

CRANFIELD UNIVERSITY

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**Preliminary Thermodynamic Assessment for
Advanced-cycle Marine Gas Turbines**

School of Aerospace, Transport and Manufacturing

Ph.D. Thesis

SCHOOL OF AEROSPACE, TRANSPORT AND MANUFACTURING

Full-time Ph.D.

Academic Years 2011–2014

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Preliminary Thermodynamic Assessment for Advanced-cycle Marine Gas Turbines

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December 2014

This thesis is submitted in partial fulfilment of the requirements for the degree of Doctor of Philosophy

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*A mi familia;
y en especial a mi abuela Mérita,
mi apoyo más incondicional y más indulgente.*

Executive Summary

Nowadays, around 80% to 90% of the international trade is made by sea, and the vast majority of the large merchant ships are powered by means of low-speed two-stroke diesel engines. Currently, gas turbines have only found niche markets in certain marine applications —such as the military or fast ferries, in which reduced size, mass, noise and vibration levels of the power plant are of paramount importance—, but their capability for entering the sector of the large cargo vessels (containers, bulk carriers, etc.) remains unexplored. Main disadvantages for the gas turbines to become a viable option for powering these ships are the very high thermal efficiency and extremely low running costs that the large diesel engines actually offer. However, as long as the current trend continues, diesel power plants are likely to lose one of their major advantages —i.e. their low operating costs— due to the ongoing regulations regarding ship emissions (IMO's MARPOL 73/78 Annex VI) and the severe reduction in the established limits for Sulphur contents in fuels for marine applications. In such a scenario, in which emissions from ships are relentlessly being cut down lower and lower, diesel engines will have to move to more expensive distillate fuels, likely equalising their fuel prices with the gas-turbine-based power plants and, therefore, opening a potential window for the latter to enter the market.

Hence, the main question posed at this point would be: given the expected equalisation in fuel prices between the gas turbines and the diesel engines, is it possible to devise an *advanced* gas-turbine-based power plant that could outcompete the low-speed two-stroke diesel engines as prime mover for marine commercial applications? Working with this hypothesis of fuel prices no longer posing a competitive advantage for diesel engines, and making the simplifying assumption (as a sensible first-order approach to the problem) that the targeted marine applications spend most of their time at cruise speed —i.e. with the prime mover operating very close to its design point—, the overall thermal efficiency at nominal conditions will be the main driving parameter to carry out the comparative assessment among the different power plants considered herein. The aim of this research is, thus, to investigate if and which sort of *advanced* gas-turbine-based power plants could be able to deliver thermal efficiencies high enough to outperform modern diesel engines in this regard, at least at design-point operation. The research comprises two markedly different phases: in phase one of this project, a wide range of candidate gas-turbine-based power plants are proposed and parametrically optimised for maximum thermal efficiency at design point, so that their relative merits against one another can be established consistently and the few most promising cycles can be downselected for further research; in phase two, these few downselected candidates will be physically characterised, carrying out preliminary size and mass estimations of the whole prime mover, along with some acquisition costs assessment.

The results presented in this thesis demonstrate the great potential of certain *advanced* thermodynamic cycles, still based on a gas turbine as the *core* of the power plant, to achieve outstanding levels of performance at design point, effectively outcompeting the diesels in this regard by a good margin, potentially between 5 and 10 percentage points. In particular, the evaporative gas turbines and the combined-cycle power plants were the most promising candidates analysed, yielding thermal efficiencies at design point in the surrounds of 60%–65%. As it will be shown, these *advanced* gas-turbine-based power plants are, in general, comparable in size to the diesel engines, but can mean huge reductions in the overall *uninstalled* dry mass of the prime mover. Despite the acquisition costs being significantly higher for the gas turbines, the great potential savings in fuel consumption —because of the higher thermal efficiencies— could make out of these *advanced* gas-turbine-based power plants a *cost-effective* solution in the long run for powering large merchant ships.

Acknowledgements

This work would not have been possible without the crucial contribution of many people, to whom I am deeply grateful.

Firstly and foremost, I want to thank Prof. Riti Singh and Dr. Vassilios Pachidis, for trusting me with the capability and giving me the opportunity of undertaking this Ph.D. research at Cranfield University. Without them, this whole experience would have never happened. All my gratitude goes as well to my supervisor Dr. Pavlos Zachos, for guiding me, for the many lessons I have learnt from him and just for dealing with me on an almost daily basis. I am not always easy.

I would like to very much acknowledge the valuable input received during the whole duration of this project from my industrial advisors Mr. Julian Groom, Mr. Paul Fletcher and Mr. Thierry Moes, whose comments and questions have always helped and encouraged me to do more and better research.

Last, but not least, a big thank you goes to my colleagues at the Department of Power and Propulsion and within the Rolls-Royce Performance U.T.C., especially to Aleixo González and Carlos Soria, for the uncountable hours spent discussing on technical and non-technical matters alike; they have helped to shape both my project and myself. And, of course, to the M.Sc. students that have participated in parallel research projects within the broader activity related to the marine environment Mr. Adam Doligalski, Mr. Giovanni Brighenti, Ms. Giulia Rossato and Mr. Syed Mashruk; they have managed to add much value to the completeness and quality of this work and I am certainly indebted to them: in spite of me being their supervisor, they have definitely taught me a lot during our different collaborations.

A Ph.D. is a big endeavour, and during these last three years, and the many motivational ups and downs I have gone through, I have had the unconditional support of my family and friends. Rather than just calling their names, I hereby want to express my most sincere gratitude and my resolution and firm commitment to make every effort possible just to ensure they actually realise what their support really meant to me, and the huge difference they made. I very much look forward to a chance for returning to them at least some of what I have been given.

To all of them, *thank you*.

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Abbreviations

A

ABC Air Bottoming Cycle.

AC aftercooler.

ASME American Society of Mechanical Engineers.

B

BPVC Boiler and Pressure Vessel Code.

C

CCGT Combined-Cycle Gas Turbine.

CCPP combined-cycle power plant.

CDT compressor delivery temperature.

CEPCI *Chemical Engineering* Plant Cost Index.

CHAT Cascaded Humidified Advanced Turbine.

CO carbon monoxide.

CO₂ carbon dioxide.

E

EC “exhaust-cooler”.

ECA Emission Control Area.

EGT exhaust gases temperature.

EvGT Evaporative Gas Turbine.

G

GA genetic algorithm.

H

HFO heavy fuel oil.

HP high-pressure.

HRSG heat recovery steam generator.

I

IC intercooler.

IFEP integrated full electric propulsion.

IMO International Maritime Organisation.

L

LP low-pressure.

M

MGO marine gas oil.

MOO multi-objective optimisation.

N

NO_x oxides of nitrogen.

O

O₃ ozone.

OPR overall pressure ratio.

P

PM particulate matter.

PPTD ‘pinch’ point temperature difference.

PR pressure ratio.

S

SBC steam *bottoming* cycle.

SECA SO_x-only Emission Control Area.

SO_x oxides of sulphur.

SOO single-objective optimisation.

STIG Steam Injected Gas Turbine.

T

TBC thermal barrier coating.

TIT turbine inlet temperature.

TRL *technology readiness level*.

V

VOC volatile organic compound.

Nomenclature

A	Area
C	heat capacity ($m \cdot c_p$)
c_p	specific heat capacity at constant pressure
D_h	hydraulic diameter
d	diameter
F	packing factor of a humidifier
f	Fanning friction factor
G	core mass velocity ($\rho \cdot u$)
G'	mass flow rate of the mixture gas stream in a humidifier
H	total enthalpy
H^+	enthalpy invariant of the saturator
h	specific enthalpy (Chapter 6, Section 8.5, and Appendix B) heat transfer coefficient (Chapter 8 except Section 8.5, and Appendix D)
h'	height
HTU	necessary height for a transfer unit
j	Colburn factor
$K_{c,e}$	contraction, resp. expansion, loss coefficient
k	thermal conductivity
L, l	length
L'	mass flow rate of the liquid stream in a humidifier
M^+	mass invariant of the saturator
m	mass flow
N	number of . . .
NTU	number of transfer units
Nu	Nusselt number
P	power
p	pressure
p'	pitch
Q	heat
PR	pressure ratio
Pr	Prandtl number
R	specific gas constant
R'	heat transfer resistance
Re	Reynolds number
r	radius
s	specific entropy
s'	separation
T	total temperature
U	overall heat transfer coefficient
u	velocity
W	ratio of lower to greater heat capacities in a heat exchanger
$X_{l,t}$	spacing between adjacent rows, resp. tubes, in a <i>tube & fin</i> heat exchanger
Z	overall packing height of the humidifier
γ	heat capacity ratio

Δ	“increment”
δ	thickness
ϵ	effectiveness
η	efficiency
Θ	temperature difference between the cold and the hot side of a heat exchanger
μ	viscosity
ρ	density
σ	ratio of core <i>free flow</i> area to <i>frontal</i> area
Φ	relative humidity
ω	specific humidity (water to dry-air ratio)

Subscripts

0,1,2	station numbering for the humidifier
1,2,3	main dimensions of a heat exchanger
AS	Adiabatic Saturation conditions
a	dry air
c	cold side of a heat exchanger
e	extended surfaces of a heat exchanger
f	fins, extended surfaces of a heat exchanger
fg	vaporisation process
g	mixture gas (dry air plus water vapour)
h	hot side of a heat exchanger
i	in, inner, inlet
is	Isentropic
m	mean
O	overall
o	out, outer, outlet
p	Polytropic (Chapter 2) <i>free flow</i> passage (Chapter 8 and Appendix D)
s	offset strips in a <i>plate & fin</i> heat exchanger
sat	saturation conditions
v	water vapour
w	liquid-phase water (Chapter 6, Section 8.5, and Appendix B) wall (Chapter 8 except Section 8.5, and Appendix D)

Chapter 1

Introduction

1.1 Background

Nowadays, around 80% to 90% of the international trade is made by sea [3]. According to the sponsor's Annual Report [2] assessing the economical balance of the year 2010, the marine sector provided in this year revenues which are worth almost £2,600m, about 24% of the company's total revenues. The total profit that this sector yielded in the year 2010 is over £330m, representing about 31% of the company's total profits for that particular year. This meant an increase —with respect to the profits that the marine sector yielded in the year 2009— of around 26%, and made the marine become the second most important sector for the company in terms of total revenues (see fig. 1.1), ahead even of the defence aerospace sector and just behind the civil aerospace.

Furthermore, the marine merchant market was responsible for over 20% of the total marine sector profits. Current estimates by the sponsor foresee that the whole marine sector market will be worth around US\$340b over the next 20 years [2], out of which US\$215b will be in the form of power and propulsion systems demand alone. The market of the large cargo vessels has grown significantly over the last years and, if the current trend continues, is expected to represent over 40% (see fig. 1.2) of the marine sector worth —this is, around \$140b on its own— over the next 20 years.

Currently, gas turbines have found niche markets in certain marine applications —such as military or fast ferries, in which reduced size, mass, noise and vibration levels of the power plant are of paramount importance—, but their capability for entering the sector of the large cargo vessels (containers, bulk carriers, etc.) remains unexplored. In these applications, the emphasis is placed upon low acquisition and operating costs of the power plant, and hence the vast majority of the large merchant ships nowadays are powered by means of low-speed two-stroke diesel engines [49, 50, 51]. Main advantages of these prime movers being [49, 50, 51, 36]: *a)* their high efficiency, somewhat around 20% to 25% greater than any simple-cycle gas turbine currently in use in the marine environment; *b)* their capability to run in residual heavy fuel oil (HFO), which can be cheaper by around 40% than the distillate fuels used for gas turbines; and *c)* the relatively low initial investment they require when compared to the gas turbines, which can represent about 20% savings on the purchasing costs. However, as long as the current trend continues, diesel power plants are likely to lose one of these three main advantages —i.e. their capability to run in extremely cheap residual fuels— within the next few years, due to the ongoing regulations regarding emissions from ships.

Exhaust emissions from diesel engines can be separated into several compounds, that have different causes for being produced and different effects on the environment [49, 50, 51, 101, 70, 40, 41]:

Carbon dioxide (CO₂) is formed whenever hydrocarbons combustion takes place and, therefore, will always be present in the exhaust gases of any power plant burning fossil fuels. CO₂ emissions can only be reduced, therefore, by improving the efficiency of the power plant, so that less fuel is required for satisfying the power demand. CO₂ is a major greenhouse effect gas.

Water vapour is produced by the oxidation of Hydrogen —always present in fossil fuels—, so it will always be present within the exhaust gases. However, except when emitted at high altitude,

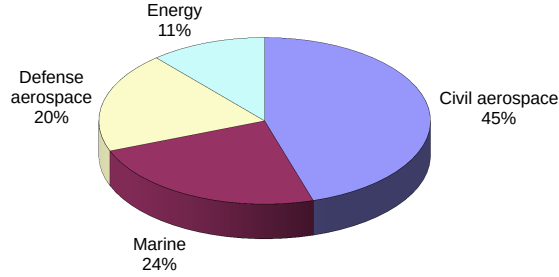


Figure 1.1: *Sponsor's total revenues by sectors in the year 2010, after [2]*

water vapour is not a major pollutant and its emission levels from ships are, therefore, not a major concern for the environment.

Oxides of nitrogen (NO_x) are formed by means of Nitrogen —always present in the air— and excess Oxygen —which is usually needed in the cylinder in order to ensure that the fuel is burnt completely— reacting at high temperatures. NO_x are a major pollutant, being responsible for the formation of tropospheric ozone (O_3) by means of a photolytic reaction [101], favoured by the presence of volatile organic compounds (VOCs), which are the major catalyst for the reaction and are also part of the exhaust emissions of diesel engines. O_3 in the troposphere is a powerful oxidant, which can have severe adverse effects on human health, such as acute irritation of skin, respiratory diseases, decreased immune functions or inflammation of lung tissue.

Oxides of sulphur (SO_x) are produced by means of the oxidation of the Sulphur content of the fuel burnt and, thus, its emission levels are only dependent on the kind of fuel being used. Residual fuels such as HFO have very high Sulphur content, though, making out of the SO_x emission levels from diesel engines a major concern, especially in the coastal areas of the Northern hemisphere. SO_x are responsible for the acidification of soils, waters and forests. This acidification process results in soils lacking the essential nutrients and in “heavy metals” —which are toxic for human beings and other animals— being released in soils and being transferred later on to the plants and other animals higher up in the food chain.

Carbon monoxide (CO) results from the incomplete combustion of hydrocarbons. CO emission levels can be reduced by simply ensuring that excess Oxygen is available in the cylinder, so that all the hydrocarbons are completely burnt. But this, as explained before, increases the levels of NO_x emissions. The trade-off between these two is not so clear, CO being such a dangerous pollutant, with great toxicity that can result in lethal poisoning levels even at very low concentrations.

Particulate matter (PM) is released because of unburned carbon deposits and lubricating oil in the cylinders. Good mixture of the fuel in the cylinder can significantly reduce the emission levels of PM, although most of this comes from the unburned lubricating oil. PM is a major concern, since long exposure for human beings, especially to the smallest particles, are the cause for cardiovascular mortality, respiratory diseases and lung cancer.

With all this in mind, governments are more and more concerned about the emission and pollutant levels. Within the coastal areas of the Northern hemisphere, shipping is one of the biggest sources

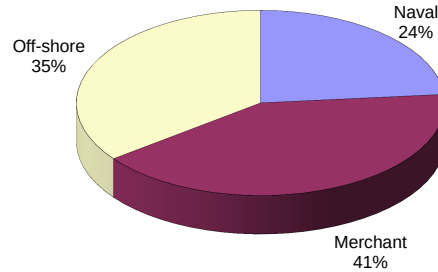


Figure 1.2: Market distribution over the next 20 years within the marine sector, after [2]

of emissions [49, 50, 51, 40, 41], responsible for around the 15% of the global NO_x , between 5% and 8% of the global SO_x and about 3% of the global CO_2 , mostly concentrated close to the biggest commercial ports in Europe and America.

For this reason, the International Maritime Organisation (IMO) –through the MARPOL 73/78 Annex VI amendments that came with the 1997 Protocol [56], which entered into force in 2005– has set the rules of the admissible emission levels depending on the class and tonnage of ships. Tier II/III of this protocol were adopted in 2008 and entered finally into force in 2010. Along with the standard limits, some Emission Control Areas (ECAs) are designated, in which NO_x and SO_x emission limits are further reduced (see figs. 1.3–1.4), for the huge amount of maritime traffic that these regions usually have and after being proved sufficiently that marine emissions play an important role on the pollution levels of these coastal areas.

There are 4 established ECAs nowadays [56, 32, 108]. Namely:

- Baltic Sea: was adopted in 1997 and came into force in 2005, it is a SO_x -only Emission Control Area (SECA), where the more stringent limits for NO_x emissions specified for “standard” ECAs do not apply.
- North Sea: came into force in 2006, being also a SECA.
- North American ECA: controlling both SO_x and NO_x emissions, was adopted in 2010.
- US Caribbean ECA: also controls both SO_x and NO_x emissions, and was adopted in 2011.

In such a scenario, in which emissions from ships are relentlessly being cut down lower and lower, and especially because of the SO_x emissions control—which will force Sulphur content in marine fuels to be lower than 0.5% globally by 2020 and could even result in the potential banning of HFO—, diesel engines will have to move to more expensive distillate fuels or to make huge investments in heavy after-treatment devices—such as selective catalytic reduction or plasma based catalytic treatment [85, 75]— and, therefore, a potential window for the gas turbines to enter the market of the large cargo vessels could be opened [49, 50, 51].

Hence, the main question posed at this point of the research would be: given the likely equalisation in fuel prices between the gas turbines and the diesel engines that could be expected when emissions regulations come into force within the next few years, is it possible to devise an *advanced* gas-turbine-based power plant, with significantly higher thermal efficiency than current simple-cycle marine gas turbines, that could outcompete low-speed two-stroke diesels as prime mover for marine commercial applications?

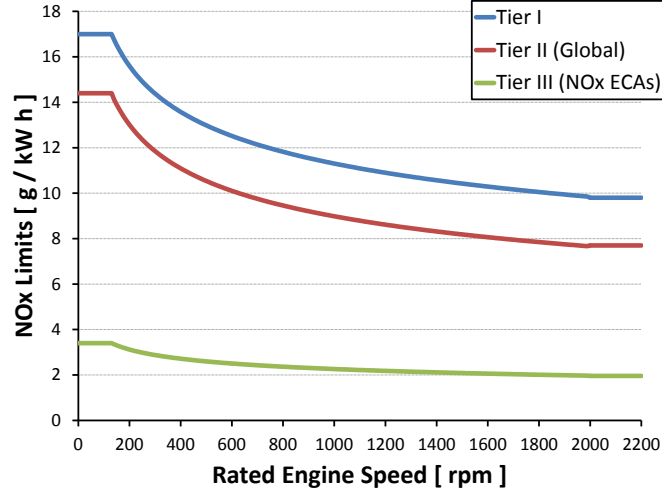


Figure 1.3: NO_x emission limits against rated engine speed, after [32]

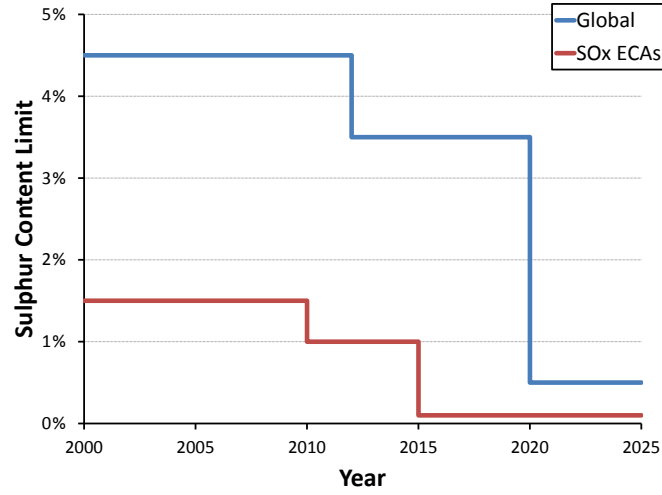


Figure 1.4: Sulphur content limits in fuel in the following years, after [32]

1.2 Aim & Objectives

As a sensible first-order approach to simplify the comparative assessment of the performances of such different power plants —i.e. gas turbines vs diesel engines—, an important assumption will be made: nominal thermal efficiency of the prime mover at design-point operation will be considered the main parameter driving the overall running costs of the power plant. Large cargo ships will have different power requirements from the prime mover along any given trip, due to the varying weather conditions they can meet, or to accommodate their speed to the maritime traffic requirements or to manoeuvre at ports. However, it will be assumed that these variations will either be small or take place during very short periods of time (compared to the overall length of the trip), whilst the vessel will spend most of the time at cruise speed, with its prime mover operating very close to its design point. And hence, thermal efficiency of the power plant at design point will become the only parameter of interest for this study.

In this regard, simple-cycle gas turbines can hardly pose a threat to the diesel engines, featuring thermal efficiencies reaching only about 40% for the most modern engines (e.g. the GE's LM2500+G4

or the RR's MT30 gas turbines [46]), far from the 50% to 55% thermal efficiency that the last-generation diesels yield [4]. However, more *advanced* thermodynamic cycles, still keeping gas turbines as the *core* of the power plant, could potentially outcompete the diesels in this aspect.

Thus, the main aim of this research will be to devise and study the thermal efficiency at design point of a wide range of *advanced* gas-turbine-based power plants, suitable for the marine environment, establishing among them a consistent comparison so that the most promising thermodynamic cycles can be identified for further research.

In order to fulfil this aim, several tasks have been identified as the necessary steps leading to any meaningful conclusion:

- Firstly, identification of the candidate power plants to be considered within this research. Given the time frame of the ongoing marine regulations, foreseen to come into force by 2020, the search for potential *advanced* gas-turbine engines that could outcompete the diesels in terms of thermal efficiency at design point will be narrowed down to only those thermodynamic cycles for which at least a demonstrator, if not a full operating power plant, already exists. These *advanced* gas turbines should be designed, developed and commercialised within the next ten years, to take full advantage of the aforementioned opportunity window, so it is of paramount importance to focus the research in technologies already proven and reliable enough.
- Secondly, comparative assessment of these potential candidates. For this purpose, a coherent set of assumptions must be developed, that will establish a solid ground for proper and fair comparison among the studied power plants, all with equivalent levels of technology. This set of assumptions should be characteristic of the technology levels, in terms of component efficiencies, that will be available for gas turbine manufacturers within the next ten years, and will be established based upon current performance data from actual engines and observable trends in the improvement rate of these performances.
- Lastly, downselection of the most promising thermodynamic cycles analysed, narrowing down the scope of work to just a few final contenders, and physical characterisation of the downselected power plants. The main components of the few *advanced* gas-turbine-based thermodynamic cycles with higher thermal efficiency at design point will be sized, their mass and cost estimated, and preliminary considerations of the installation arrangement and requirements —i.e. placement and relative position of these components within the actual power plant, total space required within the vessel for installation of the new prime mover, etc.— will be made, in order to allow for a more comprehensive comparison against the diesel engines, not only in terms of performance levels, but also in terms of vessel requirements to deploy and operate these *advanced* gas-turbine-based power plants.

1.3 Thesis Structure

Given the above, this research comprises two clearly differentiated phases: on the one hand, thermodynamic analysis of a wide range of candidate *advanced* gas-turbine-based power plants; on the other, physical definition and preliminary design of the most promising prime movers from within all the thermodynamic cycles analysed. Hence, this thesis has been divided in two parts, each following one main phase of the research.

In **Part I**, the comparative assessment of the performance of a wide range of gas-turbine-based power plants is carried out. All the considered power plants are optimised, in terms of thermal efficiency at design point, by means of a design-space exploration strategy, that will allow for the identification of the most suitable regions for design of each candidate, and will produce rough estimates of the expected maximum achievable thermal efficiency:

Chapter 2 focuses on the most traditional, open-gas-turbine ‘dry’ cycles. Intercooling, reheating and recuperation are introduced within these analyses to understand their impact on the performance of the power plants.

Chapter 3 deals with the analysis of the gas–air combined cycles, also known as Air Bottoming Cycles (ABCs). In this arrangement, waste heat from the exhaust gases of the *topping-cycle* gas turbine is recovered —as heat source replacing the combustion chamber— within a low-pressure *bottoming-cycle* gas turbine, in which no additional fuel input is required. Similarly, the effects of intercooling, reheating and recuperation, both within the *topping-cycle* and within the *bottoming-cycle* gas turbines, will be considered.

Chapter 4 contains the performance studies of the Combined-Cycle Gas Turbines (CCGTs), where waste heat recovery from the exhaust gases of the *topping-cycle* gas turbine is carried out by means of a heat recovery steam generator (HRSG). The steam raised within the HRSG works in a *closed steam bottoming cycle*, and is expanded in a steam turbine to produce additional power.

Chapter 5 treats the study of the Steam Injected Gas Turbines (STIGs), in which waste heat from the exhaust of the *topping-cycle* gas turbine is recovered by means of a HRSG, where pressurised and superheated steam is raised for its subsequent injection into the main combustor of the *topping-cycle* gas turbine, increasing the mass flow and specific heat capacity of the working fluid circulating through the turbines, and thus producing additional power.

Chapter 6 analyses the performance of the Evaporative Gas Turbines (EvGTs), in which humidity is absorbed into the air at the outlet of the compressor, by means of a humidifier, similarly increasing the mass flow and the specific heat capacity of the working fluid circulating through the expansion side of the gas turbine.

Chapter 7 closes this part, covering the downselection of the most promising cycles from within the wide range of power plants analysed in previous chapters, with in-depth explanations on the downselection rationale. The thermodynamic cycles of the few downselected power plants will be “frozen” before undertaking further research regarding the preliminary design and physical definition of this narrow set of final contenders.

In **Part II**, the preliminary design of the downselected power plants is treated. Physical sizing of the main components of the power plant, along with mass estimates, cost assessments and schematics with the final arrangement and size of the whole power plant will be produced:

Chapter 8 details the methodology for the heat transfer calculations and preliminary sizing of the heat exchangers that the downselected power plants make use of. Design trade-offs will be analysed, and scaling laws for different power outputs will be derived from these studies. These results are subsequently used to estimate the purchased-equipment cost and final mass for each and every one of the heat exchangers implemented within the considered cycles.

Chapter 9 adds some considerations on the preliminary design, sizing and mass assessment of the turbomachinery components featured within these downselected cycles. Various proposals on the final arrangement, overall size and total mass of the whole *uninstalled* power plants are discussed and some comments on the “*installation effects*” are provided.

Finally, **Chapter 10** summarises the major findings and most relevant conclusions of this research, and proposes guidelines on future work for the deployment of gas turbines as prime movers in marine commercial applications and the likely way forward for subsequent research.

Part I

Thermodynamic Analysis

Chapter 2

Advanced ‘Dry’ Cycles

The first set of cycles to be analysed are the traditional gas-turbine open ‘dry’ cycles —usually referred to as simply “*gas turbines*”—, which will serve both as the first step towards the development of an integrated simulation platform for more complex thermodynamic cycles and as the base case with which to compare the performance of more sophisticated power plants. *Advanced* gas turbines can reach higher levels of thermal efficiency at design point by means of incorporating one or more of the following [54]:

Intercooling. By splitting the compression segment of the gas turbine in two or more steps, and reducing the temperature of the working fluid in between those steps, the mean heat-rejection temperature of the cycle is reduced, which brings the thermal efficiency of the cycle closer to the ideal Carnot efficiency. Moreover, the necessary compressor work for a given pressure ratio (PR) is proportional to the compressor inlet temperature (see eq. 2.1 for a perfect gas with constant polytropic efficiency), and thus by reducing the latter a significant reduction of the former is achieved as well, which will result in higher useful power from the expansion side.

$$P_{\text{compr}} = \Delta H = mc_p(T_o - T_i) = mc_p T_i \left(\frac{T_o}{T_i} - 1 \right) = mc_p T_i \left(\text{PR}^{\frac{\gamma-1}{\gamma\eta_p}} - 1 \right) \quad (2.1)$$

Reheating. By splitting the expansion segment of the gas turbine in two or more steps, and increasing the temperature of the working fluid in between those steps, the mean heat-addition temperature of the cycle is increased, which brings the thermal efficiency of the cycle closer to the ideal Carnot efficiency. Similarly, the expansion work delivered by the turbine for a given PR (note that in a turbine $\text{PR} < 1$ always) is proportional to its inlet temperature (see eq. 2.2 for a perfect gas with constant polytropic efficiency), hence by increasing the latter a significant increase of the former is achieved as well, which will be available as useful power.

$$P_{\text{turb}} = -\Delta H = mc_p(T_i - T_o) = mc_p T_i \left(1 - \frac{T_o}{T_i} \right) = mc_p T_i \left(1 - \text{PR}^{\frac{\gamma-1}{\gamma\eta_p}} \right) \quad (2.2)$$

Recuperation. Exhaust gases from the turbine outlet are almost at ambient pressure, but still hot. By means of a heat exchanger, this heat —that would otherwise be released into the atmosphere and wasted— can be recovered within the cycle. The aim is to do this at the compressor outlet, assuming that the compressor delivery temperature (CDT) is lower than the gas-turbine exhaust gases temperature (EGT). In this way, the combustor inlet temperature will be higher due to the recovered heat, thus reducing the necessary temperature leap in the combustion chamber to reach the desired turbine inlet temperature (TIT) of the cycle, which will translate immediately into significant fuel savings.

2.1 Modelling & Assumptions

The models analysed consist of a two-spool core plus free power turbine configuration inspired by the Rolls-Royce’s MT-30 marine gas turbine, in which different arrangements of intercooler, reheaters

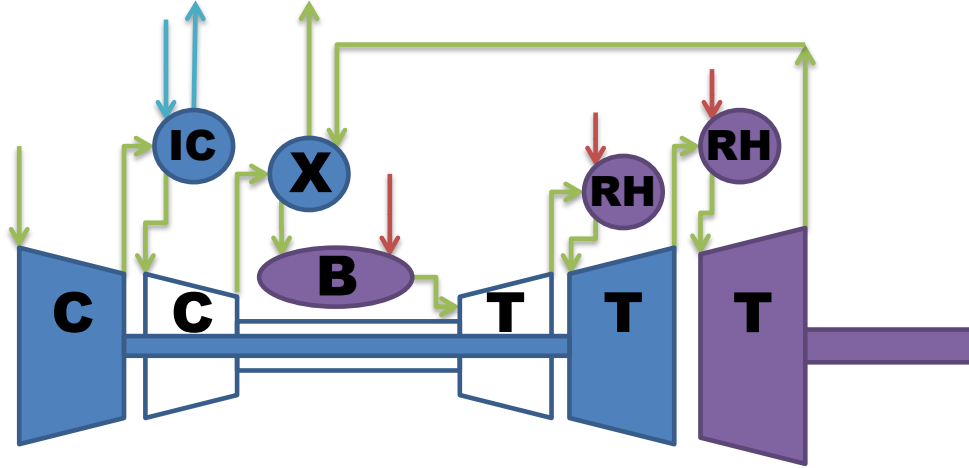


Figure 2.1: Schematic layout of the advanced gas-turbine open ‘dry’ cycle

and recuperator (see fig. 2.1) can be enabled or disabled at will, to assess the effect of introducing each of them, first separately and then altogether in conjunction.

As shown by the layout represented in fig. 2.1, an intercooler can be added between the low-pressure (LP) and the high-pressure (HP) compressors. In addition, reheaters could be added between the two compressor-driving turbines (in what will be called “*HP-reheating*”) or right upstream of the free power turbine (in what will be called “*LP-reheating*”). Also, both reheaters can be enabled at the same time, in what will be called “*dual-reheating*”. The last component, the recuperator, allows for waste heat recovery from the exhaust gases into the main gas path, right before the working fluid enters the combustion chamber, raising therefore the combustor inlet temperature.

2.1.1 Simulation Tools

In order to perform the comparative assessment of the different ‘dry’ cycles, a simulation platform has been set up in MATLAB®. The suite is divided into several blocks of code, as shown by the diagram in fig. 2.2: the first subroutine specifies the cycles to analyse and allows for splitting tasks so as to be able to parallelize the job; the second one meshes the design space within the boundaries specified for the free design parameters (see Section 2.1.2); the third one performs an iterative loop so as to converge each design point to analyse; and the fourth and last saves the results and post-processes them.

The iterative loop to converge any design point relies on an external gas-turbine performance simulation tool. TURBOMATCH, an in-house developed code, was used for this purpose. This code was initially developed by Palmer [79] and has been thoroughly revised, improved and validated since then by numerous contributors at Cranfield University [68, 109, 100, 74]. More recently, works by Tsoudis [107] and Polyzakis [82, 83] showcase the flexibility of this software and its capabilities for accurately modelling virtually any possible gas-turbine cycle, what makes TURBOMATCH inherently suitable for this research project.

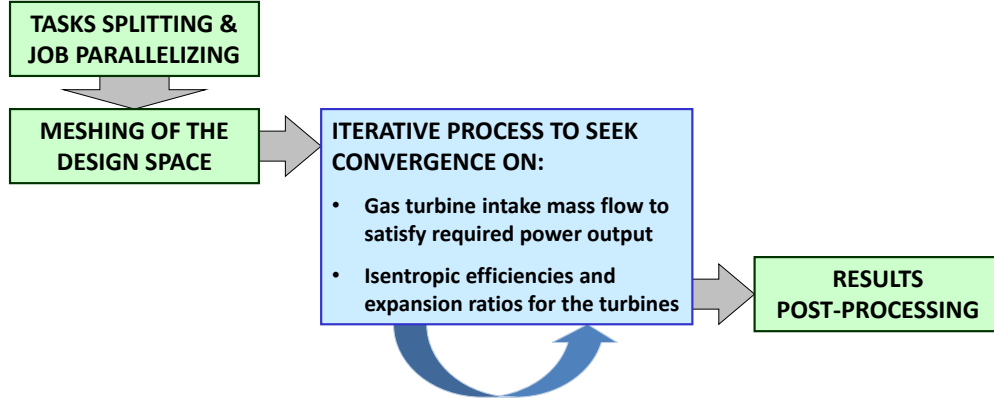


Figure 2.2: Sketch of tools developed

2.1.2 Design Space Exploration

According to the general arrangement presented in fig. 2.1, there are sixteen potential cycles to be analysed, by means of enabling or disabling the intercooler, both reheaters and the recuperator. The design space for any given cycle is meshed in terms of pressure ratios (for both compressors) and in terms of temperatures at the outlet of reheaters and intercoolers. An overall pressure ratio (OPR) alone does not define a design point on its own, since the PR split between the two shafts determines the ‘position’ of the intercooler and the HP-reheater and, hence, must be specified. Similarly, the *degree* of intercooling or reheating —i.e. the amount of heat removed from or supplied to the main gas path— featured in a gas-turbine cycle affects its performance, so the outlet temperatures of these components must also be defined to completely fix a single design point of operation.

2.1.3 Modelling Assumptions

To provide a basis in which cycles can be compared with each other fairly, certain assumptions have to be made and kept the same in all the simulations. Irrelevantly of the cycle analysed, the TIT and the required output power have been defined as 1600 K and 40 MW, respectively. TIT is chosen as a measure of the available ‘*technology level*’ that can be foreseen for the next few years, and the output power as a measure of the characteristic size of the power plant for the targeted marine commercial applications and the market of the large cargo vessels.

Other general assumptions

The working fluid is modelled as a perfect gas. The calorific properties for the gas are calculated, based on the humidity and composition of the air due to the presence of combustion products, according to the models presented by Bückner et al. [19].

Compressors

Compressors are characterised just by their PR (which value is specified for each design point via the mesh created) and their isentropic efficiency. In order to properly compare compressors with high differences in their PR, imposing a constant polytropic efficiency is recommended. Therefore,

the isentropic efficiency of each compressor can be derived from its actual PR and the polytropic efficiency according to eq. 2.3.

$$\eta_{is} = \frac{\Delta H_{is}}{\Delta H} = \frac{\text{PR}^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_o}{T_i} - 1} = \frac{\text{PR}^{\frac{\gamma-1}{\gamma}} - 1}{\text{PR}^{\frac{\gamma-1}{\gamma\eta_p}} - 1} \quad (2.3)$$

As a typical value that can be foreseen in the next ten years, the polytropic efficiency is taken as 0.90 for all the compressors, based on data provided by the sponsor and the observable trends within these data.

Turbines

Similarly, the turbines are characterised by their PR —or, in this case, their ‘expansion ratio’— and their isentropic efficiency. However, since the expansion ratio across the turbines is dependant on their isentropic efficiency (turbines need to provide the work that the compressors require, which determines the total temperature drop across the turbine and, if the isentropic efficiency is known, the expansion ratio of the turbine), and the isentropic efficiency depends as well on the expansion ratio and the polytropic efficiency (see eq. 2.4), these are unknown beforehand. Hence the need for an iterative loop that will converge to the actual value of these parameters for each design point and configuration analysed.

$$\eta_{is} = \frac{\Delta H}{\Delta H_{is}} = \frac{1 - \frac{T_o}{T_i}}{1 - \text{PR}^{\frac{\gamma-1}{\gamma}}} = \frac{1 - \text{PR}^{\frac{\gamma-1}{\gamma}\eta_p}}{1 - \text{PR}^{\frac{\gamma-1}{\gamma}}} \quad (2.4)$$

The polytropic efficiency can be taken as 0.90 for the core turbines (driving the compressors) and 0.92 for the free power turbine, as representative values of the foreseen level of technology for the next ten years, according to data provided by the sponsor and the observable trends within these data.

Burners

The combustion chamber and both reheaters are treated all in the same way. Distillate marine gas oil (MGO) is used as fuel. Combustion efficiency is taken as 99%, and a total pressure drop of 5% is assumed as typical value for modern combustors [110].

Heat Exchangers

Performances of the intercooler and the recuperator are very similar to each other, in spite of the former being a water–air heat exchanger and the latter a gas–air heat exchanger. In both cases, the heat exchanger is characterised by its *effectiveness*, defined as the temperature leap *achieved* divided by the temperature leap *available* that could be reached in the ‘ideal’ case in which all the heat would be transferred from one stream to the other. For the intercooler, the main gas path outlet temperature is specified and the amount of heat removed from the gas stream, along with its corresponding effectiveness, is calculated. For the recuperator, however, the outlet conditions for both the hot (gas) and the cold (air) sides are calculated based on the heat exchanger effectiveness and the inlet temperatures and compositions of both streams only.

Recuperators are probably the most challenging component to implement within these cycles. Main technical difficulties for recuperator designs nowadays are the high temperatures on the hot side and *leaktightness* of the component, even though flow leakages within modern recuperators are already close to nil [73]. Structural integrity due to thermal cycles and maintenance issues with corrosion used to be the main reasons for avoiding the utilisation of recuperators for many years, but the levels of reliability and the maturity of this technology today have overcome most of these concerns [73]. Rolls-Royce’s WR-21 intercooled-recuperated gas turbine accumulates thousands of running hours in a harsh marine environment, meeting the standards of the Royal Navy. In the

power generation market, Solar Turbines has successfully culminated the development of a high-performance primary-surface recuperator, compact and reliable, with effectiveness well over 90% and minimal pressure losses [80, 111]. Temperature limits on the hot side of stainless steel recuperators are usually assumed to be close to 1089 K, but much higher limits stand for Inconel super-alloys [73]. The additional costs of these components can easily be overcome in the long run by fuel savings during operation of the power plant.

For these studies, the effectiveness of the recuperator is taken as 90% as a currently achievable target, depending only on the size of the component (the bigger the heat exchanger, the higher its effectiveness). Size of the power plant is considered to be unconstrained so far —being these gas turbines aimed at replacing much heavier and bigger diesel engines— and, therefore, a relatively high value for the heat exchanger effectiveness is assumed to be achievable.

In addition to this, total pressure losses are estimated as 7.5% in the hot side and 5% in the cold side for both the intercoolers and the recuperator. The cold (water) side of the intercooler is not analysed, though, since its performances and the conditions of the outlet water are irrelevant for the thermal efficiency of the whole cycle.

Blade Cooling Considerations

For this *preliminary* comparative assessment among the different gas-turbine-based power plants, the negative effects on the overall performance of the whole power plant due to blade cooling requirements for the hot segment of the gas turbine were neglected.

On the one hand, estimating the cooling flow requirements at the very early stages of development of a gas turbine is a complicated task (accurate predictions would require deep knowledge of the detailed geometry and flow patterns within the hot segment of the gas turbine, which are unknown yet), that relies heavily in experimental correlations, somewhat acceptable for the ‘dry’ cycles presented in this section but usually without an ‘equivalent’ in accuracy readily available for more complex cycles such as the EvGTs presented in Section 6, which makes it very hard to be able to compare different power plants fairly and consistently on an even base.

On the other hand, the whole study and optimisation of the bleeds positioning —e.g. at the recuperator inlet vs at the recuperator outlet for the ‘dry’ cycles, for instance— and cooling technologies available —e.g. steam cooling vs air cooling for the CCGTs, humid air cooling vs dry air cooling for the EvGTs, etc.— is on its own a subject of research that would require a fairly large amount of time and that falls out of the scope of this work.

For this reason, and given that the effect of blade cooling requirements would affect the *absolute* levels of predicted performance for any given power plant but, at the same time, would not impact much the results of a *comparative* study as long as a consistent set of assumptions is employed for all the different power plants considered (see further discussion and references in Section 10.1), it was decided to neglect them.

Moreover, the 1600 K-TIT thermodynamic cycles presented within this thesis could be understood as some sort of ‘*thermodynamic equivalents*’ of much-higher-firing-temperature cycles with the latest materials, thermal barrier coatings (TBCs) and blade-cooling technologies that will become available for gas-turbine manufacturers within the next few years.

2.2 Results Discussion

In order to systematically compare the different cycles available, and given the number of independent variables with which a single design point is defined (OPR, PR split between spools, TIT, intercooler and reheaters outlet temperatures, etc.), direct design-space exploration would be highly time-consuming. For this reason, the comparison has been broken into smaller tasks, in which most of the independent variables will remain fixed and the effect of some few parameters on the thermodynamic cycle will be studied in detail, helping out in establishing trends and finding the best candidate in a step-by-step, logical manner.

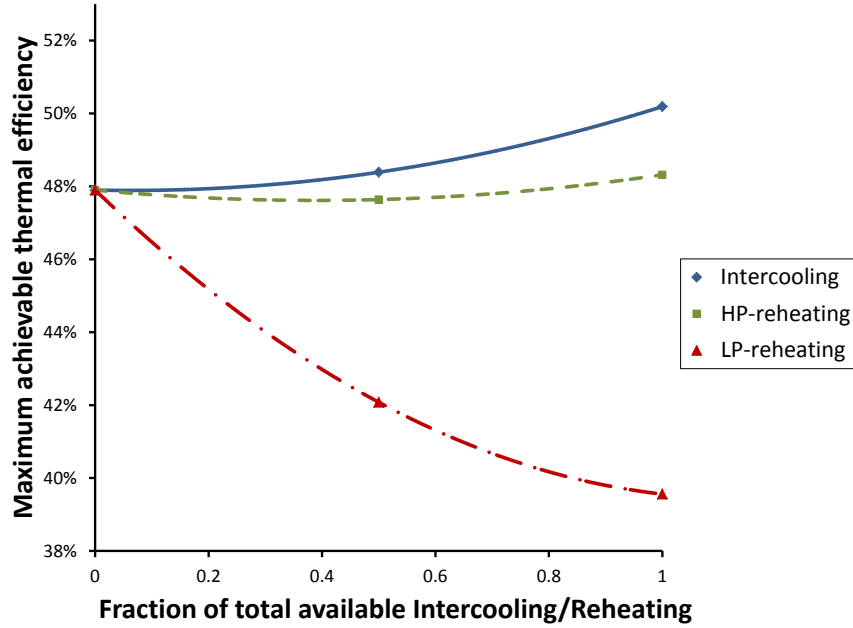


Figure 2.3: *Effect of cycle main temperatures on the maximum achievable thermal efficiency*

2.2.1 Main Cycle Temperatures

First parametric study was carried out to fully understand the effect of the main temperatures of the cycle. While the TIT is chosen as the available ‘technology level’ and is, therefore, fixed, all other temperatures remain to be decided. Namely:

Intercooler outlet temperature, which can be varied between the ambient temperature (maximum available intercooling carried out, ‘intercooling fraction’ defined as 1.00) and the low-pressure compressor delivery temperature (no intercooling carried out, ‘intercooling fraction’ defined as 0.00); and

Reheaters outlet temperature, which can be varied between the upstream turbine outlet temperature (no reheating carried out, ‘reheating fraction’ defined as 0.00) and the actual TIT of the cycle (maximum available reheating carried out, ‘reheating fraction’ defined as 1.00).

With these definitions in mind, simulations of a simple-cycle gas turbine were carried out over a wide range of OPRs, looking for the maximum achievable thermal efficiency. Same analyses within the same ranges of OPR were carried out enabling one of the intercooler/reheaters at a time, keeping the others disabled, so the effect of the intercooling/reheating fraction for that particular component in the cycle’s thermal efficiency could be assessed. Results of these analyses are shown in fig. 2.3 and will be discussed herein:

Intercooling fraction. The positive effect of applying intercooling in the main gas stream, hence reducing the necessary work to drive the HP compressor, is partly lost due to two negative effects: on the one hand, a total pressure loss associated to the heat exchanger; on the other, the extra fuel needed to reach the same TIT because of the lower CDT and lower temperature at the inlet of the combustion chamber. For this reason, at very low ‘intercooling fractions’—when the temperature at the outlet of the intercooler is not much lower than the LP CDT—the maximum achievable thermal efficiency with the intercooled cycle is slightly lower than that of the simple cycle. However, as the ‘intercooling fraction’ increases, the benefits of intercooling become more evident and eventually overtake the associated losses, so an optimised intercooled cycle (i.e. with the design point at its optimum OPR for maximum thermal efficiency) can outcompete the optimised simple cycle.

HP-reheating fraction. The negative effect of adding extra fuel plays against the benefits from being able to obtain greater work from the LP turbine, hence being able to drive the LP compressor with a smaller expansion ratio, which will directly lead to more useful work available to be extracted at the free power turbine. As happened with the intercooler, for low ‘reheating fractions’ —with small quantities of fuel added in the reheater— the maximum achievable thermal efficiency of the HP-reheated cycle is lower than that of the simple cycle. As the ‘reheating fraction’ is increased, the higher the temperature at the outlet of the reheater, the more evident the benefits from the reheating process become, until eventually these benefits overtake the initial losses and a point is reached in which an optimised HP-reheated cycle can outcompete the optimised simple cycle.

LP-reheating fraction. Adding fuel right before the free power turbine does not affect the core of the gas turbine. In this case, the more fuel that is added, the worse the overall thermal efficiency of the power plant becomes. However, the specific power is greatly increased, since more work can be extracted from the free power turbine for a constant expansion ratio. This makes LP-reheating to seem not very promising as a viable option for being implemented within the large cargo vessels’ power plants, which will aim for the highest possible thermal efficiency. However, in addition to the extra work from the free power turbine, LP-reheating also increases the temperature of the stack gases. This allows for any implemented heat recovery system to be much more efficient, as it will be shown later, and thus LP-reheating cannot be disregarded yet.

