

CRANFIELD UNIVERSITY

Performance and techno-economic analysis of humid air turbines for commercial marine applications

Giovanni David Brighenti

SATM

PhD Thesis

Academic Year: 2014 - 2017

Supervisor: Dr P. Zachos

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Abstract

The curb in SO_x and NO_x emissions, led by legislation both globally and in emission controlled areas, could eventually result in a ban of heavy fuel oil (HFO) for commercial marine applications. This could end in a window opening for advanced gas turbine-based power plants in the market of deep sea propulsion, currently dominated by slow-speed two-stroke diesel engines.

Previous studies have identified humid air turbine (HAT) as promising gas turbine-based cycles, able to compete with diesel engines in terms of maximum thermal efficiency. However, no emphasis has yet been given to the identification of the potential of the reheated HAT cycle. This cycle features notably better performance relatively to a single burner system, and may promote the exploitation of humid cycle power plants in applications where high thermal efficiency and high power-to-plant size ratios are of importance.

This thesis presents an analysis of the impact of the technology level of the main component on the performance of the HAT cycle. Part-load performance analyses of the humid air cycle as a whole system is performed. Moreover, a techno-economical study of a commercial vessel powered by a HAT cycle for a given commercial route is used to directly compare the annual cost of machinery of the cycle with a diesel engine.

The research outcomes confirm the higher thermal efficiency of the reheated HAT cycle and show the impact of technology level of the heat exchangers on the performance of the cycle. The part-load analyses of the HAT cycle show low penalties in performing away from the design condition. The comparison between the HAT and diesel engine, and advanced diesel engines, demonstrate the potential of the HAT cycle as prime mover of commercial marine vessels.

Overall this research is a step forward towards the understanding of the feasibility of humid air turbines for replacing diesel engines in commercial marine applications or for applications where the density of power is of competitive advantage.

Acknowledgements

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Abbreviations

A

ABB	Asea Brown Boveri
AC	After-cooler
AHAT	Advanced Humid Air Turbine
AK	Annual Cost of Machinery

C

CAPEX	Capital Expenditure
CC	Main Combustion Chamber, Combined Cycle
CCGT	Combined Cycle Gas Turbine
CHAT	Cascade Humid Advanced Turbine

D

DP	Design Point
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E

EC	Economiser
ECAs	Emission Control Areas
EGC	Exhaust Gas Cleaner
EGR	Exhaust Gas Recirculation
EPRI	Electric Power Research Institute
EvGT	Evaporative Gas Turbine

G

GDP	Gross Domestic Product
GE	General Electric
GT	Gas Turbine

H

HAT	Humid Air Turbine
HFO	Heavy Fuel Oil
HPC	High Pressure Compressor
HPT	High Pressure Turbine

HRSG	Heat Recovery Steam Generator
I	
IC	Inter-cooler
IFEP	Integrated Full Electric Propulsion System
IGHAT	Integrated Gasification Humid Air Turbine
IMO	International Marine Organization
ISA	International Standard Atmosphere
L	
LPC	Low Pressure Compressor
LPT	Low Pressure Turbine
LSMGO	Low Sulphur Marine Gasoil
M	
MARPOL	Marine Pollution
MDO	Marine Diesel Oil
MGO	Marine Gasoil
O	
OD	Off-Design
OPEX	Operating Expenditure
OPR	Overall Pressure Ratio
P	
PAH	Polycyclic Aromatic Hydrocarbon
PEC	Purchase Equipment Cost
pp	Percentage Point
PR	Pressure Ratio
PT	Power Turbine
R	
R&D	Research and Development
REC	Recuperator
relPR	Relative Pressure Ratio
RH	Reheater Combustion Chamber
<i>r</i>HAT	Reheated HAT cycle
RoRo	Roll-on Roll-off Ferries
S	
SAT	Saturator
SC	Simple Cycle
SOFC-HAT	Solid Oxide Fuel Cell Humid Air Turbine

SP	Selection Parameter
SPEC	Specific Purchase Equipement Cost
SRC	Selective Catalytic Reduction
ST	Steam Turbine
STIG	Steam Injected Gas Turbine
T	
TBC	Thermal Barrier Coating
TIT	Turbine Inlet Temperature
TOP-HAT	TOP Humid Air Turbine
V	
vIGV	Variable Inlet Guide Vane
W	
WAC	Water Atomisation Cooling

Nomenclature

A	heat transfer area
A_{cross}	cross-sectional area of the packed-bed of the saturator
A_{g*}	gas flow cross-sectional area at the blade through
A_{surf}	total cooled external area
A_f	total liquid-gas contact area per unit length
Bi_m	metal Biot number
C	heat capacity
C^*	heat capacity ratio C_{min}/C_{max}
C_{AF}	annual fuel cost
C_{AM}	annual maintenance cost
cp	specific heat capacity
D_{ab}	diffusion coefficient
F	fitness index
$\dot{G}$	mass flow per cross sectional area
h	specific enthalpy
H^+	enthalpy invariant coefficient
h_{fg}	specific enthalpy of evaporation
k	global heat transfer coefficient
kxA	thermodynamic heat transfer area
K_{comb}	combustion pattern factor
m	mass flow
M	molar mass, mach number
M^+	flow invariant coefficient
m_c	coolant mass flow rate
m_{c+}	dimensionless coolant mass flow rate
NTU	number of transit units
Nu	Nusselt number
p	pressure

p_{sat}	saturation pressure
P_E	vessel effective power
Per	saturator tower packed-bed geometry function
Pr	Prandtl
$\hat{P}$	power
Q	heat transferred
Q_{gf}	heat transferred from gas to interface
Q_{wf}	heat transferred from water to interface
R	gas constant
R_a	specific air gas constant
R_A	model-ship correction resistance
R_{AA}	wind resistance
R_{app}	ship appendage resistance
R_{bow}	bow-thrusters resistance
R_{SW}	sea wave resistance
R_{tot}	total vessel resistance
R_V	hull viscous resistance
R_v	specific vapour gas constant
R_W	wave resistance
Re	Reynolds number
Sc	Schmidt number
Sh	Sherwood number
St	Stanton number
t	time, static temperature
T	temperature
T_{sat}	saturation temperature
u	velocity
V	vessel cruise speed
α	weighting coefficient, shape parameter Weibull distribution, convection heat transfer coefficient
β	saturator mass flow by pass ratio, scale parameter Weibull distribution, mass transfer coefficient
γ	specific heat capacity ratio cp/cv
Γ	mass flow per unit length
δ	pressure correction factor, thickness

ΔT_{ps}	saturator pinch temperature difference
θ	temperature correction factor
λ	conductivity
μ	dynamic viscosity
ε	effectiveness
ε_0	blade cooling effectiveness
ε_f	film cooling effectiveness
η	thermal efficiency
η_c	internal flow cooling efficiency
Ξ	acquisition cost
ρ	density
Φ	relative humidity
Ψ	stage loading coefficient
ω	specific humidity

Subscripts

a	dry air
bl	mid blade section
c	cold stream
$comb$	combustion chamber
eff	effective
g	gas mixture
h	hot stream
max	maximum
min	minimum
ref	reference
stg	stage
v	water vapour-phase
w	water liquid-phase
0	pinch condition section of the saturator
1	gas inlet section of the saturator, inlet turbine stage
2	gas outlet section of the saturator, outlet turbine stage

Chapter 1

Introduction

1.1 Background

Maritime transport is the backbone of international trade and globalisation, with more than 80% in volume of the global merchandised trade transported by sea [1, 2].

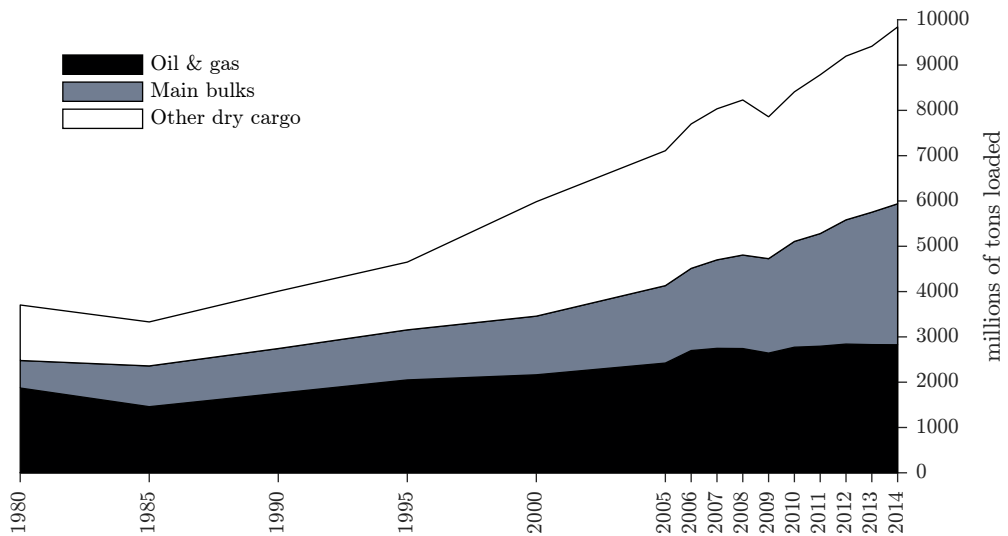


Figure 1.1: International seaborne trade, selected years [2].

The international seaborne trade has been steadily growing over the past decades and in 2014 has grown by 3.4% (see figure 1.1). In recent history, global GDP has been and remains one of the main drivers for the seaborne trade growth. However, over the past two decades the shift in patterns of developing countries, from being a source of supply of raw materials to a growing source of demand and important actors in the globalised manufacturing processes, has increasingly contributed to

the seaborne trade expansion [2] allowing the latter to grow, in average, a few percentage points faster than the global economic growth each year.

One reasons for the success of shipping as a mode of transport is that it is generally considered cost effective, with low energy consumption per tonne and kilometres when compared with aircraft and truck transportation (see figure 1.2). Nevertheless the increasing awareness of the global ship emissions levels and of their impact on the coastal and ocean environment have risen concerns on the sustainability of the industry with the current growth rates [3–5].

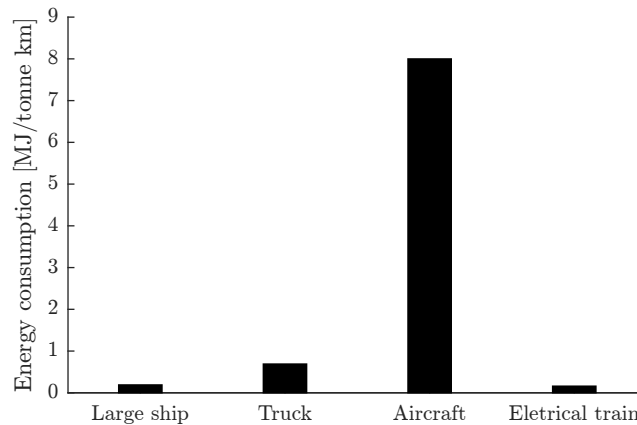


Figure 1.2: Comparison between modes of transport [6].

The vast majority of large commercial ships are powered by low-speed two-stroke diesel engines. The long-established supremacy of these prime movers in the deep sea propulsion sector are reflection of their overall perceived operational economy, simplicity and reliability [7]. The heavy investments by the marine engine designers to reduce the operational cost of these engines have paid off in efficient prime movers with thermal efficiency, in some cases, higher than 55% [7]. This and the rise in the bunker oil prise, have limited the penetration of gas turbine power plants to warship and fast ferries propulsion market.

However, several studies ([5, 8, 9] among others) have supported that shipping contributes by around 15% and 4-9% respectively to all global anthropogenic NO_x and SO_x emissions. Some 70% or more of these emissions occur within 400 km of land, potentially able to affect the pollution of coastal communities and population health, especially in areas of Europe and southeast Asia [5, 8, 9], possibly undercutting the efforts in reducing the SO_x and NO_x land-based anthropogenic emissions.

Nitric oxides (NO_x) contribute to photochemical smog and add to the problem of acid rain. Moreover, these emissions lead to the formation of ozone in the

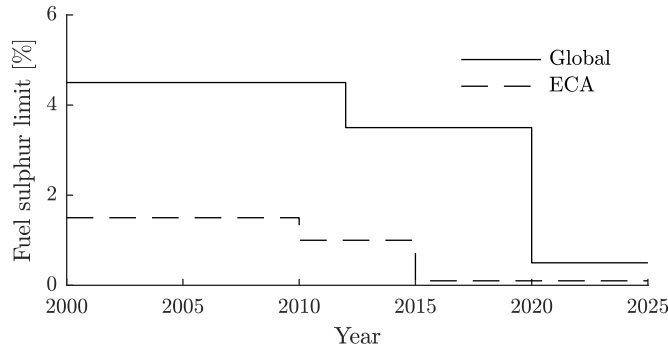
troposphere that could result, when subjects are overexposed, in respiratory illnesses, impaired vision and allergies [10]. Nitric oxide (NO) in diesel engines is produced mainly by the oxidation of nitrogen with excess atmospheric oxygen in high-temperature regions (>1850 K) of the cylinder during combustion and subsequently only a fraction oxidises to NO_2 , therefore known as NO_x . The NO_x formation in these engines is thus mainly thermal and the main nitrogen source is the combustion air. These emissions are relatively high in diesel engines due to, the high peak temperatures and pressures, and the difficulty in controlling the combustion process.

Sulphur oxide (SO_x), mainly SO_2 and SO_3 , are the result of oxidation of the sulphur compounds in the fuel. They are toxic and corrosive and strongly contribute to the formation of acid rains by reacting with water to form sulphurous acid. SO_x emissions are high in marine transport because most diesel engines (70 to 80%) run on residual fuels with high average sulphur content (2.4-2.7%) [4, 5].

The efforts to improve and reduce the impact of shipping on the environment are mainly driven by regulation. The international regulations regarding emission levels were covered in MARPOL AnnexVI [11] by the International Maritime Organization (IMO).

In 2008, IMO approved further amendments to curb SO_x emissions [12] (see figure 1.3a) by: (i) reducing the fuel sulphur limit to 1% within the emission control area (ECA)s by March 2010, (ii) cutting the global fuel sulphur limit from 4.5% to 3.5% by January 2012, (iii) further limiting the fuel sulphur down to 0.1% within ECAs from January 2015, and (iv) reducing of the global fuel sulphur limit to 0.5% by January 2020 after a feasibility review not later than 2018, otherwise postponed to 2025.

In the 2008 amendments [12], IMO also approved to progressively further tighten NO_x emissions standards for new engines: (i) Tier I limits the NO_x to 17 g/kWh for diesel engines installed in ships constructed between January 2000 and January 2011, (ii) Tier II limits the NO_x to 14.4 g/kWh for diesel engines installed in ships constructed after January 2011, whereas (iii) Tier III limits the NO_x to 3.4 g/kWh for diesel engines installed in ships constructed after January 2016 operating in the ECAs, otherwise Tier II applies. These values of emissions are valid for engine of speed lower than 130 rev/min, for higher rotational speeds the limits are stricter and are show in figure 1.3b. In 2009 and then in an amended version in 2015 "Guidelines for the Exhaust Gas Cleaning Systems" [13] was published by IMO and has mandatory effect under MARPOL AnnexVI. Emission ratios (SO_x/CO_2 % v/v) are set as equivalent sulphur fuel contents, that must be met at any operating conditions, including transient. The norms also regulate the composition and



(a) Sulphur concentration limits in fuels

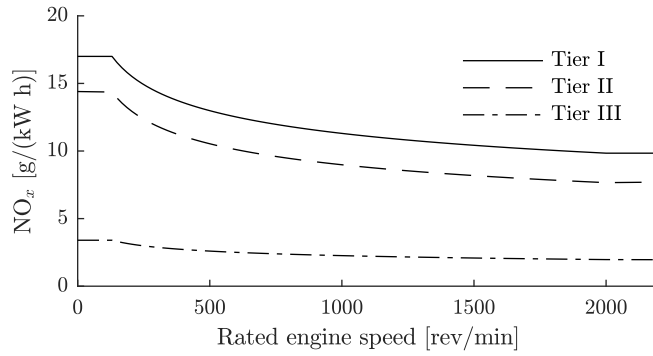
(b) NO_x emission limits

Figure 1.3: Emission limits covered in MARPOL AnnexVI 2008 amendments [12].

pH of the discharge wash water used in the scrubbers. Much tougher curbs on NO_x have been introduced by several regional authorities [7] and there have been several cases of voluntary self regulation in the shipping industry in response and in anticipation of greater demands for social responsibility from the market [2].

Vessels that sail in ECAs currently may adopt two different solutions to meet the SO_x limits: switch to higher distillate fuels with low sulphur concentration (<0.1%) as low sulphur marine gas oil (LSMGO) or install Exhaust Gas Cleaning (EGC) systems known also as exhaust gas scrubbers. The additional capital expenditure for these systems and the volatility of the fuel prices makes it difficult to draw a firm conclusion on the profitability of EGC systems as this strongly depends on the mission profile of the vessel and the variation in price of the fuels in the future [14]. However, Lloyd's Register & UCL London in their Global Marine Fuel Trend 2030 study predict that half of the fuel shares in the marine sector will remain HFO [15].

Two different technologies are currently competing for the reduction of NO_x emissions in diesel engines: Selective Catalytic Reduction systems (SCR) and Exhaust

Gas Recirculation systems (EGR). These systems have been used for decades in power generation and in the automotive industry and are, therefore, mature and robust technology. Nevertheless, a few challenges emerge in implementation on two-stroke diesel engines. The SCR, to be effective, needs to operate with high exhaust temperatures (around 330-350 C°), whereas the current exhaust temperatures of two-stroke engines are around 250 C° [16]. Therefore, the SCR usually needs to be installed upstream of the turbine of the turbocharger, adding complexity to the system. The EGR systems takes part of the exhaust gas and mixes it back with the inlet air. The reduction in NO_x is a result of a richer fuel to air mixture and lower peak temperatures during combustion. Both systems suffer the high concentration of sulphur in HFO [16], and the size and capital cost of both systems would benefit from using low-sulphur content fuels. The EGR system must include a scrubber system to avoid corrosion. The SCR needs to be designed in order to suppress the reaction that transforms SO₂ in SO₃, that would allow the formations white plumes [16]. According to initial cost estimation, the EGR and EGC systems is approximately 10-13% of the vessel price [17]. The operating expenditure will also increase significantly, due to the increase in maintenance costs and the urea consumption.

Although it is not yet clear what solutions shipowners will adopt to comply with the ever stringent regulations, the overall result is most probably an increase in both the capital expenditure (CAPEX) and operation expenditure (OPEX) costs of the power plants [14, 16]. In this convergence of events, gas turbine-based power plant may eventually benefit from the advancement of gas turbine technology [7], as a result of the massive investment in R&D in aero engines and the increase in competitiveness in the propulsion market. In particular this could occur as more stringent regulations on the emission are adopted.

Current marine gas turbines have lower NO_x than diesel engines. The continuous combustion process in gas turbines and the thermodynamic conditions (lower peak pressure and temperature) at which it occurs allow effective control of the residence time at high temperature and therefore lower thermal NO_x generation. Further, NO_x emission reduction is achievable with pre-mixed staged combustion without the expensive selective catalytic systems often required in diesel engines [7]. Moreover, gas turbines run on distillate fuels and therefore would run on low sulphur content fuels to comply with the IMO regulations.

Table 1.1 summarises the main historic qualitative differences between gas turbines and diesel engines. Although it is expected for the OPEX and CAPEX of diesel engines to rise with the implementation of IMO regulations, gas turbines need to reduce substantially the operation expenditure in order to compete within the

Table 1.1: Qualitative comparison of IMO I diesel engines and currently employed marine gas turbines [18].

Criteria	Diesel Engines	Gas Turbines
Noise & vibration	-	✓
Weight & size	-	✓
CAPEX	✓	-
OPEX	✓	-
Availability	-	✓
Fuel price	✓	-
Plant Efficiency (CO ₂ emissions)	✓	-
SO _x & NO _x emissions	-	✓
Workload for crew	-	✓

market. This could be possible with the development of advanced gas turbine based power plants able to achieve high thermal efficiency (>55%).

The study by Sanchez-de-Leon [18] on viable gas turbine-base power plants for the marine market concluded with the down-selection of the humid air turbine (HAT) and combined gas turbine (CCGT) as most promising candidates for marine applications based on thermal efficiency analysis. The first order analysis of the cycles (no account for turbine blade cooling flows) found that both cycles achieve thermal efficiency considerably higher than other gas turbine based cycles and most two-stroke diesel. The density of power was also comparable with the diesel engines resulting in power plants of similar size. HAT cycles, often referred to also as evaporative gas turbine (EvGT) cycles, differ from gas turbine cycles by use of the exhaust heat to increase the flow capacity in the hot section of the cycle, by evaporating water in the gas path of the gas turbine upstream of the combustion chamber. CCGT are combined by two thermal cycles: the gas turbine Brayton cycle that operates at the highest temperature is the *topping cycle*, whereas the waste heat produced by this cycle is used to power a Rankine steam cycle (*bottoming cycle*) and therefore operates at lower temperature.

Although both cycles achieve similar thermal efficiencies and density of power, several studies have found that HAT cycles are expected to present advantages over CCGT cycles, in terms of operability [19–21], emissions [22–24] and maintainability [23] (see table 1.2).

Table 1.2: Qualitative comparison of HAT and combined cycle.

Comparison		HAT	Combined cycle
Operation	Start-up speed	⊙ Faster start-up	ST and HRSG require warming up
	Load responsiveness	⊙ Equivalent to standalone GT	Base
	Ease of control	○ Equivalent to standalone GT	Requires control of ST
Emissions	NO _x reduction	○ Low NO _x due to combustion with humid air	Low NO _x can be achieved using low-NO _x combustion chamber
Economics	Construction lead time	○ Short due to simple equipment configuration with no ST	Base
	Piping	– Thin-wall piping as GT outlet pressure is low, but wide-diameter	Thick-wall piping but small-diameter
	Maintenance cost	○ Simple equipment configuration, low GT combustion peak temperature, no high pressure steam and condensation system	Includes HRSG, ST, and condenser systems, high GT combustion peak temperature, high pressure steam
	Investment cost	○ Includes four HE and a saturator tower	Includes HRSG and ST

⊙: HAT notably superior, ○: HAT superior, –: HAT and CCGT equivalent

ST: steam turbine, GT: gas turbine, HRSG: heat recovery steam generator

HE: heat exchanger

Most research on HAT cycles has focused on land-based electric generation application. Therefore, in order to reduce the feed water mass flow required to operate the cycle, the cycle configurations feature closed water loop. Moreover, almost all the water injected in the the cycle can be recuperated from the exhaust of the gas turbine. However, the HAT cycle layout analysed by Sanchez-de-Leon [18] for marine application directly injects sea water within the cycle. This is possible because the saturator, coupled with a highly effective droplet eliminator, acts as an air and water filter. The cycle therefore does not need a complex water treatment system, with further reduction in CAPEX, size and weight of the plant.

Sanchez-de-Leon [18] also analysed a reheated version of the HAT cycle. The preliminary analysis of the cycle have shown significant benefits in terms both of specific power and thermal efficiency at design point, with no significant change in size of the power plant.

Although the theoretical benefits in operating a HAT cycle with a reheated combustion chamber are well established [18, 20, 25, 26], an analysis of the performance of the cycle across the design space, the feasibility of the design of all components

and the economics of the cycle are still missing in literature. Several studies have focused on the impact of the layout of the heat exchanger in the cycle [27–29]. However, the impact of the technology level of the heat exchangers is yet to be investigated.

A significant number of publications (among others [22, 30–32]), both experimental and analytical, have studied the performance of the saturator tower, the truly novel component of the cycle. Limited research is, instead, available for the part-load performance of the HAT cycle as a whole and for interaction between components in off-design. Moreover, the economic viability of the HAT cycle, in any of its layout, for marine applications remains unexplored.

1.2 Aim & Objectives

The aim of this research is to analyse the feasibility, the across-the-envelope performance and economic viability of the humid air turbine power plant as a prime mover for commercial marine applications.

The objective of this thesis are to:

- Develop an optimisation scheme for the exploration of HAT cycle design space.
- Develop a scheme for part-load and off-design performance analysis of HAT cycles.
- Quantify the sensitivity of the performance of the cycle to the technology level of selected components. Moreover, the objective is to determine the trade-offs between the maximum thermal efficiency and minimum purchase equipment cost of the power plant.
- Analyse the HAT cycle performance across a range of power load setting and in off-design conditions, and verify than the emission levels fall within the limits imposed by the IMO regulations.
- Evaluate the impact different cycle design vectors have on the economics of HAT power plant. Moreover, the objective is to compare the annual cost of machinery¹ of a HAT cycle and a diesel engine over a give marine route.

Overall this research is a step forward towards the understanding of the feasibility of advanced gas turbine-based systems for replacing diesel engines in commercial

¹annual cost of machinery = CAPEX + OPEX

marine applications or for applications where the density of power is of competitive advantage.

1.3 Thesis Structure

This first chapter has described the background and research rational behind this research. Moreover, the aims and objectives of this thesis have been defined. The following chapters of this document are structured as following:

Chapter 2 constitutes a summary of the research available in literature on the humid air turbines. The gaps in the literature are highlighted and the selection of the analysed layout of the cycles is justified by previous research and findings.

Chapter 3 focuses on the design space exploration of the *reheated* and *non-reheated* layout of the HAT cycle. The developed methodology used for the exploration of the design space of the HAT cycles is presented. The impact of the technology level of the heat exchangers and saturator tower on the design point performance of the cycle and on the purchase equipment cost of the power plant are analysed.

Chapter 4 analyses the part-load and off-design performance of the *reheated* HAT cycle. The methodology developed for this purpose is presented. The analysis focuses also on the several *handles* available for driving the power plant in part-load. The estimated emission levels of NO_x , SO_x , CO_2 and CO across the range of power settings are also reported.

Chapter 5 presents economic analyses the *reheated* HAT cycle. The methodology used to estimate the operating cost of the power plants when powering a vessel over a defined marine route is presented. Selected designs from the design space exploration are compared in terms of annual cost of machinery. Moreover, a direct comparison of the annual cost of machinery of the HAT cycle and a diesel engine is presented.

Chapter 6 summarises the major findings and outcomes of this research. Guidelines for future research in assessing and developing humid air turbines for commercial applications are outlined.

Part I

Humid Air Turbine

Chapter 2

HAT cycle — *Characteristics and Variations*

2.1 *Humidified* Gas Turbine

Humid air turbine cycles fall within a particular group known as *humidified* gas turbines, where water is injected, either in liquid or gaseous phase, upstream of the combustion chamber. The common idea behind these cycles is that increasing the mass flow and the heat capacity of the flow through the turbine, augments the specific power output of the gas turbine. The work extracted from the turbine increases, whereas the work required by the compressors remains unchanged — the water is injected usually downstream of the latter—, and the work required to increase the pressure of the water is lower than for the air. If the heat from the exhaust of the turbine is used to heat water or generate steam then the thermal efficiency of the cycle increases. Compared to *dry* recuperated cycles, due to the increases the heat capacity and flow rate in the hot exhaust, the recovery of the exhaust heat is also more effective.

Three main categories of *humidified* gas turbine cycles can be found:

- Gas turbines where water is injected in liquid phase and evaporates within the gas turbine. To boost the power on hot days the water can be injected in the intake, whereas it is introduced in the compressor for inter-cooling and downstream of the compressor for increasing the heat recovery in the recuperator. The water can also be injected in the combustion chamber for reduction of NO_x emissions.
- Gas turbines where the heat from the exhaust of the gas turbine is used to generate steam then injected upstream of the combustion chamber. These

cycles are commonly known as STIG cycles or Cheng cycles. Several of these cycles are available commercially [33].

- Gas turbines where a fraction of the water heated by the exhaust of the gas turbine is evaporated in contact with the dry air upstream of the combustion chamber. The evaporation process occurs in a saturation tower, also known as humidification tower. These cycles are generally known as humid air turbines (HAT) or evaporative gas turbines (EvGT).

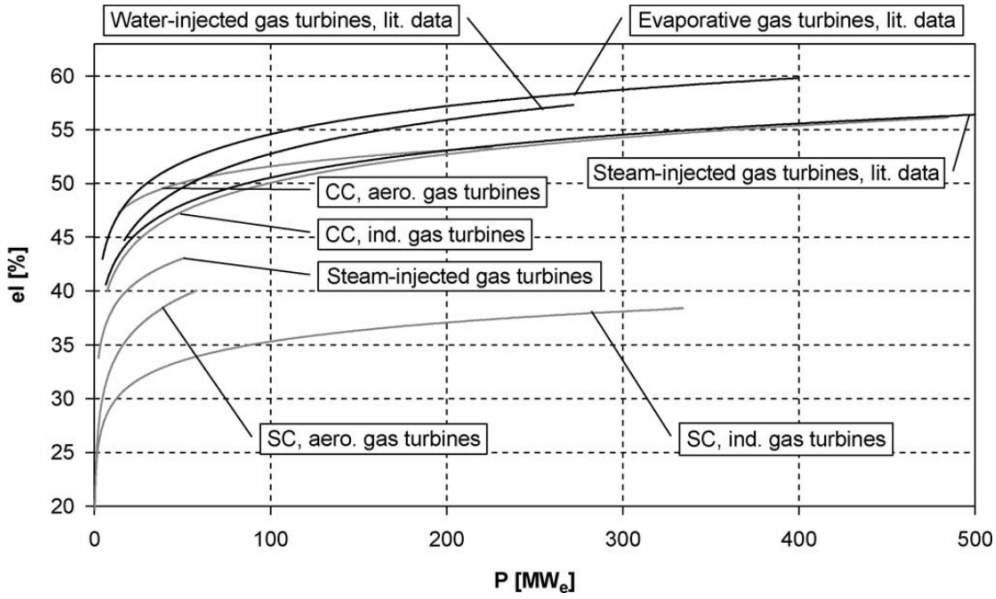


Figure 2.1: Trend lines for literature data (before 2005) of electrical efficiency (el) and power output (P) for humidified gas turbine cycles compared with trend lines for existing simple cycle gas turbines (SC), steam-injected gas turbines and combined cycles (CC) [33].

Figure 2.1 interpolates performance data of these advanced cycles from selected publications before 2005. Although most of the cycles reported by Jonsson et al. [33] in figure 2.1 were simulated under different assumptions —i.e., different turbine inlet temperature, pressure drop in the heat exchanger, etc.— the comparison highlights how the HAT cycle achieves higher thermal efficiency than the remaining *humidified* cycles, and also than the combined cycles.

2.2 Humid Air Turbine

The HAT cycles can achieve such high efficiency due to the effective utilisation of *low-quality* heat in the saturator tower, by means of low temperature evaporation.

The evaporation of the air in the saturator tower occurs in direct contact with the air and allows the generation of steam at a lower temperature than if evaporated as a pure species (as in a boiler). The evaporation in presence of dry air, in fact, occurs not at the single component saturation pressure but at a lower partial pressure —i.e., Dalton’s law.

The dry air enters the bottom of the saturator tower and the hot water, in liquid state, is injected from the top of the tower and dribbles down the packed-bed in the saturator. The heat and mass is transferred from the stream of hot water circulating counter-currently through evaporation. To enhance the evaporation process, by increasing the temperature difference between the two flows, the air at the inlet of the saturator is cooled previously in the after-cooler. The water is heated within the economiser downstream of the gas turbine, and in the inter-cooler and after-cooler, recovering otherwise wasted heat. Inter-cooling is particularly beneficial for the performance of the cycle since: not only the compressor work is reduced, but also the heat extracted is regenerated in the saturator tower. A recuperator is, usually positioned downstream of the saturator and upstream of the combustion chamber to recuperate the *high-quality* heat from the exhaust of the turbine. Nevertheless, several layouts variation of the HAT cycle have been presented and studied during the years with additional or less components.

2.2.1 HAT with Full-flow Humidification

Interest in gas turbines with saturation tower started in 1980s and rose in the 1990s, when in the USA a research programme, that involved Electric Power Research Institute (EPRI), was commenced on the HAT cycle patented by Roe and Galif [34], illustrated in figure 2.2. In this configuration the water-loop is closed and a cooler is used to reduce the water temperature at the inlet of the inter-cooler.

The reports by EPRI [35, 36] compared the performance of the HAT cycle based on available industrial (gas turbine from GE and ABB) and aero-derivative gas turbines (Pratt and Whitney PW4000) with triple pressure reheated combined cycles based on the same gas turbines. The power outputs of the cycles were over 200 MW_e and natural gas was used as fuel. In both cases the specific investment cost of the HAT cycles exceeded the combined cycles. However, due to higher thermal efficiency of the cycles (53.5%–57.4%) by in average 4 percentage points (pp) the cost of electricity was lower for the HAT cycles. The research highlighted that the mass flow rate at the outlet of the turbine would be 20-30% greater than the compressor inlet mass flow. Therefore, aero-derivative gas turbine resulted more suitable, being able to run with higher mass flow rate variations. The comparison of part-load performance, by Rao et al. [19], of a redesigned natural gas-fired FT

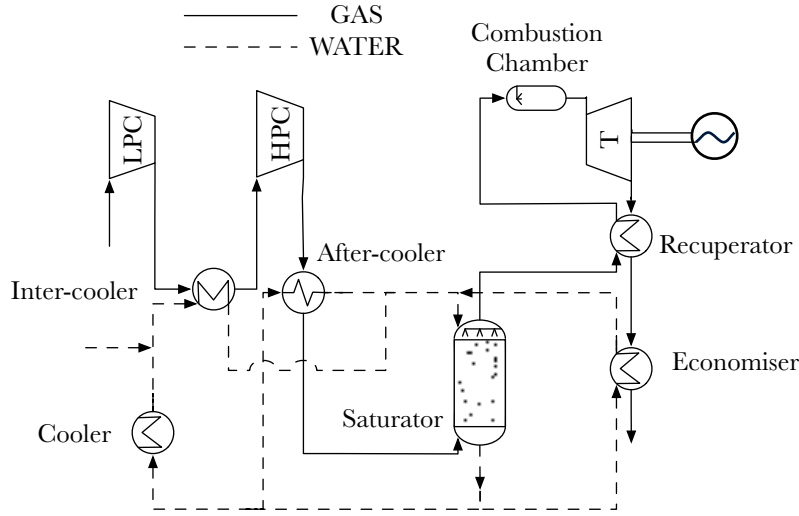


Figure 2.2: Schematics of the HAT cycle studied by Rao et al. [34, 35].
LPC: low-pressure compressor, HPC: high-pressure compressor, T: turbine.

4000 HAT cycle and a combine cycle concluded that the HAT cycle performed better than the combined cycle over the whole range of power settings analysed, and was less sensitive to variations in ambient conditions. Other studies were also performed for HAT cycles integrated with fuel gasification (IGHAT) [19] and also for HAT cycles integrated with solid-oxide fuel cells (SOFC-HAT) [37, 38].

Cascade humid advanced turbine (CHAT) was first proposed by Nakhamkin et al. [20]. The cycle was composed by a gas turbine with two-non-coaxial-shaft and the power-generator on the low pressure shaft. This configuration allowed to solve the miss match in mass flow rate between the compressors and turbines using heavy duty gas turbines and industrial compressors and expanders. The CHAT was an inter-cooled, humidified, recuperated and reheated cycle and several configurations of the latter were studied for large scale power plants ($\sim 300 \text{ MW}_e$) [20, 25]. One of the configuration studied had no low pressure compressor and the therefore the low pressure turbine was effectively a power turbine. It was found that the latter cycle configuration, with a turbine inlet temperature (TIT) of the high pressure shaft of 1149 C° and a TIT of the low pressure turbine of 1394 C° , would reach a thermal efficiency of 60%, and a value of 63% if the TIT of the high pressure shaft could reach 1482 C° [25]. The research also concluded that CHAT cycles (possibly more in generally HAT cycles) had faster start-up time (similar to standalone gas turbines), better part-load performance and retained their power at high ambient conditions better than combined cycles [20].

Chiesa et al. [26] focused on the optimisation of three *humidified* cycles. The HAT cycle studied was inter-cooled, humidified and recuperated. It also featured

two extra coolers on the gas path down stream of the inter-cooler and after-cooler to further reduce the compressor work and increase the temperature leap in the saturator. For fixed overall pressure ratio and TIT he optimised for maximum thermal efficiency the position of the inter-cooler, the water to air mass flow ratio in the saturator and the water mass flow split between the inter-cooler, after-cooler and economiser. He found that HAT cycle was at 2% more efficient than the other *humidified* cycle configuration over a range of pressure ratios and TIT with a maximum value of 57% ($PR^1=33$ & $TIT=1500\text{ C}^\circ$). He also mentioned the possibility of further improving the thermal efficiency and the specific power of the cycle by adding a reheater. The exergy analysis showed that the exergy destruction due to the evaporation of water in the saturator tower was lower than in the mixing of steam and air in the STIG cycles.

Another study on the optimisation of HAT cycles was performed by Lazzaretto et al. [27, 28]. The system of heat exchangers was viewed as a "*back box*", where the heat transfer between the hot and cold streams occurred regardless of the number of heat exchangers and their arrangement. The boundary condition between the cycle components and the "*back box*" were optimised so that the heat transfer in the "*back box*" was maximised independently from the arrangement of the heat exchangers. The main outcomes included that the inter-cooler position was close to the position that minimised the compressor work, the pressure ratio had little impact on the maximum thermal efficiency [27] and that the heat exchangers necessary to obtain maximum thermal efficiency were one inter-cooler, one after-cooler, one economiser and one recuperator [28].

Kavanagh et al. [39, 40] performed a systematic comparison of three *humidified* cycles (STIG, HAT & TOP-HAT²) using consistent thermodynamic models and fluid properties. Multi-Objective optimisation of all three cycles were performed and the TOP-HAT resulted the most efficient and with the lowest cost of electricity, with a maximum thermal efficiency of 53.4% against 51.7% of the HAT cycle.

Nyberg and Thern [41] compared the performance of five different gas turbine cycles: simple cycle, recuperated cycle, recuperated cycle with saturator tower, recuperated cycle with saturator tower and economiser, and HAT cycle with inter-cooler, after-cooler, economiser and recuperator (complete HAT cycle). The impact of TIT and overall pressure ratio was investigated for all cycles, whereas the impact of the heat exchangers technology level was not considered. The performance of the heat exchangers for all cycles was defined by the minimum pinch

¹PR: Pressure ratio.

²TOP-HAT: Quasi-isothermal compression cycle with humidity injected during the compression of the air.

temperature difference between the two streams. The study highlighted the importance of the economiser and after-cooler to generate first of all as high water temperature as possible and then as high water mass flow as possible, to increase the enthalpy exchange in the saturator tower. The best position to bleed air to cool the turbine blades was found to be downstream of the after-cooler (lowest air temperature position) but no considerations on the thermal stress in the blades were made.

2.2.2 HAT with Part-flow Humidification

In 1993 another research program on humid air turbine (named EvGT consortium) was commenced in Sweden. The objective of the program was to demonstrate the technology needed for the HAT cycle, by building a 600 kW rated power pilot plant [33].

A new layout of the humid air cycle was proposed by Westermarck [29], where a fraction of the air delivered from the compressor by-passed the humidification process and mixed with the humid air from the saturator before the recuperator. The layout of the part-flow cycles resulted in a reduction of the heat transfer area of the heat exchangers and the volume of the saturator tower, and therefore in a reduction in the investment cost.

