CRANFIELD UNIVERSITY

ÁLVARO PENA LÓPEZ

GAS TURBINE PERFORMANCE STUDIES FOR LH_2-FUELLED ENGINES

SCHOOL OF AEROSPACE, TRANSPORT AND MANUFACTURING
Thermal Power - Aerospace Propulsion

MSc
Academic Year: 2020–2021

Supervisor: Dr Theoklis Nikolaidis, Dr Soheil Jafari
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This thesis is submitted in partial fulfilment of the requirements for the degree of MSc.

Abstract

The aim of this Individual Research Project is to lay solid foundations for the investigation of liquid hydrogen utilisation in aero gas turbines employing Cranfield University's Turbomatch software.

To this day, including key components like heat exchangers is considerably limited in this software. As heat exchangers are believed to be vital in order to take advantage of the unique properties of liquid hydrogen, this thesis aims to create the possibility to carry out investigations in this field without depending on similar tools from outside Cranfield University.

To do so, a code is created in Matlab to serve as a heat exchanger library in which key parameters like outlet temperatures, overall pressure drops or geometrical characteristics are calculated in a split second. This code is validated, and results obtained from it are shown, compared and discussed.

Different engine configurations are modelled in Turbomatch (in which the location of the heat exchangers is varied from one model to the other). The outputs from the code are introduced in these Turbomatch models and simulations are run. Results from these simulations are plotted and analysed, highlighting the impact of these heat exchangers according to the location they are on.

Finally, some conclusions are extracted from all of the above and recommendations for future researches are listed.

Keywords

Liquid hydrogen; heat exchangers; cryogenic fuels; thermal management; aero gas turbines.

Acknowledgements

First of all, I would like to express my sincerest gratitude to Dr Theoklis Nikolaidis and Dr Soheil Jafari for the amazing job they have done as co-supervisors and for their flexibility, support and counselling throughout the entire academic year. This pandemic has not been easy for anybody, but their work has proven to be as professional as ever.

Secondly, I would like to wish the best of luck to my colleague Akshay Nag Srinath, who is developing his Individual Research Project in parallel to this one.

Last but not least, I deeply want to thank my parents, Josetxo and Mari Jose, my sister Marina and my friends Adrián, Arturo, Andrea and Juan for all the affection and support they have constantly shown me throughout this amazing journey.

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List of Abbreviations

LH_2	Liquid hydrogen
COT	Combustor outlet temperature
HEX	Heat exchanger
IRP	Individual Research Project
HPT	High pressure turbine
LPT	Low pressure turbine
LMTD	Log-mean temperature difference
NTU	Number of transfer units
OPR	Overall pressure ratio
SFC	Specific fuel consumption
A	Heat exchange area
b	Height of a channel
C	Specific heat ratio or selection
c	Width of a channel

c_p	Specific heat at constant pressure
D	Diameter of a shell
D_h	Hydraulic diameter
d	Diameter of a pipe
f	Friction factor
I	Thermal inertia
k	Thermal conductivity
L	Length of a plate, tube...
Nu	Nusselt number
N	Number of passes
N_b	Number of baffles
n	NTU per number of shell pass
Q	Heat
U	Overall heat transfer coefficient
$\dot{m}$	Mass-flow-rate
P	Pressure
Re	Reynolds number
T	Temperature
v	Velocity of the fluid

Greek symbols

ε	Effectiveness of a heat exchanger
ρ	Density

Indices

c	Cold flow
h	Hot flow (except in D_h)
drop	Pressure drop
e	Equivalent
global	Actual heat being transmitted in the system
i	Inner pipe
o	Outer pipe
in	Inlet condition
out	Outlet condition
max	Maximum among two or more values
min	Minimum among two or more values
s	Shell side
t	Tube side
1p	One-pass calculation for multiple-pass case

Chapter 1

Introduction

Aviation is always at the cutting edge of technology because of the role it plays in connecting people. ICAO [6], in its annual report, depicts the habits and practices of the society, revealing that, in 2019, about 8.7 billion passengers (an increase of 4.9 % from 2018) and 57.6 million tones of cargo (- 2.9 %, the first time this number decreases in almost a decade) were transported. This key sector is strictly regulated, and recently, important countries from all over the Globe have joined their forces to fight a common enemy: climate change. As a main character, aviation must join that fight by becoming greener in any possible aspect.

Liquid hydrogen (LH_2) is one of the options currently examined to substitute traditional aviation fuels. Moreover, its unique properties might be exploited to improve the engine cycle by implementing heat exchangers in strategic locations. In this project, a code has been created to model different types of heat exchangers, obtaining as outputs their geometric and thermodynamic characteristics; once the type of heat exchanger is chosen, the outputs are introduced in certain Turbomatch files to run some simulations and obtain as a result the impact of, firstly, using LH_2 as a fuel, and secondly, implementing heat exchangers to modify the cycle.

In Chapter 2, the literature review carried out for this thesis is briefly presented, discussing, among others, some of the concepts mentioned above these lines. In Chapter 3, a method to simulate heat exchangers is selected and thoroughly explained, together with the discussion of the strategy followed to calculate the pressure drop according to each type of heat exchanger; a flowchart of the code is included at the end of this Chapter. In Chapter 4, some results obtained from the code are plotted and analysed, and then the results obtained from Turbomatch simulations are shown and assessed. Finally, in Chapter 5, the conclusions extracted from the whole project are compiled and presented to the reader, followed by some recommendations for future researchers aiming to continue this project.

1.1 Objectives and problem statement

Cranfield University's Turbomatch software is designed to allow the user model and customise an engine stage by stage (or brick by brick, as it is called in its technical jargon). However, when trying to insert in the model heat exchangers that do not use the airflow inside the engine for both the cold and hot sides (that is to say, external fluids like fuel or oil are used), the programme crashes. Furthermore, there is no option to choose the type of heat exchanger that wants to be employed, its size either, etc.

This thesis pursues the following objectives to minimise, or completely avoid, the mentioned limitations:

- To create a code employing Matlab that must serve as a "heat exchanger library", from which the user can select the type of heat exchanger to be used and the whole code must adapt to that choice.
- To structure that code in such a way that the outputs it gives are both geometrical and thermodynamic parameters, i.e., solves *sizing* and *thermal* problems. More-

over, the pressure drop happening inside the engine must be an output of the code as well.

– To validate the code to assure as much accuracy and credibility as possible.

– To run Turbomatch simulations with heat exchangers incorporated to the model, and the parameters of those heat exchangers must be obtained from the outputs of the code. These simulations must be analysed and, if possible, validated. Effects of each location of the heat exchanger on the engine must be assessed.

Chapter 2

Literature Review

2.1 Introduction

Probably neither Wilbur nor Orville Wright could even envisage what the future held for their most famous invention. If the reader pauses for thought for a brief second and looks back over the past of aviation, a feeling of awe might bud in its body.

Figure 2.1: *Wright Flyer*'s first flight, with Orville as the pilot and Wilbur on the ground, observing. [8]

Everything started in December 1903, when the Wright brothers achieved the first successful engine-operated flight; such a moment was immortalised in Figure 2.1. That first flight has been followed by a whole aviation industry. The design, manufacturing and maintenance of the planes we know today is the result of a century of investigation and development.

Going through the steps of the history of aviation is beyond the scope of this project, but some interesting matters need to be discussed. Aviation has always been tightly linked to the society, adapting itself to trends, behaviours and different economical cycles (like during petroleum crisis or COVID global pandemic), with the ability to keep itself at the fore of technological advances. These state-of-the-art technologies have pushed the boundaries of aviation up to today's levels, specially referring to enhanced comfort[1] in civil aviation (because of better cabin pressurising, cabin light adjustments, noise reduction, etc.) and improved performance, both in civil and military aviation, among other aspects.

Figure 2.2: The A350 is one of Airbus's newest proposals for long-haul flights. [14]

[1]When comfort is mentioned, the design of cabin seat distribution is obviated, as these days priority is given to costs rather than luxury in many airlines.

Speaking of performance improvement, it is true that refined aerodynamics have definitely played a vital role in it, but it is also indisputable that engines are key to increase performance, no matter whether they are propellers, turbojets[2] or turbofans. To this day, gas turbines have experienced a more than significant enhancement in terms of power output while being more and more fuel-efficient and environmentally friendly. These characteristics come from better heat transfer comprehension (leading to better cooling techniques), new manufacturing techniques, i.e., more precision and ability to create more complex structures, the emergence of computational fluid dynamics (CFD hereafter), accurate empirical measurements e.g. for combustor design parameters, and so on. This evolution can be observed in Figure 2.3.

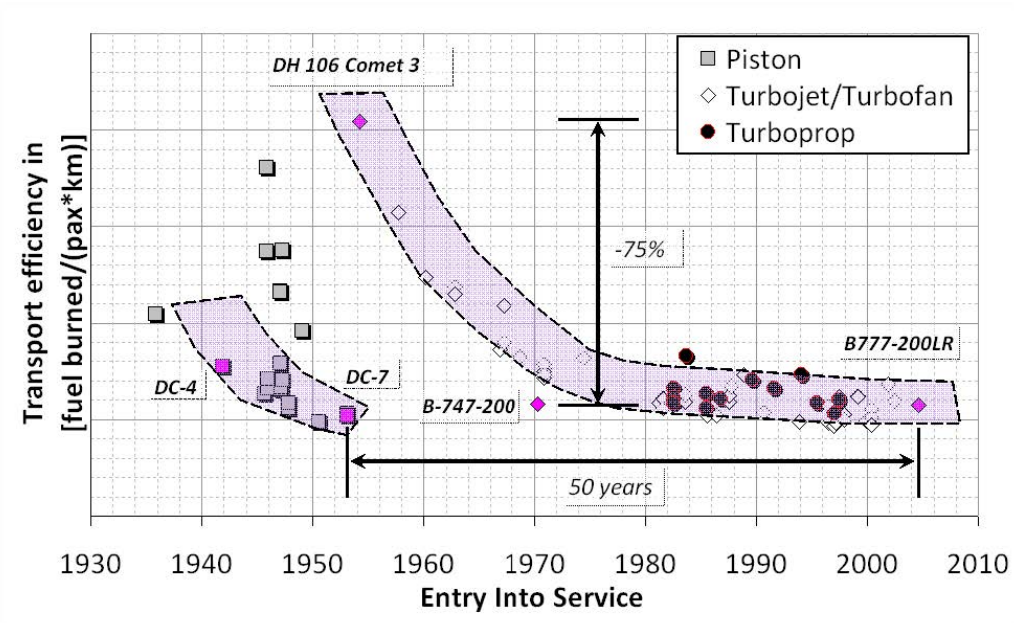


Figure 2.3: Evolution of aircraft and engine efficiency throughout the last decades. [30]

2.1.1 Turbofans: in pursuit of efficienct and green engines

As mentioned, it is not rare to find exhaustive cost monitoring, and fuel is one of the most expensive parts of aircraft functioning [30], [31]. To reduce fuel costs, among many

[2]When propellers or turbojets are used for military purposes, efficiency is left aside and performance plays the leading role.

different techniques, one can find:

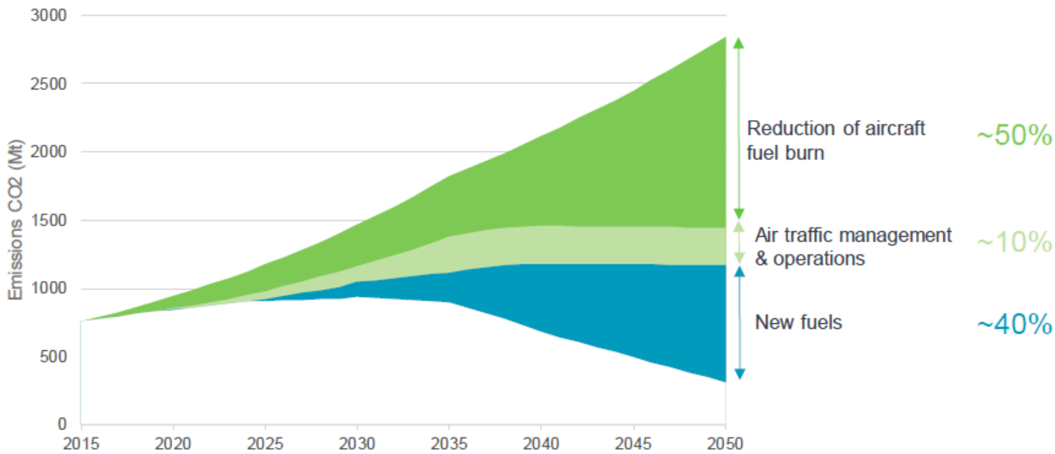


Figure 2.4: Air Transport Action Group's (ATAG) forecast to halve total CO_2 emissions (2005-2050 period), SRIA, EU. [29]

– Reducing the amount of fuel put in each airliner per flight.

– Employing new, greener fuels.

– Using more fuel-efficient airliners.

The first one could be risky, as low fuel could cause an emergency landing or worse. The second and third options are expensive, because they involve considerable research and development to, for example, increase aerodynamic efficiency by reducing the drag, or to increase engine efficiency. Second point is on what this Chapter (and, in general terms, the whole project) is going to focus hereafter.

Today's aircraft engines, including propellers but specially gas turbines, are visually different from the ones one could find twenty or thirty years ago. These days, turbofans are larger than ever, with nacelle diameters wide enough to accommodate considerably big intake fans. See Figure 2.5 for visual reference (timeline is exaggeratedly big to ease the understanding of size evolution).

(a)

(b)

Figure 2.5: Evolution of engine sizes: a) De Havilland Comet (1949) [18]; b) Boeing 777X (2019) [22].