Keeping in mind all the above, the temperatures at the outlet of the intercooler and reheaters will take their most extreme values in all subsequent simulations: the lowest temperature possible will be chosen for the outlet of the intercooler and the highest achievable temperature will be selected for the outlet of any reheater. This will reduce the number of parameters involved in the optimisation of the candidate ‘dry’ gas-turbine cycles to just two: the individual PR of each of the compressors. The selected temperatures are the ones yielding either the highest potential for overall thermal efficiency of the power plant (this is the case of the lowest possible temperature imposed at the intercooler outlet or the highest possible temperature imposed at the HP-reheater outlet) or the highest potential for waste heat recovery (this is the case of the maximum possible temperature imposed at the LP-reheater outlet). A parametric optimisation has been carried out with these constraints, which will help to identify the approximate optimum OPR —as well as the optimum PR split between spools; i.e. the optimal position for the intercooler or the HP-reheater— for each cycle, and to estimate its corresponding maximum achievable thermal efficiency.

2.2.2 Effects of intercooling

Intercooling can easily be implemented in any marine commercial application. Sea water is effectively an almost infinite heat sink that can be utilised for this purpose. The whole design space for the intercooled cycle is plotted in fig. 2.4, which shows the contours of thermal efficiency at design point against the design-point OPR and LPPR. From this plot, the local thermal efficiency maxima for a particular OPR can be extracted to get the thermal efficiency vs OPR curve of the intercooled cycle. This is plotted in fig. 2.5, and compared against the thermal efficiency achievable by means of a simple-cycle gas turbine.

As can be seen in fig. 2.5, the intercooled cycle only outcompetes the simple cycle for extremely high OPRs. These high thermal efficiency design points correspond, as shown in fig. 2.4, to cycles with extremely low LPPR (almost negligible compression carried out by the LP spool) and extremely high HPPR, which represent a rather ‘infeasible’ region of the design space, in which the effects appreciated from the intercooled cycle are equivalent to a reduction of the inlet temperature in a simple cycle (without any intercooling afterwards), and not a consequence of the intercooler itself. Therefore, it can be concluded that the intercooled cycle is not a valuable option in terms of thermal efficiency by itself.

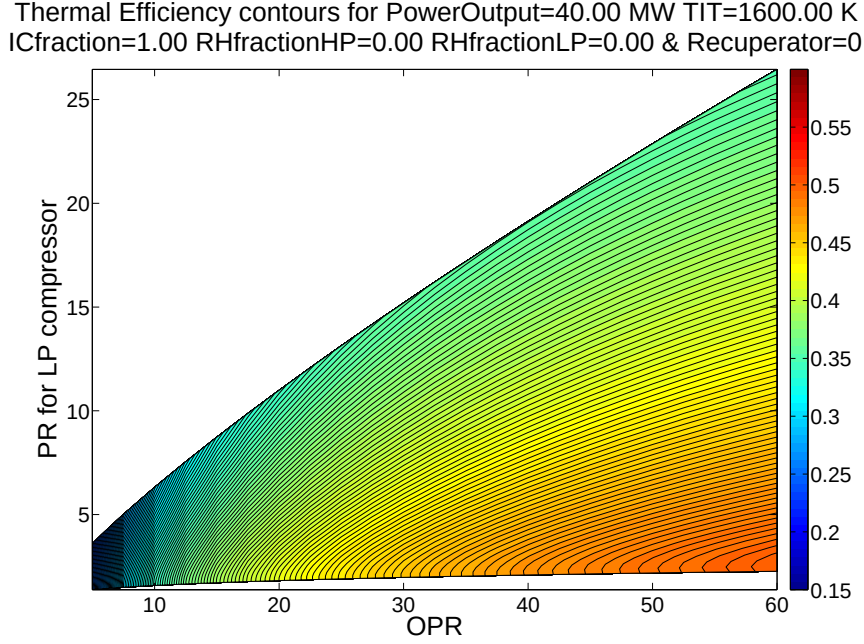


Figure 2.4: *Contours of design point thermal efficiency for the intercooled cycle*

2.2.3 Effects of reheating

In the case of the HP-reheated cycle, the optimum point of the design space that can actually outcompete the optimised simple cycle appears (see fig. 2.6) also in a rather ‘infeasible’ region of the design space—as it occurred with the intercooled cycle—but right at the other edge: with an LP compressor carrying out the greatest part of the compression work, followed up by an almost negligible HP compressor. Very little expansion is therefore carried out at the HP turbine, before applying the reheat process and doing almost all the necessary work extraction to drive the compressors at the LP turbine.

As happened with the intercooled cycle, in this case the gain in thermal efficiency must not be understood as the effect of reheating itself. Effectively, the almost negligible expansion carried out before reheating takes place can also be seen as a slight increase in the TIT from the main combustion chamber, with no reheating at all.

The LP-reheated cycle, on the other hand, has already been shown yielding very low thermal efficiencies if no waste heat recovery system is implemented. So, in the same way, one can conclude that the reheated cycles (irrelevantly of the pressure level at which the reheating is applied) are not a promising option as a viable power plant for marine commercial applications: HP-reheating optimum falls within an ‘infeasible’ region of the design space and LP-reheating is simply outcompeted by the simple-cycle gas turbine.

2.2.4 Effects of recuperation

Recuperated cycles show the optimum thermal efficiency at much lower OPRs than simple-cycle gas turbines. The gains in thermal efficiency when compared with the simple cycle are just about a 2% points improvement (see fig. 2.8), reaching almost 50% maximum thermal efficiency at OPRs around 7.5:1. Recuperation is identified, therefore, as the means by which the greatest potential for high thermal efficiencies can be achieved, and recuperators will be the key components in improving the

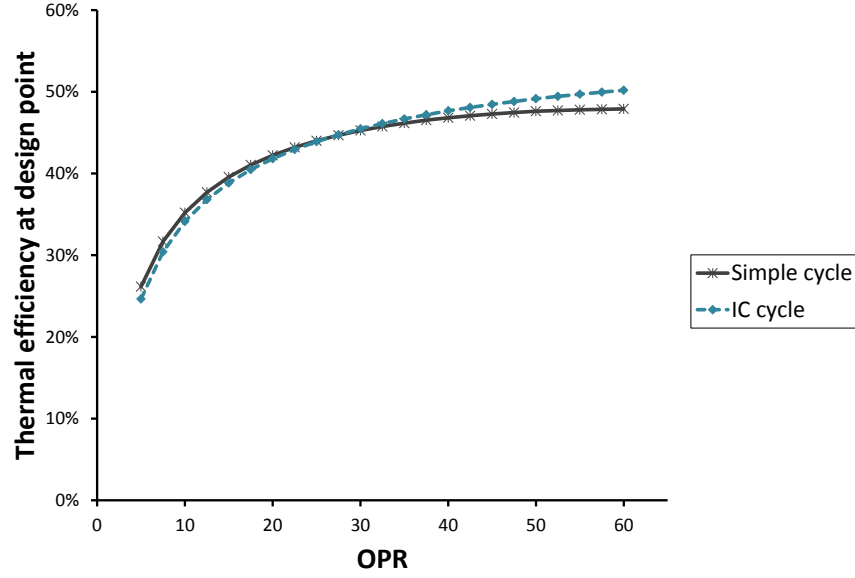


Figure 2.5: Comparison between simple- and intercooled-cycle

performances of the gas turbine power plants, especially when combined with either intercooling, reheating, or both.

2.2.5 The Intercooled-recuperated (IC-R) cycle

The benefits of the intercooled cycle can be fully exploited by means of adding a recuperator. Being one of the main drawbacks of the intercooled cycle the low CDT, which implies high quantities of fuel injected in the combustion chamber to reach the specified TIT, its synergy with the recuperated cycle appears immediately as an appropriate way of overcoming that disadvantage by just increasing the temperature at the inlet of the combustor via the heat recovery system. Furthermore, the lower the CDT, the more heat can be recovered from the exhaust gases, hence increasing the positive effects of the recuperator on the whole cycle's thermal efficiency. The overall thermal efficiency at design point for such a cycle is shown in fig. 2.8, compared against the simple-cycle and the simple-recuperated gas turbines.

Several studies have already dealt with the design and development of the IC-R cycle during the last thirty years, the main outcome of these being the WR-21 advanced cycle gas turbine [28, 99]. However, the approach taken in this study is substantially different, since the requirements for the WR-21 specified part-load performances that had to be met for the usual mission profile of the naval applications to which it was targeted: cruise mode, at low power output, during most of the time, plus 'boost' mode for high speeds during shorter periods. This resulted in the range of 20%–30% of the maximum power output to be the more demanding requirement to be met, and thus the design of the WR-21 was mainly driven by part-load considerations, rather than by trying to achieve the maximum thermal efficiency from this type of cycle. For all the above, it can be noticed that the maximum thermal efficiency achievable with an IC-R cycle (slightly above 54%) is far beyond the WR-21's thermal efficiency at design point (which barely surpasses the 40% mark), proving this cycle to be an option to consider, if appropriately designed, also for marine applications with high thermal efficiency requirements at design point, and not only because of part-load performances.

2.2.6 The Reheated-recuperated (RH-R) cycle

When recuperation is enabled, reheating does not only allow for more power to be extracted from the turbines at the expansion side, but also provides the cycle with greater EGT, and thus additional

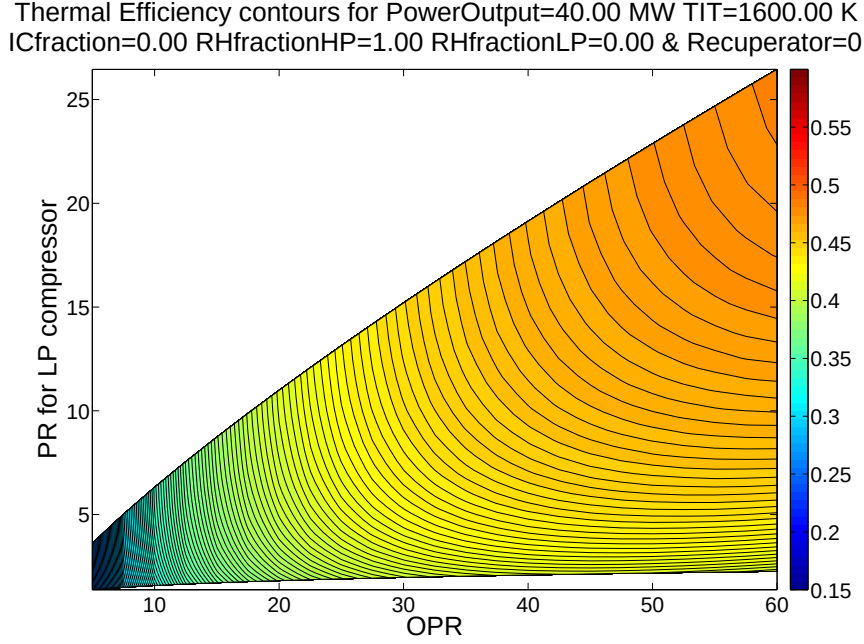


Figure 2.6: *Contours of design point thermal efficiency for the HP-reheated cycle*

waste heat available that can potentially be recovered into the cycle. Three configurations are possible for this sort of cycle: *a)* the HP-reheated, recuperated cycle; *b)* the LP-reheated, recuperated cycle; and *c)* the dual-reheated, recuperated cycle. Dual-reheating can immediately be discarded, as the increased cooling flows requirement due to having three cooled turbines, rather than just two, outweighs the benefits from the increased heat-addition mean temperature and makes this cycle to yield the lowest thermal efficiency of the three configurations. In addition to this, it would incur in greater maintenance costs and higher levels of emissions, so it is easily outperformed in every aspect by any of the single-reheater configurations.

As for the LP-reheated or the HP-reheated cycles, both yield very similar efficiencies (the latter just about 0.25% points lower) at similar OPRs. However, further considerations have to be made if it is to be decided which of the two configurations is the most promising for the purposes discussed herein. In fig. 2.7, the contours of thermal efficiency for an HP-reheated recuperated cycle are shown. It can be seen that the optimum achievable thermal efficiency with this configuration is reached at the bottom edge of the design space, for an OPR of around 10:1, where a very small region with overall thermal efficiencies just over 51% appears. This means that the optimum cycle would feature an almost negligible compression/expansion in the LP spool, while most of the compression/expansion would be carried out in the HP spool. Effectively, the optimum thermal efficiency achievable with an HP-reheated cycle would be, precisely, with the reheater placed as close as possible to the free power turbine. This is, as close as possible to the LP-reheated configuration. This suggests that the most promising cycle to carry on forward in further analyses is the LP-reheated configuration, and thus the HP-reheating will be disregarded for all subsequent simulations, considering only the possibility of placing a reheater in between the LP core (compressor-driving) turbine and the free power turbine.

When compared with the IC-R cycle, it is apparent that the synergy between reheating and recuperation does not quite match the potential of intercooling and recuperation working in conjunction (see fig. 2.8). However, this study proves that there is merit for the reheated gas turbines to be implemented within power plants featuring waste-heat recovery systems, especially when those systems

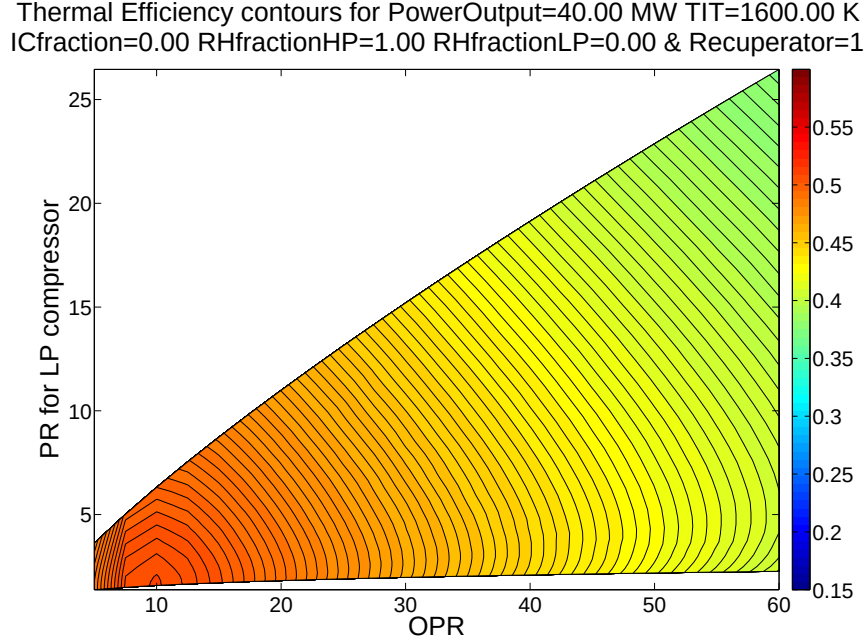


Figure 2.7: *Contours of design point thermal efficiency for HP-reheated-recuperated cycle*

do not rely on a low CDT to maximise their effectiveness. Such would be the case of combined-cycle power plants featuring separate *bottoming* cycles, rather than the analysed recuperation method.

2.2.7 The Intercooled-reheated-recuperated (IC-RH-R) cycle

When both the benefits of the intercooler and the reheater are added up and combined with a recuperator, the resulting cycle offers low CDT along with high enthalpy available in the exhaust gases, maximising the potential for heat recuperation. The maximum thermal efficiency achieved by means of this thermodynamic cycle is raised by almost 2.5% points when compared with the IC-R cycle. This yields ultimately the most promising configuration among the ‘dry’ cycles analysed, with thermal efficiencies as high as 56.5% (see fig. 2.8) at OPR around 35:1.

2.3 Summary

A broad range of Brayton ‘dry’ cycles featuring intercooling and/or reheating and/or recuperation have been analysed and parametric optimisation through systematic design-space exploration has been carried out for each of them, in terms of finding their maximum thermal efficiency at design point and identifying the most promising regions of each configuration’s design space. Figure 2.8 summarises all the previous results, plotting thermal efficiency curves against OPR for the four most promising cycles, as well as the ‘reference’ line for the equivalent simple-cycle gas turbine. The main conclusions that can be withdrawn from these studies are:

- The highest possible thermal efficiency that can be achieved by the simple-cycle gas turbine, with the technology level that can be foreseen for the next ten years, is around 48%, at OPRs greater than 50:1.
- Intercooling or reheating, without any waste heat recovery system, are not promising cycles in terms of thermal efficiency. Being either outcompeted by the simple-cycle gas turbine or being

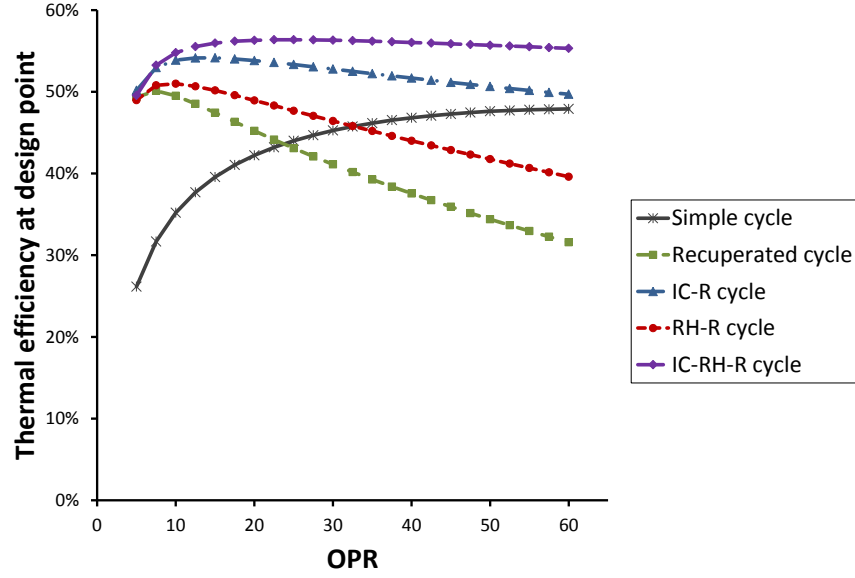


Figure 2.8: Thermal efficiency against overall pressure ratio for the open ‘dry’ cycles

barely able to outperform the simple-cycle optimum in what have been identified as rather ‘infeasible’ regions of the design space.

- Recuperation appears as a simple method to implement waste-heat recovery without incurring in the additional costs of featuring an additional *bottoming* cycle (such as in combined-cycle power plants). Recuperators can by themselves raise the thermal efficiency of the simple-cycle gas turbine by almost 2% points, pushing the optimum OPR towards much lower values (50% thermal efficiency at OPR around 7.5:1), and become even more promising if coupled with intercooling and/or reheating.
- The highest possible thermal efficiency for a ‘dry’ cycle gas turbine, assuming the technology level that can be foreseen for the next ten years, is achieved by means of an IC-RH-R cycle, reaching values around 56.5%, at OPR around 35:1, followed up closely by the IC-R cycle, with just about 2.5% points less (54% thermal efficiency at OPR around 15:1).
- The fact that the IC-R cycle is simpler than its reheated counterpart, along with the lower maintenance costs and increased life of the free power turbine (which operates at a much lower inlet temperature in the non-reheated configuration), makes it impossible to actually identify one of these as the clear ‘winner’ just based on the simulations carried out.
- RH-R and simple-recuperated cycles, both exhibit very similar optimum thermal efficiencies, around 50%–51%, which would suggest that the addition of the reheater in this case is worthless, since it would increase the complexity of the power plant without providing any real advantage when compared against the simple-recuperated cycle. Nevertheless, if different systems for waste heat recovery were to be implemented, rather than the simple recuperator, reheated-recuperated cycles still have great potential, as Alstom’s GT-24/26, for instance, implements such a cycle for their combined-cycle power plants.

Chapter 3

Air Bottoming Cycles

In an ABC, waste heat from the “main” (*topping-cycle*) gas turbine is recovered by means of a different *bottoming-cycle* gas turbine, in which the combustion chamber is replaced by a heat exchanger responsible for this heat recovery process. Waste heat from the *topping* cycle is then used as the main heat input for the *bottoming* cycle, that is able to extract some additional power from this low-temperature heat source without burning any additional fuel [16]. A typical arrangement of such power plant is presented in fig. 3.1

Due to the low bottoming-cycle TIT, the work required by the bottoming-cycle compressors is indeed very similar to the available work that can be extracted from the turbines, and thus the amount of additional useful power generated in the bottoming cycle is very sensitive to the slightest variation in the compressors required work. In this regard, intercooling applied to the bottoming cycle is an excellent way of improving the overall efficiency of the power plant: reducing the work required by the bottoming cycle’s HP compressor allows for more useful power to be extracted in the bottoming cycle’s free power turbine at no extra thermodynamic cost, at the same time that lowering the bottoming cycle CDT allows for more heat to be recovered from the topping cycle exhaust gases [61, 62, 84, 77].

Air bottoming cycles are traditionally an interesting option for re-powering existing units. Keeping old simple-cycle gas turbines as the core (*topping*) cycle and adding the bottoming gas turbine can have advantages in terms of simplicity and price: *a*) compared to the combined gas–steam (Joule–Rankine) cycle, an ABC does not require bulky steam equipment nor water feed, it can operate unmanned and requires considerably less maintenance; *b*) compared to other types of advanced ‘wet’ cycles (such as STIGs or EvGTs), the ABC does not require the modification of the core gas turbine to accommodate the extra mass flow through the expansion side. The gains in thermal efficiency when re-powering old units by means of simply implementing an ABC can be as high as 10% points (compared against the original simple-cycle gas turbines), while the specific power can be increased as well by as much as 30%, depending on the configuration [61, 62, 84, 77].

3.1 Modelling & Assumptions

The models, as shown in fig. 3.1, consist of dual gas turbines, both featuring two-spool plus free power turbine configurations, identical to those described in Section 2.1 with the sole exception of the waste-heat-recovery heat exchanger arrangement: rather than recuperating waste heat into the topping-cycle gas turbine, it is recovered as the main heat source for the bottoming cycle instead. The effects of implementing intercooling and/or reheating within the topping-cycle gas turbine will be studied as well. Given the similarities with the open ‘dry’ cycles analysed in Chapter 2, the same set of assumptions described in Section 2.1.3 apply to the ABCs. As always, the total power output requirement for all the ABC power plants analysed is taken as 40 MW—which characterises the size of the power plant—and the topping-cycle gas turbine TIT as 1600 K—which characterises the foreseen ‘technology level’ that will be available for gas-turbine manufacturers within the considered time frame—.

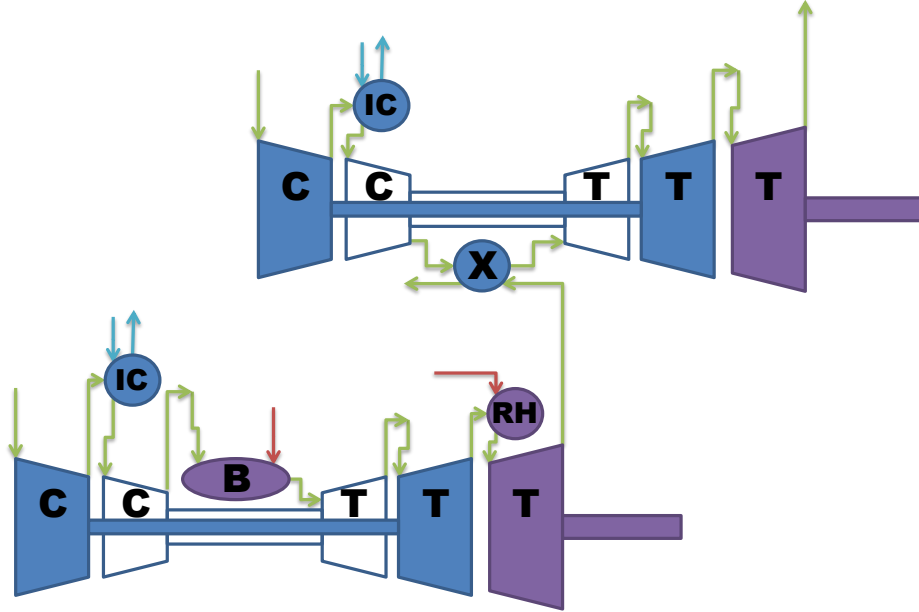


Figure 3.1: Schematic layout of an air bottoming cycle power plant

3.1.1 ‘Novel’ Proposals

Following the analyses presented in Section 2.2, it is apparent that reheating is another means by which to increase the useful power extraction in the expansion side of any gas turbine. Given the sensitivity of the bottoming cycle to any slight variation on the compressors required or the turbines extracted work, the effect of implementing reheating within the bottoming-cycle gas turbine will be studied as well.

As a result of this new heat input, the EGT from the bottoming cycle is expected to raise, and the possibility of recovering waste heat from the bottoming cycle arises: the most useful place in which this waste heat can be utilised is, as it happened with the open ‘dry’ cycles, right upstream of the main combustor of the power plant, at the outlet of the topping-cycle HP compressor, as long as the topping cycle’s CDT is lower than the bottoming cycle’s EGT. The final arrangement of the ‘novel’ ABC power plant proposed is depicted in fig. 3.2.

3.2 Results Discussion

In order to reduce the number of parameters involved in the simulations, a preliminary analysis was carried out, to try to relate the mass flow in the bottoming cycle with the mass flow in the topping cycle for optimum heat recovery. Figure 3.3 shows the variation of the two most relevant temperatures of the ABC power plant —this is, the two outlets of the main waste-heat-recovery heat exchanger from fig. 3.1, “X”, namely: bottoming-cycle TIT, the cold (air) side outlet; and topping-cycle EGT, the hot (gas) side outlet— with the relative mass flow of the bottoming cycle (defined as the bottoming-cycle mass flow divided by the topping-cycle mass flow), along with its corresponding overall thermal efficiency of the whole power plant. All other free design parameters (both gas turbines’ OPRs, main combustor TIT, etc.) are kept constant throughout this study, so both gas turbines are always the same and the effect of only varying the relative mass flow circulating through each of them can be assessed.

As shown in fig. 3.3, if the mass flow through the bottoming cycle is lower than the mass flow through the topping cycle (relative mass flow < 1), the TIT for the bottoming cycle has reached

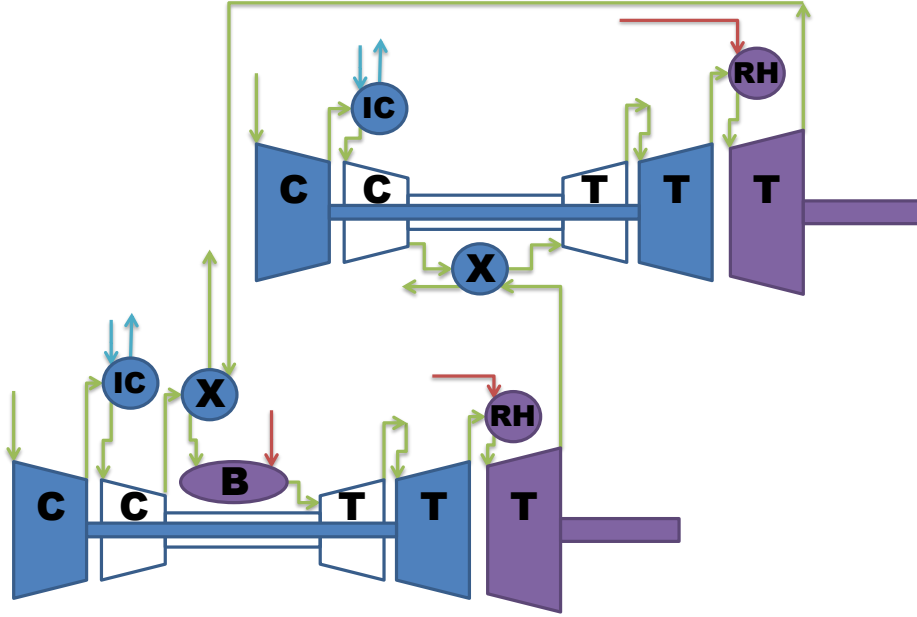


Figure 3.2: Schematic layout of the ‘novel’ air bottoming cycle power plant

its maximum (limited by the turbine outlet temperature from the topping cycle), but the EGT for the topping cycle increases as the bottoming cycle’s mass flow decreases. This means that not all the enthalpy available in the stack gases is being recovered and part of it is being released into the atmosphere. Furthermore, as the mass flow through the bottoming cycle’s free power turbine is reduced, less power output will be obtained from it. As a consequence of both, the overall thermal efficiency of the power plant decreases steadily.

On the other hand, in fig. 3.3 it can also be seen that, if the mass flow through the bottoming cycle is greater than the mass flow through the topping cycle (relative mass flow > 1), the EGT for the topping cycle reaches its minimum (limited by the CDT of the bottoming cycle), but the TIT for the bottoming cycle drops as the bottoming cycle’s mass flow raises. For lower TITs, the thermal efficiency of any cycle drops, so this means that even though the maximum amount of waste heat from the topping cycle is being recovered, it is not being used for power extraction as efficiently as it could be if the bottoming cycle’s TIT was higher. Hence the overall thermal efficiency drops as well.

This makes apparent, following the trends in fig. 3.3, that for achieving maximum thermal efficiency from an ABC configuration, the mass flow through both the topping- and the bottoming-cycle gas turbines must be the same, and so it will be taken in all subsequent simulations, effectively removing this parameter from the independent variables that define the design space of the ABC power plants.

3.2.1 Design Space Exploration

Having already established the relative mass flow distribution between the topping- and the bottoming-cycle gas turbines, keeping the design TIT of 1600 K and the required power output of 40 MW specified in Section 2.1.3, and making use of the results and conclusions of Section 2.2.1 to define the temperature at the outlet of intercoolers and reheaters, only four remain as free design parameters for the ABC power plants: the four individual PRs of each compressor of each gas turbine.

As the number of independent variables increases, so does the complexity of the design-space exploration analyses, since the number of design points to analyse increases exponentially. Further

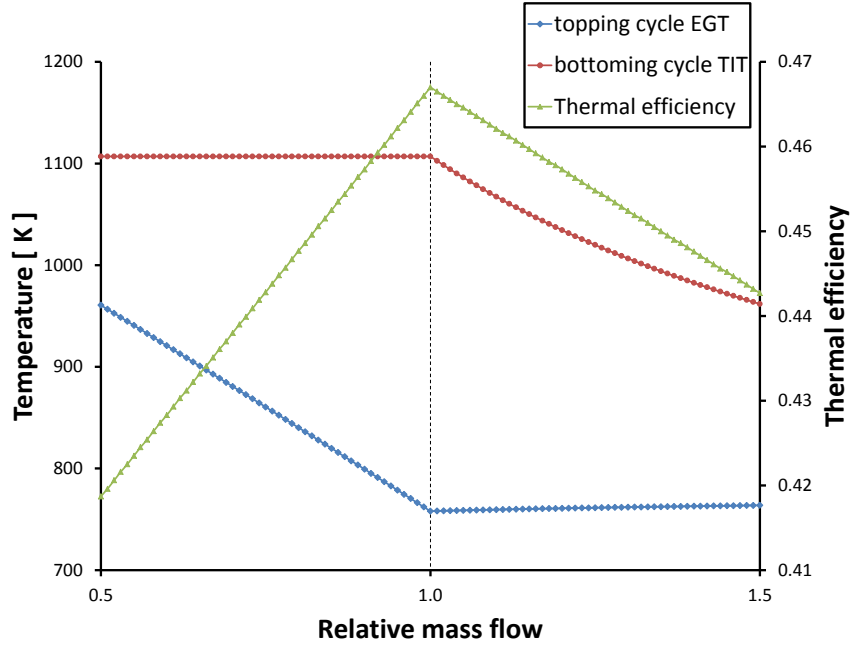


Figure 3.3: *Bottoming-cycle relative mass flow study for an ABC power plant*

details on the methodology followed and complete results from these analyses can be found in Appendix A, whilst only the most relevant findings will be discussed herein.

3.2.2 Intercooled Bottoming Cycle

Figure 3.4 shows the overall thermal efficiency against topping-cycle OPR curves for the most interesting among the ‘traditional’ ABCs analysed. Comparison against the simple-cycle ‘dry’ gas turbine from Section 2.2 showcases the typical $\sim 10\%$ points gain in thermal efficiency—at least in the lower range of OPRs analysed—by means of deploying a simple-ABC to recover waste heat from the topping cycle.

As discussed before, implementing intercooling within the bottoming cycle has a dual effect: it reduces the bottoming cycle’s CDT, allowing for more enthalpy to be recovered from the topping cycle exhaust gases; at the same time that reduces the work required by the bottoming cycle’s HP compressor, allowing for more useful power to be extracted at the bottoming cycle’s free power turbine. Both these effects prove to have great synergy with each other and result in significant improvements in the overall thermal efficiency of the power plant for the whole range of OPRs analysed (see fig. 3.4).

3.2.3 Reheated Topping Cycle

Reheating applied to the topping-cycle gas turbine, as used to happen with the open ‘dry’ cycles in Section 2.2.6, also proved to increase the overall thermal efficiency of the power plant even further in the whole range of OPRs analysed (see fig. 3.4). On the one hand, higher temperature at the inlet of the topping cycle’s free power turbine means that more power can be extracted from the same expansion ratio; on the other, it will also result in higher temperature at the outlet of the topping cycle’s free power turbine, and thus higher TIT in the bottoming cycle and better utilisation of the recovered waste heat.

3.2.4 Intercooled Topping Cycle & Recuperation

Figure 3.5 shows the overall thermal efficiency against topping-cycle OPR curves when recuperation (as described in Section 3.1.1 and fig. 3.2) is enabled, for the most interesting configurations among

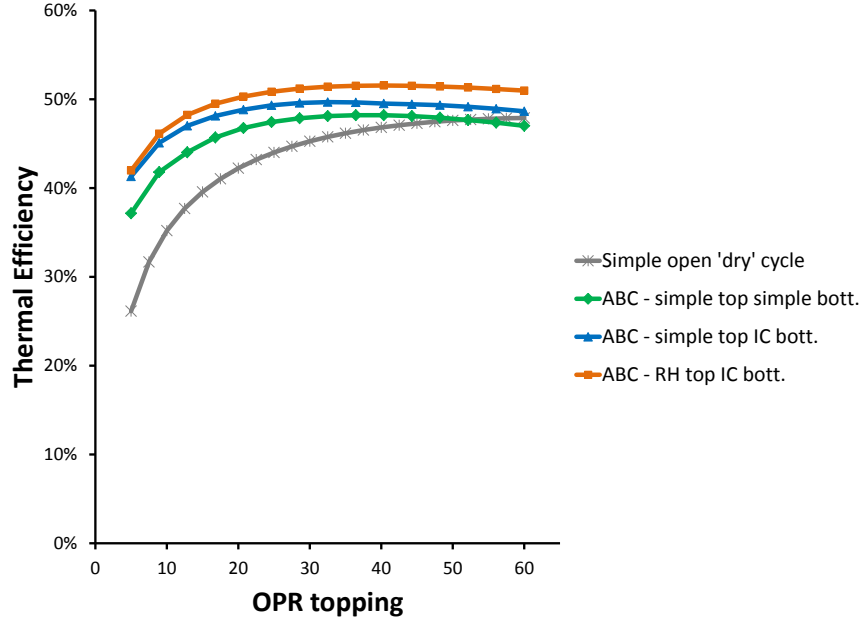


Figure 3.4: Thermal efficiency against topping-cycle overall pressure ratio for the ‘traditional’ ABC power plants

the ‘novel’ ABCs analysed. In the vast majority of ABC configurations considered within this study, recuperation of the waste heat from the bottoming cycle into the topping cycle’s compressor outlet is not feasible, except for a very narrow region at very low topping-cycle OPRs where the topping-cycle CDT is lower than the bottoming-cycle EGT. As an example, see for instance the curve for the reheated topping-cycle gas turbine with an intercooled-ABC in fig. 3.5: the ‘kink’ point that appears at $OPR \sim 12:1$ separates the region of low topping-cycle OPRs in which waste-heat recuperation from the bottoming cycle is actually feasible from the region of the design space in which it is not.

However, when intercooling is implemented within the topping-cycle gas turbine, the reduction in the topping-cycle CDT allows for waste-heat recuperation from the bottoming cycle to be feasible in a much broader region of the design space, as shown by the curve for the intercooled-reheated topping-cycle gas turbine with an intercooled-ABC in fig. 3.5, for which the ‘kink’ point appears at $OPR \sim 42:1$. Moreover, intercooling within the topping-cycle gas turbine means lower work required by the topping cycle’s HP compressor, which in turn increases the amount of useful power that can be extracted at the topping-cycle expansion side, hence increasing even further the overall thermal efficiency and outperforming the reheated-only topping-cycle gas turbine with intercooled-ABC for the whole range of OPRs analysed.

3.2.5 Reheated Bottoming Cycle & Recuperation

Applying reheating to the bottoming cycle expands even more the region of OPRs in which waste-heat recuperation from the bottoming cycle into the topping cycle’s compressor outlet is actually feasible, and further increases the effectiveness of this recuperation due to the increased bottoming cycle’s EGT, thus having the potential to achieve even higher thermal efficiencies at relatively low OPRs. In fig. 3.5, it can be seen that no ‘kink’ point appears for the intercooled-reheated topping-cycle gas turbine featuring an intercooled-reheated-ABC within the range of OPRs considered for this study, showcasing that waste-heat recuperation from the bottoming cycle is feasible for this configuration in every single analysed design point. Also, the thermal efficiency yielded by this configuration—in which both the intercooler and the reheater are enabled in both the topping- and the bottoming-cycle

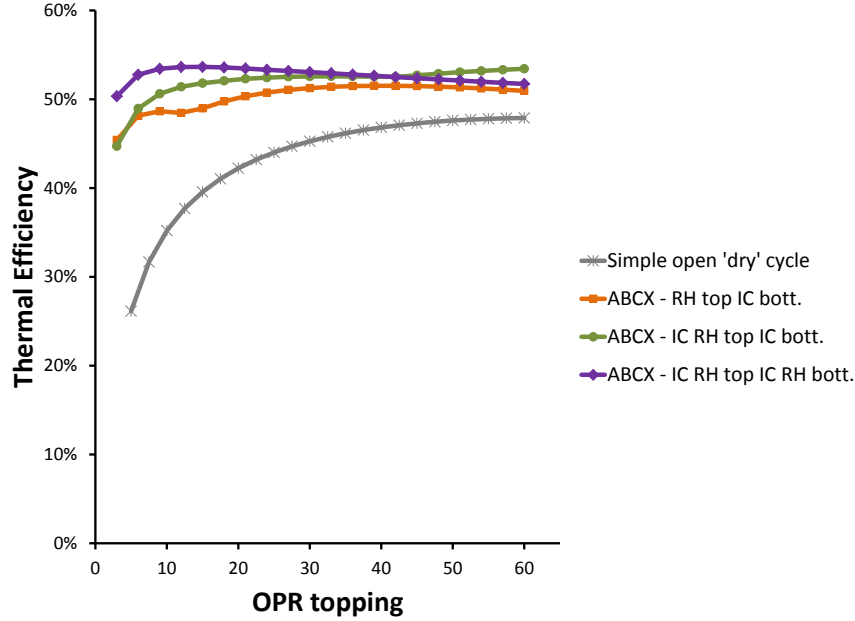


Figure 3.5: Thermal efficiency against topping-cycle overall pressure ratio for the ‘novel’ ABC power plants

gas turbines—is the greatest achievable from all the ABC power plants analysed, reaching $\sim 53.5\%$ at topping-cycle OPR $\sim 15:1$.

3.3 Summary

According to the schematics presented in fig. 3.2, there are 32 potential ABC power plants to analyse, by means of enabling or disabling the intercoolers, the reheaters and/or the recuperator. Among this broad range of thermodynamic cycles considered, just a few are worth noticing:

- Simple-cycle gas turbines’ thermal efficiency can be improved, at least within the lower range of OPRs typical for industrial applications, by means of implementing simple-ABCs as waste-heat recovery systems (see fig. 3.4). At extremely high OPRs, though, the topping-cycle turbine outlet temperature becomes too low, and additional power extraction by means of a bottoming gas-turbine cycle becomes infeasible.
- Intercooling within the bottoming cycle increases the potential of the ABC power plant —by increasing the waste-heat recovery capability of the bottoming cycle and by allowing for more useful power extraction at the bottoming-cycle expansion side—, raising its overall thermal efficiency at design point by about 2% points for the whole range of OPRs analysed.
- Reheating within the topping cycle enhances the performance of the ABC power plant —by means of allowing for greater useful power extraction at the topping-cycle expansion side and by raising the bottoming-cycle TIT—, resulting in additional gains in overall thermal efficiency at design point of about 2% points for the whole range of OPRs analysed, when compared against the simple-topping-cycle gas turbine with intercooled-ABC.
- The synergy between intercooling within the topping cycle and waste-heat recuperation from the bottoming cycle has been shown, making it possible for the intercooled-reheated topping-cycle gas turbine with intercooled-ABC and recuperation implemented to reach overall thermal

efficiencies at design point around 52.5% at topping-cycle OPR $\sim 25:1$, in spite of its non-recuperated counterpart being able to outperform this configuration at extremely high topping-cycle OPRs.

- Reheating within the bottoming cycle has good synergy with the previous configuration, making out of the intercooled-reheated topping-cycle gas turbine with intercooled-reheated-ABC and recuperation the most promising among the 32 possible configurations analysed, reaching thermal efficiencies as high as 53.5% at topping-cycle OPR $\sim 15:1$.

Still, the most promising configuration from within the ABC power plants analysed cannot quite match the performance of the most promising among the open ‘dry’ cycles discussed in Section 2.3, even though the ABCs have increased complexity —i.e. twice the number of every component—, which would result in bulkier, heavier and more expensive power plants overall, with higher acquisition and maintenance costs. From this study, it can be concluded that ABCs can, therefore, have their potential as a rather inexpensive option to re-power old units, but are not the way forward to design brand-new high-thermal-efficiency *advanced* gas-turbine-based power plants.

Chapter 4

Combined Cycle Gas Turbines

Combined-cycle power plants (CCPPs) —also known as CCGTs— have become the major trend in power generation systems during the last decades, mainly due to the high thermal efficiencies that they can achieve. Since the early stages of their development, it was clear the synergy between the high firing temperatures of the gas-turbine *topping* cycle and the low stack temperatures of the steam *bottoming* cycle (SBC) —gas turbines are typically high-TIT but low-OPR cycles, while steam power plants work at low TITs but much higher OPRs—, so that waste heat from the former could be utilised as heat input for the latter, hence increasing the difference between the mean heat-supply temperature and the mean heat-rejection temperature and achieving, therefore, higher Carnot efficiencies for the whole CCPP thermodynamic cycle.

There is a broad range of arrangements in which combined gas–steam cycles can work (see, for instance, [113, 88, 89, 17, 60]), and thus one of the major goals of this study is to establish, for the particular set of assumptions taken within the framework of this research project —aimed at providing a consistent basis with which to be able to carry out fair comparison among the very different thermodynamic cycles considered—, the optimal configuration for both the gas-turbine *topping* cycle and the SBC that, working together, would yield the maximum overall thermal efficiency at design point.

4.1 Modelling & Assumptions

Gas turbine models analysed consist of two-spool core arrangements with free power turbine similar to those presented in Section 2.1, in which an intercooler and/or a reheater can be enabled or disabled, so that the effects of these components on the overall cycle performance can be assessed (see fig. 4.1). Waste heat from the exhaust gases of the topping-cycle gas turbine is recovered by means of a HRSG, that raises steam working in a closed *bottoming* cycle to extract additional power. Several arrangements for the SBC have been considered, namely: single- and dual-pressure HRSGs, with or without reheat, have been analysed (see fig. 4.2).

4.1.1 ‘Novel’ Proposals

Equation 4.1 shows the formula for the overall thermal efficiency of a CCPP, as a function of the gas-turbine topping-cycle thermal efficiency and the efficiency of the heat recovery system (consisting of the HRSG plus the SBC), showcasing the need for optimising both to work together in order to maximise the performance of the power plant: the maximum achievable overall CCPP thermal efficiency will happen not for the power plant with the most efficient gas turbine, nor for the power plant with the most efficient SBC, but for the power plant in which both the topping cycle and the heat recovery system (including the performance of the HRSG) are designed to work the best together.

$$\eta_{\text{CCPP}} = \eta_{\text{GT}} + (1 - \eta_{\text{GT}}) \cdot \eta_{\text{HRSG}} \cdot \eta_{\text{SBC}} \quad (4.1)$$

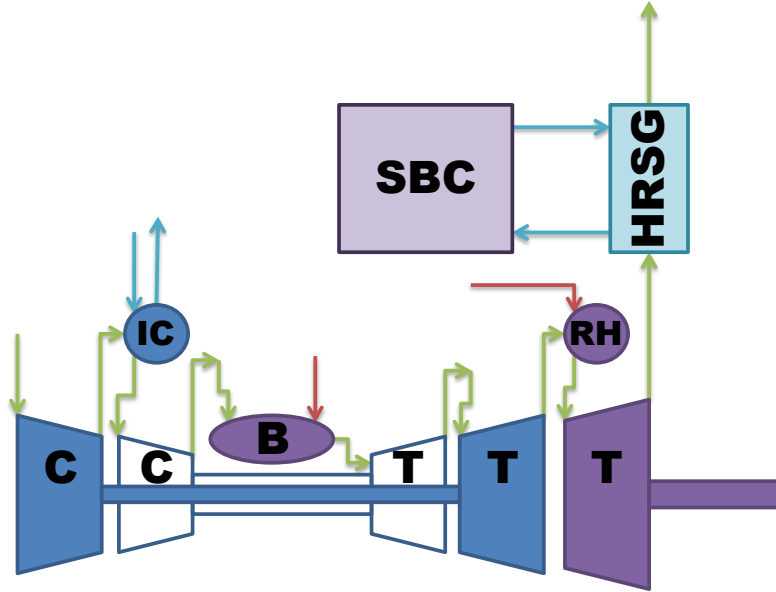


Figure 4.1: *Schematic layout of the combined-cycle gas turbine power plant*

The goal is, thus, to optimise the gas-turbine EGT so that the best possible balance between the amount of available waste heat to be recovered and the efficiency in the utilisation of this recovered waste heat for power extraction can be accomplished by means of the HRSG plus SBC system. Due to the constant rate of increase in gas-turbine firing temperatures over the years, the means of achieving this goal have changed drastically. In the past, supplementary firing within the HRSG used to be implemented to reach an adequate temperature for the SBC to provide with any actual benefit, but this became unnecessary as soon as the gas-turbine firing temperatures increased enough [113, 22]. Immediately it was apparent that, as firing temperatures would go on increasing, gas-turbine EGT could be optimised for the HRSG by either completely removing the supplementary firing or by *moving* the reheater upstream within the gas turbine (hence implementing a reheated gas-turbine topping cycle), so that additional power could be extracted from the expansion side of the gas turbine as well [88, 89]. If firing temperatures keep increasing further, which seems to be the case nowadays, it seems reasonable to think that new strategies for optimising the gas-turbine EGT may arise: in particular, waste-heat recuperation within the gas-turbine topping cycle itself was proposed and will be analysed herein. The schematics for this new layout are depicted in fig. 4.3.

