performance figures of the three engines are compared against each other in section 5.2.1. Afterwards, the potential of the combined cycle concept to meet the Flightpath 2050 goals is discussed in section 5.2.2. Finally, section 5.2.3 compares the attainable fuel

burn benefits with those claimed for competing novel engine concepts to indicate whether it is advisable to spend further time and resources on the development of the concept.

5.2.1 Performance relative to each other

Table 5-1 shows the main performance figures, the engine weight and the fan diameter for each of the three novel engine cores and for the TF2050. All engine parameters are listed as per maximum cruise conditions, to allow for a like-for-like comparison.

The net thrust levels of all four engines are almost identical, which is not surprising given the relatively small variations in mission fuel burn. The percentage of thrust provided by the bypass flow however increases noticeably for the combined cycle engines from the already very high 91 % of the TF2050 to about 93 % and 96 % for the two combined cycle arrangements. By comparison, the bypass section of the LRTF2000 contributes only about 79 % of the net thrust of the engine and more contemporary large civil engines usually also feature shares between 80 and 85 %. The increased influence of the bypass also means, that the bypass pressure losses are increasingly important. This is one of the reasons why it is crucial to calculate the variation of the bypass pressure losses due to a flow path contraction around the pre-cooler or an intercooler.

The bypass ratio of the CCTF2050_ITR is significantly higher than that of the other three configurations, because of the active reheat combustor and the more powerful bottoming cycle. As a result the core mass flow is significantly reduced and the core specific power, which is calculated according to equation 5-1, is significantly increased. The specific power of the TF2050_ITR is actually lower than that of the TF2050, because the reheat combustor is unlit but still induces a 2 % pressure loss.

$$sP_{wr_{CO}} = \frac{\text{Sum of power produced by turbines (and bottoming cycle)}}{\text{core inlet mass flow}} \quad 5-1$$

Usually, the specific thrust of an engine (sF_N) reduces if the bypass ratio increases, but the opposite seems to be the case for the combined cycle engines. This is due to a combination of reasons. The main reason is that the jet velocity ratio between the core and the bypass is different to that of a modern turbofan. Part of the heat that is removed from the core exhaust is used to produce additional shaft power in the bottoming cycle, but the majority (~80 %) is transferred to part of the bypass flow. As a result, the outlet temperature of the coolant airflow is increased, increasing the jet velocity out of the pre-

cooler duct nozzle. The pressure losses across the diffusers and the heat exchanger matrix offset much of the thrust gain, but the jet velocity is still significantly higher.

Furthermore, the combination of the increased core specific power of the combined cycle engines and the fact that higher air-side inlet pressures improve heat transfer in the pre-cooler, lead to an increase in optimal fan pressure ratio. Increasing the fan pressure ratio naturally increases the specific thrust of the engine. This is especially true for the CCTF2050_ITR, where the core specific power and the FPR are both increased considerably.

Table 5-1: Maximum Cruise Performance Comparison of year 2050 engines

Parameter	Units	TF2050	TF2050_ITR	CCTF2050	CCTF2050_ITR
F_N	kN	37.3	37.4	37.1	37.0
Rel. $F_{N,BP}$	%	91.0	90.5	92.9	96.0
BPR	-	16.7	16.4	16.8	21.8
FPR	-	1.40	1.41	1.41	1.43
OPR	-	63.3	63.3	63.1	61.6
TET_1	K	1580	1580	1580	1580
TET_2	K	-	1347	-	1580
WS_{HPT}	-	-	0.45	-	0.45
BC Pwr	kW	-	-	634	1022
$\eta_{therm,BC}$	%	-	-	19.3	20.0
$sPwr_{core}$	MW/(kg/s)	1.12	1.11	1.13	1.29
η_{therm}	%	53.9	53.6	55.5	55.7
η_{prop}	%	81.9	81.7	81.7	81.2
sF_N	m/s	86.5	87.4	88.8	91.6
SFC_{DP}	g/(kN*s)	12.88	12.98	12.53	12.56
ΔSFC	%	0.0	+0.8	-2.7	-2.5
SFC_{ave}	g/(kN*s)	12.64	12.74	12.30	12.33
d_{Fan}	m	2.84	2.83	2.79	2.75
Weight	kg	5161	5125	5505	5306
$\Delta Weight$	%	0.0	-0.7	+6.7	+2.8
ΔFB_{y2050}	%	0.0	+0.95	-3.17	-3.25
ΔFB_{y2000}	%	-45.85	-45.34	-47.57	-47.61

The overall pressure ratio is virtually constant between the TF2050, the TF2050_ITR and the CCTF2050. The OPR of the CCTF2050_ITR is slightly reduced. This is primarily due to the change in bypass ratio and hence core mass flow. The significantly smaller core mass flow of the CCTF2050_ITR means that the blades at the back of the HPC are noticeably smaller, leading to a lower polytropic efficiency of the component. Hence, the compressor delivery temperature limit is reached at a lower pressure ratio. Nonetheless, the fuel burn benefit did not improve, when the OPR was reduced further in an effort to alleviate the HPC penalty due to excessively small HPC blades.

The fuel burn estimate is directly dependent on the mission average SFC (SFC_{ave}), the engine weight and the fan diameter. The specific fuel consumption of the optimal combined cycle engine with inter-turbine reheat is approximately 2.5 % lower than that of the TF2050, which signifies a smaller reduction than that of the CCTF2050. However, the fan diameter is even smaller due to the more power dense core engine. The fan of the CCTF2050_ITR is approximately 5 cm smaller than the fan of the CCTF2050 and almost 10 cm smaller than the fan of the TF2050. Additionally, the CCTF2050_ITR weighs only about 144 kg more than the TF2050, while the CCTF2050 weighs almost 350 kg more than the TF2050. The combination of the lower engine weight and smaller fan diameter of the CCTF2050_ITR offsets the slight increase in SFC relative to the CCTF2050 and even enable a small reduction in mission fuel burn. It should be noted, that the combined cycle concept improves the thermal efficiency of the engine relative to the TF2050, while the propulsive efficiency remains the same for the CCTF2050 and is slightly reduced for the CCTF2050_ITR. These trends were to be expected as the bottoming cycle was conceptualised to improve thermal efficiency, while the increase in specific thrust naturally led to a reduction in propulsive efficiency. The effect is more prominent for the CCTF2050_ITR, because the fan pressure ratio and the heat transfer to part of the bypass flow are greater.

The fact that the CCTF2050_ITR offers a fuel burn benefit over the CCTF2050, no matter how small, despite the fact that inter-turbine reheat alone actually worsens the efficiency of a year 2050 turbofan engine (see TF2050_ITR), proves that there is a synergistic effect between the two technologies. The increased mean temperature in the expansion process allows for a smaller engine core, while the increased LPT EGT is utilised by the bottoming cycle rather than wasting the heat energy to the environment.

Figure 5-1 illustrates the fuel burn benefits determined in this investigation relative to the TF2050 and their respective uncertainties. The TF2050_ITR does not offer a fuel burn

benefit as mentioned before, but at least appears to suffer a relatively small uncertainty based on the discussion in this chapter. The CCTF2050 and the CCTF2050_ITR on the other hand may offer small fuel burn benefits of around 3.2 %, but these estimates are much less certain. In a worst case scenario, both combined cycle engines could actually burn more fuel on a 7000 nm mission than the TF2050. The uncertainties around the fuel burn estimates compound a number of different effects, which are not all equally likely to manifest. A probability analysis (e.g. Monte-Carlo Analysis) would illustrate the most likely fuel burn benefits in the identified ranges by evaluating each uncertainties' individual contribution. Such an analysis was beyond this work, but would add further confidence in the results and could help evaluate the concepts more definitively.

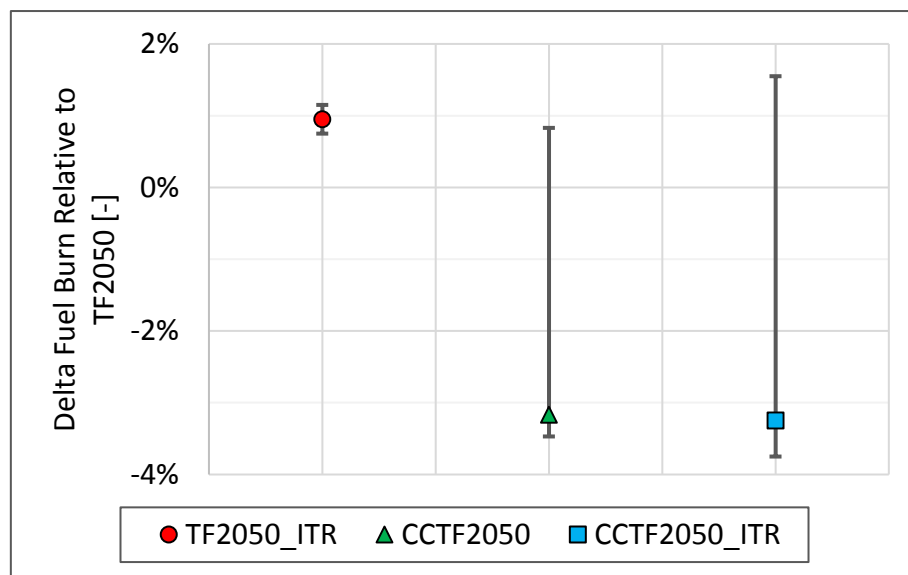


Figure 5-1: Comparison of fuel burn estimates and uncertainties

Overall, the CCTF2050 appears to be the more promising of the two combined cycle arrangements. The fuel burn benefit of the CCTF2050_ITR may yet increase more than the CCTF2050 due to fully automated optimisation studies, but the increased uncertainty in the results may negate that completely. Especially the likely necessary but unaccounted for weight changes due to the reheat combustor and the lifing concerns for HPT2 and LPT are likely to impede the performance of the CCTF2050_ITR. The higher working temperatures through the MHX also mean, that the weight estimates for the MHX are less dependable, despite the incorporation of FEM results in the design process. Finally, the added complexity of the engine arrangement will make it more difficult to maintain, increasing the direct operating costs of the engine even more than for the CCTF2050.

5.2.2 Performance relative to Flightpath 2050

Flightpath 2050 calls for a 75 % reduction in energy consumption and CO₂ emissions for air traffic in the year 2050 relative to the year 2000. 7 % of that reduction is predicted to be attainable through changes in air traffic management measures, leaving a 68 % reduction target for future aircraft split between the airframe and the engine. The conventional turbofan architecture still has some room for further improvements and is projected to enable a 45.8 % fuel burn reduction relative to a representative year 2000 baseline configuration when deployed on an advanced tube-and-wing (ATW) aircraft. Novel engine cycles therefore need to be developed that can deliver the remaining 22.2 percentage points of fuel burn reduction in order to meet the flightpath 2050 target. The remaining necessary 22.2 % fuel burn reduction relative to the baseline LRTF2000 means that the new concept needs to provide a 40.9 % fuel burn reduction relative to the reference TF2050. It is unlikely that any of the novel core engine concepts will enable such reductions alone.

Neither the CCTF2050, nor the CCTF2050_ITR enable fuel burn reductions that are even close to those targets. While they are predicted to enable a fuel burn reduction of about 3.17 and 3.25 % over the TF2050 respectively, not even the most optimistic modelling assumptions would allow for a large enough improvement to meet the Flightpath goals. Furthermore, the fuel burn estimates are still subject to substantial uncertainties and detailed design studies of the unaccounted for effects and components will likely reduce the predicted benefits. Combining novel technologies further to exploit synergies between them may raise the fuel burn benefits, but the Flightpath 2050 goals will likely remain extremely ambitious with a tube-and-wing aircraft configuration and two main engines.

5.2.3 Performance relative to competing concepts

The remaining loss sources of the conventional turbofan architecture can be exploited in a number of ways by different technologies, some of which may be seen as direct competitors to the CCTF2050 and the CCTF2050_ITR (see section 2.2). This section compares the fuel burn benefits predicted for the combined cycles to the estimates of those competing and some potentially subsidiary concepts.

Intercooling

Intercooling has been shown to lower the mission fuel burn of a geared turbofan engine similar to the TF2050 by about 4.9 % [29], although with the caveat of an increased

turbine entry temperature at cruise. Some of that fuel burn benefit may be traded for a reduction in NO_x emissions, because the compressor delivery temperature of an intercooled engine will be lower than that of a conventional turbofan with the same OPR. These benefits appear similar to those determined for the combined cycle engines investigated in this work, but the uncertainty of the intercooled numbers is supposedly significantly lower. Additionally, intercooling requires only one additional major component, if the likely necessary variable area nozzle of the intercooler duct is neglected, while the combined cycle engine requires at least four additional components and a host of auxiliary systems. Moreover, intercooling concepts have been researched and tested for a number of years now. Hence, many of the detailed design solutions for the heat exchanger and its sub-components have already been identified and accounted for, even if they are not yet ready for mass production.

Theoretically, adding intercooling to the CCTF2050 or the CCTF2050_ITR should improve the fuel burn benefit. However, the pre-cooler and the intercooler would compete for the same cooling flow and position on the engine, and those limitations may prevent any substantial benefits to arise. Intercooling may still increase the fuel burn benefit of the CCTF2050, if an even more complex engine arrangement were considered. The combined cycle could feature a reverse flow core arrangement (see [25],[26]), with the intercooler at the back of the engine. That intercooler would not transfer heat from the IPC exhaust flow to part of the bypass air, but to the S-CO_2 exiting the compressor. That way the intercooler and the pre-cooler of the bottoming cycle would not compete for the same coolant mass flow and the S-CO_2 would be pre-heated before entering the MHX. The preheating effect may be used to reduce the size of the MHX or raise the peak temperature in the bottoming cycle. However, such a configuration would be extremely complex, difficult to control and may not actually offer substantial fuel burn improvements, because the temperature levels of the IPC delivery air and the S-CO_2 compressor outlet flow are fairly similar. The work split between the IPC and HPC could be changed to raise the IPC delivery air temperature, but intercooling is most effective if applied early in the compression process [24]. Ultimately, the feasibility of such an arrangement would have to be assessed in a dedicated design and optimisation study with similarly, if not more, detailed component models as employed in this investigation.