This allows to exploit the capabilities of a good equilibrium in terms of performance between the amount of air passed converted into primary (or hot) flow, that is, the flow that goes into the different stages of the turbofan's core (compressor, combustion chamber, turbine and nozzle[3]) and secondary (or cold) flow. The parameter of control of this equilibrium is commonly known as *bypass ratio*. For greater bypass ratios, one needs, among other things, wider nacelles. As a curious fact, new GE9X engine's fan diameter is almost three and a half meters wide [5]. Another critical aspect of an aero gas turbine's performance is the pressure ratio, which has also been increased throughout the last decades. Explaining the rationale behind engine manufacturers' quest for bigger

[3]In turbofans, one can find nozzles in which hot and cold flows are mixed before exhausting them or others in which hot and cold flows still run separately, that is to say, there are two nozzles. [30]

engines goes beyond the scope of this Individual Research Project (IRP hereafter), but without getting into too much detail, it could be said that increasing the bypass ratio in between reasonable values increases performance and reduces fuel consumption; there are some values above which fuel consumption is jeopardised, hence increasing the ratio over those limits is unsubstantiated.

But, is the bypass ratio the only matter the Industry can focus on to improve the efficiency? Definitely not; the type of fuel one uses determines, among other things, the combustion products found in the exhaust. This question was asked by the Industry many decades ago, but it is today when we can start to see the consequences of exploring that question. On the one hand, electric aircraft are about to be a reality; no combustion means no pollution from engine side. On the other hand, an old, familiar character has come into the spotlight once more: hydrogen, principally in its liquid form.

2.2 Liquid hydrogen: a promising resource

Liquid hydrogen, commonly known and abbreviated as LH_2, possesses many promising characteristics [1], from which one might remark: the main combustion product is water, and if injection and combustor geometry are optimal, it produces a very homogeneous flame during combustion (hot spots are rare rather than common); this avoids the creation of hazardous products that need higher temperatures to be formed (like nitrogen products, NO_x) [31]; besides, in order to be liquid, hydrogen needs to be cryogenic, reaching temperatures around 20 K (-253.15 ºC) while stored inside the fuel tanks at a considerable pressure, and these very low temperatures open the door to an appealing option: to employ the fuel (LH_2) as a heat sink to cool down key stages of the engine while heating it up to reach ideal injection temperatures.

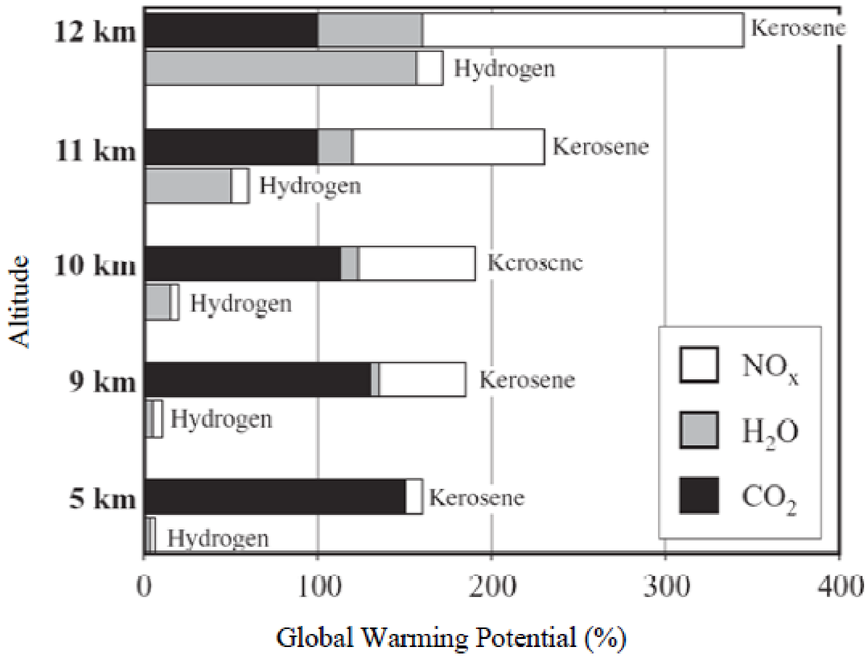


Figure 2.6: Net greenhouse effects of kerosene and hydrogen. [42]

The European Union has conducted a thorough research on this matter (apart from funding key projects seeking revolutionary technological advancements to push Europe towards the greenest continent in the Globe), and a summary of this investigation can be read on its ”Strategic research and innovation agenda” [29].

However, LH_2 is not a panacea, albeit benefits look promisingly bigger than the costs. It presents two major drawbacks:

- It requires significantly bigger tanks than traditional fuels. In terms of mass, LH_2 possesses almost three times the energy content of gasoline (120 against 44 MJ/kg) but, in terms of volume, its density is four times smaller (8 versus 32 MJ/kg) [1], [15].

- H_2 molecules can escape the tank considerably easily, increasing the risks while manipulating the fuelling system, that is, when loading the aircraft (explosions are a serious risk if hydrogen runs out of its storage) [1], [9], [15].

Nevertheless, this project wants to focus on engine performance, leaving aside aircraft design issues or operational risk management. If the reader wants to explore these concepts, [26], [27] offer thorough reviews. Combustion chamber design is also omitted in this paper, but some good information regarding this point can be found in [31], [35]. Thus, the key aspect this IRP emphasises is the management of the aforementioned cryogenic temperatures, that is to say, heat management, or more precisely, heat *transfer* management. Concretely, this IRP focuses on heat transfer when employing heat exchangers.

2.3 Heat exchangers for LH_2 usage in aircraft engines

It has been mentioned that cryogenic characteristics offer an interesting opportunity in terms of performance. This occurs because a certain part of the combustion energy is transformed into thermal energy instead of kinetic. By absorbing some of that heat, the temperature of the LH_2 increases to reach an optimum value for combustion, that is to say, that heat is being used for cycle optimisation purposes instead of being exhausted to the outer air.

There are two approaches to this matter. One is to completely redesign the engines with specific geometries that pursue the best design for LH_2 utilisation. This approach is being followed by companies/projects like EnableH2 [19]. The other option is to adapt already existing engines that run on aviation fuel to run on LH_2, that is, to minimise structural changes to avoid ruling out engines that are already operative. Although it might look like brand new technology, it is an option that was already explored decades ago. NASA [1] studied the possibility of designing an LH_2 aircraft from scratch, analysing from body's structural/aerodynamic technical details to the smallest subsystems, such as engine's oil circuit. When focusing on the design of the engines, they dedicated a entire section to heat exchanging technology, and proposed several engine configurations (each one depending on the location of the heat exchangers). Each configuration presents its own pros and cons, and broadly speaking, they are as follows:

– Compressor precooler: this option was quickly scrapped by NASA as the benefits could not compete with the problems this technology presents, i.e., severe air side freezing, considerable pressure drops and, in terms of practicality, it requires too much hardware changes and it is heavy. These conclusions have been recently validated by [34]. A sketch of it is given in Figure 2.7.

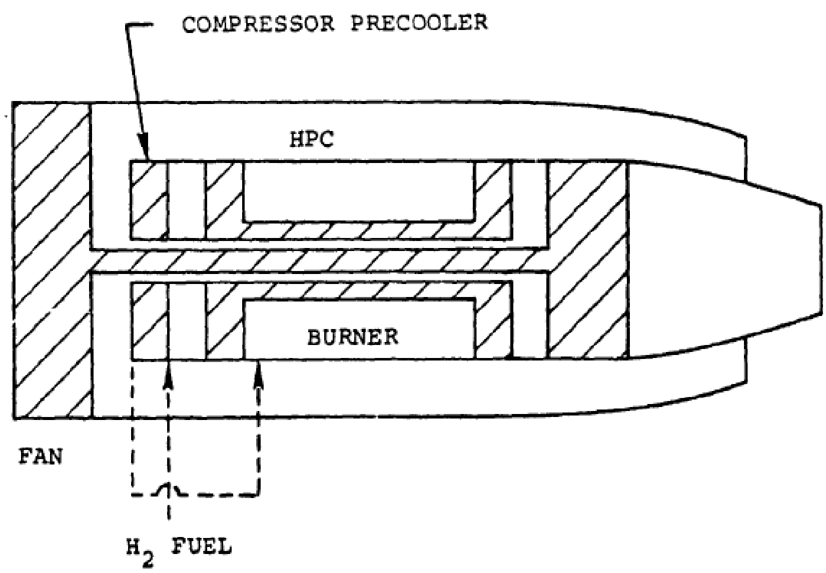


Figure 2.7: NASA's proposal for compressor precooling using a heat exchanger. [1]

– Compressor intercooler: although it presents serious benefits, drawbacks are still significantly important. Also scrapped by NASA. [34] and specially [19] demonstrate that it is not worth it when the approach is to minimise structural changes. If a brand new compressor is designed, then this design can be made to match the sizing requirements of the intercooler, minimising its drawbacks (similar to the ones present in the precooler). A visual representation of it is given in Figure 2.8.

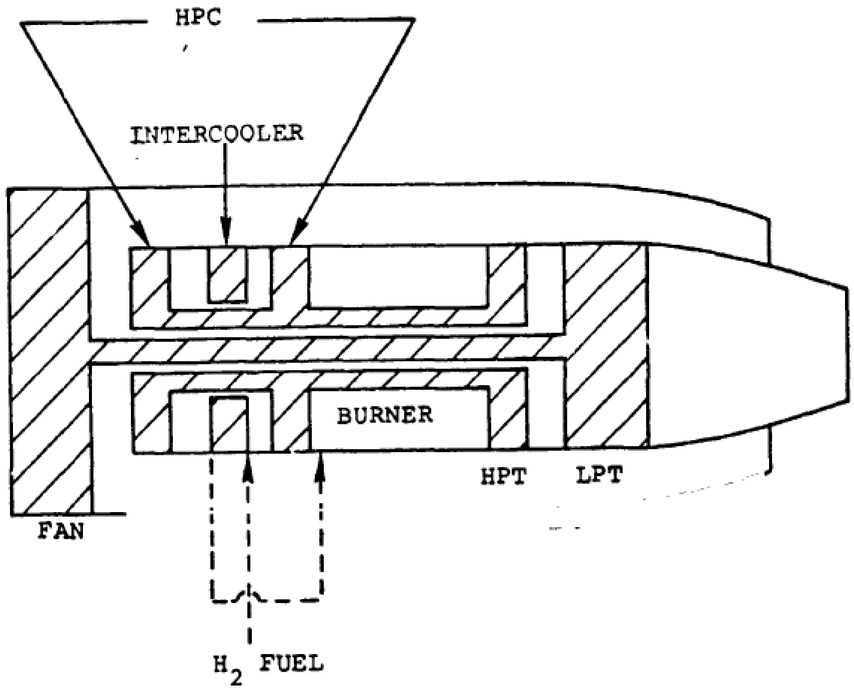


Figure 2.8: NASA's proposal for compressor intercooling using a heat exchanger. [1]

– Hydrogen cooling of turbine cooling air: the idea is to cool down the air meant to cool down turbine blades. Located between the high-pressure (HPT) and low-pressure turbines (LPT), it shows promising results, as it is not structurally complex to incorporate it to current engines. It is also important to recall that it reduces the effect of vane flow (originally determined to diminish the effect of cooling flow) on turbine's efficiency. [34] shared NASA's conclusion regarding this technology. It is described in Figure 2.9.

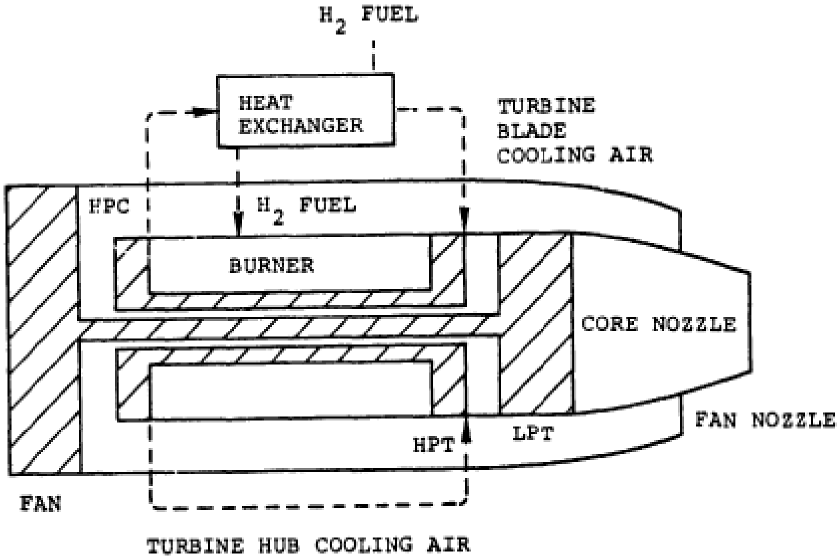


Figure 2.9: NASA's proposal for cooling of turbine cooling air using a heat exchanger. [1]

– Fuel heating with exhaust gas: this technology shows the most promising results, e.g., in terms of structure (high mass-flow-rates allow reasonably small heat exchangers). NASA concluded it could only be located in the main flow. However, [2] and [34] theorised that secondary flow could also be useful to control LH_2 injection temperature. This technology can be observed in Figure 2.10.

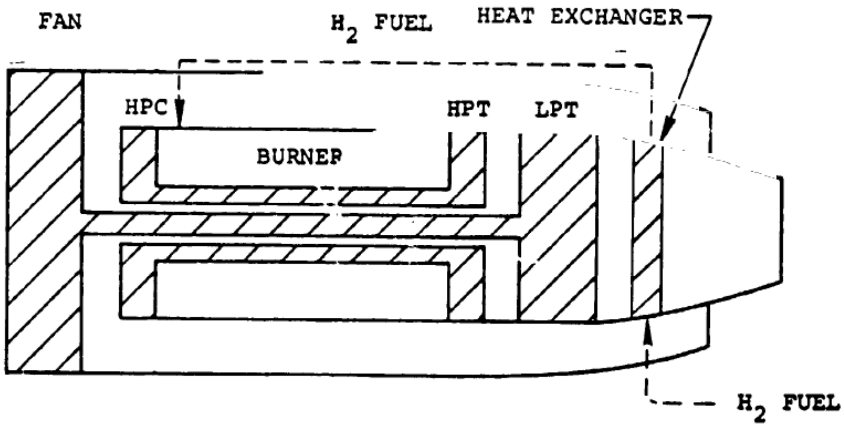


Figure 2.10: NASA's proposal for LH_2 heating with exhaust gas in the nozzle using a heat exchanger. [1]

2.3.1 Adaptability of heat exchangers to cryogenic fluids

Another aspect which requires investigation is the fact that maybe materials commonly used for heat exchangers are not eligible for cryogenic fluids. Although this aspect deserves some researches fully focused on it, a couple of preliminary calculations and conclusions are introduced in this thesis via the exploration of a concept typically known as thermal inertia, I. According to [3], thermal inertia can be defined as the *property of a material that expresses the degree of slowness with which its temperature reaches that of the environment*; or, focusing on a definition for materials' thermal inertia, *capacity of a material to store heat and to delay its transmission.* [2] also explores this concept in a slightly different way[4] (by analysing thermal conductivity, k), and gives some enriching examples of utilisation. Generally speaking, thermal inertia can be defined as:

$$I = \sqrt{\rho c_p k} \tag{2.1}$$

where eq.(2.1) shows that it is the square root of the density, ρ, times the specific heat at constant pressure, c_p, times the thermal conductivity, k. I is usually expressed in $\mathrm{J/(m^2 \cdot s^{1/2} \cdot K)}$, ρ in $\mathrm{kg/m^3}$, c_p in J/(kg·K) and k in W/(m·K).