Whilst many studies highlight the multiple benefits of reheated gas turbines for CCPPs, recuperated gas turbines have traditionally been considered to be poorly suited for this task [60]. Even though some studies have dealt with such a configuration for a CCPP, the open literature lacks a consistent comparative study that would clarify the potential of gas-recuperation within CCGT power plants. In particular, reheated *or* recuperated topping-cycle gas turbines have thoroughly been analysed, but the potential synergy of a combination of the two in a reheated-recuperated CCGT remains to be assessed.

Carcasci & Facchini [20] modelled the performances of CCPPs featuring a reheated gas turbine, based on the Alstom's GT24/26, against a simple-cycle H-class heavy-duty gas turbine. The two alternatives had as well different cooling systems (air cooling for the reheated gas turbine and steam cooling for the H-class), so comparison could not be made directly between them, but the potential

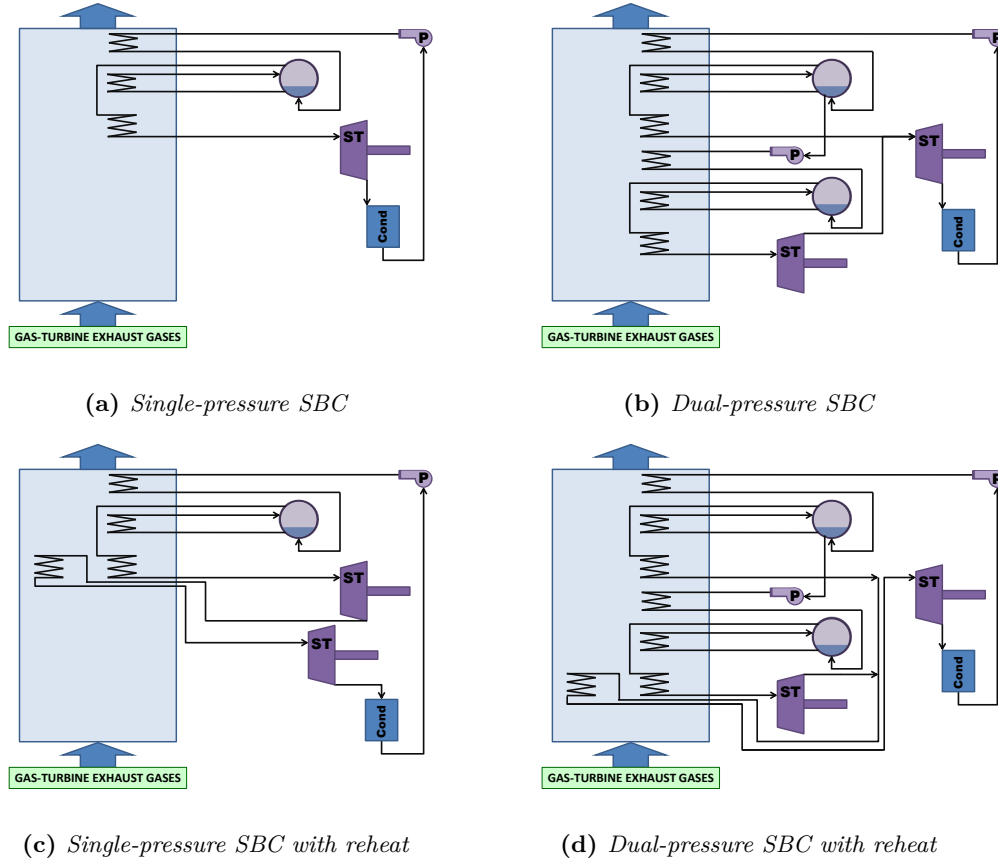


Figure 4.2: Schematic layout of the different steam bottoming cycles considered within the CCGT power plants analysed

of the reheated gas turbine was showcased. Chiesa & Macchi [24] performed analyses on CCPs featuring different blade cooling methods for both a simple-cycle and a reheated gas turbine, and reported on the superiority of the latter. Sanjay et al. [95, 96, 97] carried out similar calculations on even a wider range of blade cooling technologies, reaching the same conclusions. However, none of these works considered recuperated gas turbines within their researches.

More comprehensive studies on the optimal gas turbine configuration for CCPs were carried out by Polyzakis et al. [82, 83], analysing the performances of intercooled, reheated and intercooled-reheated gas turbines. Although their work briefly discusses the intercooled-reheated-recuperated gas turbine, this configuration is immediately discarded due to the low temperature of the gas-turbine exhaust gases. Still, no consideration is made of the (non-intercooled) reheated-recuperated gas turbine. Similarly, Briesch et al. [18] analysed the influence of intercooling, reheating and recuperation within the gas-turbine cycle. They found recuperated gas turbines for CCPs to outcompete the simple-cycle ones due to the waste heat being recovered at the gas-turbine thermal efficiency (always greater than the SBC efficiency), but still reheated gas turbines could outperform the recuperated ones. Intercooled-recuperated and intercooled-reheated-recuperated gas turbines were considered within this study, and discarded for the same reason mentioned above. Again, no consideration is made within this research about the (non-intercooled) reheated-recuperated gas turbine.

Recuperated gas turbines have their greatest potential nowadays in the solar energy market, where lower-temperature waste heat can be recovered by means of organic Rankine cycles, better suited for these applications [52]. Gogoi [48] reported on the performances of CCPs featuring recuperated gas turbines as the topping cycle, and showcased these being able to outperform the simple-cycle gas turbines by a good margin. However, their work did not consider the possibility of implementing

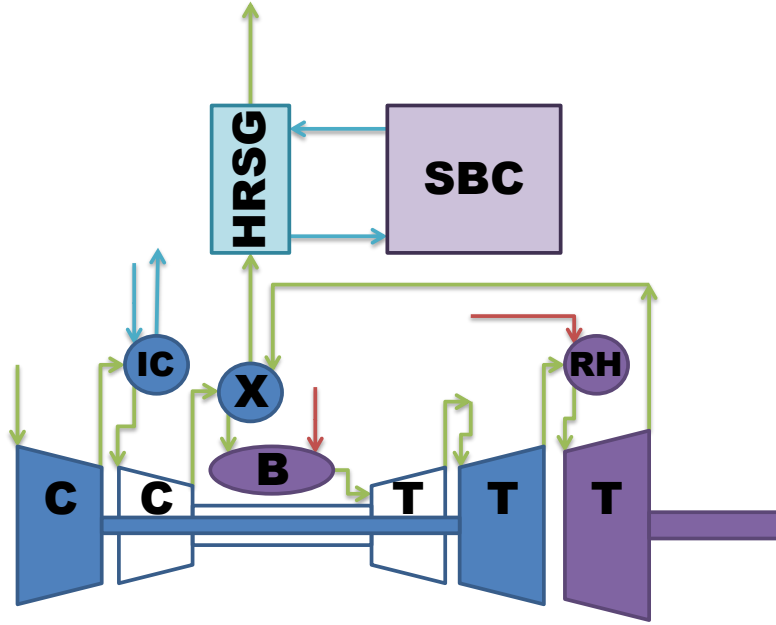


Figure 4.3: Schematic layout of the recuperated combined-cycle gas turbine power plant

reheating within the gas turbine as well. Analyses and optimisation of the reheated-recuperated gas turbine as the topping cycle for CCGTs were carried out by Bassily [13, 14, 15], and their performances compared against current state-of-the-art simple-cycle gas turbines. Bassily reported on the superiority of the reheated-recuperated gas turbines, even though no comparison was made of this configuration against the reheated (non-recuperated) one.

The results presented herein (some of which were already reported in [34]) focus, therefore, in analysing and comparing all possible configurations of *advanced* gas turbines as the topping cycle of CCGT power plants, with a consistent and coherent set of assumptions. These *advanced* gas turbine cycles will feature intercooling and/or reheating and/or recuperation, as shown in fig. 4.3, and will be coupled with different arrangements of SBCs as well, as depicted in fig. 4.2.

4.1.2 Simulation Tools

The simulation platform developed in MATLAB® and discussed in Section 2.1.1 has been enhanced, for the simulation of the CCGT power plants, by including a module capable of calculating the performance of the SBC. In this regard, STEAMOMATCH, another in-house tool developed at Cranfield University, was used. The current version of the code has undergone major rewriting of most of its functionalities by different authors over the years [31, 78, 11, 93], and thorough validation against commercially available software has been carried out and presented in [93] for single-, dual- and triple-pressure systems, with or without reheat.

4.1.3 Gas Turbine Modelling Assumptions

Given the similarities with the open ‘dry’ cycles analysed in Chapter 2, the same set of assumptions described in Section 2.1.3 apply to the gas-turbine *topping* cycle of the CCGTs. To provide a basis for consistent comparison with all other power plants considered within this research project, the total

power output requirement for the whole CCPP and the gas-turbine TIT were taken, as always, as 40 MW and 1600 K, respectively.

4.1.4 Steam Bottoming Cycle Modelling Assumptions

Steam properties are calculated by means of the *freesteam* [1] libraries, an open-source implementation of international-standard IAPWS-IF97 steam tables [55].

Heat recovery steam generator

A notional $T - h$ diagram for the typical single-pressure HRSG is presented in fig. 4.4. The main parameters that define the performance of this component are the evaporator-inlet/superheater-outlet ‘pinch’ point temperature differences (PPTDs) —i.e. the minimum temperature differences between the hot (gas) side and the cold (water/steam) side at the most critical points, for the heat transfer process to take place effectively— and the ‘approach’ point temperature margin at the economiser outlet for safe operation of the power plant. For this study, the following values will be assumed:

- Evaporator PPTD is taken as 10 K, as a typical value for the gas–water heat transfer process in modern HRSGs [30].
- The ‘approach’ point temperature margin is taken as 2 K, as a typical value for safe operation of water-drum type steam boilers nowadays [30].
- Superheater PPTD is taken as 20 K, as a typical value for the gas–steam heat transfer process in modern HRSGs [30].
- No pressure losses are accounted for within the HRSG.

Steam turbomachinery

Steam turbines’ isentropic efficiency will be taken as 90%, while water pumps’ isentropic efficiency will be taken as 85%, irrelevantly of the operating PR of any of them [30]. The LP-turbine outlet pressure —i.e. condenser pressure— will be chosen at 5 kPa, as suggested for modern SBCs by [30].

In addition to these assumptions, there are limitations to be kept in mind for feasible operation of the SBC. Typically:

- Steam temperature limit at the turbine inlet is defined as 850 K for material considerations (this is the actual temperature limit for typical steels utilised nowadays [60]).
- Steam quality limit at the turbine outlet is defined as 85% for maintenance reasons (excessive water content results in quick erosion of the last stages of the steam turbine, and thus much higher maintenance costs [60]).

4.1.5 Design Space Exploration

In addition to the ranges analysed for the free design parameters —i.e. independent variables— of the gas-turbine *topping*-cycle design space (identical to those discussed in Section 2.1.2), similarly, for the *bottoming* steam-cycle design space, the free design parameters are the operating ‘expansion ratios’ of each steam turbine. In this regard, *sub*-critical steam is always assumed, and a wide range of steam pressures (from 5 kPa to 22 MPa) will be analysed for each of the operating steam pressure levels.

With the PPTDs specified in Section 4.1.4, the steam mass flow operating in each of the steam loops can be calculated. However, these PPTDs can sometimes lead to infeasible operating points, due to two main reasons:

- The specified superheater PPTD of 20 K can sometimes be too low and result in superheated steam above the 850 K limitation for the steam temperature at the turbine inlet.

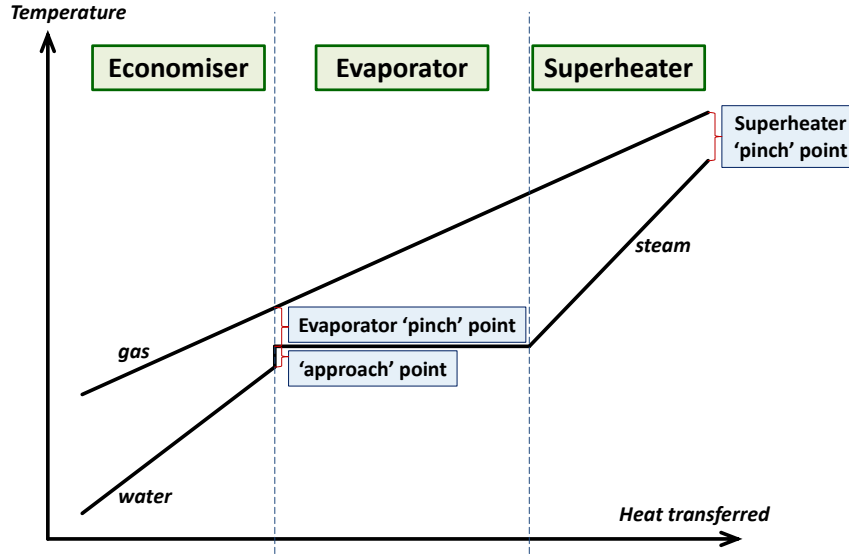


Figure 4.4: Notional $T - h$ diagram for a typical single-pressure HRSG

- The specified evaporator PPTD of 10 K can sometimes be too low and result in high steam mass flows through the economiser, that would make the enthalpy lines in the $T - h$ diagram (see fig. 4.4) to cross each other.

Should any of these two circumstances occur, the corresponding PPTD will be increased in small steps, until the minimum PPTD for feasible operation is found, and the 850 K TIT limitation is complied with or the enthalpy cross-over disappears.

4.2 Results Discussion

Among the different configurations for the SBC considered for this study (see fig. 4.2), dual-pressure systems can outperform the single-pressure HRSGs by a margin of about 2% points in the whole range of OPRs analysed. However, when it comes down to the implementation of reheating within the SBC, the benefits are barely noticeable: the dual-pressure with reheat HRSG arrangement outcompetes its non-reheated counterpart by a mere 0.25%, and only within a narrow margin of OPRs that optimise the performance of such a power plant.

Moreover, the steam pressure necessary to optimise the reheated configuration is well above 20 MPa, while the non-reheated dual-pressure SBC provides almost the same level of performance working at around 12 MPa only. For simplicity of operation and safety reasons, most of the industrial CCPPs nowadays do not exceed the 16 MPa threshold [33, 47], above which water-purity requirements to avoid deposits within the steam boiler tighten significantly. For this reason, the non-reheated dual-pressure HRSG depicted in fig. 4.2b is chosen as the *bottoming* cycle configuration that all the CCPPs will implement in order to find the optimal *topping* gas-turbine cycle.

4.2.1 Optimal gas-turbine configuration

Figure 4.5 shows the overall thermal efficiency against gas-turbine OPR for the different gas-turbine configurations analysed, all featuring the same SBC arrangement but with steam working at *optimised* pressure levels to maximise the overall thermal efficiency of *each* CCPP. As it can be seen, gas-turbine cycles with intercooling implemented (dotted lines) do feature lower thermal efficiencies than

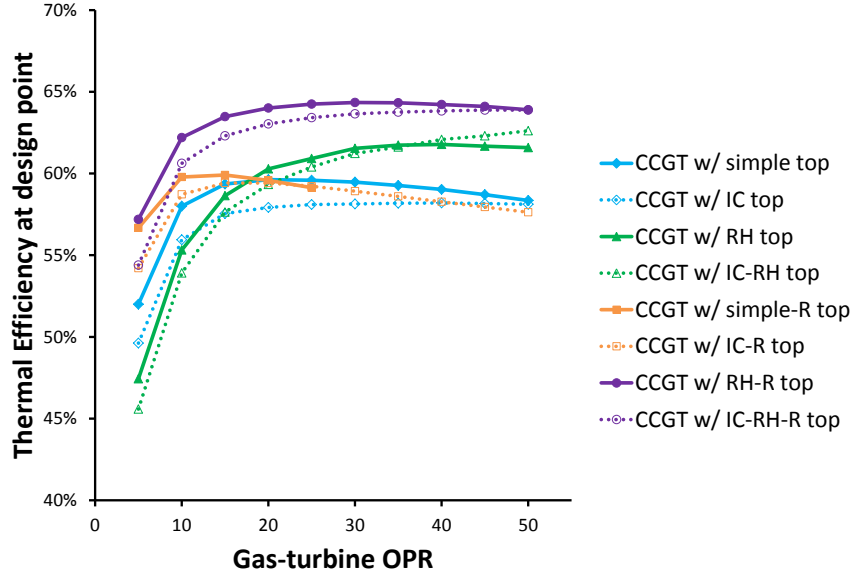


Figure 4.5: Thermal efficiency against gas-turbine overall pressure ratio for several CCGTs

their non-intercooled counterparts (solid lines). There are a few exceptions, such as the intercooled-reheated topping-cycle gas turbine outperforming the reheated-only gas-turbine cycle at extremely large OPRs (see fig. 4.5) but, in general, one can conclude that implementing an intercooler within the topping-cycle gas turbine is not beneficial for maximising the overall thermal efficiency of a CCGP.

Gas-turbine reheating, on the other hand, proves to always increase the potential to achieve high overall thermal efficiencies of any CCGP in which it is implemented, when compared against the same thermodynamic cycle without reheating implemented in the gas turbine. This does not occur for the whole range of OPRs analysed, though, and the simple-cycle topping-cycle gas turbine outperforms the reheated gas-turbine cycle for the range of lower OPRs (see fig. 4.5), for instance. Still, when both the reheated and non-reheated cycles are optimised for their maximum achievable thermal efficiency at design point, the reheated gas turbine shows greater levels of performance.

Similarly, recuperation implemented within the topping-cycle gas turbine increases the potential for higher overall thermal efficiencies of the CCGP that features such component. In spite of the difference in maximum achievable thermal efficiency at design point between the simple-cycle and the recuperated-cycle gas turbines being relatively small (see fig. 4.5), when gas-turbine reheating is implemented as well, the gains in overall thermal efficiency are significant: about 4% points for the whole range of OPRs analysed (see, for instance, the comparison between the reheated-only topping-cycle gas turbine and the reheated-recuperated gas-turbine cycle in fig. 4.5).

It is apparent, in sight of the results presented in fig. 4.5, that the RH-R topping-cycle gas turbine yields the higher thermal efficiency among all the CCGT configurations analysed, greatly outcompeting its contenders across the whole range of OPRs analysed, and making out of this gas-turbine cycle the most promising for implementation within CCGPs for maximum overall thermal efficiency at design point.

4.3 Summary

The synergy between the high-TIT low-OPR gas-turbine *topping* cycle and the low-TIT high-OPR *bottoming* steam cycle has been proven, and its potential to achieve higher overall thermal efficiencies than the ‘dry’ cycles presented in Chapter 2 is showcased in fig. 4.6.

A wide range of *advanced* gas-turbine *topping* cycles have been analysed, in search for the optimal configuration to be implemented within a CCGP. In this regard, the RH-R gas turbine proved to be

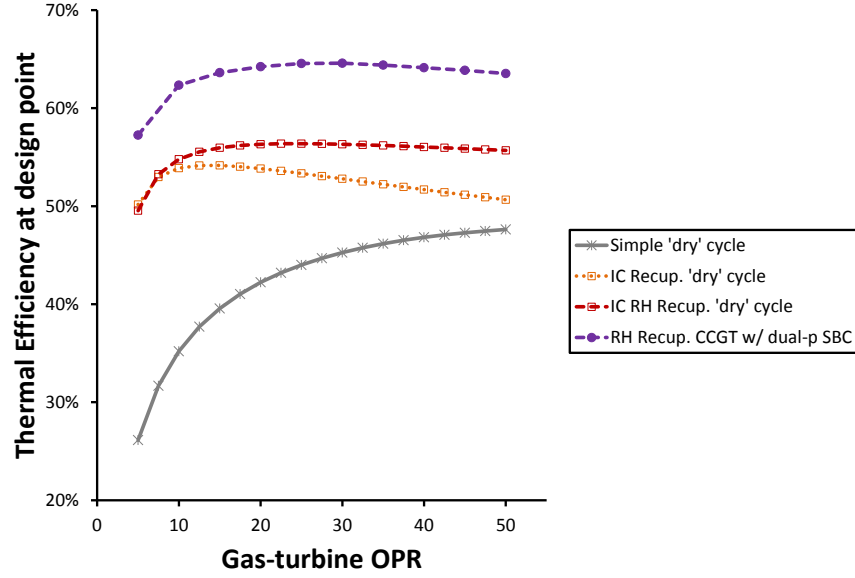


Figure 4.6: Comparison between the most promising among the open ‘dry’ cycles and the CCGTs analysed

the configuration with the highest potential to achieve maximum overall thermal efficiency at design point, yielding thermal efficiencies of around 64.5% at $OPR = 30:1$, and well above 62% for the whole range of $OPRs > 10:1$, when coupled with an optimised SBC with dual-pressure arrangement.

Dual-pressure SBC configurations outperform single-pressure arrangements in all cases. However, in spite of steam reheating implemented within dual-pressure SBCs being able to raise significantly the overall performance of the CCPP when simple-cycle (or even reheated-cycle) gas turbines are utilised as the topping-cycle configuration, when different *advanced* gas turbines are implemented instead (and particularly when the RH-R gas-turbine cycle is chosen), the optimisation of the non-reheated SBC proved to be almost equally able to reach the same levels of overall thermal efficiency.

This highlights the importance of optimising the *topping* and the *bottoming* cycles of any CCPP *together*. The ‘traditional’ approach usually followed by the industry, by which the gas-turbine *topping* cycle is typically ‘frozen’ and the best possible SBC is designed to fit it, can result in excessively complex *bottoming* cycles (such as the reheated SBCs) that, many times, are greatly outperformed by *optimised* gas-turbine *topping* cycles with simpler *bottoming* configurations.

Chapter 5

Steam Injected Cycles

Within the STIG cycles, waste heat from the exhaust gases of the *topping*-cycle gas turbine is recovered by means of a HRSG, where the available enthalpy contained in these exhaust gases is used to preheat and evaporate some water into steam, and to superheat to some extent this resulting steam that will thereafter be injected into the gas-turbine main combustor. In this way, heat that used to be rejected into the atmosphere in the ‘dry’-cycle gas turbine is now used to raise steam that will be fed back into the gas-turbine cycle, allowing for better heat utilisation overall.

As in any other ‘wet’ cycle, the working principle is rather simple: by increasing both the specific heat capacity and the mass flow circulating through the expansion side of the gas turbine —by means of injecting humidity prior to it; typically, right upstream of or directly into the combustion chamber—, the imbalance between the compressor required work and the work generated by the turbines will increase as well, allowing for greater amounts of useful work to be extracted from the free power turbine, and hence yielding higher overall thermal efficiencies for the whole power plant. Extensive literature on the STIG cycle can be found in [23, 42, 45, 66, 54, 60], just to cite a few.

In addition to the potential gains in thermal efficiency and power output, the presence of high levels of humidity in the primary zone of the main combustion chamber, due to the higher specific heat capacity of the injected steam, greatly contributes to reduce the temperature peaks resulting from the combustion in turbulent flow, thus reducing the rate of NO_x production. Injection of small quantities of steam for this purpose, with the aim to reduce the overall emission and pollution levels of the power plant —and not with the aim of obtaining increased power output and thermal efficiency— is a common strategy nowadays for gas turbine manufacturers around the globe [46].

One last advantage of the STIG cycle is the possibility of achieving short power boosts at low extra cost in situations where this may result useful. Figure 5.1 shows the output power and thermal efficiency curves against the amount of steam injected in a simple-cycle STIG, taken from the NGT studies carried out by Rolls-Royce plc. in [90].

As shown in fig. 5.1, the power output increases linearly with the amount of steam mass flow injected; on the contrary, the overall thermal efficiency of this particular STIG cycle analysed exhibits a maximum at steam mass flows ~ 20 kg/s. For steam mass flows lower than the *optimal*, the superheater outlet of the HRSG will be ‘pinched’ (see $T - h$ diagram in fig. 4.4), so the steam will be injected at the maximum achievable temperature into the combustion chamber, but lower steam mass flows injected will mean lower work imbalance between the compression and the expansion side of the gas turbine and hence lower useful power available. On the other hand, for steam mass flows greater than the *optimal*, an internal ‘pinch’ will occur at the inlet of the evaporator (see fig. 4.4, again) that will result in the steam temperature at the outlet of the HRSG superheater to drop, thus increasing the fuel consumption necessary to reach the desired TIT for this particular STIG cycle, due to the injection of greater quantities of *cooler* steam in the main combustor. The *optimal* steam mass flow of ~ 20 kg/s appears, therefore, where both the *internal* pinch at the evaporator inlet and the *external* pinch at the superheater outlet happen at the same time: this means that the maximum possible amount of steam —with the available enthalpy from the gas-turbine exhaust gases— is being raised within the HRSG of the STIG cycle, while still maintaining the maximum achievable superheated-steam temperature —limited by the gas temperature at the outlet of the free

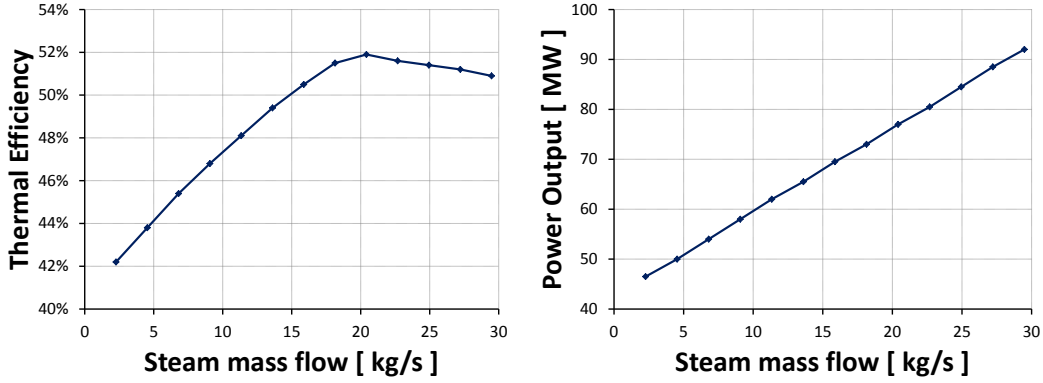


Figure 5.1: Thermal efficiency and power output curves at gas-turbine design point against steam mass flow injected, after [90]

power turbine of the *topping* cycle—for injection of “maximum quality” steam into the gas-turbine main combustor.

It is apparent then in fig. 5.1 that the increased fuel consumption resulting for injecting steam mass flows greater than the *optimal* is partially compensated by the increased useful power generated due to the increased mass flow and specific heat capacity of the humid air circulating through the expansion side of the gas turbine, while the loss in useful power extraction for injected steam mass flows lower than the *optimal* totally outweighs the fuel savings in this region. For this reason, the thermal efficiency drops much more steeply for reduced steam mass flows than it does for increased steam mass flows, and so short power boosts by simply increasing temporarily the water mass flow circulating through the HRSG pumps are achievable without great penalisation in terms of fuel consumption.

5.1 Modelling & Assumptions

Gas turbine models analysed consist of two-spool configurations with free power turbine, as depicted in fig. 5.2, in which intercooling and reheating can be enabled or disabled, to assess their impact on the STIG cycle performance. Figure 5.2a shows the basic schematics for a “*plain*” STIG cycle, in which steam is raised by means of a single-pressure HRSG (see diagram in fig. 5.3a) directly at the injection pressure.

Following the discussion in Section 4.1.1, the effect of implementing gas-recuperation within the topping-cycle gas turbine (as in fig. 5.2c) of a STIG cycle will be studied as well, in spite of the fact that recuperated gas turbines have traditionally been considered as not very well suited for this sort of power plants.

Even though the most common arrangements for the STIG power plants comprise only the single-pressure HRSG and direct injection of the superheated steam into the gas turbine main combustion chamber, some modifications have as well been proposed over the years (see [45, 54, 60], for instance) in which steam is initially raised at a much higher pressure than the operating pressure of the gas turbine main combustor, and thus it can be partially expanded in a steam turbine before the injection takes place, hence producing some additional power. These cycles are usually referred to as “*combined*”-STIGs, which can also feature recuperated or non-recuperated topping-cycle gas turbines, as in figs. 5.2b, 5.2d.

When considering the “*combined*”-STIG cycles, however, a broader range of configurations for the HRSG arise, other than the single-pressure arrangement. Similarly to the SBCs considered in Section 4.1, a whole range of *open* steam cycles could be devised, in which the LP steam turbine and the condenser from figs. 4.2a–4.2d would be replaced by steam injection into the gas-turbine main combustor, and fresh water would be provided to the LP water pump.

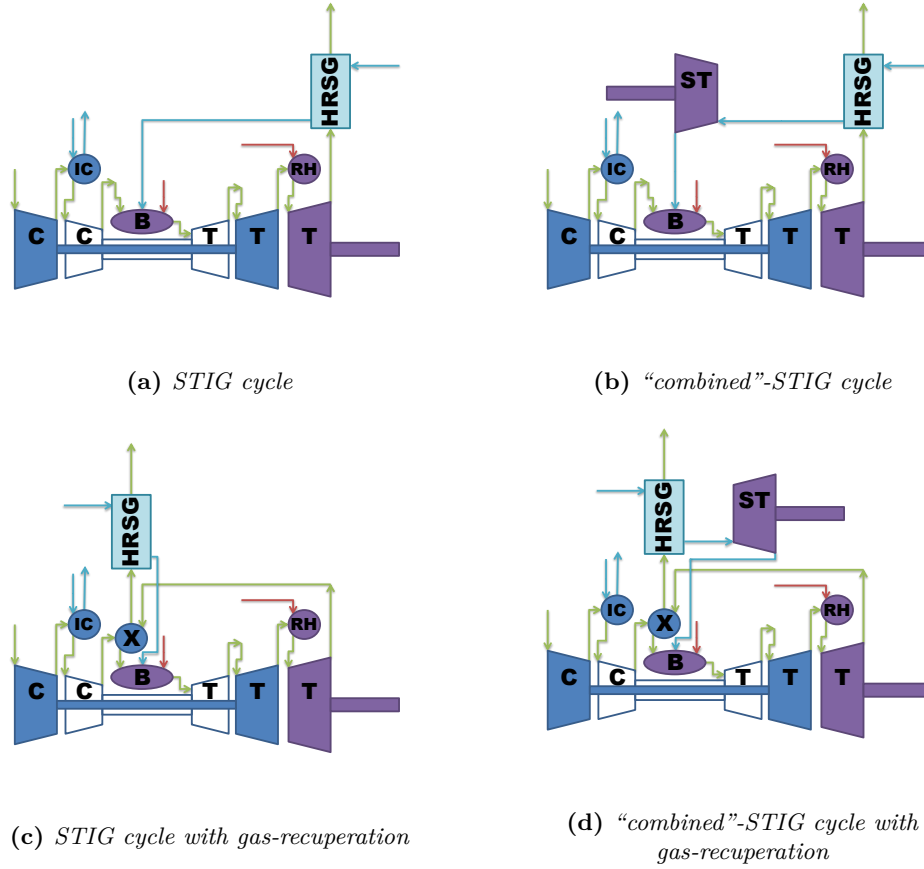


Figure 5.2: Schematic layout of the different STIG arrangements considered

Diagrams of the *open* SBC for the different STIG configurations analysed are represented in fig. 5.3. Figure 5.3a corresponding to the “*plain*” STIG—or, simply, STIG—cycle, the rest of figs. 5.3b–5.3d depict the different HRSG arrangements considered for the “*combined*”-STIGs. Namely:

- dual-pressure HRSG where the HP steam is mixed with the LP steam after expansion in the HP steam turbine and then injected into the gas turbine main combustor (see fig. 5.3b), in what is usually called the combined-STIG (or C-STIG) cycle *per se*; or
- single-pressure HRSG with reheated steam after the expansion carried out in the HP steam turbine (see fig. 5.3c), in what will be referred to as the reheated-steam injected gas turbine—or, simply, reheated-STIG (R-STIG) cycle—; or, finally,
- dual-pressure HRSG with reheated steam after mixing of both the LP and the HP steam streams (see fig. 5.3d), in what will be called the combined-reheated-STIG (or C-R-STIG) cycle.

5.1.1 Gas Turbine Modelling Assumptions

Given the similarities with the open ‘dry’ cycles analysed in Chapter 2, the same set of assumptions described in Section 2.1.3 apply to the gas-turbine *topping* cycle of all the STIG power plants analysed herein. Humid air is treated within the gas turbine as an ideal mixture of ideal gases by TURBOMATCH. To provide a basis for consistent comparison with all other power plants considered within this research project, the total power output requirement for the whole STIG cycle and the gas-turbine TIT were taken, again, as 40 MW and 1600 K, respectively.

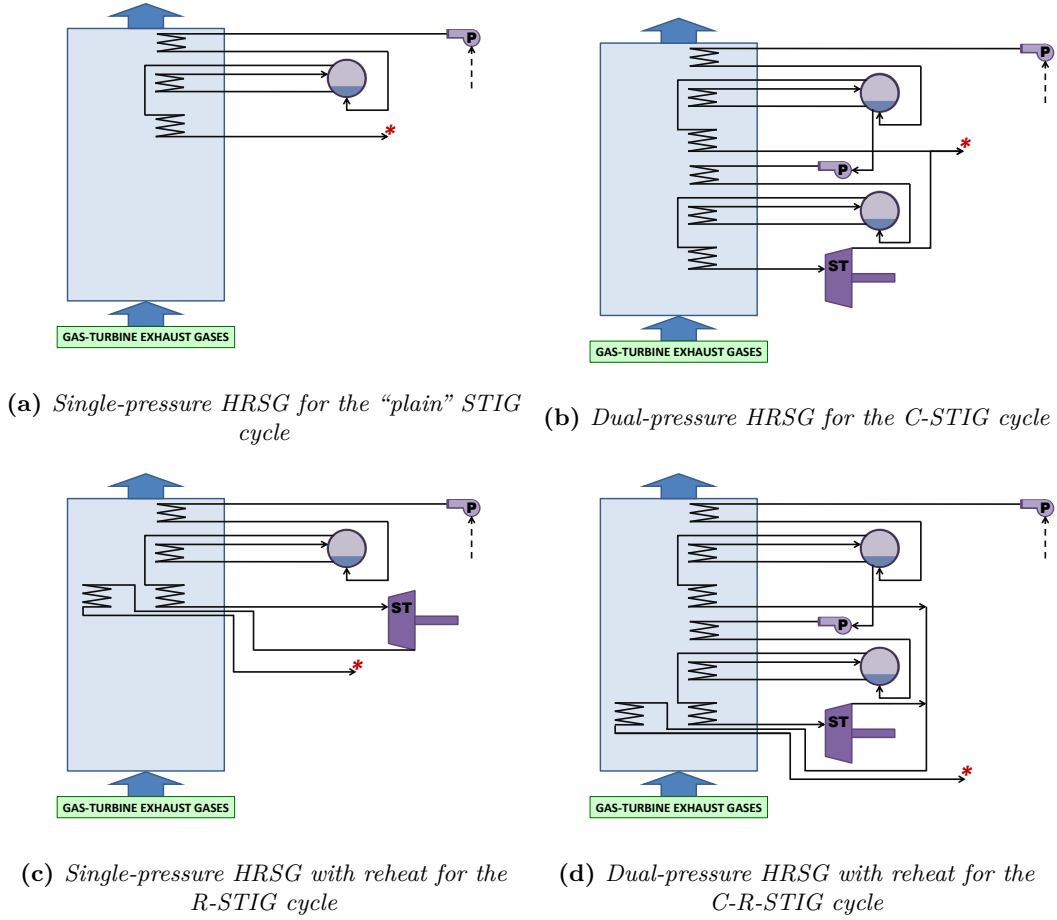


Figure 5.3: Schematic layout of the different HRSG arrangements considered within the different STIG cycles analysed

5.1.2 Steam Bottoming Cycle Modelling Assumptions

Steam operates within the STIG power plants in an *open* bottoming cycle but, other than that, the thermodynamic performances of the HRSGs, water pumps and steam turbines are identical to those presented in Chapter 4 for the CCGTs. Thus, the same set of assumptions described in Section 4.1.4 applies to the *open* SBCs of the STIG cycles.

5.2 Results Discussion

The aim, as always, is to optimise both the *topping* cycle gas turbine and the heat-recovery steam *bottoming* cycle to work best together, so as to maximise the overall thermal efficiency of the power plant at design point. According to figs. 5.2–5.3, there are 32 possible configurations —between “plain” STIGs and “combined”-STIGs— to analyse, which will be parametrically optimised in terms of compressor PRs for the gas turbine and steam operating pressures and mass flows for the HRSG, following the same procedure detailed in Section 4.1.5.

The most relevant findings of such studies will be highlighted herein.

5.2.1 Optimal gas-turbine configuration

STIG cycles with intercooling implemented within the gas-turbine topping cycle do, in general, feature lower thermal efficiencies than those without intercooling. Just the same as with the gas-turbine

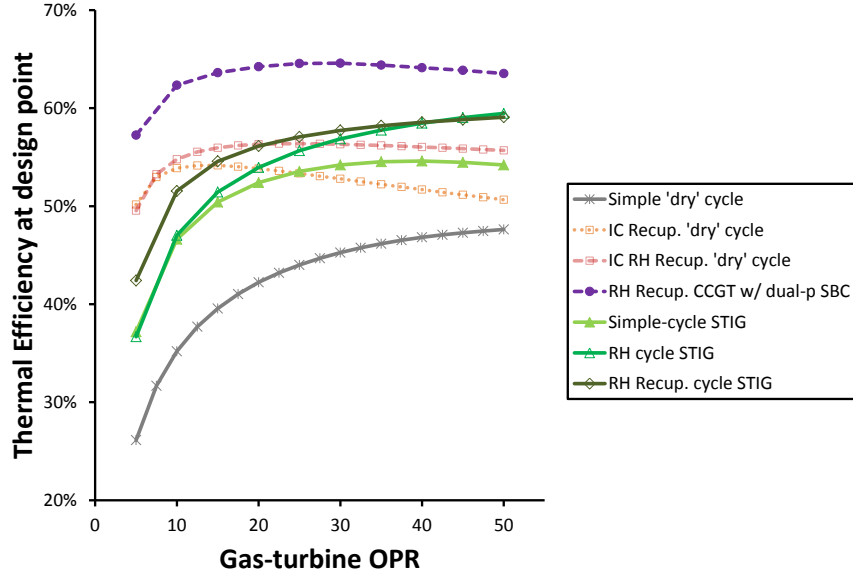


Figure 5.4: Thermal efficiency against gas-turbine overall pressure ratio for several STIG power plants with different gas-turbine topping cycle configurations

configuration within the CCGTs presented in Section 4.2.1, there are a few exceptions in which the performance levels of an *optimised* STIG featuring an intercooled topping-cycle gas turbine are almost equal to the levels achieved by its *optimised* non-intercooled counterpart, but there are no cases in which the intercooled gas turbine can clearly outperform the non-intercooled one. And so it is concluded that intercooling within the gas-turbine topping cycle is not beneficial for maximising the overall thermal efficiency of the STIG power plants either.

STIG power plants with reheating implemented within the gas-turbine topping cycle, on the other hand, do always prove to yield higher overall thermal efficiencies than those with non-reheated gas turbines. The additional enthalpy available in the exhaust gases, once recovered by means of the HRSG and fed back into the cycle, allows for more power extraction in the expansion side of the gas turbine, greatly outweighing the extra fuel being burnt in the reheater. Figure 5.4 shows the potential gains in thermal efficiency for the “*plain*” STIG cycle, when featuring a *simple-cycle* gas turbine or a *reheated* gas turbine instead—these gains are most significant at high gas-turbine OPRs—and compared as well against some of the most promising open ‘dry’ cycles and the best among the CCGTs studied, just for reference.

Gas-turbine recuperation within the STIG cycles, using the high-temperature waste heat within the gas turbine before streaming the gas-turbine exhaust gases into the HRSG for raising steam, does in general drop slightly the maximum achievable thermal efficiency of non-reheated gas turbines or, in the best case scenario, does not improve its thermal efficiency at all. However, when the topping-cycle gas turbine does feature reheating as well, the synergy between these two components becomes apparent, and the overall thermal efficiency of such reheated-recuperated STIG power plant does actually improve by a fairly large margin when compared against the reheated-only STIG, as shown in fig. 5.4, especially for the range of lower gas-turbine OPRs. Overall, the RH-R STIG cycle can outperform the simple-cycle STIG by about 4% points for the whole range of gas-turbine OPRs.

For all the above, reheated-recuperated (RH-R) gas-turbine topping cycles usually show the best potential for thermal efficiency from within all the configurations studied.

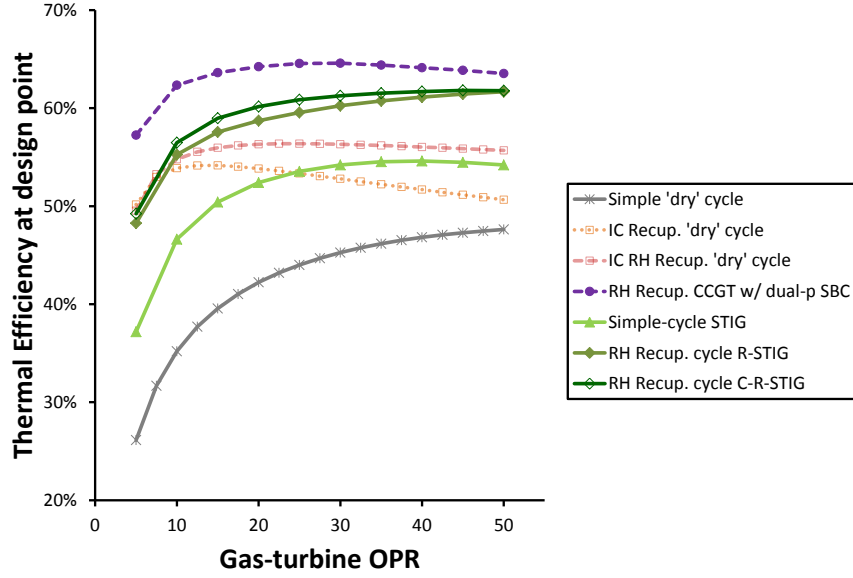


Figure 5.5: Thermal efficiency against gas-turbine overall pressure ratio for several STIG power plants with RH-R gas-turbine topping cycle and different HRSG configurations

5.2.2 Optimal heat-recovery steam *bottoming* cycle arrangement

Once established the best possible configuration for the gas-turbine *topping* cycle, fig. 5.5 shows the comparison between the most promising *bottoming* steam cycles analysed for the STIG power plants, compared against the simple-cycle “*plain*” STIG for reference.

As shown, the most complex dual-pressure HRSG with reheated steam (C-R-STIG cycle) in fig. 5.5 yields the highest overall thermal efficiency among all the STIG power plants analysed, outperforming the “*plain*” STIG with *identical* RH-R topping-cycle gas-turbine configuration plotted in fig. 5.4 by at least 2% points across the whole range of gas-turbine OPRs considered, and exhibiting as much as 4% points improvement for the lower end of this range. However, single-pressure HRSGs with reheat (R-STIG cycles) perform almost equally as well (see fig. 5.5 again), which highlights the importance for any among the “*combined*”-STIG cycles of featuring steam-reheating after the HP steam-turbine expansion, hence injecting the steam into the gas-turbine combustion chamber at the highest possible temperature.

5.3 Summary

A broad range of different arrangements for the STIG power plants has been studied and each of them parametrically optimised for maximum thermal efficiency at design point. Among these, the C-R-STIG with RH-R topping-cycle gas turbine is the most promising configuration, yielding overall thermal efficiencies as high as 62% for gas-turbine OPRs of about 45:1 (see fig. 5.5). However, this cycle does have a few drawbacks:

- Extremely high steam pressures, well above 20 MPa, are required in the HP steam loop of the C-R-STIG cycle to yield such high thermal efficiencies. As discussed in Section 4.2, the risk of silica and other salts depositions increases greatly above the 16 MPa threshold, tightening significantly water-purity requirements for operation of HRSGs at higher pressure levels. Evaporated salts entering the tubes of the HRSG can lead to deposits or blockages, that can eventually result in hazardous leakages of superheated high-pressure steam or even total failure of the boiler [33, 47], thus making safe operation of these power plants that much more challenging and greatly increasing the acquisition and maintenance costs for such components.

- In spite of the greater operating HP steam pressure and increased complexity of the HRSG for this C-R-STIG cycle, CCGTs with (non-reheated) dual-pressure SBCs and an *identical* RH-R topping-cycle gas-turbine arrangement can greatly outperform the C-R-STIG power plants (see fig. 5.5, again), operating at much lower steam pressures (about 12 MPa, only) and even at lower gas-turbine OPRs.

For these two reasons, the C-R-STIG cycle is not considered a very promising option for the purposes of this research.

To avoid the high steam operating pressures, “*plain*” STIG cycles are great candidates. These cycles, in turn, have the additional advantage of implementing a single power turbine within the gas-turbine topping cycle and no steam turbines whatsoever, and thus deployment of such power plants is much simpler due to all their useful power being delivered in the same shaft, without the need for any gearboxes. However, the maximum achievable thermal efficiency by means of a RH-R gas turbine working in “*plain*” STIG cycle configuration is barely above the 59% mark, at gas-turbine OPRs > 50:1 (see fig. 5.4).

This means that “*plain*” STIGs, even when featuring the most suitable *advanced* gas turbines as the topping cycle, are vastly outcompeted by CCGTs (see fig. 5.4, again); and even the open ‘dry’ cycles (see results presented in Section 2.2 and summary in Section 2.3) do not fall that far behind. And thus, “*plain*” STIG power plants do not seem a viable option for powering large cargo vessels, either.

Chapter 6

Evaporative Gas Turbines

One of the main limitations of the STIG cycle is the unavoidable exergy losses within the HRSG. As shown in fig. 4.4, pinch conditions occur at the inlet of the evaporator in normal operation of the typical boiler, hence the heat transfer process for raising and superheating the steam is carried out across great temperature differences between the air and the steam sides of the heat exchangers. The greater the temperature difference between both sides of a heat exchanger, the greater its exergy losses will become, and thus an attempt is made with the EvGT cycle to overcome this issue and achieve the highest possible exergetic efficiency within the process of ‘injecting’ humidity into the gas-turbine cycle.

For this purpose, in the EvGT, the air stream from the compressor outlet of the gas turbine is circulated through an aftercooler (AC), where its temperature will drop. Hot water coming from the water side of this AC, as well as from another gas–water heat exchanger —the “exhaust-cooler” (EC)— that will carry out the cooling of the exhaust gases, and even from an hypothetical intercooler (IC) implemented within the cycle, will be sprayed from the top of a packed-bed humidifier (labelled “H” in fig. 6.1), whilst cool air from the AC circulates from the bottom of this saturation tower. This process, in which water and air streams are directly in contact, results in both *heat* and *mass* transfer from the water into the air stream—even below the boiling temperature of water, recovering a great amount of low-temperature (i.e. *low-quality*) waste heat back into the cycle—, and creates a gradient of temperatures through the height of the humidifier in which the temperature difference between both streams is always kept to a minimum.