Intercooled recuperation

Possibly the most directly competing concept to the combined cycle engine is the intercooled recuperated aero engine (IRA) concept. Both concepts draw heat energy

from the LPT core exhaust via a heat exchanger, though the energy is transferred back to the compressor delivery air in case of the IRA concept instead of to a secondary power cycle. IRA configurations have also been researched for decades now, with a number of component tests having been completed already. Novel heat exchanger arrangements have been developed recently at Aristotle University of Thessaloniki (AUTH) and supposedly these have led to SFC improvements of around 8 % over the TF2050 [37]. That particular configuration uses a secondary fluid recuperator (SFR) to transport the heat from the back of the engine to the back of the compressor without having to recirculate the compressor delivery air. The fuel burn benefit of the IRA engine with an SFR has not been declared yet, though it is likely to be between 8 and 9 %, as the concept introduces multiple large and heavy components that will offset some of the fuel burn reduction that arises from the reduced SFC. If true, that means the IRA engine concept is predicted to offer more than twice the fuel burn reduction of the CCTF2050_ITR, while featuring fewer additional components to maintain.

One of the potential bottlenecks of the SFR concept is the fact that sodium is currently assumed to be used as the heat transfer fluid. Pure sodium is extremely volatile and highly flammable and may therefore not be feasible for aero application for safety reasons. Other sodium based fluids may be less flammable, but that does not necessarily make them safe for aero application. Different heat transfer fluids may offer similar benefits, though that has not been shown as of yet. If the SFR concept has to be abandoned altogether, the more conventional MTU Lancette recuperator concept developed in the NEWAC programme would only offer a more modest SFC benefit of about 2.6 %, which is very similar to the figures of the CCTF2050 and CCTF2050_ITR. Hence, the fuel burn benefit would likely be comparable as well. Nonetheless, the IRA concept would probably still be the preferred concept for further development, because it requires fewer novel components than the combined cycle and the technology readiness level (TRL) of the Lancette recuperator is higher.

The two concepts are unlikely to be combined in one engine, as they compete for the same heat source. The recuperator could be moved further upstream in the expansion process in order to further increase the combustor inlet temperature and leave the LPT exhaust gas heat for the bottoming cycle, but the resulting pressure losses between turbines stages and the longer shafts will likely mitigate any potential benefit of such a design.

Topping cycles

Intercooling, recuperation and bottoming cycles all address the core exhaust heat losses of modern turbofan engines. The combustor irreversibilities that arise in conventional 'constant pressure' combustors present another major loss source. Topping cycles are devices that raise the maximum cycle pressure between the last stage of the turbo-compressor and the turbine by including a 'constant volume' combustor. The topping cycle may simply provide pressure-rise combustion without a mechanical connection to the turbomachinery of the engine or it may provide additional power to one of the shafts. Different arrangements have been investigated in the past [148]–[150], with the Napier E.145 Nomad-II engine [151] being one of the best-known examples.

Xisto et al. investigated different design aspects of a pulse detonation combustion (PDC) engine [148], [152] and determined, that the concept may offer a fuel burn benefit of up to 10 % when coupled with intercooling. However, the pressure ratios of the IPC and HPC were increased so much in the most fuel efficient configuration, that the compressor delivery air reached the temperature limit of the TF2050 again, negating any possible NO_x reductions. Additionally, the 'constant volume' combustion process in the pulse detonation chambers leads to significantly higher localised flow temperatures than a conventional 'constant pressure' combustor. The residence times of those hot spots is very short, but the NO_x emissions are significantly increased nevertheless. The most fuel burn efficient PDC engine with intercooling therefore emits considerably more NO_x than the TF2050 due to the compounding effects of the high compressor delivery temperature and the increased flame temperatures in the PDC chambers. Fortunately, multi-objective optimisation studies have shown that the OPR may be changed so the fuel burn benefit is reduced by only about 1 %, while the NO_x emissions are actually reduced significantly relative to the TF2050. It should be noted that this requires the PDC core to operate with lean mixtures. Xisto et al. also determined that even a non-intercooled PDC engine could offer significant fuel burn reductions (up to 7.2 %), but those engines also emit significantly more NO_x than the TF2050. The efficiency of the engine could once again be traded against lower NO_x emissions by lowering the OPR, but the necessary reductions are so extensive that the PDC engine without intercooling may not offer any fuel burn benefit at all.

Bauhaus Luftfahrt investigated the competing 'composite cycle engine' (CCE) concept during the LEMCOTEC and ULTIMATE projects [153], [154]. The CCE includes two V10 piston engines wrapped around the LP shaft that delivers shaft power and is conceptually

similar to the Napier E.145 Nomad-II. Piston engines are always going to be less power dense than gas turbines at aircraft-relevant mass flows, even if they are turbo-charged, which is why the CCE is approximately 43 % heavier than the TF2050. However, the CCE also features extremely high overall pressure ratios (>100), which means that the SFC is reduced significantly. Consequently, the CCE is estimated to offer a 9.3 % fuel burn reduction over the TF2050 despite the additional weight. If the CCE is furthermore intercooled, the fuel burn benefit is predicted to rise to about 11.2 %. However, similarly to the PDC engine, the peak temperatures in the CCE are significantly higher than in the TF2050 combustor. Hence, the CCE without intercooling is predicted to emit approximately 15 % more NO_x than the TF2050. The intercooled CCE however is expected to offer a 12 % NO_x reduction relative to the TF2050 [154].

The pulse detonation engine and the CCE both promise significantly larger fuel burn benefits than the CCTF2050 or the CCTF2050_ITR. However, both concepts also involve a significant number of additional components, whose design implications could not all be accounted for yet. The claimed benefits may therefore be reduced in the future. Additionally, neither concept directly competes with a closed cycle bottoming cycle, which means that there is at least the theoretical possibility to combine the technologies in an even more complex, but possibly more efficient engine. Unfortunately, the intercooler would probably have to be dropped from such a cycle, due to the aforementioned conflict with the pre-cooler of the bottoming cycle. Not incorporating an intercooler may pose a problem in terms of NO_x emissions.

Sebastiampillai [155] currently investigates the feasibility of yet another topping cycle mechanism, the nutating disk engine [156], [157]. However, the technology was originally developed for much smaller power ratings (e.g. UAVs) and is feared not to scale well with mass flow. As a result, the concept is currently only applied to a short-range open-rotor engine with a significantly smaller core mass flow than the TF2050. Nonetheless, the fuel burn benefit is currently estimated to 14.7 % over a projected year 2050 reference engine, offering the highest relative improvement of all the concepts discussed so far.

Combination of concepts

Rolt [27] currently investigates whether combining several of the discussed concepts in one engine might enable a fuel burn saving closer to that required to meet the Flightpath 2050 goals. His research is currently focused on a geared intercooled turbofan engine with an open air bottoming cycle, nutating disk combustion and a reverse flow core

arrangement. Initial studies suggest that the SFC of such an engine may be up to 15 % lower than that of the TF2050, but the corresponding fuel burn benefit that accounts for the component weight and other design implications is yet to be determined.

5.3 Qualitative Emissions, Noise and Cost Discussion

The development of dedicated models to quantify gaseous and noise emissions for the combined cycle aero engines was beyond the scope of this investigation. However, emission reductions are among the main objectives of ACARE and Flightpath 2050. Additionally, any future aero engine will likely have to be shown to emit substantially fewer gaseous emissions than year 2000 or even more recent turbofan engines to be considered for product development by the OEMs. During times of high public awareness, OEMs have to make socially conscious decisions for their products to be accepted. Additionally, initially voluntary goals as in Flightpath 2050 are often used to drive future legislation, even if in a less stringent form. This means it is entirely possible for global emission taxes to be introduced in the future in order to combat global warming, which would give emission reductions an economic incentive.

Gaseous emissions

The gaseous emissions of the CCTF2050 are bound to be slightly reduced relative to the TF2050. The reduced fuel flow will correlate to an essentially equivalent reduction in CO₂ emissions, because the temperatures in the combustor remain the same. The NO_x emissions will furthermore be reduced, though to a lesser extent as Jet-A does not contain much Nitrogen and therefore does not produce much fuel-bound NO_x. Thermal NO_x production will stay the same due to the same flame temperatures. All investigated configurations, including the TF2050, feature a lean-direct injection combustor [21], [22] that is assumed to reduce emissions substantially relative to the year 2000 baseline technology.

The emission characteristics of the CCTF2050_ITR are more difficult to predict. CO₂ will also be reduced equivalently to the fuel burn reduction, but the reheat combustor makes the quantification of total thermal NO_x hard. The inlet mass flow to the reheat combustor already contains NO_x, CO₂ and other gaseous emissions that may affect the formation of further emissions in the combustor. The study of the oxy-poor flame design by Martinez-Bueno [43] showed how important it is to manage the flame temperature in the reheat combustor to manage the NO_x formation processes. The oxy-poor flame combustor can theoretically be designed to produce a much more homogenous

temperature distribution than a more conventional annular combustor, leading to fewer hot spots and therefore NO_x emissions. The oxy-poor flame combustor also operates with fuel-air mixtures closer to stoichiometric conditions, which should help reduce NO_x emissions. Overall, a lot of further research would be necessary to reliably quantify the emissions produced by the reheat combustor, but given the small benefit of the CCTF2050_ITR over the CCTF2050, it is probably not worth the effort for this configuration alone.

Noise

The CCTF2050 and the CCTF2050_ITR are both likely to reduce engine noise noticeably. The heat exchangers in the core exhaust and part of the bypass flow will likely attenuate a considerable portion of the noise produced by the upstream turbomachinery components. The pre-cooler obviously only interacts with part of the bypass flow, reducing its muffling effect. The extent of the noise attenuation effects of heat exchangers is still debatable and is an active research area around the world [32], [94]. Researchers from Aristotle University of Thessaloniki (AUTH) have been investigating the noise attenuation effects of heat exchangers in NEWAC and ULTIMATE, but their results have not yet been published.

The jet noise of the combined cycle engines should also reduce significantly, because the core jet velocities are significantly reduced. The fan is likely the dominant noise source of the combined cycle engines, which may be addressed with advanced liners in the fan casings. Furthermore, the fan diameter of the combined cycle engine is slightly reduced relative to the TF2050, which should help reduce the tip Mach numbers and hence the fan noise.

Development and operating costs

Given the relatively small fuel burn benefits predicted for the CCTF2050 and the CCTF2050_ITR relative to the TF2050 and the complexity of the engines, it is unlikely that either configuration would be an economic success for manufacturers or operators. Further improvements in component performance and weight would be necessary to improve the fuel burn benefit significantly.

The number of additional components is quite high and their development is at a very preliminary stage, which means considerable investments in their development, testing and necessary manufacturing techniques would be necessary. Those additional costs would be at least partially passed on to the operators, but fuel burn savings of about

1.6 t per flight are likely not enough to offset the additional investment and the almost definitely higher maintenance costs. The higher temperatures in the turbines of the CCTF2050_ITR will also affect their component life, which will have a further detrimental effect on the maintenance cost. Finally, even most new engines with a more conventional layout experience significant teething problems within the first few operating years due to slight design changes. Introducing this many new components means that more components can potentially fail and, in a worst case, lead to a grounding of the aircraft.

A direct operating cost estimation tool was developed in ULTIMATE [158] based on the models presented and used by Nalianda. This model was briefly applied during this investigation in an effort to quantify the costs for operators to produce and for manufacturers to operate a combined cycle aero engine with an S-CO₂ bottoming cycle. The model estimated that approximately 625 aircraft with either the CCTF2050 or the CCTF2050_ITR would have to be sold in order for the manufacturer to break even on the investment, which represents a substantial order book given the long-range application. The maximum projected internal rate of return was 9.1 % for 1287 units sold. The TF2050 needed 623 aircraft to be sold to break even and the maximum IRR was 9.3 % at 1319 units sold. Unfortunately, the model does not account for the increased development, manufacturing and unit costs of a combined cycle engine relative to a conventional turbofan. Hence, these estimates are likely not realistic for the present engine concepts. The engine weight is the only indicator of increased manufacturing and development costs for the manufacturer, which is why the lighter TF2050 appears to be only slightly more economical even though it is much less complex than the combined cycle engines.

Ultimately, the CCTF2050 and the CCTF2050_ITR do not appear to be particularly attractive engine configurations from a direct operating cost perspective. Instead, less complex, but similarly or more fuel burn efficient configurations such as intercooled or intercooled-recuperated engines seem like more worthwhile investments.

6 Conclusions and Further Work

This thesis proposes an integrated performance and preliminary design estimation approach to assess the feasibility of two geared turbofan engine arrangements with supercritical CO₂ bottoming cycles. The geared turbofan with only an S-CO₂ bottoming cycle is called the CCTF2050, while the combined cycle that also includes inter-turbine reheat is called the CCTF2050_ITR. The performance of a third engine without a bottoming cycle but with inter-turbine reheat (TF2050_ITR) was also investigated. The TF2050_ITR was primarily simulated to enable a differential analysis of the individual and the synergistic effects of the bottoming cycle and the reheat combustor in the CCTF2050_ITR. However, the performance results of the TF2050_ITR furthermore present a valuable contribution to knowledge, as the studies showed that adding only a reheat combustor to a projected year 2050 geared turbofan engine model does not yield a fuel burn benefit.

The previous chapters explained the details of the models, the assumptions that had to be made and the results of the conducted parametric studies. Furthermore, the implications of the fixed assumptions and boundary conditions were investigated, the individual model uncertainties discussed and an approximate cumulative uncertainty estimated for both combined cycle engines.

This chapter bottom-lines the findings and gives a feasibility assessment of the combined cycle engine concept with an S-CO₂ bottoming cycle in section 6.1. Section 6.2 outlines a range of future research areas of the concept. These discussions combine developments that would be necessary to validate the current estimates with the proposition of further design change options. Those alternatives range from individual component design changes to more radical cycle variations. Finally, section 6.2.7 proposes a technology roadmap that incorporates the discussed investigations in what is thought to be a sensible development strategy for the concept, should further research be conducted.

6.1 Feasibility of Concept

The considerable level of detail of the heat exchanger models applied in this study allow for a good approximation of the optimum performance of the CCTF2050 (-3.17 % fuel burn) and the CCTF2050_ITR (-3.25 % fuel burn), despite the fact that actual optimisations were not possible and parametric studies had to be conducted instead.