The influence of thermal inertia is further stated in Chapter 4.

2.3.2 The future of heat exchangers

Heat exchangers are a technology in constant development, as each new material, new manufacturing technique, even new fluid (mostly in terms of popularity) represents a novel opportunity and challenge for these devices. Many different types of heat exchangers and their technological characteristics are discussed in Chapter 4, including microchannel cross-flow heat exchangers.

[4] As eq.(2.1) shows, studying the influence of thermal conductivity is directly related to studying thermal inertia's.

Speaking of novel techniques, a company called MEGGITT and specially its heat exchanger manufacturing division, Heatric [43] produce what they call ”Printed Circuit Heat Exchangers” (PCHE), which are manufactured using diffusion-bonding techniques to create heat exchanger cores without welds or joints.

This is just an example of how companies are really considering heat exchangers as a profitable line of business for the near future, and this directly translates into technological advances that are about to come.

2.4 Simulating heat exchangers

In order to analyse and put into practice all the literature review, it is necessary to use some software to simulate the performance of an engine. This software must be flexible enough to work with different types of engines, while leaving the user modify the properties of each stage, introduce some new stages, ducts, bleedings... and remove existing ones. In other words, the chosen software must leave the user customise any engine model in as many ways as possible.

Thankfully, Cranfield University developed a software, Turbomatch, which is very customisable and is set to be the main tool for this project. Turbomatch divides the different (main) components of an engine into *bricks*, and each brick possesses different parameters which can be adjusted independently.

As the reader might have deduced, there are bricks for heat exchangers. However, this software presents a major drawback: it cannot consider external flows. In other works, it only contemplates the air passing through the intake and exhausted by the nozzle. As we want to use the fuel (LH_2) as the cold fluid in those heat exchangers, we would be using an external flow. Hence, Turbomatch cannot reach a solution.

In order to avoid this limitation, an external tool must be used to simulate those heat exchangers in a particular way that gives the outlet conditions as an output. By doing this, both Turbomatch and the aforementioned tool need to be used at the same time to reach a solution. This matter leads to the following questions (question 1 is answered in Chapter 3, and the other two are answered below):

1. Among the different, available general methods of heat exchanger simulation, which one shall be used?
2. Should this method be specific or generic? That is to say, should it be focused on a certain type of heat exchanger (including its geometry), or should it give the opportunity to choose among different types of heat exchangers and develop all the calculations from a general approach, without focusing on geometries or individual characteristics?
3. Which software shall be used to develop this tool?

A general approach to this matter is preferred to give as much flexibility as possible to the user. The tool must be developed in such a way that it is easy to alternate between different types of heat exchangers and their boundary conditions without giving too many details. In terms of software choice, Matlab was selected as the main tool for this purpose as it is a very common software in the engineering world. The code is described and explained in Chapter 3.

Chapter 3

Heat Exchanger Modelling and Simulation

There are several design methods for heat exchangers. This Chapter focuses on some of the methods available for two-fluid heat exchangers. The most typical methods are:

- The log-mean temperature difference (LMTD).
- The effectiveness ε-NTU method, where NTU stands for *Number of Transfer Units*.
- Dimensionless mean temperature difference (Ψ-P).
- (P_1-P_2) method.

LMTD method is a logarithmic average. The larger the LMTD, the more heat is transferred (for heat exchangers with constant heat transfer coefficient, U and area, A). It can be used when both fluids' mass-flow-rates and inlet temperatures, and one fluid's outlet temperature are known. Constant mass-flow-rate and (fluid) thermal properties is assumed, together with a temperature change rate proportional to the temperature difference between both fluids [7]. This last assumption means that LMTD can only be applied when both fluids present constant specific heat, which usually is quite a good description of fluids over a small range of temperature. However, in this thesis a cryogenic fluid is going to be used as cold flow, and air from the aero engine as hot flow (air coming out of the turbine can be around 800 K hot, while LH_2 can be found at only 20 K in the fuel tank), and this considerable difference, albeit LH_2 could be coming into engine systems at around 200 K [1], is considered to be a good-enough argument to rule this simulation method out.

The ε-NTU method, on the other hand, can be used when outlet temperatures are not known, that is to say, it looks like it might give more flexibility for code development as it requires less information than LMTD and, at the same time, it requires less iterations than LMTD. ε-NTU is also semi-empirical: each type of heat exchanger possesses a specific equation for it in which effectiveness (ε) and the number of transfer units (NTU) are related. This is shown later on.

Ψ-P and $P_1 - P_2$ are graphical methods, require plotting and therefore they are not the most suitable options.

Summing up, ε-NTU is the modelling method chosen for this thesis. Its general procedure is introduced below these lines.

3.1 ε-NTU method

Before starting, it is important to note that, throughout this thesis, subscripts h and c stand for *hot* and *cold*, respectively.

In order to make this model work, some boundary conditions need to be specified:

– Inlet mass-flow-rates of hot fluids, $\dot{m}_h$ and $\dot{m}_c$.

– Inlet temperatures, $T_{h,in}$ and $T_{c,in}$.

– Inlet total pressures, $P_{h,in}$, and $P_{c,in}$.

– Both fluids' specific heat at constant pressure, $c_{p,h}$ and $c_{p,c}$.

Additionally, two more boundary conditions are asked, Reynolds numbers and inlet speeds. The reason for this is explained later on, after introducing pressure loss calculations.

Some ratios need to be calculated to make things easier. These are as follows:

$$C_h = \dot{m}_h c_{p,h} \tag{3.1}$$

$$C_c = \dot{m}_c c_{p,c} \tag{3.2}$$

where C_h and C_c are the product of the first and last boundary conditions aforementioned. From them, another two ratios can be obtained:

$$C_{max} = max\{C_h, C_c\} \tag{3.3}$$

$$C_{min} = min\{C_h, C_c\} \tag{3.4}$$

With these last ratios from eq.(3.4), the maximum transferred heat can be obtained with the following expression:

$$Q_{max} = C_{min}\left(T_{h,in} - T_{c,in}\right) \tag{3.5}$$

This heat is the maximum exchange that can happen between both fluids. Thus, it can be considered as a limit, a *boundary condition*, to obtain a range of temperatures in which all the calculations can be carried out.

3.1.1 *Sizing* problem

This method, which gives information about outlet temperatures and also about the necessary size of the heat exchanger to achieve those outlet parameters, can be solved via two different paths (or *problems*), identified in this thesis as *sizing problem* and *thermal problem*. In the first one, the main target is to find out the heat transfer area (A) the heat exchanger must possess to allow the desired heat exchange to occur. In this first problem, outlet parameters are not considered a target or priority, but an input the user can do or let the method evaluate the heat exchanger for the whole range of valid temperatures. In the second path, the *thermal problem*, the size of the heat exchangers becomes an input of the user, which, depending whether it is just a value or a range of values, gives a value or a range of values for the outlet temperatures. In other words, the first method answers the question ”to obtain these specific outlet conditions, how big does the heat exchanger need to be?”, and the second one to the question ”for a certain heat exchanger, known its transfer surface, what outlet temperatures can be obtained?”.

In this Section, the solution to the first problem is given; the solution to the other one is given later on. Nevertheless, to solve any of them an energy balance must be introduced. As one might have deduced, if there is a maximum transferable heat that applies to both fluids, this could mean that both fluids’ variations should show an equilibrium among themselves. And, in fact, that is the case. Hence:

$$C_c\left(T_{c,out}-T_{c,in}\right)=C_h\left(T_{h,in}-T_{h,out}\right) \tag{3.6}$$

This expression can be evaluated for the whole range of temperatures (delimited by the maximum transferable heat) or for just a exact, valid value of one of the two outlet temperatures, obtaining as a result the other outlet temperature by solving eq.(3.6) for the remaining outlet parameter. For example, imagining $T_{h,out}$ has already been determined by the desires of the user:

$$T_{c,out} = T_{c,in} + \frac{C_h}{C_c}\left(T_{h,in} - T_{h,out}\right) \tag{3.7}$$

These last two expressions can be shortened if we express each side of eq.(3.6) not as the maximum transferable heat, but as the actual heat that is being transferred *globally*, i.e., affecting both fluids in a balanced way, and it can be expressed as follows:

$$Q_{c,global} = C_c\left(T_{c,out} - T_{c,in}\right) \tag{3.8}$$

$$Q_{h,global} = C_h\left(T_{h,in} - T_{h,out}\right) \tag{3.9}$$

So, introducing eq.(3.9) into eq.(3.6) and eq.(3.7), eq.(3.10) and eq.(3.11) are obtained, respectively:

$$Q_{c,global} = Q_{h,global} = Q_{global} \tag{3.10}$$

$$T_{c,out} = T_{c,in} + \frac{Q_{h,global}}{C_c} = T_{c,in} + \frac{Q_{global}}{C_c} \tag{3.11}$$

It has been mentioned several times that Q_{max} marks the limit of the *valid* outlet temperatures. It is now time to show how these ranges can be calculated. On the one hand, the worst thing that could happen is that the heat exchanger would not work; there would be no heat exchange occurring. Thus, outlet conditions would be identical to the inlet ones. In other words, the highest possible outlet temperature for the hot fluid is its inlet temperature, therefore the lowest possible temperature for the cold fluid is its inlet temperature:

$$T_{h,out,max} = T_{h,in} \tag{3.12}$$

$$T_{c,out,min} = T_{c,in} \tag{3.13}$$

On the other hand, the other extreme of the range corresponds to the one marked by the maximum transferable heat, Q_{max}. When $Q_{global} = Q_{max}$, expressions like eq.(3.11) and its homologous equation for hot fluid's outlet temperature (when cold fluid's outlet temperature is given) can be employed as follows:

$$T_{h,out,min} = T_{h,in} - \frac{Q_{max}}{C_h} \tag{3.14}$$

$$T_{c,out,max} = T_{c,in} + \frac{Q_{max}}{C_c} \tag{3.15}$$

Thus, temperature ranges can be expressed as:

$$T_{h,out} \in \{T_{h,out,min}, T_{h,out,max}\} \Rightarrow T_{h,out} \in \{T_{h,in} - \frac{Q_{max}}{C_h}, T_{h,in}\} \tag{3.16}$$

$$T_{c,out} \in \{T_{c,out,min}, T_{c,out,max}\} \Rightarrow T_{c,out} \in \{T_{c,in}, T_{c,in} + \frac{Q_{max}}{C_c}\} \tag{3.17}$$

The next step of the effectiveness-NTU method is, indeed, to calculate the effectiveness (ε) of the heat exchanger. This is done as shown in eq.(3.18), where one can observe that the effectiveness is the quotient between the actual heat being transferred, Q_{global}, and the maximum transferable heat, Q_{max}:

$$\varepsilon = \frac{Q_{global}}{Q_{max}} \tag{3.18}$$

Below these lines one can find the analytical expressions to calculate the number of transfer units (NTU) once the effectiveness is known. A different expression relating both needs to be applied according to the type of heat exchanger. To ease their typing and comprehension, the following steps are taken:

– For all the expressions, the C rate is introduced:

$$C = \frac{C_{min}}{C_{max}} \tag{3.19}$$

– For shell-and-tube heat exchangers, where several shell passes (N_s) are considered[1],

[1] It must be borne in mind that the most typical scheme for shell-and-tube heat exchangers is: n shell passes, $2n$, $4n$, $6n$ tube passes (for one shell pass, 2, 4, 6... tube passes).

the folowing simplification is proposed:

$$n = \frac{NTU}{N_s} \tag{3.20}$$

Also, for multiple shell passes, the effectiveness of each case[2] is calculated (ε_{1p}).

From [24] and [28], the following expressions are used depending on each type of heat exchanger:

Double-pipe:

Parallel flow $$\varepsilon = \frac{1-e^{-NTU(1+C)}}{1+C} \tag{3.21}$$

Counter-flow $$\varepsilon = \frac{1-e^{-NTU(1-C)}}{1-C\cdot e^{-NTU(1-C)}} \tag{3.22}$$

Counter-flow, case $C = 1$ $$\varepsilon = \frac{NTU}{NTU+1} \tag{3.23}$$

Cross-flow (single-pass):

Both fluids unmixed $$\varepsilon = 1-e^{\frac{NTU^{0.22}}{C}\left[e^{-C\cdot NTU^{0.78}}-1\right]} \tag{3.24}$$

C_{max} mixed $$\varepsilon = \frac{1}{C}\left(1-e^{1-C\left[1-e^{-NTU}\right]}\right) \tag{3.25}$$

C_{min} mixed $$\varepsilon = 1-e^{-\frac{1}{C}\left[1-e^{-C\cdot NTU}\right]} \tag{3.26}$$

Both fluids mixed $$\varepsilon = \left[\frac{1}{1-e^{-NTU}}+\frac{C}{1-e^{-C\cdot NTU}}-\frac{1}{NTU}\right]^{-1} \tag{3.27}$$

[2]One should notice that the expression for ε_{1p} is the same as the one for the effectiveness of the *one-pass* case, but with n instead of NTU in it.