Air circulating through the humidifier absorbs both *heat*, increasing its temperature, and *mass* from the water stream, due to the difference in water concentration between both streams and not due to water boiling, which results in most of the water impurities and salts remaining within the liquid stream circulating downwards through the saturation tower, and effectively makes the humidifier behave as a water-treatment device *per se*. This fact, along with the high levels of moisture content achieved in the exhaust gases and the possibility of condensing most of this humidity within the EC, facilitates self-sufficient operation of the EvGTs in terms of water requirements, with the whole water circuit working in a closed-loop arrangement, and without the need for make-up water from the exterior [8, 5, 12], which in turn means lower specific investments for the deployment and operation of the EvGT power plants.

Still, humid air coming from the humidifier is at relatively low temperatures (always lower than the water boiling temperature at the working pressure of the humidifier), and thus the implementation of waste-heat recuperation from the exhaust gases within the gas turbine cycle is always feasible and greatly improves the overall thermal efficiency of the power plant. In this way, high-temperature waste heat from the gas turbine exhaust gases is utilised to increase the temperature of the humid air before entering the combustion chamber (so that less fuel needs to be burnt to reach the specified TIT), whilst only the low-quality heat remaining afterwards circulates through the EC to produce additional hot water that will be injected into the humidifier.

The diagram in fig. 6.1 shows as well the available by-pass at the compressor outlet that will result in the so-called ‘part-flow’ EvGTs, in which only a fraction of the total compressed air mass flow circulates through the AC and the humidifier (in an evaporative cycle) and the remaining frac-

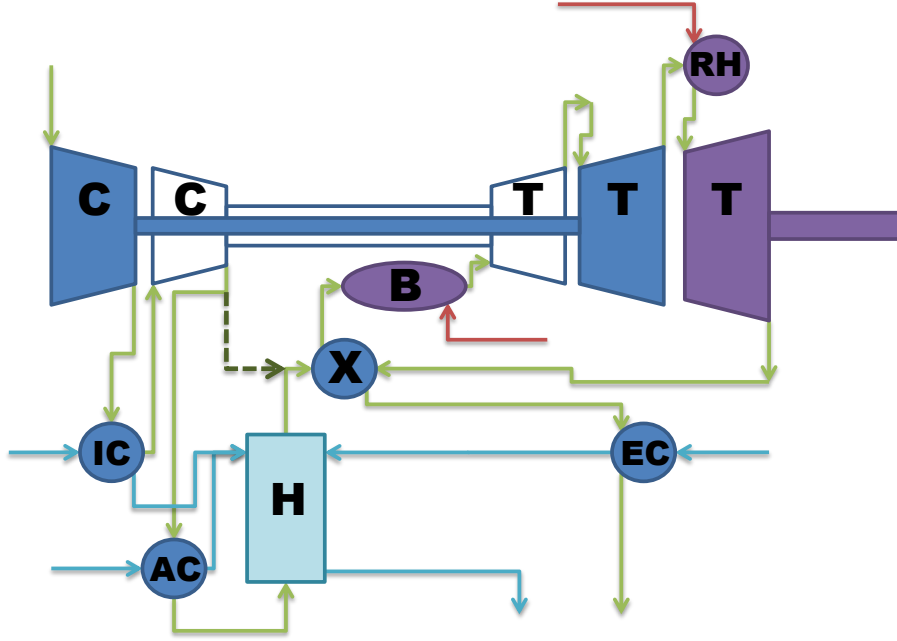


Figure 6.1: Schematic layout of the evaporative gas turbine cycle

tion by-passes both components just to be streamed directly into the recuperator (experiencing the thermodynamic processes of a recuperated ‘dry’ cycle). By adjusting this by-pass split ratio, the optimum trade-off between the ‘wet’ and the ‘dry’ cycle features can be found, and the potential of achieving even greater overall thermal efficiencies than just by either of the two arises. Further and extensive literature on this sort of arrangement, as well as many other similar proposals, can be found in [45, 38, 86, 67, 5, 91, 54], amongst others.

6.1 Modelling & Assumptions

Models analysed consist of two-spool plus free power turbine arrangements, as depicted in fig. 6.1, which are essentially the conversion into evaporative ‘wet’ cycles from those models presented in Section 2.1. Due to the particularities of the saturation process taking place within the humidifier, the AC, the EC and the recuperator (labelled “X” in fig. 6.1) are always necessary in these cycles to achieve maximum thermal efficiency. Intercooling and/or reheating, however, can be enabled or disabled at will, to assess their impact on the EvGTs design and performance.

To allow for fair comparison among the very different power plants considered within this research, the power output and TIT for the EvGT power plants are taken as 40 MW and 1600 K, respectively, identical to the values assumed for every other contender analysed.

6.1.1 Simulation Tools

Gas turbines, as always, will be simulated with TURBOMATCH. In addition to this software, a new model for quick assessment of the performances of the humidifier at design point was developed in the context of this research, aimed at calculating the outlet conditions of the humid air from the saturation tower knowing only the inlet conditions for both the air and water streams. The main objective is to be able to simulate a huge number of design points of EvGT power plants in reasonable

times, and so the model described herein lacks the flexibility and robustness of, for instance, the work presented by Enick et al. [39], but can accomplish the evaluation of a humidifier at design point in just a fraction of the time, which is extremely convenient for design-space exploration strategies.

Additionally, the water sides of the IC, the AC and the EC have to be evaluated as well, as these are the sources of hot water that will be streamed into the humidifier. Given a certain amount of enthalpy available (at a certain temperature) in the gas side of any gas–water heat exchanger, there is a minimum amount of water mass flow that can ensure heat transferred between the two streams will match the specified effectiveness of the heat exchanger. However, above this minimum, water mass flow can be increased, still removing all the available enthalpy from the gas stream but accomplishing this by means of a smaller temperature leap in the water side. It is apparent, therefore, that there is a certain trade-off to be studied between the amount of water injected into the humidifier and the temperature at which this stream of water is injected. Depending on the particularities of the design-point pressure and temperature profiles for a given EvGT, the effect of this trade-off can be of great importance.

Humidifier Modelling

Many works on the thermodynamics of the saturator columns are available in the literature, showcasing different strategies to approach the modelling of the humidifier tower with various degrees of complexity and detail [39, 65, 105, 29, 91, 57]. Most of them, though, rely heavily in discretising the humidifier into a number of small steps and evaluating heat and mass transfer in each of the small sections of the tower, sometimes requiring additional simplifying assumptions or experimental coefficients and correlations to fine-tune the model. These models are, generally speaking, time-consuming, and require additional inputs, such as the detailed geometry of the column, for instance, that may be unknown at the preliminary stages of design of new EvGT power plants. As a first-order approach, and given that the only focus of this research is on design-point performances, the decision was made to implement a simpler model for this study, based only upon first thermodynamic principles.

Equations governing a saturation process can be found in [9]. To a particular differential section of the height of the saturator, we can apply the equation for the conservation of mass to the dry air (eq. 6.1a), equation for the conservation of mass to the water side (eq. 6.1b), and equation for the conservation of energy to the whole system (eq. 6.1c).

$$m_a = \text{const.} \quad (6.1a)$$

$$m_v - m_w = \text{const.} \quad (6.1b)$$

$$m_a h_a(T_g) + m_v h_v(T_g) - m_w h_w(T_w) = \text{const.} \quad (6.1c)$$

Dividing eqs. 6.1b and 6.1c by eq. 6.1a, we can define the two invariants of the humidifier, which remain constant throughout any section of the saturator column. Equation 6.2 defines the *mass flow invariant* of the humidifier, not to be confused with the ratio of liquid-water mass flow to dry-air mass flow. Equation 6.3, after some very straightforward algebra, defines the *enthalpy invariant* of the humidifier.

$$\frac{m_w}{m_a} - \omega = M^+ \quad (6.2)$$

$$h_a(T_g) + \omega [h_v(T_g) - h_w(T_w)] - M^+ h_w(T_w) = H^+ \quad (6.3)$$

Knowing the air inlet conditions and the water inlet conditions to the saturator column is not enough information to determine both invariants. And thus an additional hypothesis has to be made, i.e. it will be assumed that the air exiting the humidifier from the top of the tower is completely saturated, which is a reasonable assumption for any humidifier working at design point [6, 7]. Hence, calculating the water to dry-air ratio for the gas outlet section can be done by simply replacing $\Phi = 1$ in eq. 6.4 (assuming that the humid air behaves as an ideal mixture of ideal gases).

$$\omega = \frac{\Phi p_{\text{sat}}(T_g)}{p - \Phi p_{\text{sat}}(T_g)} \frac{R_a}{R_v} \quad (6.4)$$

Three main stations are defined within the saturation tower. Namely:

Station “0”, corresponding to the ‘pinch’ conditions, in which the operating and saturation enthalpy lines of the humidifier would be tangent to each other for an *ideal* process. At this station, the air is assumed to be completely saturated by definition and, even though these conditions (which pressure, temperature and specific humidity are unknowns beforehand) do not have to actually take place within the humidifier, analytical expressions can be derived for the invariants in this case, which makes it a very interesting point from the thermodynamic point of view [9]. In a real process, however, a finite separation between the operating and the saturation enthalpy lines, ΔT , will occur. This separation, characteristic of the *technology level* and the size of the humidifier, has been assumed to be of 5 K, as a representative typical value for modern saturation towers [9].

Station “1”, corresponding to the bottom section of the saturator —i.e. air inlet/water outlet—, for which the unknowns are the water conditions (temperature and mass flow). Air is not completely saturated at this station, but all its properties are known from the calculations of the compression/aftercooling processes.

Station “2”, corresponding to the top of the column —i.e. water inlet/gas outlet—, for which the unknowns are the gas conditions (temperature, specific humidity and, hence, mass flow). Air is assumed to be completely saturated at this point, as per hypothesis made, and with this additional equation the system is closed and water and air conditions can be solved throughout the humidifier, at the predefined stations. Water conditions are calculated from the gas–water heat exchangers implemented within the cycle, assuming constant specific heat capacity for the liquid-phase water, as will be detailed next.

Full details on the final, simplified expressions of the system of equations solved for a particular design point of the humidifier are provided in Appendix B.1, along with the detailed explanation of the necessary checks that must be performed to ensure proper implementation of this model. Validation of the current model is provided in Appendix B.2.

Water Calculations

Water must be injected into the humidifier always in liquid phase. To ensure this, calculations for the water side of the gas–water heat exchangers implemented within the EvGT cycle —i.e. IC, AC and EC— are performed in the following way:

1. The maximum temperature that water can be injected at is taken as the boiling temperature at 90% of the actual operating pressure of the humidifier, to allow for a small safety margin.
2. For each heat exchanger —i.e. IC, AC and EC—, a check is made to assess whether, based on the inlet temperature of the hot (gas) side, the water can actually be heated up till the maximum temperature of injection.
 - (a) If so, water outlet temperature of the heat exchanger is taken equal to the maximum temperature of injection.
 - (b) If not, water outlet temperature is that one corresponding to a 90% *effectiveness* heat exchanger.
3. For each heat exchanger, water mass flow is calculated, based on the enthalpy drop from the hot (gas) side, corresponding to the selected water outlet temperature.
4. Water streams from the three heat exchangers are thereafter mixed and injected into the humidifier.
5. An additional degree of freedom is introduced within the models, to allow for increased water mass flow through the heat exchangers, so that the water temperature will drop. This way, the trade-off between injecting higher mass flows of water at lower temperatures (or *vice versa*) into the humidifier will be explored.

6.1.2 Modelling Assumptions

Other than what has been previously discussed regarding the modelling of the humidifier and the water side of the IC, AC and EC, and given the similarities between the EvGT models and the open ‘dry’ cycles presented in Chapter 2 except for the increased water content in the expansion side of the ‘wet’ cycles, the typical set of assumptions for the gas turbines discussed in Section 2.1.3 apply to the EvGTs as well.

6.2 Results Discussion

According to the schematics in fig. 6.1, there are four possible EvGT cycles to analyse, by means of enabling or disabling intercooling and/or reheating. However, prior to assessing the effect of these components in the cycle’s performance, a brief study and discussion on the performance of the isolated humidifier, and the influence of the water mass flow and temperature injected into it, is paramount.

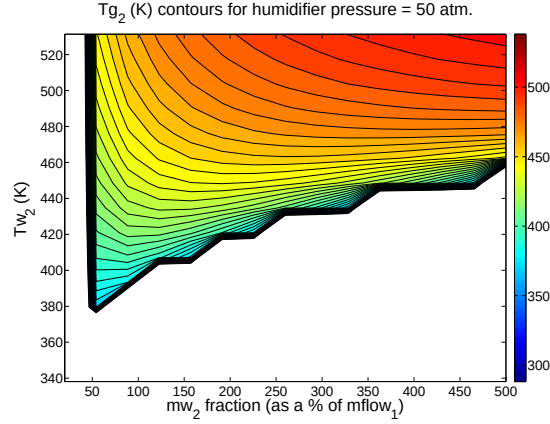
6.2.1 Design Space Exploration

In figs. 6.2a–6.2b, contours of specific humidity —i.e. water to dry-air ratio, in mass— and temperature of the humid air at the outlet of the humidifier are plotted for a wide range of different water inlet conditions, assuming a constant set of known air inlet conditions (temperature, pressure, specific humidity and total mass flow). Water mass flow injected into the humidifier is varied between $1/5$ and 5 times the air mass flow circulating from the bottom of the column. Water inlet temperature is varied from slightly over ambient temperature to slightly below its boiling point at the operating pressure.

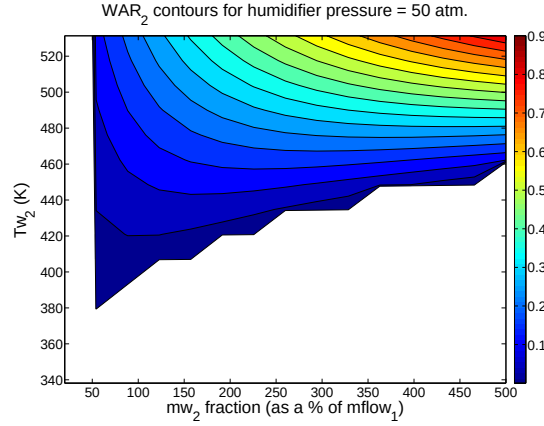
The maps in fig. 6.2 show that the greater the enthalpy injected through the water inlet into the humidifier —i.e. the greater the water mass flow and the greater the water temperature—, the more water will be absorbed into the air stream and the higher the temperature of the humid air at the saturator outlet will be. Two blank regions —in which the model fails to produce any predictions of the resulting thermodynamics— appear in the aforementioned maps, that must be taken into consideration when assessing the limitations of the presented model for the humidifier:

- First, for water injected mass flows lower than 50% of the air mass flow, a blank region appears that corresponds to lack of enough water mass flow to actually saturate the air stream. Hence, this is not a region of infeasible design points, but a region in which the hypothesis made of completely saturated air at the humidifier outlet is invalid. Feasible humidifiers with partial saturation could be designed in this region; however, it is anticipated that these design points are far away from the points of maximum overall thermal efficiency of the whole power plant (in such case in which the water mass flow injected into the humidifier would be so limited, part-flow EvGTs with the optimal amount of air by-passing the aftercooler and the humidifier would perform better than the partial-saturation humidifiers), and thus are not of much interest for this study.
- Second, for any water injected mass flow, there is a minimum water temperature below which the humidification process is not feasible. The temperature of the water is not high enough to heat the humid air up to a point in which additional absorption of humidity into the air would happen, and water would be condensed rather than evaporated through the humidifier. These are indeed infeasible design points and are not due to model limitations.

Once coupled with the rest of the EvGT cycle, the amount of water injected into the humidifier and its temperature are not independent variables any more, but related to each other by means of the water calculations described in Section 6.1.1: given a particular design point for a particular EvGT configuration, there is a minimum water mass flow that can be circulated through the different gas–water heat exchangers (IC, AC and EC), which will produce the highest achievable water temperature; however, higher water mass flows are also feasible, that will result in lower water temperatures. This effectively defines a single *curve of feasible humidifier designs* within the maps in figs. 6.2a–6.2b for each design point of each EvGT configuration. This *curve* can be parametrised with a single



(a) Contours of temperature of the humid air at the outlet of the humidifier, T_{g2} (K), for varying water inlet conditions



(b) Contours of water content of the humid air at the outlet of the humidifier, ω_2 , for varying water inlet conditions

Figure 6.2: Performance maps of the isolated humidifier, for constant air inlet conditions (100 kg/s at 288.15 K), operating at 50 atm

independent variable, as discussed in Section 6.1.1, which effect on the overall thermal efficiency of the EvGT power plants will be studied, along with the effect of the available by-pass discussed in Section 6.1 (see also fig. 6.1).

6.2.2 Performance of the EvGT Power Plants

Figure 6.3 plots thermal efficiency against OPR curves for the four EvGT configurations analysed, along with the thermal efficiency curves for the most promising among the CCGTs and the open ‘dry’ cycles. From this figure, it is apparent the huge potential of the EvGT power plants to be deployed in applications in which the main requirement is to provide the highest possible achievable thermal efficiency at design point, as is the case for the prime movers of large cargo vessels.

The two most interesting configurations analysed correspond to the ‘equivalent’ EvGTs of the IC-R and IC-RH-R ‘dry’ cycles presented in Sections 2.2.5 and 2.2.7. Comparison between the thermal efficiencies achieved by these ‘wet’ EvGTs and their ‘dry’ versions highlight the potential of water absorption as a means to raise the overall thermal efficiency of *advanced-cycle* gas turbines.

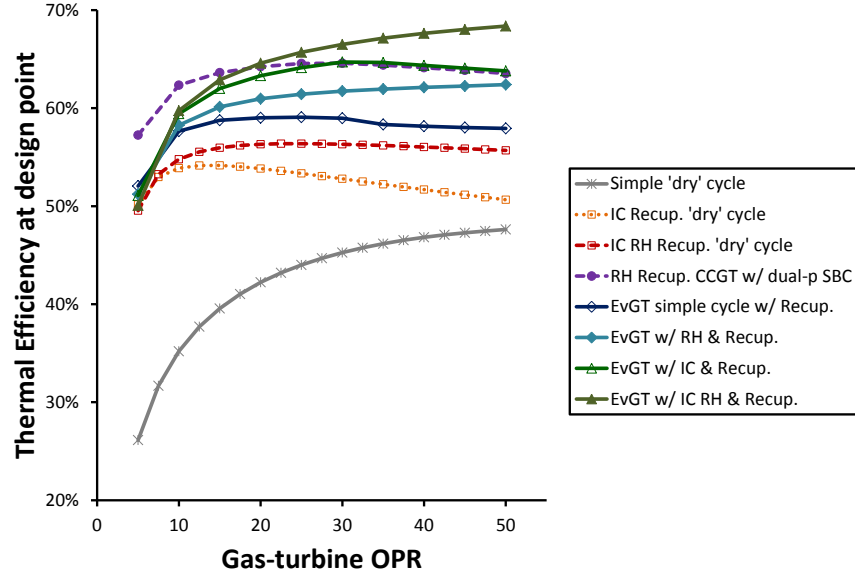


Figure 6.3: Comparison between the most promising among the CCGTs and the open 'dry' cycles, and the EvGTs analysed

6.3 Summary

A new model to perform quick evaluations of the performance of humidifiers at design point was developed and implemented within the integrated simulation platform developed in MATLAB®. The results from the analyses carried out, based on this model, on the thermal efficiency of the EvGT power plants are the most promising among all the cycles considered within this project.

Qualitatively speaking, EvGT power plants present some interesting features, compared against the rest of contenders studied within this research:

- Compared against the open 'dry' cycles...
 - ◊ EvGTs feature lower NO_x emissions, due to the presence of high levels of humidity in the main combustor of the gas turbine.
 - ◊ EvGTs feature much higher thermal efficiencies across the whole range of OPRs analysed.
- Compared against the CCGT cycles...
 - ◊ EvGTs feature lower NO_x emissions, due to the presence of high levels of humidity in the main combustor of the gas turbine.
 - ◊ EvGTs feature comparable, if not greater, thermal efficiencies than the CCGT power plants. In this regard, and even though off-design operation of the power plant has not been considered as a key feature to consider for this comparative assessment, Nakhamkin et al. [76] proved that their Cascaded Humidified Advanced Turbine (CHAT) power plant—an intercooled-reheated-recuperated EvGT cycle developed with current technology and readily available components—exhibits superior part-load performances than modern CCGTs, faster transients and start-ups (almost comparable to the characteristic times of simple-cycle gas turbines), and better behaviour when operating in varying ambient conditions.
 - ◊ Moreover, EvGTs extract all their useful work from the free power turbine at the expansion side of the gas-turbine *topping cycle only*, and deliver all this useful power directly in a single shaft, hence getting rid of the needs for additional weight and investments in steam turbomachinery and gearboxes.

- Compared against the “*plain*” STIG cycles...
 - ◊ EvGTs also benefit from reduced NO_x emissions, due to the high levels of humidity in the combustion chamber of the gas turbine.
 - ◊ Both power plants deliver their useful power in a single shaft, without the need for additional steam turbomachinery nor gearboxes.
 - ◊ EvGTs vastly outcompete the thermal efficiencies achievable by the STIG power plants across the whole range of OPRs analysed, due to their capability to recover much more of the low-temperature waste heat and produce higher levels of humidity in the expansion side of the gas turbine.
- Compared against the “*combined*”-STIG cycles...
 - ◊ EvGTs also benefit from reduced NO_x emissions, due to the high levels of humidity in the combustion chamber of the gas turbine.
 - ◊ EvGTs deliver all their useful power in a single shaft, without the need for additional steam turbomachinery nor gearboxes.
 - ◊ Still, EvGTs can outperform even the “*combined*”-STIG power plants, due to their capability to recover much more of the low-temperature waste heat and produce higher levels of humidity in the expansion side of the gas turbine.

Among the four different EvGT configurations studied herein, it is worthwhile noticing two:

The IC-R EvGT cycle, yielding thermal efficiencies slightly above the 64.5% mark, at OPRs around 35:1; essentially matching the performance levels of the RH-R gas turbine working in CCGT arrangement with dual-pressure SBC presented in Section 4.2, but with the advantage in terms of lifing and maintenance costs both for the free power turbine and the recuperator working at lower temperatures due to the lack of reheater.

The IC-RH-R EvGT cycle, yielding the highest thermal efficiency among all the different power plants considered, well above 68% at $\text{OPR} = 50:1$.

Chapter 7

Comparison & Downselection

After all the analyses for the whole range of *advanced* gas-turbine-based power plants have been carried out, the aim is to narrow down the scope of research to just a few —the most promising— candidates, to establish the preliminary design, sizing, physical characterisation and estimates on the mass and acquisition costs for each one of them; and to compare these figures against the conventional low-speed two-stroke diesel engines.

The main parameter to base the downselection upon will be, as previously stated, the thermal efficiency at design point. In this regard, low-speed two-stroke diesel engines feature thermal efficiencies as high as 53.5% nowadays (see, for instance, [4]) in simple-cycle configuration. However, their potential to further increase that thermal efficiency by means of implementing combined-cycle arrangements is significantly lower than that of the gas turbines, improving the overall thermal efficiency of the power plant by at most 1%–2% points, according to the literature [103, 69].

With this in mind, a threshold of 55% thermal efficiency is taken as the figure of merit to beat by any gas-turbine-based power plant aimed at replacing the diesels as prime movers for large cargo vessels. Table 7.1 summarises the maximum achievable thermal efficiency for the most promising cycles analysed, along with their corresponding *optimal* gas-turbine OPR that results in this maximum. These are only approximate values, though, considering that the results presented herein come from the analysis of a relatively coarse mesh covering the whole design space of each candidate; nevertheless, the amount of design points analysed (in the order of 10,000 to 20,000 for each cycle) ensure that the absolute optima will not be far away from these approximate values.

Taking a closer look at the values in tab. 7.1, the cycles that do meet the requirement of exceeding the 55% thermal efficiency at design point can be identified as potential candidates to outcompete the diesels in the marine environment. Namely, these cycles are:

From the open ‘dry’ cycles, only the IC-RH-R gas turbine features thermal efficiency in excess of the 55% threshold, and just by a narrow margin. Moreover, the open ‘dry’ cycles do not offer any other particular advantage over the rest of candidates, and so these will be disregarded for any further research.

From the CCGTs, many are the configurations that yield thermal efficiencies well in excess of 60%. The most interesting configuration analysed consists of a RH-R topping-cycle gas turbine featuring a dual-pressure (non-reheated) SBC, as discussed in Section 4.3. This level of performance makes out of the CCGTs a clear contender to be considered further.

From the “*plain*” STIGs, RH-R topping-cycle gas turbines have the potential to outcompete the diesels. However, when compared against other *advanced* gas-turbine-based power plants analysed, STIG cycles are vastly outperformed. The EvGTs, for instance, offer greater levels of thermal efficiency at design point, with at least the same potential to reduce NO_x emissions due to the humidity present in the main combustor, and all the useful power delivered in a single shaft as well (see discussion in Section 6.3), and thus the STIG cycles can be disregarded in favour of the EvGTs.

Table 7.1: Summary of maximum achievable thermal efficiencies (and optimal OPRs) for the most promising among the advanced gas-turbine-based power plants analysed

Cycle		Maximum η_{th}	Optimal OPR
Open ‘dry’ cycles	Simple-cycle gas turbine	47.9%	> 50:1
	Simple-recuperated gas turbine	50.1%	$\sim 7.5:1$
	RH-R gas turbine	51.0%	$\sim 10:1$
	IC-R gas turbine	54.2%	$\sim 15:1$
	IC-RH-R gas turbine	56.4%	$\sim 25:1$
CCGTs	Simple-cycle CCGT + dual-p. SBC	59.9%	$\sim 25:1$
	RH-R CCGT + dual-p. SBC	64.6%	$\sim 30:1$
“plain” STIGs	Simple-cycle STIG	54.6%	$\sim 40:1$
	RH-R cycle STIG	59.1%	> 50:1
“combined”-STIGs	Simple-cycle C-R-STIG	56.9%	$\sim 25:1$
	RH-R cycle C-R-STIG	61.8%	$\sim 45:1$
EvGTs	Simple-recuperated EvGT	59.1%	$\sim 25:1$
	RH-R EvGT	62.4%	> 50:1
	IC-R EvGT	64.7%	$\sim 35:1$
	IC-RH-R EvGT	68.4%	> 50:1

From the “combined”-STIGs, the C-R-STIG cycle is the most promising configuration analysed, especially when coupled with a RH-R topping-cycle gas turbine. Still, the thermal efficiencies achievable by means of this configuration do not quite match the performances of the CCGTs, in spite of resulting in an increased level of complexity for the SBC (see discussion in Section 5.3), and the reduced levels of NO_x emissions due to the presence of high levels of humidity in the main combustor can be achieved by means of an EvGT cycle instead, with the additional advantage for the latter of delivering all the useful power in a single shaft and avoiding the need for investing in any sort of steam turbomachinery components (see discussion in Section 6.3). Hence, the “combined”-STIG power plants will be disregarded, too.

From the EvGTs, both the IC-R and the IC-RH-R configurations are very interesting options. The IC-R EvGT can match the performance of the best CCGT analysed, with the additional advantage of reduced NO_x emissions and the useful power delivered in a single shaft (see discussion in Section 6.3), and without featuring the reheater within the gas-turbine topping cycle, which will result in cheaper maintenance and longer lifing of the free power turbine. The IC-RH-R EvGT, on the other hand, is the most efficient cycle analysed, even though the higher temperatures at the free power turbine inlet and outlet will not only incur in increased maintenance costs for the free power turbine, but also higher acquisition and maintenance costs for the recuperator, that will have to withstand much higher temperatures at every operating condition. For this reason, both the IC-R and the IC-RH-R EvGTs will be considered for further study.

The three downselected power plants will be, therefore: a) the RH-R CCGT with dual-pressure SBC, b) the IC-R EvGT, and c) the IC-RH-R EvGT; for which preliminary design, physical characterisation, and estimation of the overall size, mass and acquisition costs of their main components will be carried out in Part II of this research, in order to further evaluate the relative merits of each one of them.

The thermodynamic cycles of these three downselected power plants have been fully *optimised* (see Appendix C for the detailed results) and will be kept constant, as an input, for all subsequent studies.

7.1 Off-Design Performance Considerations

Even though the study of the off-design performances of the proposed candidates is out of the scope of this work and thermal efficiency at design point has been identified as the only key parameter to consider for this comparative assessment, cargo vessels do require off-design operation of their prime mover, for instance, to accommodate the ship's speed to the maritime traffic requirements, to navigate in rough weather, to be able to operate in different regions with great variations in ambient conditions (from tropical seas to the Arctic Ocean), or simply to manoeuvre at canals or ports.

Gas turbines, and in particular *simple-cycle* gas turbines, have traditionally been known for exhibiting very poor performances at part-load operation —much worse than the off-design behaviour of diesel engines—; however, this is not necessarily the case for other *advanced* gas-turbine-based power plants. The relatively flat thermal-efficiency-vs-OPR curves shown in fig. 6.3 (except at very low OPRs $< 15:1$) for the EvGTs, for instance, *suggest* already good off-design performances of this sort of power plants, as demonstrated also by Nakhamkin et al. [76] in their study of the CHAT cycle.

On a similar note, the very flat thermal-efficiency-vs-OPR curve shown in fig. 6.3 for the intercooled-recuperated 'dry' cycle *suggests* as well good part-load performances of this particular configuration, as it is indeed the case for the most well-known power plant implementing this sort of thermodynamic cycle: Rolls-Royce's WR-21, which was in fact designed primarily according to part-load operating requirements [28, 99].

For all the above, and despite the lack of a more consistent study of this matter, which falls out of the scope of this work, one can certainly say that there is some merit in the off-design characteristics of the most promising *advanced* gas-turbine-based power plants considered herein (especially those cycles featuring a recuperator, which becomes more and more efficient the lower the power setting of the power plant falls), and that these could definitely meet the operating requirements to make them suitable for the typical mission profiles of the large cargo ships.

Part II

Physical Definition of Downselected Power Plants

Chapter 8

Heat Exchangers Design

Heat exchangers are the key components in raising the overall thermal efficiency of any *advanced* gas-turbine-based power plant.

Essentially, a gas turbine *per se* is a thermodynamic cycle that takes heat from a high-temperature source (the fuel burnt), extracts some useful power, and then releases the remaining waste heat into a cold sink (the ambient). In doing so, the overall thermal efficiency of this thermodynamic cycle depends obviously on the efficiencies of the turbomachinery components (compressors and turbines) utilised, but is rather limited, nevertheless, by the compression & heat addition & expansion processes of the cycle itself.

To achieve higher degrees of thermal efficiency, the goal is, generally speaking, to recover somehow as much of the waste heat released into the ambient as possible, and driving the recovered heat back into the cycle for additional power extraction. In this regard, gas turbines are particularly well suited for implementing waste-heat recovery systems, given that the vast majority of the heat losses are contained in the exhaust of the gas turbine, in the form of high-temperature gases, and thus high-quality waste heat is concentrated almost exclusively in a single source, as opposed to the diesel engines where the lubricating oil and the coolant of the refrigeration system and the exhaust gases are all different sources of lower-quality waste heat.

The means for recovering waste heat from the exhaust gases of a gas turbine can vary substantially—as showcased by the different power plants analysed in Part I of this research—but, in essence, this is accomplished by implementing some sort of heat exchanger that will transfer heat (or, some times, heat and mass) back into the cycle.

Heat exchangers are bulky, heavy and expensive components, that are expected to represent a big portion of the total investment required in the acquisition, installation and operation of any *advanced* gas-turbine-based power plant. Therefore, a big emphasis is placed upon the preliminary design and sizing of the heat exchangers featured within the three power plants downselected in Chapter 7. Moreover, it has been discussed already the fundamental role of the heat exchangers in raising the performance levels of gas-turbine-based power plants, and so a high degree of correlation between the acquisition costs of these heat exchangers and the maximum achievable thermal efficiency of the power plant is expected to be found and studied in detail.

8.1 Heat Transfer Theory & Optimal Design

The mindset and methodology followed for the preliminary design of the heat exchangers is very similar to the one presented by Franco and Giannini [43, 44], whereby a hierarchical approach to the optimisation of the design of compact heat exchangers is established by means of: *a*) the optimisation of the thermodynamic cycle of the whole power plant; *b*) the optimisation of the operating parameters of each component implemented within the thermodynamic cycle; and *c*) the optimisation of the internal design of each component. The thermodynamics of the downselected cycles have already been established (see Appendix C for the details), and thus the remaining work to do in this case is just to optimise the internal geometry of the heat exchangers, *constrained* by the specified thermodynamics and operating parameters for each of them.

The overall heat transfer carried out by any heat exchanger can be written as per eq. 8.1:

$$Q = UA \cdot \Delta\Theta_m \quad (8.1)$$

where Q is the total amount of heat transferred, U the *overall* heat transfer coefficient, A the heat-transfer surface area, and $\Delta\Theta_m$ is the mean temperature difference between both streams across the length of the heat exchanger.

Given that the thermodynamics are pre-defined as an input to this optimisation study, the total heat transferred —i.e. the total “duty” of the heat exchanger— is known beforehand. The *overall* heat transfer coefficient depends on the internal geometry of the heat exchanger and the thermodynamic properties of both fluids; however, it is usually more convenient to treat the product UA as a whole, depending as well on the internal geometry and thermodynamic properties of both streams, for the purpose of heat exchanger design and sizing. The mean temperature difference, although easily calculated for *counter-flow* arrangements (see, for instance [98, 63], or [59, 102] for a more numerical approach to the topic), it is not defined that straightforwardly for *cross-flow* heat exchangers —all the heat exchangers featured within the downselected power plants will be of the *cross-flow* type—, and therefore the ϵ – NTU method was adopted instead.

The ϵ – NTU method tries to find a relation between the *effectiveness*, ϵ , of a heat exchanger, and the *number of transfer units*, NTU, required to achieve that effectiveness. The number of transfer units can be understood as the “number of times” that the mean *driving force*, $\Delta\Theta_m$ —some times also called the *driving potential*—, acts in the heat transfer process, and is a measure of the thermodynamic size of a heat exchanger.

Considering that the fluids can have different (mean) heat capacities, $C = m \cdot c_p$, the number of transfer units is defined usually as per eq. 8.2:

$$\text{NTU} = \max \left\{ \frac{\Delta T_c}{\Delta\Theta_m}, \frac{\Delta T_h}{\Delta\Theta_m} \right\} \quad (8.2)$$

where ΔT represents the temperature leap (in *absolute value* always) achieved, from the inlet to the outlet, on either side of the heat exchanger; and the sub-indexes c, h refer either to the cold or the hot streams, respectively.

Given that the heat transferred to/from one stream can be expressed as per eq. 8.3:

$$Q = C \cdot \Delta T = m \cdot c_p \cdot \Delta T \quad (8.3)$$

by equating eqs. 8.1 and 8.3, and substituting in eq. 8.2, the definition of the number of transfer units can also be written as per eq. 8.4:

$$\text{NTU} = \max \left\{ \frac{UA}{C_c}, \frac{UA}{C_h} \right\} \quad (8.4)$$

The effectiveness of a heat exchanger is defined, similarly, as per eq. 8.5:

$$\epsilon = \max \left\{ \frac{\Delta T_c}{\Theta_{\text{span}}}, \frac{\Delta T_h}{\Theta_{\text{span}}} \right\} \quad (8.5)$$

where Θ_{span} is the maximum temperature difference, between the cold and the hot stream inlets, $\Theta_{\text{span}} = T_{h,i} - T_{c,i}$.

With these definitions in mind, the relation between the *effectiveness* and the *number of transfer units* of a cross-flow heat exchanger can be derived (see, for instance, [59]), depending on whether the streams are assumed to be *mixed* —this is, with uniform temperature for any cross section— or *unmixed*.

In the case of *tube & fin* heat exchangers, for instance, the cold fluid (usually water) circulates *inside* the tubes, therefore being an *unmixed* stream —the temperature profiles inside the tubes will be, in general, different to one another—, while the hot fluid (usually gas) circulates ‘freely’ on the *outside* of the tubes, being therefore a *mixed* stream. Depending on whether the *mixed* or the *unmixed* stream has the greatest heat capacity of the two —usually referred to as C_{max} —, and defining the ratio of the lower to the greater heat capacities of both streams as $W = C_{\text{min}}/C_{\text{max}}$ —so

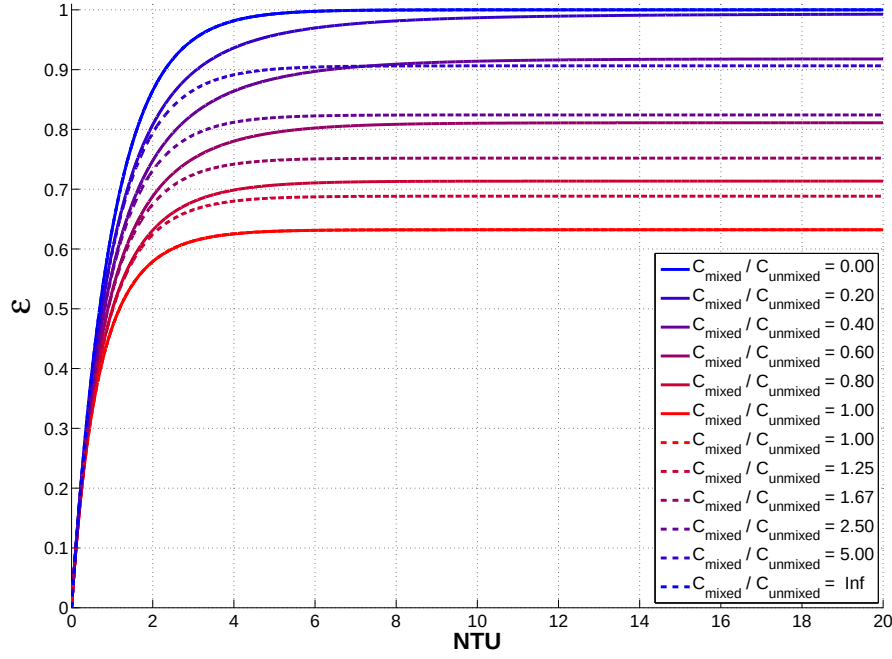


Figure 8.1: $\epsilon - NTU$ curves for the mixed-unmixed single-pass cross-flow heat exchanger

that $W \leq 1$ in all cases—, eq. 8.6 relates the number of transfer units, NTU, to the effectiveness, ϵ , of a *mixed-unmixed* single-pass heat exchanger [59].

$$NTU = \begin{cases} -\frac{1}{W} \ln(1 + W \cdot \ln(1 - \epsilon)) & \text{for } C_{\min} \text{ mixed, } C_{\max} \text{ unmixed} \\ -\ln(1 + \frac{1}{W} \ln(1 - W \cdot \epsilon)) & \text{for } C_{\max} \text{ mixed, } C_{\min} \text{ unmixed} \end{cases} \quad (8.6)$$

Equation 8.6 can also take the form of eq. 8.7 when $W \rightarrow 0$, irrelevantly of the mixed or unmixed conditions of each stream [59].

$$NTU = -\ln(1 - \epsilon) \quad (8.7)$$

Figure 8.1 plots the curves that result from eqs. 8.6–8.7 for different W , both for the case in which $C_{\min}$ is mixed and $C_{\max}$ unmixed and *vice versa*. The curves in this figure highlight the main limitation of the *single-pass* heat exchangers in *cross-flow* arrangement: there is an asymptotic value for the effectiveness, ϵ , when $NTU \rightarrow \infty$, that is the maximum achievable effectiveness for this configuration and depends only on $C_{\text{mixed}}/C_{\text{unmixed}}$ —i.e. depends on W and on whether it is the case of $C_{\min}$ mixed and $C_{\max}$ unmixed or the other way around—. To overcome this limitation, and be able to achieve the high levels of effectiveness required by the thermodynamically optimised power plants from Appendix C, *multi-pass* heat exchangers are necessary.

In a *multi-pass* arrangement, the same “basic” single-pass heat exchanger is replicated several times *in series*, with the hot and the cold streams entering the assembly on opposite edges of this series, resulting in what is called an *overall counter-flow* configuration: essentially, both fluids circulate through a number of passes in counter-flow arrangement, being each of the passes a cross-flow heat

exchanger. By means of this configuration, the limited effectiveness achievable with a single pass, ϵ_{pass} , can be raised as per eq. 8.8 for an increased number of passes, N_{pass} (see [59], for instance, for the derivation of this formula, in which it has been assumed that both the *mixed* and the *unmixed* streams are mixed between passes), getting closer and closer to the counter-flow heat exchanger effectiveness when $N_{\text{pass}} \rightarrow \infty$ [59].

$$\epsilon = \frac{\left(\frac{1 - W \cdot \epsilon_{\text{pass}}}{1 - \epsilon_{\text{pass}}} \right)^{N_{\text{pass}}} - 1}{\left(\frac{1 - W \cdot \epsilon_{\text{pass}}}{1 - \epsilon_{\text{pass}}} \right)^{N_{\text{pass}}} - W} \quad (8.8)$$

Equation 8.8 can also take the form of eq. 8.9 for the limiting case of $W = 1$ [59].

$$\epsilon = \frac{N_{\text{pass}} \cdot \epsilon_{\text{pass}}}{1 + (N_{\text{pass}} - 1) \epsilon_{\text{pass}}} \quad (8.9)$$

Given these equations and the aforementioned definitions, if the thermodynamics of a particular heat exchanger are known, eq. 8.5 will allow for the calculation of the overall effectiveness of the heat exchanger, ϵ , and, if a multi-pass heat exchanger—with a determined number of passes, N_{pass} —in *overall counter-flow* arrangement is considered, eqs. 8.8 or 8.9 will yield the target effectiveness of each one of the cross-flow passes, ϵ_{pass} , forming the multi-pass arrangement. Equations 8.6 or 8.7 can be used then to calculate the number of transfer units of a single pass, and eq. 8.4 will determine the heat transfer *conductance*, UA , that a single pass must feature in order for the whole heat exchanger to fulfil its duty.

Once this is done, all that remains is to optimise the internal geometry—and thus the mean velocities of both streams—of a *single pass* of the heat exchanger, so that the specified UA can be achieved according to the defined optimising criteria. Some times, a simplistic approach can be taken to this task, by just assuming a ‘typical’ value for the *overall* heat transfer coefficient, U , of a particular type of heat exchanger (see, for instance, values provided by Jonsson and Yan [58] for several smooth-tubes or finned-tubes heat exchangers, based on [53]) and simply working out the required total area of the primary heat transfer surface, A . In this work, however, it has been decided to fully analyse the detailed *internal* geometry of the different heat exchangers and the heat transfer characteristics resulting from this geometry: eq. D.1 provides the means for doing so, and the full detailed explanation on the heat transfer calculations for the different heat exchanger types is provided in Appendix D.

8.1.1 Tube & Fin Heat Exchangers

All gas–liquid (or gas–*phase change* or *phase change*–liquid) heat exchangers featured within the downselected cycles have been designed as *tube & fin* heat exchangers. This includes all the ICs, ACs and ECs of the EvGT power plants, as well as the HRSG and the steam condenser within the SBC of the CCPPs.

The basic geometry for a typical *tube & fin* heat exchanger is depicted in fig. 8.2, taken from [98], in which (cold) liquid water circulates through an array of parallel tubes, with individual annular fins placed on the exterior, while the (hot) gas is streamed in perpendicular to these tubes, in *cross-flow* arrangement. The suitability of this sort of heat exchangers for the considered applications (mainly gas–liquid heat exchangers, as previously stated) is due to their capability to withstand extremely high operating pressures—well in excess of 200 bar—and virtually unlimited temperatures—other than the corresponding metal allowable limits—, while still resulting in great compactness due to the extended heat-transfer surfaces [98, 63, 92].

The fact that the liquid water circulates on the inside of plain tubes helps in avoiding *fouling* issues, hence resulting in cheaper maintenance and ease of operation. At the same time, the extended surfaces, on the (hot) gas side only, increase the heat transfer coefficient there (usually much lower than on the water side), equalising both and minimising therefore the total length of the whole heat exchanger [98, 63].

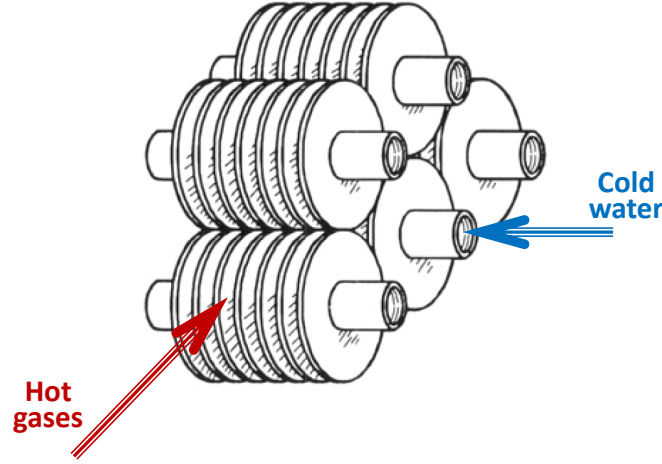


Figure 8.2: Basic geometry for the tube & fin type heat exchanger

Optimisation of the internal geometry of tube & fin heat exchangers

The internal geometry of a single pass of a tube & fin heat exchanger is shown in fig. 8.3, where the main dimensions defining the heat transfer surface and free flow areas of both streams within this single pass are depicted as well. Namely:

- L_{tubes} —or, simply, L_1 , for this is one of the three *external* dimensions that will define the overall size of the heat exchanger—, the length travelled by the water *inside* the tubes.
- d_i , the *inner* diameter of the tubes.
- d_o , the *outer* diameter of the tubes, and hence the thickness of the primary surface wall, δ_w , can be defined as $\delta_w = 1/2 (d_o - d_i)$.
- d_e , the diameter of the *extended* surfaces, with which the height of the fins, h'_f , can be expressed as $h'_f = 1/2 (d_e - d_o)$.
- X_l , the pitch between adjacent rows within the same pass of the heat exchanger, that along with the number of rows per pass, N_{rows} , define the *total length* of the single pass, L_{pass} —or, simply, L_2 , for this is another one of the three *external* dimensions that will define the overall size of a single pass of the heat exchanger—, as per $L_{\text{pass}} = N_{\text{rows}} \cdot X_l$.
- With this and the number of passes, N_{pass} , the *overall* length of the multi-pass heat exchanger —this is, the total distance travelled by the gas on the *outside* of the tubes along the whole heat exchanger— will be given by $L_{\text{overall}} = N_{\text{pass}} \cdot L_{\text{pass}}$.
- X_t , the pitch between adjacent tubes in the same row, that along with the number of tubes per row, N_{tubes} , define the *total width* of the heat exchanger, L_{width} —or, simply, L_3 , for this is the last of the three *external* dimensions that will define the overall size of the heat exchanger—, as per $L_{\text{width}} = N_{\text{tubes}} \cdot X_t$.
- N_f , the total number of fins per tube, which is defined, based on the total length of the tubes, L_{tubes} , and the pitch between adjacent fins, p'_f , as per $N_f = L_{\text{tubes}}/p'_f$.
- δ_f , the thickness of the fins.