The results of the first full HAT pilot plant experimental campaign were reported by Lindquist et al. [22, 30]. One of the objectives was to demonstrate the increase in efficiency of the HAT cycle over the simple gas turbine and recuperated cycle. The gas turbine used was a Volvo VT600 and the cycle had no inter-cooler or after-cooler. The cycle was recuperated, with hot water generated in the economiser. A bleed-off valve was positioned downstream of the compressor, to reduce the air mass flow downstream of the compressor when testing the HAT cycle, so that the simple and the HAT cycle would have the same turbine expansion line. The thermal efficiency of the HAT cycle was 13 pp higher than the simple cycle with a maximum humidity of the air at the outlet of the tower of $0.137 \text{ kg}_w/\text{kg}_a$. The HAT cycle demonstrated a reduction of 90% of NO_x emissions compared to the simple cycle with emissions lower than 10 ppm. CO emissions in the HAT cycle were ~ 0 for power loads higher than 70% but increased rapidly for power settings below 50%. The saturator tower appeared to be strongly oversized with a minimum pinch temperature in the saturator of $\sim 3 \text{ C}^\circ$. The relative humidity at the outlet resulted to be over 100% although no droplets were detected in the gas. The reason of the phenomena, according to Lindquist et al. [22], was to be found in the way the dew point was calculated at high pressure with standard steam tables.

Part-flow cycles in combination with steam generation and injection were proposed by Ågren for non-inter-cooler aero-derivative gas turbines [42] and inter-cooler industrial gas turbines [43]. Two additional heat exchangers were added in between the recuperator and the economiser, where high temperature heat was used to generate steam and to super-heat the humid air from the outlet of the saturator tower. For the aero-derivative gas turbine the maximum thermal efficiency was of 52.9% (PR=35 & TIT=1336 C°) with 12% of the inlet gas being humidified. For the industrial gas turbine full-flow resulted optimum and the maximum thermal efficiency was 52.6% (PR=20 & TIT=1358 C°).

Further studies of the cycle were performed by Bartlett [44] that through an economical study found the part-flow cycle had largest potential. Jonsson et al. [45] performed an economic assessment of the part-flow humid cycle with steam injection using existing gas turbines (GTX100 & Trent) for small-to-medium-size power plants. Both cycles were comparable with a combine cycles with dual pressure steam cycle based on the same gas turbines in terms of thermal efficiency. The investment cost resulted between 16% and 25% lower and the specific investment cost 12% to 15% lower for the humid cycles. In his study, Bartlett [44] also concluded that water self-sufficient plant was feasible, after both theoretical and experimental investigation of the water recovery system. The research also focused on the filter capabilities of the saturator tower. The saturator, given low droplet entrainment, was found to act as a filter increasing the quality of the air flow by capturing larger particles. Because of the very low transfer of impurities from the water to the air, only 1% bleed-off from the humidifying water circuit was necessary to avoid build up of impurities beyond safeguard levels.

Sanchez-de-Leon [18] investigated several *dry*, *humidified* and combined advanced gas turbine cycles for potential commercial marine applications, with the core architecture based on Rolls-Royce's MT-30 marine gas turbine. He explored several configurations of the part-flow HAT cycle: simple HAT with recuperator, reheated with recuperator, with inter-cooler and recuperator, and reheated with inter-cooler and recuperator. The water loop of the HAT cycles was open and the sea was considered the source of water. This was considered possible given very low droplet entrainment ratios at the outlet of the saturator, achievable with a suitable design of the tower and the employment of an effective droplet eliminator. The maximum thermal efficiency at design point was achieved by the inter-cooled, recuperated and reheated version of the cycle (PR=50 & TIT=1600 K) with a gain of 4 pp in efficiency on the inter-cooled, recuperated version (PR=35 & TIT=1600 K). The specific power of the reheated cycle was found to be higher, although the dimensions of the heat exchangers did not vary significantly due to the higher heat transferred in the latter cycle. The total manufacturing cost of the heat exchangers

for the reheated cycle was found to be 29% higher than for the only inter-cooled, recuperated cycle. This was due to the required use of Inconel alloy instead of stainless steel for the recuperator (to withstand the higher temperatures) and the higher density of passages in the inter-cooler of the reheated cycle. The HAT cycles also achieved efficiencies comparable with a dual-pressure steam combined cycles with the same gas turbine technology. The performance calculations of the cycles were performed not considering the effect of blade cooling, as the main objective was to compare a vast number of cycles and the impact of blade cooling was assumed to be similar for all of the cycles.

2.2.3 Advanced Humid Air Turbine

Another project on humid air turbines was commenced in 2004 by Hitachi Ltd, Central Research Institute of the Electric power Industry and Sumitomo Precision Products. The research was focused on a variation of the full-flow HAT cycle with water injection at the inlet of the compressor, water atomisation cooling (WAC), and therefore no a surface inter-cooler. The objective was to first build a AHAT pilot plant (phase I) and then a 40 MW class AHAT demonstrator power plant (phase II).

Takahashi et al. [46] compared several *humidified* gas turbine cycle configurations. HAT and AHAT resulted to have very similar maximum thermal efficiencies, within 0.5 pp, and similar specific powers. Nevertheless, HAT and AHAT cycles achieved maximum thermal efficiency with lower humidity levels at the inlet of the combustion chamber and lower pressure ratios than the other *humidified* cycles.

Higuchi et al. [21] and Araki et al. [24] presented the results of the experimental campaign on a 3.68 MW pilot plant (run up to 4 MW), by Hitachi Ltd. in Hitachinaka City, Japan. Rashing Super-Ring™, a fourth generation random packing, was used for the saturator tower and the gas turbine core was a two-stage radial compressor with total pressure ratio of ~ 8 and a two-stage axial turbine. The flexibility of the AHAT cycle was confirmed, and NO_x emissions were measured as low as 10 ppm, whereas the CO emissions were measured between 200 and 300 ppm. The combustion efficiency with humidification resulted to be 99.6%, lower than the target 99.9%. It was reported that the high CO emissions and low combustion efficiency was a consequence of not operating all the clusters of fuel nozzles in the combustion chamber, as the combustion chamber had already been tested separately in high humidity conditions and ~ 0 ppm of CO had been achieved in that occasion [47]. The cooling flow requirement for the turbine, calculated according to Higuchi et al. [48], resulted adequate for steady-state and transient operations. A spray-type system was used for the water recovery at the outlet

of the economiser. It was assessed that the non-uniform velocity distribution of the gas flow made the recovery system less effective than anticipated and no full condensation was achieved. An optimisation of control system and an improved selection of materials for the water quality system was found to be required to avoid the concentration of impurities.

The results of the tests on a 40 MW-class AHAT cycle, by Hitachi Ltd. in Japan, were reported by Takeda et al. [49]. The cycle tested was a simplified version of a AHAT, as the main objective of the test was to verify the practicality of operating an AHAT cycle with a heavy duty industrial gas turbine. The after-cooler and economiser were not included. The humidification occurred through a spray-type saturator (humidity at the inlet of the combustion chamber $\sim 5\%$) and only $1/60$ of the exhaust gas was treated by a new packing-type water recovery system. The heavy-duty industrial gas turbine used was the H-50, single shaft with multi-can combustion chamber. The test verified that key components of a medium-class industrial gas turbine could be used for a AHAT cycle. The cold start-up from ignition to rated power test showed that this could be achieved in ~ 60 min, compared to the 180 min of a combined cycle. The new water recovery system proved to be able to achieve water recovery ratio on basis of humidification more than 100%.

Phase III of the research programme started in 2012 and was expected to run for four years [50]. The objective was the development of technologies for enhancing reliability and for scaling-up in preparation for a medium-class demonstration plant.

2.2.4 Saturator Tower

The saturator tower is the truly novel component that has not been used previously in a power cycle. The majority of the models used and developed for the simulation of this component are derived by chemical engineering methods for cooling tower design. The substantial difference in pressure and temperature, between the fluids used usually in chemical applications and in the saturator tower, has required the development of specific models for the HAT cycles.

Two main types of saturation towers have been studied: packed bed towers and tubular towers. The packed bed tower consists of a shell with packing (structured or random), water distributor, bed support and droplet eliminator. The packing increases the contact between the water and air, favouring the evaporation. The tubular tower is considered convenient for small cycles as it eliminates the economiser [51] and consists of a shell and a vertical tube bank. The exhaust gas

flows externally to the tubes, cooling down, whereas the water and the air flow are brought into contact inside the tubes. The different methods of humidifying the air were discussed by Dalili [51], and Dalili and Westermark [52] performed experimental investigation on the packed-bed tower.

One of the first algorithms for the simulation of a saturator tower in a HAT cycle was proposed by Enick et al. [53] based on energy balance and mass transfer equations. Aramayo-Prudencio and Young extensively studied performance of the saturator tower [31, 32]. True invariants of the system were defined along with the expressions for calculating the thermodynamic state of the air for the pinch condition in the tower [31]. A rigorous control volume formulation was also proposed for the simulation of the saturator, and analytical expressions were derived from the heat and mass transfer coefficients for structured packed-bed tower [32]. Lindquist et al. [54] studied the impact of the diffusion constant model in the simulation of the tower. The model developed predicted both the outlet temperature of the air from the tower and the theoretical height of the structured packed-bed with an uncertainty of 10% against experimental data.

Other models were also developed by Parente [55], Dalili [51], Ågren [56]. Araki et al. [57] also proposed a saturator model in the design study of the tower for an AHAT. He also analysed the impact of the cycle required power on the dimensions of the latter. Dalili [51] also studied the tubular saturator tower and developed a model for the simulation of the latter.

2.2.5 Part-load and Off-design Performance of HAT Cycles

In more than 20 years of research on HAT cycles, limited amount of literature is available on part-load and off-design simulations of these cycles. The performance of the saturator has been studied separately from the cycle, whereas only in a few cases the overall performance of the power plant was considered.

Nakhamkin et al. [20] in the study of the CHAT cycle reported, among other, the variation of heat rate of the CHAT cycle compared to a combined and simple cycle (same gas-turbine). The efficiency of the CHAT cycle dropped from 54.7% to 51.5% when passing from full load to 50% load. For the same variation in power output the combined cycle efficiency dropped from 54.8 to 47.8. The impact of the ambient temperature was also considerable lower for the CHAT cycle with a 1.5% loss of power and 1% increase in heat rate for an increase in temperature from 15 C° to 32.2 C°. For the same variation the combined cycle had a 10% loss of power

and 3% increase in heat rate. The same study also reported that from a cold-start the CHAT cycle could reach full power within 30 minutes compared to the over 2 hours of the combined cycle. Nevertheless, no methodology, assumptions or description of the models used to achieve these results were published.

The impact of ambient conditions on the performance of the full-flow HAT cycle was studied by Kim et al. [58]. The cycle studied was a single shaft HAT, similar to the one shown in figure 2.2. The impact of ambient conditions was studied in terms of design thermal efficiency and specific power —no off-design modelling of the gas turbine, heat exchangers or saturator tower were used. The variation in power output was calculated by multiplying the specific power by the new off-design mass flow rate due to the variation in ambient air density. This simplified approach showed that HAT cycles have lower thermal efficiency and power deterioration than combined cycles, calculated with the same methodology. Nevertheless, the variation in isentropic efficiency of the turbomachinery operating away from the design point was not considered. Moreover, the constraints of the physical dimensions of the heat exchangers (heat transfer area) and saturator (tower height) on the heat and mass transfer were neglected.

Aramayo-Prudencio in his PhD thesis [59] reported a study on the off-design performance of a full-flow HAT cycle. The cycle was simplified and did not have an inter-cooled or the economiser —all the water injected in the saturator was heated in the after-cooler. The gas turbine was based on a three-shaft gas turbine with three compressors and three turbines, where the power was extracted from the low pressure shaft. The gas turbine was designed as a dry cycle and then *operated* as a HAT cycle. The impact of the combustion outlet temperature was assessed within the range of 94-103% of the design value. For the matching of the operating point general compressors map were used, whereas the parameters necessary for the turbine were calculated analytically. The saturator outlet conditions of the gas were calculated with the control volume formulation model of the saturator [32] by keeping the physical dimensions constant. The study highlighted the importance of using a model of the saturator tower able to calculate the evaporation rate for a fixed geometry saturator at different inlet conditions, when calculating the performance of the whole power plant. The study did not consider cooling flows for the turbines and the off-design performance of the heat exchangers.

The off-design performance of the full flow HAT cycle was studied also by Wang et al. [60] by comparison with other *humidified* cycles. The isentropic efficiency of the compressor and turbine at off-design were calculated with analytical correlations, and off-design models were used for the heat exchangers and saturator tower. The saturator model was based on Perente's et al. [55] work. The HAT

cycle performed better than the other *humidified* cycles when operating at higher ambient temperature. However, both the simple cycle and the STIG cycle performed better than the HAT cycle in terms of relative thermal efficiency when operating in part load. According to Wang et al. [60] the part-flow HAT cycle could perform better in off-design by regulating the air mass flow through the saturator.

Takahashi et al. [61] explored the methods that could be used to drive the AHAT cycle in part-load and compared the part-load performance of the AHAT cycle with the part-load of combined cycles. Four handles for driving the AHAT cycle in part-load were analysed: the reduction of the inlet air by closing the vIGV, the reduction of the combustor outlet temperature, the decrease of humidity at the inlet of the compressors and the stop of water injection. Four part-load schedules were studied, where the cycle was driven in off-design by a combined use of the studied handles. The schedule with the highest thermal efficiency across the analysed power load was to reduce the inlet mass flow down to 80% of the design value and then, for lower power outputs, reduce the TIT. The AHAT authors also concluded that the analysed AHAT cycle has higher part-load performance than combined cycles. No details on the methods and assumptions used for the cycles off design performance simulations were revealed within the paper, both for the AHAT and the combined cycle.

2.3 Summary

This chapter described the benefits of humidification of the working fluid within the Brayton cycle, the difference between the categories of *humidified* gas turbines, and the past investigations of humid air turbines. Most of the research on HAT cycles has focused on land-based electric generator applications. Therefore, in order to reduce the feed water mass flow required to operate the cycle, the cycle configurations feature closed water loop. Almost all the water injected in the cycle can be recuperated from the exhaust of the gas turbine. Research on HAT cycles for marine application focused on the open water loop with direct use of sea water. This is possible given very low droplet entrainment ratios at the outlet of the saturator, achievable with a suitable design of the tower and the employment of an effective droplet eliminator. The power plant system is simplified because of the absence of the bulky and expensive water treatment loop, and would be a major advantage when powering a vessel.

The potential benefits of a reheated version of the HAT in terms of thermal efficiency have been analysed. Preliminary analysis of the cycle have shown significant

benefits in terms both of specific power and thermal efficiency, with no significant change in size of the heat exchangers. Further analysis of the performance within the design space and at part-load, and techno-economics studies are necessary to determine the feasibility of this cycle layout for marine applications.

Performance analysis and techno-economic studies of the HAT cycle have been focused on the selection of the layout of the cycle, and on the position and number of heat exchangers used. The impact on the thermal efficiency and specific power of the turbomachinery technology and the maximum temperature in the cycle has been assessed for several layouts of the cycle. However, the impact of the technology level of the heat exchangers on the performance and on the design vector of the cycle has not been discussed. Moreover, in techno-economic studies focus has not been set on the impact of the technology level of the heat exchangers on the equipment purchase cost of the power plant.

In the past experimental and analytical research has focused on the performance of the saturation tower. Over the years several models for the performance of the saturator with fixed geometry have been developed and validated against experimental data from test rigs. Nevertheless, limited research is available on the interaction between the heat exchangers, turbomachinery and saturator tower when operating away from the design point. Moreover, limited research is available on the impact of ambient condition and part-load settings on the performance of the cycle.

The theoretical benefits in operating a HAT cycle with a reheater combustion chamber are well established, however an analysis of the performance of the cycle across the design space and a feasibility analysis of all cycle components and of the economics of cycle are still missing. There is not sufficient knowledge regarding the cycle behaviour at part load operation, and the interaction between the cycle components in off-design has not been explored. Therefore, the cycle behaviour should be investigated at early stages to confirm the high performance of the cycle across a wide range of power outputs. The economical viability of this cycle for commercial marine applications remains also to be assessed.

In this chapter, the previous research available in literature for the humid air turbines is outlined, both for land-based power generation or cogeneration, and for marine applications. The gaps in the literature that need to be covered for the fulfilment of the aims and objectives of this thesis have been defined. Moreover, the selection of the open water loop layout of the cycle and the investigation in the reheated version of the HAT are justified by previous research outcomes here reported.

Chapter 3

Design Point Performance Analysis

This chapter presents the study of the design point performance of the HAT cycles across the design space defined as a function of selected design parameters, representative of the technology level of main cycle components. First the HAT cycles layout and the methodology used to estimate the performance at design point are reported. The impact of the technology level of selected components on the performance and cost are therefore discussed. Finally the impact on the design point performance of the variation in the design point environment conditions is explored.

The definition of *design point* used within this thesis refers the operating point with maximum temperature at the outlet of the combustion chambers, for which the power plant can continuously operate in time. Therefore the cycles are designed for maximum power output.

3.1 Cycle Description

The layout of the 40 MW class cycle herein studied, with a dual-shaft and free power turbine gas-turbine arrangement, based on the MT-30 (Rolls Royce) marine gas turbine architecture is illustrated in figure 3.1. Two version of this cycle are herein studied: (i) *reheated* —i.e., with a reheater combustion chamber— and (ii) *non-reheated* —i.e., without a reheater combustion chamber.

Water absorption into the compressed air takes place in the packed-bed saturator (see figure 3.2). The dry air enters the bottom of the saturator tower and the hot water, in liquid state, is injected from the top of the tower and flows down the packed-bed in the saturator. The hot water and the dry air flow in direct contact

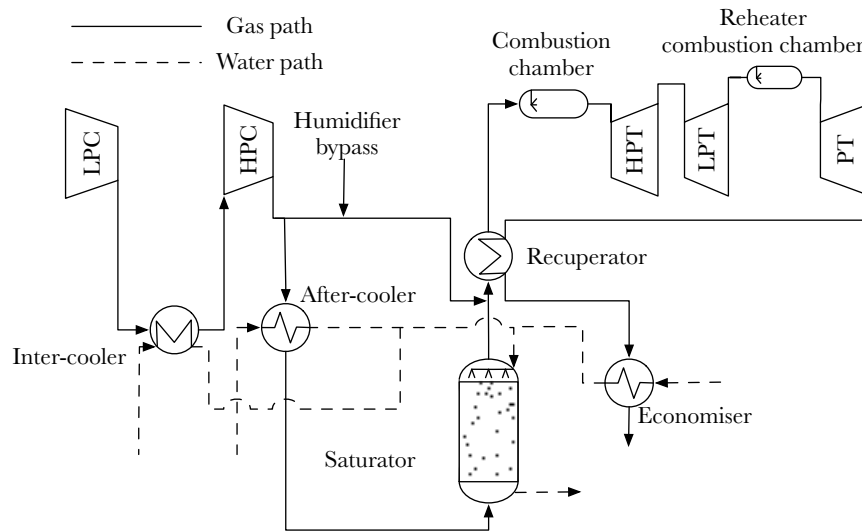


Figure 3.1: Humid Air Turbine cycle layout.

LPC: low-pressure compressor, HPC: high-pressure compressor, HPT: high-pressure turbine, LPT: low pressure turbine, PT: power turbine

in counter-current arrangement. The heat and mass transfer occur by means of evaporation, driven the difference in temperature and water concentration in the two streams. To enhance the evaporation process, by increasing the temperature difference between the two flows, the air at the inlet of the saturator is cooled upstream in the after-cooler. The water is heated within the economiser downstream of the gas turbine, and in the inter-cooler and after-cooler, recovering otherwise wasted heat.

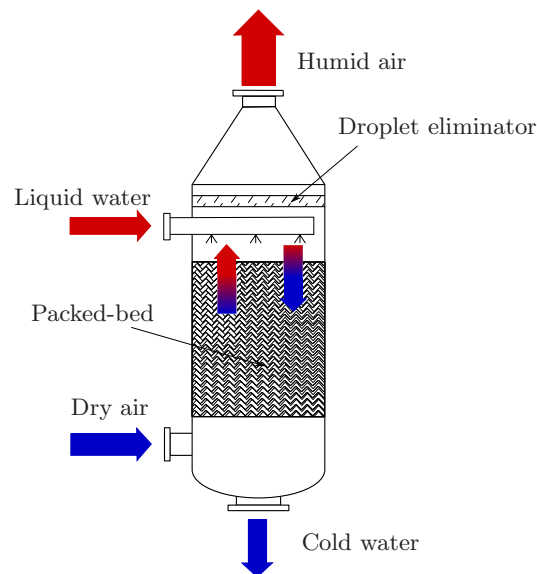


Figure 3.2: Schematic drawing of the saturator tower.

Due to the relatively low temperatures of the humid air exiting the saturator, the HAT cycle benefits significantly from implementing a recuperator that will recover the high-quality —i.e., high-temperature— waste heat from the power turbine exhaust gas, before streaming the humid air into the combustion chamber.

A by-pass of the after-cooler and saturator for the air stream is made available from the outlet of the high-pressure compressor (HPC) to the inlet of the recuperator, so that a certain fraction of dry air can by-pass the humidification process completely. The dry air from the HPC and the humid air from the saturator mix upstream of the recuperator. The fraction of air that from the outlet of the HPC ($m_{g,HPC\ out}$) flows through the after-cooler and saturation tower ($m_{g,SAT\ in}$) is defined by $\beta = m_{g,SAT\ in}/m_{g,HPC\ out}$.

In this study, an open loop is used to feed water the inter-cooler, after-cooler and economiser. The system results simplified and the bulky and expensive water treatment system are not necessary. The evaporation process in the saturator and an efficient droplet eliminator make the saturator a water-treatment device per se [44], allowing the use of direct sea water without undermining the life of the components.

3.2 Cycle Design Point Performance

The thermodynamic design point performance of the HAT cycle is calculated with a system of modules for single or groups of components. The performance of the gas turbine and recuperator are resolved by Turbomatch, a software developed in Cranfield University that, through extensive use, has been proven reliable, accurate and flexible [62–64]. The blade cooling mass flow requirements, and the performance of the inter-cooler, after-cooler, economiser and saturator tower are resolved by separate modules. Due to the non-linearity of the performance of the cycle components, the convergence of the design point is obtained using an iterative solver: a Broyden method [65].

The algorithm used to resolve the design performance of the HAT cycle is shown in figure 3.3 and requires as input the component design parameters, ambient conditions and the required power output. To initialise the solver and run the gas turbine module, the gas turbine inlet air mass flow, the turbine rows cooling flows, and the temperature and humidity of the gas at outlet the inter-cooler, economiser, and saturator are guessed. The results from the Turbomatch simulation are then used to resolve the mass flow requirement for the cooling of the turbine rows and the performance of the remaining heat exchangers and saturator. The errors from

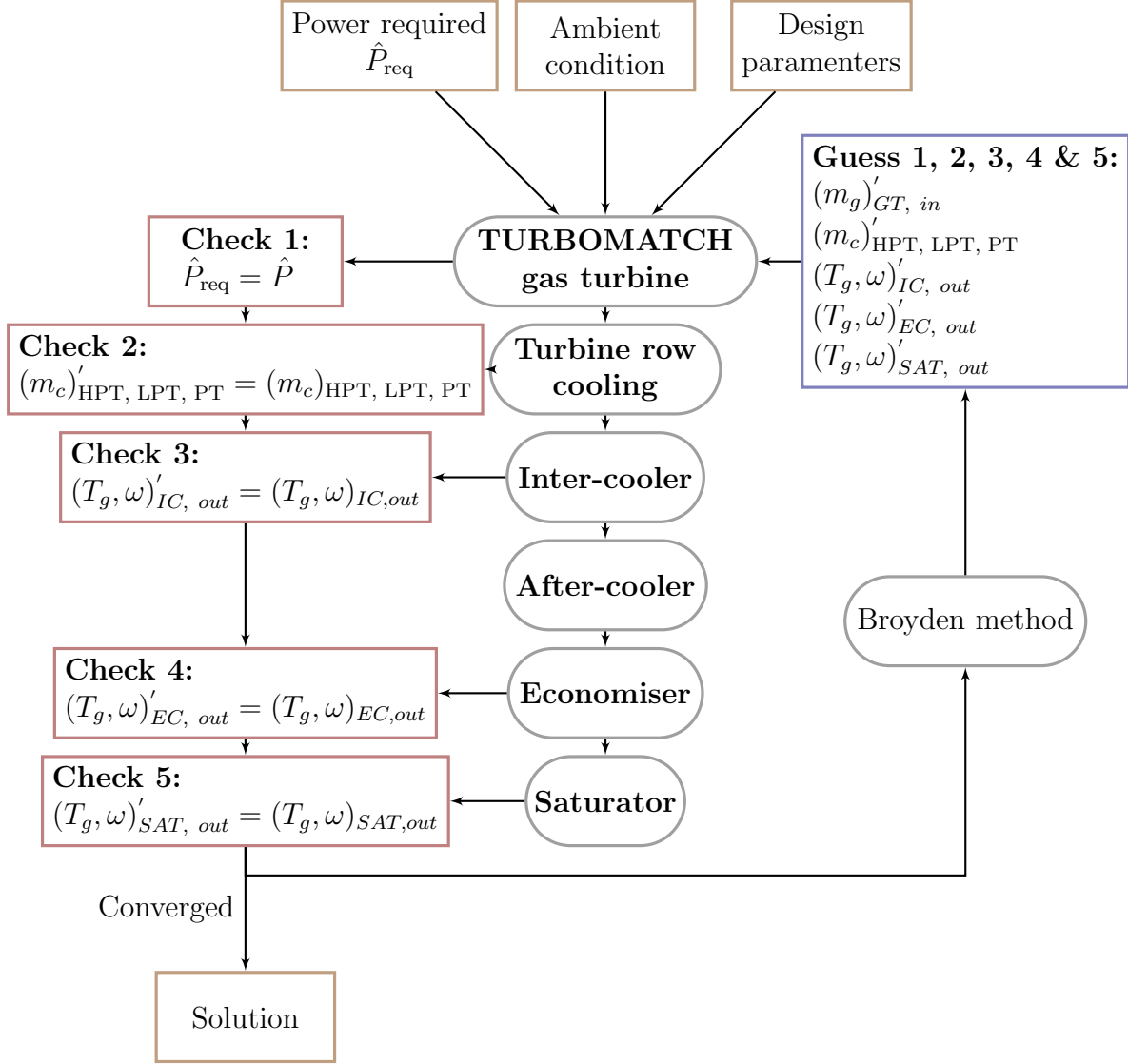


Figure 3.3: HAT cycle design point simulation algorithm

the checks on the power output, turbine cooling flow requirements and thermodynamic outlet conditions of the inter-cooler, economiser and saturator are used to update the Broyden matrix until convergence is achieved.

Some general assumptions are made throughout all the simulations:

- All components are adiabatic
- No account for mechanical losses

Gas Turbine & Turbine Row Cooling

Turbomatch is a 0D performance simulation software. Since all the design point component parameters are specified by the user, the design point performance is

resolved by subsequent calculations of the components along the main gas path. Iteration within Turbomatch at design point is needed only for the calculation of the outlet conditions of the the two streams of the recuperator.

A 90% polytropic efficiency is assumed for the compressors and the core turbines, whereas 92% for the power turbine. This reflects the current technology levels of the components. The turbine inlet temperature (TIT) downstream of the combustion chamber is 1600 K. The impact of the TIT of the combustion chambers and of the polytropic efficiency of the turbomachinery on the thermal efficiency of the *reheated* HAT are reported in appendix B.1. A pressure loss of 5% is accounted for through both combustion chambers. The fuel used is Marine Diesel Oil (MDO) and the chemical proprieties of the fuel used in combustion calculations are specified in Goodger [66].

The turbine rows, if necessary, are air cooled and the required coolant mass flows are calculated using the methodology proposed by Young and Wilcock [67]. This method accounts for the cooling requirement of the disks, the stationary and rotating blades when cooled with internal convection, film cooling technology and thermal barrier coating (TBC). Typical values for maximum metal temperature ($T_{m,max} = 1300$ K), film cooling effectiveness ($\varepsilon_f = 40\%$) and internal flow cooling efficiency ($\eta_c = 70\%$) are fixed in agreement with Horlock [68]. The cooling flows for all turbines are extracted downstream of the high pressure compressor and upstream of the aftercooler. Detailed description of the model used for the calculation of the cooling flow requirements are presented in appendix A.4.

Heat Exchangers

The outlet conditions from all gas-water heat exchangers, namely inter-cooler, after-cooler and the economiser, are resolved by imposing the effectiveness ($\varepsilon = Q/Q_{max}$) and the heat capacity ratio ($C^* = C_{min}/C_{max}$) between the two streams in the assumption that the water side is always the stream with the highest heat capacity ($C_w = C_{max}$). The salinity of the water is neglected in the calculation of the water proprieties, due to the lack of data at high pressure and temperature.

Water condensation in the air stream can take place in the inter-cooler and after-cooler due to the high pressure and low temperature of the gas at the outlet of the heat exchangers. Significant condensation occurs in the economiser, as a result of the high quantity of water in the exhaust gas of the recuperator and the low outlet temperature. The latent heat of condensation of the water in the humid air variates the heat capacity of the gas (C_g) when condensation occurs and generates a pinch point between the two streams within the heat exchanger (figure 3.4).

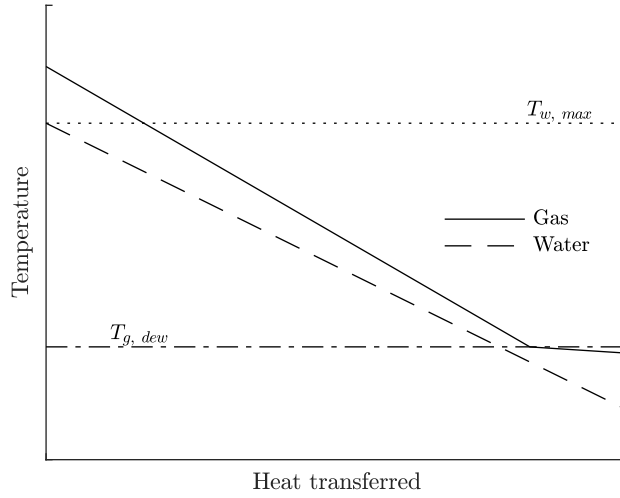


Figure 3.4: Heat transferred in the economiser.

Water is injected in the saturator at liquid state to facilitate the humidification process. The temperature of the water at the outlet of the heat exchangers is therefore limited by the water saturation temperature. A safety coefficient of 0.9 is introduced to avoid any steaming in the saturator as follows:

$$T_{w,out} \leq T_{w,max} = T_{sat}@ (0.9p_{SAT}) \quad (3.1)$$

Where $T_{w,out}$ is the water outlet temperature of the heat exchangers, T_{sat} is the water saturation temperature and p_{SAT} is the pressure in the saturator. When the water temperature at the outlet of the heat exchangers exceeds the limit temperature $T_{w,max}$ a new value of C^* is calculated overwriting the imposed value.

In the recuperator all the inlet proprieties of both streams are defined. Therefore, the outlet conditions of the two streams are calculated with the definition of the effectiveness of the heat exchanger.

In the inter-cooler, after-cooler and economiser, 5% pressure loss ($P_{loss} = \Delta P/P_{in}$) is estimated in both the water and air streams. In the recuperator, 7.5% pressure losses is estimated in the cold stream, whereas 5% pressure loss is estimated in the hot stream. This difference in pressure loss between the hot and cold sides of the recuperator is due to the difference in cross-sectional area of channels —i.e., a consequence of the difference in gas density in the two flows.

Saturator

The saturator, used for the evaporation of water into the cycle, is modelled according to Sanchez's methodology [18] and based on Aramayo-Prudencio's and Young's theory of pressurised saturators [31]. The methodology is designed for only design point performance calculations, for reduced required computational time and with a minimum amount of input information from the user to calculate the outlet conditions of the two streams:

- Inlet conditions of the two flows (water & dry-air)
- Pinch temperature difference ΔT_{ps} between the operating and the saturated air enthalpy lines in the saturator.

The detailed assumptions and model description are reported in appendix A.1. This model has been validated against experimental results of a HAT pilot cycle by Lindquist et al. [54] and the maximum error in the outlet properties of the two streams is of 1.5% as reported in table A.1. The pressure loss through the saturation tower was assumed to be a conservative 5% of the inlet gas pressure.

3.3 Purchase Equipment Cost Estimation

This section reports the methods used for the estimation of the purchase equipment cost of the HAT power plant and of the single components. It is assumed that the power plant would be installed on a vessel in integrated full electric propulsion (IFEP) configuration. The acquisition cost comprises of the cost of the gas generator and power turbine, the packing, the electric generator, the heat exchangers, and the saturator tower, as shown in eq. 3.2. Since this research focuses specifically on the power plant and not on the integration and architecture of the vessel, no account is made for auxiliary engines, cost of installation or electric drivers for the single or possibly multiple propellers.

$$PEC = \Xi_{\text{bare GT}} + \Xi_{\text{packing}} + \Xi_{\text{gen}} + \Xi_{\text{SAT}} + \sum \Xi_{\text{HE}} \quad (3.2)$$

The specific purchase equipment cost is defined as $SPEC = PEC/\hat{P}$ where $\hat{P}$ is the design power output of the cycle.

The acquisition cost (Ξ) of the bare gas turbine —i.e., no account for the generator or packing— is calculated as the sum of the cost of the compressors, turbines, combustion chambers and ancillaries:

$$\Xi_{\text{bare GT}} = \sum \Xi_{\text{comp}} + \sum \Xi_{\text{turb}} + \sum \Xi_{\text{comb}} + \Xi_{\text{ancillaries}} \quad (3.3)$$

The cost of the turbomachinery and generator is calculated as presented by Traverso et al. [69]. The cost of the ancillaries is assumed to be 40% of the cost of the turbomachinery and the cost of the packing is estimated to be 40% of the bare gas turbine as suggested by Kavanagh et al. [40].

Heat Exchangers

The construction geometry and the flow arrangement of the cycle heat exchangers have been selected and analysed by Orts-Gonzalez as reported in Brighenti et al. [70] and are summarised in table 3.1.

Table 3.1: Heat exchanger geometry and flow arrangement

	Geometry construction	Flow arrangement
Inter-cooler & After-cooler	<i>plate-fin:</i>	counter-current multi-pass cross-flow
	serrated fin — air side	
	plain fin — water side	
Recuperator	<i>plate-fin:</i>	counter-currant multi-pass cross-flow
	serrated-fin — both sides	
Economiser	<i>fin-tube:</i>	counter-currant multi-pass cross-flow
	fin tube — non-condensing section	
	plain tube — condensing section	

The serrated fin arrangement in the inter-cooler, after-cooler and recuperator is selected to maximise the heat transfer coefficient and enhance the structure reliability. On the water side plain fin arrangement is used to reduce the pumping power required for circulation and reduce the fooling and passage blocking risk. The plain tube arrangement is used in the section of the economiser where condensation occurs to ease the flow of the condensate water.

For exception of the recuperator in the reheated version of the cycle, all heat exchangers are made out of 90-10 Copper-Nickel alloy, in order to endure corrosion and organic fouling [71]. The hot pass in the recuperator, in the *reheated* version of the cycle, operates at temperature that exceed 1050 K. Therefore, Inconel 625 alloy is used for the high temperature pass of the recuperator, as previous research, among others, by McDonald [72], Maziasz [73], and Min [74] have found that Inconel 625 presents excellent creep resistance properties. The remaining passes of the recuperator are made out of stainless steel.

The detailed design of the heat exchangers was performed by Orts-Gonzalez and reported in Brighenti et al. [70]. The heat transfer area for all heat exchangers is calculated with ε -NTU method using correlations, from ESDU [75], specific to the geometry construction.

The heat transfer coefficient in the gas streams of the *plate-fin* heat exchangers are calculated according to Yang and Li [76], whereas the heat transfer coefficient for the water streams are calculated according to Rohsenow et al. [77] and ESDU [78] respectively for turbulent and laminar flow.

The heat transfer coefficient in the gas stream in the non-condensing section of the economiser is calculated according to ESDU [79], in the condensing section according to VDI Heat Atlas [80]. The heat transfer coefficient in the water stream calculated as reported in ESDU [78].

The acquisition cost of the heat exchangers is calculated using correlations based on the heat transfer area. For the *plate-fin* heat exchangers the formulation proposed by ESDU [81] is used with the material correction factors by Towler et al. [82] to account for the different alloys employed. The acquisition cost of the economiser is, instead, calculated according to Casarosa et al. [83]. All the material costs are update to the 2015 values by means of the chemical engineering plant cost index [84].

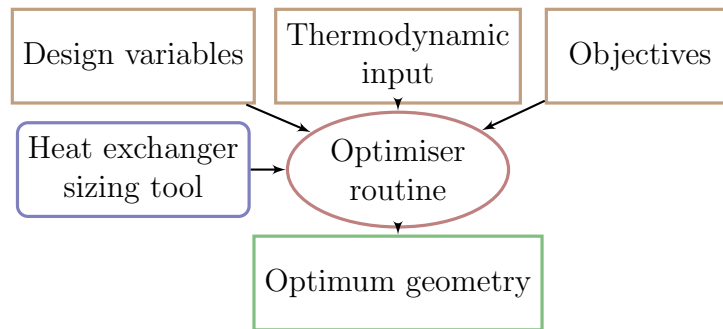


Figure 3.5: Heat exchangers design algorithm

Numerous heat exchanger designs can match a given heat exchanger performance. The tool developed by Orts-Gonzales [70] follows the algorithm shown in figure 3.5. A multi-objective optimisation of the heat exchangers design is performed to achieve a Pareto front, where compactness is traded-off for minimum acquisition cost. The genetic algorithm in MATLAB, based on Deb [85], was used for the optimisation process. The selection of the design from the Pareto front is based on the minimum value of the heat exchanger design selection parameter SP (see eq 3.4).

$$SP(x) = \sqrt{\sum_{i=1}^n \alpha_i \left(\frac{F_i(x)}{\max(F_i)} \right)^2}; \quad \sum_{i=1}^n \alpha_i = 1 \quad (3.4)$$

$F_i(x)$ represents the fitness of each individual in the Pareto —i.e., cost and perimeter—, and α_i represents the weighting coefficients. These coefficients allow the designer

to place more emphasis on any of the objectives when selecting the heat exchanger from the Pareto front. Their values were set to 0.5, in order to select the configuration that trades equally both objectives.

Saturator

The saturator tower is composed of a pressurised shell, the packed-bed, droplet eliminator, water-spray nozzles and other auxiliary components. However, the main dimension and cost of the tower depend on the size of the packing.

The cross sectional area of the packing is calculated so that the velocity of the gas is 60% of the flooding velocity and so that the pressure losses are never higher than 300 Pa/m as recommended by Dalili [51]. The height of the packing is calculated iteratively by imposing the inlet conditions of both the water and the dry air streams and by updating a guessed height until the outlet conditions of the two flows do correspond to the design values. The detailed assumptions and model description are further reported in appendix A.2.

The acquisition cost calculation of the tower is based on the weight of the shell and volume of the packing. The cost of the packing is assumed to be 3800 \$/m³ whereas the cost of the shell is assumed to be 8800 \$/ton [86]. An additional fixed cost of 14000 \$ is added for the installation of the droplet eliminator, water-spray nozzles and remaining auxiliary components [86].

3.4 Parametric Design Space Exploration — *Reheated* HAT

3.4.1 Methodology — *Reheated* HAT

Cycle analysis of the *reheated* HAT have shown that the maximum thermal efficiency is always achieved with the maximum technology level of the cycle components. Therefore, the impact of the technology level of the heat exchangers on the thermal efficiency, heat transfer area and capital investment cost is assessed by a parametrical study.