The modelling approach also enables the evaluation of the remaining uncertainties around those fuel burn benefits without excessive further effort. The overall approach has therefore proven its merit for aero application and promises to be equally beneficial when applied to other applications that experience similarly strict design constraints, such as marine propulsion or small-scale off-shore power plants. However, it might be more efficient to use the more traditional iterative design approach with separate performance and preliminary design models, if the size of the engine is a secondary concern and the individual components are not integrated in such a way that they affect each other's performance as much as the see pre-cooler and outer bypass for example.

The thermal efficiency of the bottoming cycles in either configuration is relatively constant between 18 and 21 % near the fuel burn optimum, as long as the turbomachinery components are able match the modelling assumptions. The pressure ratio of the compressor and the component efficiencies largely determine the range of attainable fuel burn benefits, which is why the development of more detailed turbomachinery models is crucial.

The useful power output and hence the improvement potential of the bottoming cycle primarily depends on the S-CO₂ mass flow. The S-CO₂ mass flow in turn is determined by the design of the pre-cooler and the MHX, and the compressor inlet conditions. The closer the compressor inlet temperature and pressure are to the critical point, the greater is the variation of the specific heat capacity of the S-CO₂ flow inside the pre-cooler. High specific heat capacities mean that the pre-cooler has to transfer a large amount of heat to achieve a small temperature drop in the S-CO₂. Hence, a compromise has to be found between the heat exchanger designs and the operating point of the compressor, which is only really possible with detailed component models as were developed here. The compressor inlet conditions have to deviate from the critical point to increase the amount of S-CO₂ the pre-cooler can cool, but this means that the compressor does not benefit much from real gas behaviour anymore. The compressor of the CCTF2050 for example consumes around 51 % of the power produced by the two S-CO₂ turbines, which is significantly more than hoped for in an S-CO₂ cycle. The compressors of some of the highly efficient stationary S-CO₂ power cycles are predicted to consume less than 30 % of the produced power [55]. Less volatile real gas behaviour may simplify the engine control system, but is obviously contradictory to the original idea of an S-CO₂ cycle.

The number of tubes in the two heat exchangers can also be used to increase the heat duty of the pre-cooler and manage the outlet temperature of the MHX, but they also affect

the external pressure losses. This leaves the coolant mass flow, which determines the length of the individual pre-cooler tubes at a given inlet Mach number, as one of the few seemingly independent ways to increase the heat transfer. However, the pre-cooler outer diameter may force a contraction of the outer bypass duct, raising the pressure losses in the remaining cold flow.

Assuming that real gas turbomachinery design and development proceeds quickly and far enough, the results clearly show that the feasibility of the combined cycle engine pivots around the performance and design of the pre-cooler. Design improvements for the pre-cooler are the most likely to improve the fuel burn predictions and therefore the feasibility of the concept, but many modelling assumptions of this study may already be too optimistic.

Ultimately, the present models and simulations suggest that the concepts of geared turbofan engines with a supercritical CO₂ bottoming cycle, and possibly inter-turbine reheat, do not offer more significant fuel burn savings than other novel engine concepts. Unfortunately, many competing concept studies do not outright declare the uncertainties of the fuel burn benefits due to modelling assumptions. It is therefore difficult to ascertain a like-for-like comparison. It is likely that some of the seemingly more promising concepts have underestimated the performance implications of some design constraints, giving rise to significant uncertainties around the claimed fuel burn benefits. However, for some of these concepts those uncertainties would have to be significantly larger than the uncertainties of the present models to lower their fuel burn benefits to the range of the CCTF2050 (-3.17 % fuel burn) and the CCTF2050_ITR (-3.25 % fuel burn).

6.2 Further Work

The models described in chapter 3 incorporate more design details than many other concept evaluation studies. This enables a more reliable assessment of the concept's feasibility, but it also introduces a potential problem. The modelling assumptions and the chosen component designs may be wrong or at least far from optimal, which would limit the possible performance benefit of the concept. Moreover, the detailed models are slower and less stable than more simplistic models that use prescribed component parameters such as efficiencies and pressure losses.

This section outlines possible topics of further work that should either be pursued to improve the stability and reliability of the present models, or which may facilitate further performance improvements.

One imminent problem that should be addressed prior to any further investigations is the resolution of the fluid property files. Recent side studies showed, that the S-CO₂ fluid property file is adequately precise close to the critical point, but the pressure level at which the resolution is decreased (8 MPa) is too close to the critical point. As a result, the non-linear trends of S-CO₂ in that region might be underestimated and the parametric studies presented in this work might not be as precise as initially thought based on the assessment of Belmonte [61]. A case study was performed to gauge the error that may have been introduced to the present simulation because of this. The optimal CCTF2050 configuration presented in section 4.2.1 was recalculated with more refined fluid property tables. The fuel burn benefit was reduced from 3.174 % to 3.173 % as a result of this, while the calculation time rose from around 4.5 to 12 seconds. These results suggest that it is not reasonable to increase the fluid property tables beyond the level used throughout this thesis. However, it cannot be guaranteed that the introduced error will be as low as it is in this case for every simulated configuration and that the trends presented in sections 4.2.2 and 4.3.2 may not have been slightly altered because of these inaccuracies. A broader sensitivity study should be conducted to confirm that the fluid property tables that were used in this investigation suffice when pursuing the mission fuel burn minimum.

6.2.1 Model stability and optimisation

The component and engine models created for this investigation are theoretically capable of design and off-design performance simulations. Unfortunately, the strong-nonlinear behaviour of the fluid properties, the large amount of components and the level of detail of some of those components reduce the mathematical stability of the models. In their current state, the models' convergence depends very much on how the large number of iterative parameters (algebraics) is that are initialised prior to the execution of the models. The sensitivity can be so extreme, that changing the initial estimate of the flow temperature in one section of the pre-cooler by only 1 K can be prohibit convergence. Furthermore, it does not necessarily help convergence to have initial estimates close to the solution, as the simulations have shown.

In order to have a chance of reaching convergence during off-design simulations, it is necessary to simulate a large number of operating points between cruise and Top-of-Climb conditions for example and initialise each algebraic manually over and over again. Due to these instabilities it is currently not possible to pass the engine models to an

external optimiser such as Isight to determine the optimal design and off-design performance of the two concepts.

Therefore, the first priority of any possible future development of the concept should be to improve the stability of these models and enable off-design simulations. The models could then be used in an optimisation environment to prove or refute the results of this investigation. The stability may be helped by splitting the component code of the heat exchangers into more sub-models with a wider inheritance tree. Choosing a different set of boundary conditions and algebraics may also help.

6.2.2 Preliminary turbomachinery design

The stability of the models and the accuracy of the results may be improved simultaneously by extending the S-CO₂ turbomachinery models to use mean- and streamline performance estimation models instead of scaled component maps. Including such capabilities will increase the number of equations to be solved and may therefore increase the number of iterations, but it will also help mitigate some inherent problems of using scaled real gas maps.

The S-CO₂ component maps implemented here were derived for a specific set of boundary conditions which differ considerably from the boundary conditions encountered here. Non-dimensional component characteristics should theoretically be scalable to other applications, but for real gas compressors for example the relationship between enthalpy rise and pressure ratio will change dramatically depending on the inlet conditions, even if the non-dimensional inlet mass flow is the same. The maps may not capture these dependencies accurately enough, leading to long and ultimately unsuccessful iterations or unrealistic results. Additionally, the default scaling methodology for component maps of PROOSIS is not accurate when applied to real gas maps. 1D-preliminary design and performance estimation models on the other hand could directly take the changes in fluid properties into account for every condition, which might improve model stability. These methods could also account for design details a scaled map could not if the original component did not include them. Mean- and streamline design methods could for example simulate the effects of variable inlet guide vanes, no matter if the compressor from Barber-Nichols Inc. (BNI) had some or not. The methods could furthermore relate component size and manufacturing limits such as minimum leading edge radii or tip clearances. Ideally, the weight of the compressor and

turbine casings would also be estimated by these models, alongside the performance and weight of the necessary seals and bearings.

The main benefit of such models would be that pressure ratios, component efficiencies and component weights would be directly linked, which would substantially increase confidence in the claimed fuel burn benefits. Even more detailed methods such as CFD and FEM would obviously further improve confidence in the results, but the resulting calculations would be so complex and slow, that whole engine optimisation could not be realistically performed without investing extreme resources. The potential improvements identified by such methods are not likely to warrant such expenses this early in the design process. Instead, it would be more beneficial to include preliminary design methods for the shafts and the power transmission systems of the engine in the simulations. That way, mechanical constraints and their thermodynamic repercussions could be better understood and accounted for.

6.2.3 Alternative heat exchanger designs

The involute spiral tube heat exchanger design implemented for the pre-cooler and the MHX is highly compact and features relatively low drag. However, the performance of the engine is still limited by some of its geometric constraints and design implications.

First of all, the tubes have to be arranged perpendicular to the engine axis. This means that the outer diameter of the heat exchanger can get very large and affect the performance of other components. If an arrangement were employed where the tubes are leaned against the engine centreline, the outer diameter of the heat exchanger would reduce significantly for the same inlet area. Unfortunately, such an arrangement also requires for the external flow to be turned toward the inlet of the heat exchanger in the preceding duct, which has been shown to be difficult in the past [15]. The Lancette and CORN recuperators of MTU and AUTH (see sections 2.5.2.1 and 2.5.2.2) follow the design philosophy of an inclined heat exchanger, but their tubes are arranged differently, making a like-for-like comparison difficult. Those recuperators would likely enable shorter diffuser designs and maybe even better overall performance for the MHX, but the lack of publicly available design and performance data would have necessitated further dedicated design studies. Dispensing such resources for uncertain and limited gains, and without a way to adequately validate the findings, was deemed unreasonable during the concept development stage, but may be an option for further work.

The performance of the involute spiral heat exchanger may also be improved, by integrating the heat exchanger and the diffuser. The first rows of tubes could actually sit inside the diffusing duct, rather than just behind it, in order to shorten the diffuser. The flow Mach number between those tubes could be kept low by not having a continuous front of tubes in the diffuser, but by arranging them in a wider zig-zag pattern (see Figure 6-1). That way, the flow is not accelerated as much as it would be with a straight front. The heat transfer coefficients around those lead tubes may be lower, due to the lower flow velocities around the tubes, and the total pressure losses across the heat exchanger may be higher if the flow Mach number is not diffused as far as originally intended before reaching the full front of the tube stacks. However, such a design might still be more effective than the configuration investigated here. Additionally, the first tubes could feature a stronger internal design to protect the downstream tubes from foreign object damage.

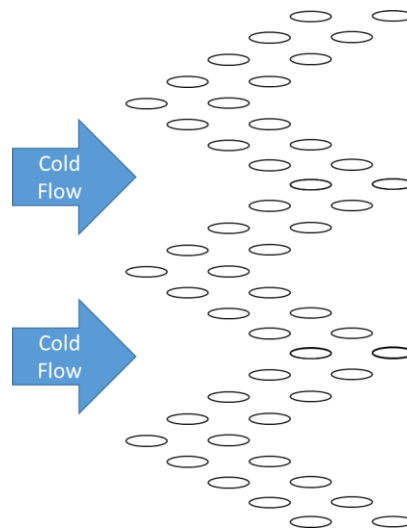


Figure 6-1: integrated pre-cooler and diffuser

Alternatively, the heat exchangers could feature finned elliptical tubes to increase the heat transfer surface area and offset the low heat transfer coefficients of the external flow. However, it remains to be seen if the performance of the engine would actually improve, because the drag and weight of the individual tubes would also increase.

Considering the high internal pressures and subsequently high stresses in the tube walls, it may also be possible, though unlikely, that circular tubes could actually offer a fuel burn benefit over the currently assumed elliptical tubes with an aspect ratio of eight. Circular tubes would produce more drag and transfer less heat, because of the reduced heat transfer surface area for a given internal hydraulic diameter. However, the circular

cross-section most likely means that the solidity of the tubes could be reduced, possibly lowering the component weight. Lower aspect ratio elliptical tubes may present a sweet spot between the two designs, and the combination of Rengel's [100] results and the MSc research of D'Antino should enable their assessment.

For a more significant design change, the more conventional tubes could be replaced completely with carbon nanotubes, though their performance and manufacturing processes are still being researched at the moment [159].

Previous studies also investigated the feasibility of heat pipes [85], [86] for aero engine intercoolers and recuperators. The biggest drawback of a heat pipe design for the pre-cooler is that the S-CO₂ would still need to be directed toward the pre-cooler duct, because the heat pipes would have to be straight and as short as possible. Additionally, the S-CO₂ side pressure losses are very likely to be higher than inside the involute spiral tubes, so it is not certain that the mission fuel burn would be reduced. However, heat pipes would improve the safety of the concept, as no S-CO₂ would leak from the bottoming cycle if a pipe broke. Maintenance of the heat exchangers would also be easier, as whole heat exchanger modules could be replaced without having to evacuate the tubes. The studies of Camilleri et al. [85] suggest that a heat pipe design can actually be lighter than some other intercooler arrangements investigated in NEWAC [24], but it is not certain that the same can be said when compared to the involute spiral heat exchanger with elliptical tubes, which is why the idea was dismissed by Anselmi Palma [26] in LEMCOTEC. Camilleri et al.'s study also did not explain if and how the inlet and outlet manifolds and the rest of the heat exchanger structure were accounted for in the weight estimates.

Overall, there are a number of different design changes that could be applied to see if the fuel burn of the CCTF2050 and the CCTF2050_ITR might be improved. However, it seems unlikely that any of these changes are going to substantially reduce the fuel burn.

6.2.4 Design of additional components

The uncertainty around the fuel burn figures determined using the present CCTF models can be reduced, if the performance and weight of further components were appropriately accounted for. Unfortunately, the weight of the S-CO₂ itself is not properly accounted for in the current models, because the main connecting ducts are not included as dedicated models yet. Such models would be necessary to calculate flow velocities and hence the total fluid volume. Furthermore, such models would not only allow for a more realistic

estimate of the weight of the ducts themselves, but their weight could be traded against internal pressure losses via their diameters for example.

Moreover, simple models to design the valves, internal bypass ducts and possible fluid storage tanks would further increase confidence in the weight estimates of the combined cycle engine. The weights depend on the chosen type of control system for the bottoming cycle, which depends on the optimal off-design performance. Consequently, an integrated assessment of the component weight of the control system is the only realistic way to find the fuel burn optimal design and operation of the combined cycle engines.