Shell-and-tube:

One shell pass
$$\varepsilon = 2\left[1+C+\sqrt{1+C^2}\frac{1+e^{-NTU\sqrt{1+C^2}}}{1-e^{-NTU\sqrt{1+C^2}}}\right]^{-1} \tag{3.28}$$

Multiple shell passes
$$\varepsilon = \frac{\left[1-\varepsilon_{1p}C/1-\varepsilon_{1p}\right]^n - 1}{\left[1-\varepsilon_{1p}C/1-\varepsilon_{1p}\right]^n - C} \tag{3.29}$$

Special case $C=1$
$$\varepsilon = \frac{n\varepsilon_{1p}}{1+(n-1)\varepsilon_{1p}} \tag{3.30}$$

Special case for all types:

Special case $C=0$
$$\varepsilon = 1-e^{-NTU} \tag{3.31}$$

The only step remaining is to calculate the overall heat transfer coefficient, U and the heat exchange surface, A. However, unless one of them is specified, only the product $U \cdot A$ can be obtained. If the exact thermal or geometrical characteristics of a certain heat exchanger are known, then it would be possible to calculate U and A separately. However, as the intention of this thesis is to keep every step as generic as possible, the chance of specifying (if wanted) each of these parameters is given to the user, but for theoretical explanations, everything is described from a general perspective. Anyway, the expression that relates the heat transfer coefficient with the exchange surface and the number of transfer units is:
$$NTU = \frac{UA}{C_{min}} \tag{3.32}$$

3.1.2 *Thermal* problem

This path could be described like ”reading *sizing* problem from the bottom to the top”. First, either $U \cdot A$ or U and A separately are specified. They could be either an exact value or a range of values that wants to be studied. The next step is to obtain the number of transfer units from eq.(3.32).

According to the type of heat exchanger to be analysed, its corresponding equation from page 25 shall be chosen and employed, bearing in mind the necessity to use eq.(3.19) and eq.(3.20) first.

Once the proper equation from page 25 is put into practice, the effectiveness is obtained. As Q_{max} is calculated from the initial boundary conditions, i.e., it is known, Q_{global} can be obtained from eq.(3.18). Hence, introducing eq.(3.10) into eq.(3.11), cold fluid's outlet temperature is obtained, meaning that, by solving eq.(3.6) for hot fluid's outlet temperature, the whole heat exchanger is thermally characterised and the solution for the problem has been found.

3.2 Pressure drop calculation

Although all the presented parameters are now explained and calculated, there is another vital aspect in order to create a tool that can match Turbomatch's requirements for heat exchanger implementation: the calculation of the pressure drop happening inside the heat exchanger for each fluid.

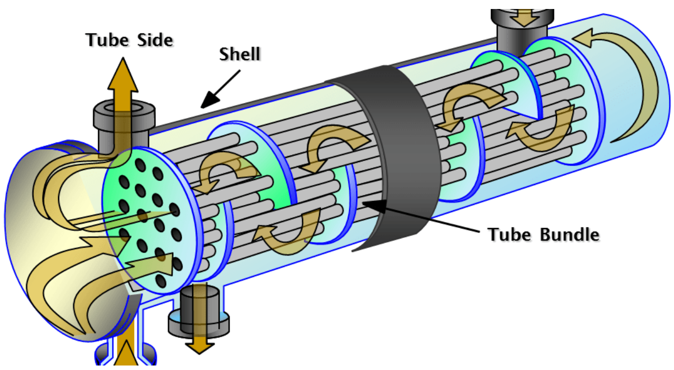


Figure 3.1: A generic example of a shell-and-tube heat exchanger. [25]

As the reader might have thought, pressure drops are tightly related to the specific geometry of the heat exchanger under consideration. However, and as it has been aforementioned, the intention of this paper is to transmit a general method, without focusing on very specific details (like geometrical distribution of the tubes inside a heat exchanger). Fortunately, there is plenty of literature focused on developing general correlations for different types of heat exchangers that allow the user to calculate the occurring pressure drop almost without knowing any geometrical aspect from the heat exchanger they are studying.

Of course, this approach presents several limitations that need to be assessed when speaking of pressure drop calculation accuracy:

– Heat exchangers are typically divided into different parts, identifiable for the sake of this explanation as *inlet*, *body* and *outlet*. Each part induces pressure drops that need to be calculated, sometimes separately, taking into account the specific geometrical characteristics of that certain part. Using a general correlation that covers the whole heat exchanger is a risk in terms of accuracy.

– General correlations for pressure drops typically rely heavily on Reynolds (*Re*) or Nusselt (*Nu*) numbers. In this thesis, all the pressure drop correlations include, at most, calculations which only require the determination of *Re*.

– Calculating *Re* is not an easy task when a general approach wants to be employed. It requires the user to specify parameters like the (dynamic) viscosity, which is approximately constant for fluids like air, but varies a lot when cryogenic fluids like LH_2 are assessed. As LH_2 is quite a novel technology, there are not too many sources to consult and obtain accurate viscosity values, although [20] offers a good library of values for different parameters of LH_2, and [21] presents a complete index of contents related to hydrogen properties. To avoid having to input a certain dynamic viscosity for each temperature that wants to be studied, *Re* is going to be considered an input, a *boundary condition* to be added to the ones presented on page 20.

– Some correlations also require determining the speed and one or two geometrical values (like, for example, the length of the pipes of a double-pipe heat exchanger). Hence, the inlet speed of each fluid is also considered an input from now onwards, and must be incorporated to the list on page 20. The mentioned geometrical values are not considered members of that list because each type of heat exchanger (thus, each general correlation) requires some geometrical parameters or others; these are specified *after* declaring the inlet conditions and choosing the type of heat exchanger.

To ease the comprehension of these matters, the list presented on page 20 is now shown again and updated with the aforementioned parameters. The inlet boundary conditions to be specified are:

– Both fluids' inlet mass-flow-rates, $\dot{m}_h$ and $\dot{m}_c$.

– Inlet temperatures, $T_{h,in}$ and $T_{c,in}$.

– Inlet total pressures, $P_{h,in}$, and $P_{c,in}$.

– Specific heats at constant pressure, $c_{p,h}$ and $c_{p,c}$.

– Reynolds numbers, Re_h and Re_c.

– Inlet velocities, v_h and v_c.

3.2.1 Pressure drop general correlations

Double-pipe heat exchangers: subscripts *i* and *o* stand for *inner* and *outer* pipe, respectively. The expressions below these lines need to be particularised for each pipe. Peccini et al. [4] present and validate in their paper the following expressions for friction factor (f) and pressure drop (ΔP_{drop}) calculations for double-pipe heat exchangers:

Laminar ($Re < 1311$) $$f = \frac{64}{Re} \tag{3.33}$$

Transitory ($1311 \leq Re \leq 3380$) $$f = 0.0488 \tag{3.34}$$

Turbulent ($Re > 3380$) $$f = 0.14 + \frac{1.056}{Re^{0.42}} \tag{3.35}$$

Pressure drop $$\Delta P_{drop} = \rho f \frac{Lv^2}{2d} \tag{3.36}$$

where ρ is the density of the fluid, and L and d are the length and diameter of each pipe, respectively.

Cross-flow heat exchangers[3]**:** the following expressions for friction factor calculation are obtained from [12] and validated by employing the method described in [10]. On the other hand, the formula for ΔP_{drop} is obtained from [23]:

Laminar ($Re \leq 3000$) $$f = \frac{1.33}{\sqrt{Re}} \tag{3.37}$$

Turbulent ($Re > 3000$) $$f = \frac{0.074}{\sqrt[5]{Re}} \tag{3.38}$$

Pressure drop $$\Delta P_{drop} = 4f \frac{\rho v^2}{2} \frac{L}{D_h} \tag{3.39}$$

where D_h is the hydraulic diameter and it is calculated as follows: for a generic cross-flow heat exchanger in which one can find channels with a rectangular section (referring to the height of the channel as b and as c to the channel width), the hydraulic diameter is obtained by applying the following equation [12]:

$$D_h = \frac{4bc}{2(b+c)} \tag{3.40}$$

Shell-and-tube heat exchangers: an example of this kind of heat exchangers can be

[3]Authors disagree on the Re ranges in which the flow is laminar/turbulent. Hence, the most general, shared value found in the literature is given in this thesis.

observed in Figure 3.1. All the equations are taken from [16]. Nevertheless, the reader is invited to review the method proposed in [17] in which this matter is carefully analysed and expanded:

Tubes (cold fluid, subscript t):

Laminar ($Re < 2300$) $$f_t = \frac{16}{Re} \tag{3.41}$$

Turbulent ($Re \geq 2300$) $$f_t = \frac{1}{(1.58\ln(Re) - 3.28)^2} \tag{3.42}$$

Pressure drop $$\Delta P_{drop,t} = 4\left(\frac{f_t L_t}{d_i} + 1\right) N \frac{1}{2}\rho v^2 \tag{3.43}$$

where L_t is the tube length, d_i is the inner diameter of the tube and N is the number of passes.

Shell (hot fluid, subscript s): The shell side is a bit trickier to calculate. The first thing to keep in mind is that the layout of the tubes in it heavily determines the pressure drop that one can find when using these heat exchangers. In this thesis, two significantly common layouts have been chosen: triangular and square pitch layouts.

Each layout has a particular formula to calculate its *equivalent diameter*, D_e:

Triangular $$D_e = \frac{8\left(\frac{\sqrt{3}}{4} Pitch^2 - \frac{\pi {d_o}^2}{8}\right)}{\pi d_o} \tag{3.44}$$

Square $$D_e = \frac{4\left(Pitch^2 - \frac{\pi {d_o}^2}{4}\right)}{\pi d_o} \tag{3.45}$$

where d_o is the outer diameter of the tube and $Pitch$ is, literally, the pitch of the tube (in meters).

Once the equivalent diameter is calculated, the pressure drop in the shell side can be calculated:

$$\text{Friction factor} \qquad f_s = e^{0.576-0.19\cdot\ln(Re)} \tag{3.46}$$

$$\text{Pressure drop} \qquad \Delta P_{drop,s} = f_s \frac{D_s}{D_e}(N_b+1)\frac{1}{2}\rho v^2 \tag{3.47}$$

where D_s is the inner diameter of the shell, and N_b the number of baffles.

All the explanations given here are entirely focused on obtaining the pressure drops inside a shell-and-tube heat exchanger. However, [16] expands this method and shows how to fully characterise a shell-and-tube heat exchanger.

Although all the expressions are taken from several-times-cited literature, the reader might have noticed that the particularities of each heat exchangers induce remarkable changes in the pressure drop calculations. In other words, even if this generic method is considered good enough, it will always lack from accuracy when comparing it to any model specifically developed for a certain heat exchanger. That is why validating the results these equations might give back is quite a task. Nevertheless, for general purposes, this pressure drop evaluation is considered sufficiently good [4], [10] [16].

3.3 Code flowchart

On page 34, a flowchart of the code developed for this thesis can be found. However, the reader should bear in mind this is just one of the many ways of schematising the information detailed throughout this Chapter. Some additional information concerning the flowchart is now given.

User inputs:

(1) All the inlet conditions listed on page 29 must be specified by the user.

(2) The type of heat exchanger has to be selected by the user. The types of heat exchangers available on this code are many of the most typical ones:

- Double-pipe: parallel-flow or counter-flow.
- Cross-flow (single-pass): both fluids unmixed, C_{max} fluid mixed, C_{min} fluid mixed, both fluids mixed.
- Shell-and-tube: one shell pass, N shell passes.

(3) To solve the *thermal problem*, either U, A or $U \cdot A$ need to be specified by the user. The programme detects which one has been specified and which one has not.

(4) Throughout Section 3.2.1, it has been highlighted that, depending on the type of heat exchanger selected in (2), the geometric parameters to be specified vary. The user must introduce the corresponding ones to calculate ΔP_{drop}.

Scripts:

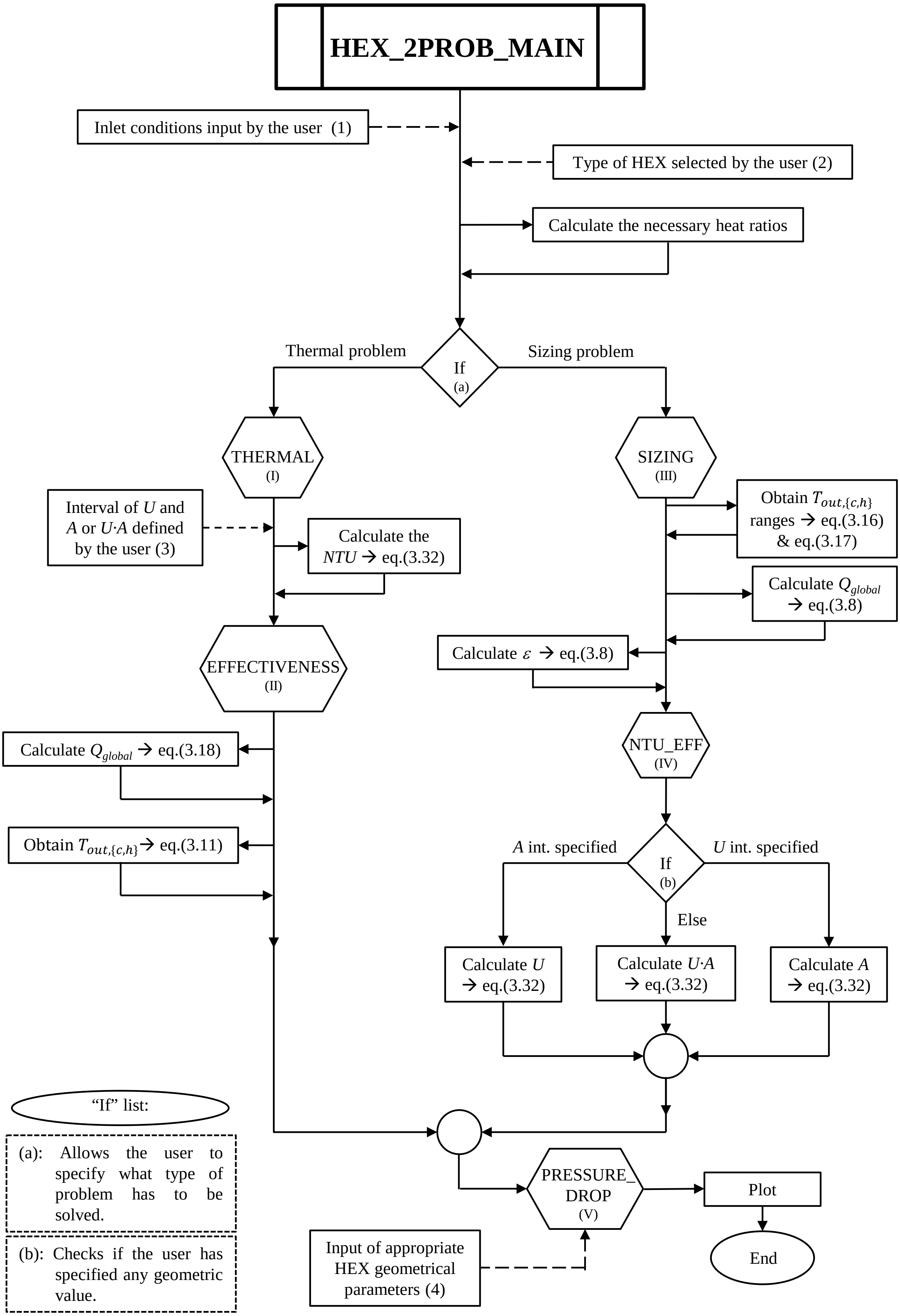
(I) THERMAL: this script solves the *thermal problem* described in this thesis. All the solved equations are explicit.