The design parameter that best describes the technology level of the heat exchangers is the effectiveness ($\varepsilon = Q/Q_{max}$). The maximum effectiveness of the water-air heat exchangers was assumed to 95%, whereas 90% was considered for the recuperator.

The design parameter that most affects the cost and size of the saturator tower is the pinch temperature difference (ΔT_{ps}) within the component, as it represents the driver for heat and mass transfer process within the tower. The minimum pinch temperature difference —i.e. maximum technology level— was assumed to be 5 K.

Cycle analysis of the *reheated* HAT have shown that the optimum OPR that maximises the thermal efficiency, for the imposed TIT, is beyond the technology level today available (OPR>50:1). The impact of reducing the OPR is studied as a design parameter that defines the technology level of the gas turbine.

The design envelope of the above-mentioned variables are summarised in table 3.2 and represent the range across which the parametric campaign was conducted, for a total of 486 cycle designs.

Table 3.2: Range of parameters across which the design space exploration is conducted.

Parameter		Range	
IC, AC, EC effectiveness	$\varepsilon = Q/Q_{max}$	[0.75 0.8 0.95]	(-)
REC effectiveness	$\varepsilon = Q/Q_{max}$	[0.8 0.85 0.9]	(-)
SAT pinch temperature difference	ΔT_{ps}	[5 10 15]	(K)
Overall pressure ratio	OPR	[30:1 40:1]	(-)

IC: inter-cooler, AC: after-cooler, EC: economiser, REC: recuperator, SAT: saturator

The remaining design variables are: the heat capacity ratio of the water-air heat exchangers, relative pressure ratio and saturator by-pass ratio and define respectively the water mass flow through the water-air heat exchangers (inter-cooler, after-cooler and economiser), the position of the inter-cooler within the compression process, and the air mass flow that by-passes the humidification process ($\beta = m_{g,SAT\ in}/m_{g,HPC\ out}$). These variables are optimised to maximise the thermal efficiency of the cycle. The single objective genetic algorithm in MATLAB® is used for the optimisation of the before mentioned design parameters. The bounds of the variables are reported in table 3.3

Table 3.3: Domain of free design variables.

Optimised design variable		Range
IC, AC, EC heat capacity ratio	$C^* = C_{min}/C_{max}$	[0.3 - 0.95]
Relative pressure ratio	$relPR = \log_{OPR} PR_{LPC}$	[0.25 - 0.75]
Saturator bypass	$\beta = m_{g,SAT\ in}/m_{g,HPC\ out}$	[0 - 1]

Therefore, the thermodynamic *design vector* comprise the effectiveness of the heat exchangers, the OPR and the humidifier pinch point as the free design variables,

and the heat capacity ratios of the water-air heat exchangers, the relative pressure ratio and the saturator bypass as the optimised design variables.

The cycles in this section are designed for fixed environmental conditions. The ambient temperature and pressure were respectively assumed to be 288.15 K and 101325 Pa, and the relative humidity of the air to be 80%. The sea water temperature was also assumed to be 288.15 K.

3.4.2 Results — *Reheated* HAT

A number of cycles have been selected to describe the effect of the design vector on the performance, cost and size of the power plant. The heat exchanger design vectors of the selected cycles (A→E) are shown in table 3.4 and figure 3.7. When referring to the cycles in the text, the overall pressure ratio is defined by the superscript L^{OPR} and the pinch temperature difference in the saturator by the subscript $L_{\Delta T_{ps}}$.

The cycle with maximum thermal efficiency is achieved with maximum effectiveness of all heat exchangers, maximum OPR and minimum pinch temperature

Table 3.4: Cycles nomenclature definition.

	$A^{\text{OPR}}_{\Delta T_{ps}}$	$B^{\text{OPR}}_{\Delta T_{ps}}$	$C^{\text{OPR}}_{\Delta T_{ps}}$	$D^{\text{OPR}}_{\Delta T_{ps}}$	$E^{\text{OPR}}_{\Delta T_{ps}}$	$F^{\text{OPR}}_{\Delta T_{ps}}$
ε_{IC}	0.95	0.75	0.95	0.95	0.95	0.75
ε_{AC}	0.95	0.95	0.75	0.95	0.95	0.75
ε_{EC}	0.95	0.95	0.95	0.75	0.95	0.75
ε_{REC}	0.9	0.9	0.9	0.9	0.8	0.8

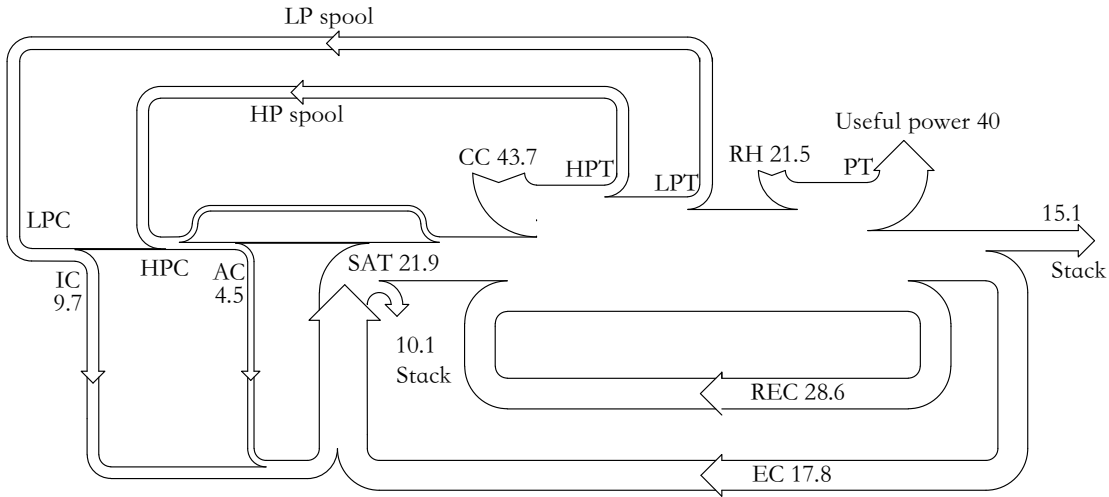


Figure 3.6: Sankey diagram of the *reheated* HAT cycle A_5^{40} .
Total enthalpy in MW

difference in the humidifier, hence maximum technology level (cycle A_5^{40}). The design vector as well as its performance are shown in table 3.5, and used as baseline in the results analysis.

Table 3.5: Design variables values of the baseline cycle A_5^{40} .

	Values	Constrains
Performance		
Thermal efficiency	61.3%	
Power output ($\hat{P}$)	40 MW	
Compressor		
Overall pressure ratio	40:1	
Relative pressure ratio (relPR = $\log_{\text{OPR}} \text{PR}_{\text{LPC}}$)	0.526	$0.1 < \text{relPR} < 0.9$
Combustion chambers		
Turbine inlet temperature	1600 K	
Heat exchangers		
Inter-cooler effectiveness (ε_{IC})	95%	
After-cooler effectiveness (ε_{AC})	95%	
Recuperator effectiveness (ε_{REC})	90%	
Economiser effectiveness (ε_{EC})	95%	
Inter-cooler heat capacity ratio (C_{IC}^*)	0.95	$0.3 < C_{\text{IC}}^* < 0.95$
After-cooler heat capacity ratio (C_{AC}^*)	0.95	$0.3 < C_{\text{AC}}^* < 0.95$
Economiser heat capacity ratio (C_{EC}^*)	0.72	$0.3 < C_{\text{EC}}^* < 0.95$
Saturator tower		
Pinch temperature difference (ΔT_{ps})	5 K	
Global parameters		
Saturator bypass (β)	50.2%	$0\% < \beta < 100\%$
Air inlet mass flow	40.89 kg/s	

The interaction between the components and the total enthalpy transferred within the cycle A_5^{40} can be appreciated in the Sankey diagram in figure 3.6. Within this cycle 28.6 MW are recovered within the recuperator. The heat exchanged between the gas and water in the inter-cooler, after-cooler and economiser is respectively 9.7 MW, 4.5 MW and 17.8 MW. 55.5% of the hot water injected in the saturator tower is generated in the economiser, whereas 30.4% and 14.1% is generated in the inter-cooler and after-cooler. Among the 32 MW exchanged in the inter-cooler, after-cooler and economiser only 21.9 MW are recovered within the cycle by means of heat and mass transfer in the saturator. The humidity achieved at the inlet of the combustion chamber is 0.196 kg_v/kg_a.

Impact of the Heat Exchanger Effectiveness on the Cycle Performance

The impact of the heat exchanger effectiveness on the cycle thermal efficiency is shown in figure 3.7, with constant OPR ($\text{OPR} = 40:1$) and pinch temperature difference in the saturator ($\Delta T_{ps} = 5 \text{ K}$). The x -axis and y -axis of the contour plots are, respectively, the after-cooler and economiser effectiveness, and the contour lines show the estimated cycle thermal efficiency. Moving vertically and horizontally through the contour plots, the effectiveness of the recuperator and inter-cooler vary respectively. Therefore, the top-right point of the top-right contour plot corresponds to the cycle A_5^{40} shown in table 3.5.

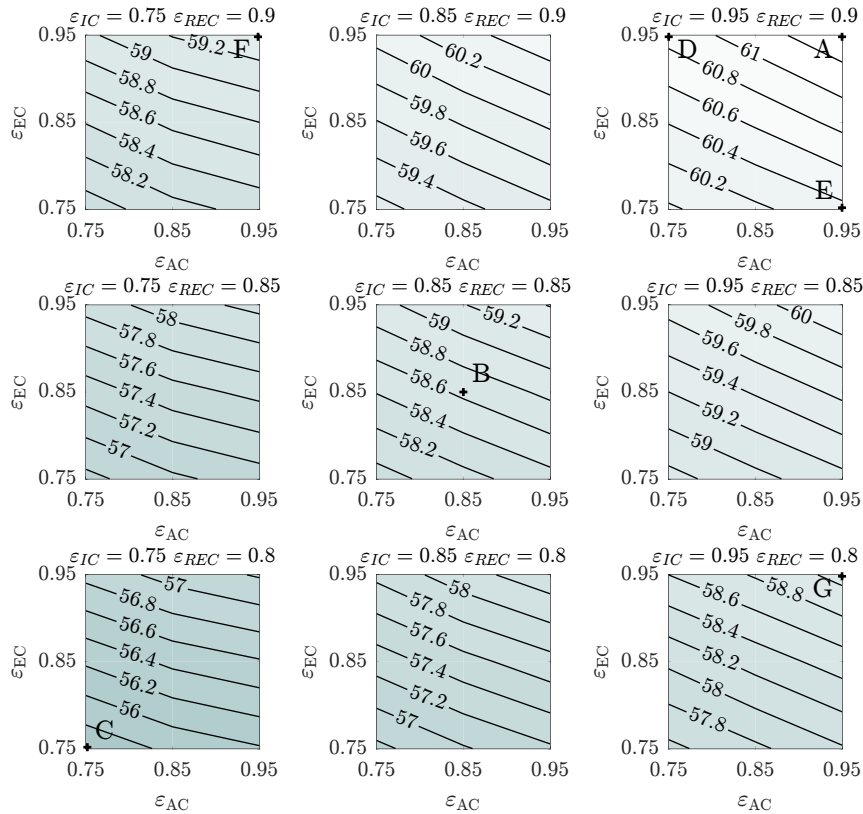


Figure 3.7: Cycle thermal efficiency in function of the heat exchanger effectiveness.

$\text{OPR} = 40:1$ & $\Delta T_{ps} = 5 \text{ K}$

The dependency between the thermal efficiency and the heat exchanger effectiveness remains linear when changing the effectiveness of all heat exchangers together. The comparison between cycles A_5^{40} and E_5^{40} highlights the impact of the recuperator effectiveness and results in a reduction of 2.28 pp in cycle thermal efficiency when recuperator effectiveness becomes 0.8 from 0.9. The impact of inter-cooler effectiveness is shown by comparing cycle A_5^{40} against B_5^{40} . A reduction in the effectiveness of the inter-cooler from 0.95 to 0.75 causes a reduction of 2 pp in the

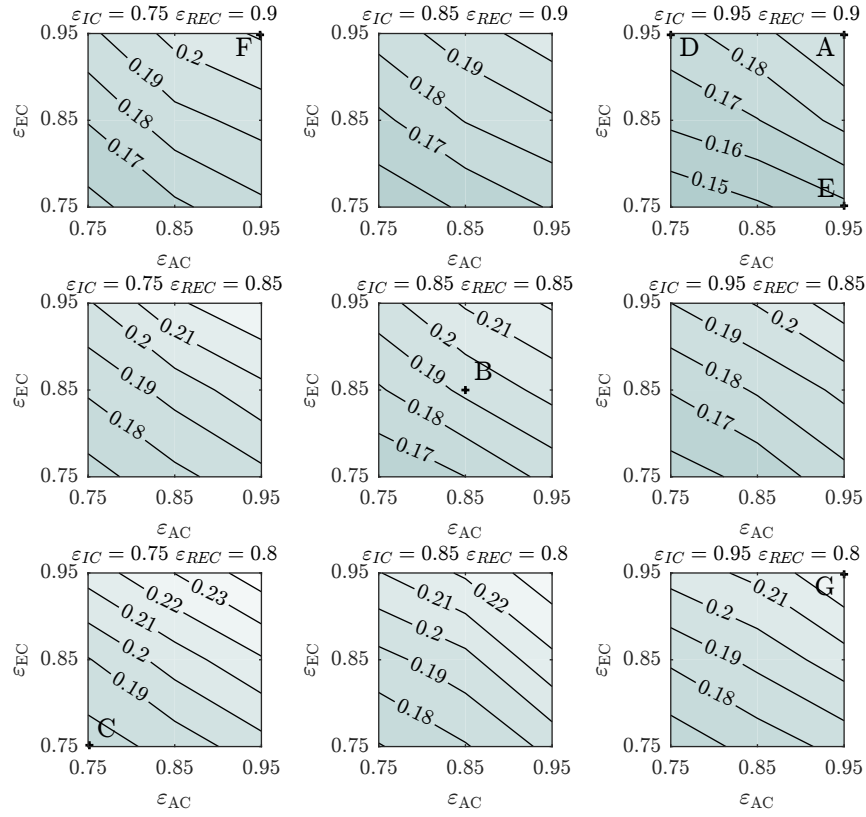


Figure 3.8: Absolute humidity [kg_v/kg_a] at the inlet of the combustion chamber across the design space as a function of the heat exchanger effectiveness.
 $\text{OPR} = 40:1$ & $\Delta T_{ps} = 5 \text{ K}$

cycle thermal efficiency. The impact of the after-cooler and economiser is shown by the comparison of cycle A_5^{40} with cycle C_5^{40} , and of A_5^{40} with cycle D_5^{40} , with respectively a reduction of 0.48 pp and 1 pp in thermal efficiency. The combined effect of the reduction in effectiveness of all the heat exchangers, is shown by the comparison of cycle A_5^{40} with cycle F_5^{40} with a drop in thermal efficiency by 5.5 pp.

Figure 3.8 shows the effect of the heat exchanger effectiveness on the absolute humidity at the inlet of the combustion chamber for constant values of overall pressure ratio ($\text{OPR} = 40:1$) and saturator pinch temperature difference ($\Delta T_{ps} = 5 \text{ K}$). The maximum humidity in the design space is $0.246 \text{ kg}_v/\text{kg}_a$, which ensures stable combustion in a modern combustion chamber as reported by experimental studies [47, 87].

The impact of the heat exchanger effectiveness on the optimised saturator bypass ratio, on the consequent humidity (ω) at the inlet of the combustion chamber and on cycle thermal efficiency is illustrated in figure 3.9 for constant $\text{OPR} = 40:1$ and $\Delta T_{ps} = 5 \text{ K}$.

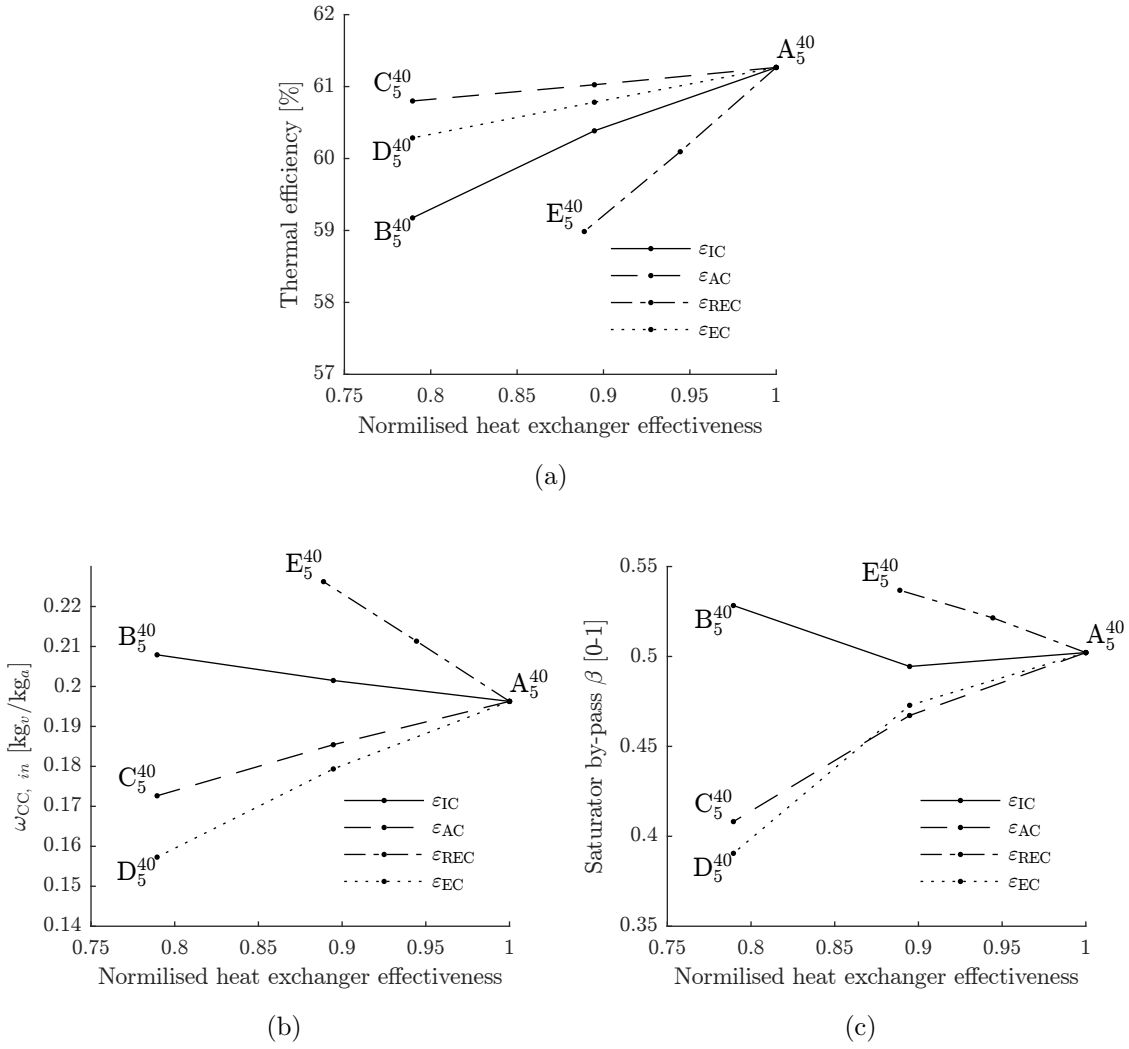


Figure 3.9: Influence of the effectiveness of the single heat exchanger on the: a) HAT thermal efficiency b) Humidity at the inlet of the combustion chamber c) Saturator by-pass ratio.
OPR = 40:1 & $\Delta T_{ps} = 5$ K

The reduction in effectiveness of the after-cooler and economiser has an impact on the amount of heat and mass transfer that occurs within the saturator, and therefore on the thermal efficiency of the cycle. For an after-cooler effectiveness of 75% ($A_5^{40} \rightarrow C_5^{40}$ – see figure 3.9), the water mass flow in the saturator is reduced by 0.5%, while a 1.2% reduction is observed in its temperature ($C_{AC}^* = 0.95$ across the range). The temperature of the gas at the inlet of the saturator increases by 13.8%, reducing the temperature leap in the saturator and, therefore the heat and mass transfer that occurs within the device. Hence, β decreases by 19% (figure 3.9c) — i.e. the fraction of mass flow that bypasses the saturator increases — and the humidity at the inlet of the combustion chamber also decreases from 0.196 kg_v/kg_a to 0.173 kg_v/kg_a (figure 3.9b). The reduction in heat recuperated at low

temperature in the cycle, due to the after-cooler, affects the thermal efficiency of the cycle, with a 0.48 pp reduction between cycles A_5^{40} and C_5^{40} (figure 3.9a).

When the economiser effectiveness reduces from 95% to 75% ($A_5^{40} \rightarrow D_5^{40}$ —see figure 3.9), the mass flow of the water heated in the heat exchanger falls by 22%, whereas the water outlet temperature falls only by 0.3% —i.e. C_{EC}^* increases. As for the after-cooler, the same reduction in effectiveness results in a decrease in β by 26% as shown in (figure 3.9c), and a reduction in humidity at the inlet of the combustion chamber from 0.196 to 0.15 (figure 3.9b). This reduction in heat recuperated in the cycle at low temperature, as for the after-cooler, entails a decrease in thermal efficiency of 1 pp between cycles A_5^{40} and D_5^{40} (figure 3.9a). For effectiveness of the economiser lower than 75% the C_{EC}^* remains constant with a value of 0.95. Therefore, β increases, whereas the humidity of the gas at the inlet of the combustion chamber continues to decrease (see figure 3.10).

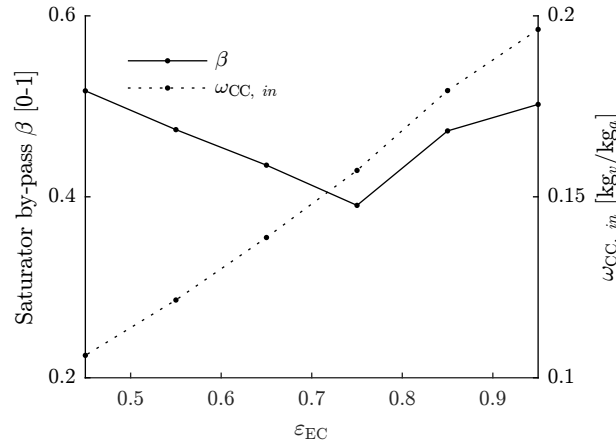


Figure 3.10: Impact of the economiser effectiveness on β and $\omega_{CC,in}$.
 OPR = 40:1, $\Delta T_{ps} = 5$ K, $\epsilon_{IC} = 0.95$, $\epsilon_{AC} = 0.95$, $\epsilon_{REC} = 0.9$

The effectiveness of the inter-cooler affects the amount of waste heat that is recuperated both at high and low temperatures within the cycle. When the effectiveness becomes 75% from 95% ($A_5^{40} \rightarrow B_5^{40}$ —see figure 3.9), the total amount of water injected in the saturator increases by 10% due to the increase of heat available in the gas at the inlet of the economiser —i.e. the temperature at the outlet of the recuperator on the hot side increases due to the increase in temperature of the gas at the inlet of the cold side. Therefore, β increases by 2% (figure 3.9a) — i.e. the fraction of mass flow that bypasses the saturator decreases— and the humidity at the inlet of the combustion chamber increases by 6% (figure 3.9b). This highlights the importance of the inter-cooling process, because not only the compression work required reduces but also the heat extracted from the gas between the two compressors is partly recovered in the saturator and not rejected

entirely as in most common inter-cooled gas turbines. Thus, the reduction of the effectiveness of the inter-cooler from 0.95 to 0.75 entails a reduction of 2 pp in the cycle thermal efficiency ($A_5^{40} \rightarrow B_5^{40}$ —see figure 3.9a).

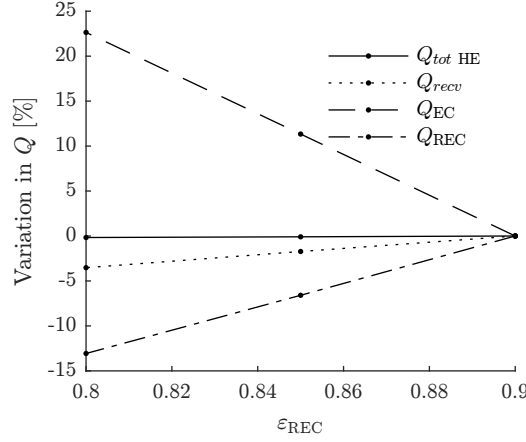


Figure 3.11: Impact of the recuperator effectiveness on the heat transferred in the heat exchangers.

$Q_{tot\ HE}$: sum of the heat exchanged in the IC, AC, REC & EC

Q_{recv} : heat recovered within the cycle —i.e., sum of heat exchanged in the REC & heat and mass exchanged in the SAT—

OPR = 40:1, $\Delta T_{ps} = 5$ K, $\varepsilon_{IC} = 0.95$, $\varepsilon_{AC} = 0.95$, $\varepsilon_{EC} = 0.95$

The reduction in effectiveness of the recuperator most affects the thermal efficiency of the cycle. When the effectiveness of the recuperator is 0.8 down from 0.9 the thermal efficiency drops by 2.28 pp (see figure 3.9). Figure 3.11 shows how the sum of the heat exchanged within the heat exchangers remains almost constant when the effectiveness of the recuperator is reduced. The heat that is not transferred in the recuperator is recuperated in the economiser. Nevertheless, as the system of economiser and saturator is less effective than the recuperator in recovering high quality heat, the heat recovered within the cycle —i.e., the sum of the heat transferred in the recuperator and saturator— drops by 4%. The water mass flow injected into the saturator increases by 10%, β increases by 4.7% (figure 3.9c), and the humidity at the inlet of the combustion chamber increases from 0.196 kg_v/kg_a to 0.226 kg_v/kg_a (figure 3.9b). The specific power increase as the air mass flow at the inlet of the cycle drop by 2 kg/s.

Figure 3.12 shows the impact of the heat exchanger effectiveness on the heat transfer coefficient times the heat transfer area (kxA) of each heat exchanger for constant overall pressure ratio (OPR = 40:1) and saturator pinch temperature difference ($\Delta T_{ps} = 5$ K). For similar inlet conditions, the variation in kxA of a heat exchanger is a useful parameter to evaluate the impact of the cycle *design vector* on the variation in size and cost independently from the design choices in

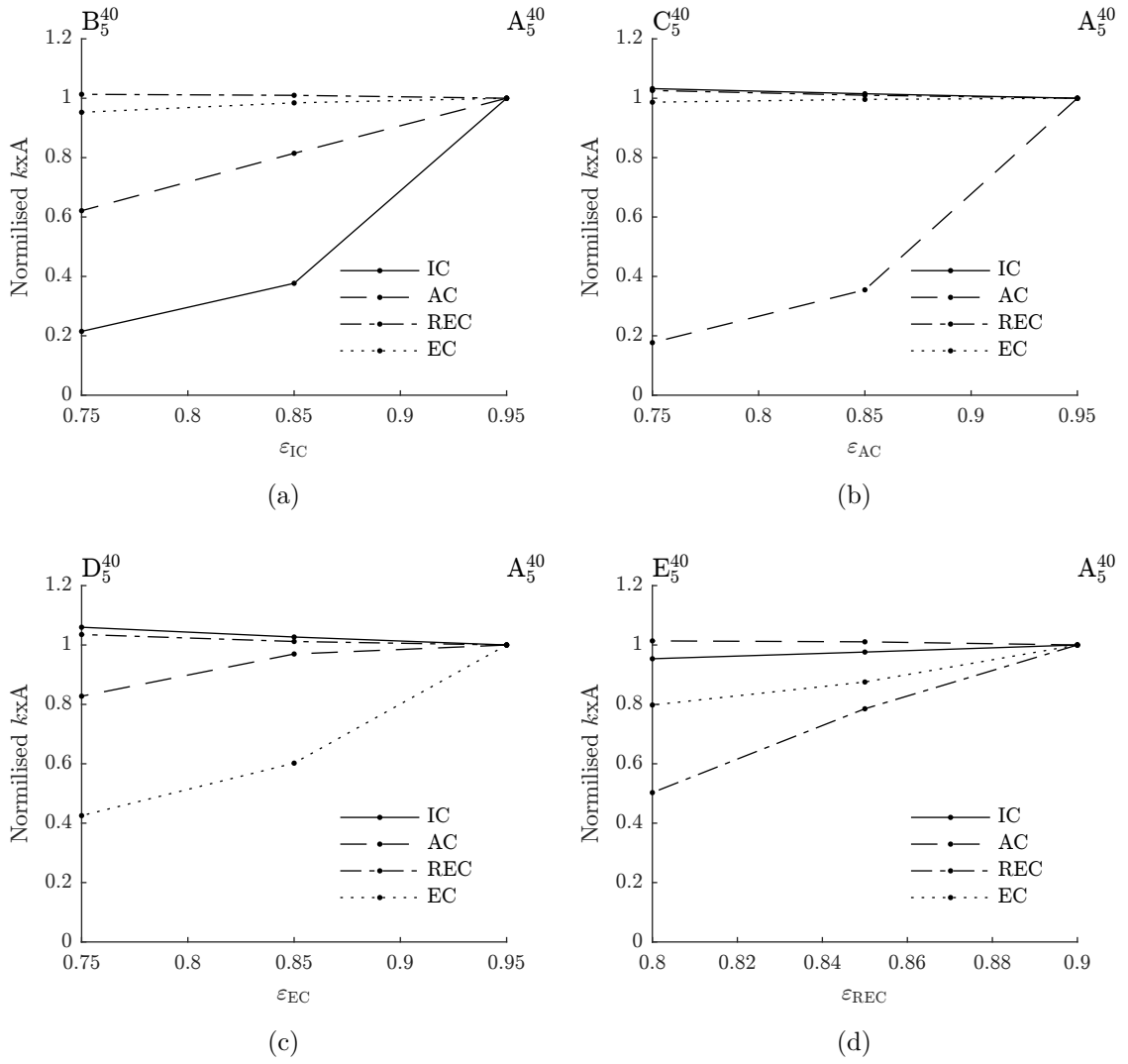


Figure 3.12: Influence of heat exchanger effectiveness, a) inter-cooler, b) after-cooler, c) economiser and d) recuperator, on the heat exchanger kxA .
 $OPR = 40:1$ & $\Delta T_{ps} = 5$ K

sizing the heat exchanger. Nonetheless, kxA should not be used to evaluate the differences in size between inter-cooler, after-cooler, economiser and recuperator due to the different inlet conditions of the two flows —i.e., different heat transfer coefficients. The values of kxA for each heat exchanger are normalised using the values of the reference cycle (see table 3.6).

The heat exchangers that are most affected by the reduction in effectiveness are the inter-cooler and after-cooler. A reduction in effectiveness of the latter heat exchangers from 0.95 to 0.75 results in a 79% ($A_5^{40} \rightarrow B_5^{40}$ —see figure 3.12a) and 82% ($A_5^{40} \rightarrow C_5^{40}$ —see figure 3.12b) reduction in the inter-cooler and after-cooler kxA .

Table 3.6: kxA values of the heat exchangers for the reference cycle A_5^{40} .

	Unit	kxA for the reference cycle
Inter-cooler	kW/K	545.5
After-cooler	kW/K	267.7
Economiser	kW/K	389.1
Recuperator	kW/K	379.4

The variation in effectiveness of a heat exchanger, however has a knock-on effect on the kxA of the remaining heat exchangers. The aftercooler kxA is strongly affected by the variation in the inter-cooler and economiser effectiveness, with a 38.5% and 22% reduction when effectiveness of these heat exchangers is 0.75 down from 0.95 ($A_5^{40} \rightarrow B_5^{40}$ in figure 3.12a and $A_5^{40} \rightarrow D_5^{40}$ figure 3.12c). In addition, the kxA of the economiser is strongly affected by the variation in effectiveness of the recuperator ($A_5^{40} \rightarrow E_5^{40}$), with a reduction of 21%. A light knock-on effect on the recuperator and inter-cooler kxA was found by the effectiveness changes in the economiser with respectively an increase in kxA by 3% and 6%.

Impact of the ΔT_{ps} on the Cycle Performance

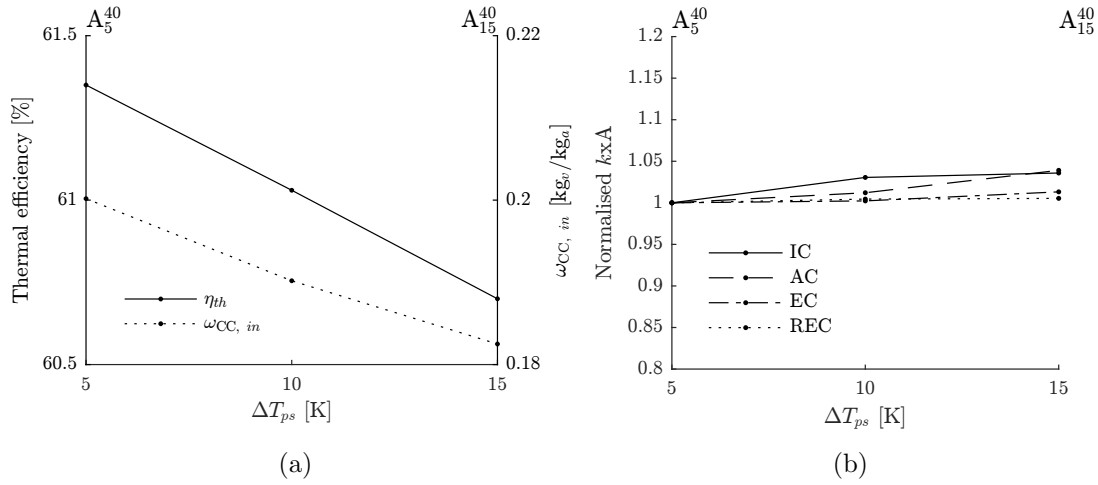


Figure 3.13: Influence of the saturator pinch temperature difference: (a) on the thermal efficiency and absolute humidity at the combustion chamber inlet and (b) on the kxA of the heat exchangers.

$$\text{OPR} = 40:1, \varepsilon_{IC} = 0.95, \varepsilon_{AC} = 0.95, \varepsilon_{EC} = 0.95 \text{ \& } \varepsilon_{REC} = 0.9$$

The increase of the saturator pinch temperature difference ΔT_{ps} from 5 K to 15 K ($A_5^{40} \rightarrow A_{15}^{40}$) has a linear effect on the thermal efficiency and on the absolute humidity of the air at the inlet of the combustion chamber (figure 3.13a), with a

reduction in thermal efficiency of 0.64 pp from the baseline cycle. The kxA of all heat exchangers is not significantly affected by the pinch temperature difference in the saturator (figure 3.13b), with a maximum increase of 4% in inter-cooler kxA for $\Delta T_{ps} = 15$ K. The increases in kxA (cycle $A_5^{40} \rightarrow A_{15}^{40}$) of the heat exchangers are attributed to the 3.2% higher gas mass flow required to produce the prescribed power output of 40 MW. This is triggered by a the lower water content in the hot section of the power plant.

Impact of the Overall Pressure Ratio on the Cycle Performance

The impact of the overall pressure ratio at design point on the performance and design of the cycle is also analysed. An optimum cycle configuration with thermal efficiency of 60.6% is achieved when reducing the overall pressure ratio from 40:1 to 30:1 ($A_5^{40} \rightarrow A_5^{30}$) —i.e., a reduction of 0.8 pp with regards to the baseline cycle with $OPR = 40:1$. The inlet gas mass flow required to deliver the prescribed power output of 40 MW increases by 10.4%. The cycle that achieves the maximum thermal efficiency with $OPR = 30:1$ features all heat exchanger with maximum

Table 3.7: Design variables values of the baseline cycle A_5^{30} .

	Values	Constrains
Performance		
Thermal efficiency	60.6%	
Power output	40 MW	
Compressor		
Overall pressure ratio	30:1	
Relative pressure ratio ($\text{relPR} = \log_{OPR} PR_{LPC}$)	0.54	$0.1 < \text{relPR} < 0.9$
Combustion chambers		
Turbine inlet temperature	1600 K	
Heat exchangers		
Inter-cooler effectiveness (ε_{IC})	95%	
After-cooler effectiveness (ε_{AC})	95%	
Recuperator effectiveness (ε_{REC})	90%	
Economiser effectiveness (ε_{EC})	95%	
Inter-cooler heat capacity ratio (C_{IC}^*)	0.95	$0.3 < C_{IC}^* < 0.95$
After-cooler heat capacity ratio (C_{AC}^*)	0.95	$0.3 < C_{AC}^* < 0.95$
Economiser heat capacity ratio (C_{EC}^*)	0.69	$0.3 < C_{EC}^* < 0.95$
Saturator tower		
Pinch temperature difference (ΔT_{ps})	5 K	
Global parameters		
Saturator bypass (β)	44.3%	$0\% < \beta < 100\%$
Air inlet mass flow	45.16 kg/s	

effectiveness and minimum pinch temperature difference in the humidifier as shown in table 3.7. The humidity levels at the inlet of the main burner is reduced by 10.2%, while the saturator bypass ratio β decreases by 6.1 pp when reducing the overall pressure ratio from 40:1 to 30:1. This is attributed to the lower water temperature at the outlet of the inter-cooler, after-cooler and economiser, and to the reduction in temperature of the gas at the outlet of the high pressure compressor.

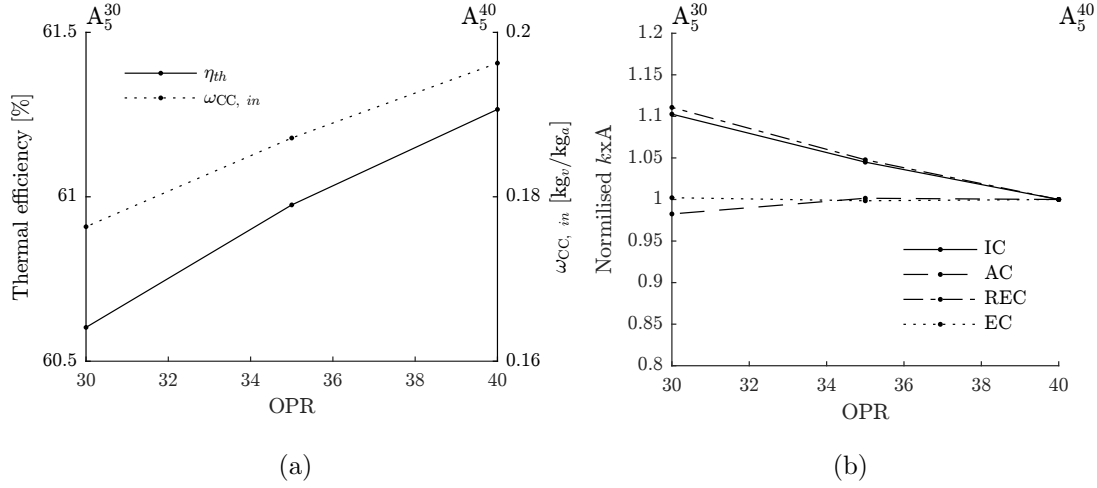


Figure 3.14: Influence of the overall pressure ratio: (a) on the thermal efficiency and absolute humidity at the combustion chamber inlet and (b) on the kxA of the heat exchangers.

$$\Delta T_{ps} = 5 \text{ K}, \varepsilon_{IC} = 0.95, \varepsilon_{AC} = 0.95, \varepsilon_{REC} = 0.9, \varepsilon_{EC} = 0.95$$

Although the reduction in overall pressure ratio implies cheaper and more compact turbomachinery, the decrease in specific power results in a 11.2% and 10.3% increase in kxA (figure 3.14) of the inter-cooler and recuperator respectively, whereas the after-cooler and economiser kxA decrease by approximately 4%.