The performance of the MHX diffuser would probably be improved, if the LPT were redesigned with a lower hub-to-tip ratio or a lower exit Mach number, or both. However, such a study would need to investigate how the performance of the LPT might change because of this, making it increasingly complex. The design of the LPT should probably be integrated in the combined cycle engine process anyway, because the power output is noticeably reduced due to the bottoming cycle. The loading of the turbine stages is therefore reduced and the efficiency could increase, or the number of stages might be reduced. However, it is not clear how large the effect of these changes would be on mission fuel burn and if they are worth the added complexity in the engine model. A separate parametric design study might clarify the situation and could help derive a less complex surrogate model to be integrated in the combined cycle simulations.

6.2.5 Safety and Failure Scenarios

A further critical area that will affect the chances of the combined cycle engine concept to ever make it into production are potential failure scenarios. The combined cycle entails a whole host of additional components and systems, of which only the largest, most performance-relevant were discussed in detail in this thesis. In order for the concept to enter production however, every valve, coupling and seal would have to be investigated for failure potential. The amount of redundancies that would be necessary to ensure the safety of the passengers and crew may increase the total engine weight considerably and could therefore reduce or even negate the predicted fuel burn benefits. Typically such design considerations are not accounted for in concept studies such as this, which might only develop a design to about TRL 2. However, they should be addressed when moving toward TRL3 or 4 at the latest, due to the significant effect these additional components may have on mission fuel burn.

Overspeed scenarios for the S-CO₂ turbomachinery, while necessary to investigate, are unlikely to be critical, because their low mass and inertia can probably be contained by the engine casings and would pose little to no threat to the safety of the aircraft. Foreign object damage (FOD) may pose a more significant threat to the bottoming cycle however. FOD to the MHX is relatively unlikely as it is located downstream of the LPT. Only a major incident like an LPT blade-off event in the core engine, a disc burst or a shaft failure would lead to a damaged MHX. Those events are more dangerous in their own right however and the engine would be shut-down either way. If however, a foreign object were to find its way into the pre-cooler bypass duct, the tubes of the heat exchanger could burst and the S-CO₂ be rapidly released. Emergency shut-off valves would be included in the S-CO₂ ducts leading to and from the pre-cooler to minimise the amount of S-CO₂ that would be set free, but this would not mitigate the risk completely. CO₂ is not flammable and the bypass flow is at a relatively low temperature anyway, so there would be no risk of igniting the bypass flow. However, the immense pressure differential between the cooling air and the S-CO₂ may lead to a violent rupture of the damaged tubes. The mass of the tubes and their broken pieces is relatively small however, which may keep their impact at bay. In order to retain the remains of the tubes, it may be beneficial to line the casing around the pre-cooler with Kevlar, though this will increase the weight and cost of the engine. Alternatively, the first rows of the heat exchanger could be made stronger in order to act as a first defence against incoming debris. Simulations and tests would have to be conducted in order to assess whether or not any of these measures are necessary and dependable.

Finally, the effects of dust, soot or volcanic ash depositions on the heat exchanger tubes would need to be investigated. They might simply reduce the effectiveness of the heat exchangers and hence the bottoming cycle. Depending on how much either of the two heat exchangers are affected, this may impact the performance and safety of the engine differently. If the heat effectiveness of the pre-cooler were reduced significantly more than the MHX, which is likely given their position in the engine, the compressor inlet temperatures of the cycle could rise considerably. In a best case scenario, the bottoming cycle would find a new stable operating condition and its performance would simply be diminished. In a worst case, the higher temperature levels in the bottoming cycle lead to either an overspeed scenario or the MHX tubes break, because their structural integrity is lowered by the higher metal temperatures. The problem may be aggravated if the heat exchanger tubes were made from ceramic composites. Previous heat exchanger concepts have investigated ceramic materials in order to increase their integrity at high

temperatures [110] and lower their weight. However, TBC coatings of HPT NGVs and blades have shown to be susceptible to soot induced failures. The particles can stick to the relatively rough surface and create hot spots, weakening the structure of the blade. A similar scenario for the MHX would have to be avoided at all cost, though the working temperatures of the MHX are significantly lower than those in the HPT.

6.2.6 Alternative cycle designs

The performance figures of the CCTF2050 and the CCTF2050_ITR naturally depend on the assumed component designs, but also on the cycle arrangement. This section proposes alternative cycle arrangements that could be investigated to possibly increase the fuel burn benefit of the combined cycle engines.

Recuperated bottoming cycle

The combined cycle aero engine concept may benefit from including an S-CO₂ to S-CO₂ recuperator yet. The current study determined that a printed circuit heat exchanger would not improve the mission fuel burn of the CCTF2050, because the heat exchanger would be too heavy for the SFC benefit it provides. However, a lighter design may increase the heat transfer to weight ratio of the recuperator enough to actually offer a fuel burn benefit. The printed circuit heat exchanger design was originally chosen, because it is extremely compact and able to withstand high internal pressures, but it may also be possible to design tube type heat exchangers with similar traits.

The pressure differential between the hot and cold side is actually smaller than for the MHX, so it should be possible to create sufficiently strong heat exchanger tubes for the S-CO₂ to S-CO₂ heat exchanger as well. A potential problem of a tube heat exchanger is the pressure loss of the flow around the tubes. Those pressure losses frequently exceed 5% (see air-side losses across pre-cooler) and may negate any benefit of pre-heating the compressor delivery S-CO₂ before entering the MHX. Furthermore, the working pressures of both fluids are quite high, which means the necessary flow area is relatively small, limiting the length of a pass in a cross-flow arrangement. Only a pure counter-flow arrangement is likely to offer any significant heat transfer.

As a further option, the recuperator could be replaced with a regenerator. Unfortunately, regenerators are cyclic devices, which can be problematic when working with continuous flow machines such as gas turbines. Additionally, regenerators have to be manufactured from materials of high thermal conductivity in order to release the heat energy to the cold

flow again, which causes heat conduction in the flow direction and therefore limits the achievable effectiveness.

Even if these design options should not improve the mission fuel burn of the CCTF2050, the CCTF2050_ITR may profit from a recuperator. The higher overall temperature levels in the bottoming cycle of the CCTF2050_ITR mean, that the recuperator may transfer a larger amount of heat and therefore offer a greater improvement. The temperature drop in the S-CO₂ before entering the pre-cooler also means, that either the pre-cooler dimensions can be reduced, the compressor inlet conditions can be reduced or the S-CO₂ mass flow can be increased. The same would theoretically be true for the CCTF2050 of course, but the deltas are relatively small.

Alternative working fluids

Supercritical CO₂ was originally chosen as the working fluid of the bottoming cycle, because it offers great power density, good thermal efficiency in the relevant temperature range, is not flammable and relatively cheap (see section 2.3 for more details). The present investigation has shown that the concept may offer a fuel burn improvement over the reference TF2050, but the benefit is smaller than anticipated and the pre-cooler is the main limiting factor. An open air bottoming cycle would not need a pre-cooler as the working fluid could be reunited with the core mass flow after being expanded in the turbine. This may enable a greater useful power output by the bottoming cycle and therefore a greater fuel burn improvement, despite the less favourable fluid properties of air and the subsequently lower thermal efficiency of the bottoming cycle. Rolt currently investigates such an arrangement in his PhD research. Organic fluids and steam have already been investigated in previous studies [12], [13] and were determined not feasible for aero application. Investigations investigating Kalina cycles as aero engine bottoming cycles were not found in the public domain at this point, which is likely due the very complex operation of the cycle even at design conditions. Even more detailed component models would be necessary to adequately represent a Kalina cycle. Nevertheless, Kalina cycles supposedly offer great thermal efficiencies using low grade heat sources [50], so they might be worth considering for aero application after all.

Triple cycle concept

Further continuous research at Cranfield University into possible ways to improve the combined cycle concept includes the MSc projects of Zhu [91] and Beretens [160]. These studies aim to reduce the dependency of the bottoming cycle's performance on the air-

pre-cooler. The triple cycle concepts splits the total heat duty of the air-pre-cooler of the CCTF2050 between the air-pre-cooler and a cryogenic pre-cooler working in tandem. First, the S-CO₂ is cooled to within approximately 3 – 10 K of the compressor inlet temperature in the air-pre-cooler. The large change in specific heat capacity in the S-CO₂ means that the remaining few degrees temperature drop still require a substantial amount of heat to be transferred. That heat is transferred to cryogenic hydrogen via a second pre-cooler located between the air-pre-cooler and the S-CO₂ compressor. The heated hydrogen is subsequently burnt in the main engine's combustor, replacing Jet-A. The combustor would have to be redesigned in order to efficiently burn hydrogen instead of Jet-A. The flame speed for example is much higher for hydrogen than it is for Jet-A and thermo-acoustic instabilities are a major concern in a hydrogen combustor. Micro-mix combustors [161] or the oxy-poor flame combustor developed in the AHEAD project [162] might be feasible design solutions. An added benefit of hydrogen combustion can be that the NO_x emissions might be reduced, because no fuel-bound NO_x is generated and the low residence times stifle the formation of thermal NO_x.

The heat transfer in the cryogenic pre-cooler is limited by the fuel flow necessary for the main combustor, because large amounts of heated hydrogen cannot be safely stored on board. The problem is aggravated by the fact that hydrogen has an LHV of about 120 MJ/kg, while Jet-A only has an LHV of about 43 MJ/kg, which means the hydrogen combustor requires only about 36 % of the fuel flow a Jet-A combustor would need. The cryogenic system may feature multiple pressure levels to increase the effectiveness of the heat exchanger.

The triple cycle concept could be used in three ways:

- I. Keep the S-CO₂ compressor inlet conditions constant and reduce the size of the air-pre-cooler for a given total heat duty
- II. Keep the compressor inlet conditions and the size of the air-pre-cooler constant and increase the heat duty of the cooling system and hence the S-CO₂ mass flow
- III. Lower the compressor inlet conditions and keep the total heat duty of the cooling system constant to reduce the power consumption of the S-CO₂ compressor and increase the bottoming cycle's thermal efficiency.

After passing through the cryogenic pre-cooler and before entering the main combustor, the hydrogen might furthermore be used to cool the electrical systems of the aircraft or the cabin air. Additionally, the hydrogen mass flow could be increased if part of it were used in the fuel cells that are envisioned to replace the APU on the advanced tube-and-

wing aircraft. This would increase the total heat duty of the pre-cooler and should therefore help to reduce the mission fuel burn further.

The ongoing investigations will have to show which of the three ways to integrate the triple cycle concept offers the greatest fuel burn reductions. The triple cycle concept can also improve the off-design performance of the bottoming cycle, because the compressor inlet temperature is no longer limited by the coolant air temperature alone. The increased fuel flow at TOC and EOR will also increase the total heat transfer in the cooling system.

Distributed Propulsion

Finally, the bottoming cycle concept could be integrated with the electric distributed propulsion (EDP) concept to utilise the useful power of the bottoming cycle more effectively. The main engine of an aircraft using electric distributed propulsion contributes little to no thrust to the aircraft. Instead, the engine expands the exhaust flow to near-ambient conditions and provides the excess power to multiple fans distributed along aircraft fuselage. If a bottoming cycle were to be added to such an engine, the expansion ratio of the LPT would be slightly reduced to accommodate for the pressure loss across the MHX. A considerable portion of the size limitations of the pre-cooler would be alleviated in such a scenario. The pressure losses across the pre-cooler and in a possible outer bypass duct of the engine would not matter much in the overall scheme of things. Additionally, the large degree of integration of engine and aircraft in the EDP concept means that the pre-cooler could more easily use the fuselage surface area to reject the excess heat of the bottoming cycle. Consequently, the heat duty of the pre-cooler could be increased considerably, leading to a greater useful power output of the bottoming cycle. Rejecting the heat via the aircraft fuselage would offer the added bonus of heating the aircraft boundary layer, reducing the drag and hence the thrust requirements of the engine. It is important however, to ensure that the efficiency of the propulsors of the EDP aircraft would not be lowered due to higher inlet temperatures and more turbulent flow. The decay of the pre-cooler effectiveness during hot-day operation or when dirty would be a further concern. Adding intercooling to such an aircraft may offer further fuel savings, because the pre-cooler and the intercooler would not necessarily compete for the same coolant anymore.

An integrated EDP aircraft with an S-CO₂ bottoming cycle, and possibly intercooling, has the potential to offer significantly larger fuel burn benefits than the CCTF2050 and CCTF2050_ITR arrangements investigated in this study, though a detailed analysis is yet to be conducted. Such a configuration would furthermore offer a natural synergy with

hydrogen combustion and the triple cycle concept, as the hydrogen could be used to cool the necessary electrical systems of an EDP aircraft.

6.2.7 Technology Roadmap

This section proposes a possible order in which the mentioned developments could be initiated to best mature the combined cycle concept and increase confidence in its potential. This roadmap (see Table 6-1) only addresses a development of the concept to about technology readiness level (TRL) three. High fidelity simulations (CFD, FEM) and component tests would be necessary to validate the models and improve their accuracy further, in order to mature the concept to TRL 4 and beyond.

The development has been broken down into three development phases:

- Phase one primarily addresses shortcomings of the existing models and introduces meanline design tools for the S-CO₂ turbomachinery
- Phase two proposes the inclusion of more of the engine's sub-systems in the design and performance evaluations to increase confidence in the results
- Phase three explores alternative component designs to further optimise performance

This order is proposed, because it is crucial to ensure the existing models are stable and reliable before they can be extended and the designs can be improved. This will not only help to quantify the improvements due to each design change individually, it will also save time and computational resources that would otherwise be spent on attempting to collect the necessary data from unstable models. Table 6-1 only includes short titles of the larger individual steps of each phase. References to the relevant sections of the proposed further work are included should further details be required. The timeline and duration of each phase would depend on the resources provided to the project and are therefore almost impossible to ascertain at this point.

In the interest of trying to avoid wasting unnecessary heavy computational resources, it is may also be argued to perform some of the stage three tasks, such as the triple cycle concept, before or in parallel to phase one. However, the results would be even less dependable than the results presented in this work due to the added complexity and subsequently less stable and applicable models. Currently, Mikel Beretens [160] is in the process of evaluating the triple cycle concept for his MSc project, but his work clearly illustrates the dependency of his results on the data from the presented combined cycle

engine data, which aggravates the effect of the underlying modelling uncertainties. In the interest of accuracy it is therefore not recommended to further develop Beretens' models until the underlying combined cycle engine models have been improved upon as suggested in phase one and two of the technology roadmap.