(II) EFFECTIVENESS: this script is called by THERMAL to calculate ε and contains all the equations presented from page 25 onwards.

(III) SIZING: this script solves the *sizing problem* stated in this Chapter. It solves implicit and explicit equations.

(IV) NTU_EFF: this script is almost identical to EFFECTIVENESS, but it is prepared to solve the most complex equations from page 25 onwards implicitly.

(V) PRESSURE_DROP: the aim of this script is to ask the user for the necessary geometrical parameters according to the type of heat exchanger selected in (2) and, with that information, calculate the pressure drops occurring inside it.



HEX_2PROB_MAIN
Inlet conditions input by the user (1)
Type of HEX selected by the user (2)
Calculate the necessary heat ratios
Thermal problem
If (a)
Sizing problem
THERMAL (I)
SIZING (III)
Interval of U and A or U·A defined by the user (3)
Calculate the NTU → eq.(3.32)
Obtain $T_{out,\{c,h\}}$ ranges → eq.(3.16) & eq.(3.17)
Calculate Q_{global} → eq.(3.8)
EFFECTIVENESS (II)
Calculate ε → eq.(3.8)
Calculate Q_{global} → eq.(3.18)
NTU_EFF (IV)
Obtain $T_{out,\{c,h\}}$ → eq.(3.11)
A int. specified
If (b)
U int. specified
Else
Calculate U → eq.(3.32)
Calculate U·A → eq.(3.32)
Calculate A → eq.(3.32)
"If" list:
(a): Allows the user to specify what type of problem has to be solved.
(b): Checks if the user has specified any geometric value.
PRESSURE_ DROP (V)
Plot
End
Input of appropriate HEX geometrical parameters (4)

Chapter 4

Results

This Chapter is divided into three main sections: the first one presents the capabilities of the code by showing some results obtained for many different types of heat exchangers; the second one describes what happens when this tool is used together with Turbomatch; the last one briefly outlines some preliminary calculations carried out to address the potential impact of thermal inertia.

4.1 Results obtained from the code

In this Section, some charts are presented in which the calculations undertaken by the code are plotted. The general inlet conditions below have been taken from [1] and [20], and are as follows:

- Mass-flow-rates [kg/s]: $\dot{m}_h = 0.832$; $\dot{m}_c = 0.351$.
- Specific heat at constant pressure [J/(kg K)]: $c_{p,h} = 1100$; $c_{p,c} = 13849$.
- Inlet temperatures [K]: $T_{h,in} = 791$; $T_{c,in} = 200$.

– Inlet pressures [kPa]: $P_{h,in} = 1515$; $P_{c,in} = 5068$.

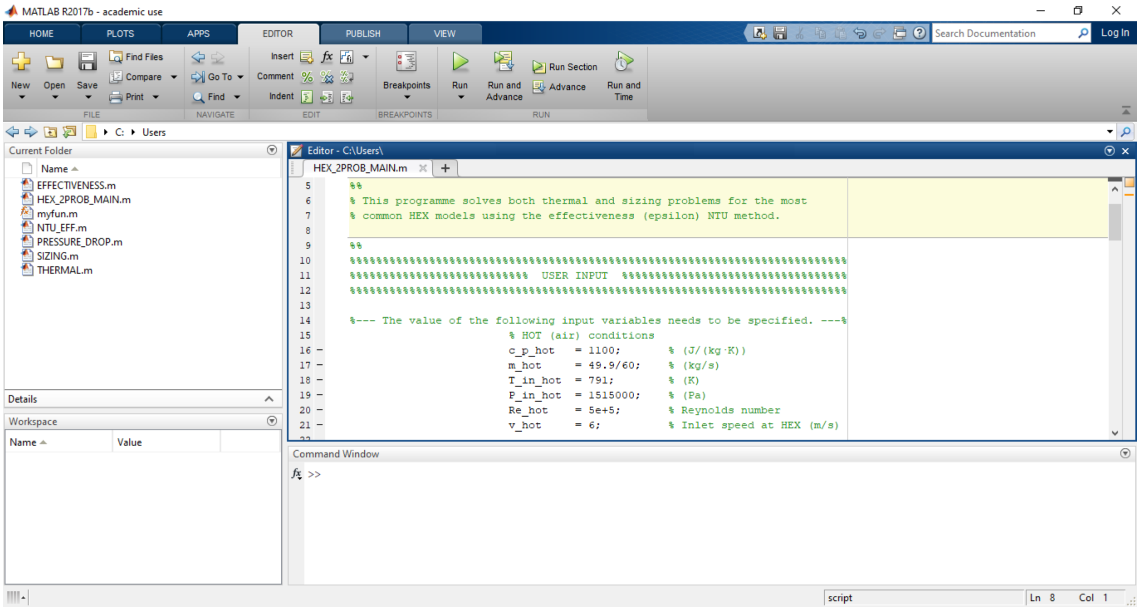


Figure 4.1: Matlab's main screen with the principal script, HEX_2PROB_MAIN open. A few of the aforementioned inlet conditions can be spotted.

The geometrical conditions for pressure drop calculations are mentioned later on for each type of heat exchanger. To maintain some kind of coherence, all the graphs shown below these lines have been obtained by solving the *sizing problem*. The *thermal problem* would be useful if a certain value (and not a range) of the different parameters is wanted. However, studying the *sizing problem* allows to appreciate the variation of those parameters, and therefore it is consider more useful. Neither U nor A are specified.

4.1.1 Double-pipe heat exchangers

Figure 4.2 and Figure 4.3 show the variation of the effectiveness of parallel- and counter-flow double-pipe heat exchangers against the variation of the number of transfer units and the product $U \cdot A$, respectively.

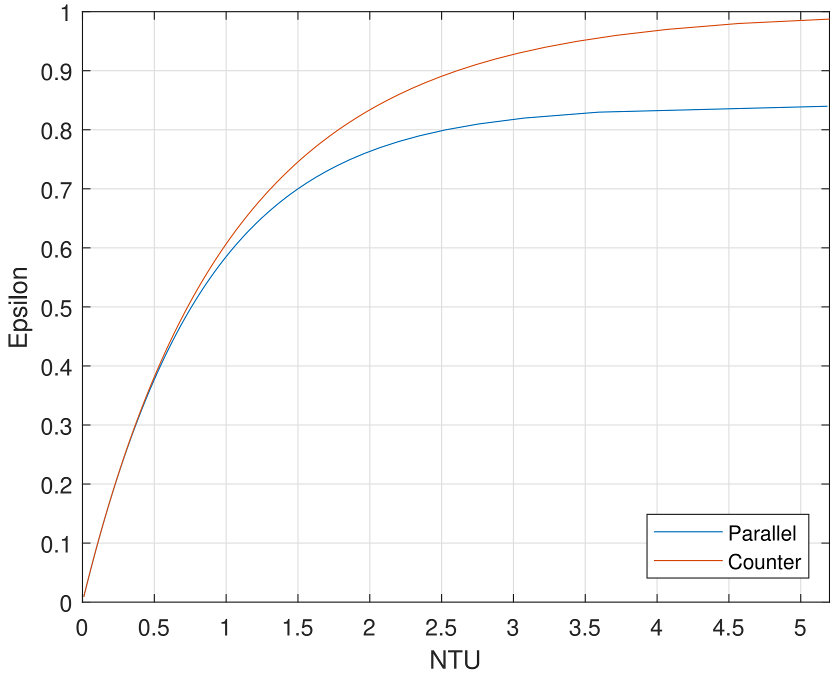


Figure 4.2: NTU vs ε for parallel- and counter-flow double-pipe heat exchangers.

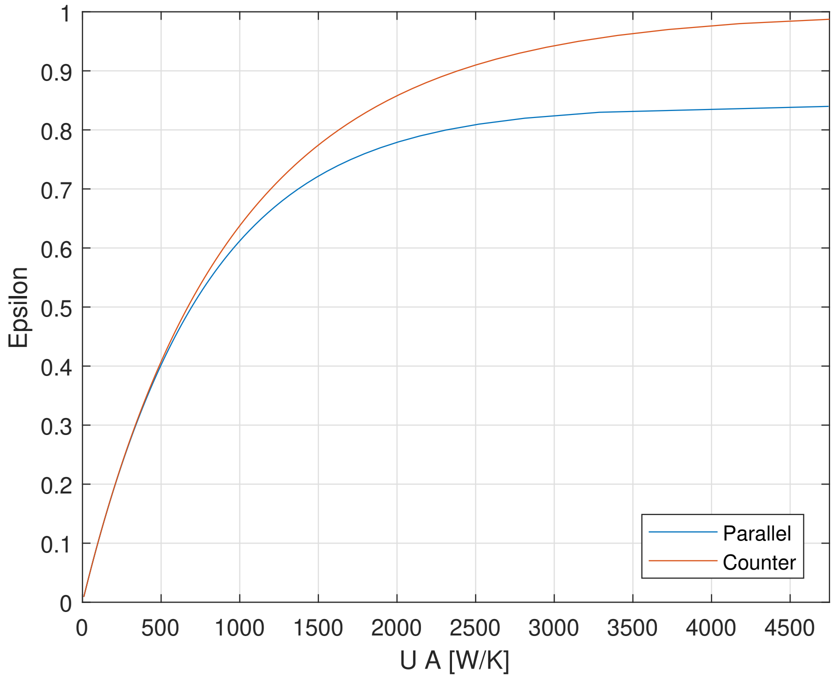


Figure 4.3: $U \cdot A$ vs ε for parallel- and counter-flow double-pipe heat exchangers.

As it can be observed, counter-flow heat exchangers offer better performance than parallel-flow ones. This is because counter-flow heat exchangers create a more uniform temperature difference between both fluids over the entire length of the pipe [7]. Both present similar performances up till $NTU = 1$; from there onwards, counter-flow ones show a significant improvement.

4.1.2 Cross-flow heat exchangers

Figure 4.4 and Figure 4.5 show a representative part of the variation of the effectiveness of cross-flow heat exchangers against the variation of the number of transfer units and the product $U \cdot A$, respectively.

To find out if these results are logic, one must go deep into Wilhelm Nusselt's *Fundamental Law of Heat Transfer* [36] in which he demonstrated that, in terms of effectiveness, the best configuration is that in which both fluids are unmixed, followed by those in which one of the two fluids is mixed, being the both-mixed configuration the one that shows the poorest effectiveness. Furthermore, if boundaries are pushed, C_{max}*-mixed* and *both-mixed* configurations roughly go over NTU = 6. However, the other two (specially the *both-unmixed* configuration) easily reach NTUs greater than 10. Figure 4.6 shows a comparison between unmixed and C_{max}*-mixed* configuration.

According to the literature, cross-flow heat exchangers develop the best performance of all the types of heat exchangers for high NTU values, and the second best behind counter-flow (double-pipe) heat exchangers for NTU ranges of 0-5 (the most used range) [7]. They might not be the most common in the Industry (shell-and-tube heat exchangers are the clear winners here), but as explained in Section 2.3.2, significant R+D efforts are being put in cross-flow technology because of their promising characteristics (compact size, reduced pressure drops...). New materials and novel manufacturing solutions are opening the door to improved cross-flow technology.

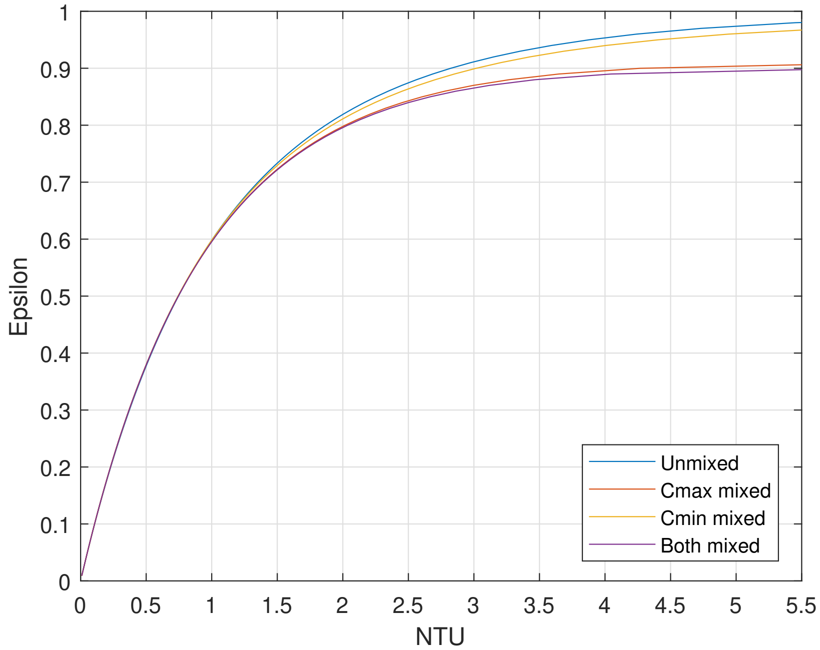


Figure 4.4: NTU vs ε for unmixed-unmixed, mixed-unmixed and mixed-mixed configurations of cross-flow heat exchangers.