Impact of Component Technology Level on SPEC

The variations in specific purchase equipment cost (SPEC) is referred to the base cycle A_5^{40} and the detailed purchase cost per component is reported in table 3.8.

The most expensive heat exchangers are the recuperator and inter-cooler, and account for respectively for 54.1% and 30% of the purchase cost of the heat transfer devices, whereas the saturator and the economiser are the cheapest heat transfer devices. The cost of the generator is constant as it is function of the only power output that is constant across the design space.

Table 3.8: Breakdown of the specific purchase equipment cost (SPEC) for the reference cycle A_5^{40} .

Cycle SPEC	$SPEC_{cycle}$	353.9 \$/kW
Gas turbine & packing	Ξ_{GT}	57.4% of PEC_{cycle}
Heat exchangers & saturator	Ξ_{HE}	23.3% of PEC_{cycle}
Inter-cooler	Ξ_{IC}	30.0% of Ξ_{HE}
After-cooler	Ξ_{AC}	12.0% of Ξ_{HE}
Recuperator	Ξ_{REC}	54.1% of Ξ_{HE}
Economiser	Ξ_{EC}	3.1% of Ξ_{HE}
Saturator	Ξ_{SAT}	1.8% of Ξ_{HE}
Generator	Ξ_{gen}	19.3% of PEC_{cycle}

Figure 3.15a illustrates the impact of the technology level of the components on the purchase cost of the heat exchanger devices, including the saturation tower, and on the purchase cost of the gas turbines including the ancillaries and packing. The nomenclature used to identify the cycles is reported in table 3.4. Figure 3.15b summarises the impact of the technology level of the components on both the specific purchase equipment cost (SPEC) and thermal efficiency.

It can be noticed that the reduction in the overall pressure ratio ($A_5^{40} \rightarrow A_5^{30}$) and effectiveness of the economiser ($A_5^{40} \rightarrow D_5^{40}$), and the increase in the saturator pinch temperature difference ($A_5^{40} \rightarrow A_{15}^{40}$) result in an increase in the SPEC of the cycle by respectively 3.8%, 1% and 0.8%. The reduction in OPR results in an increase in kxA of the inter-cooler and recuperator, as previously shown in figure 3.14b, and leads to an increase in the purchase cost of the heat exchangers by 9.9% (see figure 3.15a). Due to the reduction in specific power the cost of the gas turbine and packing also increases by 2.6%. The reduction in effectiveness of the economiser ($A_5^{40} \rightarrow D_5^{40}$) results in an increase in purchase cost of the heat exchangers (see figure 3.15a) by 1%, due to both the knock-on effect on the inter-cooler and recuperator (see figure 3.12c), and the marginal contribution of the economiser to the total acquisition cost of the heat exchangers. Moreover, the reduction in specific power results in an increase by 1.4% of the cost of the gas turbine and packing. The increase in ΔT_{ps} , due to the reduction in specific power and consequent knock-on effect on the other cycle components, results in an increase in purchase cost of both the heat exchangers and gas turbine by 1.4% and 0.8%.

Figure 3.15a shows how a reduction in effectiveness of the inter-cooler ($A_5^{40} \rightarrow B_5^{40}$) and after-cooler ($A_5^{40} \rightarrow C_5^{40}$) result in a drop in acquisition cost of the heat exchangers and saturator but also in an increase in purchase cost of the gas turbine

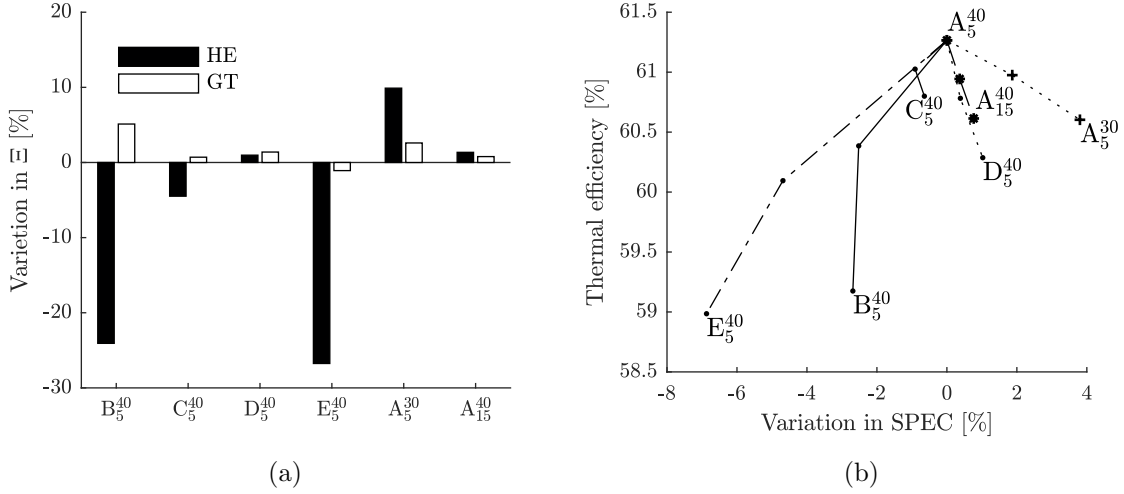


Figure 3.15: (a) Impact of the technology level of the components on the purchase cost of the heat exchangers and gas turbine (A_5^{40} reference cycle).
 (b) Impact of the technology level of the components on the thermal efficiency and cycle SPEC (A_5^{40} reference cycle).
 $HE=IC+AC+EC+REC+SAT$ & $GT=bare\ GT+ancillaries+packing$

and packing. Moreover, the variation in specific power, and therefore acquisition cost of the gas turbine, is linear to the effectiveness of either the inter-cooler or after-cooler whereas the cost of the heat exchangers is not linear to the latter — generally exponential. Therefore, a reduction in effectiveness of only the inter-cooler or after-cooler below a certain value would result in an increase in the SPEC of the power plant. This can be noticed in figure 3.15b when reducing the effectiveness of the after-cooler ($A_5^{40} \rightarrow C_5^{40}$): the SPEC increase when reducing the effectiveness from 0.85 to 0.75.

As mentioned in the previous section, the reduction in effectiveness of the recuperator ($A_5^{40} \rightarrow E_5^{40}$) results in a reduction in the kxA of the heat exchangers (see figure 3.12d) and a consequent 26.7% reduction in the acquisition cost of the latter and saturator (see figure 3.15a), but also in an increase in specific power that results in a 1.1% drop in purchase cost of the gas turbine and packing.

3.4.3 Conclusions — *Reheated* HAT

The parametric study of a *reheated* humid air turbine system conducted in the current work has shown the high efficiency potential of this cycle across the design space, with a maximum thermal efficiency of 61.4%, achieved with maximum technology level of the heat exchangers and saturator, and maximum overall pressure

ratio of the gas generator. The absolute humidity at the inlet of the combustion chamber across the design space is within the limits for flame stability, never exceeding $0.25 \text{ kg}_v/\text{kg}_a$.

The parameter that is found to primarily drive the thermal efficiency is the inter-cooler and recuperator effectiveness. For a change in inter-cooler effectiveness from 0.95 to 0.75, the thermal efficiency falls by 2 pp, whereas for the recuperator, a change from 0.9 to 0.8 causes a reduction in thermal efficiency of 2.28 pp. Cycle efficiency is not significantly affected by after-cooler and economiser effectiveness.

It was found that the recuperator is the only heat exchanger that when reduced its effectiveness results in both a cheaper gas-turbine and heat exchangers devices, due to the increase in specific power. The initial reduction in effectiveness of the inter-cooler and after-cooler also result in overall lower purchase cost of the heat transfer devices. However, the gas-turbine purchase cost increases as the specific power drops. Therefore, a reduction in effectiveness of only the inter-cooler or after-cooler below a certain value ($0.85 > \varepsilon_{AC} > 0.75$) would result in an increase in the SPEC of the power plant.

The reduction in OPR and effectiveness of the economiser, and the increase in the pinch temperature difference of the saturator, all result in a rise of the SPEC of the power plant. The knock-on effect, due the reduction in specific power, results in the increase in the purchase cost of both the heat exchangers and gas turbine.

3.5 Parametric Design Space Exploration — *Non-reheated* HAT

3.5.1 Methodology — *Non-reheated* HAT

Cycle analysis of the *non-reheated* HAT have shown that the maximum thermal efficiency is always achieved with the maximum technology level of the cycle heat exchangers. Therefore, the impact of the technology level of the latter on the thermal efficiency, heat transfer area and capital investment cost is assessed by a parametrical study.

As for the *reheated* HAT cycle, the effectiveness ($\varepsilon = Q/Q_{max}$) and the pinch temperature difference in the saturator (ΔT_{ps}) are selected as parameters that best describe the technology level of respectively the heat exchangers and saturator tower. The maximum effectiveness of the water-air heat exchangers was assumed to be 95%, whereas 90% was considered for the recuperator.

The design envelope of the above-mentioned variables are summarised in table 3.9 and represent the range across which the parametric campaign was conducted for a total of 243 cycle designs.

Table 3.9: Range of parameters across which the design space exploration is conducted.

Parameter		Range	
IC, AC, EC effectiveness	$\varepsilon = Q/Q_{max}$	[0.75 0.8 0.95]	(-)
REC effectiveness	$\varepsilon = Q/Q_{max}$	[0.8 0.85 0.9]	(-)
SAT pinch temperature difference	ΔT_{ps}	[5 10 15]	(K)

Contrary to what found for the *reheated* HAT, cycle analysis of the *non-reheated* HAT have shown that there is an optimum OPR that maximises the thermal efficiency for the imposed TIT, and is within the technology level today available (OPR<50:1). The optimum value of this parameter nevertheless depends on the technology level of the other cycle components. Therefore, along with remaining design variables —i.e., the heat capacity ratio of the water-air heat exchangers (C^*), relative pressure ratio (relPR) and saturator by-pass ratio (β)—, the OPR is optimised to maximise the thermal efficiency of the cycle. The single objective genetic algorithm in MATLAB[®] is used for the optimisation of the before mentioned design parameters. The bounds of the variables are reported in table 3.10.

Table 3.10: Domain of free design variables.

Optimised design variable		Range
IC, AC, EC heat capacity ratio	$C^* = C_{min}/C_{max}$	[0.3 - 0.95]
Relative pressure ratio	$relPR = \log_{OPR} PR_{LPC}$	[0.25 - 0.75]
Saturator bypass	$\beta = m_{g,SAT in}/m_{g,HPC out}$	[0 - 1]
Overall pressure ratio	OPR	[10:1 - 50:1]

Therefore, the thermodynamic *design vector* comprise the effectiveness of the heat exchangers and the humidifier pinch point as the free design variables and the heat capacity ratios of the water-air heat exchangers, the relative pressure ratio, the saturator bypass and the overall pressure ratio as the optimised design variables.

The cycles in this section are designed for fixed environmental conditions. The ambient temperature and pressure were respectively assumed to be 288.15 K and 101325 Pa, and the relative humidity of the air to be 80%. The sea water temperature was also assumed to be 288.15 K.

3.5.2 Results — *Non-reheated* HAT

A number of cycles have been selected to describe the effect of the design vector on the performance, cost and size of the power plant. The heat exchanger design vectors of the selected cycles (A→E) are shown in table 3.11 and figure 3.17. When referring to the cycles in the text, the pinch temperature difference in the saturator by the subscript $L_{\Delta T_{ps}}$.

Table 3.11: Cycles nomenclature definition.

	$A_{\Delta T_{ps}}$	$B_{\Delta T_{ps}}$	$C_{\Delta T_{ps}}$	$D_{\Delta T_{ps}}$	$E_{\Delta T_{ps}}$	$F_{\Delta T_{ps}}$
ε_{IC}	0.95	0.75	0.95	0.95	0.95	0.75
ε_{AC}	0.95	0.95	0.75	0.95	0.95	0.75
ε_{EC}	0.95	0.95	0.95	0.75	0.95	0.75
ε_{REC}	0.9	0.9	0.9	0.9	0.8	0.8

Table 3.12: Design variables values of the baseline cycle A_5^{40} .

	Values	Constrains
Performance		
Thermal efficiency	58.3%	
Power output ($\hat{P}$)	40 MW	
Compressor		
Overall pressure ratio	41.2:1	10:1 < OPR < 50:1
Relative pressure ratio (relPR = $\log_{OPR} PR_{LPC}$)	0.525	0.1 < relPR < 0.9
Combustion chambers		
Turbine inlet temperature	1600 K	
Heat exchangers		
Inter-cooler effectiveness (ε_{IC})	95%	
After-cooler effectiveness (ε_{AC})	95%	
Recuperator effectiveness (ε_{REC})	90%	
Economiser effectiveness (ε_{EC})	95%	
Inter-cooler heat capacity ratio (C_{IC}^*)	0.95	0.3 < C_{IC}^* < 0.95
After-cooler heat capacity ratio (C_{AC}^*)	0.95	0.3 < C_{AC}^* < 0.95
Economiser heat capacity ratio (C_{EC}^*)	0.93	0.3 < C_{EC}^* < 0.95
Saturator tower		
Pinch temperature difference (ΔT_{ps})	5 K	
Global parameters		
Saturator bypass (β)	62.2%	0% < β < 100%
Air inlet mass flow	49.7 kg/s	

The cycle with maximum thermal efficiency is achieved with maximum effectiveness of all heat exchangers and minimum pinch temperature difference in the

humidifier, hence maximum technology level (cycle A_5). The design vector as well as its performance are shown in table 3.12, and used as baseline in the results analysis. The maximum thermal efficiency of the *non-reheated* HAT, compared to the *reheated* HAT, is lower by 3 pp. The specific power also decreases by 17.7% due to the increase in air inlet mass flow.

The interaction between the components and the total enthalpy transferred within the cycle A_5 is shown in the Sankey diagram in figure 3.16. Within this cycle 21 MW are recovered within the recuperator. The heat exchanged between the gas and water in the inter-cooler, after-cooler and economiser is respectively 12 MW, 7.1 MW and 16.6 MW. 46.4% of the hot water injected in the saturator tower is generated in the economiser, whereas the remaining 33.6% and 20% is generated in the inter-cooler and after-cooler. Among the 35.7 MW exchanged in the inter-cooler, after-cooler and economiser only 24.6 MW are recovered within the cycle by means of heat and mass transfer in the saturator. The humidity achieved at the inlet of the combustion chamber is $0.186 \text{ kg}_v/\text{kg}_a$.

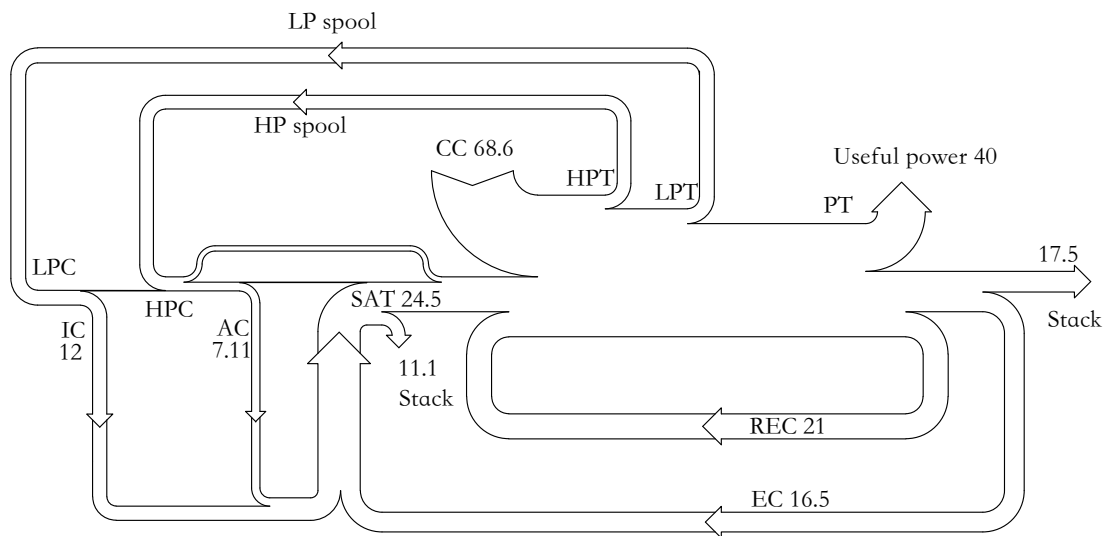


Figure 3.16: Sankey diagram of the *non-reheated* HAT cycle A_5 .
Total enthalpy in MW

Impact of the Heat Exchanger Effectiveness on the Cycle Performance

The impact of the heat exchanger effectiveness on the cycle thermal efficiency is shown in figure 3.17, for constant pinch temperature difference in the saturator of 5 K. The x -axis and y -axis of the contour plots are, respectively, the after-cooler and economiser effectiveness, and the contour lines show the estimated cycle thermal efficiency. Moving vertically and horizontally through the contour plots, the effectiveness of the recuperator and inter-cooler vary respectively. Therefore,

the top-right point of the top-right contour corresponds to the cycle A_5 shown in table 3.12.

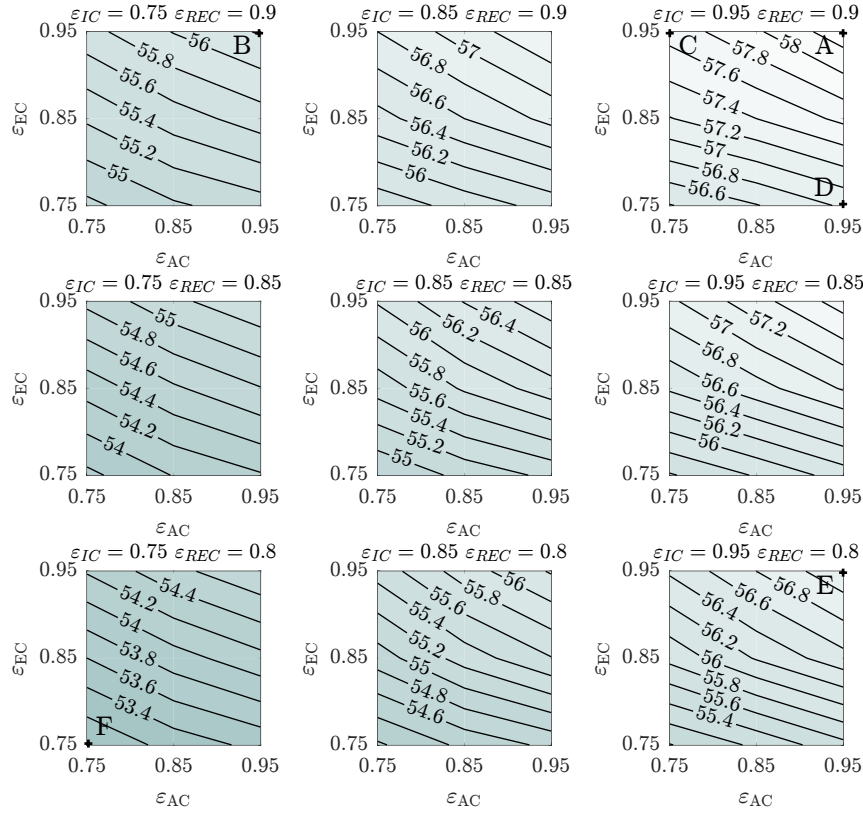


Figure 3.17: Cycle thermal efficiency in function of the heat exchanger effectiveness.

$$\Delta T_{ps} = 5 \text{ K}$$

The comparison between cycles A_5 and E_5 highlights the impact of the recuperator effectiveness and results in a reduction of 1.17 pp in cycle thermal efficiency when recuperator effectiveness is 0.8 down from 0.9. The impact of inter-cooler effectiveness is shown by the comparison cycle A_5 and B_5 . A reduction in the effectiveness of the inter-cooler from 0.95 to 0.75 causes a 2.1 pp reduction in the cycle thermal efficiency. The impact of the after-cooler and economiser is shown by the comparison of the cycle A_5 with the cycle C_5 , and of A_5 with the cycle D_5 , with respectively a 0.64 pp and 1.49 pp reduction in the thermal efficiency. However, the impact of the after-cooler decreases with a reduction in effectiveness of the economiser and increases with the reduction in effectiveness of the recuperator. The impact of the economiser, on the other hand, decreases with a reduction in effectiveness of the inter-cooler and increases with the reduction in effectiveness of the recuperator. The combined effect of the reduction in effectiveness of all the heat exchangers is shown by the comparison of the cycle A_5 with the cycle F_5 with

a drop of 5.3 pp in thermal efficiency. The humidity at the inlet of the combustion chamber in the reference cycle A_5 is $0.186 \text{ kg}_v/\text{kg}_a$. The maximum humidity in the design space is $0.221 \text{ kg}_v/\text{kg}_a$, which ensures stable combustion in a modern combustion chamber as reported by experimental studies [47, 87].

The impact of the heat exchanger effectiveness on the optimised saturator bypass ratio, on the absolute humidity (ω) at the inlet of the combustion chamber, on the overall pressure ratio and on cycle thermal efficiency is illustrated in figure 3.18 for constant $\Delta T_{ps} = 5 \text{ K}$.

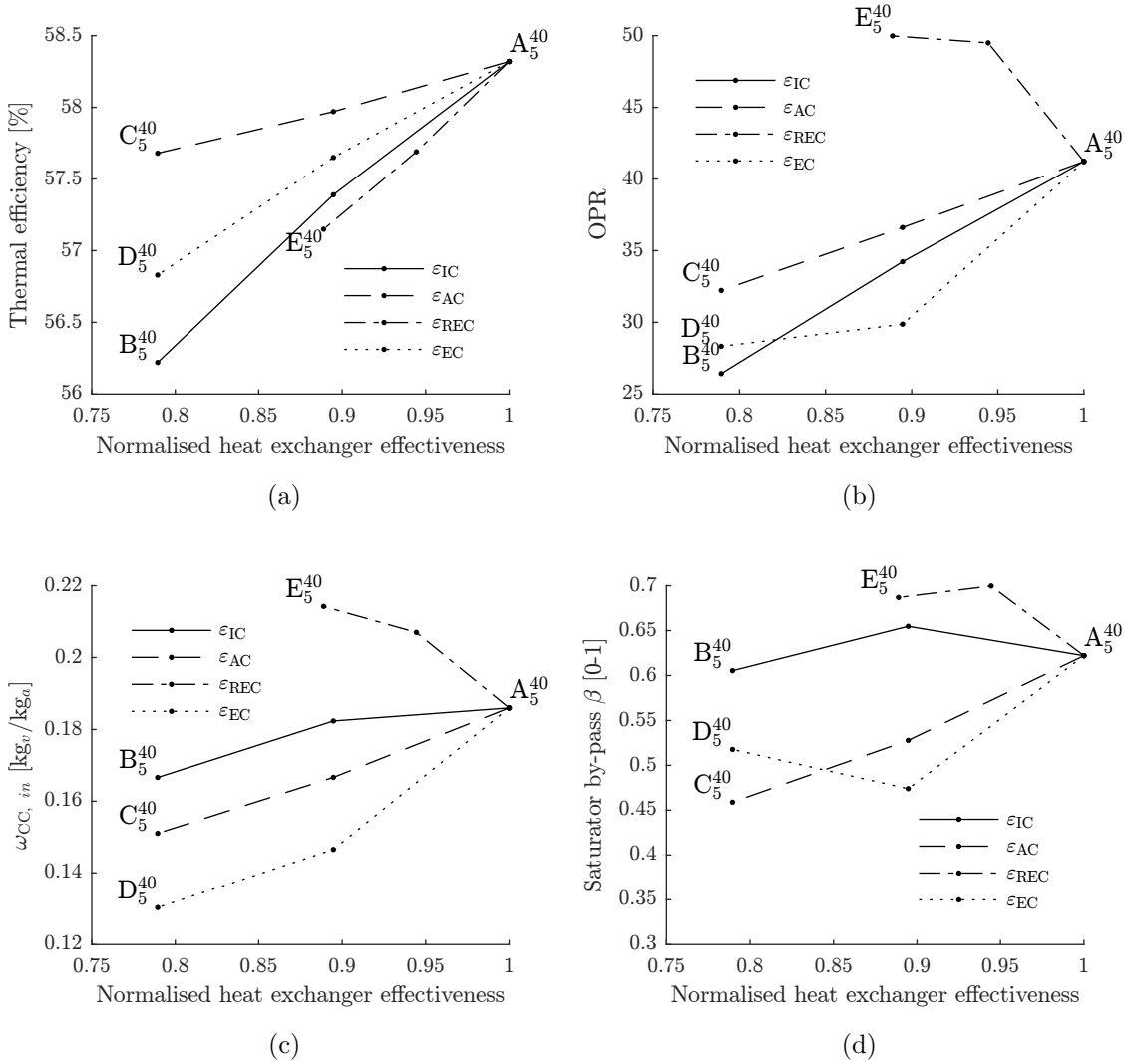


Figure 3.18: Influence of the effectiveness of the single heat exchanger on the:
a) HAT thermal efficiency b) Overall pressure ratio c) Humidity at the inlet of
the combustion chamber d) Saturator by-pass ratio.
 $\Delta T_{ps} = 5 \text{ K}$

The reduction in effectiveness of the after-cooler and economiser has an impact on the amount of heat and mass transfer that occurs in the saturator and therefore

on the thermal efficiency of the cycle. For an after-cooler effectiveness of 0.75 ($A_5 \rightarrow C_5$ — see figure 3.18), the water to gas mass flow ratio in the saturator ($m_{w,1}/m_{g,2}$) increases by 22.5%, while a 4% reduction in water temperature is observed ($C_{AC}^* = 0.95$ across the range). The temperature of the gas at the inlet of the saturator increases by 12.2%, reducing the temperature leap within the saturator and therefore the heat and mass transfer that occurs within the device. β drops by 26.2% (figure 3.18c) —i.e. the fraction of mass flow that bypasses the saturator increases— and the humidity at the inlet of the combustion chamber decreases from 0.186 kg_v/kg_a to 0.151 kg_v/kg_a (figure 3.18c). The reduction in effectiveness of the after-cooler also results in the drop of the optimal overall pressure ratio of the cycle from 41.2:1 to 32.2:1 (figure 3.18b). The reduction in heat recuperated at low temperature in the cycle, due to the after-cooler, affects the thermal efficiency of the cycle, with a 0.64 pp reduction between cycles A_5 and C_5 (figure 3.18a).

When the economiser effectiveness reduces from 95% to 75% ($A_5 \rightarrow D_5$ —see figure 3.18), the mass flow of the water heated in the heat exchanger increases by 9%, whereas the water outlet temperature falls by 7.4%. The heat capacity ratio in the economiser increases when the effectiveness of the latter component is decreased. When the effectiveness reduces from 95% to 85%, C_{EC}^* increases from 0.93 to 0.95 and remains constant for further reductions in effectiveness. As a consequence β initially decreases by 23.7% when the effectiveness is reduced from 95% to 85% but then increase by 9.2% when the effectiveness is further reduced from 85% to 75% as shown in figure 3.18d. The humidity at the inlet of the combustion chamber decreases from 0.186 kg_v/kg_a to 0.13 kg_v/kg_a (figure 3.18c) and the optimal overall pressure ratio drops from 41.2:1 to 28.3:1 (figure 3.18b). This reduction in heat recuperated within the cycle at low temperature entails a 1.49 pp decrease in thermal efficiency between cycles A_5 and D_5 (figure 3.18a).

The effectiveness of the inter-cooler most affects on the thermal efficiency of the cycle, with a 2.1 pp reduction in thermal efficiency when the effectiveness is reduced to 75% from 95% ($A_5 \rightarrow B_5$ —see figure 3.9a). The overall pressure ratio drops from 41.2:1 to 26.4:1 when the effectiveness of the inter-cooler is reduced from 95% to 75% (figure 3.18b). The C^* of both the economiser and after-cooler drops respectively from 0.93 to 0.80 and from 0.95 to 0.79 due to the lower maximum water temperature (see eq. 3.1) and higher heat available in the latter heat exchangers. This results in an increase of water injected in the saturator at a lower temperature. Therefore, contrary to what occurs in the design space of the *reheated* cycle, the reduction in effectiveness of the inter-cooler ($A_5 \rightarrow B_5$) results in a drop in humidity at the inlet of the combustion chamber from 0.186 kg_v/kg_a to 0.166 kg_v/kg_a (figure 3.18c).

The recuperator is only heat exchanger whose effectiveness reduction results in an increase in OPR of the cycle, from 41.2:1 to 50:1 —i.e., the higher boundary of the free design variable in the optimiser— when it is reduced to 80% from 90% (see figure 3.9). As for the *reheated* cycle, the heat that is not transferred in the recuperator is recuperated in the economiser. However, the heat recovered within the cycle —i.e., the sum of the heat transferred in the recuperator and saturator— reduces. The mass flow and the temperature of the water injected into the saturator increases respectively by 1.8% and 2.1%. The humidity at the inlet of the combustion chamber increases from 0.186 kg_v/kg_a to 0.214 kg_v/kg_a (figure 3.9b). The specific power increase as the air mass flow at the inlet of the cycle drop by 3.66 kg/s. Compared to the *reheated* HAT cycle, the impact of the recuperator of the *non-reheated* HAT on the thermal efficiency is diminished.

Table 3.13: kxA values of the heat exchangers for the reference cycle A_5 .

	Unit	kxA for the reference cycle
Inter-cooler	kW/K	673.9
After-cooler	kW/K	388.6
Economiser	kW/K	852.5
Recuperator	kW/K	478.9

Figure 3.19 shows the impact of the heat exchanger effectiveness on the heat transfer coefficient times the heat transfer area (kxA) of each heat exchanger for constant saturator pinch temperature difference ($\Delta T_{ps} = 5$ K). The values of kxA of each heat exchanger are normalised using the values of the reference cycle which is reported in table 3.6.

The reduction in effectiveness of the inter-cooler, after-cooler and economiser result in a reduction in kxA respectively by 75.5% ($A_5^{40} \rightarrow B_5^{40}$ —see figure 3.12a), 82.4% ($A_5^{40} \rightarrow C_5^{40}$ —see figure 3.12b) and 75.2% ($A_5^{40} \rightarrow D_5^{40}$ —see figure 3.12b) in relative terms when reducing the effectiveness of the latter from 0.95 to 0.75.

The knock-on effect that the variation of the effectiveness of a heat exchanger has on the others is more pronounced in the *non-reheated* cycle than in the *reheated* HAT. The kxA of the recuperator increases by 14%, 8% and 12% a result of the reduction in the effectiveness from 0.95 to 0.75 of respectfully the inter-cooler ($A_5 \rightarrow B_5$ in figure 3.19a), after-cooler ($A_5 \rightarrow C_5$ in figure 3.19b) and economiser ($A_5 \rightarrow D_5$ in figure 3.19c).

The reduction in effectiveness of the inter-cooler has a positive impact on the kxA of the after-cooler and economiser and both drop by 31.8% and 28.8% ($A_5 \rightarrow B_5$). The kxA of the inter-cooler is affected by the variation of the effectiveness of the

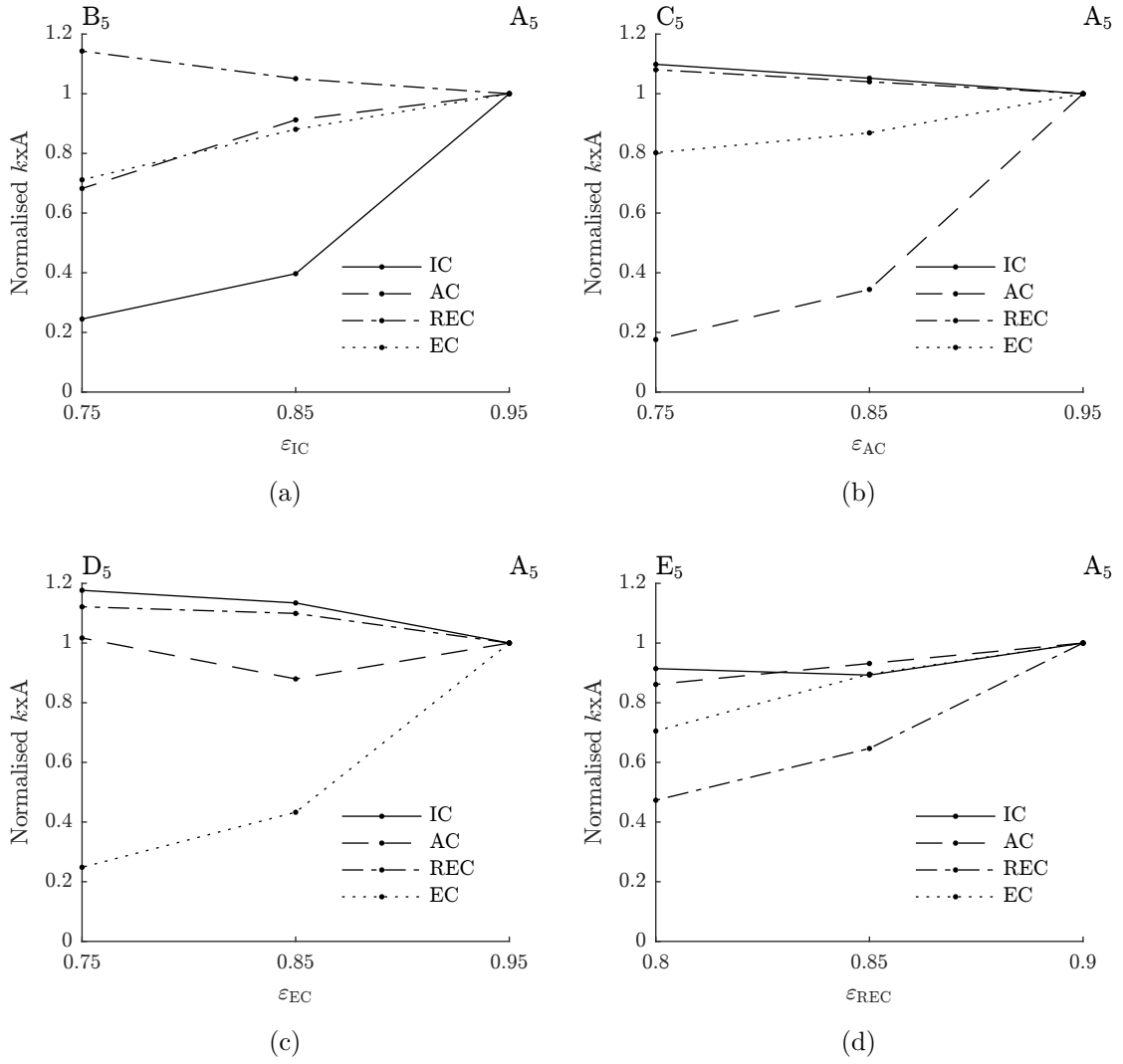


Figure 3.19: Influence of heat exchanger effectiveness, a) inter-cooler, b) after-cooler, c) economiser and d) recuperator, on the heat exchanger kxA .
 $\Delta T_{ps} = 5 \text{ K}$

after-cooler ($A_5 \rightarrow C_5$) and economiser ($A_5 \rightarrow D_5$), with an increase of 9.9% and 17.7%. The kxA of the economiser drops by 20% when the effectiveness of the after-cooler is reduced from 95% to 75%. As a result of the increase in specific power when the recuperator effectiveness is reduced from 90% to 80% (see figure 3.19d), the kxA of the inter-cooler, after-cooler and economiser decrease by 9.5%, 13.9% and 29.5%.

Impact of the ΔT_{ps} on the Cycle Performance

The increase in the saturator pinch temperature difference ΔT_{ps} from 5 K to 15 K ($A_5 \rightarrow A_{15}$) has a linear effect on the thermal efficiency (figure 3.20a), with a 0.64

pp reduction in thermal efficiency from the baseline cycle. The absolute humidity of the air at the inlet of the combustion chamber also drops from 0.186 kg_v/kg_a to 0.169 kg_v/kg_a.

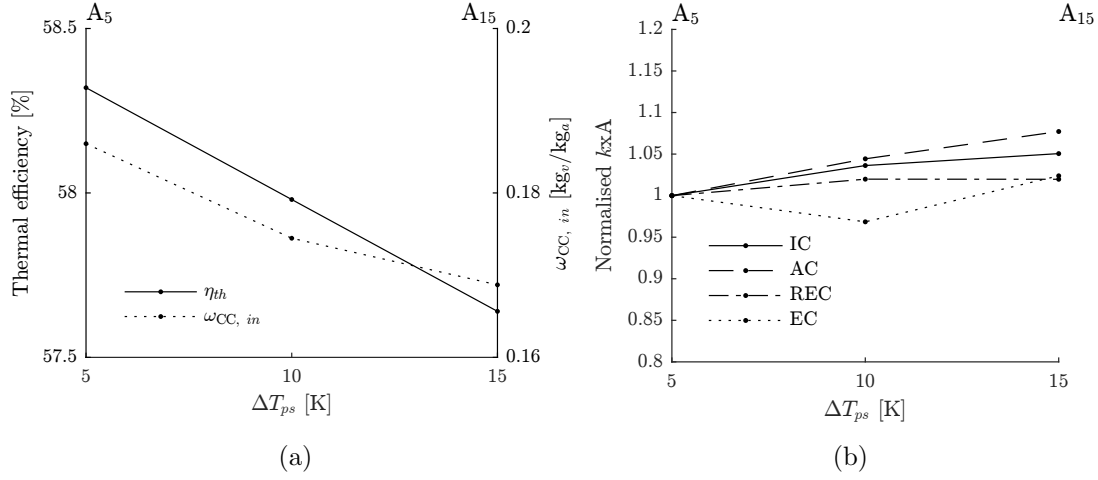


Figure 3.20: Influence of the saturator pinch temperature difference: (a) on the thermal efficiency and absolute humidity at the combustion chamber inlet and (b) on the kxA of the heat exchangers.

$$\varepsilon_{IC} = 0.95, \varepsilon_{AC} = 0.95, \varepsilon_{EC} = 0.95 \text{ \& } \varepsilon_{REC} = 0.9$$

The kxA of all heat exchangers increases as a consequence of the increase in the pinch temperature difference in the saturator (figure 3.20b), with a maximum increase of 7.7% in after-cooler kxA for $\Delta T_{ps} = 15$ K. The increases in kxA (cycle A₅→A₁₅) are attributed to the 2.7% increase in the inlet air mass flow required for the prescribed power output of 40 MW. The increase in inlet air mass flow is triggered by the lower water content in the hot section of the power plant. The initial reduction in kxA of the economiser is due to the initial reduction in C_{EC}^* due to the reduction of the maximum water temperature in the saturator. The overall pressure ratio also drops from 41.2:1 to 39.2:1.

Impact of Component Technology Level on SPEC

The variations in specific purchase equipment cost (SPEC) is referred to the base cycle A₅ and the detailed purchase cost per component is reported in table 3.14.

As for the *reheated* version of the HAT cycle, the most expensive heat exchangers are the recuperator and inter-cooler, that account for respectively 46.6% and 32.9% of the purchase cost of the heat transfer devises. The saturator and the economiser are the cheapest heat transfer devices. Moreover, it should be noticed that the heat and mass transfer purchase cost are 45% higher than in the *reheated* version of the cycle. This is due to the lower specific power and therefore larger volumes of

Table 3.14: Breakdown of the specific purchase equipment cost (SPEC) for the reference cycle A₅.