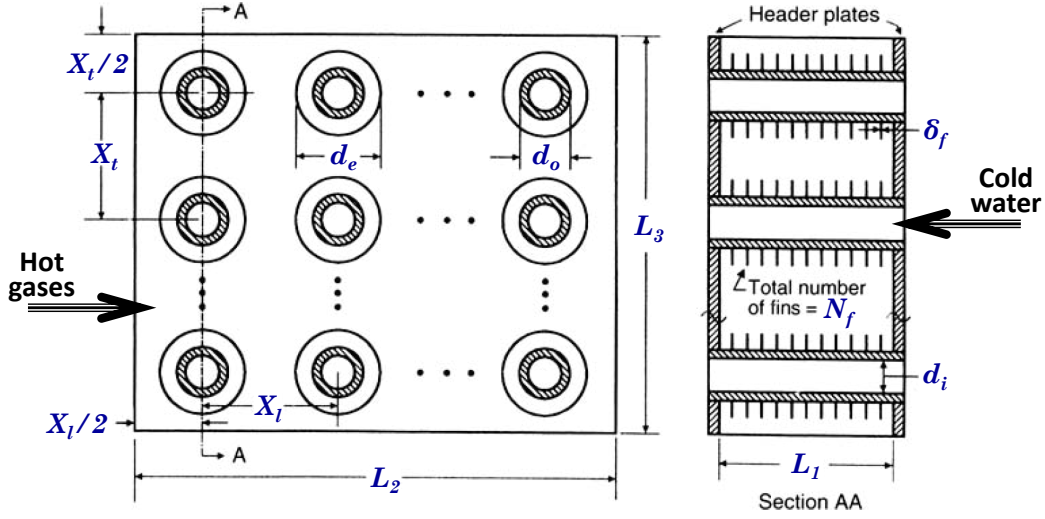


Figure 8.3: Internal dimensions definition for the tube & fin type heat exchanger

Essentially, the thermodynamics of the heat exchanger —that are known and imposed to the optimisation process— define two additional *constraints* that the final geometry of the single pass must satisfy, hence not all the listed dimensions are decision variables *per se*. Instead, an iterative process where just the right number of independent variables are specified beforehand, and the rest are calculated as a function of these independent variables plus the *thermodynamic constraints*, must be implemented. In practice, to achieve reliable convergence of this iterative process, some variables will work better as the calculated variables and some others as the independent ones. In this case, the choice of L_{tubes} and h'_f as the *calculated* variables proved to be the best. All the details on the heat transfer calculations leading to the definition of suitable heat exchanger geometries can be found in Appendix D.1.

8.1.2 Plate & Fin Heat Exchangers

The gas-gas heat exchangers featured within the downselected cycles —this is, the recuperators implemented within the different power plants— have been designed as *plate & fin* heat exchangers.

The basic geometry for a typical *plate & fin* heat exchanger is shown in fig. 8.4, taken from [98], in which the main internal dimensions defining the heat transfer surface and free flow areas of both streams within a single pass of the heat exchanger are depicted as well. Offset strip fins have been selected for this study, due to their improved efficiencies for equivalent levels of pressure drop across the heat exchanger when compared against other traditional fin geometries [98].

Plate & fin heat exchangers usually result in more compact geometries than tube & fin configurations; however, their generally lower limitations for the operating pressures, along with *fouling* issues derived from having fins in both sides of the heat exchanger, make out of them rather impractical designs for heat transfer processes where fluids in liquid phase are involved [98, 63, 92].

Optimisation of the internal geometry of *plate & fin* heat exchangers

Following the schematic representation of a single pass of a plate & fin heat exchanger in cross-flow arrangement shown in fig. 8.4, the main dimensions to consider for the design of this type of heat exchangers are:

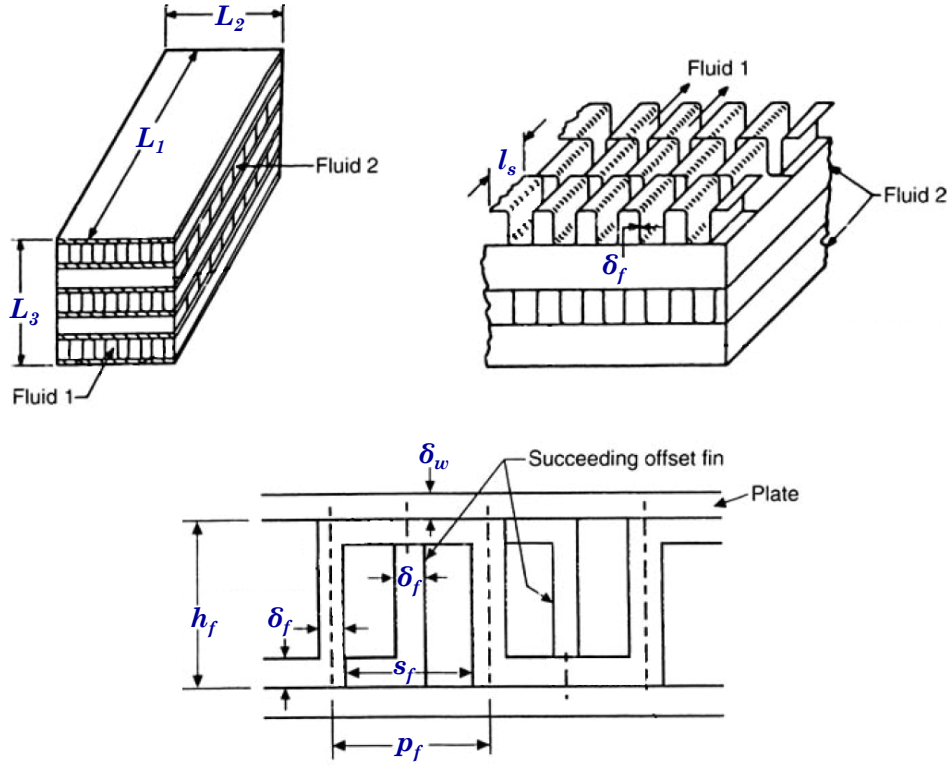


Figure 8.4: Basic geometry and internal dimensions definition for the plate & fin type heat exchanger

- $L_{\text{flow},h}$ and $L_{\text{flow},c}$ —or, simply, L_1 and L_2 , respectively—, the lengths travelled by each of the fluids within a single pass of the heat exchanger; the sub-indexes h, c refer to the hot and the cold side of the heat exchanger.
- $h'_{f,h}$ and $h'_{f,c}$, the heights of the fins —and hence the heights of the *layers*— on either side.
- $p'_{f,h}$ and $p'_{f,c}$, the pitch between adjacent fins on either side, that along with the flow lengths of the heat exchanger, $L_{\text{flow},h}$ and $L_{\text{flow},c}$, define the number of fins, $N_{f,h}$ and $N_{f,c}$, as per $N_{f,h} = L_{\text{flow},c}/p'_{f,h}$ for the hot side (and analogously for the cold side).
- δ_f , the thickness of the fins, that will be assumed the same in both sides.
- With these, the *free flow* separation between two adjacent fins can be defined as $s'_{f,h} = p'_{f,h} - \delta_f$ for the hot side (and analogously for the cold side).
- δ_w , the thickness of the plates forming the primary surface walls.
- $l_{s,h}$ and $l_{s,c}$, the length of the offset strips on either side, that along with the flow lengths of the heat exchanger, $L_{\text{flow},h}$ and $L_{\text{flow},c}$, define the number of offset strips, $N_{s,h}$ and $N_{s,c}$, as per $N_{s,h} = L_{\text{flow},h}/l_{s,h}$ for the hot side (and analogously for the cold side).
- N_{layer} , the number of layers of the *cold* side within a single pass of the heat exchanger. Given the lower density of the gases in the hot side, it will always be assumed that the number of layers of the *hot* side within a single pass will be equal to $N_{\text{layer}} + 1$, and thus these will be *embracing* one fewer layers of the cold fluid so that both the top-most and the bottom-most layers of each pass will contain hot gases.

- With this assumption, the *height* of a single pass of the heat exchanger, L_{pass} —or, simply, L_3 —, can be defined as $L_{\text{pass}} = N_{\text{layer}} \cdot h'_{f,c} + (N_{\text{layer}} + 1) \cdot h'_{f,h} + 2 (N_{\text{layer}} + 1) \cdot \delta_w$ and, subsequently, the *overall* height of the multi-pass heat exchanger as per $L_{\text{overall}} = N_{\text{pass}} \cdot L_{\text{pass}}$.

Essentially, the thermodynamics of the heat exchanger —that are known and imposed to the optimisation process— define two additional *constraints* that the final geometry of the single pass must satisfy, hence not all the listed dimensions are decision variables *per se*. Instead, an iterative process where just the right number of independent variables are specified beforehand, and the rest are calculated as a function of these independent variables plus the *thermodynamic constraints* must be implemented. In practice, to achieve reliable convergence of this iterative process, some variables will work better as the calculated variables and some others as the independent ones. In this case, the choice of $L_{\text{flow},c}$ and $h'_{f,h}$ as the *calculated* variables proved to be the best. All the details on the heat transfer calculations leading to the definition of suitable heat exchanger geometries can be found in Appendix D.2.

8.2 Pressure Losses

Pressure losses, Δp , in one side of a heat exchanger, can be calculated by means of eq. 8.10 (see [98, eq. 6.28, p. 388 and before] for the derivation), where the four terms that represent each one of the major components contributing to the total *core* pressure drop are separated.

$$\Delta p = \frac{G^2}{2} \left(\underbrace{\left(1 - \sigma^2 + K_c\right) \frac{1}{\rho_i}}_{\text{entrance effect}} + \underbrace{\frac{4 f L}{D_h} \left(\frac{1}{\rho}\right)_m}_{\text{core friction}} + \underbrace{2 \left(\frac{1}{\rho_o} - \frac{1}{\rho_i}\right)}_{\text{momentum effect}} - \underbrace{\left(1 - \sigma^2 - K_e\right) \frac{1}{\rho_o}}_{\text{exit effect}} \right) \quad (8.10)$$

Here, the entrance effect is usually a pressure drop, while the exit effect is usually a pressure raise, hence these two terms tend to balance out each other. Moreover, both terms are usually small compared to the *core friction*, and thus both can usually be neglected [98]. In many cases, however, their influence is not directly calculated, but rather accounted for by means of the Fanning friction factor, f , experimentally determined and usually available for various geometries in the form of empiric correlations.

The *core friction* term is usually the greatest source of pressure losses for most heat exchangers, accounting typically for more than 90% of the total pressure drop [98], and hence the influence of the momentum effect has been neglected as well for this preliminary design study.

With these assumptions, the pressure losses in one side of a heat exchanger can be approximated by the reduced formula given in eq. 8.11, where the idealisation $(1/\rho)_m \approx 1/\rho_m$ has been made [98].

$$\Delta p \approx \frac{4 f L G^2}{2 D_h \rho_m} \quad (8.11)$$

8.3 Mass & Costs Assessment

The work by Carlson and Conboy [21] summarises the basic guidelines for material selection and mechanical design of heat exchangers of various types, following the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (BPVC) Sections II —*Materials*— and VIII —*Rules for Construction of Pressure Vessels*—. Following these rules, the required thickness of the different walls, tubes and plates, δ_w , of both the *plate & fin* as well as the *tube & fin* geometries, can be calculated, based upon the material selected, the operating temperature of the metal alloy and the stress levels that it must withstand. With these rules, the necessary thickness of the shell containing the heat exchanger can also be assessed and, therefore, the mass estimation of the whole heat exchanger becomes a matter of simply calculating the total volume of material employed and retrieving a typical value for the density of the material chosen.

The purchased equipment costs both for a *tube & frame* and for a *tube & fin* heat exchanger, *per unit area* of primary heat transfer surface, A , can be taken from the work presented by Jonsson

and Yan [58], which correspond to US\$110/m² and US\$350/m², respectively, expressed in 2002 US\$. These costs have been transformed to their equivalent 2010 US\$ by means of the *Chemical Engineering Plant Cost Indexes* (CEPCIs) —see, for instance, [81, Table 6-2, p. 238] for a list of CEPCI and other similar indexes prior to the year 2002, and [106] for the 2010 CEPCI value—.

Similarly, the cost of a *plate & frame* heat exchanger *per unit area* of primary heat transfer surface can be taken from [106], as US\$210/m², valid for the year 2010 already. To estimate the purchased equipment cost of a *plate & fin* heat exchanger, in US\$ for the year 2010, the above is considered, multiplied by the same factor of ~ 3.182 that relates the cost of the smooth- and the finned-tubes in *tube & frame* heat exchangers according to Jonsson and Yan [58].

All these are summarised in tab. 8.1, for clearer reference.

Table 8.1: *Estimated purchased equipment cost, per unit area of heat transfer primary surface, for several heat exchangers*

Heat Exchanger Type	CEPCI 2002	Cost 2002 (US\$/m ²)	CEPCI 2010	Cost 2010 (US\$/m ²)
Tube & Frame		110.00		150.15
Finned-Tubes	390.4	350.00	532.9	477.75
Plate & Frame		—		210.00
Finned-Plates		—		668.18

Except where otherwise noted, stainless steel is chosen as the typical material for the construction of all the heat exchangers considered herein, to avoid potential *corrosion* issues due to the presence of oxygen dissolved in the water streams, especially within the humidification system of the EvGTs [58]. However, in the few cases in which Inconel alloys are necessary to withstand the elevated operating temperatures, a factor of 1.7 is applied to all purchased equipment costs, following the recommendations by [106]. In spite of Inconel alloys being usually around 10 times more expensive than plain stainless steel (up to and above \$50/kg, as opposed to the typical \$5/kg of the stainless steel nowadays) when considering raw materials, the factor of 1.7 accounts for the *non-scalability* of certain aspects of the manufacturing process affecting the final price of the product —such as the labour costs or the seller’s profits, for instance— that do not scale directly with the metal price [106].

8.4 Optimisation Results & Discussion

The main results regarding the overall size, mass and acquisition costs of the *optimal* geometries found for the main heat exchangers implemented within the downselected power plants will be discussed herein. Details on the optimisation *objectives* and the available trade-offs, as well as some initial thoughts regarding certain design alternatives, not considered initially within the methodology described in previous sections, will be provided.

8.4.1 Intercoolers, Aftercoolers and “Exhaust-coolers”

These will be, as previously stated, of the *tube & fin* type heat exchangers. The aim is to find the most *compact* geometry —following the methodology outlined in Section 8.1 and Appendix D— that would fulfil the required duty —determined by the *optimised* thermodynamic cycles presented in Appendix C— for the heat exchanger. In this regard, *maximum compactness* has been defined as the *minimum “perimeter”*, or *minimal* sum of the three external dimensions of the heat exchanger, $L_1 + L_2 + L_3$ (note that, in this case, L_2 does not stand for the length of a single pass, L_{pass} , but for the *overall length* of the multi-pass heat exchanger instead: $L_2 = L_{\text{overall}} = N_{\text{pass}} \cdot L_{\text{pass}}$).

One of the major design parameters of this type of heat exchangers is the power requirements for the pumps that make the water circulate through the cold side (tubes *inside*) of the heat exchanger itself. This is one of the major sources of overall performance deterioration for the power plant [98, 63], and the amount of power required by the circulating pumps can indeed be non-negligible in some,

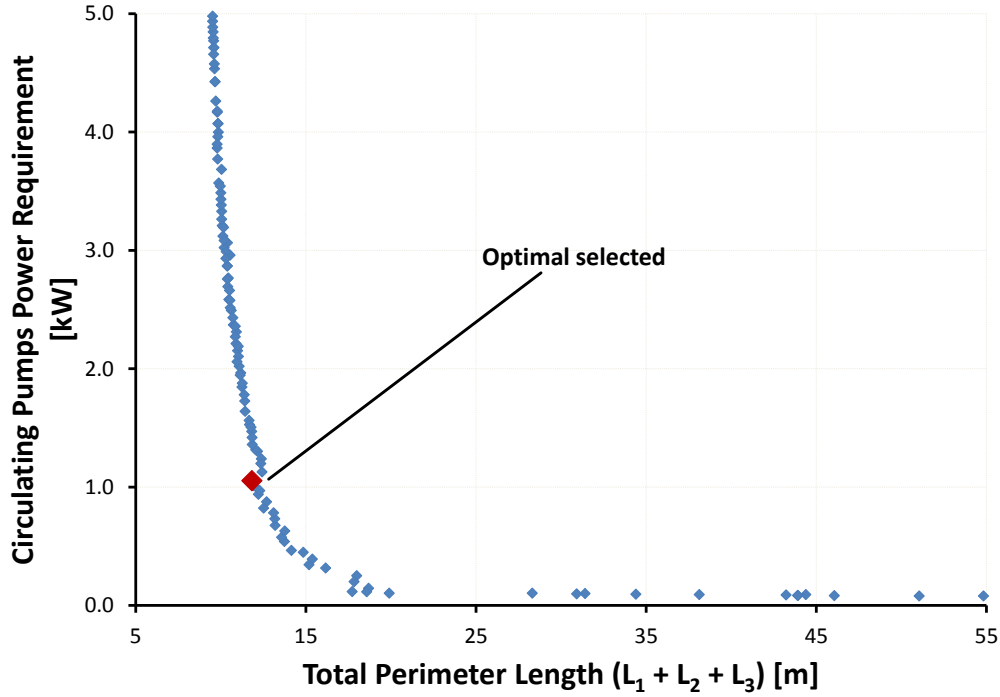


Figure 8.5: Pareto-front for the multi-objective optimisation of the “exhaust-cooler” geometry for the 40 MW IC-R EvGT power plant

very particular cases (see, for instance, Dostal [35] and the design of supercritical- CO_2 cycles for nuclear applications with *constrained* heat exchangers volume).

For this reason, a study regarding the trade-off between *compactness* and *losses* —i.e. between *minimising* the “perimeter”, $L_1 + L_2 + L_3$, and *minimising* the circulating pumps power requirement— was conducted at this point. Multi-objective optimisation (MOO) of the heat exchanger geometry was carried out, therefore, by means of the implementation of a genetic algorithm (GA) with the two aforementioned *independent objective functions*. The list of *decision variables* for the GA optimiser, as well as the convergence process to produce suitable geometries based on these *decision variables*, is described in detail in Appendix D.1. The required duty for the heat exchanger is calculated according to the ϵ –NTU method discussed in Section 8.1, taking the *optimised* thermodynamics presented in Appendix C as the only input. Once a consistent geometry has been produced, pressure losses, Δp , can be calculated as per Section 8.2 (eq. 8.11). The circulating pumps power requirement, P_{pumps} , is given then by eq. 8.12:

$$P_{\text{pumps}} = \frac{m \Delta p}{\eta_{\text{pumps}} \rho m} \quad (8.12)$$

Finally, mass and acquisition costs of a particular heat exchanger geometry can be estimated according to the assumptions outlined in Section 8.3, employing stainless steel for the construction of all the ICs, ACs and ECs featured within the downselected EvGT power plants.

Figure 8.5 shows an example of the outcome of such an optimisation, where the *pareto-front* that showcases the existing trade-off between *maximum compactness* and *minimal losses* is plotted. By definition, all points within the pareto-front are, in principle, equally suitable candidates for the design of this particular heat exchanger, and thus an additional criterion has to be defined if one point is to be selected as the *best* among the pareto-optima provided.

In this regard, it is interesting to look at the points with greater inner diameter of the tubes, d_i , that will result in simpler manufacturing of the whole heat exchanger due to the reduced total number of tubes. Such a point is marked in fig. 8.5 and, as it can be seen, falls slightly biased towards

the region of *maximum compactness*, sacrificing a little bit of the performances, for in this particular case the pareto-optima grow very rapidly in size for slight reductions in the pressure losses below the selected optimum. Table 8.2 summarises the detailed geometry of this configuration.

Table 8.2: Detailed geometry of the optimal “exhaust-cooler” selected for the 40 MW IC-R EvGT power plant

		IC-R EvGT <i>Optimal</i> EC
N_{pass}	(-)	9
N_{rows}	(-)	30
N_{tubes}	(-)	163
$L_1 = L_{\text{tubes}}$	(m)	2.15
$L_2 = L_{\text{overall}} = N_{\text{pass}} \cdot L_{\text{pass}}$	(m)	6.03
$L_3 = L_{\text{width}}$	(m)	3.64
“perimeter” $L_1 + L_2 + L_3$	(m)	11.82
$u_{m,h}$	(m/s)	39.5
$u_{m,c}$	(m/s)	1.3
Δp_c	(kPa)	0.56
P_{pumps}	(kW)	1.05

Similar analyses and optimisations have been carried out for the same downselected power plants, working with the same optimised thermodynamic cycles presented in Appendix C, but *re-scaled* to deliver increased overall power outputs in the range of 40 MW to 120 MW. Given the assumptions made so far, *re-scaling* a power plant to deliver increased power outputs can be effectively achieved by means of simply *scaling* linearly the mass flows circulating through the gas turbine and through any bottoming cycle implemented, from the 40 MW base-line cases discussed in Appendix C. With these analyses, the *scaling laws* for the construction of heat exchangers aimed at bigger power plants can be derived.

As long as the *compactness* is defined in the same way throughout these studies —i.e. understanding *maximum compactness* as *minimum overall “perimeter”*, $L_1 + L_2 + L_3$, of the heat exchanger—, the results presented in fig. 8.6 indicate that this is achieved when all the main dimensions of the heat exchanger are scaled with $\sqrt{m}$, while the fluid velocities inside the heat exchanger are kept the same. Consequently, the mass (fig. 8.6a) scales up with a power-law index slightly lower than $3/2$, due to the *non-scalability* of certain *internal* dimensions of the heat exchanger, such as the fins thickness, δ_f , for instance; and the acquisition costs (fig. 8.6b) of this component increase linearly, as does the primary heat transfer surface area, A .

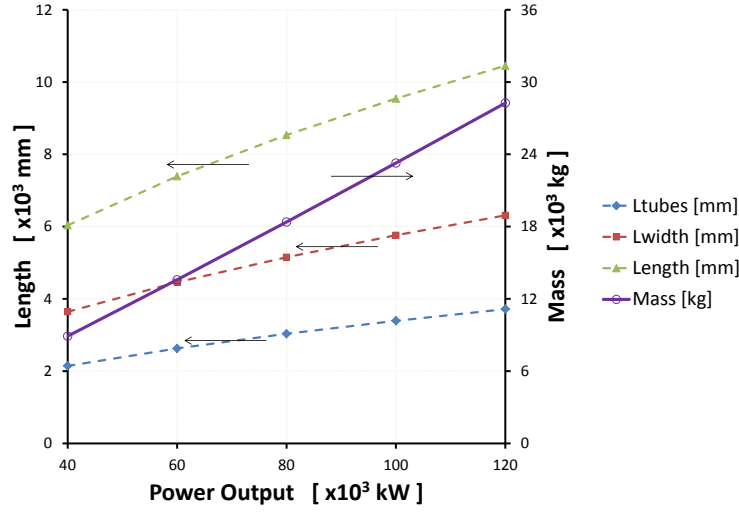
Similar trends are found for the ICs and the ACs of the IC-R EvGT cycle; as well as for the ICs, the ACs and the ECs of the IC-RH-R EvGT power plant.

8.4.2 The Steam Condenser

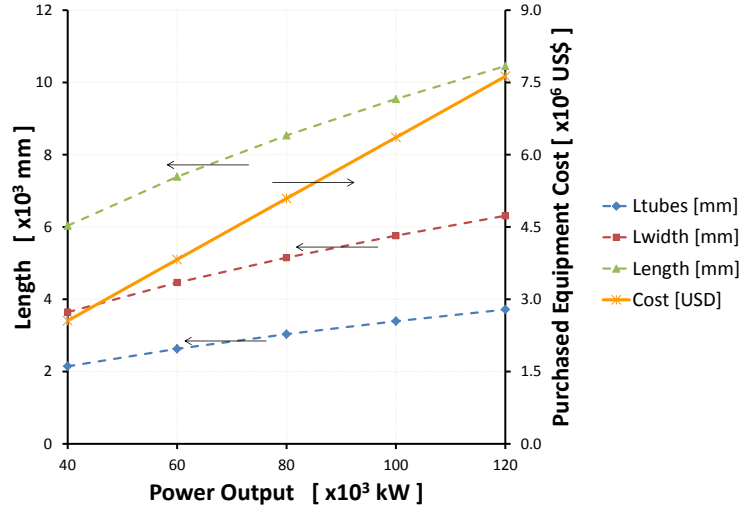
This is, analogously, a *tube & fin* type heat exchanger, manufactured as well in stainless steel. Thus the same conclusions can be withdrawn from the geometry optimisation of the steam condenser for the CCGT power plants. Figure 8.7 shows the pareto-front for the *optimised* design of the steam condenser in a 40 MW CCGT, where the *selected optimal* finned-tubes geometry has been marked. Figure 8.8 plots the derived *scaling laws* for the size and acquisition costs of such component, which follow identical trends to the EC for the EvGTs previously discussed in Section 8.4.1.

Alternative Technologies

In addition to the above, the steam condenser is a good example to showcase the flexibility of the heat exchanger design, and to provide some insight on the availability of other heat-exchanger configurations, with different pros and cons.



(a) Size and mass against overall power output



(b) Size and acquisition costs against overall power output

Figure 8.6: Size, mass and acquisition costs scaling laws for the optimal “exhaust-cooler” of the IC-R EvGT power plant

By looking in detail into the pareto-optima plotted in fig. 8.7, a special region of the pareto-front can be identified, in which the *optimal* geometries produced are indeed of the *smooth-tubes* type. This region of the *optimal* design space falls far away from the solutions with *maximum compactness*, as it is expected for a *less compact* heat exchanger arrangement such as the plain tube & frame configuration.

However, this *alternative technology* presents indeed other advantages: for starters, it exhibits very low pressure losses across the heat exchanger, even slightly lower than the *selected optimal geometry* for the finned-tubes configuration (see fig. 8.7); on the other hand, and most importantly, it is a much cheaper technology (see relevant discussion in Section 8.3 and tab. 8.1). This is much more apparent in fig. 8.8, where it can be seen that the smooth-tubes solution accomplishes the same thermodynamic duty that the finned-tubes configuration, but at less than a quarter of the price. More detailed comparison between the two available geometries for the 40 MW CCGT can be found in tab. 8.3.

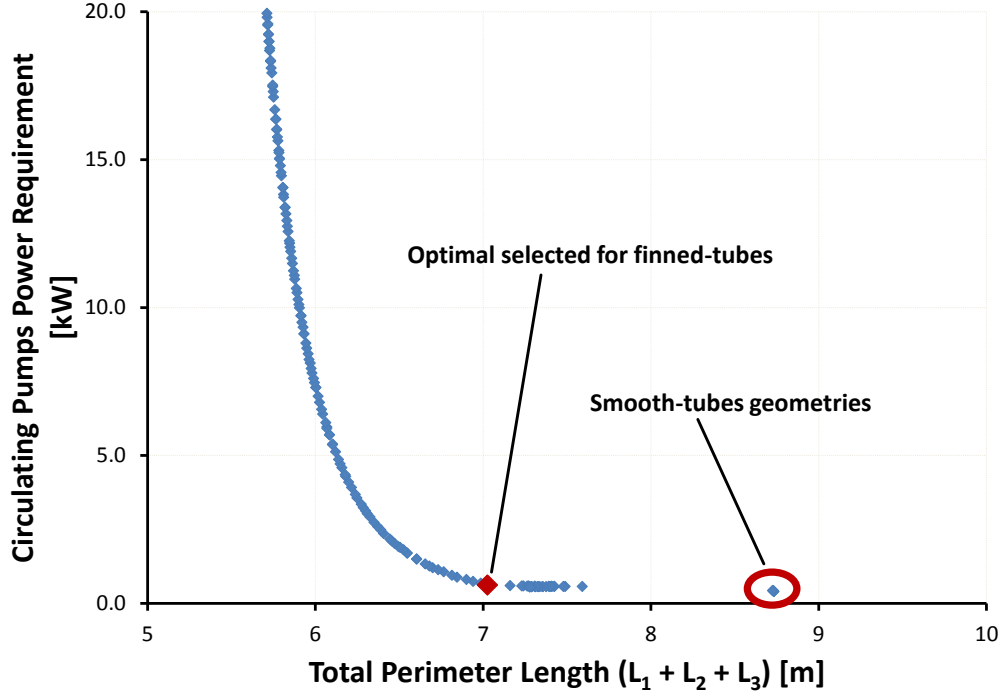


Figure 8.7: Pareto-front for the multi-objective optimisation of the steam condenser geometry for the 40 MW CCGT power plant

Table 8.3: Detailed geometry of the optimal condenser selected for the 40 MW CCGT power plant, considering either smooth- or finned-tubes configuration

		CCGT <i>Optimal</i> Condenser	
		Finned tubes	Smooth tubes
N_{pass}	(-)	1	1
N_{rows}	(-)	76	16
N_{tubes}	(-)	128	151
$L_1 = L_{\text{tubes}}$	(m)	2.65	4.11
$L_2 = L_{\text{overall}} = N_{\text{pass}} \cdot L_{\text{pass}}$	(m)	1.63	0.44
$L_3 = L_{\text{width}}$	(m)	2.75	4.18
“perimeter” $L_1 + L_2 + L_3$	(m)	7.03	8.73
d_i	(mm)	11.1	22.3
h'_f	(mm)	3.6	0.0
Δp_c	(kPa)	1.12	0.72
P_{pumps}	(kW)	0.62	0.40

In summary, plain *tube & frame* heat exchangers are less compact solutions, but can result in much cheaper designs provided that the targeted vessel has the necessary available space to accommodate these bulkier components within the deployed power plant.

8.4.3 The Heat Recovery Steam Generator

This is, again, a *tube & fin* type heat exchanger. The only particularity to take into account is that a single HRSG is formed by *six* heat-exchanger sections, manufactured all of them in stainless

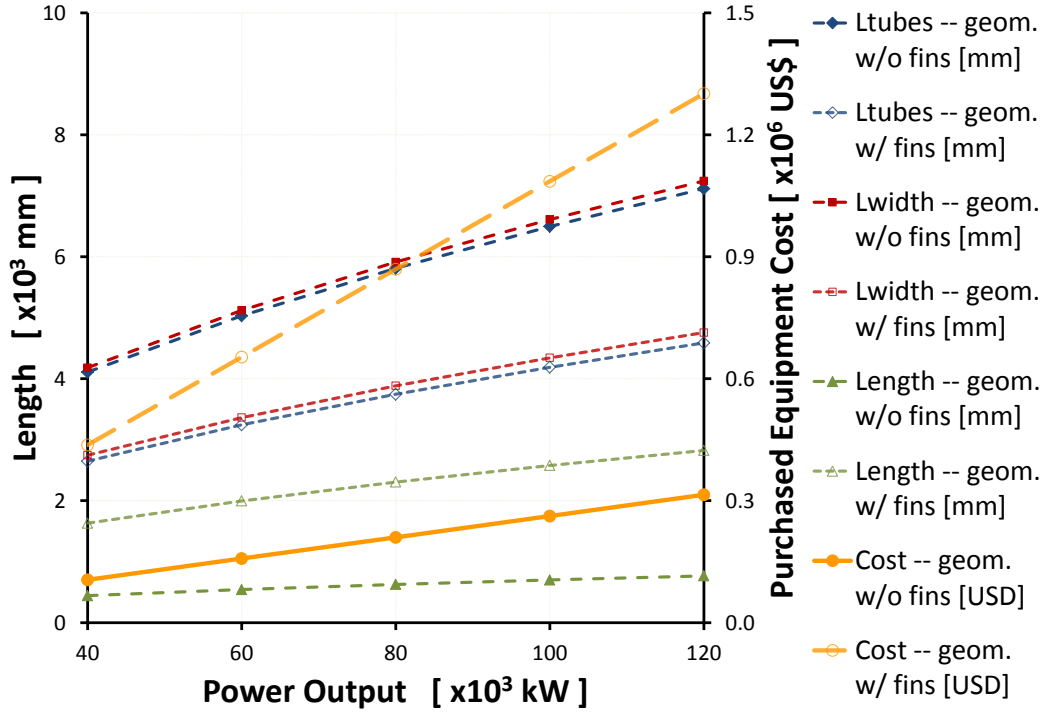


Figure 8.8: Size and acquisition costs scaling laws for the optimal steam condenser of the CCGT power plant, considering either smooth- or finned-tubes configuration

steel. Namely: (i) the LP economiser, (ii) the LP evaporator, (iii) the LP superheater, (iv) the HP economiser, (v) the HP evaporator, and (vi) the HP superheater. For this reason, the *external* dimensions of the HRSG consist of the most *constraining* —i.e. the greatest— length of the tubes, $\max\{L_{\text{tubes},i}\}$, and width, $\max\{L_{\text{width},i}\}$, among the different sections of the HRSG to determine L_1 and L_3 , respectively; and the *total sum of the overall lengths* of each section, $\sum(L_{\text{overall},i})$, to determine L_2 .

In spite of this, similar conclusions can be withdrawn from the geometry optimisation of the HRSG for the CCGT power plants. Figure 8.9 plots the derived *scaling laws* for the size and mass of such component, which follow identical trends to the EC for the EvGTs previously discussed in Section 8.4.1.

Scaling Alternatives

The HRSG is by far the bulkiest and heaviest heat exchanger implemented within the CCGT cycles. For this reason, an attempt was made to explore different strategies for the scaling of this component, to identify potential benefits of different arrangements, even at the expense of sacrificing a little bit in *compactness* as it has been defined.

In this regard, what has been called *linear-scaling* —as opposed to the $\sqrt{m}$ -scaling discussed so far, that results in masses increasing with a power-law index of almost $3/2$ — appears to be an interesting option. Rather than scaling the whole heat exchanger up congruently, this *scaling alternative* consists in taking a “base-brick” designed for lower mass flows and merely replicating this “brick”, *in parallel*, the necessary number of times. This way, the lengths in both flows directions will remain constant, while the length *perpendicular* to both flows directions scales *linearly* with the increased mass flow. Internally, rather than keeping the same number of tubes and increasing their inner diameters, the strategy is to keep the inner diameter of the tubes constant and increase the number of parallel tubes to accommodate the extra mass flow, thus keeping the fluid velocities constant as well.

Figure 8.9 compare these two different scaling strategies for the HRSG, taking the 40 MW CCGTs cycle described in Appendix C as the “base-brick” to scale up from. Cost-wise, either strategy is the

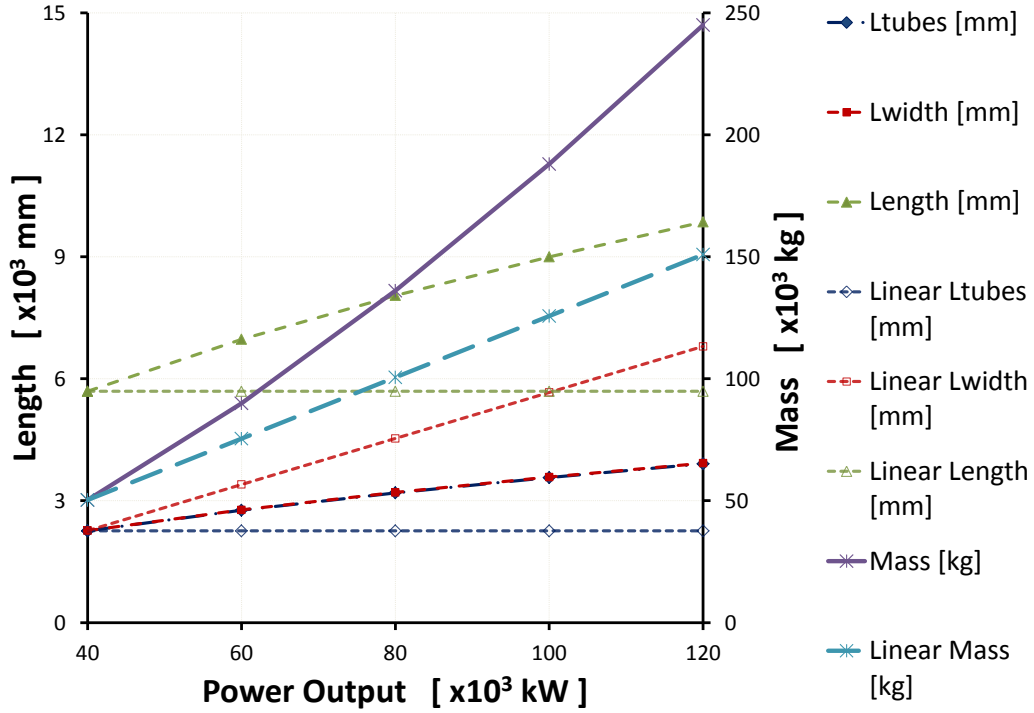


Figure 8.9: Size and mass for the optimal HRSG of the CCGT power plant, considering either linear- or $\sqrt{m}$ -scaling laws

same: the primary heat transfer surface area increases almost identically for both—in one case by increasing the number of tubes, in the other by increasing both the inner diameter and the length (and thus the *residence time*) of the tubes—, hence the acquisition costs for both are almost the same, according to the assumptions discussed in Section 8.3. This is an expected outcome, given that the price of a heat exchanger is basically dependent upon the target effectiveness and the amount of heat transferred, hence two heat exchangers fulfilling the same duty with the same effectiveness—and manufactured with the same *technology* and in the same *material*—should be almost identically priced. However, in terms of mass, the potential savings by means of linear-scaling becomes apparent, especially for the higher power outputs, where mass reductions as high as 40% can be achieved (see fig. 8.9).

8.4.4 Recuperators

These will be, as previously stated, of the *plate & fin* type heat exchangers. The aim is to find the most *compact* geometry—following the methodology outlined in Section 8.1 and Appendix D—that would fulfil the required duty—determined by the *optimised* thermodynamic cycles presented in Appendix C—for the heat exchanger. In this regard, *maximum compactness* has been defined as the *minimum “perimeter”*, or *minimal* sum of the three external dimensions of the heat exchanger, $L_1 + L_2 + L_3$ (note that, in this case, L_3 does not stand for the height of a single pass, L_{pass} , but for the *overall length* of the multi-pass heat exchanger instead: $L_3 = L_{\text{overall}} = N_{\text{pass}} \cdot L_{\text{pass}}$).

In the case of the recuperator, the pressure drop on each side has been imposed to the thermodynamic cycles, as per assumptions discussed in Section 2.1.3. For this reason, a study regarding the single-objective optimisation (SOO) of the geometry of the recuperator, by means of the implementation of a GA with *maximum compactness* for a given duty as the only *objective function*, and *constrained* by the imposed pressure losses, was conducted at this point. The list of *decision variables* for the GA optimiser, as well as the convergence process to produce suitable geometries based on these *decision variables*, is described in detail in Appendix D.2. The required duty for the

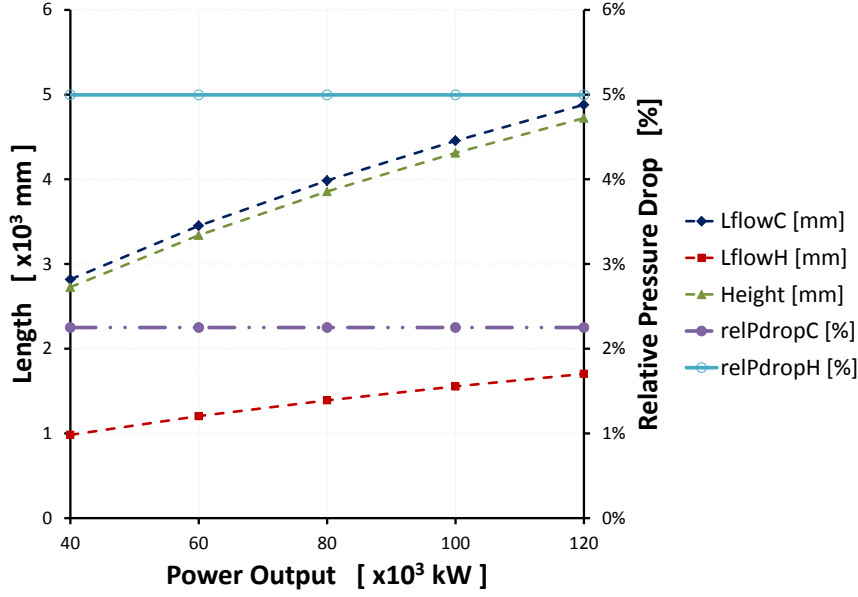


Figure 8.10: Size and pressure losses for the optimal recuperator of the IC-RH-R EvGT power plant

heat exchanger is calculated according to the $\epsilon - NTU$ method discussed in Section 8.1, taking the *optimised* thermodynamics presented in Appendix C as the only input. Once a consistent geometry has been produced, pressure losses, Δp , can be calculated as per Section 8.2 (eq. 8.11).

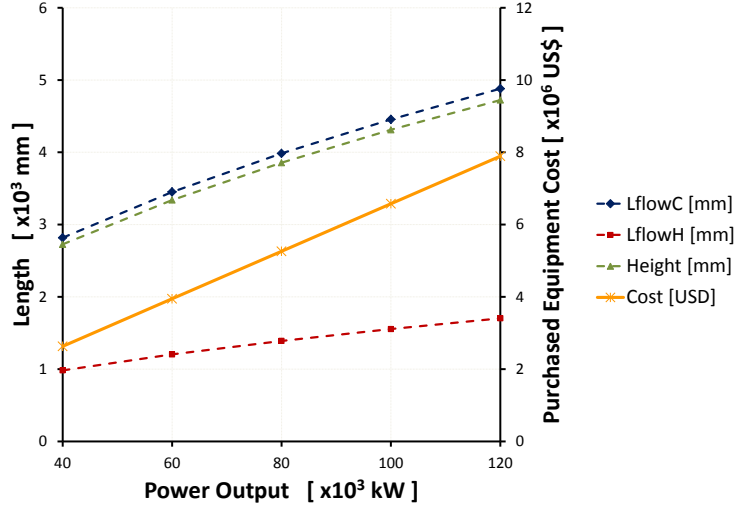
Figure 8.10 plots the external dimensions and the relative pressure losses on either side of the *optimal* recuperator for the IC-RH-R EvGT cycle, across the whole range of power outputs analysed, following once again the $\sqrt{m}$ -scaling law. It is worth noticing that out of the two *constraints* imposed by the specified pressure losses —5% in the hot side and 7.5% in the cold side—, only one is *active* in the optimum solution: the most compact recuperator is *constrained* by the pressure drop in the hot side only (see fig. 8.10).

Finally, mass and acquisition costs of a particular heat exchanger geometry can be estimated according to the assumptions outlined in Section 8.3.

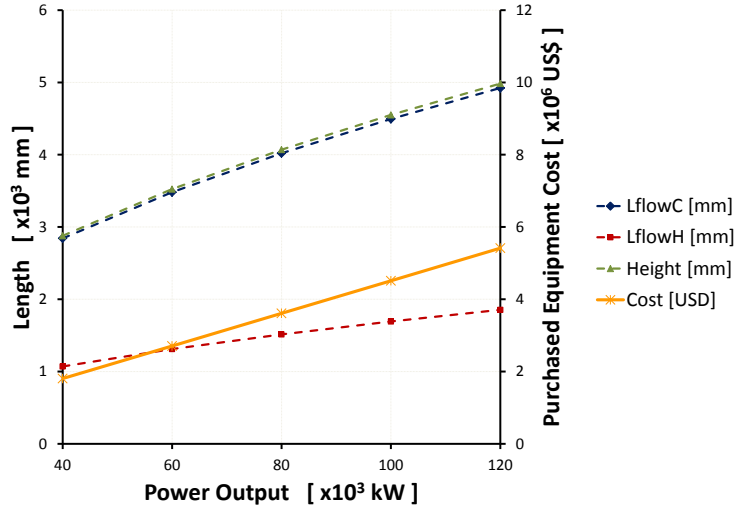
Materials Selection

In this case, however, materials considerations have to be made: in reheated cycles —this is, both the IC-RH-R EvGT and the RH-R CCGT described in Appendix C—, Inconel alloys are necessary to withstand the elevated operating temperatures, well in excess of 1000 K. The IC-R EvGT cycle from Appendix C is the only downselected power plant without a reheater, allowing therefore for the construction of its recuperator in stainless steel.

Figure 8.11 compares the size and acquisition costs, following the $\sqrt{m}$ -scaling law, of the recuperator for both the IC-RH-R and the IC-R EvGTs. The former, due to the high temperatures at the outlet of the free power turbine, featuring a recuperator manufactured in Inconel alloy; the latter featuring a recuperator manufactured in stainless steel. As it is shown, the overall size of the recuperator for the IC-RH-R EvGT (fig. 8.11a) is slightly smaller than that of the recuperator for the IC-R EvGT (fig. 8.11b). In fact, it is even lighter in terms of the total mass by about 16%. But still, it is almost 50% more expensive, due to the increased price of the selected materials. This fact serves to reinforce the statement made about the strong correlation existing between the overall thermal efficiency of a gas-turbine-based power plant and the acquisition costs of the heat exchangers that the power plant features: increased thermal efficiency comes at the price of increased heat exchangers acquisition costs, some times because of bigger and heavier devices required, some others because of the more expensive materials needed.