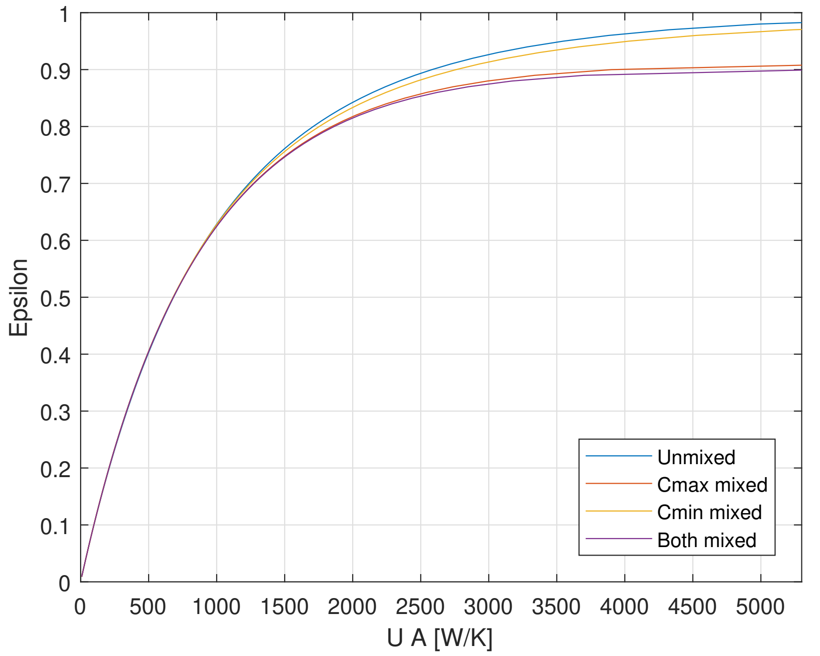


Figure 4.5: $U \cdot A$ vs ε for unmixed-unmixed, mixed-unmixed and mixed-mixed configurations of cross-flow heat exchangers.

A comparison between two configurations of cross-flow heat exchangers is given in Figure 4.6.

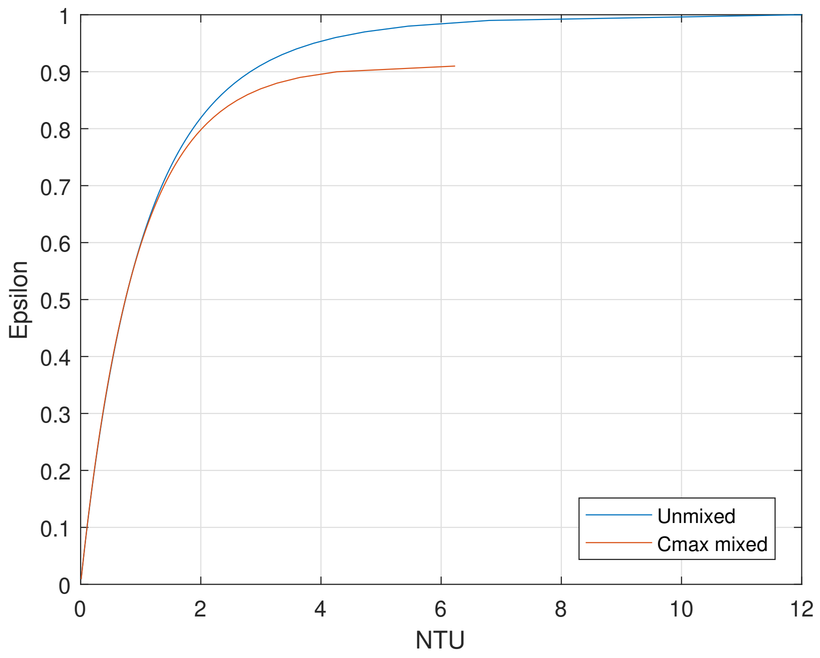


Figure 4.6: Detail of the NTU vs ε evaluation for unmixed and C_{max}-mixed configurations of cross-flow heat exchangers.

4.1.3 Shell-and-tube heat exchangers

Shell-and-tube heat exchangers are a very well-known technological solution in the Industry and in Education. They are, indeed, one of those typical examples of heat exchangers one might be introduced to during heat transfer (or even thermodynamics) lectures. In fact, if someone visits random Industry-related websites, phrases like ”Shell-and-tube heat exchangers are one of the most popular types of heat exchanger because of their flexibility to accommodate a wide range of temperatures and pressures” [37] are not uncommon. As other thoroughly-used technologies, a considerable variety of them is offered by the Industry (fixed tubesheet, U-tube, floating head...), but such is the knowledge of this heat-exchanging solution that, these days, the design of shell-and-tube heat exchangers can be customised to match the different needs companies and individuals might have.

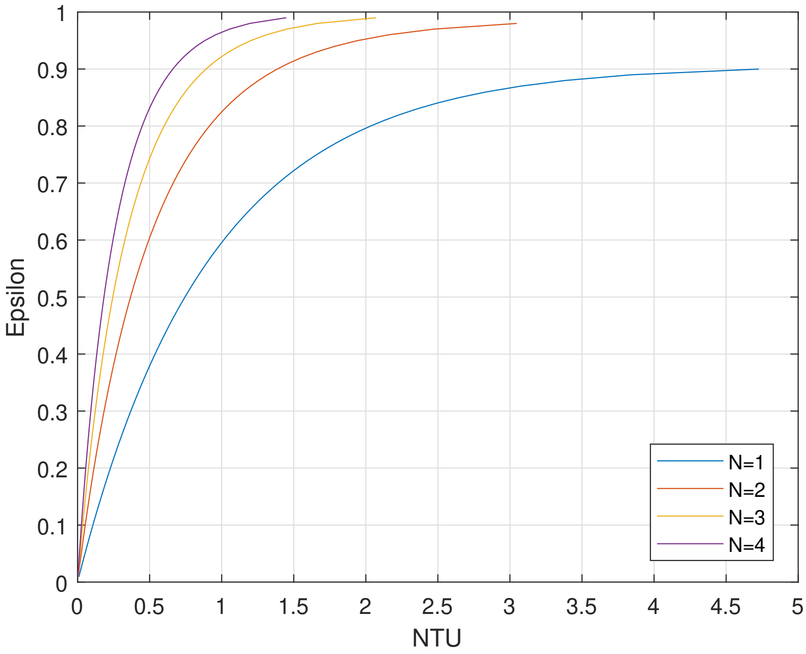


Figure 4.7: NTU vs ε for shell-and-tube heat exchangers with N_s=1,2,3 and 4 passes.

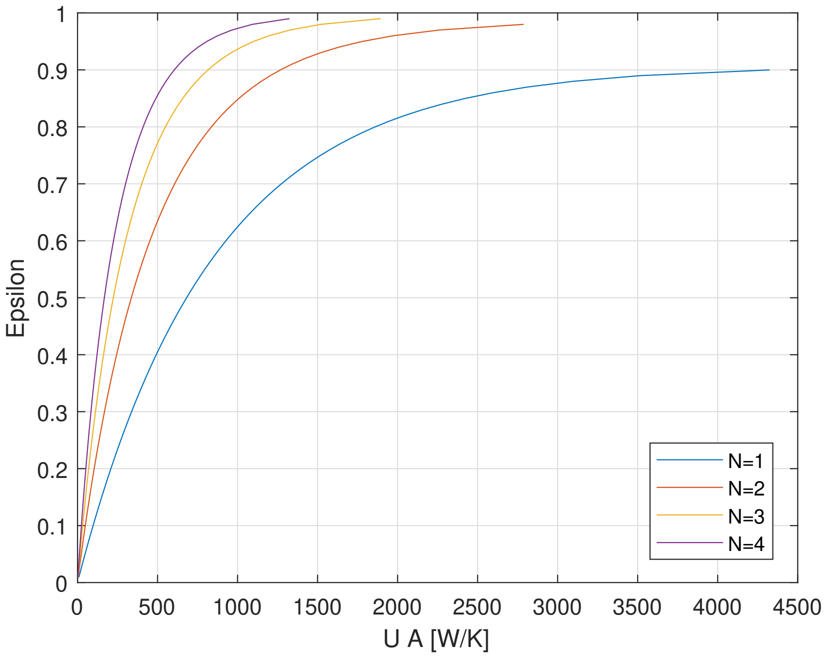


Figure 4.8: $U \cdot A$ vs ε for shell-and-tube heat exchangers with N_s=1,2,3 and 4 passes.

Another benefit of this type of heat exchangers is that their cylindrical design makes them significantly resistant to high pressures, hence their range of application becomes wider.

Figure 4.7 and Figure 4.8 are the corresponding representation of NTU vs ε and $U \cdot A$ vs ε for these heat exchangers. As it can be appreciated, as the number of passes (N_s) increases, so does the effectiveness for lower values of NTU and $U \cdot A$. The number of passes is, typically, chosen according to the situation: limited space, big shells, and so on. However, one must keep in mind eq.(3.43), in which it can be appreciated that increasing the number of passes also increases the pressure drop occurring inside the heat exchanger.

Figure 4.9 and Figure 4.10 are the result of combining all of the above into some charts.

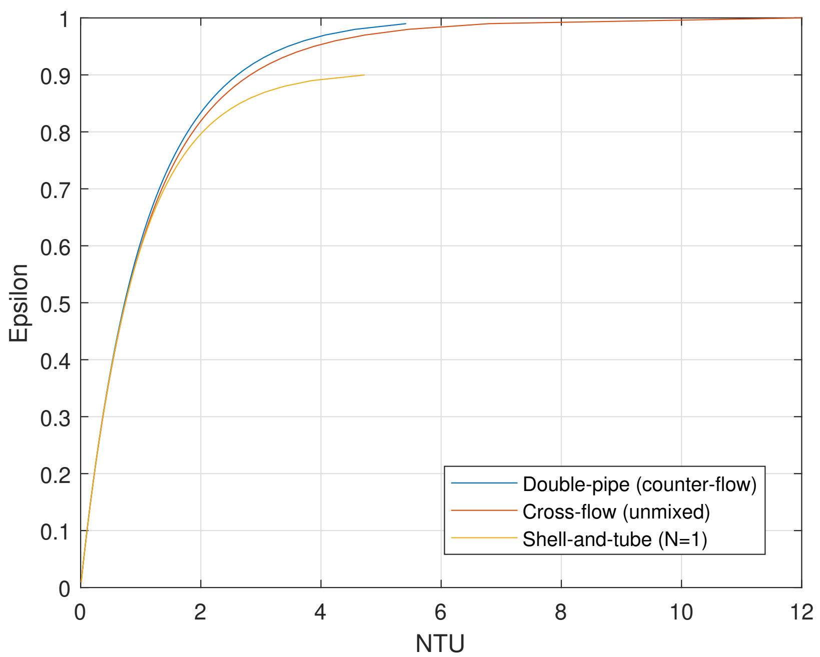


Figure 4.9: NTU vs ε for shell-and-tube heat exchangers with N_s=1,2,3 and 4 passes.

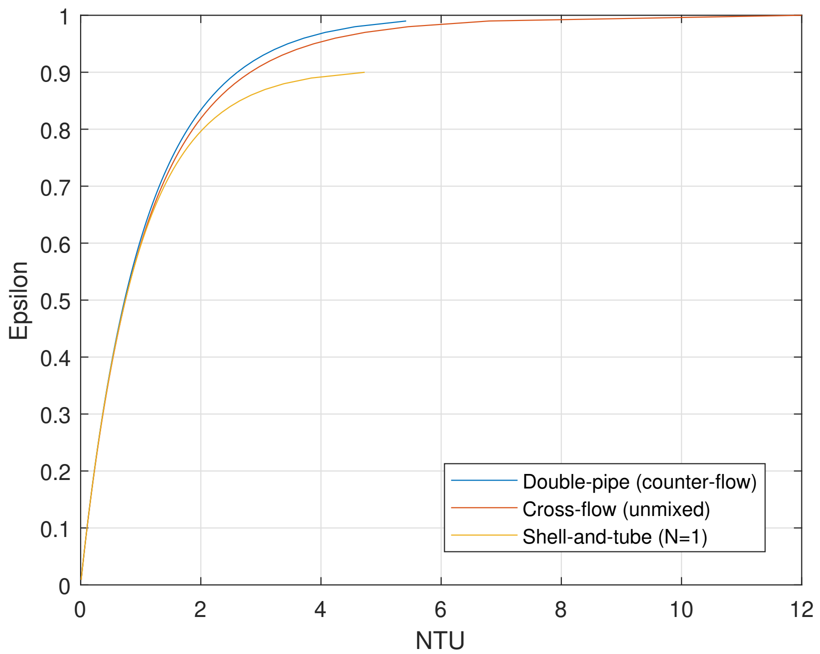


Figure 4.10: $U \cdot A$ vs ε for shell-and-tube heat exchangers with N_s=1,2,3 and 4 passes.

Nevertheless, these results must be accompanied by those from the pressure drop calculations to fully validate the code for its application in Turbomatch-based analysis.

4.1.4 Pressure drop results

Validating any pressure drop given by the code is quite a tricky task. The approach of this thesis is general, i.e., without getting into *too much* detail, and almost all the literature is focused on specific examples of heat exchangers, determining many geometrical parameters that have not even been mentioned in this project. Moreover, the conditions in which the heat exchangers of this research work are considerably extreme (very high and low temperatures, pressures measured in megapascals, and low mass-flow-rates.

As mentioned in Chapter 3, the expressions employed to calculate the pressure drop are already validated by the literature. However, papers like [39] might be used to contrast orders of magnitude to check if what the code gives as result is, indeed, believable.

Pressure drop in shell-and-tube heat exchangers

The following values have been taken from [39] to imitate as purely as possible the heat exchanger analysed in it:

- Tube inner diameter, d_i: 16 mm.
- Tube outer diameter, d_o: 20 mm.
- Shell inner diameter, D_s: 0.8 m.
- Tube length, L_t = 5 m.
- Number of baffles[1], N_b = 6.
- Tube pitch, $Pitch$ = 25 mm.
- Square pitch layout.