Cycle SPEC	SPEC _{cycle}	331.8 \$/kW
Gas turbine & packing	Ξ_{GT}	43.5% of PEC _{cycle}
Heat exchangers & saturator	Ξ_{HE}	36.0% of PEC _{cycle}
Inter-cooler	Ξ_{IC}	32.9% of Ξ_{HE}
After-cooler	Ξ_{AC}	15.0% of Ξ_{HE}
Recuperator	Ξ_{REC}	46.6% of Ξ_{HE}
Economiser	Ξ_{EC}	3.9% of Ξ_{HE}
Saturator	Ξ_{SAT}	1.6% of Ξ_{HE}
Generator	Ξ_{gen}	20.5% of PEC _{cycle}

the devices. The lower SPEC of the *non-reheated* HAT compared to the *reheated* cycle is a result of the absence of the reheater combustion chamber and therefore cheaper gas turbine.

Figure 3.21a illustrates the impact of the technology level of the components on the purchase cost of the heat exchanger devices —i.e., heat exchangers and saturator tower— and on the purchase cost of the gas turbines, ancillaries and packing. The nomenclature used to identify the cycles is reported in table 3.11. Figure 3.21b summarises the impact of the technology level of the components on both the specific purchase equipment cost (SPEC) and thermal efficiency.

It can be noticed that all the cycles B₅, C₅ and D₅ where the effectiveness of the water to air heat exchangers is reduced —i.e., inter-cooler, after-cooler and economiser—, the purchase cost of the gas turbine and packing increases by respectively 3.8%, 1.6% and 2.1%, due to the reduction in specific power. Moreover, the variation in specific power and therefore in acquisition cost of the gas turbine, is linear to the effectiveness of either the inter-cooler or after-cooler. The cost of the heat exchangers is not linear to the latter and is generally exponential. Therefore, a reduction in effectiveness of the inter-cooler or after-cooler below a certain value would result in an increase in the SPEC of the power plant. This can be noticed in figure 3.21b (see A₅→C₅), where the initial reduction in effectiveness of the after-cooler from 0.95 to 0.85 results in a reduction in SPEC of the power plant, whereas the reduction in effectiveness of the after-cooler from 0.85 to 0.75 results in an increase in SPEC.

As for the *reheated* HAT the specific power increases with a reduction in effectiveness of the recuperator. Therefore, the purchase cost of the gas turbine and packing drop by 1.14%, and the purchase cost of the heat exchangers and saturator

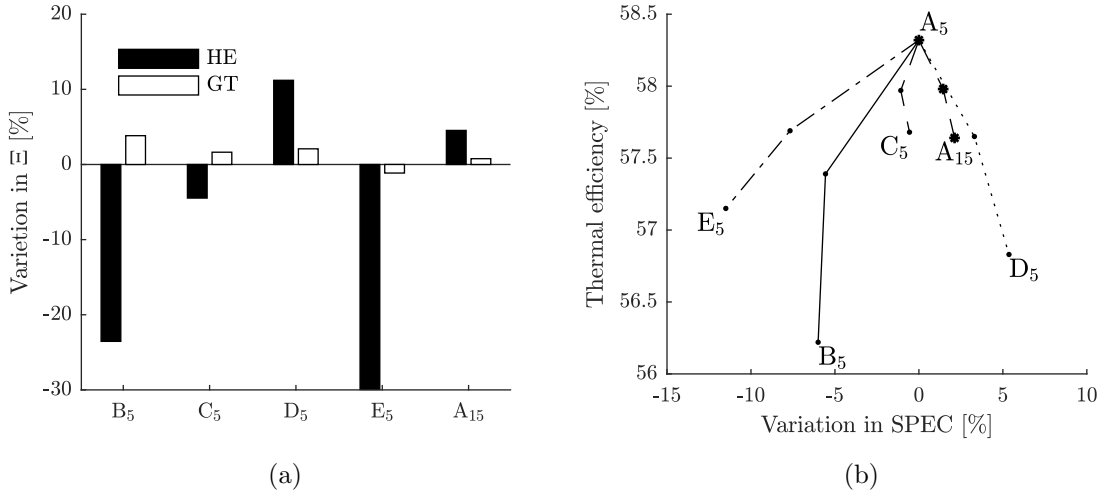


Figure 3.21: (a) Impact of the technology level of the components on the purchase cost of the heat exchangers and gas turbine (A₅ reference cycle).
 (b) Impact of the technology level of the components on the thermal efficiency and cycle SPEC (A₅ reference cycle).
 HE=IC+AC+EC+REC+SAT & GT=bare GT+ancillaries+packing

tower drops by 30% (A₅→E₅). As a result the SPEC of the power plant drops by 11.5%.

The reduction in effectiveness of the economiser (A₅→D₅) results in an increase in purchase cost of the heat exchangers (see figure 3.15a) by 11.21%, due to the knock-on effect on the inter-cooler and recuperator (see figure 3.19c) and due to the marginal contribution of the economiser to the total acquisition cost of the heat exchangers. Therefore, the combined increase of both the gas turbine and heat exchanger purchase cost results in a rise in SPEC of the power plant by 5.4% when reducing the effectiveness of the economiser from 95% to 75%. The increase in the saturator pinch temperature difference (A₅→A₁₅) also results in an increase in purchase cost of both the heat exchangers and gas turbine by 4.5% and 0.8%. As a result, the SPEC of the power plant increases by 2.1%.

The reduction in effectiveness of the heat exchangers in the *non-reheated* HAT has a stronger effect in reducing the SPEC of the power plant if compared to the *reheated* HAT. This can be noticed in figure 3.22 where the SPEC and thermal efficiency of selected cycles from the design space exploration of the *reheated* and *non-reheated* HAT are compared. The cycle with maximum thermal efficiency within the *non-reheated* HAT design space (A₅) has a 6.3% lower SPEC than the *reheated* cycle A₅⁴⁰ at the cost of a 4.8% lower thermal efficiency.

Figure 3.22 also shows how, for the same reduction in effectiveness of the recuperator or inter-cooler, the *non-reheated* HAT benefits from the highest reduction

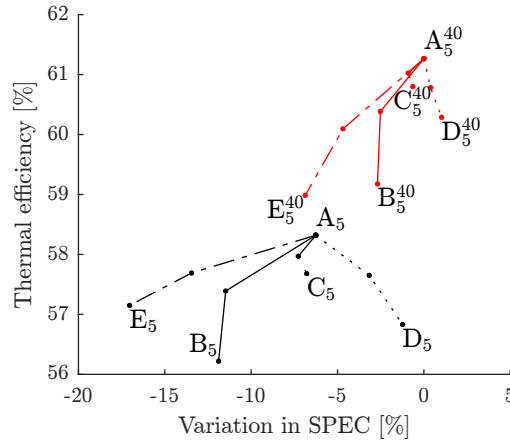


Figure 3.22: Comparison between the impact of the technology level of the components on the thermal efficiency and SPEC of the cycle of the *reheated* and *non-reheated* HAT cycles.

RED: *reheated*, BLACK: *non-reheated*, A_5^{40} reference cycle

in SPEC of the power plant. Moreover, the penalties in thermal efficiency for the *non-reheated* cycle are also lower, especially when reducing the effectiveness of the recuperator (cycle $A_5 \rightarrow E_5$). The purchase cost of the heat exchangers of the *non-reheated* HAT A_5 is 44% higher than for the *reheated* cycle and this explains why the variation in purchase cost of the heat exchangers has a higher impact on SPEC of the power plant.

3.5.3 Conclusions — *Non-reheated* HAT

The parametric study of a *non-reheated* humid air turbine system conducted in the current work has shown that the cycles achieves lower thermal efficiency than the *reheated* version across the whole design space. The maximum thermal efficiency of 58.3% is achieved with the maximum technology level of the heat exchangers and saturator and is 2.9 pp lower than for the correspondent *reheated* HAT with maximum thermal efficiency. The absolute humidity at the inlet of the combustion chamber across the design space for this cycle layout is also within the limits for flame stability, never exceeding $0.22 \text{ kg}_v/\text{kg}_a$.

The thermal efficiency of the *non-reheated* version of the cycle is found to be less sensitive to the reduction in effectiveness of the recuperator than the *reheated* versions of the cycle. A change in recuperator effectiveness from 0.95 to 0.75 results in a 1.17 pp drop in the thermal efficiency compared to the 2.28 pp reduction observed in the *reheated* HAT.

The parameters that were found to primarily drive the thermal efficiency are the inter-cooler and recuperator but also economiser effectiveness. A change in the

inter-cooler effectiveness from 0.95 to 0.75, similarly to the *reheated* HAT, results in a 2.1 pp reduction in the thermal efficiency, whereas for the economiser, the same change in effectiveness causes a 1.49 pp reduction in thermal efficiency. The cycle efficiency is least affected by after-cooler effectiveness, with a drop of 0.64 pp when the effectiveness is reduced from 0.95 to 0.75. Compared to the *reheated* layout of the cycle, the thermal efficiency is significantly more sensitive by lower effectiveness of the after-cooler and economiser.

The parametric study has shown that the technology level of each heat exchanger affects the thermal efficiency and purchase cost of the power system differently. The recuperator is found again to be the only heat exchanger that reduced its effectiveness results in the a cheaper gas-turbine and heat exchangers devices, due to the increase in specific power.

In power plants for commercial marine applications the fuel cost is usually the dominant cost voice, accounting for more than 85% of the annual machinery cost¹. Therefore, due to the considerably lower thermal efficiency of the *non-reheated* version of the HAT cycle, further analysis on the HAT cycle are focused on the *reheated* version.

3.6 Impact of the Design Ambient Conditions on the Cycle Design & Performance of the *reheated* HAT

In the previous section, all the design analysis of the cycles are based on the assumption that the maximum design power of 40 MW (design point) is achieved with air and the sea water temperature of 288.15 K (15 °C). However, the ambient and sea temperature change significantly depending on the latitude and if navigations is in open or closed seas. Therefore, depending on the navigation routes the design values of the air and water temperatures may significantly vary. In this section the sensitivity of the design thermal efficiency and purchase equipment cost of the *reheated* HAT power plant to the ambient and sea temperature are evaluated.

¹annual machinery cost = CAPEX + OPEX

3.6.1 Methodology

The analysis of the impact of air and water temperature on the design and performance of the *reheated* cycle is performed by keeping the design parameters that defined cycle A_5^{40} in section 3.4 constant and are reported in table 3.15.

Table 3.15: Constant design parameters in the analysis of the impact of air and water temperature on the design and performance of the *reheated* HAT.

Parameter		Value	
IC, AC, EC effectiveness	$\varepsilon = Q/Q_{max}$	0.95	(-)
REC effectiveness	$\varepsilon = Q/Q_{max}$	0.9	(-)
SAT pinch temperature difference	ΔT_{ps}	5	(K)
Overall pressure ratio	OPR	40:1	(-)

The remaining cycle design variable —i.e., heat capacity ratios of the water-air heat exchangers, the relative pressure ratio and the saturator bypass— are optimised to maximise the thermal efficiency of the cycle for the different sea and ambient temperatures. As in section 3.4, the single objective genetic algorithm in MATLAB® is used for the optimisation of the before mentioned design parameters. The bounds of the variables are reported in table 3.3 (see section 3.4.1).

The impact of the sea water and ambient temperature on the design performance of the cycle is studied as following: (i) the only ambient temperature is varied from 278.15 K to 298.15 K by steps of 5 K, whereas the sea water temperature is kept constant at 288.15 K, (ii) the only sea water temperature is varied from 278.15 K to 298.15 K by steps of 5 K, whereas the ambient temperature is kept constant at 288.15 K and (iii) both ambient and sea water temperatures are changed contemporary from 278.15 K to 298.15 K by steps of 5 K.

3.6.2 Results — Reheated Cycle

The outcome of the design performance sensitivity of the cycle to the ambient and water sea temperature is analysed using as reference the *reheated* cycle A_5^{40} , with air and water temperature of 288.15 K. The design vector and detailed performance of this cycle are reported in table 3.5, whereas SPEC and the break-down of the purchase equipment cost of the power plant are reported in table 3.8, both in section 3.4.2.

The sensitivity of the performance of the *reheated* HAT to the variation of the design ambient and sea water temperature are shown in figure 3.23. The thermal efficiency is most affected by the variation of the ambient air design temperature

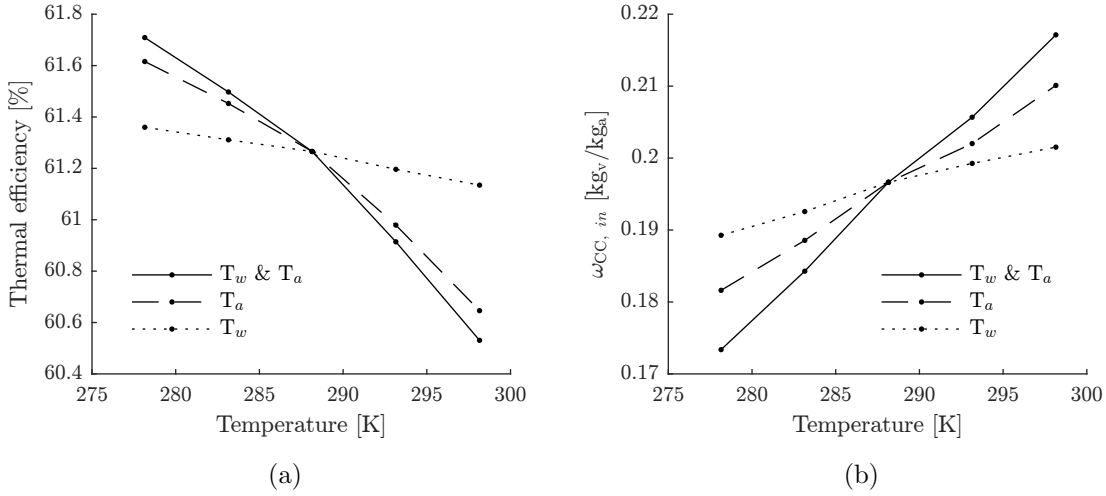


Figure 3.23: Influence of sea water and ambient temperature on: (a) HAT thermal efficiency (b) Humidity at the inlet of the combustion chamber.
 T_a : T_a varies and $T_w = 288.15$ K, T_w : T_w varies and $T_a = 288.15$ K, and T_a & T_w : T_a and T_w vary.

(see figure 3.23a). The 40 MW cycle, designed for ambient temperature of 278.15 K achieves thermal efficiency 0.35 pp higher than if design for 288.15 K, whereas and increase in design ambient temperature from 288.15 K to 298.15 K results in a 0.62 pp drop in thermal efficiency —i.e., sea water temperature in both cases is 288.15 K. The design thermal efficiency of the cycle is only marginally affected by the design sea water temperature. A gain of 0.09 pp and a drop by 0.14 pp are respectively achieved when the design water temperature is changed from 288.15 K to 278.15 K and from 288.15 K to 298.15 K —i.e., ambient temperature in both cases is 288.15 K. When both design ambient and sea water temperature are both changed from 288.15 K to 278.15 K and from 288.15 K to 298.15 K the respective impact on the thermal efficiency is an increase by 0.44 pp and a reduction by 0.74 pp.

The main driver for the change in the thermal efficiency, when both ambient and water temperature are changed, is the variation in power required by the compressors. When the design ambient and sea water temperature are respectively reduced from 288.15 K to 278.15 K the power of the gas generator drops by 1.7% and 0.5%. In the case of the reduction in air temperature this is the result of the air temperature drop at the inlet of both compressors, whereas in the case of the reduction in water temperature this is due to the only reduction in temperature of the air at the outlet of the inter-cooler —i.e., inlet of the HP compressor.

The humidity at the inlet of the main combustion chamber (see figure 3.23b) drops with both the reduction of the design ambient temperature and the sea

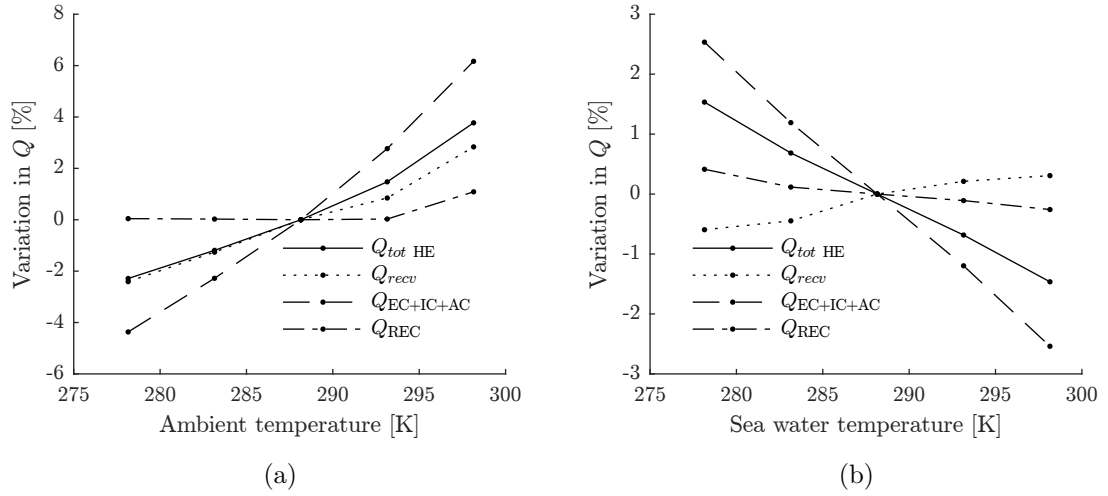


Figure 3.24: Influence of sea water and ambient temperature on the heat recovered within the cycle: (a) T_a varies and $T_w = 288.15$ K, (b) T_w varies and $T_a = 288.15$ K.

Q_{tot} HE: sum of the heat exchanged in the IC, AC, REC & EC

Q_{recu} : heat recovered within the cycle —i.e., sum of heat exchanged in the REC & heat and mass exchanged in the SAT—

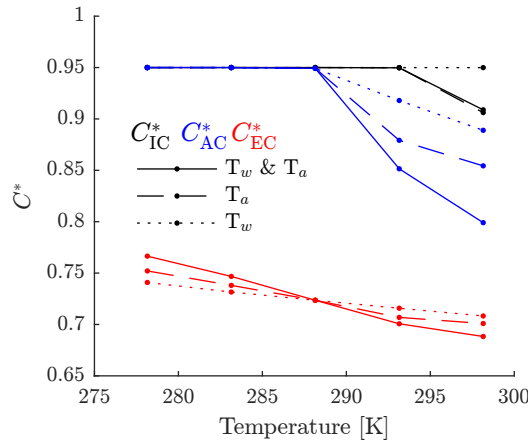


Figure 3.25: Influence of sea water and ambient temperature on the heat capacity ratio within the gas-water heat exchangers.

BLACK: heat capacity ratio of the inter-cooler, BLUE: heat capacity ratio of the after-cooler, RED: heat capacity ratio of the economiser

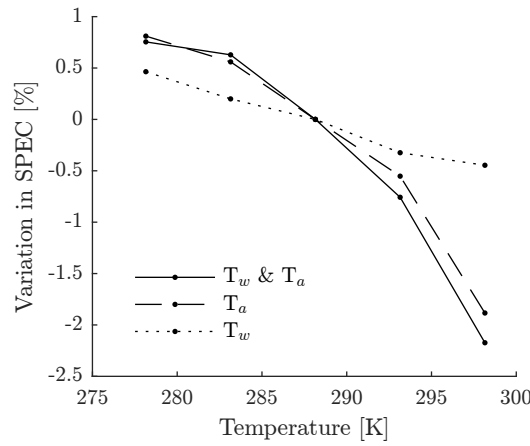
T_a : T_a varies and $T_w = 288.15$ K, T_w : T_w varies and $T_a = 288.15$ K, and T_a & T_w : T_a and T_w vary.

water temperature. However, the humidity within the cycle is most sensitive to the design ambient temperature. The variation in humidity is the result of the variation in heat recovered within the cycle —i.e., heat and mass transferred in the saturator and heat transferred in the recuperator. Figure 3.24 shows the variation in total heat exchanged within the heat exchangers and the heat recovered within

the cycle when ambient (figure 3.24a) and sea water (figure 3.24b) temperature are changed. In figure 3.24a it can be noticed that the increase in design ambient temperature, from 288.15 K to 298.15 K (sea water temperature is constant at 288.15 K), results in a 3.8% increase in total heat exchanged in the heat exchangers, but also a 2.8% increases in the amount of heat recovered within the cycle. The temperature of the gas at the inlet of the inter-cooler, after-cooler and economiser increases and therefore also the difference between the inlets of the water-gas heat exchangers increases. The heat capacity in the economiser drops (see figure 3.25) from 0.72 when the ambient temperature is 288.15 K to 0.70 when ambient temperature is 298.15 K and as a result the amount of hot water generated per unit of air mass flow in the gas turbine increases. For ambient temperatures above 288.15 K for the after-cooler and above 293.15 K for the inter-cooler, the increase of ambient temperature results in the decrease in the heat capacity ratio within these heat exchangers. Therefore, the mass flow of hot water injected in the saturator rises —i.e., the temperature of the water injected in the saturator remains constant. This enhances the evaporation rate in the saturation tower when the design ambient air temperature is increased. The humidity increases to 0.210 kg_v/kg_a when the ambient temperature is 298.15 K, whereas reduces to 0.182 kg_v/kg_a when the ambient temperature is 278.15 K (see figure 3.23b).

If the design water temperature is increased from 288.15 K to 298.15 K, the heat recovered within the cycle increases by 0.3% (see figure 3.24b). However, the heat exchanged in the heat exchangers drops by 1.5% as a result of the decrease in temperature difference between the inlet flows of the heat exchanger. The heat capacity in the economiser (see figure 3.25) drops from 0.72 when the ambient temperature is 288.15 K to 0.71 when ambient temperature is 298.15 K. As a consequence, the evaporation rate in the saturator is enhanced. For sea water temperatures above 288.15 K the heat capacity ratio within the after-cooler also decreases enhancing the generation hot water then injected in the saturator. The humidity increases to 0.201 kg_v/kg_a when the sea water temperature is 298.15 K, whereas reduces to 0.189 kg_v/kg_a when the sea water temperature is 278.15 K (see figure 3.23b).

The specific purchase cost of the power plant is only marginally affected by the reduction in both sea water and ambient design temperature (see figure 3.26). The reduction in the water design temperature from 288.15 K to 278.15 K results in a 0.46% increase in SPEC, whereas for the same reduction in design ambient temperature the SPEC increases by 0.81%. When both ambient and sea water temperature are decreased to 278.15 K a 0.75% increase in SPEC is observed.



(a)

Figure 3.26: Influence of sea water and ambient temperature on the specific equipment cost (SPEC).

$\mathbf{T}_a$: T_a varies and $T_w = 288.15$ K, $\mathbf{T}_w$: T_w varies and $T_a = 288.15$ K, and $\mathbf{T}_a \& \mathbf{T}_w$: T_a and T_w vary.

The SPEC is most sensitive to the increase in ambient temperature above 293.15 K. A reduction by 1.9% in SPEC is observed when the ambient temperature is increased from 288.15 K to 298.15 K, whereas a drop by 2.2% is observed when both ambient and sea water temperatures are increased by the same amount. The purchase cost of the inter-cooler is the main contributor to such reduction when the design ambient temperature exceeds 293.15 K. The reduction in heat capacity observed in figure 3.25 results in a reduction of the purchase cost of the inter-cooler, that drops by 21.6%. The purchase cost of the after-cooler also drops when the heat capacity ratio of the latter heat exchanger is diminished.

3.6.3 Conclusions

The study of the impact of design ambient and sea water temperature on the performance of the *reheated* humid air turbine has shown that the thermal efficiency and the specific purchase equipment cost of the power plant are most affected by the variation in ambient temperature.

The cycle designed to operate at ambient and sea water of 278.15 K achieves a design thermal efficiency of 61.7% —0.6 pp higher than if designed for ambient and sea water temperature of 288.15 K. However, the SPEC of the power plant increases by 0.75%. When design sea water and ambient temperature is 298.15 K the maximum design thermal efficiency is 60.5% —0.74 pp lower than if designed for ambient and sea water temperature of 288.15 K. Nevertheless, the SPEC of the power plant results lower by 2.17%.

The humidity at the inlet of the combustion chamber reduces when both the design temperatures of the sea water and ambient are reduced. The humidity drops to $0.173 \text{ kg}_v/\text{kg}_a$, when the sea and ambient design temperature is 278.15 K, and rises to $0.217 \text{ kg}_v/\text{kg}_a$ when the sea and ambient design temperature is of 298.15 K.

3.7 Summary

An optimisation scheme for the exploration of both the *reheated* and *non-reheated* HAT cycle design spaces has been developed, for a cycle with maximum power output of 40 MW. The design parameters that reflect the technology level of the components have been selected and were varied parametrically to quantify the trade-off in thermal efficiency and specific purchase cost of the power plant, with the remaining design variables optimised for maximum thermal efficiency. The maximum thermal efficiency achieved within the design space of 61.2% and 58.3% respectively for the *reheated* and *non-reheated* HAT are both achieved with the maximum technology level of all heat exchangers. The SPEC of the *non-reheated* HAT with maximum thermal efficiency was found to be 6.3% lower than for the *reheated* cycle with maximum thermal efficiency.

The main outcomes of the design space exploration of the HAT cycles are the following:

- In the *reheated* HAT, the parameter that was found to primarily drive the thermal efficiency is the inter-cooler and recuperator effectiveness. For a change in inter-cooler effectiveness from 0.95 to 0.75, the thermal efficiency falls by 2 pp, whereas for the recuperator, a change from 0.9 to 0.8 causes a 2.28 pp reduction in thermal efficiency. Cycle efficiency is not significantly affected by after-cooler and economiser effectiveness.
- In the *non-reheated* HAT, the main parameters that affect the thermal efficiency are the effectiveness of the inter-cooler, recuperator and economiser. For a change in inter-cooler effectiveness from 0.95 to 0.75 the thermal efficiency falls by 2.1 pp, whereas the same change in effectiveness of the economiser causes a reduction in thermal efficiency of 1.49 pp. A reduction in effectiveness of the recuperator from 0.9 to 0.8 results in a 1.17 pp drop in thermal efficiency. The cycle efficiency is least affected by after-cooler effectiveness, with a 0.64 pp drop in efficiency when the effectiveness is reduced from 0.95 to 0.75.

- In both the *reheated* and *non-reheated* HAT cycles only the reduction in effectiveness of the recuperator resulted in both a reduction of purchase cost of both the gas turbine, and the heat exchangers and saturator. This was found to be consequence of the increase in specific power, facilitated by the increase in evaporation rate in the saturator. A reduction in effectiveness of the recuperator from 0.9 to 0.8 results in a reduction by 6.87% and 11.5% in SPEC for respectively the *reheated* and *non-reheated* cycle. In terms of thermal efficiency this results in a drop by 2.28 pp for the *reheated* cycle and by 1.17 pp for the *non-reheated* cycle.
- The reduction in effectiveness of the inter-cooler and after-cooler result in a drop in purchase cost of the heat exchangers and a rise in the cost of the gas turbine, as the specific power decreases. It was found that the reduction in effectiveness of only the inter-cooler or after-cooler below a certain value ($0.85 > \varepsilon_{AC} > 0.75$) would result in an increase in the SPEC of the power plant.
- The reduction in effectiveness of the economiser, the reduction in OPR for the *reheated* HAT and the increase in the pinch temperature difference in the saturator all results in an increase in the SPEC of the power plant, as a result of the knock-on effect on the other components that become all more expensive.
- Due to the dominant voice operating cost has in the annual cost of machinery for prime mover in commercial marine applications, the *reheated* version of the HAT cycle was selected for further studies.

The study on the impact of the design ambient and sea water temperature on the thermal efficiency and cost of the power plant has been performed by re-optimising the free design variables. The study has been performed for the cycle A_5^{40} of the *reheated* HAT that achieves the maximum thermal efficiency. The main outcomes are the following:

- The ambient design temperature has a higher impact on the design thermal efficiency and SPEC of the power plant than the design sea water temperature.
- The humidity at the inlet of the combustion chamber decreases with both the reduction in design ambient and water temperature.
- The design thermal efficiency of a cycle designed for cold sea water and ambient temperatures (278.15 K) is 0.66 pp higher than for a cycle design

for temperate water and ambient conditions (288.15 K). The SPEC of the power plant results 0.75% higher for the cycle design for cold conditions than for the cycle design for temperate conditions.

- The design of the power plant for sea and ambient tropical conditions (298.15 K) achieves a maximum thermal efficiency 0.74 pp lower than the cycle designed for temperate conditions. However, the SPEC is reduced by 2.17% as a result of the reduction of the heat exchanger purchase cost.

This chapter has covered and fulfilled the objective of the development of an optimisation scheme for the exploration of HAT cycle design space. The sensitivity of the performance of the cycle to the technology level of selected components has been quantified. Moreover, the objective of determining the trade-offs between the maximum thermal efficiency and minimum purchase equipment cost of the power plant was fulfilled.

Chapter 4

Part-load/Off-design Performance Analysis

Commercial ships spend most of the time at cruise speed, which requires the power plant to operate between 75% and 80% of the maximum power output. Moreover, the power required and the performance of the power plant are affected by the wind and sea conditions, and by the ambient and sea temperature. The HAT cycle, therefore, would operate most of the time at part-load, away from the maximum power output setting for which it is designed (see chapter 3). In order to have realistic estimates in terms of fuel consumption it is, therefore necessary to evaluate the performance of the cycle at part-load and in off-design conditions.

This chapter presents the part-load and off-design performance analysis of the *reheated* HAT cycle. The study is focused on the performance of cycle A₅⁴⁰ previously analysed in the design space studies (see table 3.5). The first section presents the methodology for the modelling and simulation of the steady state part-load performance across a range of operating conditions. The second section analyses the impact of the inlet temperature of both burners, and of other proposed variables, on power output and thermal efficiency. Moreover, the impact of the design vector of the cycle part-load performance is presented. The impact of the ambient air and water temperature on the cycle performance is finally examined. The last section of this chapter addresses the emission production of the *reheated* HAT cycle, by evaluating the emission rate of NO_x, SO_x, CO and CO₂ across a range of operating conditions, and verify that the latter stand within the limits impose by MARPOL AnnexVI [12].

4.1 Off-design Cycle Thermal Performance Modelling

The methodology here presented was developed for the evaluation of the performance of the cycle at early stages of the design and, therefore without the detailed design of the components. Due to the impact the humidity in the hot section of the cycle has on the performance of the power, it is important to properly simulate the evaporation process in the tower at off-design. Therefore, a numerical scheme for the calculation of heat and mass transfer is used for resolving performance of the saturator tower.

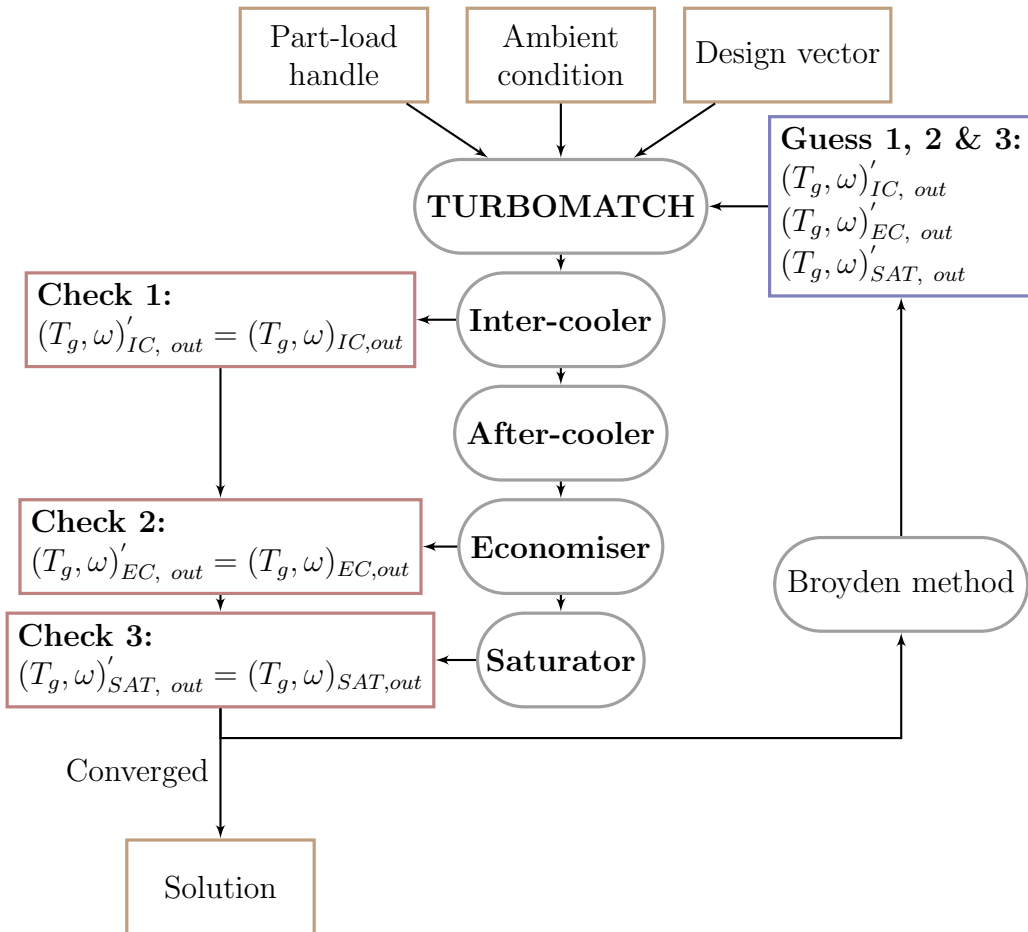


Figure 4.1: HAT cycle off-design simulation algorithm

As for the design point calculations, the off-design performance of the cycle is resolved by a system of subroutines and software. Due to the non-linearity of the performance of the cycle components, the off-design steady state performance of the HAT cycle is calculated with a Broyden method [65] iterative solver. The performance of gas turbine and recuperator are simulated by Turbomatch whereas

the performance of the inter-cooler, after-cooler, economiser and of the saturator are resolved by separate subroutines.

The simulation algorithm, to resolve the off-design performance of the HAT cycles, is shown in figure 4.1. It requires as inputs the design vector of the cycle, the ambient conditions and the off-design *handles* —i.e., free variables used to control the engine operation. To initialise the solver and run the Turbomatch software, the temperature and humidity of the gas at outlet the inter-cooler, economiser, and saturator are initially guessed. The results from the Turbomatch simulation are then used to resolve the remaining heat exchangers and saturator. The difference between the guessed and calculated gas outlet thermodynamic conditions of the heat exchangers and saturator are used to update the Broyden matrix until convergence is achieved.

In the analysis of the off-design performance of the cycle the following general assumptions are made:

- All components are adiabatic
- No account for mechanical losses

Gas Turbine

Turbomatch requires the design vector to perform a first design point simulation of the cycle. In this simulation the components maps are scaled to match the design point specifications such as mass flow, pressure ratio, turbomachinery isentropic efficiency, etc.

To perform the off-design performance, the free parameter called *handle*, typically the turbine inlet temperature of the high pressure turbine or power turbine, is specified. The new steady-state conditions for the specified values of the *handle* are calculated with an iterative method due to the non-linearity of the components characteristics. The system of *checks* and *guesses* used within Turbomatch is well described by Pellegrini [64].

The compressor, turbine and combustion chamber characteristics in off-design are described by maps, their values, therefore depend on the equilibrium operating point. Within the assumption that the power plant would be installed on a vessel with integrated full electric propulsion (IFEP), the power turbine is operated at constant rotational speed as it is connected to a generator.

The percentage of air bled from the outlet of the high-pressure compressor, for the cooling of the turbines rows, does not vary in part-load and off-design operating points.

Heat Exchangers

The performance model of the heat exchangers relies on the variation of kxA , and does not require specific geometry dimension for the solving of the model. In the assumption that the flow properties of the two streams do not vary substantially from the design values, Prandtl number and the thermal conductivity of the fluid can be considered constant. Thus, the main driver for the variation of the heat transfer is the variation of the Reynolds number and consequently of the stream mass flow. Using the Dittu-Boelter heat transfer correlation, it is possible to scale the kxA of each heat exchanger at off-design conditions as shown in eq. 4.1. The derivation of the equation and the assumptions made are reported in detail in appendix A.3.

$$kxA = kxA_{ref} \left(\frac{m_c^{0.8} m_h^{0.8}}{m_{c,ref}^{0.8} m_{h,ref}^{0.8}} \right) \left(\frac{m_{c,ref}^{0.8} + m_{h,ref}^{0.8}}{m_c^{0.8} + m_h^{0.8}} \right) \quad (4.1)$$

where:

- $_c$ is related to the cold stream.
- $_h$ is related to the hot stream
- $_{ref}$ is the reference value at design point

The new effectiveness of the heat exchanger at off-design is calculated within the assumption that the flow arrangements are approximated to counter-current.

$$\varepsilon = \frac{1 - \exp[-NTU(1 - C^*)]}{1 - C^* \exp[-NTU(1 - C^*)]} \quad (4.2)$$

where $NTU = kxA/C_{min}$ is the number of transfer units and $C^* = C_{min}/C_{max}$ is the heat capacity ratio between the two streams. Once the new effectiveness is defined the outlet conditions of the heat exchanger are calculated with the definition of effectiveness ($\varepsilon = Q/Q_{max}$). In order to avoid steaming within the heat exchanger, a threshold at the water outlet temperature is applied based on the water saturation temperature at 90% of the pressure in the saturator:

$$T_{w, max} = T_{sat}@ (0.9p_{SAT}) \quad (4.3)$$

The pressure losses in each stream of the heat exchangers are calculated as suggested by Walsh et al. [88] as following:

$$\frac{\Delta p}{p_{in}} = \alpha \left(\frac{m \sqrt{T_{in}}}{p_{in}} \right)^2 \quad (4.4)$$

where α is a coefficient calculated at design point.

Saturator

The model for the off-design performance calculations of the saturator is based on the Aramayo-Prudencio et al.[32] and Lindquist et al. [54] for the heat and mass transfer physics of the system, and is based on Dalili [51] for the calculation of the cross-section of the packed-bed.

The full model description is reported in appendix A.2. The model has been validated against experimental results of a HAT pilot cycle by Lindquist et al. [54], by imposing the dimensions of the packed-bed, height and cross-sectional area. The maximum error in the outlet proprieties of the two streams is of 1.5% as reported in table A.3.

4.2 Off-design Cycle Thermal Performance Analysis

The off-design performance analysis is performed on the cycle A_5^{40} , the design with the highest thermal efficiency within the design space of the 40 MW class *reheated* HAT cycle (see section 3.4.2). The design vector and the design performance of the cycle are reported in table 3.5. The effectiveness of the inter-cooler, after-cooler and economiser are all 0.95 whereas the effectiveness of the recuperator is 0.9. The OPR and the pinch temperature difference in the saturator tower are respectively 40:1 and 5 K.

Several of the handles that can be used to drive the *reheated* HAT cycle in part-load are here analyses. Their impact on the cycle performance in terms of thermal efficiency and power output are evaluated and focus is also set on the main thermodynamic drivers that govern the cycle performance.

4.2.1 Impact of Turbine Inlet Temperature

The TITs of the main burner (TIT_{CC}) and reheater (TIT_{RH}) are used as handles for the part-load performance across the range reported in table 4.1. The by-pass ratio of the saturator (β) is kept constant at the design value.

As long as no additional water is streamed into the inter-cooler, after-cooler or economiser to prevent steaming in the saturator, the water to gas mass flow ration in each of these heat exchanger is kept constant at the design value. As such,

Table 4.1: Variation of the TIT of the main and reheater combustion chamber across for part load performance.