The investigation of combined cycle engines using other working fluids, and the integration of the combined cycle engine into an EDP aircraft configuration, would have to start at a lower point in the development process than the other developments discussed so far. Consequently, if these concepts were explored, it would not necessarily be part of the development process of the current models. That is why these two concepts are labelled as 'Parallel Efforts' in Table 6-1 and why they were not categorised as part of the main development phases.

Table 6-1: Proposed research order of further work

Research area			Relevant section
Phase One	I	Stability of models and off-design simulations	6.2.1
	II	Mission Optimisation	6.2.1
	III	First meanline design tools for S-CO ₂ turbomachinery (size, efficiency, integrity)	6.2.2
	IV	Mission Optimisation	6.2.1
Phase Two	I	Effects of MHX diffuser length & outer diameter	6.2.3
	II	Extended meanline S-CO ₂ turbomachinery models (VSVs, OTL, seals, bearings)	6.2.2
	III	Preliminary design of shafts and transmissions	6.2.2
	IV	Ducts, S-CO ₂ storage tank, Control System	6.2.4
	V	Safety and Failure Scenario Analysis	6.2.5
	VI	Mission Optimisation	6.2.1
Phase Three	I	Alternative recuperator design	6.2.6
	II	Triple cycle concept	6.2.6
	III	Alternative MHX and Pre-cooler arrangements	6.2.3
	IV	Mission Optimisation	6.2.1
Parallel Efforts	Electric Distributed Propulsion		6.2.5
	Alternative working fluids		6.2.5

It is proposed that each phase is concluded with a fuel burn optimisation run. Not only will this help track progress and increase confidence in the results, but it will allow for the quantification of each phase's contribution to developing the concept further. Only phase one of the proposed plan proposes two optimisation runs. The first run should be conducted after the stability problems of the existing models are mitigated. This will increase confidence in the fuel burn estimates and might clarify the general feasibility of the concept. Should the fuel burn benefit of the mission optimised CCTF2050 and CCTF2050_ITR exceed the predictions of this work, the concept may be more attractive than anticipated and further development is advisable. If the fuel burn benefit were to virtually disappear however, it is unlikely that more detailed models will make a large enough difference to warrant further research. The second optimisation run of phase one of the proposed research agenda would help quantify the impact of the meanline design tools of the S-CO₂ turbomachinery.

In a similar fashion to the first optimisation of phase one, the optimisation runs at the end of each development phase should be used to re-assess the feasibility of the concept. The assessments could potentially be used to halt all future developments if necessary, avoiding unnecessary expenditure of resources.

REFERENCES

- [1] Airbus, "Flying on Demand 2014 2033 - Global Market Forecast," 2014.
- [2] Boeing, "Current Market Outlook 2014 - 2033," 2014.
- [3] Report of the High Level Group on Aviation Research, "Flightpath 2050," *Isbn 978-92-79-19724-6*, p. 28, 2011, DOI: 10.2777/50266.
- [4] T. Grönstedt, A. Rolt, N. Tantot, *et al.*, "GT2016-56123: Ultra Low Emission Technology Innovations for Mid-century Aircraft Turbine Engines," in *Proceedings of ASME Turbo Expo 2016: Turbine Technical Conference and Exposition*, 2016, pp. 1–13.
- [5] Ultimate, "ULTIMATE project website," 2016. [Online]. Available: <http://www.ultimate.aero/>.
- [6] IATA, "Fact Sheet Climate Change - International Air Transport Association," 2017.
- [7] R. Singh, "Fifty Years of Civil Aero Gas Turbines," Cranfield, 1996.
- [8] F. Ali, C. Xisto, P. Heinemann, *et al.*, "GTP-18-1147: Civil Transport Aircraft Fuel Efficiency ; A Long-Term Technology Maturation Scenario," *J. Eng. Gas Turbines Power*, pp. 1–55, 2018.
- [9] J. A. Pertuze, S. Sato, Z. S. Spakovszky, *et al.*, "N + 3 Aircraft Concept Designs and Trade Studies , Final Report Volume 1," vol. 1, no. December, 2010.
- [10] S. W. Ashcraft, A. S. Padron, K. A. Pascioni, *et al.*, "Review of Propulsion Technologies for N + 3 Subsonic Vehicle Concepts October 2011," no. October, 2011.
- [11] C. M. DeServi, L. Azzini, M. Pini, *et al.*, "GPPF-2017-78: Exploratory Assessment of a Combined-Cycle Engine Concept for Aircraft Propulsion," in *Proceedings of the 1st Global Power and Propulsion Forum*, 2017, p. 11.

- [12] C. A. Perullo and D. N. Mavis, "GT2013-95734: An Integrated Assessment of an Organic Rankine Cycle Concept," in *Proceedings of ASME Turbo Expo 2013: Turbine Technical Conference and Exposition*, 2013, pp. 1–8.
- [13] A. Lundblad, T. Grönstedt, M. Genrup, *et al.*, "ISABE 2015-20101: High Power Density Work Extraction from Turbofan Exhaust Heat," in *International Symposium on Air Breathing Engines*, 2015, pp. 1–10.
- [14] P. P. Walsh and P. Fletcher, *Gas Turbine Performance*, 2nd Editio. Oxford: Blackwell Publishing company, 2004.
- [15] "NEWAC Webpage." [Online]. Available: <http://www.newac.eu/>.
- [16] "LEMCOTEC Webpage," 2016. [Online]. Available: <http://www.lemcotec.eu/>.
- [17] "VITAL Cordis Webpage European Commission."
- [18] T. Grönstedt, D. Au, K. Kyprianidis, *et al.*, "GT2009-60201: Low Pressure System Component Advancements and Its Influence on Future Turbofan Engine Missions," in *Proceedings of ASME Turbo Expo 2009 Power for Land, Sea and Air*, 2009, pp. 1–12.
- [19] F. Jacob, J. Sebastiampillai, A. Rolt, *et al.*, "Deliverable 2.6 – Year 2000 and Initial Year 2050 Reference Configurations," 2017.
- [20] GE.Aviation, "Dust vs. CMCs: A Cool Winner," 2017. [Online]. Available: http://www.geaviation.com/press/ge90/ge90_20170119.html. [Accessed: 27-Jan-2017].
- [21] K. M. Tacina, D. P. Podboy, F. Lee, *et al.*, "ISABE 2017-22606: Gaseous Emissions Results from a Three-Cup Flametube Test of a Lean Direct Injection Combustion Concept," in *International Symposium on Air Breathing Engines*, 2017, pp. 1–19.
- [22] L. Li, Y. Lin, Z. Fu, *et al.*, "Emission Characteristics of a Model Combustor for Aero Gas Turbine Application," *Exp. Therm. Fluid Sci.*, vol. 72, pp. 235–248, 2016, DOI: 10.1016/j.expthermflusci.2015.11.012.

- [23] ISAE, "D3.1 Propulsor and Integration Technologies," 2018.
- [24] A. M. Rolt and K. G. Kyprianidis, "Assessment of New Aero Engine Core Concepts and Technologies in the EU Framework 6 NEWAC Programme," in *27th International Congress of the Aeronautical Sciences*, 2010, pp. 1–11.
- [25] W. Camilleri, "An Assessment of High Overall Pressure Ratio Intercooled Engines for Civil Aviation," PhD Thesis, Cranfield University, 2014.
- [26] E. Anselmi Palma, "Design and Analysis of Environmentally Friendly Civil Brayton Cycles," PhD Thesis, Cranfield University, 2015.
- [27] A. Rolt, J. Sebastianpillai, F. Jacob, *et al.*, "ISABE 2017-22660: Selecting Combinations of Advanced Aero Engine Technologies," in *International Symposium on Air Breathing Engines*, 2017, pp. 1–17.
- [28] X. Zhao, T. Grönstedt, and K. Kyprianidis, "Assessment of the Performance Potential for a Two-pass Cross-flow Intercooler for Aero Engine Applications," *Int. Symp. Air Breath. Engines*, pp. 1–10, 2013.
- [29] X. Zhao and T. Grönstedt, "ISABE 2015-20018: Aero Engine Intercooling Optimization using a Variable Flow Path," in *International Symposium on Air Breathing Engines*, 2015, pp. 1–11.
- [30] C. F. McDonald, A. F. Massardo, C. Rodgers, *et al.*, "Recuperated Gas Turbine Aero Engines, part I: Early Development Activities," *Aircr. Eng. Aerosp. Technol.*, vol. 80, no. 2, pp. 139–157, 2008, DOI: 10.1108/00022660810859364.
- [31] C. F. McDonald, A. F. Massardo, C. Rodgers, *et al.*, "Recuperated Gas Turbine Aero Engines, part II: Engine Design Studies Following Early Development Testing," *Aircr. Eng. Aerosp. Technol.*, vol. 80, no. 3, pp. 280–294, 2008, DOI: 10.1108/00022660810873719.

- [32] C. F. McDonald, A. F. Massardo, C. Rodgers, *et al.*, “Recuperated Gas Turbine Aero Engines. part III: Engine Concepts for Reduced Emissions, Lower Fuel Consumption, and Noise Abatement,” *Aircr. Eng. Aerosp. Technol.*, vol. 80, no. 4, pp. 408–426, 2008, DOI: 10.1108/00022660810882773.
- [33] R. Colin, “The WR-21 Intercooled Recuperated Gas Turbine Engine – Integration Into Future Warships,” in *Proceedings of the International Gas Turbine Congress*, 2003, no. c, pp. 1–6.
- [34] L. Xu, K. G. Kyprianidis, and T. U. J. Grönstedt, “Optimization Study of an Intercooled Recuperated Aero-Engine,” *J. Propuls. Power*, vol. 29, no. 2, pp. 424–432, 2013, DOI: 10.2514/1.B34594.
- [35] D. Misirlis, Z. Vlahostergios, M. Flouros, *et al.*, “Optimization of Heat Exchangers for Intercooled Recuperated Aero Engines,” *Aerospace*, vol. 4, no. 14, pp. 1–16, 2017, DOI: 10.3390/aerospace4010014.
- [36] S. Boggia and K. Rüd, “Intercooled Recuperated Gas Turbine Engine Concept,” in *41st AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit*, 2005, pp. 1–11.
- [37] C. Salpingidou, M. Flouros, M. T. U. Aero, *et al.*, “GT2018-76778: Investigation and Assessment of the Performance of Various Recuperative Cycles Based on the Intercoole Recuperation Concept,” in *Proceedings of ASME Turbo Expo 2018: Turbine Technical Conference and Exposition*, 2018, pp. 1–10.
- [38] K. Vogeler, “The Potential of Sequential Combustion for High Bypass Jet Engines,” in *International Gas Turbine & Aeroengine Congress & Exhibition*, 1996, p. 10.
- [39] S. Köhler, “Performance Increase of the Rolls-Royce UltraFan through Inter-Turbine Reheat,” MSc Thesis, Cranfield University, 2015.

- [40] AnsaldoEnergia, "GT36: Superior Value," 2018. [Online]. Available: <https://www.ansaldoenergia.com/business-lines/new-units/gas-turbines/gt36>. [Accessed: 13-Mar-2018].
- [41] Y. Levy, V. Sherbaum, V. Erenburg, *et al.*, "Flameless Oxidation Combustor Development for a Sequential Combustion Hybrid Turbofan Engine," in *Proceedings of ASME Turbo Expo 2016: Turbomachinery Technical Conference and Exposition*, 2016, pp. 1–10.
- [42] DelftUniversity, "Ahead Project Website," 2016. [Online]. Available: <http://www.ahead-euproject.eu/project-description/>. [Accessed: 17-May-2018].
- [43] V. Martinez-Bueno, "3D LES - RANS CFD Studies of a Novel Secondary Combustor Concept for Low-NO_x Emissions," MSc Thesis, Cranfield University, 2018.
- [44] O. Petit, C. Xisto, X. Zhao, *et al.*, "GT2016-57920: An Outlook for Radical Aero Engine Intercooler Concepts," in *Proceedings of ASME Turbo Expo 2016: Turbine Technical Conference and Exposition*, 2016, pp. 1–11.
- [45] GEPower, "Breaking the Power Plant Efficiency Record," 2016. [Online]. Available: <https://powergen.gepower.com/about/insights/power-plant-efficiency-record.html>. [Accessed: 22-Mar-2017].
- [46] S. Clemente, D. Micheli, M. Reini, *et al.*, "Bottoming Organic Rankine Cycle for a Small Scale Gas Turbine : A Comparison of Different Solutions," *Appl. Energy*, vol. 106, pp. 355–364, 2013.
- [47] S. Masheiti, B. Agnew, and S. Walker, "An Evaluation of R134a and R245fa as the Working Fluid in an Organic Rankine Cycle Energized from a Low Temperature Geothermal Energy Source," *J. Energy Power Eng.*, vol. 5, pp. 392–402, 2011.
- [48] EIA, "Emission of Greenhouse Gases - Other Gases," 2003. [Online]. Available: http://www.eia.gov/oiaf/1605/archive/gg03rpt/other_gases.html.

- [49] A. Kalina, "Low Temperature Geothermal System," US 6820421 B2, 2004.
- [50] H. A. Mlcak, "An Introduction to the Kalina Cycle," in *Reprint from: Proceedings of the International Joint Power Generation Conference*, 2004, pp. 1–11.
- [51] O. Bolland, M. Forde, and B. Hande, "Air Bottoming Cycle: Use of Gas Turbine Waste Heat for Power Generation," *J. Eng. Gas Turbines Power*, vol. 118, pp. 359–368, 1996.
- [52] E. G. Feher, "The Supercritical Thermodynamic Power Cycle," *Energy Convers.*, vol. 8, pp. 85–90, 1968.
- [53] G. Angelino, "Carbon Dioxide Condensation Cycles For Power Production," *J. Eng. Power*, no. July, pp. 287–295, 1968.
- [54] G. Angelino, "ASME 69-GT-102: Real Gas Effects in Carbon Dioxide Cycles," in *International Gas Turbine Conference & Products Show*, 1969, pp. 1–12.
- [55] V. Dostál, "A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors," PhD Thesis, 2004, DOI: MIT-ANP-TR-100.
- [56] M. Persichilli, A. Kacludis, E. Zdankiewicz, *et al.*, "Supercritical CO₂ Power Cycle Developments and Commercialization : Why sCO₂ Can Displace Steam," p. 16, 2012.
- [57] Y. A. Gengel and M. A. Boles, *Thermodynamics - An Engineering Approach*, 5th Editio., no. 5th. .
- [58] H. Frutschi, *Closed Cycle Gas Turbines*, 1st ed. Wiley, 2005.
- [59] D. Y. Peng and D. B. Robinson, "A New Two-Constant Equation of State," *Ind. Eng. Chem. Fundam.*, vol. 15, no. 1, pp. 59–64, 1976, DOI: 10.1021/i160057a011.
- [60] R. Span and W. Wagner, "A New Equation of State for Carbon Dioxide Covering the Fluid Region from the Triple-Point Temperature to 1100K at Pressures up to 800 MPa," *J. Phys. Chem. Ref. Data*, vol. 25, no. 6, 1996.