Temperatures in Dae et al. [39] are significantly *smoother*, around 230 K and 280 K. In order to maintain a certain fidelity to the aeronautical purpose of this thesis, the inlet temperatures suggested by NASA [1] (200 K and 791 K) have been respected. Additionally, inlet velocities[2] of around 6-8 m/s have been chosen for both fluids (again, inside the range of values commonly used in the literature aforementioned). Lastly, [39] suggests employing mass-flow-rates for the cold flow of around 25 kg/s, and 1000 kg/s for the hot. However, the values (suggested in [1]) of 0.351 kg/s for the cold fluid and 0.832 kg/s for the hot have been maintained. Although this could be remarkable when comparing the quotients $\Delta P_{drop}/P_{in}$, it is negligible for the absolute values, as demonstrated by [39] and shown in Figure 4.11 later on.

The code results after introducing the parameters mentioned above are:

- $\Delta P_{drop,h}$ = 908.23 Pa.
- $\Delta P_{drop,c}$ = 169095.1 Pa.

[1]This value is not explicitly specified in [39], but it is a typical value found in the literature, thus it has been considered representative.

[2]*Re* values have been adjusted to the suggestions of the paper.

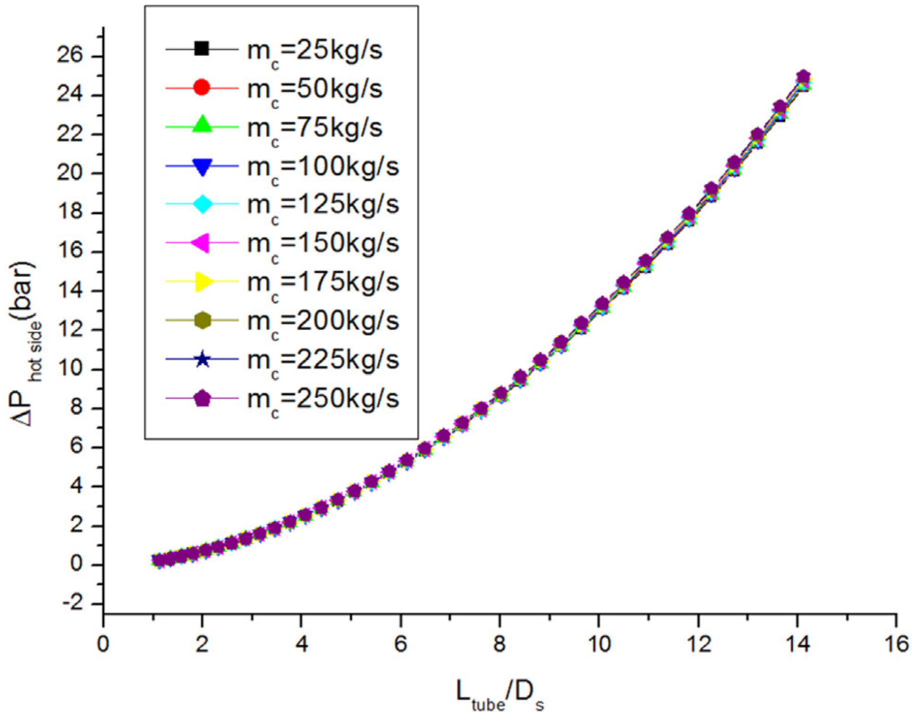


Figure 4.11: Pressure losses on the tube side (cold flow) according to tube length-shell inner diameter ratio for different mass-flow-rates. [39]

As it can be observed in Figure 4.11, the lower the mass-flow-rate, the slightly-lower pressure drops for low L_t/D_s values and the slightly-greater drops for high L_t/D_s values. In this case, the employed ratio is 5, and even without supposing that a line for a mass-flow of 0.35 kg/s would be a bit under the ones shown there, it can be seen that both [39] and the code suggest pressure drops on the cold side of around 200 kPa.

Pressure drop in cross-flow heat exchangers

This type of heat exchanger, is, without any doubt, the most difficult type to validate, because there is not a generic model as there is in shell-and-tube category; for each type, there are many specified parameters, and each model relies a lot in matters like whether plates are corrugated or not. In other words, it is hard to generalise. Considering microchannel cross-flow heat exchangers (probably the ones with the *simplest* geometry (in terms of easiness to characterise it), [41] presents an interesting study that matches quite reasonably the approach taken in this paper for this type of heat exchangers, such as low mass-flow-rates, prismatic channels and so on (see Chapter 3 for further information). Micro cross-flow heat exchanger are considered one of the main lines of investigation in the near future of the heat transfer world. If the reader wants to enhance its knowledge of them, Tsopanos et al. [40] might be a good starting point. The parameters described in [41] (and incorporated to the code) are as follows:

- Channel height, b and width, c: 490 μm.
- Plate dimensions, L: 50x50 mm.
- Number of channels (each), N: 31.

The rest of the parameters (mass-flow-rates, inlet temperatures and pressures, inlet velocities and Re) maintain the values given in shell-and-tube validation as [41] does not give too much information about it apart from some ranges, and those ranges are similar to the ones mentioned in [39]. Results obtained from the code are as follows:

- $\Delta P_{drop,h}$ = 2996.4 Pa.
- $\Delta P_{drop,c}$ = 2765.4 Pa.

The results presented[3] in [41] are compiled in Table 4.1:

Parameters	Elements: 7.86·E+5	Deviation [%]	Elements: 1.01·E+6	Deviation [%]	Elements: 1.18·E+6
ΔP_{drop} [Pa]	2558.0	2.9	2638.1	1.3	2665.5

Table 4.1: Study of grid independence: it shows ΔP_{drop} values obtained with different mesh densities. [41]

[3] In Table 4.1, ”Elements” stands for *Number of mesh elements*.

Pressure drop in double-pipe heat exchangers

As seen in Chapter 3, pressure drop calculation for this type of heat exchangers is the simplest of all. In fact, there are even online calculators that allow the user to introduce some values and, in return, the website gives the pressure drop. The tool from [32] has all the steps traced and referenced, hence it is considered trustworthy. Moreover, there is an example given in there, and although differences with respect to these thesis's parameters are similar to the ones mentioned for the other two types (*warmer* temperatures, flowrates and inlet pressures, among others), they are small and even negligible in some cases, the rest of the parameters are considerably similar, the geometric parameters to be defined are almost the same ones identified in this project and the resolution method is similar. All in all, [32] obtains a pressure drop of $\Delta P_{drop} \sim 2200$ Pa. The employed equations are not identical to the ones chosen for this project, and parameters like pipe roughness, which have not been considered in this research, are taken into account in [32].

The parameters introduced into the code are taken from a a paper from IJSTM [33]:

- Diameter of the inner pipe, d_i: 26 mm.
- Diameter of the outer pipe, d_o: 34 mm.
- Length of the inner tube, L_i: 1.2 m.
- Length of the outer tube, L_i: 1.2 m.

Consequently, the code gives the following pressure drop results:

- $\Delta P_{drop,h}$ = 3555.7 Pa.
- $\Delta P_{drop,c}$ = 2765.4 Pa.

As the reader might have observed, the results are quite similar but not identical. This might be due to different aspects already mentioned, probably being the general approach, i.e., lack of accuracy in parameter choosing and determination, the main cause of these differences. However, the order of magnitude is, indeed, identical, and that is considered more important for a generic method than accuracy down to the last pascal.

A pressure drop comparison between the three types makes no sense, and even less in the circumstances surrounding this thesis. The purpose of this Section was to demonstrate that the code is capable of emulating some experiments from the literature.

4.2 Results obtained from Turbomatch

It must be highlighted before continuing that all the employed ambient parameters have been obtained from NASA's report [1] and from the International Standard Atmosphere (ISA) [44]. The chosen engine has been the Rolls Royce Avon, the first turbojet Rolls-Royce produced. Although it is not precisely an example of modernity, state-of-the-art of clean emissions, or even a turbofan, it is simple to model and the intention of this thesis is to build a solid basis for future investigations, in which there is enough room for all the desired complexity; it is better not to run when one is learning how to walk. Engine parameters have been left as default. Explaining how Turbomatch works is beyond the scope of this thesis, but in [38] there is plenty of information. In terms of Combustor Outlet Temperature (COT), the following values have been selected for the off-design evaluation: COT = 1400 (design point), 1500, 1600 and 1700 K.

First of all, an Avon working on kerosene is modelled, to serve as a guideline. Secondly, the same simulation is done but this time the fuel type is changed to LH_2. Results are analysed and turbine inlet and outlet temperatures are noted[4]. Lastly, two simulations are made: one with a heat exchanger cooling the turbine cooling air, and one with

[4]This is done to particularise Matlab code to the current situation, so the appropriate pressure drops and effectiveness can be calculated for both engine + heat exchanger configurations mentioned throughout Chapters 2, 3 and 4.

a heat exchanger right before the nozzle entrance. Because of the design of both Matlab- and Turbomatch-developed codes, the user could imitate these simulations for each of the heat exchanger types that have been presented here and introduced in Matlab code's structure by just selecting-deselecting a couple of code lines and clicking on "Run" on both programmes.

Nevertheless, only results for one of the most promising technologies (microchannel unmixed cross-flow heat exchangers) have been attached here to avoid redundancies in this text. The parameters selected for this heat exchanger are taken from all of the above and from [1]:

- $b = c = 500\ \mu$m.
- $L_c = 10$ cm.
- $L_h = 50$ cm.
- Number of cold channels = Number of hot channels = 31 (each).

Turbine cooling air heat exchanger case:

Speeds are kept in between logical values of ~ 5 m/s. With all this data, a value for $U \cdot A$ of 1270 W/K was fixed because it is a reasonable value [11] for which reasonable[5] effectiveness of $\varepsilon \sim 70$ % and NTU of ~ 1.4 is obtained.

Nozzle heat exchanger case: To match the situation described in Chapter 2, hot side inlet speeds of this heat exchanger need to be much bigger. In fact, NASA calculated that $v_h \sim 100$ m/s would be feasible. $U \cdot A$, ε and NTU maintain the same values from last case.

[5]According to [13], industrial applications of cross-flow heat exchangers reach values of up to 75 %.

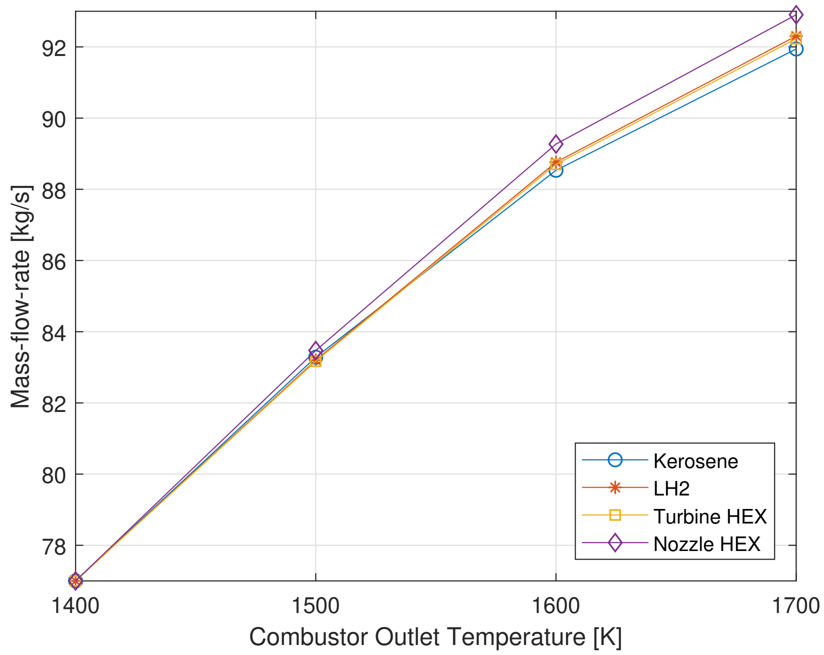


Figure 4.12: Mass-flow-rate ($\dot{m}$) results obtained from Matlab + Turbomatch.

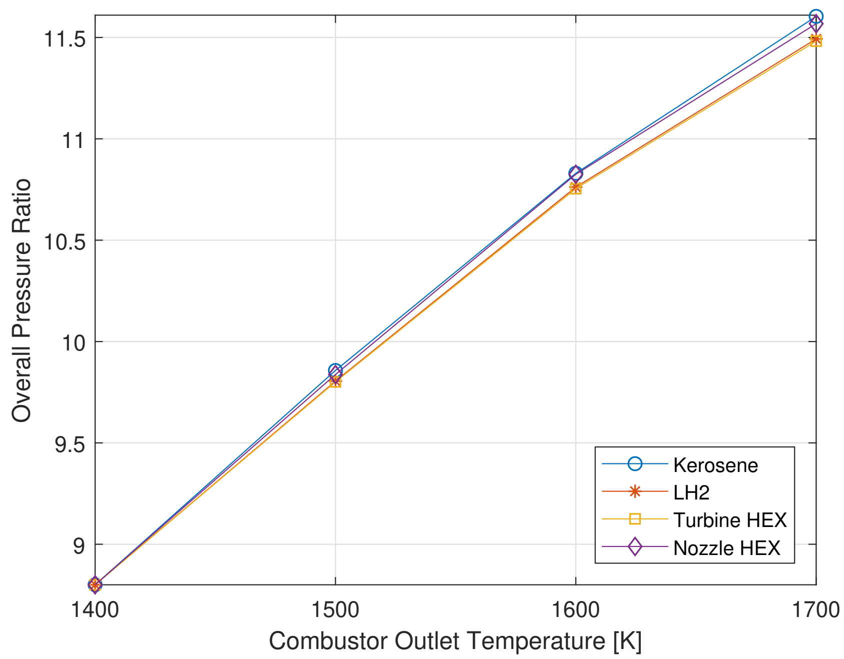


Figure 4.13: Overall pressure ratio (OPR) results obtained from Matlab + Turbomatch.

As one might have foreseen, mass-flow-rates and pressure ratios behave very similarly.