TIT_{CC}	1600 : -25 : 1275 (K)
TIT_{RH}	1600 : -25 : 1200 (K)

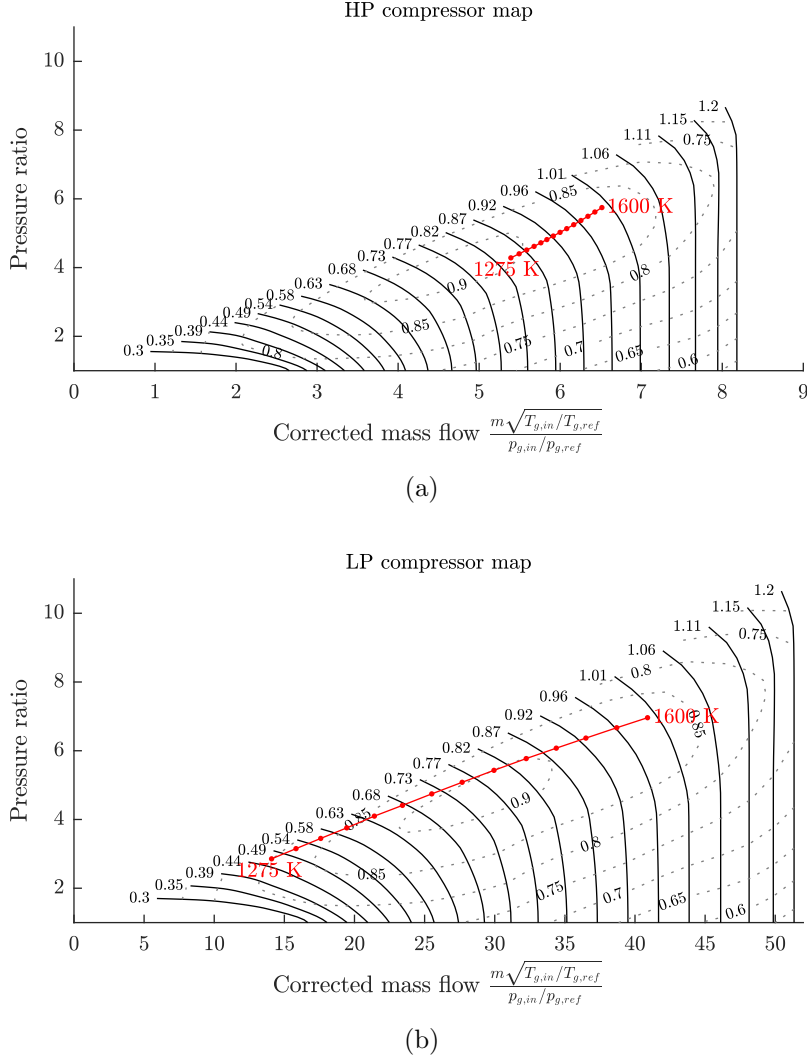


Figure 4.2: Running line on the compressors map when TIT_{CC} is used as handle:
(a) HP compressor map (b) LP compressor map

the variation of the heat transfer coefficient of the heat exchanger (see eq. 4.1) becomes a function only of the gas mass flow.

Figure 4.2 illustrates the systems running lines across a range of TIT_{CC} between 1600 K —i.e., design point— and 1275 K whereas, the TIT_{RH} is kept constant at 1600 K. The reduction in TIT_{CC} results in a 65% reduction of the inlet gas mass flow and a reduction in pressure ratio in both compressors due to the reduction in energy available for the core turbines. In both compressors, the new equilibrium

operating point when reducing the TIT_{CC} results in a lower corrected mass flow and pressure ratio. The rotational speed of the low-pressure shaft drops to 47% of the design value when the TIT_{CC} is reduced from 1600 K to 1275 K. In comparison, for the same TIT_{CC} reduction, the rotational speed of the high-pressure shaft drops to 84% of the design value. Although the different variation in the rotational speed of the two shafts is normal in three shaft gas turbines, the difference is further enhanced by the presence of the inter-cooler. The temperature of the gas at inlet of the HPC drops by only 11 K when the TIT_{CC} is reduced from 1600 K to 1275 K, this results in the smaller reduction in non-dimensional mass flow than if in absence of the inter-cooler. The proximity of the operating point of the LPC at low TIT_{CC} to the surge line could raise issues on the operability of the cycle in such conditions. Nevertheless, this would depend on the compressor map and systems to avoid operation in that region could be implemented.

When the TIT_{RH} is reduced from 1600 K to 1375 K and TIT_{CC} is kept constant at 1600 K (figure 4.3a-b), a 5.2% increase in the inlet flow of the cycle can be observed. The non-dimensional mass flow between the outlet of the LPT and the PT inlet increases due to the temperature rise in the reheater combustion chamber which, neglecting the fuel mass flow, it can be written:

$$\left(\frac{m \sqrt{T}}{p} \right)_{PT, in} = \left(\frac{m \sqrt{T}}{p} \right)_{LPT, out} \sqrt{\frac{T_{PT, in}}{T_{LPT, out}}} \frac{p_{LPT, out}}{p_{PT, in}} \quad (4.5)$$

where $\sqrt{\frac{T_{PT, in}}{T_{LPT, out}}}$ is the temperature ratio across the reheater, whereas $\frac{p_{LPT, out}}{p_{PT, in}}$ is the pressure ratio across the reheater combustion chamber.

Given that the inlet non-dimensional mass flow does not vary significantly, the non-dimensional mass flow at the outlet of the LPT increases to satisfy eq. 4.5 and the expansion ratio in the LPT also increases. This justifies the above-mentioned increase in the mass flow at the inlet of the gas turbine. This increase, nonetheless depends on the variation in humidity in the hot section of the cycle: if the humidity decreases, the increase in inlet gas mass flow is more significant than if the humidity remains constant or increases. Due to the increase in power extracted by the LP turbine by 11%, the LP compressor pressure ratio increases from 6.96:1 to 7.36:1. On the contrary, the power extracted by the HP turbine decreases by 1.6%, therefore a marginal decrease in the HP compressor pressure ratio from 5.75:1 to 5.46:1 is observed.

Having demonstrated the operability of the cycle across a range of TIT of both combustion chambers, figure 4.4 shows the thermal efficiency, power output and humidity at the inlet of the main burner of the HAT cycle across a range of main

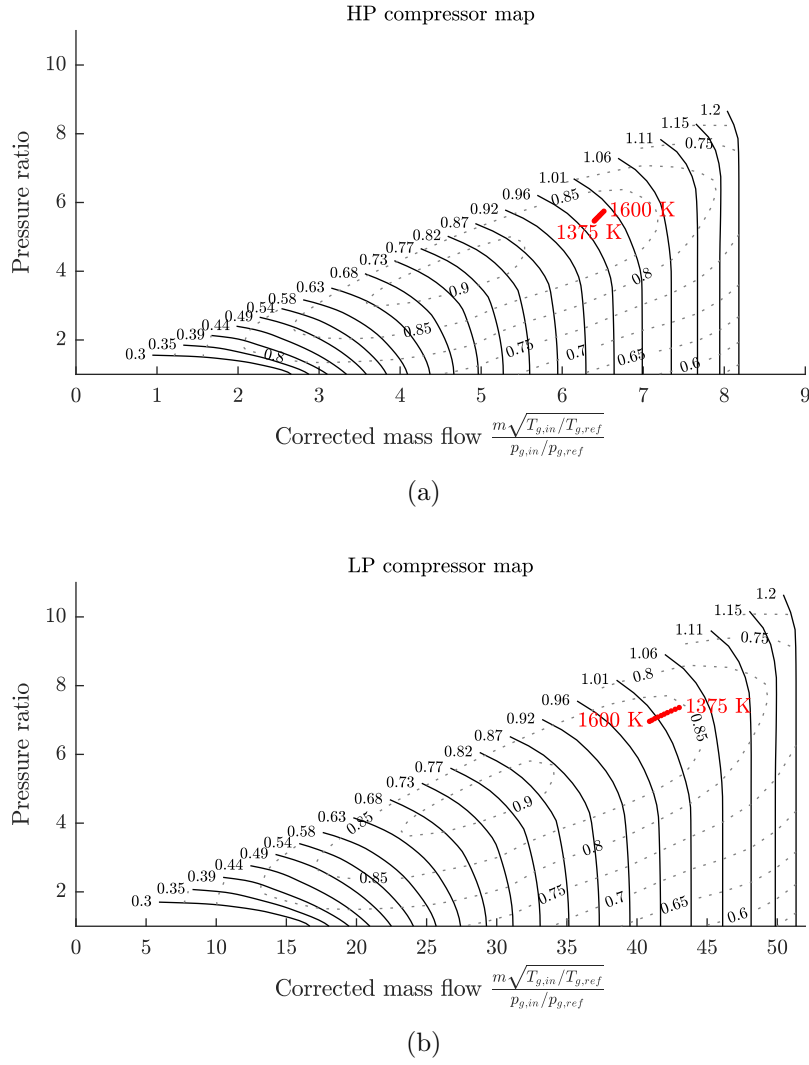


Figure 4.3: Running line on the compressors map when TIT_{RH} is used as handle:
 (a) HP compressor map (b) LP compressor map

burner and reheater outlet temperatures. Two discrete regimes can be identified within the part load operating envelope of the cycle (A, B in figure 4.4a). Region A (figure 4.4a) where the system power output is predominantly affected by the TIT of the main burner (TIT_{CC}), while the reduction in TIT_{RH} is found to increase the power output. In region B (figure 4.4a), although the TIT_{CC} remains the main driver of the power, a reduction in TIT_{RH} at constant TIT_{CC} causes a slight reduction in the power output.

In terms of the system thermal efficiency (figure 4.4b), there are again two discrete regions following the trend of the power output. Region A, where the thermal efficiency is mainly dependent upon the TIT_{CC} and region B, where the TIT_{RH} is found to have a detrimental effect on the efficiency of the cycle while TIT_{CC} is only affecting it weakly.

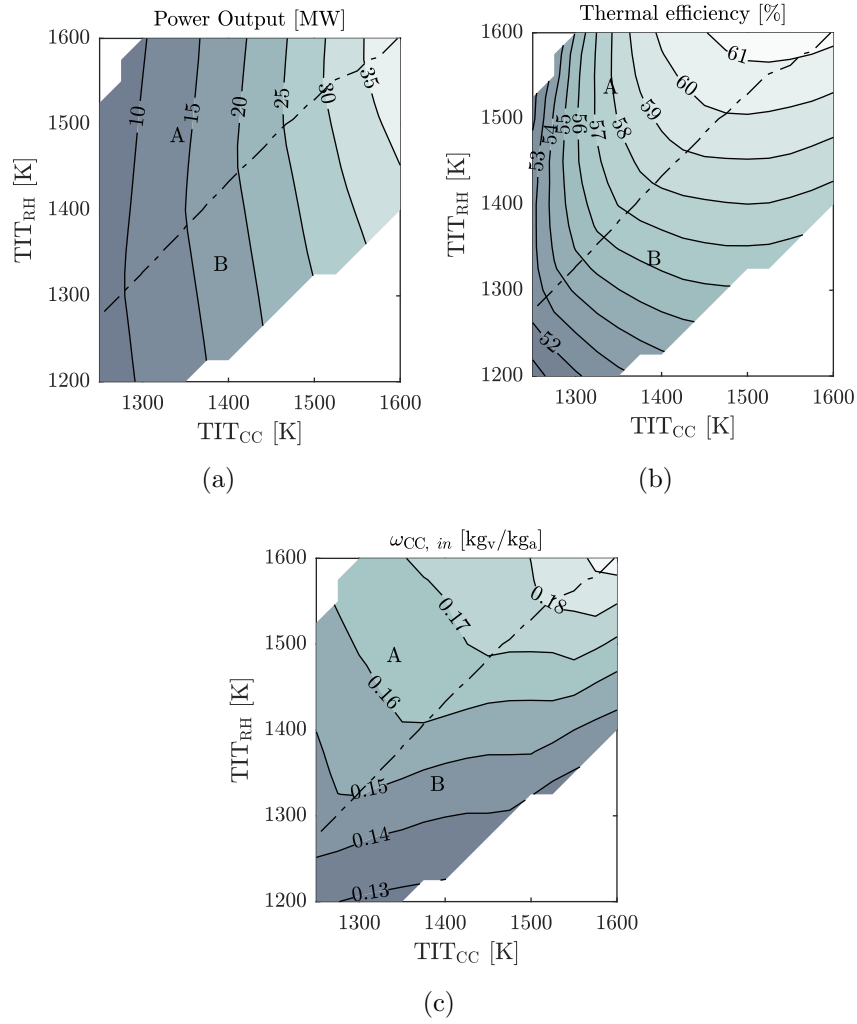


Figure 4.4: Part-load performance of the HAT cycle across a range of main burner and reheater outlet temperatures.

(a) Power output, (b) thermal efficiency and (c) humidity at the inlet of the main burner.

The behaviour of the system power output across these identified part-load regimes A and B (see figure 4.4), is predominantly driven by the variations in gas humidity and temperature at the inlet of the combustion chamber, as shown in figure 4.4c. Gas humidity (ω) depends on the pressure in the saturator tower (p_{SAT}), the relative gas humidity (Φ) and the air saturation pressure $p_{sat}(T_g)$ evaluated at the temperature of the gas T_g according to the following equation:

$$\omega = \frac{\Phi p_{sat}(T_g)}{p - \Phi p_{sat}(T_g)} \frac{R_a}{R_v} \quad (4.6)$$

Gas temperature (T_g) is in turn dependent upon the mass flows of the water and gas stream as well as upon the water inlet temperature at the saturation tower as described in the off-design saturator model in appendix A.2.

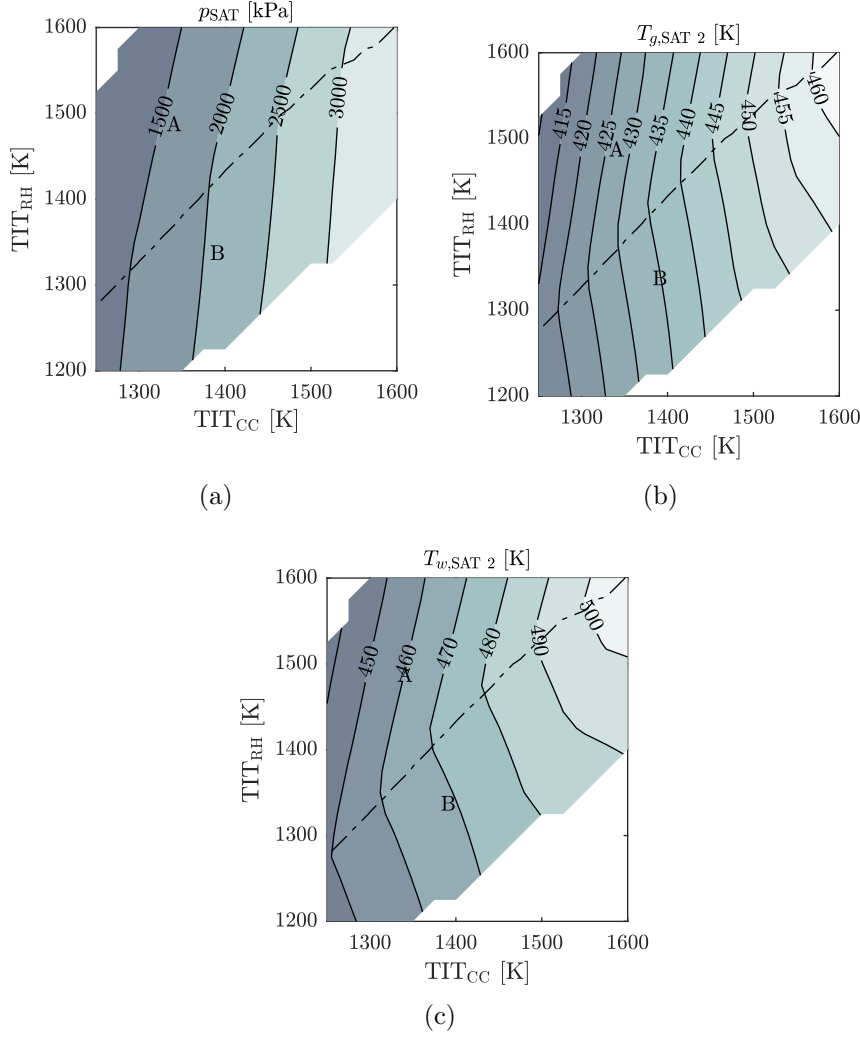


Figure 4.5: (a) Saturator pressure, (b) gas outlet temperature and (c) water inlet temperature as a function of TIT_{CC} and TIT_{RH} .

In region A of figure 4.4a the reduction in TIT_{CC} results in a reduction in humidity of the gas (figure 4.4c), that combined with the reduction in gas mass flow leads to a drastic reduction in power output. Although the pressure in the saturator drops (figure 4.5a), the reduction in humidity is due to the water temperature drop at the inlet of the saturator (figure 4.5c) and consequent drop in gas temperature at the top section of the saturator tower (figure 4.5b).

In region B, TIT_{CC} has only a minor impact on the humidity reduction (figure 4.4c), as the water inlet temperature (figure 4.5c) in the saturator is less affected by TIT_{CC} than in region A. The smaller impact of the TIT_{CC} on the humidity leads to a smaller sensitivity of the power output to the variation in TIT_{CC} . TIT_{RH} is found to have a detrimental effect on the humidity of the gas in region B compared to region A (figure 4.4c). Although for reductions in TIT_{RH} the pressure in the

saturator increases in both regions (A, B in figure 4.5a), the water temperature (figure 4.5c) in region B drops whereas in region A it increases.

The different behaviour of the water temperature at the inlet of the saturator is due to the water limited temperature at the outlet of the economiser in region A (see eq. 4.3). In region A, the economiser outlet temperature of the water (figure 4.6b) is directly affected by the pressure in the saturator, thus the temperature increases for reductions in TIT_{RH} , moreover the water to gas mass flow ratio in the economiser (figure 4.6a) varies and depends on the maximum water temperature. In region B, as the water to gas mass flow ratio is constant, the temperature is affected by the heat available in the economiser, thus the temperature drops for reductions in TIT_{RH} .

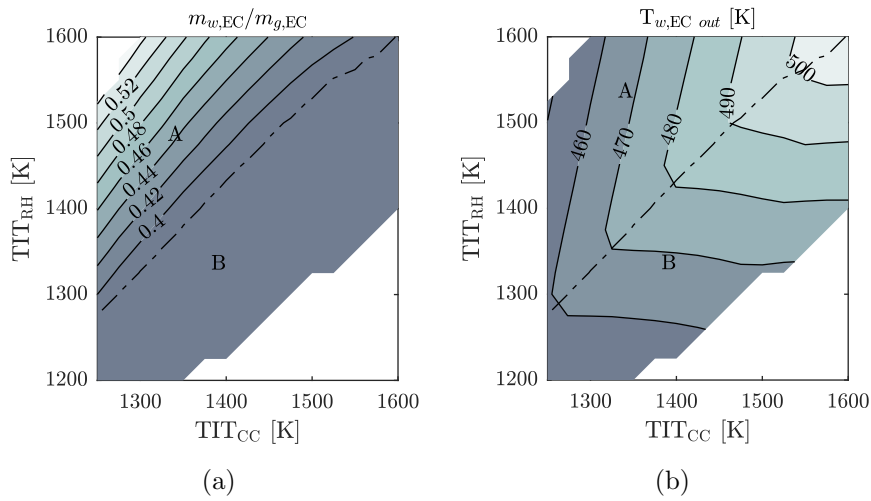
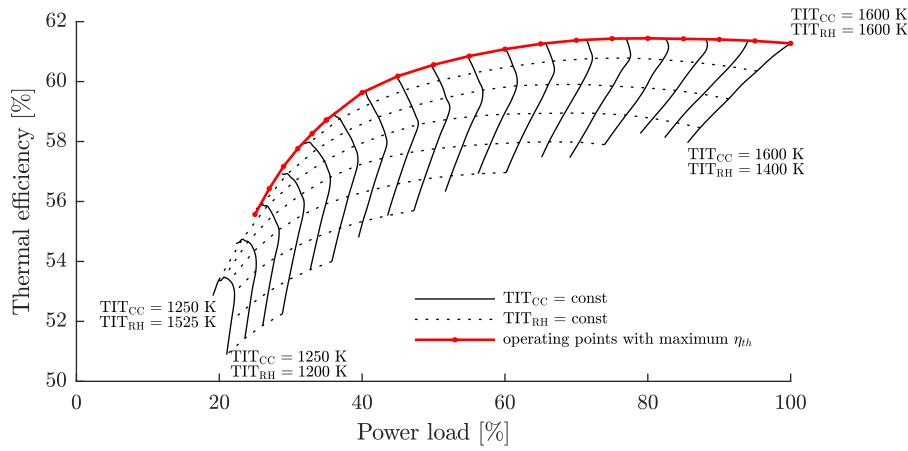
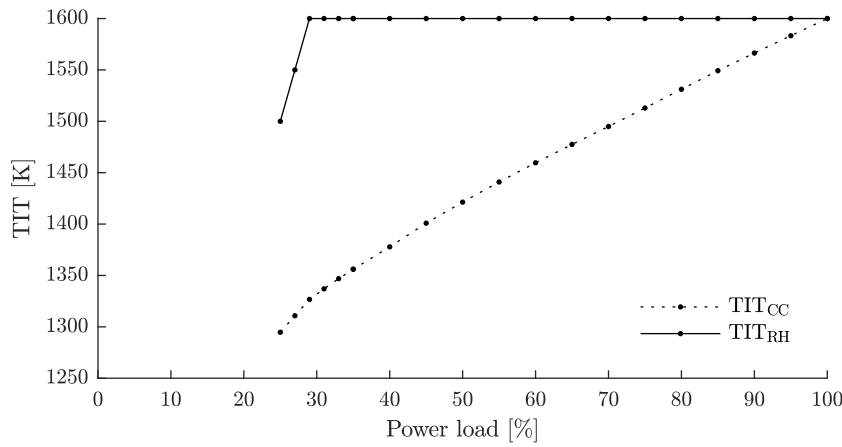


Figure 4.6: (a) Water to gas mass flow ratio and (b) outlet temperature of the water in the economiser as a function of TIT_{CC} and TIT_{RH} .

The cycle part-load control across a range of operating conditions is fundamental to maintain high thermal efficiencies across all power settings. Figure 4.7a shows the variation of both power output and thermal efficiency across a range of TIT_{CC} and TIT_{RH} . The red curve in figure 4.7a represents the operating points with maximum thermal efficiency for a power load between 100% and 25% of the design point power output. As shown in figure 4.7b, the highest thermal efficiency between 100% and 29% power output is achieved with constant $TIT_{RH} = 1600$ K and a variable TIT_{CC} to control the load. Between 29% and 25% of power load a reduction in TIT_{RH} from 1600 K to 1500 K is also required as well as a reduction in TIT_{CC} to 1295 K. The efficiency of the cycle, between design point and 67% of power load, remains higher or equal to the design value, with a maximum increase of 0.16 pp at 80% of the power load. When at 50% of the power load, the thermal efficiency drops by 0.72 pp from the design value.



(a)



(b)

Figure 4.7: (a) Thermal efficiency vs. power load in function of TIT_{CC} and TIT_{RH} and (b) TIT_{CC} and TIT_{RH} for maximum thermal efficiency across a range of power output.

The cycle can be operated down to approximately 50% of power load featuring thermal efficiency higher than 60%. Although the high thermal efficiency would limit the operating costs related to fuel consumption, the high TIT_{RH} needed across the power range to achieve such efficiencies may impact the life of the power turbine blade rows which may cause a knock-on effect on the maintenance cost. A full techno-economic analysis of the cycle for a specific operation would be necessary to assess the optimum part-load management of the cycle from a cost point of view.

An alternative common way for part-load control is via compressor variable inlet guide vanes (vIGVs) [61]. The performance of the reheated HAT cycle when closing the vIGVs is studied by keeping the water to air mass flow ratio constant at the design values for all water to air heat exchangers. Both TIT_{RH} and TIT_{CC} are

kept constant at 1600 K. Figure 4.8 illustrates the impact vIGVs regulated inlet mass flow on the thermal efficiency and power output. When the cycle operates at 85% of the power output (34 MW) the thermal efficiency drops by 1.13 pp. By comparison, for the same power load, achieved with the reduction of the TIT_{CC} , the thermal efficiency increases by 0.15 pp (figure 4.7a). As such, operating the cycle at part load by using TIT_{CC} as handle, rather the vIGVs, seems a more viable approach from an efficiency prospective as well as from a maintenance cost viewpoint.

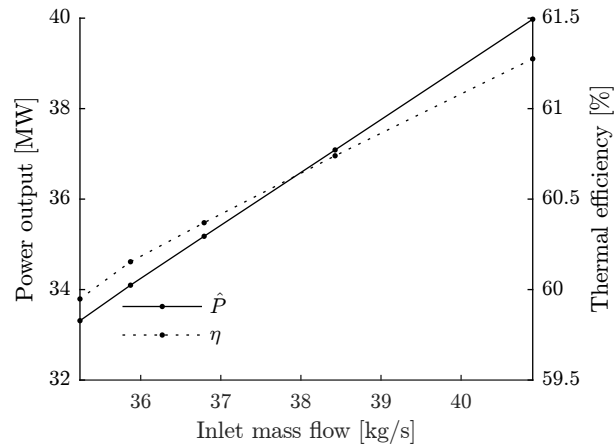


Figure 4.8: Power output and thermal efficiency for performance control via vIGVs.

4.2.2 Impact of the Saturator By-pass

In this section, the performance of the cycle is analysed when the saturator by-pass (percentage of the gas mass flow that flows through the after-cooler and saturator out of the total gas mass flow at the inlet of the gas turbine) is used as a handle to drive the cycle in part load. The saturator by-pass ratio is varied from 20% to 80.3% when the design value is 50.2% with constant TIT of both burners at the design values of 1600 K. When $\beta = 100\%$ all the air mass flow flows through the after-cooler and saturator tower whereas when $\beta = 0\%$ no air is humidified in the saturator tower.

When the by-pass ratio is changed, the variation in performance of the system is driven by the humidity in the hot section of the cycle. It can be noticed that any variation in β from its design value results in both a reduction in thermal efficiency and power output. When β is reduced to 20% the thermal efficiency drops by 43 pp and the power output by 13.25% (figure 4.9a), due to the reduction in inlet gas mass flow from 40.9 kg/s to 38.7 kg/s and the reduction in the humidity at the

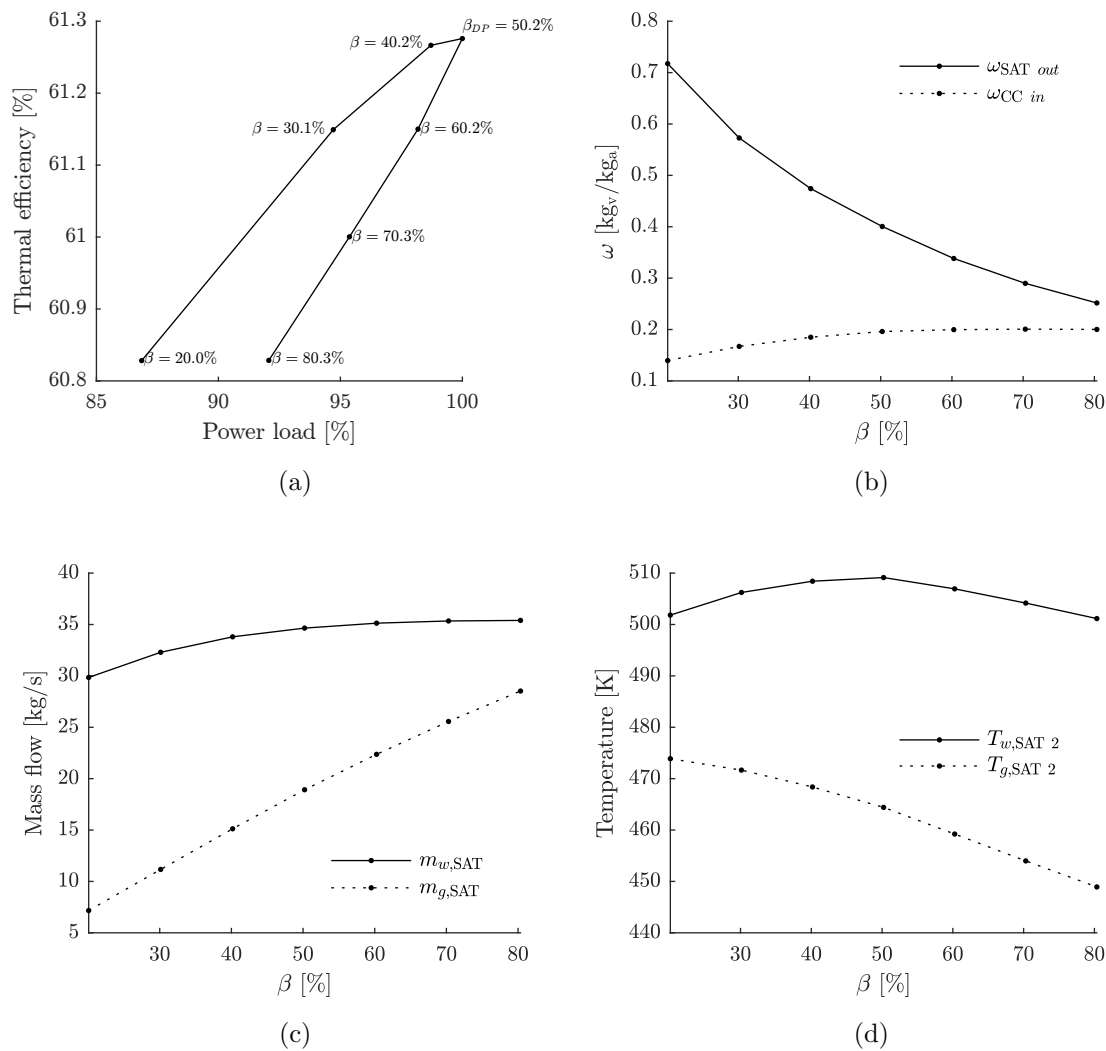


Figure 4.9: a) Thermal efficiency vs. power output when the saturator by-pass is varied.
 b) Humidity of the gas at the inlet of the main combustion chamber and at the outlet of the saturator tower as a function of β .
 c) Water and gas mass flows streamed in the saturator tower as a function of β .
 d) Temperature of the gas and water respectively at the outlet and inlet of the saturator tower as a function of β .

inlet of the combustion chamber from 0.196 to 0.140 (figure 4.9b). As β reduces, the humidification process in the saturator improves: the gas inlet mass flow in the tower drops by 62% (figure 4.9c), whereas the water mass flow drops by 14%. The reduction in water mass flow is a consequence of the reduction in gas mass flow in the gas turbine —i.e., the water to air mass flow ratio in the heat exchangers is kept constant— and due to the further reduction in gas mass flow through the after-cooler. Although the temperature of the water reduces from 509 K to 502 K, the temperature of the gas at the outlet of the saturator tower increases by 9.5 K,

consequently to the increase in water to air mass flow ratio in the saturator tower. As a result, the humidity of the gas at the outlet of the tower increases 0.400 to 0.717 (figure 4.9b).

When β is increased to 80.3%, although the humidity in the hot section of the gas turbine increases from 0.196 to 0.200 (figure 4.9b), the power output drops by 8% (figure 4.9a) because of the reduction in gas mass flow in the gas turbine from 40.9 kg/s to 38.5 kg/s. As β increases, the combined reduction in temperature of the water at the inlet of the saturator tower and the reduction in the water to gas mass flow ratio in the saturator results in the gas temperature at the outlet of the tower to drop by 15.5 K. Thus, the humidity of the gas at the outlet of the tower also drops from 0.400 to 0.252. However, the mass flow of water evaporated in the cycle increases.

Figure 4.10a compares the operating points with maximum thermal efficiency when reducing the power load from 100% to 30% when both TIT_{CC} and β are used as handles, and when only TIT_{CC} is varied. It can be noticed that at high power settings the difference in thermal efficiency is negligible, whereas at low power settings the difference in thermal efficiency reaches 0.4 pp. β decreases from 50.2% to 40.5% when the power load is reduced from 100% to 50%, whereas decrease to 25% when the power load is reduced to 30% (see figure 4.10b). For the same power load the TIT_{CC} increases when β is varied to maximise the thermal efficiency: $TIT_{CC} = 1330$ K is required for the power plant to operate at 30% of power output when β is not varied, whereas $TIT_{CC} = 1353$ K is required for the same power load when the β handle is optimised for maximum thermal efficiency.

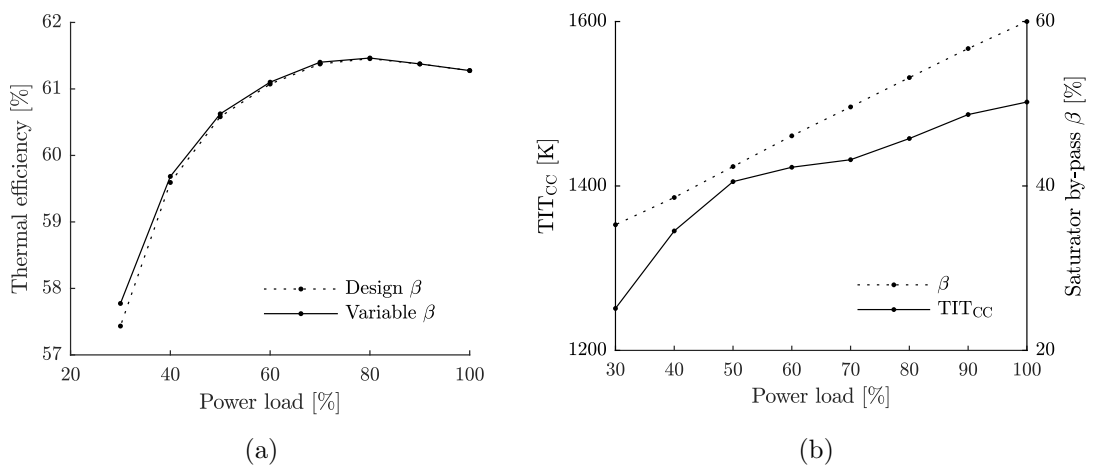


Figure 4.10: (a) Thermal efficiency vs. power load and (b) TIT_{CC} and β variation for maximum thermal efficiency across a range of power output.

The use of only the saturator by-pass as a handle is not effective to regulate the part-load, due to the lower power range achievable and the high losses in thermal efficiency. When used as a handle together with the TIT_{CC} of the cycle, no significant gain in thermal efficiency is achieved if not at low power settings with a maximum gain of 0.4 pp in thermal efficiency against the baseline at 30% power load. This handle is expected to be significant in part-load management only for power loads below 30%.

4.2.3 Impact of the Water to Gas Mass Flow Ratio in the Economiser

The water to air mass flow ratio in the economiser at design point is the result of the optimisation process of the heat capacity ratio of the heat exchanger, as described in section 3.4. The variation of the mass flow in the economiser can impact the performance of the saturator and therefore of the cycle. In this section the impact the water to air mass flow ratio is analysed for a range of TIT of both combustion chamber (see table 4.2).

Table 4.2: Variation of the TIT of the main and reheater combustion chamber across for part load performance for the analysis of the of the water to gas mass flow ratio in the economiser.

TIT_{CC}	1600 : -50 : 1300	(K)
TIT_{RH}	1600 : -50 : 1300	(K)

The water to air mass flow ratio in the heat exchangers can be controlled independently from the power output of the power plant, as the water mass flow is governed by electrical driven pumps. The water to gas mass flow in the economiser is varied from 80% to 120% of the design value. The results are compared to the previous case, where the water to gas mass flow ratio is kept constant at the design value. If, with the given water to air mass flow ratio, the water temperature at the outlet of the heat exchanger exceeds the water temperature limit $T_{w, max}$ (see eq. 4.3) the water mass flow is increased to attain the limit.

Figures 4.11a-b illustrate respectively the power output $\hat{P}_i$ and figures 4.12c-d the thermal efficiency η_i , where i is the percentage of the design water to air to mass flow ratio in the economiser. The over imposed dashed line contour plot is the difference in power ($\Delta\hat{P} = \hat{P}_i - \hat{P}_{100}$ MW) and thermal efficiency ($\Delta\eta = \eta_i - \eta_{100}$ pp) against the reference study with the design water to gas mass flow ratio. $\hat{P}_{100}$ and η_{100} are the power outputs and thermal efficiency achieved with the design water to air mass flow ratio, across the part load operating space.

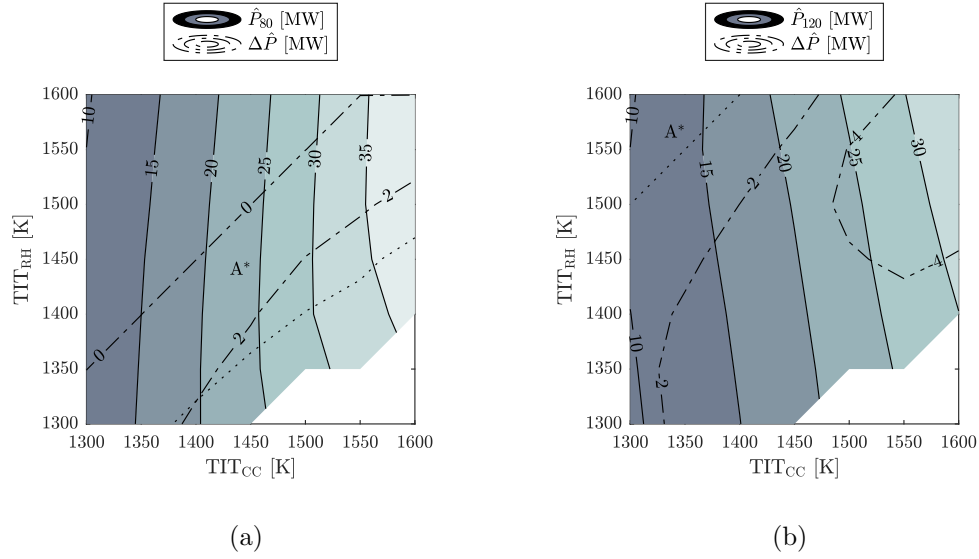


Figure 4.11: Power output of the cycle when changing the water to gas mass flow ratio in the economiser to (a) 80% (b) 120% of the design value and the design value and the power and efficiency difference with design water to gas mass flow ratio in the economiser.

When the water to gas mass flow in the economiser is reduced to 80% of the design value, the power output (figure 4.11) and the thermal efficiency (figure 4.12) both increase. This applies to the region beyond region A* described in figure 4.12. The maximum increase in power output of 3.3 MW is achieved with $TIT_{CC} = 1600$ K and $TIT_{RH} = 1450$ K, whereas the maximum gain in thermal efficiency of 0.78 pp is achieved $TIT_{CC} = 1400$ K and $TIT_{RH} = 1300$ K. The region A*, delimited in figure 4.11a by the dotted line and where the water to gas mass flow rate is defined by $T_{w, max}$, expands to lower TIT_{CC} and higher TIT_{CC} .

When the water to gas mass flow in the economiser is increased to 120% of the design value, the power output (figure 4.11) and the thermal efficiency (figure 4.12) both decrease. This applies to the region beyond region A*, described in figure 4.11a by the dotted line and where the water to gas mass flow rate is defined by $T_{w, max}$. The maximum drop in power output of 5.8 MW is achieved with $TIT_{CC} = 1600$ K and $TIT_{RH} = 1600$ K, whereas the maximum drop in thermal efficiency of 1.34 pp is achieved with $TIT_{CC} = 1300$ K and $TIT_{RH} = 1300$ K.