- [61] M. Belmonte, "Development of a Performance Simulation Model of a Turbofan Engine Integrated With a Supercritical CO₂ Bottoming Cycle," MSc Thesis, Cranfield University, 2015.
- [62] DoE, "Quadrennial Technology Review 2015, Chapter 4: Advancing Clean Electric Power Technologies - Supercritical Carbon Dioxide Brayton Cycle," 2015.
- [63] M. L. Bauer, R. Vijaykumar, M. Lausten, *et al.*, "Pathways to Cost Competitive Concentrated Solar Power Incorporating Supercritical Carbon Sioxide Power Cycles," in *The 5th International Symposium - Supercritical CO₂ Power Cycles*, 2016, pp. 1–22.
- [64] T. J. Held, "Supercritical CO₂ Cycles for Gas Turbine Combined Cycle Power Plants," in *Power Gen International*, 2015, pp. 1–20.
- [65] K. Brun, P. Friedman, and R. Dennis, *Fundamentals and Applications of Supercritical Carbon Dioxide (sCO₂) Based Power Cycles*. Duxford: Woodhead Publishing - Elsevier, 2017.
- [66] G. D. Brighenti, "Performance and Assessment of S-CO₂ Bottoming Cycles for GT-based Power Plants for Commercial Marine Applications," MSc Thesis, Cranfield University, 2014.
- [67] F. Crespi, G. Gavagnin, D. Sanchez, *et al.*, "Supercritical Carbon Dioxide Cycles for Power Generation: A Review," *Appl. Energy*, vol. 195, pp. 152–183, 2017, DOI: 10.1016/j.apenergy.2017.02.048.
- [68] S. A. Wright, R. F. Radel, M. E. Vernon, *et al.*, "Operation and Analysis of a Supercritical CO₂ Brayton Cycle," Albuquerque, 2010.
- [69] E. M. Clementoni, T. L. Cox, and M. A. King, "Off-Nominal Component Performance in a Supercritical Carbon Dioxide Brayton Cycle," pp. 1–9, 2015, DOI: 10.1115/1.4031182.

- [70] E. M. Clementoni, M. A. King, and T. L. Cox, "Initial Transient Power Operation of a Supercritical Carbon Dioxide Brayton Cycle with Thermal-hydraulic Control," 2016, pp. 1–10.
- [71] J. Wilkes, T. Allison, J. Schmitt, *et al.*, "Application of an Integrally Geared Compressor to a sCO₂ Recompression Brayton Cycle," in *The 5th International Symposium - Supercritical CO₂ Power Cycles*, 2016, pp. 1–24.
- [72] J. E. Cha, S. W. Bae, J. Lee, *et al.*, "Operation Results of a Closed Supercritical CO₂ Simple Brayton Cycle," in *The 5th International Supercritical CO₂ Power Cycles Symposium*, 2016, pp. 1–10.
- [73] DoE, "SunShot Program Website," 2013. [Online]. Available: <https://www.energy.gov/eere/solar/concentrating-solar-power-sunshot-research-and-development>. [Accessed: 10-Apr-2016].
- [74] A. Moisseytsev and J. J. Sienicki, "Simulation of S-CO₂ Integrated System Test With Anl Plant Dynamics Code," in *The 5th International Symposium - Supercritical CO₂ Power Cycles*, 2016, pp. 1–19.
- [75] C. Lettieri, D. Yang, and Z. Spakovszky, "An Investigation of Condensation Effects in Supercritical Carbon Dioxide Compressors," *J. Eng. Gas Turbines Power*, vol. 137, pp. 1–8, 2015, DOI: 10.1115/1.4029577.
- [76] F. Jacob, "Feasibility Study of Supercritical CO₂ Bottoming Cycles for Aero Application," MSc Thesis, Cranfield University, 2015.
- [77] D. Sánchez, R. Chacartegui, J. M. Muñoz De Escalona, *et al.*, "Performance Analysis of a MCFC & Supercritical Carbon Dioxide Hybrid Cycle Under Part Load Operation," *Int. J. Hydrogen Energy*, vol. 36, no. 16, pp. 10327–10336, 2011, DOI: 10.1016/j.ijhydene.2010.09.072.
- [78] W. M. Kays and A. L. London, *Compact Heat Exchangers*, 3rd ed. New York: McGraw-Hill, 1984.

- [79] L. Chordia, "Thar Energy Manufacturer of Heat Exchangers for sCO₂ Power Cycles," in *University Turbine Systems Research Workshop*, 2015, p. 25.
- [80] S. F. J., "Finned Elliptical Tubes and Their Application in Air-Cooled Heat Exchangers," *J. Eng. Ind.*, vol. 88, no. 2, pp. 179–190, 1966.
- [81] A. D. Kraus, A. Aziz, and J. Welty, *Extended Surface Heat Transfer*, no. 1. New York: John Wiley & Sons, Inc., 2001.
- [82] J. E. Hesselgreaves, *Compact Heat Exchangers*, no. January. Elsevier Science & Technology Books, 2001.
- [83] Y. Ito, T. Goto, and T. Nagasaki, "Effect of Airflow on Heat Transfer of Air-to-Refrigerant Airfoil Heat Exchanger," in *53rd AIAA Aerospace Sciences Meeting*, 2015, no. January, pp. 1–11.
- [84] B. R. Kramer, B. C. Smith, J. P. Heid, *et al.*, "Drag Reduction Experiments Using Boundary Layer Heating," in *37th AIAA Aerospace Sciences Meeting & Exhibit*, 1999.
- [85] R. Camilleri, S. Ogaji, and P. Pilidis, "Applying Heat Pipes to a Novel Concept Aero Engine PART 1 – Design of a Heat-Pipe Heat Exchanger for an Intercooled Aero Engine," *Aeronaut. J.*, vol. 115, no. 1169, pp. 393–402, 2011.
- [86] R. Camilleri, S. Ogaji, and P. Pilidis, "Applying Heat Pipes to a Novel Concept Aero Engine PART 1 – Design of a Heat-Pipe Heat Exchanger for an Intercooled-Recuperated Aero Engine," *Aeronaut. J.*, vol. 115, no. 1169, pp. 403–410, 2011.
- [87] L. Xu and T. Grönstedt, "Design and Analysis of an Intercooled Turbofan Engine," *J. Eng. Gas Turbines Power*, vol. 132, pp. 1–4, 2010, DOI: 10.1115/1.4000857.

- [88] X. Zhao and T. Gronstedt, "Conceptual Design of a Two-pass Cross-flow Aeroengine Intercooler," *Proc. Inst. Mech. Eng. Part G J. Aerosp. Eng.*, vol. 229, no. 11, pp. 2006–2023, 2015, DOI: 10.1177/0954410014563587.
- [89] G. O. Musgrove, D. Shiferaw, S. Sullivan, *et al.*, "Tutorial : Heat Exchangers for Supercritical CO₂ Power Cycle Applications," in *The 5th International Symposium - Supercritical CO₂ Power Cycles*, 2016, pp. 82–91.
- [90] Heatric, "Heatric - Company Website," 2017. [Online]. Available: www.heatric.com. [Accessed: 07-Sep-2017].
- [91] P. Zhu, "Feasibility Study of Cryogenic Pre-coolers for Supercritical CO₂ Bottoming Cycles for Aero Applications," MSc Thesis, Cranfield University, 2017.
- [92] H. Grieb and W. Schlosser, "Profiled Tube Having an Elliptical or Lancet-Shaped Cross-Section, for Use in Tubular Heat Exchangers, and Method for its Manufacture," EP0239814 (B1), 1993.
- [93] M. Kormann and R. Schaber, "GT2014-26387: An Intercooled Recuperative Aero Engine for Regional Jets," in *Proceedings of ASME Turbo Expo 2014: Turbine Technical Conference and Exposition*, 2014, pp. 1–11.
- [94] S. Donnerhack, "NEWAC Sub-Programme 2 Public Workshop - Intercooled Recuperative Engine," London, UK, 2010.
- [95] X. Zhao, "On Aero Engine Intercooling," Degree of Licentiate of Engineering, Chalmers University of Technology, 2013.
- [96] Aviamarket, "Aviamarket - Lyulka Al-31 Turbofan Engine Family," 2016. [Online]. Available: <http://www.aviamarket.org/reviews/engines/428-al-31f-turbofan-engine-family.html>. [Accessed: 25-Nov-2016].
- [97] Z. Kussior, "Leteckemotory - Al-31," 2008. [Online]. Available: <http://www.leteckemotory.cz/motory/al-31/index.php?en>. [Accessed: 25-Nov-2016].

- [98] R. S. Matos, J. V. C. Vargas, T. A. Laursen, *et al.*, “Optimization Study and Heat Transfer Comparison of Staggered Circular and Elliptic Tubes in Forced Convection,” *Int. J. Heat Mass Transf.*, vol. 44, pp. 3953–3961, 2001.
- [99] H. Schoenenborn and E. Ebert, “Thermomechanical Design of a Heat Exchanger for a Recuperative Aeroengine,” vol. 128, no. October, pp. 736–744, 2006, DOI: 10.1115/1.1850510.
- [100] J. Rengel, “Structural Analysis of an Involute Spiral Heat Exchanger for S-CO₂ Bottoming Cycles,” MSc Thesis, Cranfield University, 2017.
- [101] J. Rengel, F. Jacob, A. Rolt, *et al.*, “GT2018-76208: Structural Analysis of Profiled Tubes for a Turbofan Engine Supercritical-CO₂ Bottoming Cycle Heat Exchanger,” in *Proceedings of ASME Turbo Expo 2018: Turbine Technical Conference and Exposition*, 2018, pp. 1–8.
- [102] F. Jacob, A. M. Rolt, J. M. Sebastiampillai, *et al.*, “Performance of a Supercritical CO₂ Bottoming Cycle for Aero Applications,” *MDPI Appl. Sci. Spec. Issue Gas Turbines Propuls. Power*, vol. 7, no. 3, 2017, DOI: 10.3390/app7030255.
- [103] R. Varvill, “Heat Exchanger Development at Reaction Engines Ltd .,” *Acta Astronaut.*, vol. 66, no. 9–10, pp. 1468–1474, 2010, DOI: 10.1016/j.actaastro.2009.11.010.
- [104] F. M. White, *Fluid Mechanics*, 7th ed. New York: McGraw-Hill, 2009.
- [105] C. A’Barrow, “Aerodynamic Design of the Coolant Delivery System for an Intercooled Aero Gas Turbine Engine,” PhD Thesis, Loughborough University, 2013.
- [106] Sandmeyersteel, “Specification Sheet : Alloy 347 / 347H.” pp. 1–2, 2014.
- [107] Haynes® International Inc., “HAYNES ® 282 ® alloy, Datasheet.” pp. 1–25, 2017.

- [108] M. S. Walker, A. M. Kruizenga, E. A. Withey, *et al.*, “Short Duration Corrosion Performance of Carbon Steels in S-CO₂ at 260 C,” in *The 5th International Symposium - Supercritical CO₂ Power Cycles*, 2016, pp. 1–19.
- [109] M. Kapoor, Ö. N. Doğan, K. Rozman, *et al.*, “Diffusion Bonding of H230 Ni-Superalloy for Application in Microchannel Heat Exchangers,” in *The 5th International Supercritical CO₂ Power Cycles Symposium*, 2016, pp. 1–21.
- [110] C. Lewinsohn, J. Fellow, N. Sullivan, *et al.*, “Ceramic , Microchannel Heat Exchangers for Supercritical Carbon Dioxide Power Cycles Introduction & Overview,” in *The 5th International Symposium - Supercritical CO₂ Power Cycles*, 2016, pp. 1–22.
- [111] L. Fine Tubes, “High Strength Tubing Solutions for Critical Aerospace Applications,” 2016.
- [112] N. D. Baltadjiev, C. Lettieri, and Z. S. Spakovszky, “GT2014-26180: An Investigation of Real Gas Effects in Supercritical CO₂ Centrifugal Compressors,” in *Journal of Turbomachinery*, 2015, pp. 1–13.
- [113] B. Monge, “Design of Supercritical Carbon Dioxide Centrifugal Compressors,” PhD Thesis, Univeristy of Seville, 2014.
- [114] J. Qi, T. Reddell, K. Qin, *et al.*, “Supercritical CO₂ Radial Turbine Design Performance as a Function of Turbine Size Parameters,” *J. Turbomach.*, vol. 139, no. August, pp. 1–11, 2017, DOI: 10.1115/1.4035920.
- [115] L. Moroz, B. Frolov, M. Burlaka, *et al.*, “GT2014-25385: Turbomachinery Flowpath Design and Performance Analysis for Supercritical CO₂,” in *Proceedings of ASME Turbo Expo 2014: Turbine Technical Conference and Exposition*, 2014, pp. 1–8.
- [116] Concepts_NREC, “Preliminary Design of Multistage Axial Compressors and Turbines,” 2018.