(a) IC-RH-R EvGT cycle: recuperator in Inconel alloy



(b) IC-R EvGT cycle: recuperator in stainless steel

Figure 8.11: Size and acquisition costs scaling laws for the optimal recuperators of the EvGT power plants: materials selection

8.5 Preliminary Design of the Humidifier

The humidifier is a particular type of heat exchanger, in which both *heat* and *mass* transfer takes place. The schematics of a saturation column are depicted in fig. 8.12, taken from [29]. As shown, the air circulates from the bottom of the column upwards, through the metal packing, heating up and absorbing evaporated water in the process. Hot water is sprayed onto the packing from the top of the column, cooling down as it travels downwards, creating a gradient of temperatures through the height of the saturator. The droplet eliminator, at the top of the column, avoids water droplets entrainment into the recuperator and the combustor, effectively providing the humidifier with a *scrubber*-like function that cleans the air from dust and other particles. This, assuming that appropriate care has been taken to ensure the necessary quality of the water being injected into the humidifier, facilitates self-sufficient operation of the EvGT [8, 5, 12], since most of the impurities tend to remain within the liquid-phase water, meaning no hazard for the integrity of the components in the hot segment of the gas turbine. Water can, therefore, work in a closed-loop circuit, without the need for any additional make-up water supplied from the exterior, by simply condensing moisture content from the exhaust

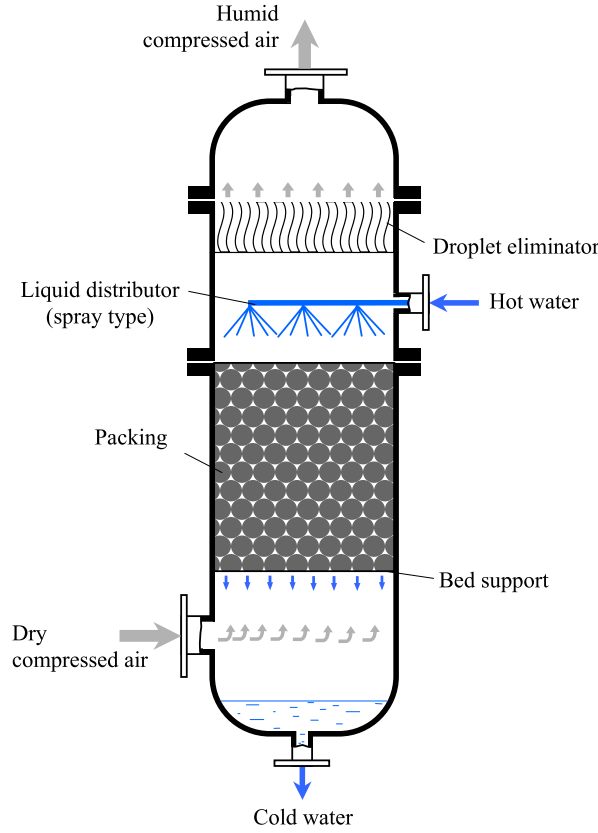


Figure 8.12: *Schematics of a saturation tower, taken from [29]*

gases of the EvGT and subsequently treating this condensate. Further details on the water-circuit chemistry, necessary blow-downs to avoid the building-up of salts concentration within the liquid-phase water, condensing temperature to ensure that sufficient water is recovered from the exhaust gases, and components selection for water treatment and self-sufficient operation of the EvGT can be found in [8, 5].

Sizing of the humidifier will be split in two parts. Namely: *a)* calculating the necessary packing *length*, to ensure that the saturation tower fulfils its duty; and *b)* calculating the *cross-sectional area* of the saturation tower, to keep pressure losses below a certain threshold.

Length of the Humidifier

The necessary length of the packing, Z , to fulfil the saturator duty will be calculated as per eq. 8.13, depending on the number of *transfer units*, NTU, of the thermodynamic process for the *optimal* EvGT cycles described in Appendix C, and based upon available data on the *height of a transfer unit*, HTU, which is the required packing length for a thermodynamic process of $\text{NTU} = 1$ to happen.

$$Z = \text{NTU} \cdot \text{HTU} \quad (8.13)$$

The detailed description of this sizing method can be found, among others, in [6, 7, 5, 39]. Basically, the definition of the NTU given in Section 8.1 as the “number of times” that the mean *driving force* acts in a heat transfer process applies here as well. The only difference is that, due to the particularities of the saturation process, the *driving force* cannot be defined straightforwardly in terms of temperature differences between the two streams alone, but has to account for the whole *heat* and *mass* transfer process instead.

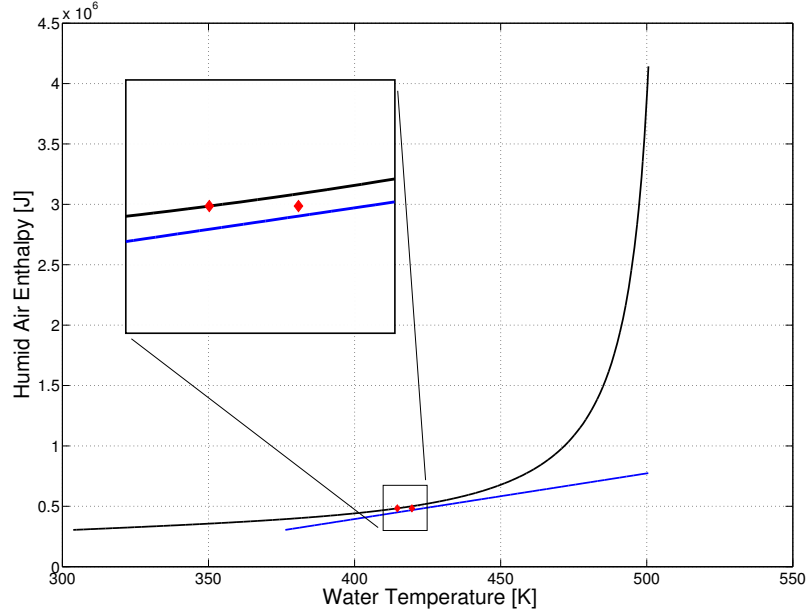


Figure 8.13: Operating and saturation lines for a typical humidifier

To understand the definition of the *driving force* for a saturation process, the operating and saturation lines of a typical humidifier are plotted in fig. 8.13, where the ‘pinch’ conditions at (T_0, h_0) and $(T_{w0} = T_0 + \Delta T, h_0)$ have been marked as well. To simplify the calculations, the operating line has been assumed to be straight, with constant slope between the top and the bottom of the column, and the error produced by this assumption when compared with the calculated ‘pinch’ conditions has been magnified (see detail in fig. 8.13), just for reference purposes. A more detailed discussion on the loss of accuracy due to this commonly made assumption can be found in the work presented by Enick et al. [39], and the errors reported are still small.

Following the diagram presented in fig. 8.13, at a given section of the saturation tower, water will be at a certain temperature, T_w , and the specific enthalpy of the humid air can be expressed as a (*linear*, because of the assumption made) function of this water temperature only: $h_g = h_g(T_w)$. The actual temperature and water content of the humid air at this section are unknown. In between of the water and the humid air streams, an *interphase* will form. This *interphase* layer of humid air is commonly assumed to be at the same temperature that the water stream, T_w , but completely saturated with moisture [39], so that its specific enthalpy will be given by the enthalpy of the saturation line: $h_{\text{interphase}} = h_{\text{sat}}(T_w) > h_g(T_w)$. The *driving force* can then be defined as the difference between the *interphase* and the humid air enthalpies, $h_{\text{sat}}(T_w) - h_g(T_w)$, and thus the number of transfer units, NTU, can be calculated as per eq. 8.14:

$$\text{NTU} = \int_{h_{g1}}^{h_{g2}} \frac{dh_g}{h_{\text{sat}} - h_g} = \int_{T_{w1}}^{T_{w2}} \frac{dh_g/dT_w}{h_{\text{sat}}(T_w) - h_g(T_w)} dT_w \quad (8.14)$$

where dh_g/dT_w is constant, due to the assumption made.

The *driving force* distribution along the height of the humidifier, therefore, can be readily calculated, and the term inside the second integral in eq. 8.14 has been plotted for the previous case in fig. 8.14. Effectively, the area below the curve plotted in fig. 8.14 is equal to the total number of transfer units for the aforementioned humidifier. As shown, the variations in the relative contribution from the different sections of the column to the total *length* of the humidifier are rather significant,

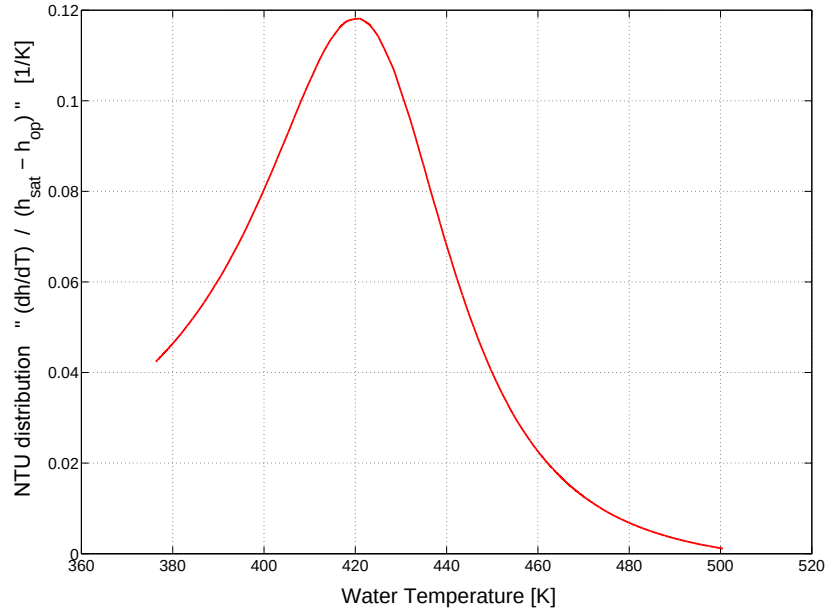


Figure 8.14: Driving force *distribution* for a typical humidifier

being the sections closer to the ‘pinch’ conditions the ones in which the *driving force* is minimal, and hence their relative contribution to the integral the greatest.

The sections with lower *driving force* are the least *effective* within the whole saturation column, and require greater lengths to produce similar changes in enthalpy than those sections with higher *driving force*. This means that the last few centimetres of packing in a humidifier are the most *effective* ones: the temperature of the humid air gets closer to the boiling point of water, the *volatility* of water in air increases close to this temperature, and so the specific enthalpy of the humid air increases more rapidly there than anywhere else in the column. This highlights the importance of injecting the hot water at the top of the humidifier as close as possible to its boiling point (minus a small safety margin), for this will yield the best saturator performances.

Cross-sectional Area of the Humidifier

The necessary cross-sectional area of the saturation tower can be determined, establishing a certain criterion so as to limit the pressure losses across the packing, according to the method described in [29, 27], among others. Essentially, fig. 8.15, taken from [27], provides generalised pressure drop correlations for packed towers, depending on the mass flow rates of the two streams (or, more specifically, on the *ratio* between the two, L'/G'), on the thermodynamic properties of the fluids (in particular, on their densities, ρ , and viscosities, μ), and on the characteristics of the selected packing through the F *packing factor* (which is a modification of the *specific surface area* of the selected packing [27]). Each line in this plot represents a constant level of *pressure drop per unit length of selected packing*, with the dashed line on top representing *flooding* conditions.

For safe operation of the humidifier, enough margin to the *flooding* line must be ensured at all times. Typically, the design pressure drop per unit length of packing will not exceed the equivalent to 60% of the flooding conditions, with the most common values chosen between 300 Pa/m [29] and 400 Pa/m [6, 7]. If the pressure drop per unit length is specified, the ratio of liquid-to-gas flow rates L'/G' is known, and the F packing factor determined according to the type of packing selected, then the *gas mass flow rate per unit of cross-sectional area*, G' , can be determined with the

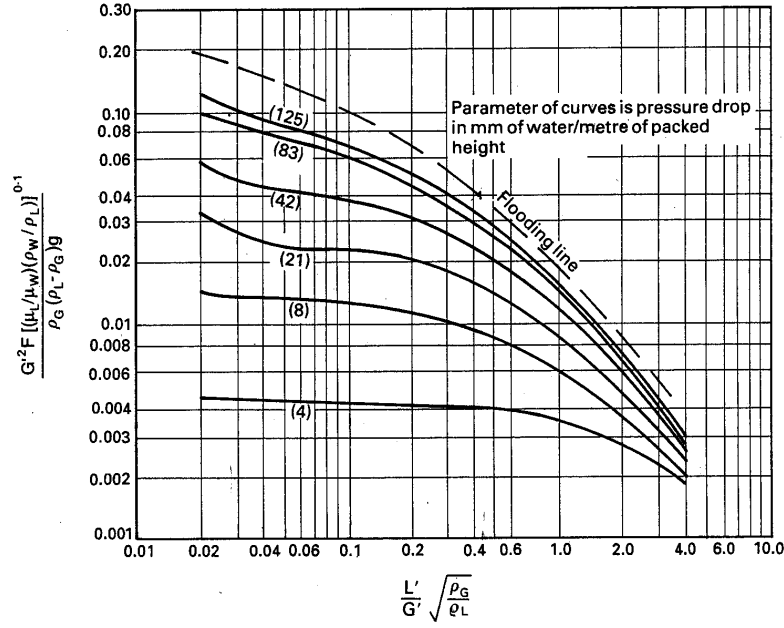


Figure 8.15: Generalised pressure drop correlations for packed towers

correlations provided in fig. 8.15, and the actual cross-sectional area of the tower can be worked out to accommodate the actual mass flow rates: $A = m_g/G'$.

Extended and more accurate pressure drop correlations can be found in Leva [64], for instance, valid for a greater range of different fluid mixtures and operating conditions. However, for the purposes of preliminary design of this study, the correlations provided in fig. 8.15 are considered to be accurate enough.

8.5.1 Final Size, Mass & Costs of the Humidifiers

For this study, $HTU \sim 0.60$ m was taken, similar to the one used in [6, 7], despite the fact that much lower values (between 0.25 m and 0.35 m) have been predicted and reported for certain packings in [29], among others, and thus it could be possible to achieve the same *heat* and *mass* transfer process with much lower packing heights. Following guidelines in [6, 7] as well, and additional 1 m was added to the total length of the column, to account for the necessary space for the installation of manifolds, flow distributors, droplet eliminator, etc.

To keep consistency with the above packing selection, $F \sim 50 \text{ m}^2/\text{m}^3$ was assumed, after sizing of the different cases analysed in [6, 7] for a 400 Pa/m pressure drop and alignment of the calculated cross-sectional areas with the results presented by Ågren et al. F packing factors for different packing choices can be found in [27], for instance.

For the mass assessment of the packed bed, a typical *void fraction* of 0.95 was chosen as representative for the stainless steel structured package selected [27]. With this, the total volume of the packing and the material density, mass of the packed bed alone can be readily calculated. The shell containing the packing, flow distributors, etc. is a pressure vessel, that can be sized, following the ASME BPVC Sections II and VII—as reported by Carlson and Conboy [21]—for the corresponding diameter and length of the column, similarly in stainless steel.

The overall purchased equipment cost of the humidifier can be estimated as per correlations given in [106, Table 7-2, p. 322 and after], both for the *stainless steel structured packing* and for the *stainless steel vertical pressure vessel*, depending on the total volume of the former and on the mass of the latter, expressed in 2010 US\$ already.

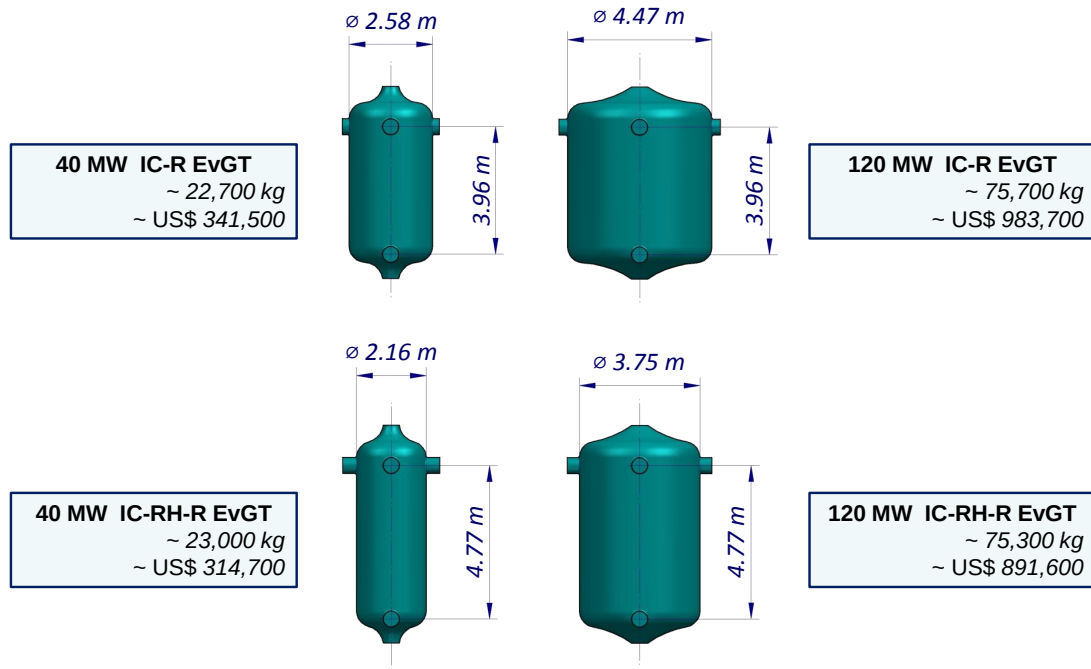


Figure 8.16: Summary of the results from the sizing, mass assessment and purchased equipment cost estimation for the humidifiers of several optimised EvGT power plants

With these assumptions, the overall size, mass and cost for the cases on both edges of the power outputs range analysed (40 MW to 120 MW), both for the IC-R and for the IC-RH-R EvGT power plants described in Appendix C, are summarised in fig. 8.16. It is worth noticing the *scaling laws* that apply to the design of saturation columns, and can be derived from these results: the length of the packed bed depends only on the thermodynamic process carried out, and not on the overall power output of the power plant —i.e. depends only on the temperature and pressure profiles and on the ratio of liquid-to-gas mass flows, but not on the absolute value of the mass flow rates circulating through the column—. Increased mass flows, on the contrary, affect the cross-sectional area of the saturation tower, that scales linearly with the overall power output of the power plant —hence the diameter of the column scaling with $\sqrt{m_g}$ —.

Chapter 9

Final Comparison of Downselected Power Plants

Based upon the results presented in Chapter 8, the goal is now to work on the overall arrangement and spacial distribution of all the components featured within each of the downselected power plants. With this, the overall size and mass of these *advanced* gas-turbine-based prime movers will be assessed, so that their individual merits can be compared, firstly against one another, and then against the diesel engines.

At these *early stages* of the design process, no specifications in terms of available space nor other particular requirements for the targeted cargo vessels are considered, hence the *proposed* arrangement of the downselected power plants presented herein must be understood as *one among many other possibilities*, aimed only at giving a visual idea on the overall size of the prime mover and the “connectivity” between the different components implemented within.

9.1 Turbomachinery design

A parallel research project, conducted by Mr. S. Mashruk, was launched at this point to deal with the preliminary design and sizing of the turbomachinery components (both for the gas turbine as well as for the steam turbomachinery featured within the CCGTs), based on the assumption of *constant mean diameter*, *constant stage loading* and *free vortex design* as the main simplifying hypotheses to carry out such design, and subsequently aligning these results against data provided by the sponsor regarding size and mass of the MT-30 gas turbine package. Further explanations on the materials selection and first principles in which the preliminary design of the turbomachinery components is based, along with extensive literature on the topic and the complete description of the detailed results can be found in Mashruk M.Sc. thesis [71]. A brief summary of the results reported in [71] that are relevant to the present research is included in tab. 9.1.

9.2 Evaporative Gas Turbines

Figure 9.1 depicts the schematic arrangement and *proposed* spacial distribution of the different heat exchangers and gas-turbine package featured within the 40 MW IC-R EvGT power plant, giving an idea on the overall size of such prime mover, if it were to be deployed in this way.

9.2.1 Comparison among the 40 MW power plants

To illustrate the differences between the IC-R and the IC-RH-R EvGT cycles, a comparison is to be made between both sorts of power plants, both designed for a power output of 40 MW, in terms of their heat exchangers size, mass and acquisition costs. Table 9.2 provides the break-down of the contribution to the total of ICs, ACs, ECs, recuperators and humidifiers.

Table 9.1: Gas and steam turbines size and mass estimations, based on [71]

	Size (m) <i>length</i> $\times$ <i>width</i> $\times$ <i>height</i>	Mass (kg)
Gas turbines		
IC-RH-R EvGT		
40 MW	$6.0 \times 1.5 \times 2.0$	4 900.
60 MW	$7.4 \times 1.9 \times 2.5$	8 500.
80 MW	$8.5 \times 2.2 \times 2.9$	12 600.
100 MW	$9.5 \times 2.4 \times 3.2$	17 000.
120 MW	$10.4 \times 2.6 \times 3.5$	21 800.
IC-R EvGT		
40 MW	$6.6 \times 1.9 \times 2.5$	10 800.
60 MW	$8.0 \times 2.3 \times 3.1$	18 700.
80 MW	$9.3 \times 2.7 \times 3.5$	27 600.
100 MW	$10.4 \times 3.0 \times 3.9$	37 300.
120 MW	$11.4 \times 3.2 \times 4.3$	47 700.
RH-R CCGT		
40 MW	$8.0 \times 2.1 \times 2.8$	11 700.
60 MW	$9.8 \times 2.6 \times 3.4$	20 200.
80 MW	$11.3 \times 3.0 \times 4.0$	29 800.
100 MW	$12.6 \times 3.3 \times 4.4$	40 300.
120 MW	$13.8 \times 3.6 \times 4.8$	51 500.
Steam turbines		
40 MW	$4.0 \times 4.0 \times 4.0$	9 200.
60 MW	$4.9 \times 4.9 \times 4.9$	15 900.
80 MW	$5.7 \times 5.7 \times 5.7$	23 500.
100 MW	$6.3 \times 6.3 \times 6.3$	31 800.
120 MW	$6.9 \times 6.9 \times 6.9$	40 600.

Several facts can be highlighted upon seeing this data:

- Regarding the IC, AC and EC, both cycles feature similarly-sized gas–water heat exchangers. However, the mass of these components within the IC-RH-R EvGT is much greater, especially in the case of the IC and the AC. This is due to the *internal* geometry of such components: in the IC-RH-R EvGT power plant, these heat exchangers have to accomplish greater heat-transfer duties, but work with much lower mass flows, as compared to the IC-R EvGT. For this reason, the *external* size of the gas–water heat exchangers in both sorts of power plant is somewhat comparable, since one fact tends to compensate for the other; but, *internally*, these components within the IC-RH-R have a greater number of smaller tubes with bigger extended surfaces on the hot side, hence lower *void fractions* resulting in heavier devices. This effect is particularly noticeable in the IC and, above all, in the AC, where the differences between the operating pressures on the gas (hot) side for the two EvGT cycles (see Appendix C.1 for the details) are greater.
- As discussed in Section 8.4.4, the recuperator of the IC-RH-R EvGT power plant is relatively smaller, and lighter, than that within the IC-R EvGT. Despite its reduced size and mass, which is due to the reduced mass flows circulating in the IC-RH-R EvGT cycle, the high temperatures at the outlet of the free power turbine of the *reheated* cycle forces the manufacturing of its recuperator in Inconel alloys, which are much more expensive.

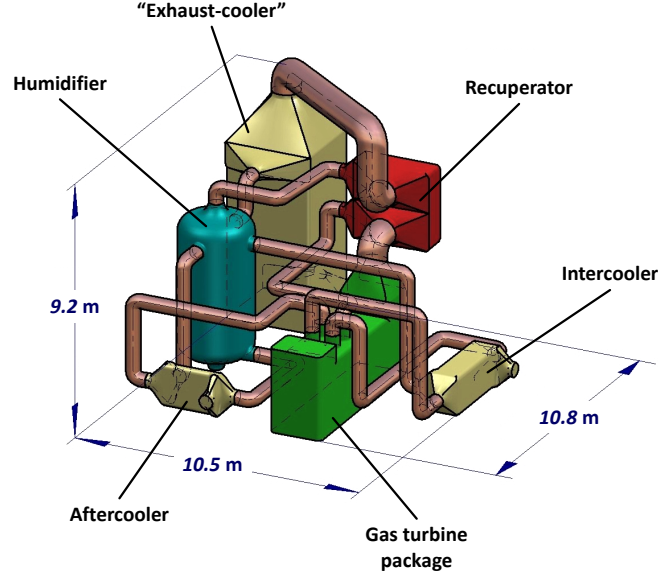


Figure 9.1: Overall size and proposed arrangement for the 40 MW IC-R EvGT power plant

- The humidifier represents in both cases the most important contribution to the overall mass of the heat exchangers implemented within the different power plants. In spite of the EC being the biggest component, in terms of size, the intricate metallic structure of the packed bed and the shell containing the humidifier are indeed much heavier, even more than the recuperator.

Table 9.2: Size, mass and acquisition costs break-down and comparison of the different heat exchangers implemented within the 40-MW-power-output EvGTs

	40 MW EvGT power plants					
	IC-R EvGT			IC-RH-R EvGT		
	Size (m) $L_1 \times L_2 \times L_3$	Mass (kg)	Cost (kUS\$)	Size (m) $L_1 \times L_2 \times L_3$	Mass (kg)	Cost (kUS\$)
Intercooler	1.1×2.3×1.1	1 100.	258.6	1.0×3.0×1.4	2 200.	523.6
Aftercooler	0.2×2.0×1.5	800.	120.3	0.2×1.3×1.5	2 600.	104.3
“Exhaust-cooler”	2.1×6.0×3.6	8 900.	2 554.	2.8×6.4×3.2	10 100.	2 861.
Recuperator	2.8×1.1×2.9	17 200.	1 804.	2.8×1.0×2.7	14 500.	2 631.
Humidifier	∅2.6 × 4.0	22 700.	341.5	∅2.2 × 4.8	23 000.	314.7

Compared with the **diesel engines**, taking the Wärtsilä X92 as a reference of a typical modern engine’s size and mass, 40 MW of power could be deliver by a 7-cylinder configuration [4], which would be 13.2 m long, 8.8 m wide, and 15.8 m height (to which an additional 2.5 m clearance has to be added for the removal of the engine’s pistons for maintenance purposes), according to the data provided by the manufacturer. This is somewhat comparable in size with the typical 40 MW EvGT power plant depicted in fig. 9.1. However, the overall mass of such a configuration would be around 1,260 t for the diesel engine, which represents a ~ 21 -fold increase in the prime mover mass with respect to any of the 40-MW EvGT power plants described herein (see tabs. 9.1 and 9.2).

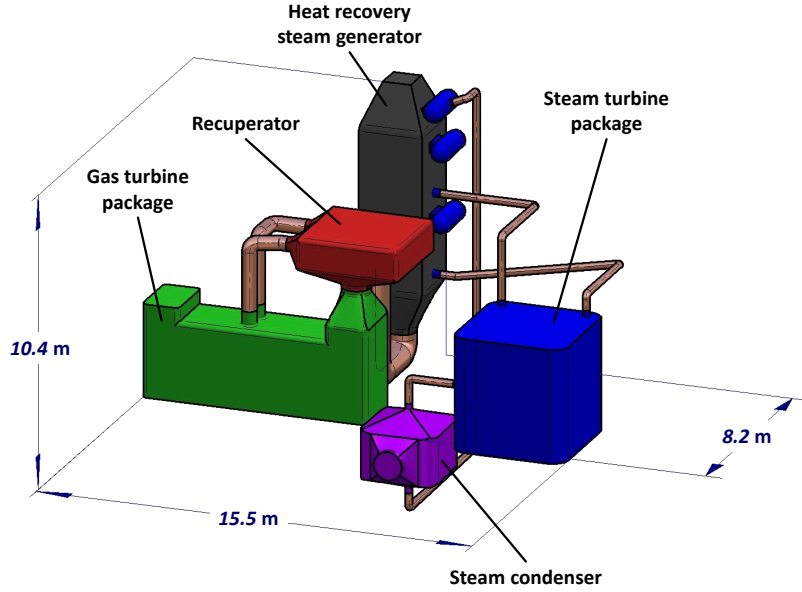


Figure 9.2: Overall size and proposed arrangement for the 40 MW CCGT

9.3 Combined Cycle Gas Turbines

Figure 9.2 depicts the schematic arrangement and *proposed* spatial distribution of the different heat exchangers and gas- and steam-turbine packages featured within the 40 MW RH-R CCGT power plant, giving an idea on the overall size of such prime mover, if it were to be deployed in this way. Comparison against the 40 MW EvGT represented in fig. 9.1 is thus made possible, and highlights the similitude—at the very least in terms of size—between the two prime movers. Given the inherent flexibility of any of the presented *advanced* gas-turbine-based power plants to have their components re-arranged in a different way, or even to have their heat exchangers re-designed to meet specific constraints, the deciding factor in order to choose one above the other will definitely not be their overall size, but rather their thermal efficiency, acquisition costs or total mass.

9.3.1 Comparison among the 120 MW power plants

To illustrate how the overall size of the power plant is *scaled* with its power output—considering firstly that the heat exchangers have been scaled for *maximum compactness* (understood as *minimum “perimeter”*) and, therefore, resulting in what has already been referred to as $\sqrt{m}$ -scaling—, fig. 9.3 depicts, side by side, the 40 MW and the 120 MW RH-R CCGT power plants.

Nevertheless, further considerations on the topics previously discussed in Sections 8.4.2 and 8.4.3—i.e. the manufacturing of the steam condenser with smooth-tubes geometries and the *linear*-scaling of the HRSG, respectively—can be made at this point. Data regarding the different *alternatives* for the heat exchangers of the 120 MW RH-R CCGT power plant, and already presented in figs. 8.8 and 8.9, are summarised in tab. 9.3.

Out of these data, it is worth noticing:

- The recuperator represents the highest contribution to the overall heat-exchangers acquisition costs of the RH-R CCGT power plant considered herein. This is due to mainly three circumstances: *a)* the high temperatures at the outlet of the free power turbine of a *reheated* cycle, that force its recuperator to be manufactured in Inconel alloys; *b)* the elevated mass flows circulating through both sides of the recuperator—much higher than those present in the recuperator of the IC-RH-R EvGT, for instance—, that result in bulkier components; and

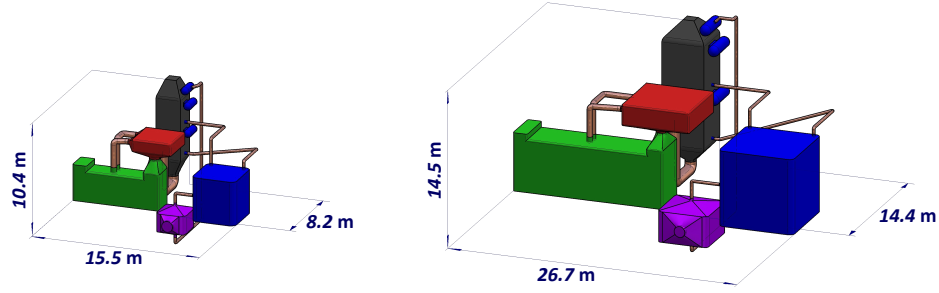


Figure 9.3: Overall size comparison between the 40 MW and the 120 MW CCPPs

c) the 90% recuperator *effectiveness*, across relatively low temperature differences between both streams, that yield as a consequence big heat transfer surfaces and long *residence times* to fulfil its duty.

- The HRSG is the heaviest of the heat exchangers implemented within the RH-R CCGT, accounting for about half of the overall mass of the power plant, including both the heat exchangers and the gas- and steam-turbine packages (see tabs. 9.1 and 9.3), when following the $\sqrt{m}$ -scaling law. For this reason, it is especially interesting to consider the *linear*-scaling of the HRSG instead, that will result in very similar acquisition costs, but also in a much lighter component.
- The acquisition costs of the steam condenser with finned tubes are rather similar to the acquisition costs of the HRSG, but the two of them together represent still the smaller fraction when compared with the acquisition costs of the recuperator for the RH-R CCGT. For this reason, the potential savings by choosing the smooth-tubes configuration for the condenser tend to “dilute” and lose relevance within the overall heat-exchangers purchased equipment costs, and their effect is much less noticeable than what one may anticipate from a solution that cuts down the cost of this component by about 75%. In addition to this, the mass comparison between the two steam condenser *alternatives* can be misleading in this case: even though the *dry* mass of the smooth-tubes configuration presented herein *seems* lighter than the finned-tubes geometry, that difference will become much smaller during operation, when the heat exchangers are filled with the working fluids, since the smooth-tubes arrangement has increased *residence times*, and therefore will have approximately an extra 50% of water total mass circulating through its cold side at any given point.

Table 9.3: Size, mass and acquisition costs break-down and comparison of the different alternatives for the heat exchangers implemented within the 120-MW-power-output CCGTs

	120 MW CCGT power plants		
	Size (m) $L_1 \times L_2 \times L_3$	Mass (kg)	Cost (kUS\$)
Recuperator	6.1×1.9×6.3	135 300.	12 660.
HRSG — $\sqrt{m}$ -scaling	3.9×9.9×3.9	245 000.	1 355.
HRSG — <i>linear</i> -scaling	2.3×5.7×6.8	150 900.	1 360.
Condenser — finned-tubes	4.6×2.8×4.8	13 000.	1 301.
Condenser — smooth-tubes	7.1×0.8×7.2	3 700.	314.5

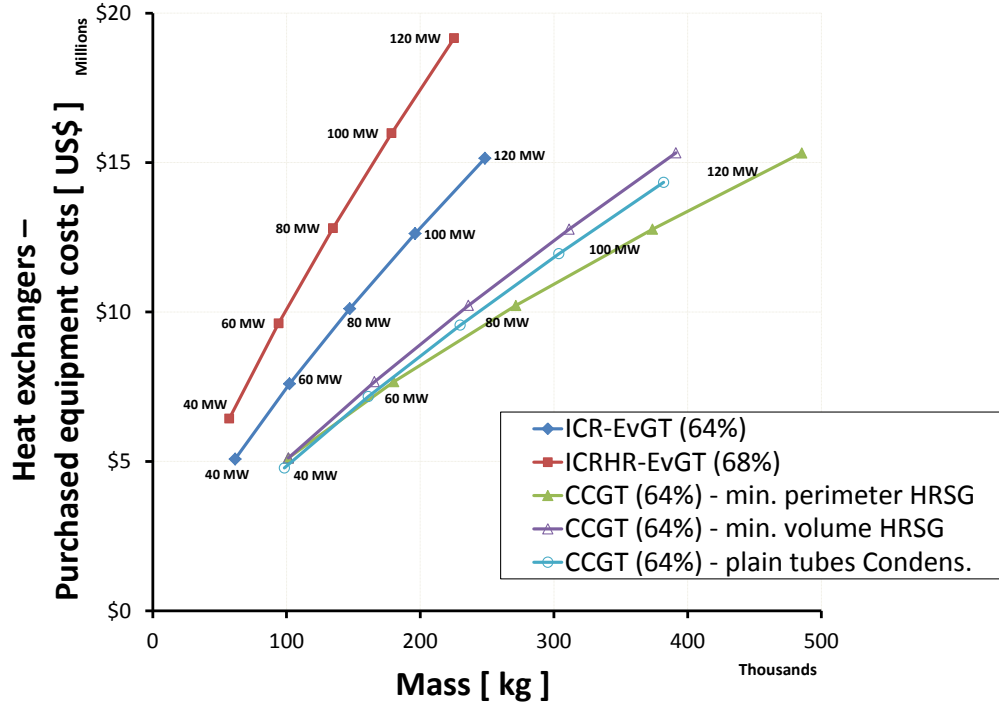


Figure 9.4: Heat exchangers acquisition costs against overall mass of the power plant for the downselected candidates, within the 40 MW to 120 MW range of power outputs

Compared with the **diesel engines**, taking the Wärtsilä X92 as a reference of a typical modern engine's size and mass, 120 MW of power could be delivered by **two** 10-cylinder configurations working in parallel [4], which would be 19.5 m long, 8.8 m wide, and 15.8 m height (to which an additional 2.5 m clearance has to be added for the removal of the engine's pistons for maintenance purposes) **each**, according to the data provided by the manufacturer. This is somewhat comparable in size with the typical 120 MW CCGT power plant depicted in fig. 9.3. However, the overall mass of such a configuration would be around 3,580 t for the two diesel engines, which still represents a ~ 7 -fold increase in the prime mover mass, even when compared against the $\sqrt{m}$ -scaled 120-MW RH-R CCGT described herein (see tabs. 9.1 and 9.3).

9.4 Summary

Figure 9.4 plots the final “design map” for the three downselected candidates, sized within the range of 40 MW to 120 MW power output, in terms of heat-exchangers purchased equipment costs against the overall mass of the whole power plant. In addition to the IC-R EvGT ($\eta_{th} \sim 64\%$) and the IC-RH-R EvGT ($\eta_{th} \sim 68\%$), three different lines are plotted for several *alternatives* in the design of the CCGT power plant ($\eta_{th} \sim 64\%$). Namely: (i) the one with all heat exchangers scaled for *minimum “perimeter”*, a.k.a. $\sqrt{m}$ -scaling; (ii) the one with the HRSG scaled for *minimum volume*, which corresponds indeed to what has been already referred to as *linear-scaling*; and (iii) the one with the HRSG scaled for *minimum volume* **and** the steam condenser designed with plain- rather than finned-tubes configuration.

Several conclusions are immediately apparent upon seeing these plots:

- There is a strong correlation between the thermal efficiency achieved by a power plant and the purchased equipment costs of the heat exchangers it features. Both the IC-R EvGT and the RH-R CCGT yield almost identical overall thermal efficiencies and the differences between their heat-exchangers acquisition costs are almost negligible, except for the single case of the CCGT

configuration that uses smooth tubes for the steam condenser —i.e. that moves into a *cheaper technology* than the compact finned-tubes heat exchangers—.

- The most efficient IC-RH-R EvGT thermodynamic cycle outcompetes the other two downselected candidates by about 4% points in thermal efficiency, but this efficiency comes at a price: the heat exchangers implemented within the IC-RH-R EvGT are over 25% more expensive, when compared with either the IC-R EvGT or the RH-R CCGT power plants.
- Moving into cheaper technologies —i.e. smooth tubes rather than finned tubes, etc.— can certainly mean a reduction in the acquisition costs of heat exchangers, provided that the deployment of bulkier components can be afforded. However, unless this solution is taken for the most crucial components, which will be proportionally more penalising for the overall size of the power plant, savings achieved by this means can “dilute” within the total cost of the prime mover and have little impact: just about a 6%–7% reduction in the heat-exchangers total acquisition costs by replacing the finned-tubes steam condenser with a smooth-tubes geometry.
- *Linear*-, as opposed to $\sqrt{m}$ -scaling of the heat exchangers, has no impact whatsoever on the overall acquisition costs of the power plant —as long as the same *technology* is chosen for all the heat exchangers featured within—, but can translate into great mass savings, especially at the higher end of the range of power outputs considered. The 120 MW RH-R CCGT with *linearly*-scaled HRSG is lighter by almost 20% than the one with $\sqrt{m}$ -scaled HRSG. If the same solution had been applied to the design of all the heat exchangers featured within this CCGT power plant, even greater mass reductions could have been achieved.
- Still, the overall mass of the CCGTs is much greater than that of the EvGTs (irrelevantly of whether the IC-R or the IC-RH-R EvGT is considered), in spite of their overall sizes being comparable, due to the lack of steam turbomachinery —which also reduces the necessary investments and specific costs for the whole power plant— in the latter.

9.4.1 Comments on the “Installation Effects”

If the masses presented in fig. 9.4 for the *advanced* gas-turbine-based power plants considered are compared against the typical masses of **diesel prime movers** —which can also be similar in terms of overall size of the whole power plant—, low-speed two-stroke diesel engines can be about 10–20 times heavier than the presented gas-turbine-based candidates for the same power output [4]. However, it must be noticed that the calculated masses presented herein correspond to the *uninstalled* prime mover, and thus there are some “*installation effects*” that would normally penalise the gas turbines and cannot be disregarded.

The installation of the different power plants into the targeted vessel and its effects on the overall mass and cost of the prime mover fall out of the scope of this work. Nevertheless, some qualitative discussion on this topic will be provided herein, mainly for reference purposes.

Intake & Outtake

Gas turbines installation requires an intake and an outtake to and from the engine room of the vessel, that usually adds a non-negligible amount to the overall mass of the power plant. These have to be designed to minimise the pressure losses occurring within —so that the performances of the gas turbine are not severely penalised—, which usually requires smooth surfaces, minimising turns and relatively big cross-sectional areas of the conducts.

Moreover, the negative effect of dust and other particles that can enter the compressor of the gas turbine from the ambient through the intake, and the quick degradation of the compressor performances in which this can result, usually requires the installation of filters in the intake conduct to avoid excessive maintenance requirements of the compressor or a severe overall efficiency reduction of the whole gas-turbine-based power plant.

The additional mass and the manufacturing and installation costs of these conducts and filters must be accounted for, if a fair and reliable comparison between the diesels and the gas turbines is to be produced.

Driving Alternatives

Whilst diesel prime movers usually implement direct drive of the vessel propellers, the rotational speed of the power shaft does make this impossible for the gas turbines. Two alternatives are feasible, namely: *a*) mechanical drive, via a reduction gearbox; or *b*) integrated full electric propulsion (IFEP), where the gas turbine drives an electric generator to satisfy all the vessel power demand. In the former, the gearbox introduces some mechanical losses, though typically small, in the system; while the net efficiency losses due to the energy conversion in the latter can reach as much as 3%–4%, which cannot be neglected (see, for instance, [36, 37, 72, 26, 112, 87], for a more comprehensive discussion on the benefits and disadvantages of either option, and how the increased flexibility of IFEP, for which gas turbines are particularly well suited, can outweigh the increased efficiency losses due to energy conversion: from mechanical into electrical and back to mechanical energy again).

On the other hand, having the gas turbine working as a generator-set to power electric motors driving the propellers allows the prime mover to operate closer to its design point at all times, generally achieving better thermal efficiency from the gas turbine, although similar effects can be achieved by implementing variable-pitch propellers in the case of a mechanical-drive arrangement.

In any case, the gearbox needed for a mechanical-drive configuration or the electric generator required for implementing IFEP both are fairly bulky components, which can be as heavy as the gas-turbine-based power plant itself (or even heavier), drastically reducing the huge difference between the power-plant masses shown in fig. 9.4 and the typical masses of modern low-speed two-stroke diesel engines.

Chapter 10

Conclusions

To summarise, the following are the most relevant findings and concluding remarks of this research project:

- From the broad range of state-of-the-art power plants analysed in Part I, the EvGTs are those which offer the greatest potential, in terms of design-point performance, closely followed by the CCGTs, which are the main trend for power generation nowadays. In particular, the three downselected thermodynamic cycles are, following the discussion in Chapter 7:
 - ◊ The intercooled-reheated-recuperated EvGT, yielding thermal efficiencies as high as 68% at OPRs $> 50:1$.
 - ◊ The intercooled-recuperated EvGT, with thermal efficiencies above 64% at OPRs $\sim 35:1$.
 - ◊ The reheated-recuperated CCGT, featuring a dual-pressure (non-reheated) SBC working at steam pressures of around 118 bar, that yields thermal efficiencies above 64% at gas-turbine OPRs $\sim 30:1$.
- In addition to their higher overall thermal efficiency at design point, EvGTs offer other advantages over the CCGTs for being deployed as prime movers in large cargo vessels, such as reduced NO_x emissions due to high levels of humidity present in the combustion chamber, reduced mass and specific capital costs due to the lack of need for steam turbomachinery, and the delivery of all the generated useful power in a single shaft without gearboxes.
- *Advanced* gas-turbine-based power plants have been shown to have great potential to outperform the diesels in terms of thermal efficiency at design point by a generous margin, up to at least 5–10 percentage points. Even though these results are not enough to fully assess the viability of gas turbines for entering the market of large cargo vessels as more *profitable* prime movers than diesel engines (part-load operation, “*installation effects*”, and overall acquisition and operating costs coupled with characteristic mission profiles for each particular marine commercial application should be carefully evaluated to fully address this question), the promising levels of predicted thermal efficiency at design point shown here can be taken as the first stone, establishing the foundation for subsequent studies and encouraging further research on this topic.
- According to the methodology and results presented in Part II, there is a strong correlation between the maximum achievable thermal efficiency of any *advanced* gas-turbine-based power plant and the acquisition costs of the heat exchangers featured within. More specifically, the higher design-point thermal efficiency of the IC-RH-R EvGT thermodynamic cycle —about 4% points higher than that of the other two downselected candidates— translates into an increase of around 20% in the acquisition costs of the heat exchangers it implements. On the other hand, both the IC-R EvGT and the RH-R CCGT, that feature almost identical thermal efficiencies at design point, have a barely noticeable difference in the acquisition costs for their heat exchangers, despite the huge differences between both configurations.

- Both EvGT downselected cycles are very similar in terms of overall size and mass of the whole power plant, despite their relatively big discrepancy in acquisition costs and performance. The overall size of the RH-R CCGT is also comparable to that of the EvGTs; however, the CCGT power plant is almost twice as heavy, due to the steam turbine mass and the heavier heat exchangers (in particular, the HRSG) implemented within.
- Still, diesel prime movers, also similar in overall size, represent a 10- to 20-fold increase in mass when compared to any of the downselected *advanced* gas-turbine-based power plants, as long as “*installation effects*” are disregarded —the installation of an intake and an outtake, the need for filters within this intake to prevent dust and other particles from entering the gas turbine core, gearboxes for configurations with mechanical drive of the propellers or electric generators for vessels implementing IFEP, etc., all increase considerably the final mass (and cost) of gas-turbine-based power plants deployed in any ship, as discussed in Section 9.4.1— and only the isolated *uninstalled* prime movers are considered.
- The only major disadvantage of the EvGT power plants over the CCGTs consists in their lower *technology readiness level* (TRL), given that combined gas-steam cycles are already well-proven technology, for which many thousand-hours of operating experience have been accumulated during the last decades, and this is not the case for the evaporative power plants. Establishing a reliable TRL for the EvGTs is not an easy task: other than the low-scale demonstrator at Lund University (see [91, 5, 12]), major work on the development and testing of a 40 MW EvGT power plant is being carried nowadays by Hitachi Ltd. (Japan), and reported so far in [114, 104].

Challenges for the development of the EvGT power plants

There are two main challenges in sight for the deployment of the EvGT cycles as viable operating power plants in the nearest future:

- Firstly and foremost, the turbomachinery design. Due to the mass flow imbalance between the compression and the expansion side of an EvGT, current gas-turbines cannot be straightforwardly adapted for operation in this sort of cycles. An attempt was made by Nakhamkin et al. in [76] to develop an intercooled-reheated-recuperated EvGT —reported as the CHAT cycle— with a currently existing industrial gas turbine as the LP (and generating) spool and with available compressors and turbines only for a parallel HP shaft in which a non-generating spool was implemented. However, if the potential of the EvGT is to be fully exploited, new designs for turbomachinery components able to accommodate the mass flows mismatch between the compression and the expansion side will be required.
- Second but most importantly, the combustor design. The high levels of specific humidity predicted for the IC-RH-R EvGT cycle, in excess of 40%, are far beyond the operating conditions tested at Lund’s demonstrator (working with approximately 20% specific humidity, at most [91, 5, 12]). While Lund’s combustor had to be only slightly modified to accommodate the extra mass flow and humidity, the much higher water-to-air ratios of the proposed power plants will most likely require brand new designs, to ensure proper combustion and flame stability. The work reported by Hitachi Ltd. [114, 104] refers only to partial humidification (with the water circuit working at only a fraction of its full capacity) and data on the maximum levels of moisture content and flame stability at this operating point are not provided. Therefore, it is hard to throw any light on this matter for the time being, and further work in this regard will be necessary in order to establish the actual limitations for combustion and flame stability with such high levels of specific humidity.

Even though combustion with these levels of water content can be theoretically achievable, an alternative solution might be to by-pass fresh dry-air from the HP compressor outlet directly into the recuperator cold-side inlet, diluting therefore the humidity absorbed into the air within the saturation column, and keeping the total water content of the air entering the combustion chamber within appropriate margins.