The maximum influence the variation in the water to air mass flow ratio in the economiser has on the thermal efficiency and power output occurs for two different operating points: at higher TIT for the power output and at lower TIT for the thermal efficiency. Nevertheless, the maximum power output and thermal efficiency for given TIT values is achieved with the water to air mass flow ratio with which $T_{w, EC out} = T_{w, max}$.

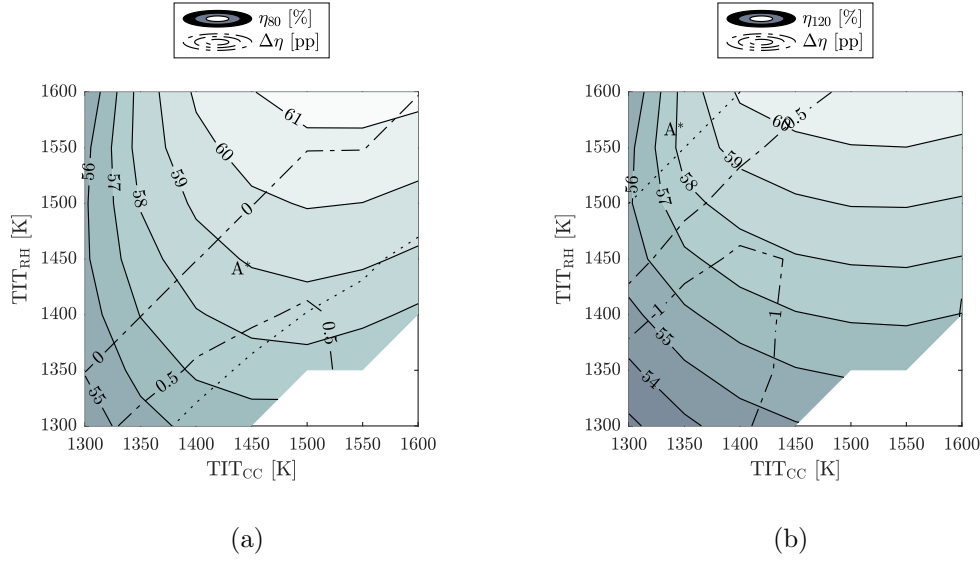


Figure 4.12: Thermal efficiency of the cycle when changing the water to gas mass flow ratio in the economiser to (a) 80% (b) 120% of the design value and the power and efficiency difference with design water to gas mass flow ratio in the economiser.

4.2.4 Impact of the Design Vector

The impact of the design vector of the cycle on the part-load performance of the power plant is assessed with the comparison of the off-design performance of selected cycles from the design space exploration of the reheated cycle (see section 3.4). The part-load performance analysis is performed for cycles B_5^{40} , C_5^{40} , D_5^{40} and E_5^{40} , with respectively reduced effectiveness of the inter-cooler, after-cooler, economiser and recuperator (see table 3.4).

In figure 4.13 the thermal efficiencies of these cycles are compared at part-load. The absolute values are presented in figure 4.13a, whereas the percentage variation, with the design value at maximum power output as a reference, are shown in figure 4.13b.

The cycle A_5^{40} demonstrates higher thermal efficiency across the whole power range here analysed (see figure 4.13a). The cycles A_5^{40} , B_5^{40} , C_5^{40} and D_5^{40} have very similar variations thermal efficiency in part-load. Among these cycles A_5^{40} has the minimum percentage gain in thermal efficiency (see figure 4.13b) when operating in part-load, with an maximum increase in thermal efficiency of 0.26% at 80% of the power load. However, cycles A_5^{40} has the minimum percentage reduction in thermal efficiency when operating at load below 35%. The cycle with reduced effectiveness of the inter-cooler, B_5^{40} , achieves the maximum percentage rise in thermal efficiency of 0.62% at 75% of the power load. Nevertheless, among these

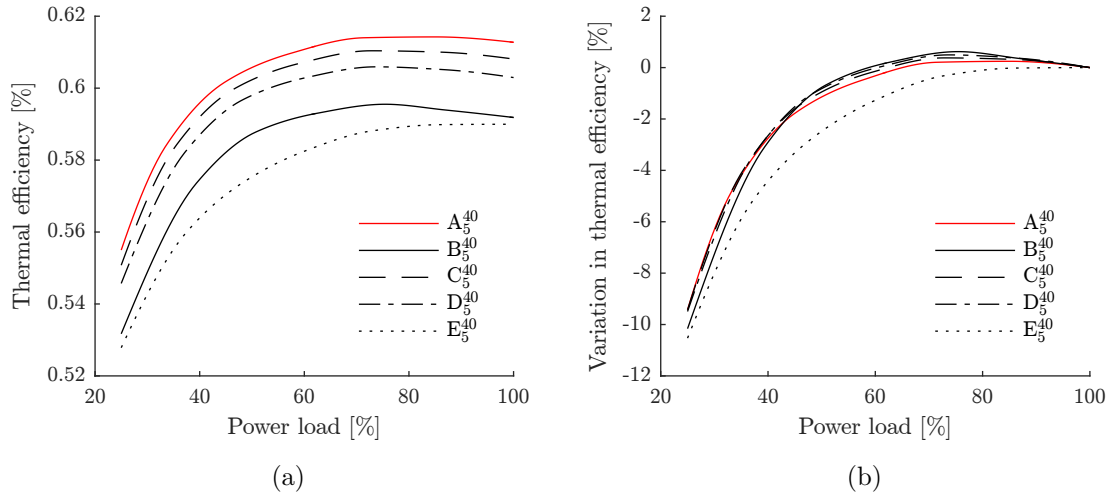


Figure 4.13: Comparison between the performance of selected cycle designs across range of power load: (a) thermal efficiency and (b) variation in thermal efficiency from the design value (reference: 100% power load)

cycles, cycle B₅⁴⁰ has the maximum percentage reduction in thermal efficiency at power loads below 40%.

The cycle with reduced effectiveness of the recuperator E₅⁴⁰, features the highest penalties in terms of both absolute (see figure 4.13a) and percentage change (see figure 4.13b) in thermal efficiency when it operates across the power load range here analysed. Therefore, although cycle B₅⁴⁰ and E₅⁴⁰ have similar thermal efficiencies at maximum power output (*design point*) with a difference of 0.19 pp, the difference in thermal efficiency reaches a maximum of 1.42 pp at 50% of the power load (see figure 4.13a). In terms of performance at part-load, the benefits of high effectiveness of the recuperator and the consequent flatter performance curves of the power plant become apparent.

4.2.5 Impact of Ambient Condition on the Cycle Performance

The ambient and sea water temperature can vary by several degrees across the time of day, the seasons and latitudes. Therefore, the impact of the air and water temperature on the performance of the cycle is valuable to predict the overall performance of the power plant across a mission route.

The impact of ambient air and sea temperature on the cycle A₅⁴⁰ performance are studied at constant TIT_{CC} and TIT_{RH} at the design value of 1600 K. The water temperature (T_w) is varied parametrically from 280.15 K to 312.15 K whereas the

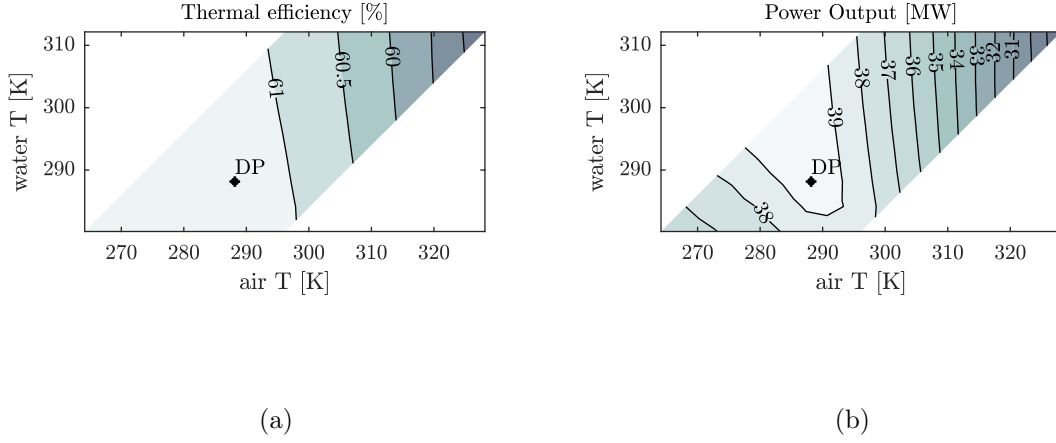


Figure 4.14: (a) Thermal efficiency and (b) power output as a function of air and water temperature.

ambient air temperature is varied as a function of the water temperature from $T_w - 16$ K to $T_w + 16$ K. The water to gas mass flow ratio is kept constant and if the temperature of the water at the outlet of the heat exchanger exceeds $T_{w, max}$ (see eq. 4.3) then the water mass flow in that heat exchanger is increased so that $T_{HE, out} = T_{w, max}$.

Figure 4.14 shows the variation in thermal efficiency and power output of the cycle as a function of water and air temperature. Thermal efficiency and power output are most affected by the ambient temperature for values higher than the design value of 288.15 K. The thermal efficiency and power output drop by 0.96 pp (figure 4.14a) and 5.1 MW (12.7% — see figure 4.14b) respectively for an increase in ambient temperature and sea water temperature of 10 K (308.15 K). Nevertheless, for the same air temperature (308.15 K), the variation of the water temperature from 292.15 K to 312.15 K cuts the thermal efficiency by 0.13 pp and the power output by 0.35 MW.

The reduction in power output and thermal efficiency is driven by the reduction in air compressibility at higher ambient temperatures and by the less effective inter-cooling at higher sea water temperatures. In both cases, the new equilibrium steady state operating point of both HP and LP shafts shifts to lower power, whereby gas mass flow (figure 4.15a) and the pressure ratio are reduced. The humidity (figure 4.15b) on the other hand increases as a result of the reduction in pressure and as a result of the increase in water to air mass flow ratio in the saturator tower. This mitigates the reduction in power output due to the reduction in compressibility of the air and poor inter-cooling observed in dry Brayton cycles.

Part of the reduction in enthalpy available in the turbines, due to the decrease in gas mass flow and pressure ratio, is compensated by this increase in humidity.

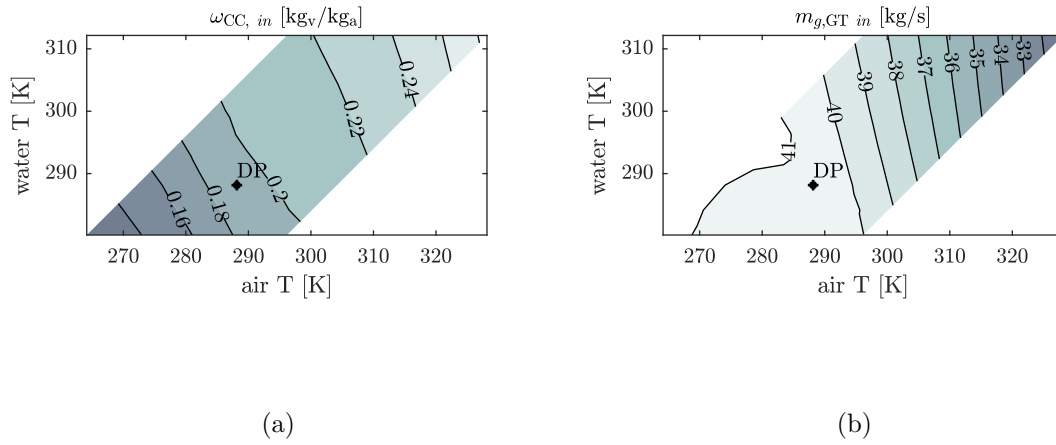


Figure 4.15: (a) Gas mass flow and (b) humidity at the inlet of main burner as a function of the air and water temperature.

For air temperatures below the design value the variations in thermal efficiency become very small. When both air and water temperatures decrease from the design values (288.15 K) to 280.15 K, the thermal efficiency drops by only 0.09 pp, whereas the power output drops to 37.6 MW. In this region, where the air temperature is lower than the design value, the water temperature is found to have greater impact on the power output than the air temperature (figure 4.14b).

4.2.6 Conclusions

The cycle methodology here presented to simulate the part-load performance of HAT cycles does not require any dimensions of the components of the cycle, allowing the prediction of the performance of the cycle across the range of power settings at very early stages of the cycles design. The use of a validated tool for the gas turbine modelling and detailed modelling of the physics within the saturator allow the study of the thermodynamic behaviour of the whole system when operating in off-design conditions.

The reheated HAT cycle was found to be capable to operate at high thermal efficiencies across a wide range of power settings. Down to 47% of the design power output it has a thermal efficiency higher than 60% and for power setting of 25% the thermal efficiency is of 55.6%. It was also found that for the cycle to operate at the maximum thermal efficiency, the TIT_{RH} remains constant at its

maximum value for most of the power settings and is reduced only when the power is below 29% of the design value.

The by-pass of the saturator tower and after cooler was found to have great impact on the power output of the cycle. Nevertheless, down to 40% of the power load the gain in thermal efficiency is negligible when the fraction of air flowing through the saturator is optimised to maximise the thermal efficiency. The impact increases for lower power settings, with a gain of 0.44 pp in thermal efficiency at 30% of power load.

The comparison between the part-load performance curves of selected cycle designs has shown how the cycle with maximum effectiveness of all heat exchangers A_5^{40} achieves higher thermal efficiency across the analysed power range. The reduction in effectiveness of the inter-cooler, after-cooler and economiser does not significantly affect the penalties in performance of the cycles at part-load. However, the reduction in effectiveness of the recuperator E_5^{40} results in higher penalties in terms of thermal efficiency. At 50% power load, the thermal efficiency of cycle A_5^{40} drops by 1.14% from the design value, whereas in cycle E_5^{40} it reduces by 2.5%.

The ambient temperature has a larger impact on the thermal efficiency and power output than the water temperature. The power output of the cycle reduces both with an increase and reduction in the water and ambient temperatures away from the design value when the TIT of both combustion chambers are kept constant. The efficiency of the cycle is almost constant when reducing the temperature of both the air and the water away from the design value, whereas it is most severely penalised by the increase in ambient temperature. This behaviour is consequent to the variation in the heat exchangers recuperated within the cycle and in the amount of heat and mass exchanged in the saturator tower.

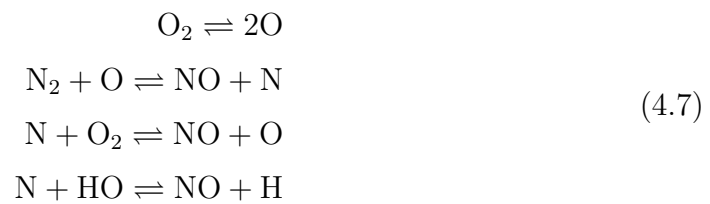
4.3 Emission estimation

The emissions formation levels have become a critical characteristic when assessing the feasibility of a propulsion platform for commercial marine applications. In the effort of curbing emissions the International Maritime Organization (IMO) has issued regulations curbing the emission levels of NO_x and SO_x in MARPOL AnnexVI [11–13], and are currently coming into place. In this section, the trends in the variation of emissions in different operating conditions are investigated along with how performance parameters affect the nitrous oxides (NO_x) and sulphur oxide (SO_x) and carbon monoxide (CO).

NO_x emissions

Thermal, fuel and prompt are the three main formation mechanism of nitrous oxide (NO) and nitrogen dioxide NO₂, known as NO_x. However, most of the nitric oxide further oxidises eventually to nitrogen dioxide.

The endothermic reactions of the oxidation of atmospheric nitrogen in high-temperature region of the flame and post-flame —i.e., formation of thermal NO_x— is described by the extended Zeldovich mechanism (see eq. 4.7). The rate of formation becomes significant when the temperature is above 1850 K [10].



Thermal nitrous oxide formation rate is influence by several factors. The increase in flame temperature results in exponential NO_x formation, but also in a reduction in time required for the chemical reaction to reach equilibrium [10]. The formation rate increases on the fuel-lean side of the stoichiometric. Fuel and nitrogen compete for oxygen and, although flame temperatures are higher in fuel-rich combustion, the fuel preferential consumes the oxygen [10]. As the flame temperature increases, the difference in emission between combustion with gaseous fuel and liquid fuel reduces [89]. Generally liquid fuels have higher thermal NO_x formation due to higher possibility that near the droplets stoichiometric combustion may occur, although the average equivalence ratio in the combustion zone may be lean. For higher flame temperatures, the temperature difference between stoichiometric and bulk flame temperatures reduces and therefore also the influence of local conditions around the fuel droplet. An increase in the inlet gas temperature also results in an increase in NO_x formation as the average temperature in the combustion chamber increases [10].

Fuel nitric oxide is produced by the oxidation of part of the organically bounded nitrogen in the fuel. Heavy fuel oils can contain up to 1.8% of nitrogen and, especially at low flame temperature, the contribution of fuel nitric oxide to the total can be considerable.

Under certain conditions of low temperature and rich fuel combustion, NO is found very early in the combustion region, in conflict with the kinetically controlled process theory. This is known as prompt nitric oxide and the formation mechanism

is not yet fully understood [10]. According to Nicol et al. [90] the initiating reaction is:



The balance of the prompt NO then involves the oxidation of HCN and N.

Nicol et al. [90] indicates that at high flame temperatures of 1900 K and equivalence ratio of 0.8, the main contributors to NO_x emissions are thermal NO_x with 60% of the total whereas 30% are prompt. For lower flame temperature and equivalence ratio the prompt contribution becomes dominant with 65%.

CO emissions

In rich-fuel combustion, carbon monoxide is a result of the inefficient combustion due to the lack of oxygen. Nevertheless, CO is also found in large quantities in stoichiometric and lean-fuel combustion, due to the dissociation of carbon dioxide. The CO formation surges in low power conditions with low peak temperatures and burning rates.

Generally the CO emissions are higher than predicted with the equilibrium theory, mainly due to incomplete fuel combustion. Commonly, the main reasons for the latter are: (i) inadequate burning rates in the primary zone of the combustion chamber, (ii) poor mixing of the air and fuel, generating zones with rich fuel combustion and areas where the fuel concentration is not sufficient for stable combustion, and (iii) quenching of the post-flame products by entrainment into the liner wall-cooling air.

SO_x emissions

Sulphur oxides are a result of the oxidation of part of the sulphur content in the fuel. Therefore, the formation rate is proportional to the fuel content. Although sulphur is not a lubricant itself, it helps to maintain the lubricity of the fuel because of the low melting point eutectic alloys it forms with nickel content in the fuel [91]. Due to its negative impact on the environment and human health, there is a growing interest and research on sulphur free fuels which use other lubricity improver additives to reduce SO_x emissions.

NO_x and CO emissions prediction

Several methods have been developed and are available in literature for the estimation of the emissions of gas turbine combustors. The three main approaches generally adopted are: empirical and semi-empirical models, reactor based models and computational fluid dynamic models.

Empirical and semi-empirical models are based on database of measured emission data. Most of these models are developed for NO_x emissions, however these methods can also be developed for other pollutants.

Lefebvre et al. [92] correlated the emission index EINO_x with the combustion pressure, the volume of the combustion chamber, the stoichiometric temperature, the temperature of the primary zone and the air mass flow rate. In this formulation the NO_x emission is suitable for conventional spray combustors and depend on the stoichiometric temperature and on the heterogeneous fuel-air mixture. Rizk et al. [93] developed a correlation that kept also into account the residence time and the evaporation time of the fuel. Tsalavoutas et al. [94] compared the semi-empirical correlations available in literature and concluded that the analysed correlations are optimistic in the estimation of both NO_x and CO. Tsalavoutas et al. also further optimised the analysed correlation in order to minimise the squares of the deviations of the predicted and measured emissions.

Another semi-empirical model, known as P3T3 method, was proposed by Zhuohui He et al. [95] for a lean direct injection combustor concept. This method correlates the emissions of NO_x with the inlet temperature and pressure of the gas at the inlet of the combustion chamber, and with the fuel to air equivalence ratio. The method relies on database of emissions of existing engines such as the one provided by the International Civil Aviation Organisation (ICAO).

Contrary to semi-empirical correlations, the reactor based models calculate the emissions based on the chemical reactions and kinetics that occur in the combustion process. The main assumptions that reactor based models take into consideration are finite rate chemistry and infinite mixing [96, 97]. The combustor is divided in a number *reactors*, flow regions with different characteristics within the combustion process. Semi-empirical correlations are used to estimate the kinetics of the chemical reactions and chemical equilibrium is used as a reference point for temperature, residence time, density and mass fraction determination. The geometry of the combustion chamber and experimental data of the specific combustor is required for the the validation and accurate use of the reactor based models [96, 97].

4.3.1 Methodology

For the purpose of this research semi-empirical approach is used for the estimation of the NO_x , CO and SO_x pollutants. The lack of data at this early stage of design of the power plants would have lead to a number of assumptions that would have jeopardised the accuracy of other methods.

NO_x & CO emissions

The correlations used for NO_x and CO emissions are based on the experimental data from the inter-cooler recuperated marine gas turbine WR-21¹ published by Butler et al. [98] and shown in figure 4.16. It was assumed that both the main and reheated combustion chamber would be similar to the combustor in the WR-21. The correlation here used for NO_x and CO calculation were proposed by Kyrou [99].

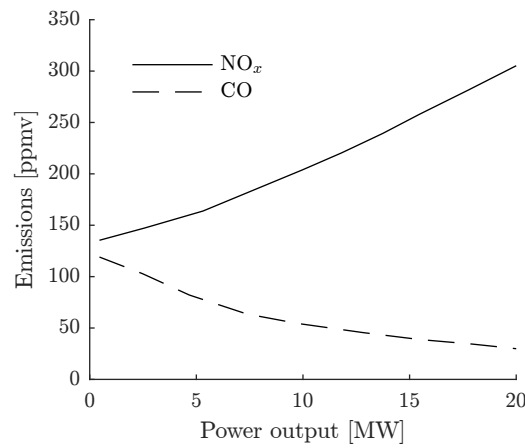


Figure 4.16: NO_x and CO emissions for WR-21 gas turbine from Butler et al. [98].

The correlations between the NO_x and CO emissions and fuel consumption calculated from the data presented in Butler et al. [98] are corrected using the *fuel flow method* presented by Deidewig et al. [100]. This method aims to provide a simplified correlation for emissions estimation.

Two correction factors were introduced to account for variations in ambient temperature and pressure:

$$\delta = \frac{p_1}{p_0} \quad p_0 = 101325 \text{ Pa} \quad (4.9)$$

$$\theta = \frac{T_1}{T_0} \quad T_0 = 288.15 \text{ K} \quad (4.10)$$

¹Rolls Royce.

where p_1 and T_1 are the inlet pressure and temperature in the gas turbine, and p_0 and T_0 are reference values.

It is assumed that for the same gas turbine, the operating point with corrected inlet pressure and temperature is equivalent to the operating point at reference conditions. Moreover, it is assumed that the pressure ratio, the corrected mass flow and the velocity remain the same [100]. Therefore, the combustion chamber inlet conditions, and fuel flow are corrected as following:

$$p_{CC, in, corr} = \frac{p_{CC, in}}{\delta} \quad (4.11)$$

$$T_{CC, in, corr} = \frac{T_{CC, in}}{\theta} \quad (4.12)$$

$$m_{f, corr} = \frac{m_f}{\delta\sqrt{\theta}} \quad (4.13)$$

Fuel flow is corrected by assuming that the added heat may be proportional to air flow and inlet air temperature for similar engine cycle operating points and constant combustion efficiencies [100].

The emission index is assumed to be a non-linear function of combustor inlet pressure and temperature. Therefore, the following relationship can be introduced:

$$NO_x(\text{or CO}) \sim T_{CC, in}^a \sim \theta^a T_{CC, in, corr}^a \quad (4.14)$$

$$NO_x(\text{or CO}) \sim p_{CC, in}^b \sim \delta^a p_{CC, in, corr}^a \quad (4.15)$$

Hence, corrected NO_x or CO can be estimated by the following formula:

$$\frac{NO_{x, corr}(\text{or CO}_{corr})}{NO_x(\text{or CO})} = \frac{1}{\theta^a \delta^b} \quad (4.16)$$

By finding the exponents a and b , the spread of the actual NO_x or CO emission index as a function of fuel flow can be collected into one fit that correlates corrected fuel flow to NO_x emissions.

A further correction is introduced for the NO_x emissions to account for the high humidity levels that are present in the combustion chamber. The impact of the humidity was estimated using the correlations proposed by Marchionna et al. [101] as following:

$$NO_x = NO_{x0} e^{-19 \omega} \quad (4.17)$$

Where NO_{x0} are the emissions with dry air and ω is the specific humidity in the combustion chamber.

SO_x & CO₂ emissions

The estimation of the sulphur oxide emissions are modelled after Jalkanen et al. [102]. The SO_x and CO₂ formations are dependent on respectively the sulphur and carbon concentration levels in the MDO fuel and on the specific fuel consumption. The moles of sulphur and carbon are calculated as following:

$$n = \frac{m}{M} = \frac{SFC \ y}{M} \quad (4.18)$$

where n [mol/kWh] are the moles of sulphur or carbon, m [g] is the mass of sulphur or carbon, M [g/mol] is the molar mass of sulphur or carbon, SFC [g/kWh] is the specific fuel consumption and y is either the sulphur content of the fuel assumed to be 1% mass or the carbon content of the fuel assumed to be 88% mass.

Since it is assumed that the moles of sulphur and carbon are the same as the moles of sulphur oxide and carbon dioxide, it is possible to calculate the sulphur oxide emissions by knowing the molar mass of the latter as following.

$$\text{SO}_x \text{ or CO}_2 = M \ n \quad [\text{g/kWh}] \quad (4.19)$$

where M this time is either the molar mass of sulphur oxide or of carbon dioxide.

4.3.2 Results

The analysis of the emissions of the *reheated* HAT A₅⁴⁰ is performed across the power setting range studied in section 4.2.1. The TIT of both main and reheater combustion chamber are used as handles to drive the cycle in part load operations.

Figure 4.17 shows the NO_x, SO_x, CO and CO₂ emissions in terms of g/kWh across the power load. The impact on the emissions rate of how the cycle is driven in off-design can also be appreciated, with the reduction of the TIT of the main combustion chamber and of reheater combustion chamber.

The NO_x emissions rate vary from a minimum of 0.09 g/kWh, at maximum power output with both TIT at at 1600 K, to a maximum emission rate of 0.37 g/kWh, at low power settings with the TIT_{RH} of 1200 K and the TIT_{CC} below 1350 K (see figure 4.17a). It is however possible to operate the cycle down to 25% of the power load with NO_x emissions below 0.15 g/kWh, when operating the cycle for maximum thermal efficiency (see section 4.2.1). NO_x emissions fall well within the limits imposed by MARPOL AnnexVI [11–13] of 1 g/kWh. Due to the suppression action humidity has on the NO_x emissions, the generation rate of the latter is

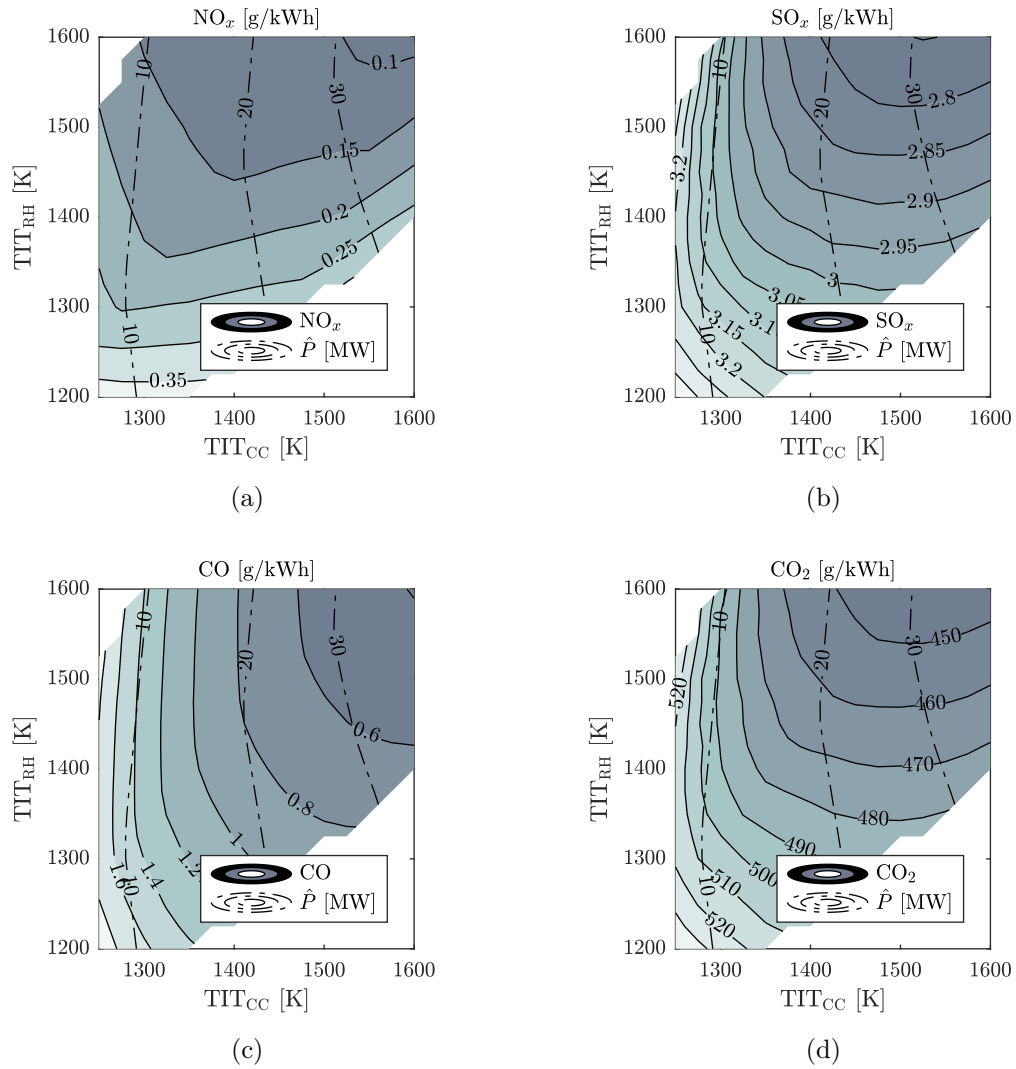


Figure 4.17: (a) NO_x , (b) SO_x , (c) CO and (d) CO_2 HAT cycle A_5^{40} emissions across part-load operations.

higher in regions of the part-load operating envelope where the humidity at the inlet of the combustion chamber is lower (see figure 4.4c).

The SO_x emissions vary across the part-load operating envelope from a minimum of 2.75 g/kWh at 70% of the power load, when the TIT of the main combustion chamber is 1500 K and the TIT of the reheater is 1600 K, to a maximum of 3.34 g/kWh when the TIT_{CC} is 1250 K and the TIT_{RH} is 1200 K. The limits imposed by MARPOL AnnexVI are on the content of sulphur in the fuel. Within this study an average 1% sulphur content in the MDO is considered and therefore withstands the global limits until 2020. However, the concentration exceeds the limits imposed in the ECAs areas where the limit is of 0.1%. When in navigation within the ECAs the power plant should switch to low-sulphur marine gasoline oil (LSMGO) to meet the imposed regulations.

The SO_x and CO_2 emissions (see figure 4.17b,d) follow the trends of the thermal efficiency across the part-load operating envelope (see figure 4.4b). The emission of CO_2 are higher at low power settings, with a maximum of 539.9 g/kWh when TIT_{CC} is 1250 K and TIT_{RH} is 1200 K. The minimum emission level of 444.3 g/kWh are instead achieved when operating at maximum thermal efficiency ($\text{TIT}_{\text{CC}} = 1500$ K & $\text{TIT}_{\text{RH}} = 1600$ K).

The minimum emission in CO of 0.39 g/kWh across the analysed part-load envelope is achieved at *design point* ($\text{TIT}_{\text{CC}} = 1600$ K & $\text{TIT}_{\text{RH}} = 1600$ K) whereas the maximum emissions of 1.95 g/kWh is achieved when TIT_{CC} is 1250 K and TIT_{RH} is 1200 K (see figure 4.17c). CO emissions, contrary to SO_x and CO_2 emissions, are function of the fuel flow consumption variation in the single combustion chambers and not of the global consumption of fuel. When the TIT_{RH} is reduced at constant TIT_{CC} , the fuel flow in the main combustion chamber increase although the total mass flow decreases. The CO emissions in the main combustion chamber tend to reduce, whereas the CO emissions in the reheater combustion chamber surge, thus the low impact TIT_{RH} has on CO emissions in figure 4.17c.

4.4 Summary

A methodology for the simulation of part-load performance of HAT cycle has been developed. This method allows the prediction of the performance of the cycle across the range of power settings at very early stages of the power plant design, as no dimensions of the cycle component are required. The study of the thermodynamic behaviour of the whole system, when operating in off-design conditions, is allowed by the use of a validated tool for the gas turbine modelling and detailed modelling of the physics within the saturator.

The main findings of the analysis on the part-load performance of the *reheated* HAT cycle are the following:

- The *reheated* HAT A_5^{40} is able operate at high thermal efficiency across a wide range of power settings. The thermal efficiency is higher than 60% down to 47% of power load, and at 25% of power output the cycle can be operated with a thermal efficiency of 55.6%. It was also found that to achieve the maximum thermal efficiency the TIT_{RH} remains constant at its maximum value across most of the power settings and reduces only when the power is bellow 29% of the design value.
- The variation in the fraction of air mass flow that is humidified in the saturator tower has a great impact on both power output and thermal efficiency.

However, it was found that when β is optimised to gain maximum thermal efficiency no significant improvement is noticed above 40% of the power-load. An increase of 0.44 pp is achieved at 30% of the power load.

- The comparison between the part-load performance curves of selected cycle designs has shown that the reduction in effectiveness of the recuperator E_5^{40} results in the highest penalties in term of thermal efficiency. At 50% power load the thermal efficiency of cycle A_5^{40} drops by 1.14% from the design value, whereas in cycle E_5^{40} it reduces by 2.5%. The variation of the effectiveness of the other heat exchangers does not significantly affect the penalties in performance of the cycles at part-load.
- Ambient air temperature has a larger impact on the thermal efficiency and power output than the sea water temperature. With constant TIT in both combustion chambers the power output decreases when both ambient and sea temperature are increased or decreased from the design value. The thermal efficiency is not significantly affected by the reduction in sea water and ambient temperature below the design value whereas it drops mainly at high ambient temperature. An increase of 10 K from the design value (288.15 K) of both water and air temperature results in a drop in efficiency by 0.96 pp and a reduction in power output by 5.1 MW.

A semi-empirical method has been developed and used for the estimation of the NO_x and CO emissions, whereas a method on the concentration of carbon and sulphur in the MDO fuel has been used for the calculation of SO_x and CO_2 emissions.

- The NO_x emissions in the HAT cycle result to be very low. For cycle $A_5^{40} reheated$ HAT the emissions are below 0.15 g/kWh when operating between 25% and 100% power load. MARPOL AnnexVI sets the limit at 1.96 g/kWh. The low NO_x emissions are a result of the high humidity in the combustion chambers. Therefore, when the cycle operates at power setting with low humidity in the cycle the NO_x emissions increase. Although, the emission rate of NO_x is estimated based on semi-empirical correlations, the emission rates obtained in this study are in line with the results published in experimental campaigns of HAT cycles [21, 22, 24, 30].
- MDO fuel, with sulphur content of 1%, can be used in open sea navigation until 2020 or 2025 depending on outcome of the IMO feasibility review planned for not later than 2018. However, low-sulphur marine gasoline oil (LSMGO) should be stored onboard and used when in navigation in the ECAs where sulphur content in the fuel limit is 0.1%.

This chapter has covered and fulfilled the objective of the development of a scheme for part-load and off-design performance analysis of HAT cycles. The performance of the HAT cycle across a range of power load setting and in off-design conditions have been analysed. Moreover, the emission levels were estimated and compared with limits imposed by the IMO regulations.

Chapter 5

Economics Analysis

In order to assess the economic feasibility of the *reheated* HAT cycle as prime mover of commercial marine vessels, it is necessary to analysis the capital investment expenditure and operation investment expenditure of the power plant integrated within a vessel. In this chapter, an economic analysis of the *reheated* HAT cycle is presented.

In the first section the methodology used for the economic analysis is presented. The simulation tools for performance calculations of the vessel and the prime mover system, and the metrics used for the financial evaluation are described. In the second section the costs analysis is performed for the 40 MW HAT cycles operating on a Panamax container ship class. Finally, a comparison is made between a diesel engine and a *reheated* HAT cycle powering a Handymax class bulk carrier operating on a defined commercial route.

5.1 Economic Analysis Methodology

The economic analysis comprises of two main parts. At first the performance of the vessel and prime mover is evaluated for defined marine routes. The fuel consumption and the annual millage are then used for the financial evaluation of the power plant.

5.1.1 Marine Route Modelling

It is assumed that the HAT power plant is installed on a vessel in integrated full electric propulsion system (IFEP) configuration —i.e., the power turbine shaft drives a generator whereas the propeller shaft is driven by an electric engine. This plant configuration has become dominant for cruise ships and liquified natural

gas carriers, and is increasingly popular in shuttle tankers, short sea and deep-sea carriers, and Baltic RoRo¹ ferries [7]. The main advantages of the IFEP configuration are the increased flexibility in positioning of the prime movers, generators and propeller electric drives, the propulsion and auxiliary systems are integrated, and economical part-load running due to the constant shaft speed of the gas turbine.

This study focuses on the main prime mover of the marine vessels propulsion system. Therefore, only in the open-sea operations are analysed in the following sections. The performance of the prime mover and the operating costs in port are not considered in this research. Moreover, the auxiliary engines, typically used in IFEP systems, are also not included.

The performance of the ship operating the *reheated* HAT cycle is evaluated with Poseidon, a marine route simulation tool previously developed in Cranfield University, and through extensive use has been proven reliable, accurate and flexible [103–106]. Given a route schedule, the software allows the calculation of the total fuel consumption and duration of the journey. The impact of adverse sea conditions and of air temperature variations during the day are also evaluated.

Figure 5.1 describes the methodology used for the route analysis. The tool is comprised of three main routines: the route management, the ship resistance model and the engine performance model.

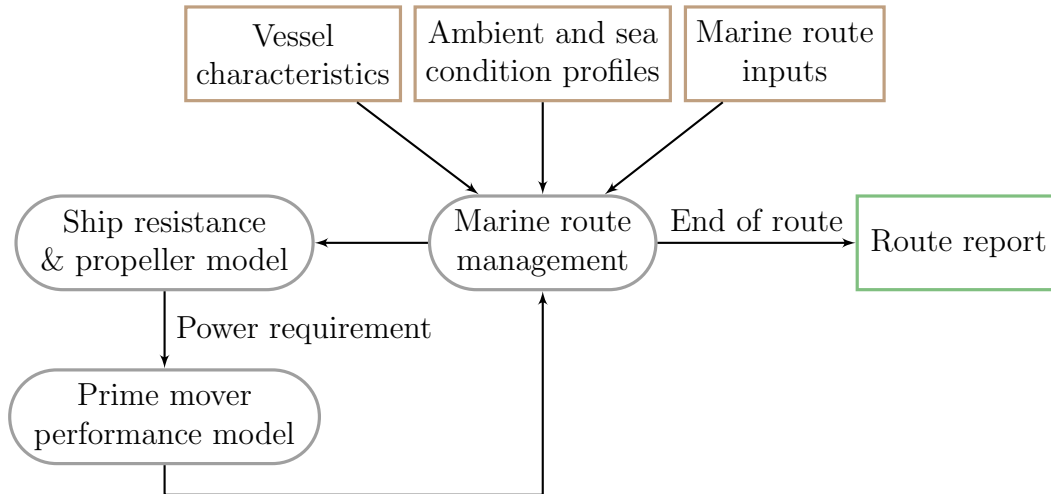


Figure 5.1: Marine route analysis methodology.