- [117] A. Sherbina, I. Klimov, and L. Moroz, "AIAA 2017-4730: Improved Model for Meanline Analysis of Centrifugal Compressors with a Large Tip Clearance," in *52rd AIAA/SAE/ASEE Joint Propulsion Conference*, 2017, pp. 1–10.
- [118] T. Conboy, "An Approach to Turbomachinery for Supercritical Brayton Space Power Cycles," in *Nuclear and Emerging Technologies for Space, NETS 2013*, 2012, pp. 1–8.
- [119] J. S. Noall and J. J. Pasch, "Achievable Efficiency and Stability of Supercritical CO₂ Compression Systems Main Compressor Design Discussion," in *The 4th International Symposium - Supercritical CO₂ Power Cycles September 2014*, 2014, pp. 1–10.
- [120] O. Canovas Ahedo, "Investigation and Design of Power Transmission Systems for Novel Propulsion Technologies," MSc Thesis, Cranfield University, 2016.
- [121] S. Donnerhack, "Intercooler and Recuperated Engines - LEMCOTEC Workshop for the Aeronautics Industry," Potsdam, 2016.
- [122] X. Zhao, "Aero Engine Intercooling Conceptual Design and Experimental Validation of an Aero Engine Intercooler," PhD Thesis, Chalmers University, 2016.
- [123] V. Sethi, "Advanced Performance Simulation of Gas Turbine Components and Fluid Thermodynamic Properties," PhD Thesis, Cranfield University, 2008.
- [124] P.-W. Kwan, D. R. H. Gillespie, R. D. Stieger, *et al.*, "GT2011-45814: Minimising Loss in a Heat Exchanger Installation for an Intercooled Turbofan Engine," in *Proceedings of the ASME Turbo Expo*, 2011, vol. 1, pp. 1–12.
- [125] S. E. Haaland, "Simple and Explicit Formulas For the Friction Factor in Turbulent Pipe Flow," *J. Fluids Eng.*, no. 105, pp. 89–90, 1983.

- [126] A. Rolt, "Personal Correspondence with Andrew Rolt." Cranfield.
- [127] Rolls-Royce, "Personal Correspondence with Rolls-Royce." Derby.
- [128] C. Xisto and T. Gronstedt, "ULTIMATE Confidential Report M1.1 – Intercooler and Pulse Detonation Technology Simulators," Gothenburg, 2015.
- [129] C. F. Colebrook, "Turbulent Flow in Pipes, with Particular Reference to the Transition Region between the Smooth and Rough Pipe Laws," *J. Inst. Civ. Eng.*, vol. 11, no. February, pp. 133–156, 1939.
- [130] D. Shiferaw, J. M. Carrero, and R. Le Pierres, "Economic Analysis of SCO₂ Cycles with PCHE Recuperator Design Optimisation," in *The 5th International Symposium - Supercritical CO₂ Power Cycles*, 2016, pp. 1–13.
- [131] R. Le Pierres, D. Southall, and S. Osborne, "Impact of Mechanical Design Issues on Printed Circuit Heat Exchangers," in *2nd International S-CO₂ Power Cycles Symposium*, 2011, p. 7.
- [132] M. Carlson, A. Kruizenga, M. Anderson, *et al.*, "Measurements of Heat Transfer and Pressure Drop Characteristics of Supercritical Carbon Dioxide Flowing in Zig-Zag Printed Circuit Heat Exchanger Channels," in *International Supercritical CO₂ Power Cycles Symposium*, 2011, pp. 1–10.
- [133] X. Zhang, X. Sun, R. N. Christensen, *et al.*, "Optimization of S-Shaped Fin Channels in a Printed Circuit Heat Exchanger for Supercritical CO₂ Test Loop," in *International Supercritical CO₂ Power Cycles Symposium*, 2016, pp. 1–14.
- [134] W.-Y. Woo, "Feasibility Study Of Cryogenic Pre-Coolers For Bottoming Cycles," MSc Thesis, Cranfield University, 2016.
- [135] O. Cakir, "Study of Etch Rate and Surface Roughness in Chemical Etching of Stainless Steel," *Key Eng. Mater.*, vol. 364, pp. 837–842, 2008, DOI: 10.4028/www.scientific.net/KEM.364-366.837.

- [136] K. R. Anderson, T. J. Gross, and C. Mcnamara, "SCO₂ Heat Exchanger Analysis for a Waste-Heat Recovery Rankine Cycle," in *International Supercritical CO₂ Power Cycles Symposium*, 2016, pp. 1–20.
- [137] P. D. Friedman and M. P. Rapp, "Parametric Analysis of SCO₂ Power Cycles Operating at Moderate Temperatures," in *International Supercritical CO₂ Power Cycles Symposium*, 2016, pp. 1–21.
- [138] J. Pasch, T. Conboy, D. Fleming, *et al.*, "Supercritical CO₂ Recompression Brayton Cycle: Completed Assembly Description," 2012.
- [139] A. Rohatgi, "WebPlotDigitizer," 2015. [Online]. Available: <http://arohatgi.info/WebPlotDigitizer/>.
- [140] L. Moroz, M. Burlaka, O. Rudenko, *et al.*, "Evaluation of Gas Turbine Exhaust Heat Recovery Utilizing Composite Supercritical CO₂ Cycle," in *Proceedings of International Gas Turbine Congress 2015*, 2015, p. 7.
- [141] "Personal Correspondence with SunShot Program Partner." San Antonio, 2016.
- [142] A. Pellegrini, T. Nikolaidis, V. Pachidis, *et al.*, "On the Performance Simulation of Inter-Stage Turbine Reheat," *Appl. Therm. Eng.*, vol. 113, no. February, pp. 544–553, 2017, DOI: 10.1016/j.applthermaleng.2016.10.034.
- [143] DassaultSystemes, "Isight 5.9 user guide," 2014.
- [144] K. Kyprianidis and A. M. Rolt, "GT2014-26064: On the Optimisation of a Geared Fan Intercooled Core Engine Design," in *Proceedings of ASME Turbo Expo 2014: Turbine Technical Conference and Exposition*, 2014, pp. 1–13.
- [145] B. Lakshminarayana, "Methods of Predicting the Tip Clearance Effects in Axial Flow Turbomachinery," *J. Basic Eng.*, vol. 92, no. 3, pp. 467–480, 1969.

- [146] P. Lolis, "Development of a Preliminary Weight Estimation Method for Advanced Turbofan Engines," PhD Thesis, Cranfield University, 2014.
- [147] Softinway, "AxStream Software Suite," 2018. [Online]. Available: <http://www.softinway.com/en/software/>. [Accessed: 01-May-2018].
- [148] C. Xisto, F. Ali, O. Petit, *et al.*, "GT2017-63776: Analytical Model for the Performance Estimation of Pre-cooled Pulse Detonation Turbofan Engines," in *Proceedings of ASME Turbo Expo 2017: Turbine Technical Conference and Exposition*, 2017, pp. 1–14.
- [149] J. Whurr, "Otto Cycle Core Turbofan Concepts," MSc Thesis, Cranfield University, 1995.
- [150] B. C. Bobusch, P. Berndt, C. O. Paschereit, *et al.*, "Shockless Explosion Combustion - An Innovative Way of Efficient Constant Volume Combustion in Gas Turbines," *Combust. Sci. Technol.*, pp. 1–14, 2013, DOI: 10.1080/00102202.2014.935624.
- [151] NPHT, "Napier Nomad," 2018. [Online]. Available: <http://www.npht.org/nomad/4579733511>. [Accessed: 07-May-2018].
- [152] C. Xisto, O. Petit, T. Grönstedt, *et al.*, "The Efficiency of a Pulsed Detonation Combustor – Axial Turbine Integration," *Aerosp. Sci. Technol.*, pp. 1–13, 2018.
- [153] S. Kaiser, A. Seitz, S. Donnerhack, *et al.*, "Composite Cycle Engine Concept with Hectopressure Ratio," *J. Propuls. Power*, vol. 32, no. 6, pp. 1413–1421, 2016, DOI: 10.2514/1.B35976.
- [154] S. Kaiser, H. Kellermann, M. Nickl, *et al.*, "A Composite Cycle Engine Concept For the Year 2050," in *Proceedings of the International Council of The Aeronautical Sciences*, 2018, pp. 1–14.
- [155] J. Sebastianpillai, "Feasibility of Nutating Disk Engine Cores for Short Range Aero Engines," PhD Thesis, Cranfield University, 2018.

- [156] P. L. Meitner, M. Boruta, and J. Jerovsek, "NASA/TM - 2006-214342: The Nutating Engine - Prototype Engine Progress Report and Test Results," 2006.
- [157] G. Smith, P. Meitner, J. Jerovsek, *et al.*, "Meyer Nutating Disc Engine: A New Concept in Internal Combustion Engine Technology," *43rd AIAA/ASME/SAE/ASEE Jt. Propuls. Conf. Exhib.*, no. July, pp. 1–8, 2007, DOI: 10.2514/6.2007-5122.
- [158] J. Sebastiampillai, A. Rolt, F. Jacob, *et al.*, "D2.4 - Operating Cost Model and Definition of Policy Scenarios," 2017.
- [159] Z. Han and A. Fina, "Thermal Conductivity of Carbon Nanotubes and Their Polymer Nanocomposites : A Review," *Prog. Polym. Sci.*, vol. 36, no. 7, pp. 914–944, 2011, DOI: 10.1016/j.progpolymsci.2010.11.004.
- [160] M. Beretens, "Working Title: Performance of a Triple Cycle Aero Engine," MSc Thesis, Cranfield University, 2018.
- [161] P. Murthy, B. Khandelwal, V. Sethi, *et al.*, "Hydrogen as a Fuel for Gas Turbine Engines with Novel Micromix Type Combustors," in *47th AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit*, 2011, no. August, pp. 1–7.
- [162] Y. Levy and T. Reichel, "Feasibility of the Hybrid Combustion Concept," 2014.
- [163] M. A. Lehar, "System and Method of Waste Heat Recovery," US 2014/0352306 (A1), 2014.
- [164] D. G. Laing and C. M. Booth, "Gas Turbine Engine with Integrated Bottoming Cycle System," US2014/0150443(A1), 2014.
- [165] C. U. Hardwicke, S. A. Leone, T. F. Taylor, *et al.*, "Combined Cycle Power Plant," US 8,408,003 B2, 2013.

- [166] S. W. Freund, R. Bodek, C. K. Bodek, *et al.*, "System and Method for using Gas Turbine Intercooler Heat in a Bottoming Steam Cycle," US 2012/0159923 (A1), 2012.
- [167] E. R. Earnest, "Combined Gas Turbine - Rankine Turbine Power Plant," US4,267,692, 1981.
- [168] I. Schnaid, "Gas Turbine with Bottoming Air Turbine Cycle," 1995.
- [169] A. Alexiou and T. Tsalavoutas, "Introduction to Gas Turbine Modelling with PROOSIS," 2011.

APPENDICES

Appendix A Performance data of LR TF2000

Parameter	Units	Cruise	TOC	EOR
Altitude	m	10668	10668	0
Mach number	-	0.82	0.82	0.25
Delta ISA	K	+0	+10	+15
F_N	kN	52.4	73.0	251.0
BPR	-	5.2	4.7	5.1
OPR	-	36.5	46.5	39.2
Fan PR	-	1.7	1.9	1.7
TET	K	1422	1691	1757
sF_n	m/s	146	191	267
SFC	g/kN/s	17.25	18.64	12.96
Fan Diameter	m		2.47	
Total Weight	kg		7927	

Appendix B Performance data of TF2050

Parameter	Units	Cruise	TOC	EOR
Altitude	m	11277.6	10668	0
Mach number	-	0.8	0.82	0.2
Delta ISA	K	+0	+10	+15
F_N	kN	32.6	50.0	183.5
BPR	-	16.8	16.1	16.1
OPR	-	62.1	75.4	60.1
Gearbox efficiency	-	0.995	0.995	0.995
Gearbox ratio		4.3	4.3	4.3
Fan PR	-	1.4	1.5	1.4
IPC PR	-	2.7	2.4	2.3
HPC PR	-	18.3	23.0	20.9
TET	K	1540	1890	1921
sF_n	m/s	85	111	183
SFC	g/kN/s	12.6	13.7	8.3
ΔFB_{2000}	%		-45.8	
Fan Diameter	m		2.84	
Total Weight	kg		5161	

Appendix C Variable Area Nozzles

A variable area nozzle (VAN) at the back of the pre-cooler bypass duct may be used to improve mission performance and control the off-design behaviour of the engine. On an intercooled engine, regulating the coolant mass flow has been shown to help improve the mission fuel burn benefit [28]. Intercooling enables significant SFC reductions at hot conditions such as climb and take-off due to the higher pressure ratios possible for a given compressor delivery temperature and the reduced compressor work. However, once the aircraft approaches cruise altitude those benefits are diminished. The core mass flow may be overcooled, reducing the compressor air temperature and increasing fuel burn at a given thrust. Additionally, the pressure losses in the diffuser and across the heat exchanger matrix may rise unnecessarily, increasing the SFC further. A variable area nozzle at the back of the intercooler duct can be used to control the pre-diffusion levels in the diffuser and the amount of heat transferred in the intercooler. The introduction of variable channel geometry that allows the core mass flow to bypass the intercooler completely has also been shown to enable further fuel burn improvements of up to 0.5 percentage points [29] by reducing the hot flow pressure losses. However, such an arrangement may be too complex for a relatively modest improvement.

The combined cycle engine is conceptualised to improve mission fuel burn primarily via SFC reductions at cruise. Hence, the heat exchangers and the respective ducts and nozzles are designed for cruise operation herein and the regulation of the coolant flow would therefore not be necessary. Nonetheless, variable area nozzles may be introduced on a combined cycle engine to give the engine control system a further handle to control the off-design behaviour of the bottoming cycle as discussed in section 2.4.3. The VAN will require hydraulic or electrical systems, possibly a motor and the variable geometry itself, which constitutes a significant weight increase for the engine. Hence, the SFC reductions due to the VAN have to be significant to offset the weight-induced fuel burn penalties. A VAN model was created, but ultimately not used in the present work, because the engine's off-design performance estimation had to be simplified as described in section 3.7.3.

Appendix D Brief Patent Review

Ever since the start of the new millennium, S-CO₂ power cycles have been the source of numerous new research efforts. However, the results of those research programmes, at least those available in the public domain, primarily focus on stationary power generation or marine applications. While it is unlikely that large Original Equipment Manufacturers (OEM) such as GE, Rolls-Royce or Pratt & Whitney (United Technologies) have not yet at least briefly considered S-CO₂ cycles for aero engines, there are no publications publicly available, be it in form of research papers, PhD Theses' or patents. Hence, it is unlikely that any existing patents specific to S-CO₂ bottoming cycles for aero application might be infringed upon by this study. However, a patent review is still necessary to ensure that no relevant developments for other applications would be passed over.