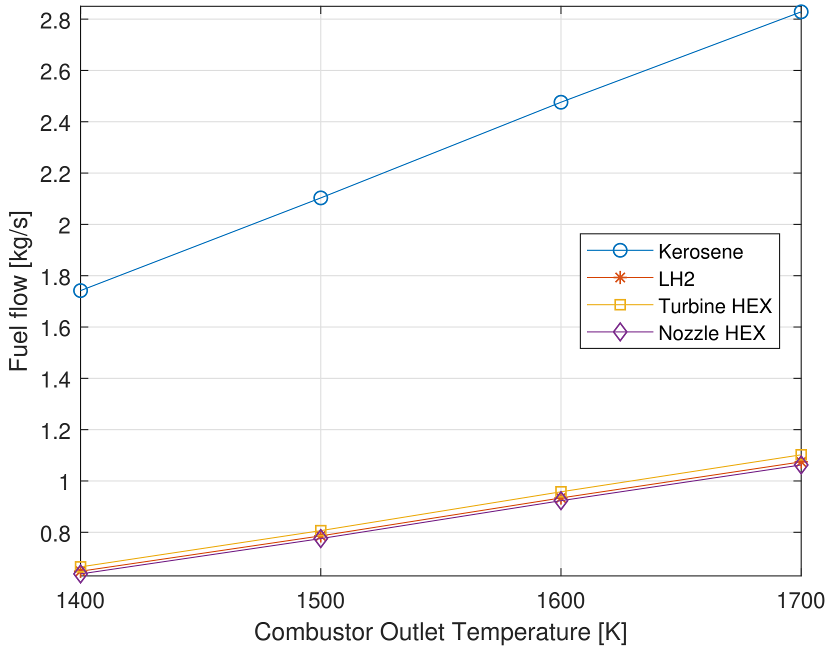


Figure 4.14: Fuel-flow-rate ($\dot{m}_f$) results obtained from Matlab + Turbomatch.

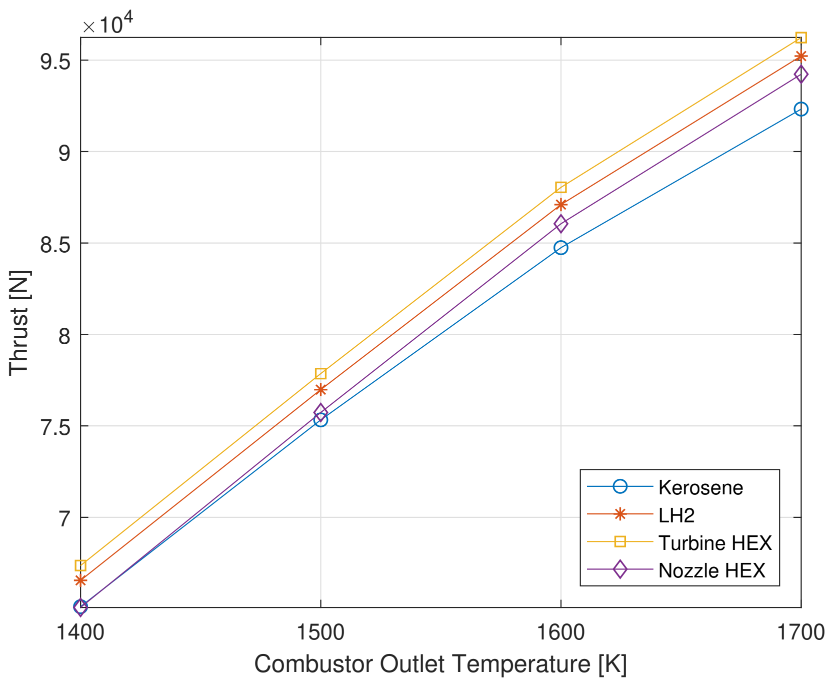


Figure 4.15: Thrust (F_n) results obtained from Matlab + Turbomatch.

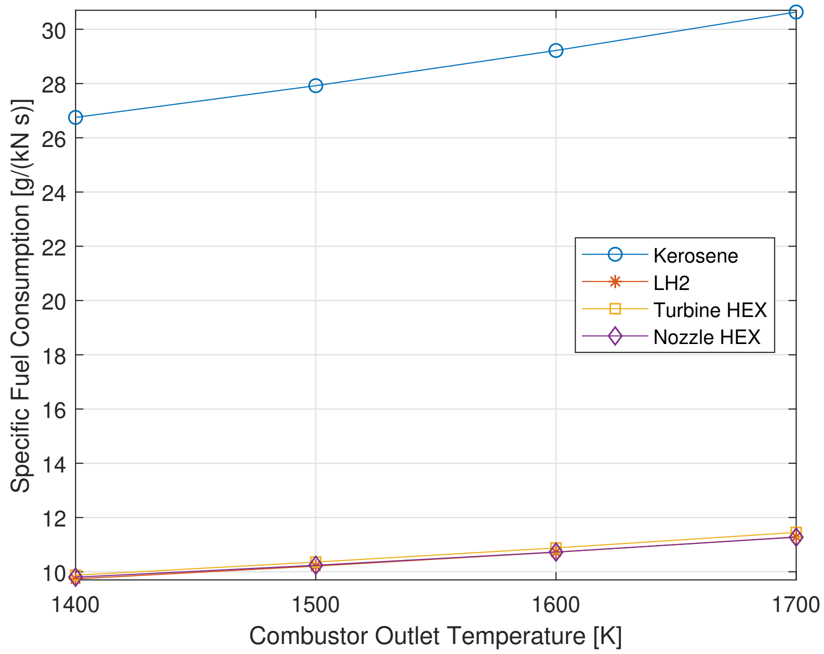


Figure 4.16: Specific fuel consumption (SFC) results obtained from Matlab + Turbomatch.

From Figures 4.12, 4.13, 4.14, 4.15 and 4.16 several conclusions could be extracted. Probably one of the most important ones is that these results resemble very significantly the ones shown (and hundreds of times cited) in [34]. As both this thesis and [34] assert, fuel flow and SFC would show considerably lower values if LH_2 would be introduced in already invented engines (see Figure 4.14 and 4.16).

A second remarkable matter could be discerned in Figure 4.15, where the effect of the different heat exchangers can be assessed. The turbine-cooling heat exchanger is indirectly producing an increase in thrust values. Explaining the theory behind this lies beyond the scope of this project, but [38] offers a complete review of gas turbine performance. The nozzle heat exchanger is slightly reducing the thrust; exactly as predicted, because the main purpose of this heat exchanger is to warm up the LH_2, i.e., to control the injection temperature of the fuel. To do so, it has been located directly inside core's flow interfering it, hence some loss of performance is inevitable. Furthermore, no studies from combustion chamber side have been included in this thesis, mainly because of time constraints, but it would be easier to study the performance of this heat exchanger in particular if they would be included in the future. Turbomatch gives the opportunity to declare a value for fuel's temperature.

The last conclusion is quite linked to what has just been mentioned: in general, it is complicated to know whether the results obtained from adding heat exchangers to the cycle are true to facts because of several reasons, some of them listed below:

– No more relevant data from the impact of the heat exchangers can be obtained from Turbomatch excepting fuel's injection temperature.

– To this day, there has been no chance to check the code behind Turbomatch's heat exchanger bricks, so it is difficult to know if these bricks are behaving properly or if some improvements to them would be feasible.

– Literature concerning this matter is still little or non-existent. Apart from [34], no papers to contrast these results have been found.

– In [34], the variation of thrust (ΔF_n) is studied, but the control parameter is not the combustor outlet temperature; it is the injection temperature of the LH_2. Thus, the smaller impact of one heat exchanger or another shown here is not necessarily wrong, simply the compared parameters are not the same.

4.3 Results obtained from thermal inertia calculations

The assumption of considering the specific heat constant in heat exchanger investigation is a common characteristic among specialised literature. This tradition might come from the often negligible variations ρ and c_p present when using air or water as working fluids in heat exchanging studies. However, when employing cryogenic fluids like LH_2 for these matters, every parameter in eq.(2.1) experiences a considerable change[6]. Hence, it is believed the aforementioned is not applicable when working with this type of fluids.

As a reference, Table 4.2 allows the reader to check the actual difference between all the parameters from eq.(2.1) at a glance. Isobaric conditions were considered for these calculations. Temperatures follow no strict rules and all the values have been obtained from [20].

	T [K]	k [W/(m·K)]	ρ [kg/m^3]	c_p [J/(kg·K)]	I [J/(m^2·s$^{1/2}$·K)]
LH_2	33	0.087	47.867	39312	403.43
Water	293	0.598	998.88	4179.8	1580.68
Ammonia	200	0.805	728.62	4222.6	1573.54

Table 4.2: Analysis of eq.(2.1) for different fluids.

An attempt to consider some variations from thermal conductivity (k) was done in [2], where, at the moment of choosing the material for the heat exchanger later designed in that project, thermal conductivity of different materials is assessed to comply with LH_2's particular thermal conductivity. In NASA's report [1], no mention is done to the method they follow to choose the materials of the heat exchangers described there. Getting into an analysis of the properties of different materials is beyond the research done for this thesis, but [2] offers some sections in which this issue is discussed.

4.3.1 Implementation considerations

Last column reflects the noticeable difference these fluids' thermal inertia presents. Again, it must be highlighted that in the case of LH_2, c_p varies significantly, and this directly affects thermal inertia. Thus, assuming the specific heat is constant might be a considerable

[6]These changes can be consulted in [20] by simply selecting liquid hydrogen from the list on the screen and introducing a wide range of temperatures for a certain pressure.

source of inaccuracy in many thermodynamic calculations. In other words, the method employed for heat exchanger design and optimisation must be capable of adapting itself to this conditioning characteristics of cryogenic fluids like LH_2.

Creating a database from some values given in [20] could be a good starting point to implement these variability in the code. If not, the user should be asked to manually introduce many different parameters that change according to the chosen temperature and pressure ranges. As it is such a big effort to extract every single number from the charts of [20], interpolations of some kind could work.

Chapter 5

Conclusions and Further Work

In this thesis, a literature review to address the current situation of liquid hydrogen (LH_2) in aviation has been carried out, highlighting the need of a change because of the environmental crisis many times mentioned throughout these last years. This change shall come from research and development of novel technologies to, e.g., employ different fuels. It is in this area where liquid hydrogen sets itself up as one of the most powerful candidates to drive the Industry (and, in general, the society) towards a greener future. To do so, it is not as simple as just switching from traditional fuels to LH_2. New fuel systems need to be developed in order to cope with the extraordinarily low temperatures and considerably high pressures hydrogen must be on in order to stay liquid. These low temperatures, on the other hand, open the door to technological solutions to take advantage of them to improve the efficiency of the whole propulsion system. Heat exchangers could be key in this. However, their compatibility with this cryogenic fuel proposal needs to be studied.

The purpose of the created (Matlab) code is double: first of all, to serve as a quick, easy option to check which type of heat exchanger could meet user's and/or situation's requirements; secondly, to avoid a limitation Turbomatch possesses, and just by extracting a small set of values among all of the parameters analysed by this Matlab code, this *hole* in Turbomatch is jumped. These two reasons are strong enough to consider a general approach for the code as the best option for this thesis. A validation of this code has been conducted, demonstrating that it possesses enough capabilities and potential to emulate different types of heat exchangers in a thorough, yet not entirely accurate way (because of the generic nature of the approach chosen for this thesis).

Rolls-Royce's Avon engine has been the model for the simulations undertaken in this project because, albeit *simple* when comparing it to modern turbofans, it is more than enough to check if, firstly, heat exchangers using LH_2 could be introduced into a Turbomatch script (with the help of the code aforementioned), and secondly to perform a preliminary study of the impact different heat exchangers (in different positions of the engine) might produce. These simulations have revealed the promising characteristics of using hydrogen as a fuel for aero gas turbines, but also imply that further research needs to be done, probably from both Turbomatch-inlet-code and Matlab-code sides, to fully understand what real effects could be expected from each type of heat exchanger and their different possible locations inside the engine. The effect of the injection temperature has proved to be very important, hence further research is required.

Besides, a preliminary analysis of thermal inertia has been included in which each parameter has been briefly addressed and some representative values of them have been added. Those values show significant variations that need to be explored to assess the impact of this parameter.

All in all, results show that, on the one hand, the code acts as a solid basis for future investigations in this field, where there is enough room for the addition of complexity (what translates into accuracy if it is implemented consistently); on the other hand, Turbomatch limitations could be avoided by undertaking a solution like the one proposed in this thesis to obtain trustworthy preliminary results. It must be added that, without assessing the internal structure of Turbomatch and its heat exchanger *bricks*, it is impossible to conclude if there is any inaccuracy coming from it.

5.1 Further work

Throughout this paper, some key points have been identified as potential paths to follow the research started by this thesis. As a summary, the following list is included to serve as a guide for future developers of this study:

- A more developed pressure drop calculation method is recommended, in which complexity is added to improve the accuracy of the whole project. Thoroughly investigate the different correlations proposed in the literature to better identify the ones that suit this investigation best.
- Introduce the c_p variation discussed in Chapter 4 by creating a library of values of the parameters from eq.(2.1). Values could be extracted from [20].
- Further research on the impact of thermal inertia is considered necessary.
- Adding the effects of roughness in pipes and walls would give credibility to the code, and the results would gain extra accuracy.
- Developing a user interface for the code is feasible and would ease the utilisation of the created tool.
- Including new types of heat exchangers and new geometrical ways to define the already included is recommended to give as much flexibility as possible to the user.
- Investigating the internal code of Turbomatch would be desirable; its heat exchanger bricks are not very popular, in fact there is no example of their utilisation in [38]. Problems in their code would instantly blur any achievable result.
- Integrating the code created in this thesis into Turbomatch to work as one would save time to the user.

– It would be optimum to repeat the simulations, but adding the effects of varying the injection temperature in Turbomatch input files. These injection temperatures, to this day, should be an output from the code when simulating the nozzle heat exchanger (the one which controls the heating of the LH_2 up to the desired point).

– Currently, new literature related to this field is being published every day. A new literature review should be conducted to update assumptions, expressions or methods employed in this thesis.

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