10.1 Future Work

Based upon the studies presented in this thesis, several tasks can immediately be identified as possible guidelines to lead subsequent research projects in this topic. Namely:

- **Introduction of blade-cooling modelling within the thermodynamic analyses.** To simplify the design-space exploration studies presented in Part I, and reduce as much as possible the required computational time, blade-cooling requirements have not been accounted for in the thermodynamic modelling of the *advanced* gas-turbine-based power plants. To ensure that this hypothesis was reasonable enough for a *comparative assessment* only, the analysis of the open ‘dry’ cycles with blade-cooling modelling implemented was carried out and presented by the author in [94]. Quick comparison between these results and those presented in Section 2.2 prove that the conclusions derived from both studies are identical. As a rule of thumb, 1% point of thermal efficiency can be deducted from the non-cooled models per combustor/re-heater present in the cycle, and therefore all the *comparative assessment* of the different cycles’ performance summarised in Chapter 7 would still hold true if blade-cooling had been implemented, in spite of the *absolute* values of thermal efficiencies presented herein being slightly overestimated. Nevertheless, for greater accuracy in predicting the overall performances of the downselected power plants within actual marine commercial applications, further research on the different cooling methods available and optimal configuration should be carried out.
- **Performance assessment at off-design conditions.** Following with the more detailed performance predictions for the downselected power plants, assessment of their part-load operation and transient behaviour is the next logical step in this regard. Despite the assumption of large cargo vessels spending most of the time at cruise speed, power plants are required to start up and change load conditions with reasonable flexibility and within reasonable times, as well as expected to operate in a wide range of ambient conditions, from tropical seas to the Arctic Ocean. In this regard, preliminary design studies carried out for the downselected power plants assuming operation in tropical seas suggest that the EvGTs are less penalised and more easily operated under varying ambient conditions than the CCGTs, which is in good agreement with the literature.
- **Power plant optimisation for a particular mission profile.** Once the performances of these power plants in different operating conditions can be assessed accurately, coupling of the power plant optimiser with a particular (or, at the very least, a ‘*more generic, typical*’) mission profile should be performed. Depending on the amount of time expected to operate at different loads, or under different ambient conditions, the power plant can be finely tuned to best suit the particular requirements of each vessel. Further studies on the overall thermo-economics of the deployment of *advanced* gas-turbine-based power plants, and their life-cycle cost analysis, in different applications with different mission profiles, can then be carried out, to better establish the optimal power plant for a particular vessel.
- **Considering design trade-offs and constraints to target individual applications.** Furthermore, the particular constraints and requirements of each vessel and the “*installation effects*” discussed in Section 9.4.1 should be accounted for in the physical design and optimisation of the power plant, especially when considering the design of the heat exchangers. The flexibility of the *advanced* gas-turbine-based power plants in terms of re-arranging their components to meet the constraints of different applications means already a big advantage over the diesel prime movers. Moreover, as discussed in Chapter 8, the design of the heat exchangers is quite flexible *per se*, which allows for even better adaptation of these power plants to the constraints and requirements of different vessels. Studies regarding the trade-offs between reducing the size of the heat exchangers featured within a particular power plant, to fit in certain applications, and the corresponding penalisation in thermal efficiency due to less *effective* heat exchangers can as well result in some interesting research, to finally and better assess the suitability of *advanced* gas-turbine-based power plants for marine commercial applications.

Bibliography

- [1] Available on the WWW. URL & download <http://freesteam.sourceforge.net/>.
- [2] Roll-Royce Group plc. Annual Report 2010. On the WWW, accessed between December 2011 and March 2012.
- [3] Rolls-Royce Group plc. Marine Investor's Briefing 2010. On the WWW, accessed between December 2011 and March 2012.
- [4] Wärtsilä X92 low-speed two-stroke diesel engine technical information. On the WWW, accessed October 2013. URL <http://www.wartsila.com/en/engines/generation-X-low-speed-merchant-marine-2-stroke/W-X92>.
- [5] N.D. Ågren. *Advanced Gas Turbine Cycles with Water–Air Mixtures as Working Fluid*. PhD Thesis, Department of Chemical Engineering and Technology, KTH, Stockholm, 2000.
- [6] N.D. Ågren and M.O. Westermark. Design Study of Part-Flow Evaporative Gas Turbine Cycles: Performance and Equipment Sizing — part I: Aeroderivative Core. *ASME J. Engng. Gas Turbines Power*, 125(1):201–215, 2003.
- [7] N.D. Ågren and M.O. Westermark. Design Study of Part-Flow Evaporative Gas Turbine Cycles: Performance and Equipment Sizing — part II: Industrial Core. *ASME J. Engng. Gas Turbines Power*, 125(1):216–227, 2003.
- [8] N.D. Ågren, M.O. Westermark, Michael Bartlett, and T. Lindquist. First Experiments on an Evaporative Gas Turbine Pilot Plant — Water Circuit Chemistry and Humidification Evaluation. *ASME J. Engng. Gas Turbines Power*, 124(1):96–102, 2002.
- [9] Andrés Aramayo Prudencio. *Thermodynamics of Water Injected Gas Turbine Cycles*. PhD Thesis, Cambridge University, 2004.
- [10] Carmen Arnaiz Franco and Emilio Díaz Ojeda. Basic Operations for Chemical Engineering. Open Course Ware, Department of Chemical and Environmental Engineering, Universidad de Sevilla, Spain. Available on the WWW (in Spanish), accessed November 2012. URL http://ocwus.us.es/arquitectura-e-ingenieria/operaciones-basicas/contenidos1/tema14/pagina_04.htm.
- [11] Raquel Ávila. Off-Design Performance Evaluation of Combined Cycle Power Plants. MS Thesis, School of Engineering, Cranfield University, 2011.
- [12] Michael Bartlett. *Developing Humidified Gas Turbine Cycles*. PhD Thesis, Department of Chemical Engineering and Technology, KTH, Stockholm, 2002.
- [13] A.M. Bassily. Analysis and Cost Optimization of the Triple-pressure Steam-Reheat Gas-Reheat Gas-Recuperated Combined Power Cycle. *Int. J. Energy Res.*, 32:116–134, 2008.
- [14] A.M. Bassily. Cost Numerical Optimization of the Triple-pressure Steam-Reheat Gas-Reheat Gas-Recuperated Combined Power Cycle that uses Steam for Cooling the First Gas Turbine. *Int. J. Energy Res.*, 32:1399–1417, 2008.

- [15] A.M. Bassily. Enhancing the Efficiency and Power of the Triple-pressure Reheat Combined Cycle by Means of Gas Reheat, Gas Recuperation, and Reduction of the Irreversibility in the Heat Recovery Steam Generator. *Applied Energy*, 85:1141–1162, 2008.
- [16] O. Bolland, M. Førde, and B. Hånde. Air Bottoming Cycle. Use of Gas Turbine Waste Heat for Power Generation. 1995. ASME Paper No. 95-CTP-50.
- [17] O.A. Bolland. Comparative Evaluation of Advanced Combined Cycle Alternatives. *ASME J. Engng. Gas Turbines Power*, 113(2):190–195, 1991.
- [18] M.S. Briesch, R.L. Bannister, I.S. Dinkunchak, and D.J. Huber. A Combined Cycle Designed to Achieve Greater than 60% Efficiency. *ASME J. Engng. Gas Turbines Power*, 117(4):734–741, 1995.
- [19] D. Bücker, R. Span, and W. Wagner. Thermodynamic Property Models for Moist Air and Combustion Gases. *ASME J. Engng. Gas Turbines Power*, 125(1):374–384, 2003.
- [20] C. Carcaschi and B. Facchini. Comparison Between Two Gas Turbine Solutions to Increase Combined Power Plant Efficiency. *Energy Conversion and Management*, 41:757–773, 2000.
- [21] Matt Carlson and Thomas Conboy. Scaling Considerations for SCO₂ Cycle Heat Exchangers. In *Proceedings of ASME Turbo Expo 2014*, Düsseldorf, Germany, June 16–20, 2014. ASME Paper No. GT2014-27233.
- [22] G. Cerri. Parametric Analysis of Combined Cycles. *ASME J. Engng. Gas Turbines Power*, 109(1):46–55, 1987.
- [23] D.Y. Cheng. “Regenerative Parallel Compound Dual-Fluid Heat Engine”. U.S. Patent No. 4,128,994, Dec. 12, 1978.
- [24] P. Chiesa and E. Macchi. A Thermodynamic Analysis of Different Options to Break 60% Electric Efficiency in Combined Cycle Power Plants. *ASME J. Engng. Gas Turbines Power*, 126:770–785, 2004.
- [25] Stuart W. Churchill and Stefan C. Zajic. Prediction of Fully Developed Turbulent Convection with Minimal Explicit Empiricism. *AIChE Journal. Fluid Mechanics and Transport Phenomena*, 48(5):927–940, 2002.
- [26] R. Colin. The WR-21 Intercooled Recuperated Gas Turbine Engine — Integration into Future Warships. In *Proceedings of the International Gas Turbine Congress 2003*, Tokyo, Japan, November 2–7, 2003. Paper No. IGTC2003Tokyo OS-203.
- [27] J.M. Coulson, J.F. Richardson, J.H. Harker, and J.R. Backhurst. *Chemical Engineering Volume 2: Particle Technology and Separation Processes*. Butterworth-Heinemann, 5th ed. 2002.
- [28] A.J. Crissalli and M.L. Parker. Overview of the WR-21 Intercooled Recuperated Gas Turbine Engine System — A Modern Engine For A Modern Fleet. In *Proceedings of ASME Turbo Expo 1993*, Cincinnati, Ohio, USA, May 24–27, 1993. ASME Paper No. 93-GT-231.
- [29] F. Dalili. *Humidification in Evaporative Power Cycles*. PhD Thesis, Department of Chemical Engineering and Technology, KTH, Stockholm, 2003.
- [30] P. Dechamps. Cranfield University lecture notes on *Combined Cycle Gas Turbines*. 2012.
- [31] Giusepina Di Lorenzo. *Advanced Low-Carbon Power Plants — the TERA Approach*. PhD Thesis, School of Engineering, Cranfield University, 2010.
- [32] DieselNet. Emission Standards: IMO Marine Engine Regulations. On the WWW, accessed between December 2011 and March 2012. URL <http://www.dieselnet.com/standards/inter/imo.php>.

- [33] Adam Doligalski. Gas–Steam Cycles Efficiency and Feasibility Studies. MS Thesis, School of Engineering, Cranfield University, 2013.
- [34] Adam Doligalski, Luis Sánchez de León, Pavlos K. Zachos, and Vassilios Pachidis. Assessing the Potential of Gas-Recuperation in Reheated Gas Turbines within Combined Gas–Steam Power Plants. In *Proceedings of ASME Turbo Expo 2014*, Düsseldorf, Germany, June 16–20, 2014. ASME Paper No. GT2014-27048.
- [35] Vaclav Dostal. A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors. PhD Thesis, Dept. of Nuclear Engineering. Massachusetts Institute of Technology, 2004. Available on the WWW, URL & download <http://hdl.handle.net/1721.1/17746>.
- [36] I.E. Douglas. Optimization of Gas Turbine Cycles for Ship Power Plant Applications. In *Proceedings of the 7th International Symposium on Marine Engineering*, Tokyo, Japan, 2005.
- [37] I.E. Douglas. Performance Evaluation of Combined Cycles for Cruise Ship Applications. In *Proceedings of ASME 2008 International Mechanical Engineering Congress and Exposition*, Boston, Massachusetts, USA, October 31–November 6, 2008. ASME Paper No. IMECE2008-67393.
- [38] M.A. El-Masri. A Modified High-Efficiency Recuperated Gas Turbine Cycle. *ASME J. Engng. Gas Turbines Power*, 110(2):233–242, 1988.
- [39] R.M. Enick, S.M. Klara, and J.J. Marano. A Robust Algorithm for High-Pressure Gas Humidification. *Journal of Computers and Chemical Engineering*, 19(10):1051–1061, 1995.
- [40] V. Eyring, H.W. Köhler, J. Van Aardenne, and A. Lauer. Emissions from international shipping: 1. The last 50 years. *Journal of Geophysical Research D: Atmospheres*, 110(17):171–182, 2005.
- [41] V. Eyring, H.W. Köhler, A. Lauer, and B. Lemper. Emissions from international shipping: 2. Impact of future technologies on scenarios until 2050. *Journal of Geophysical Research D: Atmospheres*, 110(17):183–200, 2005.
- [42] W.E. Fraize and C. Kinney. Effects of Steam Injection on the Performance of Gas Turbine Power Cycles. *ASME J. Engineering for Power*, 101:217–227, 1979.
- [43] Alessandro Franco and Nicola Giannini. Optimum Thermal Design of Modular Compact Heat Exchangers Structure for Heat Recovery Steam Generators. *Applied Thermal Engineering*, 25:1293–1313, 2005.
- [44] Alessandro Franco and Nicola Giannini. A General Method for the Optimum Design of Heat Recovery Steam Generators. *Energy*, 31:3342–3361, 2006.
- [45] H.U. Frutschi and A.A. Plancherel. Comparison of Combined Cycles with Steam Injection and Evaporation Cycles. *ASME Cogen-Turbo IGTI*, 3:137–145, 1988.
- [46] Gas Turbine World. GTW Specifications, Jan. 2005.
- [47] GE Power and Water. Handbook of Industrial Water Treatment: Boiler Water Systems. On the WWW, accessed August 2013. URL http://www.gewater.com/handbook/boiler_water_systems/index.jsp.
- [48] T.K. Gogoi. A Combined Cycle Plant with Air and Fuel Recuperator for Captive Power Application, Part 1: Performance Analysis and Comparison with Non-Recuperated and Gas Turbine Cycle with Only Air Recuperator. *Energy Conversion and Management*, 79:771–777, 2014.
- [49] F. Haglind. A Review on the use of Gas and Steam Turbine Combined Cycles as Prime Movers for Large Ships. Part I: Background and Design. *Energy Conversion and Management*, 49(12):3458–3467, 2008.

- [50] F. Haglind. A Review on the use of Gas and Steam Turbine Combined Cycles as Prime Movers for Large Ships. Part II: Previous Work and Implications. *Energy Conversion and Management*, 49(12):3468–3475, 2008.
- [51] F. Haglind. A Review on the use of Gas and Steam Turbine Combined Cycles as Prime Movers for Large Ships. Part III: Fuels and Emissions. *Energy Conversion and Management*, 49(12):3476–3482, 2008.
- [52] S. Heide, C. Felsmann, U. Gampe, S. Boje, B. Gericke, M. Freimark, U. Langnickel, R. Buck, and S. Giuliano. Parameterization of High Solar Share Gas Turbine Systems. In *Proceedings of ASME Turbo Expo 2012*, Copenhagen, Denmark, June 11–15, 2012.
- [53] G.F. Hewitt, G.L. Shires, and T.R. Bott. *Process Heat Transfer*. CRC Press, 1994.
- [54] J.H. Horlock. *Advanced Gas Turbine Cycles*. Elsevier Science, 2003.
- [55] International Association for the Properties of Water and Steam. IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam, August 2007. Lucerne, Switzerland.
- [56] International Maritime Organization. MARPOL Annex VI: regulations for the prevention of air pollution from ships, London: consolidated edition 2006.
- [57] K. Johnson. Study of Humidification and Condensation Dynamics in an Evaporative Gas Turbine Cycle. MS Thesis, Department of Energy Sciences, Faculty of Engineering, LTH, Lund, 2002.
- [58] Maria Jonsson and Jinyue Yan. Economic Assessment of Evaporative Gas Turbine Cycles with Optimized Part Flow Humidification Systems. In *Proceedings of ASME Turbo Expo 2003*, Atlanta, Georgia, USA, June 16–19, 2003. ASME Paper No. GT2003-38009.
- [59] W.M. Kays and A.L. London. *Compact Heat Exchangers*. Krieger Publishing, 3rd ed. 1998.
- [60] Rolf Kehlhofer. *Combined-Cycle Gas & Steam Turbine Power Plants*. PennWell Books, 3rd ed. 2009.
- [61] M.A. Korobitsyn. *New and Advanced Energy Conversion Technologies. Analysis of Cogeneration, Combined and Integrated Cycles*. Febroduct B.V., Enschede, 1998.
- [62] M.A. Korobitsyn. Industrial Applications of the Air Bottoming Cycle. *Energy Conversion and Management*, 43:1311–1322, 2002.
- [63] T. Kuppan. *Heat Exchanger Design Handbook*. CRC Press, 2nd ed. 2013.
- [64] Max Leva. Reconsider Packed-Tower Pressure-Drop Correlations. *Chemical Engineering Progress*, 88(1):65–72, January 1992.
- [65] T. Lindquist, M. Thern, and T. Torisson. Experimental and Theoretical Results of a Humidification Tower in an Evaporative Gas Turbine Cycle Pilot Plant. In *Proceedings of ASME Turbo Expo 2002*, Amsterdam, The Netherlands, June 3–6, 2002. ASME Paper No. 2002-GT-30127.
- [66] E. Macchi, S. Consonni, G. Lozza, and P. Chiesa. An Assessment of the Thermodynamic Performance of Mixed Gas–Steam Cycles: Part A — Intercooled and Steam-Injected Cycles. *ASME J. Engng. Gas Turbines Power*, 117(3):489–498, 1995.
- [67] E. Macchi, S. Consonni, G. Lozza, and P. Chiesa. An Assessment of the Thermodynamic Performance of Mixed Gas–Steam Cycles: Part B — Water-Injected and HAT Cycles. *ASME J. Engng. Gas Turbines Power*, 117(3):499–508, 1995.
- [68] W.L. MacMillan. *Development of a Modular Type Computer Program for the Calculation of Gas Turbine Off Design Performance*. PhD Thesis, School of Mechanical Engineering, Cranfield Institute of Technology, 1974.

- [69] MAN Diesel. Two-stroke Low Speed Diesel Engines for Independent Power Producers and Captive Power Plants. On the WWW, accessed October 2012. URL http://mandieselturbo.com/files/news/files/16119/tech_paper_low_speed.pdf.
- [70] Norwegian Marine Technology Research Institute — MARINTEK. Study of Greenhouse Gas Emissions from Ships, March 2000. MT Report: MT00 A23-038, Trondheim, Norway.
- [71] Syed Mashruk. Sizing of Combined-Cycle Components and Advanced 3D Design of Marine Power Plants. MS Thesis, School of Engineering, Cranfield University, 2014.
- [72] S.J. McCarthy and I. Scott. The WR-21 Intercooled Recuperated Gas Turbine Engine — Operation and Integration into the Royal Navy Type 45 Destroyer Power System. In *Proceedings of ASME Turbo Expo 2002*, Amsterdam, The Netherlands, June 3–6, 2002. ASME Paper No. GT-2002-30266.
- [73] C.F. McDonald and D.G. Wilson. The Utilization of Recuperated and Regenerated Engine Cycles for High-efficiency Gas Turbines in the 21st Century. *Applied Thermal Engineering*, 16, no. 8–9, pp.:635–653, 1996.
- [74] M. Mucino. *CCGT Performance Simulation and Diagnostics for Operations Optimisation and Risk Management*. PhD Thesis, School of Engineering, Cranfield University, 2007.
- [75] J. Myśków and T. Borkowski. Marine Engine Exhaust Gas Emission Aftertreatment System Concept. *Journal of KONES Powertrain and Transport*, 18(4):307–315, 2011.
- [76] M. Nakhamkin, E.C. Swensen, J.M. Wilson, G. Gaul, and M. Polsky. The Cascaded Humidified Advanced Turbine (CHAT). *ASME J. Engng. Gas Turbines Power*, 118(3):565–571, 1996.
- [77] T. Paanu, S. Niemi, and P. Rantanen. Waste Heat Recovery — Bottoming Cycle Alternatives. In *Proceedings of the University of Vaasa*, Report 175, 2012.
- [78] Emanuele Pagone. *Advanced Low Carbon Power Systems — the Advanced Zero Emissions Power Plant*. MS Thesis, School of Engineering, Cranfield University, 2009.
- [79] J.R. Palmer. *The ‘Turbocode’ Scheme for the Programming of Thermodynamic Cycle Calculations on an Electronic Digital Computer*. PhD Thesis, College of Aeronautics, Cranfield, 1967.
- [80] E.L. Parsons. Development, fabrication and application of a primary surface gas turbine recuperator. S.A.E. Paper no. 851254, 1985.
- [81] Max Peters, Klaus Timmerhaus, and Ronald West. *Plant Design and Economics for Chemical Engineers*. McGraw-Hill, 2003.
- [82] A.L. Polyzakis, C. Koroneos, and G. Xydis. Optimum Gas Turbine Cycle for Combined Cycle Power Plant. *Energy Conversion and Management*, 49:551–563, 2008.
- [83] A.L. Polyzakis, C. Koroneos, G. Xydis, and A. Malkogianni. Energy Analysis of Brayton Combined Cycles. *International Journal of Exergy*, 5, no. 5–6, pp.:652–683, 2008.
- [84] A. Poullikkas. An Overview of Current and Future Sustainable Gas Turbine Technologies. *Renewable and Sustainable Energy Reviews*, 9:409–443, 2005.
- [85] H.R. Rahai and H. Hefazi. Emission Control Technologies for Oceangoing Vessels (OGVs). California Air Resources Board: Sacramento, CA 2008.
- [86] A.D. Rao. “Process for Producing Power”. U.S. Patent No. 4,829,763, May 16, 1989.
- [87] Oliver Rath. MT30 — the Heart of Future Integrated Power and Propulsion Systems. In *Proceedings of ASME Turbo Expo 2014*, Düsseldorf, Germany, June 16–20, 2014. ASME Paper No. GT2014-26799.

- [88] I.G. Rice. The Combined Reheat Gas Turbine/Steam Turbine Cycle: Part I — A Critical Analysis of the Combined Reheat Gas Turbine/Steam Turbine Cycle. *ASME J. Engng. Gas Turbines Power*, 102(1):35–41, 1980.
- [89] I.G. Rice. The Combined Reheat Gas Turbine/Steam Turbine Cycle: Part II — The LM 5000 Gas Generator Applied to the Combined Reheat Gas Turbine/Steam Turbine Cycle. *ASME J. Engng. Gas Turbines Power*, 102(1):42–49, 1980.
- [90] Rolls-Royce plc. Next Generation Turbine (NGT) System Study. Work performed under contract DE-AC26-00NT40852, submitted to the U.S. Department of Energy, Feb. 2002.
- [91] P.M. Rosén. *Evaporative Cycles — in Theory and in Practice*. PhD Thesis, Department of Heat and Power Engineering, LTH, Lund, 2000.
- [92] Giulia Rossato. Heat Exchanger Design for Commercial Marine Applications. MS Thesis, School of Engineering, Cranfield University, 2014.
- [93] Luis Sánchez de León. Unpublished STEAMOMATCH documentation. 2013.
- [94] Luis Sánchez de León, Pavlos K. Zachos, and Vassilios Pachidis. A Comparative Assessment of ‘Dry’ Gas Turbine Cycles for Marine Applications. In *Proceedings of ASME Turbo Expo 2013*, San Antonio, Texas, USA, June 3–7, 2013. ASME Paper No. GT2013-95321.
- [95] Y. Sanjay, O. Singh, and B.N. Prasad. Energy and Exergy Analysis of Steam Cooled Reheat Gas–Steam Combined Cycle. *Applied Thermal Engineering*, 27:2779–2790, 2007.
- [96] Y. Sanjay, O. Singh, and B.N. Prasad. Influence of Different Means of Turbine Blade Cooling on the Thermodynamic Performance of Combined Cycle. *Applied Thermal Engineering*, 28:2315–2326, 2008.
- [97] Y. Sanjay, O. Singh, and B.N. Prasad. Thermodynamic Modelling and Simulation of Advanced Combined Cycle for Performance Enhancement. *Proc. IMechE, Part A: J. Power and Energy*, 222:541–555, 2008.
- [98] R.K. Shah and D.P. Sekulić. *Fundamentals of Heat Exchanger Design*. John Wiley & Sons, 2003.
- [99] S.B. Shepard, T.L. Bowen, and J.M. Chiprich. Design and Development of the WR-21 Inter-cooled Recuperated (ICR) Marine Gas Turbine. *Journal of Engineering for Gas Turbines and Power*, 117:557–562, 1995.
- [100] A.A. Sirinoglou. Implementation of Variable Geometry for Gas Turbine Performance Simulation Turbomatch Improvement. MS Thesis, School of Mechanical Engineering, Cranfield Institute of Technology, 1992.
- [101] K.O. Skjølsvik. Emission from ships: The 4th Stanley Gray lecture. *International Maritime Technology*, 112, no. 3–4, pp.:95–102, 2000.
- [102] Eric M. Smith. *Advances in Thermal Design of Heat Exchangers. A Numerical Approach: Direct-sizing, Step-wise Rating, and Transients*. John Wiley & Sons, 2005.
- [103] Tatsuo Takaishi, Akira Numata, Ryouji Nakano, and Katsuhiko Sakaguchi. Approach to High Efficiency Diesel and Gas Engines. On the WWW, accessed October 2012. URL <http://www.mhi.co.jp/technology/pdf/e451/e451021.pdf>.
- [104] Takuya Takeda, Hidefumi Araki, Yasushi Iwai, Tetsuro Morisaki, and Kazuhiko Sato. Test Results of 40MW-Class Advanced Humid Air Turbine and Exhaust Gas Water Recovery System. In *Proceedings of ASME Turbo Expo 2014*, Düsseldorf, Germany, June 16–20, 2014. ASME Paper No. GT2014-27281.
- [105] M. Thern. *Humidification Processes in Gas Turbine Cycles*. PhD Thesis, Department of Energy Sciences, Faculty of Engineering, LTH, Lund, 2005.

- [106] G.P. Towler and R.K. Sinnott. *Chemical Engineering Design: Principles, Practice, and Economics of Plant and Process Design*. Elsevier Science, 2013.
- [107] E. Tsoudis. *Technoeconomic Environmental and Risk Analysis of Marine Gas Turbine Power Plants*. PhD Thesis, School of Engineering, Cranfield University, 2008.
- [108] U.S. Environmental Protection Agency. Ocean Vessels and Large Ships. On the WWW, accessed between December 2011 and March 2012. URL <http://www.epa.gov/otaq/oceanvessels.htm>.
- [109] F. Van den Hout. Gas Turbine Performance Simulation Improvements to the Turbomatch Scheme. MS Thesis, School of Mechanical Engineering, Cranfield Institute of Technology, 1991.
- [110] Philip Walsh and Paul Fletcher. *Gas Turbine Performance*. John Wiley & Sons, 2nd ed. 2004.
- [111] M.E. Ward and L. Holman. Primary surface recuperator for high performance prime mover. S.A.E. Paper no. 920150, 1992.
- [112] Stewart Wilmshurst. Rolls-Royce MT30 Design, Certification, Launch and Growth. In *Proceedings of the International Gas Turbine Congress 2003*, Tokyo, Japan, November 2–7, 2003. Paper No. IGTC2003Tokyo OS-204.
- [113] A. Wunsch. Combined Gas/Steam Turbine Power Stations — the Present State of Progress and Future Developments. *Brown Boveri Review*, 65(10):646–655, 1978.
- [114] Manabu Yagi, Hidefumi Araki, Hisato Tagawa, Tomomi Koganezawa, Chihiro Myoren, and Takuya Takeda. Progress of the 40MW-Class Advanced Humid Air Turbine Tests. In *Proceedings of ASME Turbo Expo 2013*, San Antonio, Texas, USA, June 3–7, 2013. ASME Paper No. GT2013-95172.
- [115] Bo Yu, Hiroyuki Ozoe, and Stuart W. Churchill. The Characteristics of Fully Developed Turbulent Convection in a Round Tube. *Chemical Engineering Science*, 56:1781–1800, 2001.

Part III

Appendices

Appendix A

Design-Space Exploration Sample

Figure A.1 shows the typical overall thermal efficiency contours for the whole design space of an intercooled-only topping-cycle gas turbine featuring an intercooled-ABC. Given the four design parameters (i.e. independent variables) available, *parallel coordinates* have been employed for its representation: each *sub*-plot consists of thermal efficiency contours against bottoming cycle's OPRs in the 'x' axis and bottoming cycle's *relative*-LPPRs in the 'y' axis, while these *sub*-plots have been arranged so that the 'x' *super*-axis represents topping cycle's OPRs and the 'y' *super*-axis represents topping cycle's *relative*-LPPRs. The *relative*-LPPR is simply a non-dimensionalisation of the actual LPPR; by which $LPPR = OPR$ and $HPPR = 1.00$ if *relative*-LPPR = 1.00, and $LPPR = 1.00$ and $HPPR = OPR$ if *relative*-LPPR = 0.00.

This plot can be used to better understand the influence of each design parameter within the whole design space for such a power plant. Empty (white) areas in each of the *sub*-plots of fig. A.1 represent the regions of the design space in which waste-heat recovery from the topping cycle's exhaust by means of an ABC is not feasible. In these regions, the combination of topping cycle's EGT and bottoming cycle's CDT does not allow for additional power extraction at the expansion side of the bottoming cycle. It is apparent that, as the topping-cycle OPR increases, empty areas within each *sub*-plot become greater and greater, as a result of the topping-cycle EGT reduction, which makes the bottoming-cycle TIT to drop as well and therefore limits the bottoming-cycle OPR at which power extraction from the bottoming cycle is feasible.

Each of the *sub*-plots provides additional information on how to optimise the bottoming cycle for a particular topping-cycle design point. Figure A.2 consists of only the bottom-left-most *sub*-plot extracted from fig. A.1. For this particular topping-cycle fixed design point —with topping-cycle OPR = 12:1 and topping-cycle *relative*-LPPR = 0.25—, there are three regions within the bottoming cycle's design space worth noticing:

- Firstly, the empty region for both high bottoming-cycle OPR and high bottoming-cycle *relative*-LPPR, where the combination of these two yields too high of a bottoming-cycle LP-compressor required work so as to be able to extract any amount of useful power at the expansion side of the bottoming cycle.
- Secondly, the high-overall-thermal-efficiency region at very low bottoming-cycle OPRs ($\sim 12:1$ and below), in which the high bottoming-cycle EGT allows for waste-heat recuperation from the bottoming-cycle exhaust gases into the topping cycle's compressor outlet —this region disappears for higher topping-cycle OPRs; which result in higher topping-cycle CDTs, eventually greater than the bottoming-cycle EGTs available—.
- Lastly, the rest of the design space, in which a local optimum for bottoming-cycle OPR ($\sim 15:1$) and bottoming-cycle *relative*-LPPR (~ 0.5 ; that is, both compressors splitting the compression work evenly) appears, where maximum power output from the bottoming cycle can be extracted, thus maximising the overall thermal efficiency of the power plant. While this optimal bottoming-cycle OPR is dependent on the bottoming cycle's TIT, the even PR split between both bottoming cycle's compressors in this optimal region achieves a fine balance between lower bottoming-cycle

CDTs —that allow for better waste-heat recovery from the topping cycle’s exhaust gases— and lower bottoming-cycle HP-compressor required work —that allows for more useful work to be extracted from the free power turbine in the expansion side of the bottoming cycle—.

Finally, taking out of each *sub*-plot from fig. A.1 the absolute maximum in overall thermal efficiency, the design-space for the topping cycle *with optimised bottoming cycle* can be built up, as shown in fig. A.3. In this plot, thermal efficiency contours against topping cycle’s OPRs in the ‘x’ axis and topping cycle’s *relative*-LPPRs in the ‘y’ axis are represented. It is important to highlight that each point from the contours in fig. A.3 corresponds to a single topping-cycle design point with its *optimised* bottoming-cycle configuration: these contours provide information on the *topping-cycle design parameters* that yield maximum overall thermal efficiency for a given power plant. The *bottoming-cycle design parameters* that optimise this particular topping-cycle design point must be read from the corresponding *sub*-plot in fig. A.1, though.

Two well-differentiated areas can be highlighted from fig. A.3, namely:

- Firstly and foremost, there is a local maximum at topping-cycle OPR $\sim 15:1$ and topping-cycle *relative*-LPPR ~ 0.5 . This corresponds to the optimum design point within the region of the design space in which waste-heat recuperation from the bottoming-cycle exhaust gases into the topping cycle’s compressor outlet is actually feasible. For each topping-cycle design point that falls within this region and its surroundings, there exists an *optimised* ABC that enables waste-heat recuperation from the bottoming-cycle exhaust gases and is able to outperform the non-recuperated optimal bottoming cycle in terms of overall thermal efficiency of the power plant.
- Second but most importantly, an absolute maximum appears at the bottom-right corner of the design space. In this region, bottoming cycle’s waste-heat recuperation is not feasible, and thus the increased overall thermal efficiency of the power plant obeys only to an increased topping-cycle thermal efficiency at extremely high topping-cycle OPRs and extremely low topping-cycle *relative*-LPPRs, as it would correspond to any intercooled non-recuperated gas turbine (see fig. 2.4). Effectively, the most noticeable effect on the overall thermal efficiency of the power plant comes from the *core* (i.e. topping-cycle) gas turbine, while the bottoming-cycle is aimed solely at extracting some additional power from the *core*’s waste heat when this is possible. The fact that the non-recuperated maximum of the bottom-right corner in fig. A.3 outperforms the local maximum from the recuperated region suggests that, for this sort of power plant, penalising the topping-cycle gas turbine in order to improve the performance of the bottoming cycle does not pay off, and the emphasis should be placed in maximising the topping cycle’s thermal efficiency first.

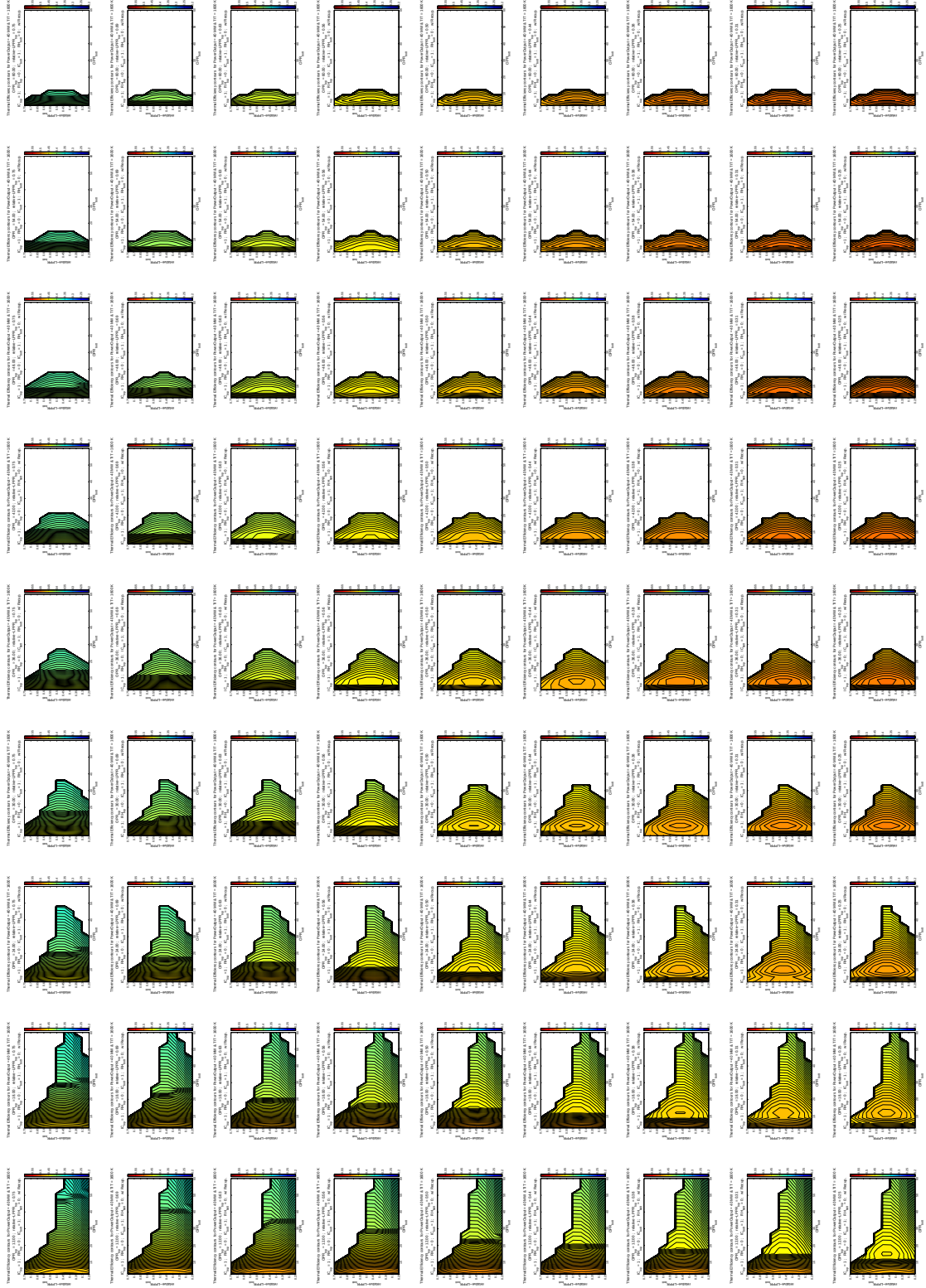


Figure A.1: *Design-space representation in parallel coordinates for the intercooled-only topping-cycle gas turbine with an intercooled-ABC*

Thermal Efficiency contours for PowerOutput = 40 MW & TIT = 1600 K

$OPR_{top} = 12.00$; $relative-LPPR_{top} = 0.25$

$IC_{top} = 1$; $RH_{top} = 0$; $IC_{bott} = 1$; $RH_{bott} = 0$; w/ Recup.

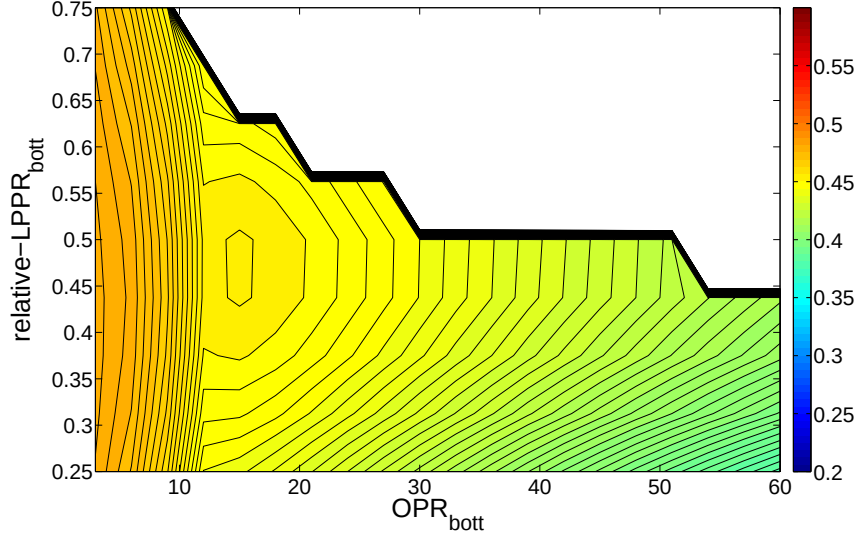


Figure A.2: Thermal efficiency contours for the intercooled-ABC with a constant intercooled-only topping-cycle gas turbine

Thermal Efficiency contours for PowerOutput = 40 MW & TIT = 1600 K

$IC_{top} = 1$; $RH_{top} = 0$; $IC_{bott} = 1$; $RH_{bott} = 0$; w/ Recup.

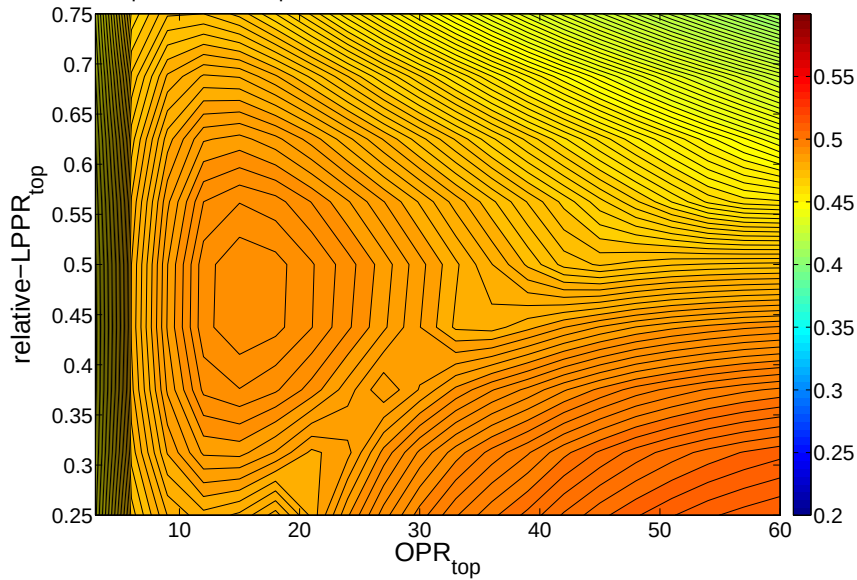


Figure A.3: Thermal efficiency contours for the intercooled-only topping-cycle gas turbine with an optimised intercooled-ABC

Appendix B

Humidifier modelling & verification

B.1 Humidifier Modelling

The system of equations that solves the performances of the saturator column at design point is given by applying eqs. 6.2–6.4 described in Section 6.1.1 to the predefined stations of the humidifier. Pressure throughout the whole column is assumed to remain constant, and equal to the pressure of the inlet air, p_1 .

Station “0”, Air is saturated by definition of the ‘pinch’ conditions (eq. B.1). Air thermodynamic properties (T_0 , ω_0) are the unknowns of the system, but it is known, by definition, that water temperature equals air temperature in the ideal saturator, whilst ΔT (assumed to be of 5 K in these calculations, as a typical value [9]) must be defined for real saturators, so that $T_{w0} = T_0 + \Delta T$. Equation B.2, after [9], is the way that the mass invariant can be rewritten for this station. Similarly, eq. B.3 results from applying the enthalpy invariant equation to this station.

$$\omega_0 - \frac{R_a}{R_v} \frac{p_{\text{sat}}(T_0)}{p_1 - p_{\text{sat}}(T_0)} = 0 \quad (\text{B.1})$$

$$M^+ - \frac{c_{pa}}{c_{pw}} - \omega_0 \left(\frac{c_{pv} - c_{pw}}{c_{pw}} + \frac{p_1}{p_1 - p_{\text{sat}}(T_0)} \frac{R_v}{c_{pw}} \left(\frac{h_{fg}(T_0)}{R_v T_0} \right)^2 \right) = 0 \quad (\text{B.2})$$

$$\begin{aligned} H^+ - h_a(T_0) - \omega_0 h_{fg}(T_0) + \\ + \omega_0 [h_w(T_0 + \Delta T) - h_w(T_0)] + \\ + M^+ h_w(T_0 + \Delta T) = 0 \end{aligned} \quad (\text{B.3})$$

Station “1”, Air thermodynamic properties (p_1 , m_{g1} , T_{g1} , ω_1 and, hence, m_a) are known from the compressor/aftercooler outlet. Water conditions (m_{w1} , T_{w1}) are the unknowns and must be calculated. Equation B.4 results from applying the mass invariant of the humidifier to the air inlet/water outlet station. Similarly, eq. B.5 comes from the enthalpy invariant equation, applied to this station.

$$M^+ + \omega_1 - \frac{m_{w1}}{m_a} = 0 \quad (\text{B.4})$$

$$\begin{aligned} H^+ - h_a(T_{g1}) - \omega_1 h_{fg}(T_{g1}) + \\ + \omega_1 [h_w(T_{w1}) - h_w(T_{g1})] + \\ + M^+ h_w(T_{w1}) = 0 \end{aligned} \quad (\text{B.5})$$

Station “2”, Water thermodynamic properties (m_{w2} , T_{w2}) are known from the water calculations of the heat exchangers implemented within the cycles. Air conditions (m_{g2} , T_{g2} , ω_2) are the

unknowns of the system, but it is known that the air will be completely saturated (eq. B.6), as per hypothesis made. Equation B.7 results from the mass invariant equation applied to the water inlet/air outlet station of the humidifier. Similarly, eq. B.8 comes from applying the enthalpy invariant to the same station.

$$\omega_2 - \frac{R_a}{R_v} \frac{p_{\text{sat}}(T_{g2})}{p_1 - p_{\text{sat}}(T_{g2})} = 0 \quad (\text{B.6})$$

$$M^+ + \omega_2 - \frac{m_{w2}}{m_a} = 0 \quad (\text{B.7})$$

$$\begin{aligned} H^+ - h_a(T_{g2}) - \omega_2 h_{fg}(T_{g2}) + \\ + \omega_2 [h_w(T_{w2}) - h_w(T_{g2})] + \\ + M^+ h_w(T_{w2}) = 0 \end{aligned} \quad (\text{B.8})$$

Equations B.1–B.3 and B.6–B.8 form a system of six equations with six unknowns that permits the solution of the ‘pinch’ conditions (T_0 , ω_0), the air outlet conditions (T_{g2} , ω_2 and, hence, m_{g2}), and the invariants of the humidifier (M^+ , H^+). After this is done, eqs. B.4–B.5 allow for the solution of the water outlet conditions (m_{w1} , T_{w1}) to be found.

Essentially, this system of equations is identical to the equations governing any condensation process. So, after the solution of the system is found, additional checks have to be performed to ensure that the saturator column is actually working as a humidifier (absorbing water into the air) and it is indeed a feasible design point. Particularly, it must be ensured that:

- $\omega_2 > \omega_1$; water must be absorbed into the air.
- $T_{g2} < T_{w2}$; air stream cannot be heated up above the water inlet temperature.
- $m_{w1} > 0$; water must actually exit the humidifier at the bottom section of the column.
- $T_{w1} > T_{AS}$; water cannot be cooled down below the *adiabatic saturation* temperature of the air inlet stream. This *adiabatic saturation* temperature, T_{AS} , can readily be calculated knowing the air inlet conditions (see, for instance, [10]) only.

If these four statements are verified, the humidifier is feasible and the model is valid for that particular design point.

B.2 Humidifier Verification

Verification of this model against data found in [6, 7] has been carried out. Saturators from the cycles studied by Ågren et al. were isolated from their power plants, their water and air inlet conditions taken as the inputs for the presented humidifier model, and the outlet thermodynamic properties of the air and water streams were compared. Tables B.1 and B.2 present the input data to the model, the inlet conditions both for the air and for the water streams. Tables B.3 and B.4 present the comparison between the air outlet conditions as calculated by this model and the air outlet conditions as calculated by Ågren et al. Errors are below 1.25% in all the cases, which satisfies sufficiently the precision needs for this first-approach study. Tables B.5 and B.6 present the comparison between the water outlet conditions as calculated by this model and the water outlet conditions as calculated by Ågren et al. Relative errors fall once again below 1.25%, which is considered to be sufficient accuracy for this research.