While there are countless patents related to steam cycles, only two patents were found specifically mentioning the use of S-CO₂ as the working fluid of a bottoming cycle. Both patents are held by General Electric Company (US), giving additional credence to the assumption that GE may already have considered S-CO₂ bottoming cycled aero engines. US2014/0352306 A1 [163] from Matthew Alexander Lehar called 'System and Method of waste heat recovery', specifies an S-CO₂ bottoming cycle at system level. However, there are a number of differences between the technology Lehar discusses and what this project entails. First of all, Lehar's S-CO₂ cycle is a recompressed cycle, a configuration that, while more efficient than the simple and recuperated cycles under investigation in the current work, is presently deemed too complex for aero engines. The second turbine component, not to mention the numerous additional S-CO₂ ducts for alternate flow paths, not only increase system weight, but also require considerably more complex control systems and redundancies. Additionally, the patented cycle is a pure electric power generation cycle for stationary applications, as specified multiple times through the mention of the generator components. The direct drive of another power system, let alone an aero engine as currently targeted in this project, is not envisioned. Should the bottoming cycle be used to generate and store energy in batteries on the aircraft however, this distinction may be disputed. The biggest difference between Lehar's and the current work however are the overall cycle parameters. It is already obvious from the main depiction on the cover page of the patent, that the cycle does not feature an S-CO₂ compressor, as no such component is included in the schematics. This is due to Lehar's cycle being a Rankine cycle as in the study of De Servi et al. [11], rather than a Brayton Cycle as in the current investigation. Hence, the cycle does not need an

S-CO₂ compressor, but features S-CO₂ pumps. Finally, while Lehar originally mentioned the design to be for an S-CO₂ cycle, he also outlines the prospect of facilitating other working fluids. In summary, although certain overlaps with Lehar's work are indisputable, the application, configuration and overall cycle parameters differ considerably from this project. The originality of the present work therefore cannot be denied based on Lehar's work.

Donald Gordon Laing and Charles Michael Booth are the registered inventors behind US 2014/0150443 A1 'Gas Turbine Engine with integrated Bottoming Cycle System' [164]. In it, Laing and Booth investigate not the standalone design of a bottoming cycle featuring S-CO₂, but a combined cycle system. The combined cycle approach increases the similarities between this investigation and their research compared to the work of Lehar, but they are again referring to stationary power applications exclusively. Laing and Booth also envision a cycle used exclusively to drive one or two generators, depending on whether or not the main engine and the bottoming cycle drive the same generator. The patented cycle also features numerous additional heating and cooling cycles between the bottoming cycle and the main engine in order to boost overall performance. While some of them may be worth further investigation for aero application in the future, most would introduce too much system complexity and weight to be feasible without considerable performance penalties. The patented concept also mentions the application of a second burner in the main engine's exhaust gases just upstream of the main heat exchanger. While this would help with start-up procedures and enhanced power delivery from the bottoming cycle, this system would not be reasonably applicable in an aero engine, as it would increase the core engine waste heat rather than lower it. The mention of such a system in the patent does however give credibility to the concept idea of employing inter-turbine reheat on the CCTF2050_ITR. Finally, Laing and Booth specifically restricted their invention to combined cycle engines featuring GE main engines. As those will not be part of the on-going research, no source of conflict can be identified. This patent, although source of many design ideas worth further investigation, ultimately does not take anything away from this work, as it entailed not only different main engines and more complex systems, but most importantly was created for a different application.

Both patents were filed in 2014, as can be seen from their patent numbers. This is another indication of the contemporary character of S-CO₂ cycle research. Even if it were not for all the differences between the patented technologies and those under

investigation in this research, the patents are held by GE and little additional data is shared in form of further publications. Hence, creating the simulation tools and monitoring experience gained with S-CO₂ component design within this project, still represents a genuine contribution to knowledge.

Another source of input with regard to research on S-CO₂ cycles for aero application is closed- and open-cycle bottoming cycle research featuring other working fluids. Steam, air and organic fluids (e.g. R245fa) are the most prominent and most widely spread alternative working fluids in power generation cycles as discussed in section 2.3. The extrapolation and application of the experiences gained from those cycles, especially the most widely used steam Rankine Cycle, proved to be a dependable way to initiate the present S-CO₂ research.

CN101737107 (A) of Hardwicke et.al. [165] and US2012/0159923 A1 of Freund et.al. [166] are examples of the great width of cycle or cycle component inventions featuring steam as their working fluid. While Freund et.al. focus on the systematic influence and design of a single component, an intercooler for the main engine that rejects its heat not to the environment but to a bottoming cycle, Hardwicke et.al. focus on wider overall cycle concepts. Most studies offer potential technology ideas that may be applicable in this research, but the minute depiction of these details would exceed the purpose of this section. Should the concept be developed further, a re-evaluation of these issues would be necessary. Considerable efforts have also been undertaken in the conceptualisation and design of cycles using organic refrigerants (e.g. US4267692 (A) [167]) or air (e.g. US5442904 (A) [168]) as the bottoming cycles working fluid. A summary of the main differences between a representative selection of the reviewed patents and the current research is given in Table 2-3.

On a component level, only EP0239814 [92] of MTU Aero engines may be of relevance at this time. As discussed before, MTU patented the design and manufacturing of elliptical heat exchanger tubes from sheet metal with an internal support structure. The heat exchanger tubes of the pre-cooler and main heat exchanger as presented in section 2.5.2.4 may be similar to those designs and the overlap with MTU's patent may depend on the details of the internal support structure and the manufacturing technique.

Appendix E Modelling in PROOSIS

E.1 General Approach

All engine performance and preliminary design methods presented in this chapter are implemented in the commercially available “Propulsion Object Oriented Simulation Software” (PROOSIS 3.6.14, Empresarios Agrupados Internacional) unless specifically mentioned otherwise.

The complexity of modelling in PROOSIS is due to its multi-layered modelling approach. A brief description of these complexities is necessary to outline the benefits and potential pitfalls of using the software. Figure E-1 illustrates the multi-layered modelling approach of PROOSIS. The components are coded in the PROOSIS-specific EL programming language. EL is an object-oriented programming language and therefore benefits from capabilities such as inheritance, polymorphism and encapsulation [169]. Hence, only new methods need to be coded in new component models, as the existing capabilities can be called upon via inheritance trees. The interdependencies of the different component codes or their organisation in libraries is not shown in the figure.

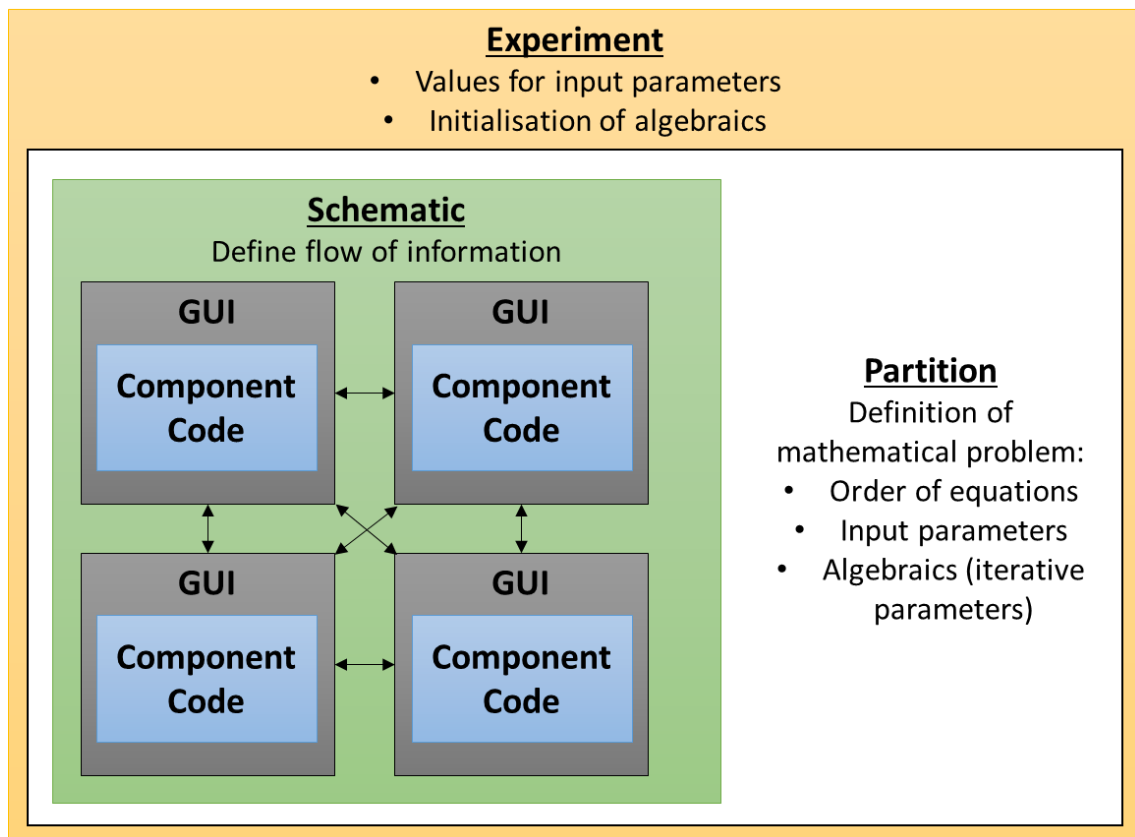


Figure E-1: Schematic structure of typical PROOSIS models

Perhaps the most important feature of EL is that it is non-sequential unless it is explicitly forced to follow a certain code sequence by the user. This means, that the order in which equations are inserted is irrelevant and will most probably not be the order in which they will be solved during the simulations. After the code is finished, a symbol is generated to be used in the graphical user interface (GUI) of PROOSIS. A whole engine can then be assembled by importing those symbols into one schematic and connecting the input and output ports of the individual components with each other as appropriate. The schematic therefore defines the flow of information between all the components and combines the source code of all components in one. Creating a partition is the next step when modelling in PROOSIS. The partition is where the user defines which of the model's parameters are used as inputs. Ideally, the user can choose any combination of component parameters that allows PROOSIS to organise the order of all the source code in the schematic.

Not every combination of input parameters will be a sensible choice as a design input in the end, not only because of their physical meaning, but because choosing them might create an extremely complex iterative mathematical problem. The partition is also where the user defines which of the parameters that are not used as inputs are iterated upon during the simulations. These iterative parameters are called algebraics in PROOSIS and the stability of the engine model is extremely sensitive to the choice of algebraics. The final modelling layer in PROOSIS is the experiment. Here, the user imposes the actual values for the chosen boundary conditions and, just as importantly, the user has to provide the model with an initial guess of all the algebraics. The more algebraics, and hence part-model iterations, are necessary to calculate the whole engine performance, the more important it gets that the initial guesses are not only reasonably close to the final results, but that they stand in a reasonable relation to the other algebraics as well. For example, the S-CO₂ fluid temperature in the first pass of the main heat exchanger should be initialised with a lower value than the second pass. The initialisation of the algebraics is not an exact science by any measure and sometimes a model takes longer or is completely unable to converge, when the initial guess is too close to the solution.

Ultimately, PROOSIS is a great simulation platform when exploring a novel engine configuration such as this, because it is not only relatively easy to code in, but it will allow for different kinds of studies to be performed with one and the same code. Possible model instabilities due to the algebraics however present potential pitfalls for the models.

E.2 Fluid property models

The fluid property model was implemented in three stages. First, look-up-tables are created that list all necessary fluid properties such as specific heat capacity, enthalpy, Prandtl number and more for a discrete set of input variables. The variables are the fluid temperature, fuel-to-air ratio (FAR) and water-to-air ratio (WAR). Secondly, interpolation routines were created to enable the homogenous depiction of the fluid properties across the whole design range. The interpolation routines are used in what are called fluid functions in PROOSIS, to calculate the exact value of a given fluid property for any given flow condition. At the start of every PROOSIS calculation, the fluid property tables are loaded once and the fluid functions are called as necessary. The third step in the implementation of fluid properties in PROOSIS is the creation of what are called therm functions in PROOSIS. Therm functions allow for total-to-static property conversions for differing sets of input parameters based primarily on the corrected mass flow equation. Different functions needed to be implemented for example depending on whether the flow area or the Mach number is known for given combination of total pressure, temperature and mass flow. All functions are solved sequentially in PROOSIS, which is why multiple implementations of the same thermodynamic equations are necessary. It would be possible to implement total-to-static property conversions directly into a component code as well, eliminating the need for multiple implementations of the same physical phenomenon depending on what inputs are used. However, this would not only make component codes less clear, it would also significantly reduce model stability.

Appendix F Work around the PhD

This dissertation was aided by a number of MSc students through their individual research projects, during which the author of this work continuously acted as a technical advisor:

Olatz Canovas Ahedo was the first of these students focusing on the “Investigation and Design of Power Transmission Systems for Novel Propulsion Technologies”. Unfortunately, the model she developed for the preliminary design of such power transmission systems could not be used in this investigation, as it became apparent that a purely mechanical system with a fixed gear ratio is most likely not feasible.

Jon Rengel Jimenez investigated the mechanical integrity of double-circular arc heat exchanger tubes for the MHX. His results have been used extensively during this investigation.

Pengju Zhu conducted a “Feasibility Study of Cryogenic Pre-coolers for Supercritical CO₂ Bottoming Cycles for Aero Applications”. The initial performance and preliminary design models for the cryogenic pre-cooler of the triple cycle concept he developed, were later picked up and extended by Mikel Beretens Izquierdo. Mikel has since developed a more intricate component design that might improve the performance of the triple cycle concept. The investigation is expected to conclude in August 2018.

Finally, Giuseppe D’Antino is in the process of investigating the aerodynamic implications of changing from an elliptical to a double-circular arc design for the tubes of the MHX and the pre-cooler. His work is also expected to conclude in August 2018.

The author of this work also took on significant tasks and responsibilities in the ULTIMATE project:

- Modelling of the LRTF2000 and the TF2050 engines
- Integrating of the engine and aircraft performance, noise, emissions and direct operating cost tools in the TERA2050 multi-disciplinary evaluation platform
- Writing of multiple project deliverables
- Leading all partner’s research efforts in WP2 of the ULTIMATE project

Finally, collaborations with project partners and colleagues at Cranfield University have led to a number of joint publications.

Appendix G PROOSIS S-CO₂ Turbine map