Table B.1: *Air inlet conditions*

Case	p_1 (bar)	m_{g1} (kg/s)	T_{g1} (K)	ω_1 (n.d.)
125-1/I-10	35.28	104.03	394.33	0.0
125-1/I-02	35.28	21.49	414.41	0.0
125-1/II-10	20.27	92.49	387.25	0.0
125-1/II-04	20.27	37.08	403.06	0.0
125-1/II-02	20.27	18.58	421.10	0.0

Table B.2: *Water inlet conditions*

Case	m_{w2} (kg/s)	T_{w2} (K)
125-1/I-10	59.52	506.11
125-1/I-02	37.61	506.13
125-1/II-10	92.49	476.15
125-1/II-04	73.42	476.15
125-1/II-02	66.51	476.17

Table B.3: *Air outlet mass flow, m_{g2} (kg/s)*

Case	Model	Data	Error
125-1/I-10	116.85	116.95	0.084%
125-1/I-02	28.87	29.13	0.885%
125-1/II-10	109.89	109.89	0.002%
125-1/II-04	49.46	49.70	0.483%
125-1/II-02	28.15	28.44	1.011%

Table B.4: *Air outlet temperature, T_{g2} (K)*

Case	Model	Data	Error
125-1/I-10	430.87	432.94	0.479%
125-1/I-02	463.15	464.88	0.371%
125-1/II-10	422.72	423.83	0.262%
125-1/II-04	438.55	439.70	0.262%
125-1/II-02	449.35	450.46	0.247%

Table B.5: *Water outlet mass flow, m_{w1} (kg/s)*

Case	Model	Data	Error
125-1/I-10	46.70	46.60	0.212%
125-1/I-02	30.23	29.96	0.894%
125-1/II-10	75.09	75.10	0.016%
125-1/II-04	61.04	60.80	0.395%
125-1/II-02	56.94	56.65	0.508%

Table B.6: *Water outlet temperature, T_{w1} (K)*

Case	Model	Data	Error
125-1/I-10	370.04	374.55	1.205%
125-1/I-02	392.83	394.40	0.399%
125-1/II-10	360.88	364.40	0.966%
125-1/II-04	378.55	379.90	0.354%
125-1/II-02	397.03	397.60	0.144%

Appendix C

Single-Objective Optimisation of Downselected Cycles

The three downselected configurations will be optimised by means of a GA in terms of design-point thermal efficiency alone —SOO—, and the optimised cycles will be “frozen” before undertaking the next stage of this research, in which their size, mass and acquisition costs will be assessed keeping the thermodynamics constant.

Results from this optimisation study are summarised —along with small modifications introduced into the models to allow for: (i) greater flexibility in the parameters that can be optimised, and (ii) increased accuracy in the thermal efficiency predictions— herein.

C.1 Evaporative Gas Turbines

The schematics of the optimised EvGT power plant can be found in fig. C.1, together with the station numbering employed to illustrate pressure and temperature profiles of this thermodynamic cycle. The models analysed consist now in a *single-spool* arrangement, in which all the turbomachinery components of the gas turbine are coupled with an electric generator in the same shaft. Since these analyses deal only with the thermodynamics of the power plant, no further considerations about the rotational speed of this shaft are made, but the only consequence of analysing a *single-spool* arrangement is that the equations for work balance between compressors and turbines in the same shaft are effectively removed from the system, hence relaxing the constraints imposed by the mechanical coupling of turbomachinery components in the positions of the intercooler and reheater. These can now be placed at any point, independently from each other, along the compression or expansion side of the gas turbine, respectively.

In addition to this, the effectiveness of all the heat exchangers present in the cycle are now independent variables that can be optimised, for which only a ‘technological limit’ has been imposed as 90% maximum effectiveness for gas–gas heat exchangers (i.e. the recuperator) and 95% maximum effectiveness for gas–liquid heat exchangers (i.e. the IC, AC and EC). Heat exchangers with lower effectiveness than these ‘technological limits’ are considered feasible for design, while higher effectiveness are considered unattainable with modern heat exchangers of *reasonable* size and cost.

Similarly, the water mass flow circulating through the IC, AC and EC, and hence the existing trade-off between the water mass flow injected into the humidifier and its injection temperature, is now controlled independently for each of the heat exchangers, rather than by a single parameter as detailed in Section 6.1.1, allowing for greater flexibility in the optimisation of the thermodynamic cycle.

Lastly, the temperatures at the outlet of the gas–water heat exchangers (namely: the IC, AC and EC), as well as the temperature at the outlet of the reheater, are now free design parameters that can be varied and do not always take their most extreme value (as discussed in Section 2.2.1), to assess their possible impact in the cycle’s performance.

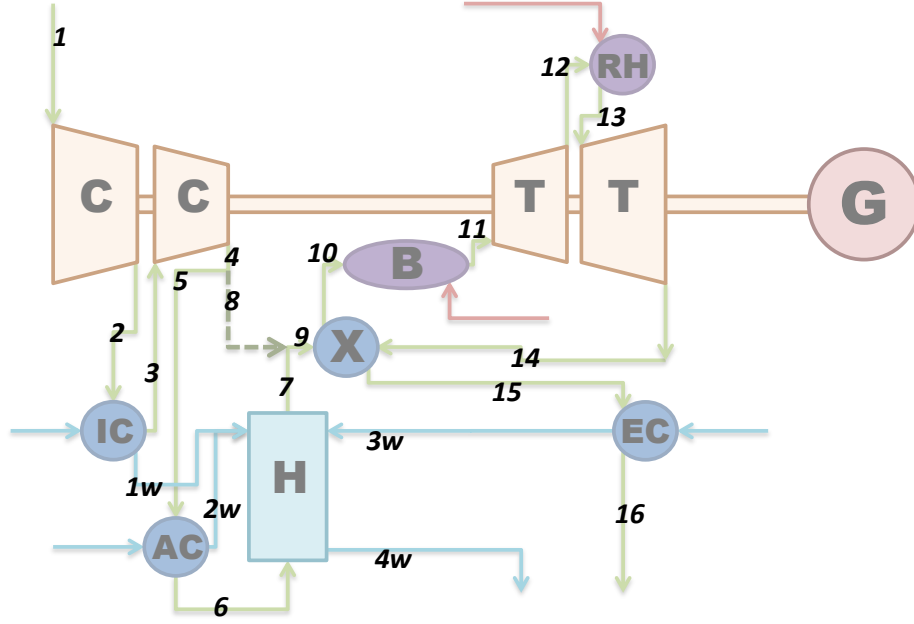


Figure C.1: Station numbering of the single-spool evaporative gas turbine

C.1.1 The IC-RH-R EvGT cycle

The main conclusions for the optimisation studies of the IC-RH-R EvGT power plant are summarised below:

- Optimal LPPR $\sim 7.36:1$ and optimal HPPR $\sim 6.77:1$; which results in an optimum OPR of around $49.8:1$, with slightly more compression work carried out in the LP spool than in the HP spool.
- The optimum position of the reheater corresponds to the supply of additional fuel at a pressure of ~ 11.8 atm, as opposed to the reheating pressure of ~ 16.2 atm that corresponds to the outlet of the compressor-driving turbines and to the inlet of the free power turbine in the dual-shaft arrangement described in Section 6.1. This means that the optimal position of the reheater for maximum thermal efficiency in such a configuration would indeed feature inter-stage reheating at some point in the middle of the free power turbine.
- The best performances of the power plant are achieved when the whole mass flow of air delivered by the compression system is circulated through the AC and through the humidification tower, and thus the available by-pass at the outlet of the HP compressor to stream the compressed air directly into the recuperator is not necessary, at least at design-point operation.
- The ‘technological limit’ of 90% maximum effectiveness constrains the design of the recuperator, that cannot get above this limit. The maximum achievable thermal efficiency of this power plant would occur for an even higher recuperator effectiveness, should this be attainable.
- Maximum thermal efficiency is achieved by means of always having the IC, AC and EC removing as much enthalpy from the hot gases as possible (that is, taking the 95% effectiveness ‘technological limit’ imposed), and by means of raising the temperature of the main gas path

back to the 1600 K imposed as the TIT limitation in the reheater as well. This is in agreement with the discussion from the ‘dry’ cycle analyses in Section 2.2.1.

- To achieve the best possible performances, the water generated at the IC, AC and EC must always be heated to the maximum achievable temperature. This means that the bare minimum of water mass flow, just enough to recover the necessary amount of enthalpy from the hot gases with 95% effectiveness, will be circulated through these gas–water heat exchangers.

With all these, the pressure and temperature profiles for this *optimised* IC-RH-R EvGT thermodynamic cycle are detailed in tab. C.1, following the station numbering from fig. C.1.

Table C.1: *Pressure and temperature profiles for the optimised IC-RH-R EvGT power plant (mass flows for 40 MW power output)*

Air–Gas conditions				
Station	Pressure (atm)	Temperature (K)	Mass flow (kg/s)	Water-to-Air Ratio
1	1.0	288.15	26.84	< 0.01
2	7.36	540.7	26.84	< 0.01
3	6.81	300.9	26.84	< 0.01
4	46.1	549.3	26.84	< 0.01
5	46.1	549.3	26.84	< 0.01
6	42.6	302.4	26.84	< 0.01
7	40.5	479.0	37.81	0.42
8	—	—	—	—
9	40.5	479.0	37.81	0.42
10	37.5	993.2	37.81	0.42
11	35.6	1600.	38.72	0.42
12	11.8	1303.	38.72	0.42
13	11.2	1600.	39.17	0.42
14	1.25	1050.	39.17	0.42
15	1.16	573.1	39.17	0.42
16	1.07	302.5	39.17	0.42

Water conditions		
Station	Temperature (K)	Mass flow (kg/s)
1w	515.5	6.883
2w	521.8	6.810
3w	521.8	41.04
4w	390.6	43.63

C.1.2 The IC-R EvGT cycle

The main conclusions for the optimisation studies of the IC-R EvGT power plant are summarised below:

- Optimal LPPR $\sim 6.44:1$ and optimal HPPR $\sim 5.55:1$; which results in an optimum OPR of around $35.7:1$, with slightly more compression work carried out in the LP spool than in the HP spool.
- The best performances of the power plant are achieved when the whole mass flow of air delivered by the compression system is circulated through the AC and through the humidification tower, and thus the available by-pass at the outlet of the HP compressor to stream the compressed air directly into the recuperator is not necessary, at least at design-point operation.

- An optimum for the recuperator effectiveness appears at around 88.9%. Although recuperators with slightly higher effectiveness are allowed, more enthalpy recovered in the recuperator means less enthalpy available to raise hot water at the EC, and the best possible trade-off between the two must be found.
- Maximum thermal efficiency is achieved by means of always having the IC, AC and EC removing as much enthalpy from the hot gases as possible (that is, taking the 95% effectiveness ‘technological limit’ imposed). This is in agreement with the discussion from the ‘dry’ cycle analyses in Section 2.2.1.
- To achieve the best possible performances, the water generated at the IC, AC and EC must always be heated to the maximum achievable temperature. This means that the bare minimum of water mass flow, just enough to recover the necessary amount of enthalpy from the hot gases with 95% effectiveness, will be circulated through these gas–water heat exchangers.

With all these, the pressure and temperature profiles for this *optimised* IC-R EvGT thermodynamic cycle are detailed in tab. C.2, following the station numbering from fig. C.1 (except for the fact that the IC-R EvGT does not feature a reheater, and hence stations 12 & 13 are not defined for this configuration).

Table C.2: *Pressure and temperature profiles for the optimised IC-R EvGT power plant (mass flows for 40 MW power output)*

Air–Gas conditions				
Station	Pressure (atm)	Temperature (K)	Mass flow (kg/s)	Water-to-Air Ratio
1	1.0	288.15	36.50	< 0.01
2	6.44	518.8	36.50	< 0.01
3	5.96	299.7	36.50	< 0.01
4	33.1	514.9	36.50	< 0.01
5	33.1	514.9	36.50	< 0.01
6	30.6	299.5	36.50	< 0.01
7	29.1	458.9	49.37	0.36
8	—	—	—	—
9	29.1	458.9	49.37	0.36
10	26.9	834.6	49.37	0.36
11	25.6	1600.	50.82	0.36
14	1.25	881.8	50.82	0.36
15	1.16	526.8	50.82	0.36
16	1.07	300.1	50.82	0.36

Water conditions		
Station	Temperature (K)	Mass flow (kg/s)
1w	495.7	9.344
2w	492.2	9.343
3w	503.0	50.58
4w	376.3	56.28

C.2 Combined-Cycle Gas Turbines

The schematics of the optimised RH-R CCGT power plant can be found in fig. C.2, together with the station numbering employed to illustrate pressure and temperature profiles of this thermodynamic cycle. The models analysed consist now in a *single-spool* arrangement, in which all the turbomachinery

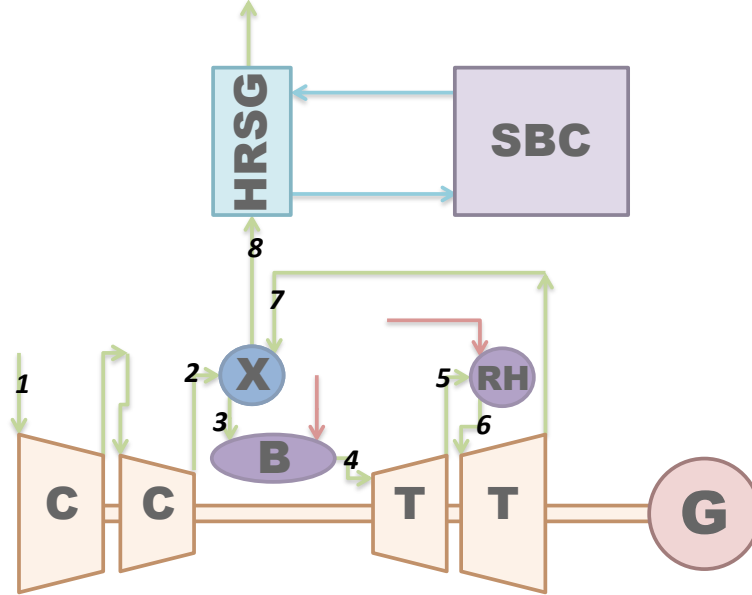


Figure C.2: Station numbering of the single-shaft combined-cycle gas turbine

components of the gas turbine are coupled with an electric generator in the same shaft. This allows for the unconstrained placement of the reheater in its optimum position, to supply the fuel addition at the optimum pressure for maximum overall thermal efficiency of the power plant.

Additionally, the effectiveness of the recuperator implemented within this cycle is now an independent variable that can be optimised, for which only a ‘technological limit’ has been imposed as 90% maximum effectiveness. Recuperators with lower effectiveness than this ‘technological limit’ are considered feasible for design, while higher effectiveness are considered unattainable with modern heat exchangers of *reasonable* size and cost.

The main conclusions for the optimisation studies of the RH-R CCGT power plant are summarised below:

- Optimal OPR for maximum thermal efficiency is $\sim 29.7:1$. The lack of an intercooler makes the PR split between the LP and the HP compressors irrelevant, from the thermodynamic point of view.
- The optimum position of the reheater corresponds to the supply of additional fuel at a pressure of ~ 6.3 atm along the expansion side of the gas turbine. This, as opposed to what happened to the IC-RH-R EvGT discussed previously, means that the reheater would be placed, in the case of a dual-shaft arrangement similar to the one presented in Section 4.1, slightly upstream of the free power turbine inlet, at some point in the middle but close to the outlet of the LP compressor-driving turbine.
- The ‘technological limit’ of 90% maximum effectiveness constrains the design of the recuperator, that cannot get above this limit. The maximum achievable thermal efficiency of this power plant would occur for an even higher recuperator effectiveness, should this be attainable.
- Maximum overall thermal efficiency of the power plant is achieved by means of implementing the SBC with LP steam pressure of around 3.5 bar and HP steam pressure slightly above

118 bar. Around 82.3% of the total steam mass flow is streamed into the HP loop, whilst only the remaining 17.7% is circulated through the LP loop.

With all these, the pressure and temperature profiles for this *optimised* RH-R CCGT thermodynamic cycle are detailed in tab. C.3. Similarly, the pressure and temperature profiles for the *optimum* dual-pressure SBC are detailed in tab. C.4. The station numbering corresponding to the water-steam closed loop, as well as the gas stations along the HRSG, are depicted in fig. C.3. A notional $T - s$ diagram for the SBC is also included in fig. C.4, just for reference purposes.

Table C.3: *Pressure and temperature profiles for the optimised RH-R CCGT power plant (mass flows for 40 MW CCPP power output)*

Station	Air-Gas conditions			
	Pressure (atm)	Temperature (K)	Mass flow (kg/s)	Water-to-Air Ratio
1	1.0	288.15	52.71	< 0.01
2	29.7	821.6	52.71	< 0.01
3	27.5	1117.	52.71	< 0.01
4	26.1	1600.	53.49	< 0.01
5	6.28	1185.	53.49	< 0.01
6	5.97	1600.	54.17	< 0.01
7	1.25	1149.	54.17	< 0.01
8	1.16	872.7	54.17	< 0.01

Table C.4: *Pressure and temperature profiles for the optimised dual-pressure SBC (mass flows for 40 MW CCGP power output)*

Water–Steam conditions				
Station	Pressure (kPa)	Temperature (K)	Mass flow (kg/s)	Steam quality
16	5.00	306.03	9.56	0.0
1	349.1	306.1	9.56	0.0
2	349.1	409.9	9.56	0.0
3	349.1	409.9	1.69	0.0
4	349.1	411.9	1.69	0.0
5	349.1	411.9	1.69	1.0
6	349.1	468.6	1.69	1.0
7	349.1	409.9	7.87	0.0
8	11 840.	411.6	7.87	0.0
9	11 840.	594.8	7.87	0.0
10	11 840.	596.8	7.87	0.0
11	11 840.	596.8	7.87	1.0
12	11 840.	850.0	7.87	1.0
13	349.1	416.0	7.87	1.0
14	349.1	425.0	9.56	1.0
15	5.00	306.03	9.56	0.8511
16	5.00	306.03	9.56	0.0

Gas conditions		
Station	Temperature (K)	Mass flow (kg/s)
1g	872.7	54.17
2g	764.5	54.17
3g	606.8	54.17
4g	488.7	54.17
5g	485.2	54.17
6g	421.9	54.17
7g	348.4	54.17

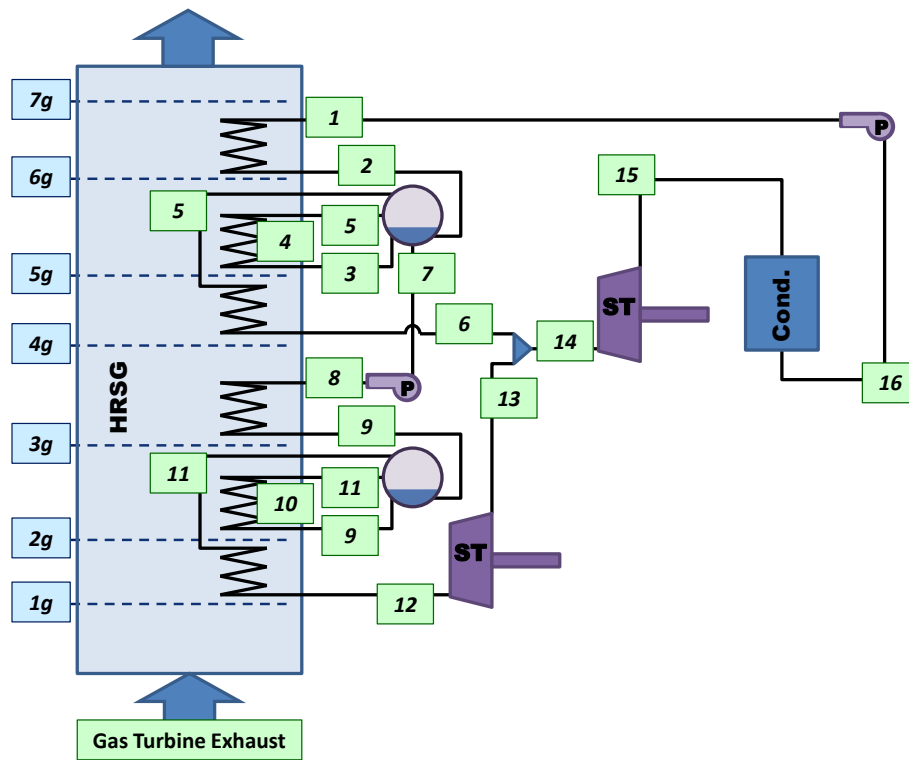


Figure C.3: Station numbering of the dual-pressure steam bottoming cycle

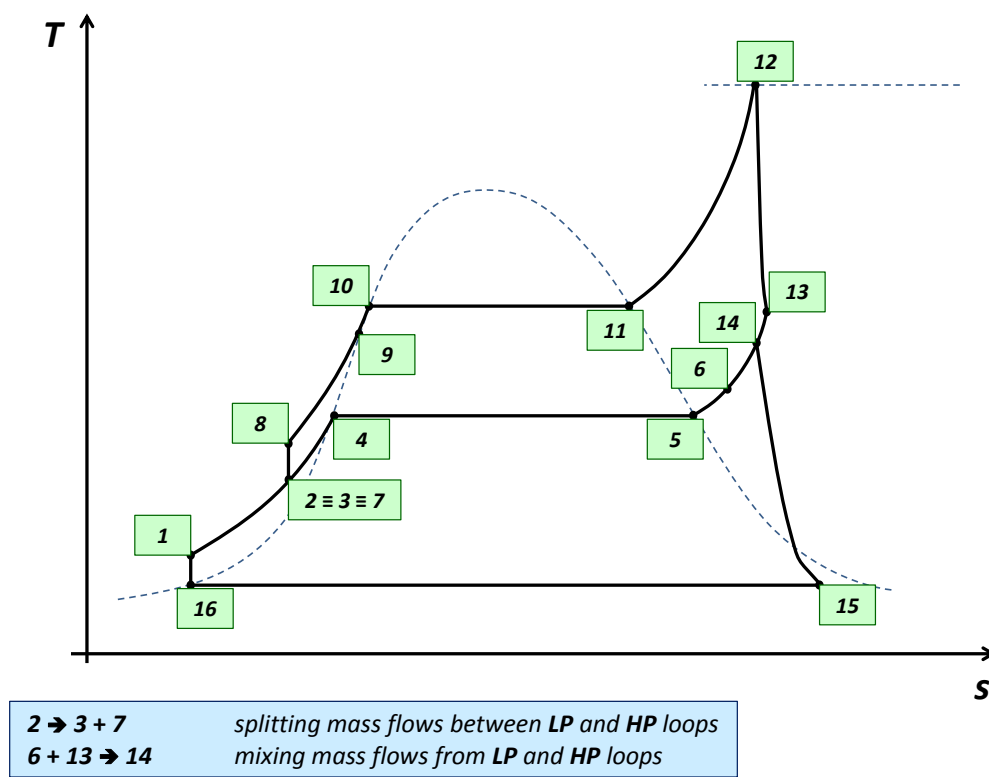


Figure C.4: Notional $T - s$ diagram for a dual-pressure steam bottoming cycle

Appendix D

Details on the Heat Transfer Calculations

The goal is to generate detailed geometries (subject to the specified type of heat exchanger) that would fulfil their required duty, by means of matching the resulting *conductance* of a single pass of the heat exchanger, UA , with the target value gotten from imposing the previously optimised thermodynamics (as discussed in Section 8.1).

Heat transfer theory establishes that the *resistance* of any heat transfer process can be solved by adding up the corresponding resistances of the different media involved—in this case: the hot and the cold fluids, and the separating wall of the primary surface of the heat exchanger—, as shown in eq. D.1, in which the contribution of each of them is broken down [98]. It is worth noticing, however, that *fouling* effects have not been taken into account in eq. D.1, and their negative influence on the heat transfer process has been completely neglected.

$$\frac{1}{UA} = \left(\frac{1}{h \eta_O A_{\text{wet}}} \right)_c + R'_w + \left(\frac{1}{h \eta_O A_{\text{wet}}} \right)_h \quad (\text{D.1})$$

The resistance to the heat transfer process from each of the fluid streams come, in the most generic case, from h , the heat transfer coefficient; η_O , the overall heat transfer efficiency of the extended surfaces (if any); and A , the heat transfer surface area of that particular side.

The term R'_w is the resistance to heat transfer due to the wall conductivity, that can be expressed as per eq. D.2, although this is usually the smallest of the three terms, negligible in many cases [98].

$$R'_w = \frac{\delta_w}{k_w A_w} \quad (\text{D.2})$$

Here, δ_w is the wall thickness, k_w the metal thermal conductivity, and A_w the overall heat transfer area of the primary surface.

Once eq. D.1 is established, and the target UA is known, the means of calculating the heat transfer coefficient h , the overall heat transfer efficiency η_O , and the heat transfer surface area A of each of the streams will depend on the type of heat exchanger being considered.

D.1 Tube & Fin Heat Exchangers

The *decision variables* chosen for the design of *tube* & *fin* heat exchangers are:

- $u_{m,c}$ and $u_{m,h}$, mean velocities of the cold and hot streams, respectively. Following Franco & Giannini [44] guidelines, the mean velocity is *constrained* according to the phase of the fluid:
 - ◊ $0.5 \text{ m/s} \leq u_m \leq 2.5 \text{ m/s}$ for liquid-phase streams;
 - ◊ $0.5 \text{ m/s} \leq u_m \leq 4.5 \text{ m/s}$ for *phase change* streams; and
 - ◊ $8 \text{ m/s} \leq u_m \leq 40 \text{ m/s}$ for gas-phase streams.

- X_l , the pitch between adjacent rows within the same pass of the heat exchanger. It must be kept in mind that a minimum spacing of $1.2 d_o + h'_f$ is recommended between tubes [44], and so it must be ensured that X_l is above this limit at all times.
- N_{rows} , the number of rows per pass.
- X_t , the pitch between adjacent tubes in the same row. It must be kept in mind that a minimum spacing of $1.2 d_o + h'_f$ is recommended between tubes [44], and so it must be ensured that X_t is above this limit at all times.
- N_{tubes} , the number of tubes per row.
- p'_f , the pitch between adjacent fins. It must be kept in mind that in order to avoid severe *fouling* issues, it is usual practice to impose a maximum fin density of 10 fins/in [44], which means that it must be ensured that $p'_f \geq 2.54 \text{ mm}$ at all times.
- δ_f , the thickness of the fins, with which the *free flow* separation between two adjacent fins can also be defined as $s'_f = p'_f - \delta_f$.
- The number of passes, N_{pass} , is also a variable that can be arbitrarily chosen, but since the process described herein deals with the design of a single pass of the heat exchanger, it is irrelevant for this task. Its impact appears only in the calculation of the target UA for the single pass of heat exchanger, that will obviously depend on the total number of passes chosen, as described in Section 8.1.

With these, the following are also immediately defined:

- $L_2 = L_{\text{pass}} = N_{\text{rows}} \cdot X_l$, *total length* of the single pass.
- $L_3 = L_{\text{width}} = N_{\text{tubes}} \cdot X_t$, *total width* of the heat exchanger.
- The total number of tubes within the single pass, $N_{\text{total}} = N_{\text{tubes}} \cdot N_{\text{rows}}$.

From the imposed thermodynamics, the following are known:

- Mass flows circulating on each side of the heat exchanger, m_c and m_h .
- Inlet and outlet temperatures for both streams, $T_{c,i}$ and $T_{c,o}$ for the cold side, $T_{h,i}$ and $T_{h,o}$ for the hot side.
- *Phase* of the streams at the inlet and outlet of the heat exchanger or, when relevant, the gas-phase to liquid-phase ratio of the mixture in *phase change* heat exchangers where the inlet and/or the outlet are not single-phase fluids.
- The operating pressure, p_c and p_h , on each side of the heat exchanger.
- The ‘target’ *conductance* of a single pass of the heat exchanger, UA , calculated according to the number of passes, N_{pass} , as detailed in Section 8.1.

Which allow for the calculation of the necessary fluid properties and other required data:

- Mean specific heat capacities, $c_{p,c}$ and $c_{p,h}$, of each one of the streams, from eq. 8.3 applied to either side.
- Mean temperatures, $T_{m,c}$ and $T_{m,h}$, *average* between the inlet and outlet temperatures for each stream.
- Mean density ρ_m , conductivity k , and viscosity μ for each of the fluids, which are *not* the averaged properties along the streams, but the values of these thermodynamic properties evaluated at the mean temperature, T_m , defined above.

- δ_w , the thickness of the plates forming the primary surface walls, can also be calculated, depending on the material chosen and the operating pressure of the heat exchanger, according to the ASME BPVC (see Section 8.3 for further details on this).

With all these, an iterative process to converge on the actual values of:

- $L_1 = L_{\text{tubes}}$, the length of the tubes; and
- h'_f , the height of the fins

that would match the ‘target’ UA will be carried out as follows.

1. **Guess** on the value of L_{tubes} , which allows for immediate calculation of:

$$\diamond \text{ the number of fins per tube, } N_f = L_{\text{tubes}}/p'_f.$$

2. **Guess** on the value of h'_f .

3. Calculate the *core mass velocities*, G , on both sides of the heat exchanger:

$$G_c = \rho_{m,c} \cdot u_{m,c} \quad (\text{D.3})$$

$$G_h = \rho_{m,h} \cdot u_{m,h} \quad (\text{D.4})$$

4. With these, the *free flow areas*, A_{flow} , on either side can be calculated:

$$A_{\text{flow},c} = \frac{m_c}{G_c} \quad (\text{D.5})$$

$$A_{\text{flow},h} = \frac{m_h}{G_h} \quad (\text{D.6})$$

5. The inner diameter of the tubes, d_i , is fixed by:

$$A_{\text{flow},c} = N_{\text{total}} \cdot \pi \left(\frac{d_i}{2} \right)^2 \quad (\text{D.7})$$

This can be used to define immediately:

$$\diamond \text{ the outer diameter of the tubes, } d_o = d_i + 2 \delta_w.$$

$$\diamond \text{ the diameter of the extended surfaces, } d_e = d_o + 2 h'_f.$$

6. With these values, it is possible to calculate the free flow area of the hot side by merely geometrical relations:

$$A_{\text{flow},h} = N_{\text{tubes}} \cdot ((X_t - d_o) \cdot L_{\text{tubes}} - \text{Area blocked by the fins}) \quad (\text{D.8})$$

where the area blocked by the fins can be expressed as $2 N_f h'_f \delta_f$. This calculated value for $A_{\text{flow},h}$ will, in general, differ from the one given by eq. D.6, and hence the first **convergence check** will be made here, trying to nullify the error between the calculation of $A_{\text{flow},h}$ according to the mass flow and core mass velocity (eq. D.6) and according to the geometrical relations only (eq. D.8).

7. Depending on the *internal* geometry only, which is already completely defined, the *wet areas* on either side of the heat exchanger can be calculated:

$$A_{\text{wet},c} = N_{\text{total}} \cdot \pi d_i L_{\text{tubes}} \quad (\text{D.9})$$

$$A_{\text{wet},h} = (\text{Area primary surface} + \text{Area extended surfaces} + \text{Area header plates})_h \quad (\text{D.10})$$

And the *hydraulic diameters* on each side are defined as:

$$D_{h,c} = 4 \frac{A_{\text{flow},c} \cdot L_{\text{tubes}}}{A_{\text{wet},c}} = d_i \quad (\text{D.11})$$

$$D_{h,h} = 4 \frac{A_{\text{flow},h} \cdot L_{\text{pass}}}{A_{\text{wet},h}} \quad (\text{D.12})$$

8. The Prandtl and the Reynolds (based on the hydraulic diameter) numbers for either side are readily available as per their definitions:

$$\text{Pr} = \frac{c_p \mu}{k} \quad (\text{D.13})$$

$$\text{Re} = \frac{G D_h}{\mu} \quad (\text{D.14})$$

9. With this, on the **cold side**, the Fanning friction factor, f_c , can be estimated by the correlation:

$$f_c = 0.00128 + 0.1143 \text{Re}_c^{-0.311} \quad (\text{D.15})$$

provided by [98] for the tubes *inside*. From the Fanning friction factor, f_c , and the Prandtl number, Pr_c , the work by Churchill et al. [25] provide the formulae to calculate the corresponding Nusselt number, Nu_c , with minimum empiricism. This work was validated against data presented by Yu et al. [115], showing minimum deviations for the widest range of operating Reynolds and Prandtl. With the Nusselt number, Nu_c , the heat transfer coefficient for the cold side can readily be calculated:

$$h_c = \frac{\text{Nu}_c k_c}{D_{h,c}} \quad (\text{D.16})$$

Since the *inside* of the tubes does not feature extended surfaces, the overall heat transfer effectiveness for the cold side will be $\eta_{O,c} = 1$.

10. Thus the heat transfer *resistance* on the cold side for eq. D.1 is readily defined by means of using the results from eqs. D.9 and D.16:

$$R'_c = \frac{1}{h_c A_{\text{wet},c}} \quad (\text{D.17})$$

11. Similarly, on the **hot side**, *outside* the tubes, the following correlation provided by [98] can be used to determine the Colburn factor, j_h , of a gas stream circulating through an array of perpendicular tubes:

$$j_h = 0.134 \text{Re}_{d,h}^{-0.319} \left(\frac{s'_f}{h'_f} \right)^{0.2} \left(\frac{s'_f}{\delta_f} \right)^{0.11} \quad (\text{D.18})$$

where $\text{Re}_{d,h}$ is the Reynolds number on the hot side based on the outer diameter of the tubes, $\text{Re}_{d,h} = G_h d_o / \mu_h$. With the Colburn factor, the corresponding Nusselt number can be calculated as per its definition:

$$\text{Nu}_h = j_h \text{Re}_h \text{Pr}_h^{1/3} \quad (\text{D.19})$$

so that the heat transfer coefficient can analogously be calculated by:

$$h_h = \frac{\text{Nu}_h k_h}{D_{h,h}} \quad (\text{D.20})$$

The overall heat transfer efficiency of the extended surfaces on the hot side, $\eta_{O,h}$, can be calculated, based on [98], as per:

$$\eta_{O,h} = 1 - \frac{\text{Area extended surfaces}}{A_{\text{wet},h}} \cdot (1 - \eta_f) \quad (\text{D.21})$$

where η_f is the fin efficiency, given by the following expression for the particular case of individual annular fins considered herein (see [98] for the derivation), in which the outer and the extended surfaces *radii*, $r_o = d_o/2$ and $r_e = d_e/2$, have been used for ease of notation:

$$\eta_f = \frac{2 r_o B}{m' (r_e^2 - r_o^2)} \quad (\text{D.22})$$

Here, the m' and B coefficients are defined as per:

$$m' = \sqrt{\frac{2 h_h}{k_w \delta_f}} \quad (\text{D.23})$$

$$B = \frac{I_1(m'r_e) K_1(m'r_o) - K_1(m'r_e) I_1(m'r_o)}{I_1(m'r_e) K_0(m'r_o) + K_1(m'r_e) I_0(m'r_o)} \quad (\text{D.24})$$

where $I_\alpha(x)$ and $K_\alpha(x)$ are the modified Bessel functions of the first and second kind, respectively.

12. With all the above, the heat transfer *resistance* on the hot side for eq. D.1 is readily defined by means of using the results from eqs. D.10, D.20 and D.21:

$$R'_h = \frac{1}{h_h \eta_{O,h} A_{\text{wet},h}} \quad (\text{D.25})$$

13. The second **convergence check**, therefore, will be made here, to try to nullify the error between the ‘target’ UA given by the imposed thermodynamics to the heat exchanger (see Section 8.1) and the UA *calculated* by means of substituting the values from eqs. D.2, D.17 and D.25 into eq. D.1.

D.2 Plate & Fin Heat Exchangers

Firstly and foremost, it is worth noticing that *plate & fin* heat exchangers work with *unmixed-unmixed* streams and, therefore, eqs. 8.6–8.7 have to be replaced by their corresponding expressions for this case. A simple analytical equation does not exist for the *unmixed-unmixed* heat exchanger, but a solution can be found, in the form of a *series expansion*, in [98, Table 3.3, p. 128]:

$$\epsilon = 1 - \exp(-NTU) - \exp(-(1+W) NTU) \cdot \sum_{n=1}^{\infty} W^n P_n(NTU) \quad (\text{D.26})$$

where:

$$P_n(y) = \frac{1}{(n+1)!} \sum_{j=1}^n \frac{(n+1-j)}{j!} y^{n+j} \quad (\text{D.27})$$

Note that eq. D.26 tends to eq. 8.7 as well, for $W \rightarrow 0$. Also, eq. D.26 does not present asymptotic values for ϵ when $NTU \rightarrow \infty$, as opposed to eq. 8.6, so any target effectiveness is theoretically achievable with a single-pass heat exchanger, irrelevantly of the value of the lower-to-higher heat capacities ratio, W . In spite of this, the number of passes, N_{pass} , has still been kept *variable*, nevertheless, to allow for greater flexibility in the optimisation of the size of the *plate & fin* heat exchangers, since it is not always the case that single-pass configurations will result in more compact devices than multi-pass arrangements.

Therefore, the *decision variables* chosen for the design of *plate & fin* heat exchangers are:

- $u_{m,h}$ and $u_{m,c}$, mean velocities of the cold and hot streams, respectively. Following Franco & Giannini [44] guidelines, the mean velocity is *constrained* according to the phase of the fluid:
 - ◊ $0.5 \text{ m/s} \leq u_m \leq 2.5 \text{ m/s}$ for liquid-phase streams;
 - ◊ $0.5 \text{ m/s} \leq u_m \leq 4.5 \text{ m/s}$ for *phase change* streams; and
 - ◊ $8 \text{ m/s} \leq u_m \leq 40 \text{ m/s}$ for gas-phase streams.
- $L_1 = L_{\text{flow},h}$, the length of the heat exchanger in the hot-flow direction.
- $N_{s,h}$ and $N_{s,c}$, the number of offset strips on each side of the heat exchanger.

- N_{layer} , the number of layers on the *cold* side, keeping in mind that it is always assumed that the number of layer on the *hot* side is equal to $N_{\text{layer}} + 1$.
- $p'_{f,h}$ and $p'_{f,c}$, the pitch between adjacent fins on either side. It must be kept in mind that in order to avoid severe *fouling* issues, it is usual practice to impose a maximum fin density of 10 fins/in [44], which means that it must be ensured that $p'_f \geq 2.54$ mm in both sides.
- δ_f , the thickness of the fins (assumed the same for both the hot and the cold sides), with which the *free flow* separation between two adjacent fins, $s'_{f,h}$ and $s'_{f,c}$, can also be defined on either side.
- The number of passes, N_{pass} , is also a variable that can be arbitrarily chosen, but since the process described herein deals with the design of a single pass of the heat exchanger, it is irrelevant for this task. Its impact appears only in the calculation of the target UA for the single pass of heat exchanger, that will obviously depend on the total number of passes chosen, as described in Section 8.1.

With these, the following are also immediately defined:

- The number of fins on the cold side, $N_{f,c} = L_{\text{flow},h}/p'_{f,c}$.
- The length of the offset strips on the hot side, $l_{s,h} = L_{\text{flow},h}/N_{s,h}$.

From the imposed thermodynamics, the following are known:

- Mass flows circulating on each side of the heat exchanger, m_c and m_h .
- Inlet and outlet temperatures for both streams, $T_{c,i}$ and $T_{c,o}$ for the cold side, $T_{h,i}$ and $T_{h,o}$ for the hot side.
- *Phase* of the streams at the inlet and outlet of the heat exchanger or, when relevant, the gas-phase to liquid-phase ratio of the mixture in *phase change* heat exchangers where the inlet and/or the outlet are not single-phase fluids.
- The operating pressure, p_c and p_h , on each side of the heat exchanger.
- The ‘target’ *conductance* of a single pass of the heat exchanger, UA , calculated according to the number of passes, N_{pass} , as detailed in Section 8.1.

Which allow for the calculation of the necessary fluid properties and other required data:

- Mean specific heat capacities, $c_{p,c}$ and $c_{p,h}$, of each one of the streams, from eq. 8.3 applied to either side.
- Mean temperatures, $T_{m,c}$ and $T_{m,h}$, *average* between the inlet and outlet temperatures for each stream.
- Mean density ρ_m , conductivity k , and viscosity μ for each of the fluids, which are *not* the averaged properties along the streams, but the values of these thermodynamic properties evaluated at the mean temperature, T_m , defined above.
- δ_w , the thickness of the plates forming the primary surface walls, can also be calculated, depending on the material chosen and the operating pressure of the heat exchanger, according to the ASME BPVC (see Section 8.3 for further details on this).

With all these, an iterative process to converge on the actual values of:

- $L_2 = L_{\text{flow},c}$, the length of the heat exchanger in the cold-flow direction; and
- $h'_{f,h}$, the height of the fins on the hot side

that would match the ‘target’ UA will be carried out as follows.

1. **Guess** on the value of $L_{\text{flow},c}$, which allows for immediate calculation of:

- ◇ the number of fins on the hot side, $N_{f,h} = L_{\text{flow},c}/p'_{f,h}$; and
- ◇ the length of the offset strips on the cold side, $l_{s,c} = L_{\text{flow},c}/N_{s,c}$.

2. **Guess** on the value of $h'_{f,h}$.

3. Calculate the *core mass velocities*, G , on both sides of the heat exchanger:

$$G_c = \rho_{m,c} \cdot u_{m,c} \quad (\text{D.28})$$

$$G_h = \rho_{m,h} \cdot u_{m,h} \quad (\text{D.29})$$

4. With these, the *free flow areas*, A_{flow} , on either side can be calculated:

$$A_{\text{flow},c} = \frac{m_c}{G_c} \quad (\text{D.30})$$

$$A_{\text{flow},h} = \frac{m_h}{G_h} \quad (\text{D.31})$$

5. The height of the fins on the cold side, $h'_{f,c}$, is fixed by:

$$A_{\text{flow},c} = N_{\text{layer}} \cdot (h'_{f,c} \cdot L_{\text{flow},h} - \text{Area blocked by the fins}_c) \quad (\text{D.32})$$

which immediately defines:

$$\diamond L_3 = L_{\text{pass}} = N_{\text{layer}} \cdot h'_{f,c} + (N_{\text{layer}} + 1) \cdot h'_{f,h} + 2 (N_{\text{layer}} + 1) \cdot \delta_w$$

6. With these values, it is also possible to calculate the free flow area of the hot side by merely geometrical relations:

$$A_{\text{flow},h} = (N_{\text{layer}} + 1) \cdot (h'_{f,h} \cdot L_{\text{flow},c} - \text{Area blocked by the fins}_h) \quad (\text{D.33})$$

This calculated value for $A_{\text{flow},h}$ will, in general, differ from the one given by eq. D.31, and hence the first **convergence check** will be made here, trying to nullify the error between the calculation of $A_{\text{flow},h}$ according to the mass flow and core mass velocity (eq. D.31) and according to the geometrical relations only (eq. D.33).

7. Depending on the *internal* geometry only, which is already completely defined, the *wet areas* on either side of the heat exchanger can be calculated:

$$A_{\text{wet},c} = (\text{Area primary surface} + \text{Area extended surfaces} + \text{Area header plates})_c \quad (\text{D.34})$$

$$A_{\text{wet},h} = (\text{Area primary surface} + \text{Area extended surfaces} + \text{Area header plates})_h \quad (\text{D.35})$$

And the *hydraulic diameters* on each side are defined as:

$$D_{h,c} = 4 \frac{A_{\text{flow},c} \cdot L_{\text{flow},c}}{A_{\text{wet},c}} \quad (\text{D.36})$$

$$D_{h,h} = 4 \frac{A_{\text{flow},h} \cdot L_{\text{flow},h}}{A_{\text{wet},h}} \quad (\text{D.37})$$

8. The Prandtl and the Reynolds (based on the hydraulic diameter) numbers for either side are readily available as per their definitions (see eqs. D.13 and D.14).

9. Since both sides are now analogous in geometry to one another, the process of calculating the overall heat transfer *resistance* for each side of a *plate & fin* heat exchanger is identical for both. For this reason, it will be described only once, without the cold- nor the hot-side sub-indexes, keeping in mind that all the equations described herein must be applied to the hot and the cold sides of the heat exchanger separately. First, the Fanning friction factor, f , and the Colburn factor, j , can be calculated on either side from correlations provided by [98], where the height of the *free flow passage*, $h'_p = h'_f - \delta_f$, has been used for ease of notation:

$$f = 9.6243 \text{ Re}^{-0.7422} \left(\frac{s'_f}{h'_p} \right)^{-0.1856} \left(\frac{\delta_f}{l_s} \right)^{0.3053} \left(\frac{\delta_f}{s'_f} \right)^{-0.2659} \dots$$

$$\cdot \left(1 + 7.669\text{E-}8 \text{ Re}^{4.429} \left(\frac{s'_f}{h'_p} \right)^{0.920} \left(\frac{\delta_f}{l_s} \right)^{3.767} \left(\frac{\delta_f}{s'_f} \right)^{0.236} \right)^{0.1} \quad (\text{D.38})$$

$$j = 0.6522 \text{ Re}^{-0.5403} \left(\frac{s'_f}{h'_p} \right)^{-0.1541} \left(\frac{\delta_f}{l_s} \right)^{0.1499} \left(\frac{\delta_f}{s'_f} \right)^{-0.0678} \dots$$

$$\cdot \left(1 + 5.269\text{E-}5 \text{ Re}^{1.340} \left(\frac{s'_f}{h'_p} \right)^{0.504} \left(\frac{\delta_f}{l_s} \right)^{0.456} \left(\frac{\delta_f}{s'_f} \right)^{-1.055} \right)^{0.1} \quad (\text{D.39})$$

With this, , the corresponding Nusselt number can be calculated as per its definition:

$$\text{Nu} = j \text{ Re } \text{Pr}^{1/3} \quad (\text{D.40})$$

so that the heat transfer coefficient can be calculated by:

$$h = \frac{\text{Nu } k}{D_h} \quad (\text{D.41})$$

The overall heat transfer efficiency of the extended surfaces on either side, η_O , can be calculated, based on [98], as per:

$$\eta_O = 1 - \frac{\text{Area extended surfaces}}{A_{\text{wet}}} \cdot (1 - \eta_f) \quad (\text{D.42})$$

where η_f is the fin efficiency, given by the following expression for the particular case of offset strip fins considered herein (see [98] for the derivation):

$$\eta_f = \frac{\tanh m'l}{m'l} \quad (\text{D.43})$$

Here, the $m'l$ product is defined as per:

$$m'l = \sqrt{\frac{2 h}{k_w \delta_f} \left(1 + \frac{\delta_f}{l_s} \right) \left(\frac{h'_f}{2} - \delta_f \right)} \quad (\text{D.44})$$

10. Thus the heat transfer *resistance* on either side of the heat exchanger is readily defined by means of using the results from eqs. D.34 or D.35, eq. D.41, and eq. D.42:

$$R = \frac{1}{h \eta_O A_{\text{wet}}} \quad (\text{D.45})$$

11. The second **convergence check**, therefore, will be made here, to try to nullify the error between the ‘target’ UA given by the imposed thermodynamics to the heat exchanger (see Section 8.1) and the UA *calculated* by means of substituting the values from eqs. D.2 and D.45 (for either side) into eq. D.1.