Marine Route Management

The route management routine simulates the route in time steps of 1 hour. The route schedule is defined by the inputs in terms of cruise speed, distance and ambient conditions. For each time step the performance of the vessel is re-evaluated

¹RoRo: Roll-on roll-off ferries

with the updated velocity and ambient and sea conditions defined by the route schedule. The wave conditions and wind intensity are defined as inputs by the sea state number, based on the *Douglas sea scale* [107] and reported in table 5.1.

Table 5.1: Sea state number [107].

Sea state number	Sea wave height [m]	Sea wave frequency [Hz]	Wind speed [m/s]
0-1	0.05	0.0	3.0
2	0.3	0.158	7.0
3	0.88	0.133	13.0
4	1.88	0.114	18.0
5	3.25	0.103	24.0
6	5.0	0.080	36.0
7	9.0	0.067	50.0
8	13	0.061	60.0

The variation in ambient temperature during the day —i.e., 24 hours— is simulated with the method presented by Huld et al. [108]. The daily temperature profile is described using three piecewise cosine functions (see eq. 5.1), dividing the day into three periods: from midnight to dawn, from dawn to the time of maximum temperature and from the time of maximum temperature to midnight.

$$T_{air}(t) = \begin{cases} T_m - T_{amp} \cos\left(\pi \frac{t_{dawn} - t}{24 + t_{dawn} - t_{peak}}\right) & 0 \text{ h} < t < t_{dawn} \\ T_m + T_{amp} \cos\left(\pi \frac{t_{peak} - t}{t_{peak} - t_{dawn}}\right) & t_{dawn} < t < t_{peak} \\ T_m - T_{amp} \cos\left(\pi \frac{24 + t_{dawn} - t}{24 + t_{dawn} - t_{peak}}\right) & t_{peak} < t < 24 \text{ h} \end{cases} \quad (5.1)$$

Where $T_{amp} = (T_{max} - T_{min})/2$ is the amplitude of the daily temperature, whereas $T_m = (T_{max} + T_{min})/2$ is the average daily temperature. It is assumed that the peak temperature is reached at 15 h.

Ship Resistance and Propeller Model

In the ship resistance model, the hydrodynamic and aerodynamic resistance of the ship are calculated. The power required by the HAT power plant to maintain the desired cruise velocity is therefore determined.

The *Holtrop and Mennen method* [109] is used for the calculation of the required propulsive power for the ship. Moreover, the aerodynamic resistance of the ship is estimated with the *Jensen method* [110]. Both the impact of shallow water on the resistance of the hull and the cavitation effects on the propeller are neglected. The total resistance R_{tot} accounts for the following resistance contributes:

$$R_{tot} = R_V + R_{app} + R_W + R_{bow} + R_A + R_{SW} + R_{AA} \quad (5.2)$$

Where R_V is the viscous resistance of the hull, R_{app} is the resistance of the ship appendage, R_W is the wave resistance, R_{bow} is the resistance of the bow-thrusters, R_A is the model-ship correlation resistance, R_{SW} is the additional resistance of sea waves and R_{AA} is the wind resistance.

The viscous resistance of the hull is estimated with the statistic method presented by Holtrop et al.[109] that utilises the ITTC-97 formulation of the frictional resistance coefficient. The resistance of the appendages is accounted for within an effective form factor as suggested in Holtrop et al.[109]. Therefore, the resistance of the hull and appendages is function of a effective form factor of the appendages, the velocity of the vessel, of the density of the sea water, of the wet surface of the bare hull and of the frictional resistance coefficient. The resistance of the bow-thrusters is accounted for separately and is function of the velocity of the vessel, of the density of the sea water, and of the number and diameter of the bow-thrusters. The model-ship correlation resistance account for the hull roughness and the still air contribution to the total resistance of the vessel are modelled with the analytical method presented by Holtrop et al.[109].

The wave resistance of the vessel is modelled according to Holtrop [111] and is function of the displacement volume of the hull, of the density of the water and of the Froude non-dimensional number. The effect of bulbous bow and transom stern on the wave resistance are accounted for in form of correction factors.

The sea wave resistance is estimated by the simplified method described in Tsoudis [103]. This method does not account for any effects on the vessels resistance by motion responses due to sea waves. The wind resistance is estimated with the *Jensen method* [110].

The power required from shaft of the prime mover —i.e., brake power— to maintain the cruise velocity is calculated according to the *propulsion of ship theory* [112] as following:

$$\hat{P} = \frac{P_E}{\eta_H \cdot \eta_O \cdot \eta_R \cdot \eta_s \cdot \eta_b \cdot \eta_t} \quad (5.3)$$

Where $P_E = R_{tot} \cdot V$ is the effective (or towing) power (V is the cruise speed), η_H is the hull efficiency, η_O is the efficiency of the propeller in open water, η_R is the

relative rotative efficiency, η_s is the stern-tube seal efficiency, η_b is the efficiency of the shaft bearing and η_t is the transmission efficiency that accounts for the losses of the generator and electrical motor.

The calculation of the open water efficiency of the propeller is based on the method presented by Oosterveld et al. [113] for Wageningen B-series propellers operating at off-design. The impact of the Reynolds number at a specific operating conditions on the propeller efficiency is accounted for as presented in Holtrop et al.[109].

Prime Mover Performance Model

The performance of the *reheated* HAT cycle is estimated by interpolation of a database of operating points of the cycle. The performance of operating points in the database is resolved as described in the methodology for the off-design performance simulation of the HAT cycle, in section 4.1. The TIT of the main combustion chamber is used as the handle to drive the cycle in part-load, whereas the TIT of the reheater combustion chamber is kept constant at the design value of 1600 K. As long as no additional water is streamed into the heat exchangers to prevent steaming in the saturator, the water mass flow in each heat exchanger is kept constant at the design value.

The database is created by parametrically simulating the performance of the cycle across a range of TIT_{CC} , sea water temperature and ambient air temperature. The range of the parametric variables are reported in table 5.2.

Table 5.2: Range of TIT_{CC} , sea water temperature and ambient air temperature used for the generation of the cycle performance database.

TIT_{CC}	1300 : 50 : 1600	K
T_w	280.15 : 8 : 304.15	K
T_a	$T_w - 8 : 4 : T_w + 8$	K

An iterative method is used to determine the TIT_{CC} , given the ambient and sea temperature, required by the power plant to deliver the brake power required by the vessel. If the maximum power output of the power plant —i.e., the power output achieved with $TIT_{CC} = 1600$ K for the given ambient and sea temperature— is lower than the required power for cruise speed, an iterative process is used to determine the new vessel speed at maximum power output. Therefore, for each time step the thermal efficiency and power output are calculated and the total fuel consumption and navigation time for the route is determined.

5.1.2 Financial Investment Evaluation

The economic analysis is conducted in *constant dollars* as no inflation rate is accounted for and only the real interest rate is considered. The main advantage over *current dollar* is that it gives a clear picture of the real cost involved in the investment and the O&M and fuel costs are not overemphasised by the inflation [114].

The financial investment is evaluated by the calculation of the annual cost of the machinery as in eq. 5.4.

$$AK = CAPEX + OPEX \quad (5.4)$$

Where CAPEX is the capital expenditure and OPEX is the operation expenditure.

The capital expenditure is the *annuities* of the *future value* of the investment and is calculated as the purchase equipment cost times the *capital recovery factor* as shown in eq. 5.5.

$$CAPEX = PEC \cdot i_r \frac{(1 + i_r)^n}{(1 + i_r)^n - 1} \quad (5.5)$$

Where PEC is calculated as described in section 3.3 (see eq. 3.2), i_r is the discount rate and n is the lifetime of the investment.

The operation expenditure is given by the following equation:

$$OPEX = C_{AF} + C_{AM} \quad (5.6)$$

Where C_{AF} is the annual fuel cost and C_{AM} is the annual maintenance cost.

The C_{AM} estimation for the HAT cycle follows the method presented in Jonsson et al. [45]. It comprises of a fixed term, that is a percentage of the PEC, and a variable term, that depends on the fuel consumption as shown in eq. 5.7.

$$C_{AM} = 0.02 \cdot PEC + 0.83 \cdot E_{fuel} \quad (5.7)$$

Where PEC is defined in dollars and E_{fuel} is the chemical fuel energy consumed over the total operating hours in a year and is defined in MWh.

When the investment of two power plants are compared, the payback period (defined in years) is the period of time required for the operation savings to recover the higher capital cost of the investment [114], as defined in eq. 5.8.

$$\text{Payback} = -\frac{\Delta CAPEX \cdot n}{\Delta OPEX} \quad (5.8)$$

Whereas the cumulative operating savings is defined as following:

$$\text{Cum. operating savings} = n \cdot \Delta\text{CAPEX} + \sum_{i=1}^n \Delta\text{OPEX}_i \quad (5.9)$$

Where ΔCAPEX is difference in capital expenditure of the two investments and ΔOPEX is the operating savings between the two investments.

5.2 *Reheated* HAT Installed on a Panamax Container Ship

In this section, the economics of selected 40 MW *reheated* HAT power plants installed on a 4500 teu Panamax vessel are examined. A comparison of annual cost of machinery is presented for selected design vectors previously analysed in design space analysis in section 3.4.2.

5.2.1 Methodology

The HAT cycles herein analysed are selected from the Pareto front —i.e., non dominated designs in terms of high thermal efficiency and low SPEC— generated in the analysis of the design space (section 3.4.2). Therefore, the selected cycles feature, among the cycle analysed in the design exploration of the design space, the best trade-off between thermal efficiency and purchase equipment cost (see figure 5.2).

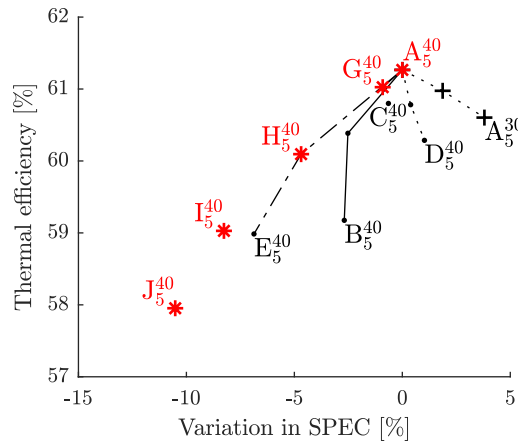


Figure 5.2: Selected cycle designs (in RED) for the economic analysis.

The design parameters, the performance at maximum power output and the SPEC of the selected cycles are presented in table 5.3. A_5^{40} is the cycle with maximum thermal efficiency, achieved with maximum effectiveness of all heat exchangers, maximum OPR and minimum pinch temperature difference in the humidifier, hence maximum technology level. The designs vectors of the cycle G_5^{40} , H_5^{40} feature respectively reduced after-cooler effectiveness ($\varepsilon_{AC} = 85\%$) and recuperator effectiveness ($\varepsilon_{REC} = 85\%$). The recuperator effectiveness in cycles I_5^{40} and J_5^{40} is 85% and 80% whereas, for both cycles, the effectiveness of the inter-cooler and after-cooler is 85%.

Table 5.3: Design variables values of the baseline cycle A_5^{40} .

	A_5^{40}	G_5^{40}	H_5^{40}	I_5^{40}	J_5^{40}
Maximum power performance					
Thermal efficiency [%]	61.3	61.0	60.1	59.0	58.0
Power output [MW]	40	40	40	40	40
Design parameters					
Overall pressure ratio [-]	40:1	40:1	40:1	40:1	40:1
Turbine inlet temperature [K]	1600	1600	1600	1600	1600
IC effectiveness [%]	95	95	95	85	85
AC effectiveness [%]	95	85	95	85	85
REC effectiveness [%]	90	90	85	85	80
EC effectiveness [%]	95	95	95	95	95
SAT pinch temperature difference [K]	5	5	5	5	5
Saturator bypass (β) [%]	50.2	46.7	52.15	48.1	49.7
Air inlet mass flow [kg/s]	40.9	41.5	39.9	40.9	40.0
SPEC [\$ /kW]	353.9	350.7	337.3	324.7	316.7

The main characteristics of the ship are reported in table 5.4 and are based on average ship particulars of the selected class of vessel as reported by MAN Diesel & Turbo [115]. In the assumption that the cruise design speed is 24.5 knots, the ship requires a power plant that produces around 40 MW at maximum continues rating (MCR) [115]. The required installed power accounts for a 10% engine margin and a 15% sea margin.

It is assumed that the vessel operates on a transatlantic round route (Rotterdam $\rightarrow$ Norfolk $\rightarrow$ Miami $\rightarrow$ Houston $\rightarrow$ Norfolk $\rightarrow$ Rotterdam) for a total of 11 000 nm per round, at a cruise speed of 24.5 knots —i.e., approximately 18 days of open sea cruise. The vessel sails about 70% of the route in temperate and 30% in tropical latitudes. The variation of ambient and sea temperature are accounted for as reported in section 5.1.1. The seasonal —i.e., summer and winter— average

Table 5.4: Panamax vessel characteristics.

Container ship class		Panamax
Ship size	teu	4 500
Scantling draught	m	13.3
Deadweight (scantling)	dwt	58 500
Length overall	m	286
Length between perpendiculars	m	271
Breadth	m	32.2
Cruise speed	knots	24.5

sea water and ambient air temperature [116] are reported in table 5.5 both for temperate and tropical latitudes. The amplitude of the variation in ambient temperature is assumed to be 20 K for both latitudes and seasons. It is also assumed that the time spent sailing in winter and summer are equal.

Table 5.5: Sea and ambient temperatures [116].

	Temperate sea		Tropical sea	
	Winter	Summer	Winter	Summer
Average sea water and ambient temperature	287.15 K	292.15 K	301.15 K	302.15 K

Figure 5.3 shows the impact of the latitude and season on the 24h average fuel consumption for cycle A_5^{40} and power required by the vessel in ideal sea state conditions —i.e., absence of wind and waves. The power required by the vessel for cruise navigation reduces when the sea water temperature increases (see figure 5.3b) as a result of the reduction in viscosity of the sea water. The break power required for navigation in tropical summer seas is 2.9% lower than the power required for cruise in temperate winter seas. Navigation at cruise speed in tropical seas results in lower fuel consumption than at temperate latitudes due to the lower power required by the vessel (see figure 5.3a). However, the thermal efficiency of the HAT power plant, as see in section 4.2.5, drops as the temperature of the air increases. Therefore, although the power requirement in summer tropical seas is 0.2% lower than in winter tropical seas, the 24h average fuel consumption in summer is 0.2% higher than in winter.

The impact of the sea state on the fuel consumption and navigation speed is shown in figure 5.4 for winter navigation in temperature latitudes. The fuel consumption increases by 12.9% when passing from sea state 1 to 5 due to the 12.7% increase in break power required for cruise speed at 24.5 knots. For sea states higher than 5

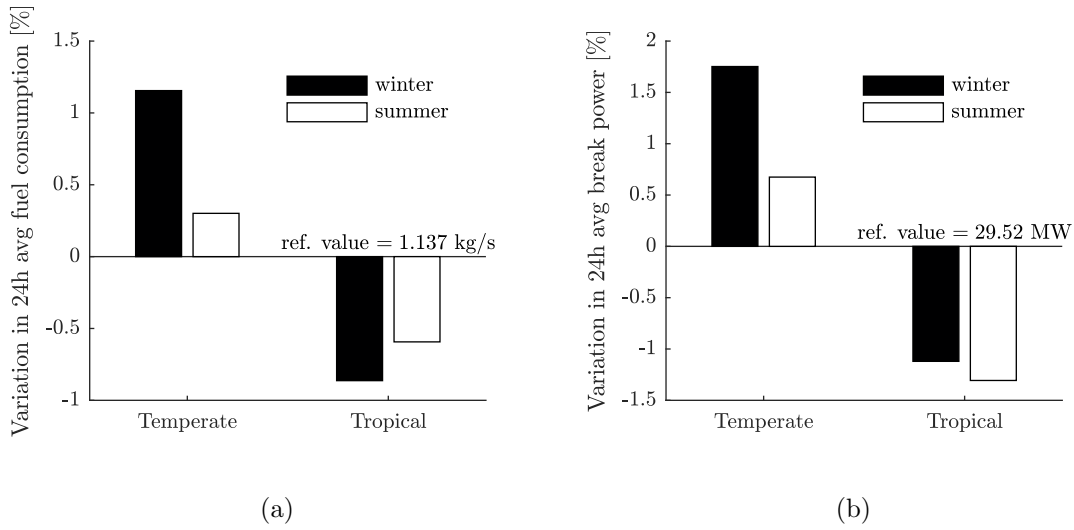


Figure 5.3: Impact of latitude and season on the 24h average (a) fuel consumption and (b) break power.
Sea state = 0.

the required break power for the cruise speed of 24.5 knots exceeds the maximum power output of the HAT power plant, the vessel speed is therefore reduced to 23.8 knots. Thus, the fuel consumption rises by 29.4% when passing from sea state 1 to 6.

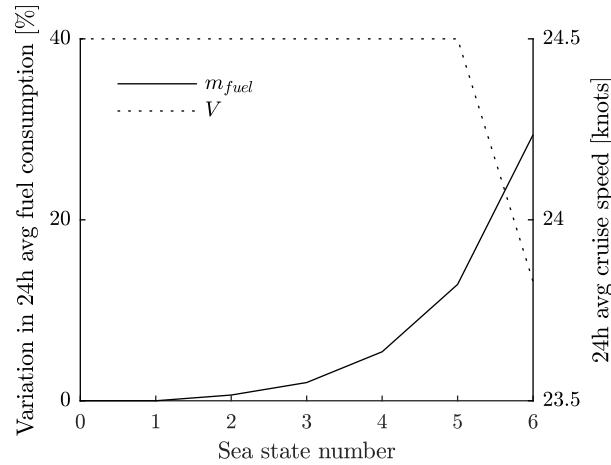


Figure 5.4: Impact of the sea state on the 24h average break power and cruise speed.
Temperate winter.

The percentage of navigation time at a given sea state is determined by the distributions of wave height in the Atlantic ocean during the summer and winter season. The distribution of the wave height is modelled with two 2-parameter Weibull probability density functions (pdf), determined by Galanis et al. [117] for

the Atlantic ocean, as shown in eq. 5.10.

$$f(x) = \frac{\alpha}{\beta} \left(\frac{x}{\beta} \right)^{\alpha-1} e^{-1(x/\beta)^\alpha} \quad \begin{array}{l} \text{(winter: } \alpha = 3.42; \beta = 2.06) \\ \text{(summer: } \alpha = 3.39; \beta = 2.73) \end{array} \quad (5.10)$$

Where x is the wave height and $f(x)$ is the wave height Weibull pdf.

In winter the vessel navigates 20.2% of the time in sea state 3, 68.8% in sea state 4 and 11% in sea state 5. Whereas, in summer the sailing time in sea state 3, 4 and 5 are respectively 8.5%, 48.6%, and 42.9%.

The lifetime of the investment is assumed to be 20 years. The impact of discount rate is analysed to reflect the risk (uncertainty) of the investment. A discount rate of 4% is assumed for the *risk-free* scenario and an 18% is adopted for the worst case scenario. For an average case scenario 10% discount rate is considered [118]. A range of MDO price, from 360 \$/t to 1080 \$/t, is analysed to account for the volatility of the fuel market.

5.2.2 Results

The power plant with the minimum annual cost of machinery across all the range of discount rates, fuel price analysed and assuming 12 round trips is cycle A_5^{40} . In the reference scenario of 12 round trips, 10% discount rate and MDO fuel price of 540 \$/t —i.e., average risk scenario and similar to current year average fuel price [119]— the CAPEX accounts for 11.4% of the AK, with 1.66 M\$/year, whereas the OPEX is 12.8 M\$/year —i.e., 88.6% of AK.

The variation in CAPEX, OPEX and AK of the analysed cycles is shown in figure 5.5a, where the cycle with the lowest capital expenditure J_5^{40} is used as reference. Within the reference scenario, plant configuration I_5^{40} , H_5^{40} , G_5^{40} and A_5^{40} feature respectively 2.5%, 6.5%, 10.7% and 11.75% higher CAPEX than cycle J_5^{40} . However, the annual cost of machinery for the above mentioned cycles is respectively 1.6%, 4.3%, 5.4% and 5.5% lower than the AK of cycle J_5^{40} , consequence of the higher thermal efficiency of the cycles with higher CAPEX, that results in lower operation costs. The annual savings², with cycle J_5^{40} as basis, are 241.7 k\$/year, 654.8 k\$/year, 834.8 k\$/year and 848.8 k\$/year for cycle I_5^{40} , H_5^{40} , G_5^{40} and A_5^{40} .

Figure 5.5b correlates the operation expenditure with the variation in capital expenditure of the analysed cycles. The variation in OPEX as function of CAPEX is linear between cycles J_5^{40} , I_5^{40} and H_5^{40} and the ratio $\Delta\text{OPEX}/\Delta\text{CAPEX}$ is -7.73 —i.e., every extra dollar spent in CAPEX results in a 7.73 \$ reduction in OPEX.

²annual savings = $\text{AK}_{J_5^{40}} - \text{AK}$

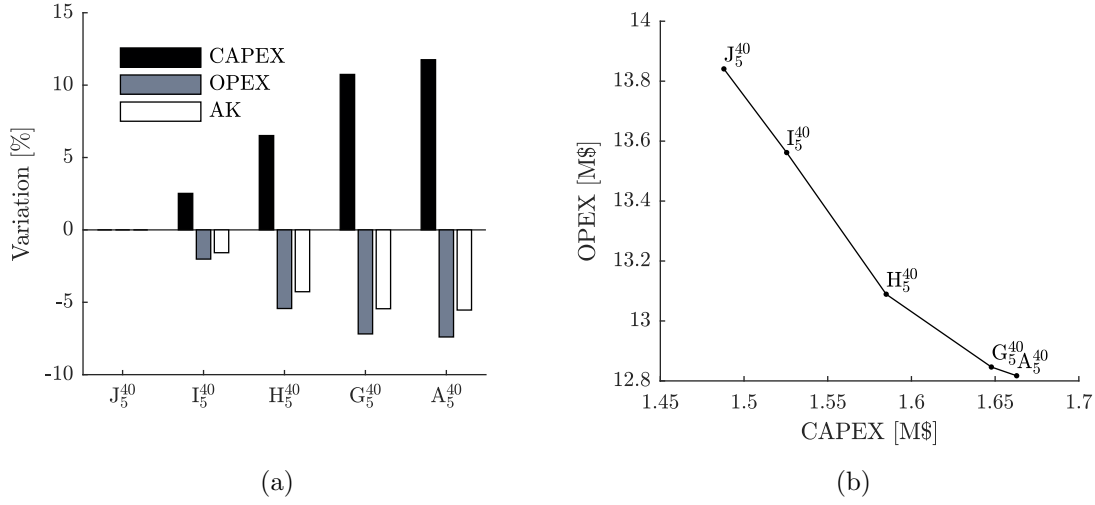


Figure 5.5: Variation in Annual cost of machinery, CAPEX and OPEX for power plants J_5^{40} , I_5^{40} , H_5^{40} , G_5^{40} and A_5^{40} with reference cycle J_5^{40} . 12 round trips/year, 10% discount rate and MDO price = 540 \$/t.

The ratio $\Delta\text{OPEX}/\Delta\text{CAPEX}$ decreases to -3.8 and -2 when comparing cycle H_5^{40} with G_5^{40} and cycle G_5^{40} with A_5^{40} . Therefore, the additional capital investment needed for cycles G_5^{40} and A_5^{40} has a smaller impact in the reduction of the OPEX compared to cycle I_5^{40} and H_5^{40} .

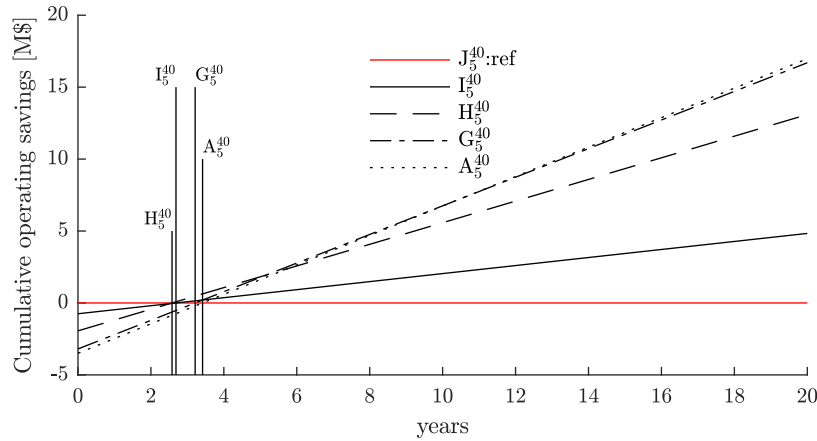


Figure 5.6: Cumulative operating savings in installing the HAT cycles I_5^{40} , H_5^{40} , G_5^{40} and A_5^{40} over the reference cycle J_5^{40} . 12 round trips/year, 10% discount rate and MDO price = 540 \$/t.

The relative cumulative operating savings and the relative payback periods for each cycle (see eq. 5.9, 5.8) are shown in figure 5.6, with cycle J_5^{40} used as basis and within the reference scenario. The payback period for cycle H_5^{40} and I_5^{40} are very similar, respectively 2.6 and 2.7 years. However, the investment in cycle H_5^{40} results in a total saving of 13.1 M\$ over 20 years, whereas the investment in cycle

I_5^{40} results in a total saving of only 4.8 M\$. The payback period of cycles G_5^{40} and A_5^{40} is 3.2 and 3.4 years, and the total savings over 20 years in operating cycle G_5^{40} and A_5^{40} instead of cycle J_5^{40} are respectively 16.7 M\$ and 17 M\$. Thus, among the analysed cycles and with cycle J_5^{40} as basis, the investment in cycle A_5^{40} allows the highest savings over the lifetime of the investment, however it is also the investment subject the highest risk, due to the highest capital cost and longer payback period. Moreover, cycle G_5^{40} can achieve comparable saving with a shorter payback period.

The impact of the number of round trips —i.e., utilisation of the power plant—, fuel price and discount rate on the annual savings and payback period are shown in figure 5.7, with cycle J_5^{40} used as basis. For each cycle, the variations in annual savings is referred to the annual savings of the specific cycle within the reference scenario —i.e., 12 round trips, 10% discount rate and MDO fuel price of 540 \$/t—, as shown in eq. 5.11.

$$\text{Var. in annual savings} = \frac{(AK_{J_5^{40}} - AK)}{(AK_{J_5^{40}} - AK)_{ref}} - 1 \quad (5.11)$$

The small difference in impact of the round trips per year, fuel price and discount rate on the variation of annual savings of each cycle is the consequence of the similar OPEX contribution to the annual cost of machinery ($\sim 90\%$) of all the analysed cycles. The impact of the number of round trips, fuel price and discount rate on the variation in payback period of the cycle is, instead, the same for each cycle and is shown with the red line in figure 5.7.

The number of round trips per year are changed from 10 to 16 —i.e., from 187 to 299 days of open sea navigation. The utilisation of the power plant affects the operation expenditure of the power plants, due to the variation of both the fuel and the maintenance cost. When the round trips per year are increased from 12 to 16 (see figure 5.7a), the maximum increase in annual savings of 40% is achieved by cycle A_5^{40} , whereas the minimum increase of 38% is achieved by cycle H_5^{40} . When the round trips per year are reduced from 12 to 10 the annual savings of cycle A_5^{40} drop by 20%, whereas by 19% for cycle H_5^{40} . As the operation savings increase for increased utilisation of the power plant the payback period decreases. If the round trips per year are increased from 12 to 16 the payback period of all analysed cycles drops by 25%, whereas it increases by 20% when the number of round trips per year are reduced to 10.

A wide range of MDO fuel price, from 360 \$/t to 1080 \$/t, is analysed to reflect the the volatility of the market. The variation of the fuel price affects the fuel cost of the power plant and therefore the saving in OPEX. In a scenario of fuel

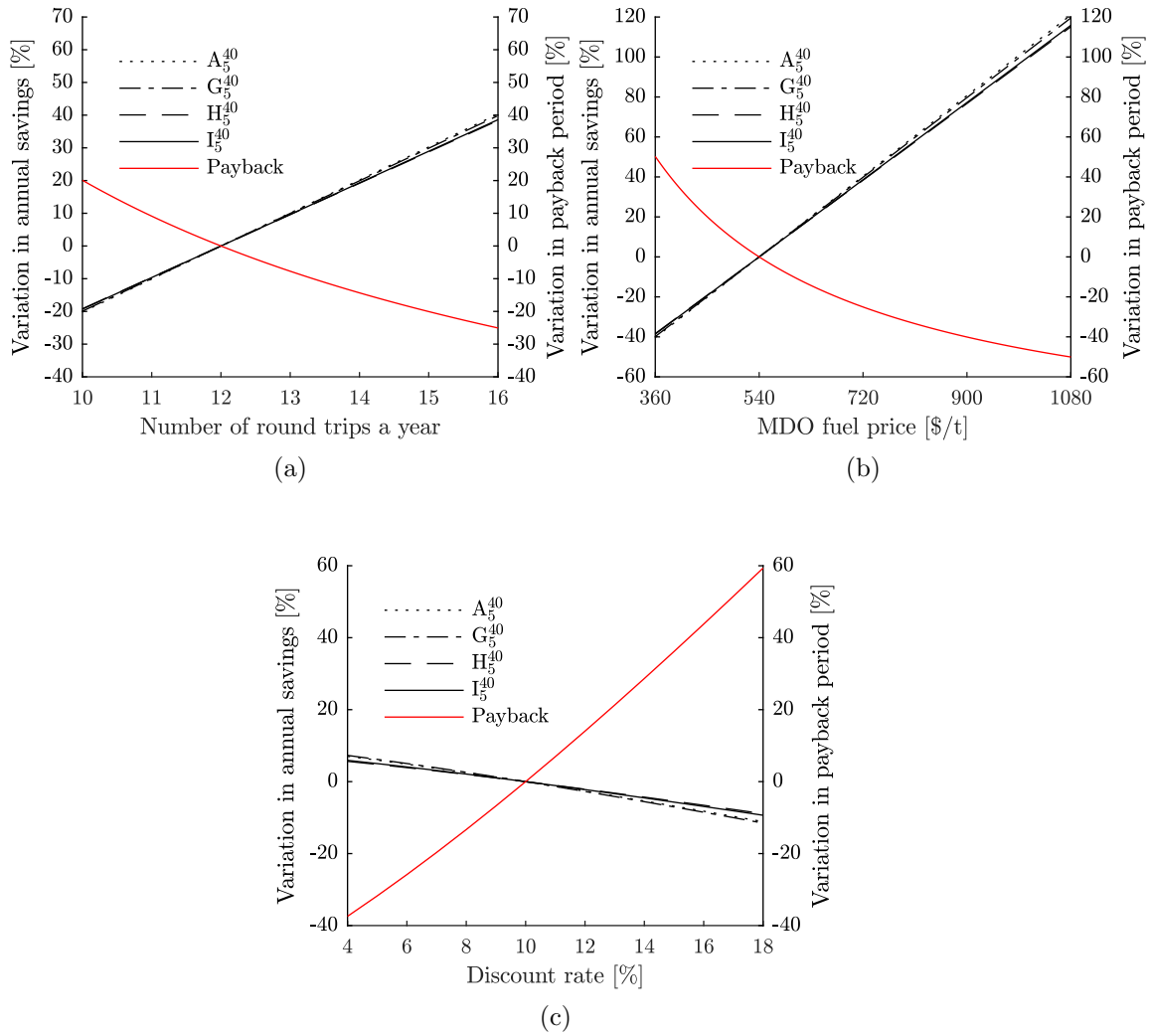


Figure 5.7: Impact of (a) round trips a year, (b) fuel price and (c) discount rate on the annual savings and payback period with cycle J_5^{40} as basis. reference: 12 round trips/year, 10% discount rate and MDO price = 540 \$/t.

price of 1080 \$/t the annual savings rise to a maximum of 120% for cycle A_5^{40} and to a minimum of 114% for cycle H_5^{40} compared to when the fuel price is 540 \$/t (see figure 5.7b). The payback period drops by 50%, meaning that all cycles have a payback time of less than 1 years and 9 months. If the fuel price is 360 \$/t the savings of cycle A_5^{40} and H_5^{40} reduces respectively by 40% and 38% and the payback period increases by 50%. The payback period of all cycles is, therefore, longer than 4 years.

Compared to the round trips per year and the fuel price, the discount rate has limited impact on the annual saving of the power plants and has sever impact the payback period of the investment (see figure 5.7c). If the discount rate is reduced from 10% to 4% —i.e., *free-risk* scenario— the payback period reduces

by 37%, and the annual savings increase from a minimum of 5.6% for cycle H_5^{40} to a maximum of 7.3% for cycle G_5^{40} . Whereas, if the discount rate is increased from 10% to 18%, the payback period increases by 59%, and the annual savings drop from a minimum of 8.9% for cycle H_5^{40} to a maximum of 11.5% for cycle G_5^{40} .

5.2.3 Conclusions

The economic analysis has been presented for selected *reheated* HAT cycles powering a Panamax container ship that operates a transatlantic route from Rotterdam to Huston. The four HAT cycles analysed are selected from the Pareto front obtained from the analysis of the design space of *reheated* HAT cycles. The estimation of the operating costs accounts for the change in performance of the power plant during the day, due to the ambient temperature variation, and of the impact of the sea water temperature on both the performance of the power plant, and on the power required to sail at cruise speed. The results also account for the impact of the wave height and wind intensity on the fuel consumption and cruise speed. A statistical model is used to estimate the navigation time in each sea state.

Cycle J_5^{40} —i.e., cycle with the lowest SPEC and lowest thermal efficiency— is used as reference for the economic evaluation of the remaining cycles. Among the analysed designs, cycle A_5^{40} —i.e., cycle with the highest thermal efficiency— achieves the lowest annual cost of machinery but also maximum capital expenditure, when operating 12 round trips per year, with a MDO fuel price of 640 \$/t and a discount rate of 10%. However, cycle A_5^{40} features a payback period of 3.4 years, the longest among the analysed cycles.

5.3 Comparison with Two-stroke Diesel Engine

This section presents a direct comparison between the annual cost of machinery of a two-stroke diesel engine compliant with IMO tier II regulations and the *reheated* HAT cycle operating a Handymax bulk carrier. The cost analysis data for the diesel engine used herein is derived from the study on the potential of waste heat recovery systems coupled with diesel engines by Theotokatos et al. [120].

For the same marine route the annual costs of the HAT cycle are also compared with updated diesel engine annual costs, which include the additional costs related to EGR and EGC systems or IMO compliant fuels necessary to comply with the tier III of IMO regulation. The capital and operation cost related to both the EGR and EGC are based on the results of the *Reduction of SO_2 , NO_x and Particulate Matter from Ships with Diesel Engines* technical report [17].

5.3.1 IMO Tier II 2-stroke Diesel vs. HAT Power Plant

The examined carrier is assumed to transport ore from Rio de Janeiro to Rotterdam. Each round trip covers a distance of 10 500 nm for a duration of approximately 30 days of navigation at cruise speed. Two scenarios of 8 and 11 round trips a year are analysed. The main characteristics of the Handymax vessel and of the reference two-stroke diesel engine are reported in table 5.6. In this section, the influence of the sea state and wind condition are not accounted for —i.e., zero sea state number. The ambient and sea temperature are kept constant at respectively 293.15 K and 288.15 K. The brake power required by the vessel for a cruise speed of 14.5 knots is 7.5 MW. Port manoeuvring and docking/undocking procedures are not considered within this study, as the prime mover is the main focus of the study —i.e., AK, CAPEX and OPEX do not account for the auxiliary engines.

Table 5.6: Handymax vessel and diesel engine characteristics [120].

Bulk carrier class		Handymax
Scantling draught	m	12.53
Deadweight (scantling)	dwt	55 000
Length between perpendiculars	m	185
Breadth	m	32
Cruise speed	knots	14.5
Engine type		Two-stroke
Bore	mm	500
Stroke	mm	2000
Brake power at MCR	kW	8815
Engine speed	r/min	122
SFOC at MCR	g/kWh	171% $\pm$ 5%
MCR: maximum continuous rating		
SFOC: brake-specific fuel oil consumption		

The *reheated* HAT cycle analysed in this section has a maximum power output (MCR) of 9 MW and is a scaled down version of the 40 MW A_5^{40} cycle analysed in section 3.4.2. The maximum power of 9 MW is achieved for ambient and sea temperature of 288.15 K and the design vector and performance of the cycle are reported in table 3.5 for exemption of the power output and inlet air mass flow that drops to 9.3 kg/s.

The purchase equipment cost of the HAT power plant is calculated as described in section 3.3, whereas the investments cost of the two-stroke diesel engine is derived

from Theotokatos et al. [120]. The diesel engine PEC value from the paper is converted from euro to US dollars at 2012 average exchange rate ($\$/\text{€} = 0.778$ [121]) and updated to 2015 value by means of the engine equipment manufacturing price index of diesel engines [122]. The lifetime of the investment is assumed to be 20 years.

The fuel and oil consumption of the diesel engine, for both the 8 and 11 round trips scenarios, is derived from Theotokatos et al. [120], whereas the fuel consumption of the HAT power plant is obtained from the simulation of the route with Poseidon as described in section 5.1.1.

Three scenarios of discount rate are analysed to reflect the risk (uncertainty) of the investment. A discount rate of 4% is assumed for the *risk-free* scenario, a 10% discount rate is considered for the average risk case [118] and a 18% is adopted for the worst scenario. A range of HFO price, from 257 \$/t to 771 \$/t, is analysed to account for the volatility of the oil market, whereas it is assumed that in general MDO price remains in time 40% higher than HFO [119, 120].

Table 5.7 and table 5.8 report the calculated annual cost of the machinery (AK), for respectively 8 and 11 round trips per year, of both power plants. The payback time of the investment is also calculated as the difference in total capital over the lifetime of the investment divide by the savings in operation expenditure. In all the scenarios herein studied the annual cost of machinery of the analysed *reheated* HAT cycle is lower than the annual cost of machinery of the analysed two-stroke diesel engine. Operating the HAT power plant, within the scenario of 8 round trips per year, free-risk discount rate and low fuel price (HFO = 257 \$/t & MDO = 360 \$/t), results in annual saving³ of 172 k\$/year. The observed investment payback time is 7 years. For the same number of round trips, but with a discount rate of 18% and high fuel price (HFO = 771 \$/t & MDO = 1080 \$/t), the annual savings are 93 k\$/year and the investment payback time is 12 years (see table 5.7). The annual savings increase and the payback time drops when the round trips are increased to eleven. For *free-risk* discount rate and low fuel price, the annual savings and payback time of the investment are respectively 279 k\$/year and 5 years, whereas for 18% discount rate and high fuel price the annual savings and payback time of the investment are respectively 182 k\$/year and 11 years (see table 5.8).

The scenario with 11 round trips per year, with a discount rate of 10% and HFO and MDO price of 386 \$/t and 540 \$/t —i.e., average risk scenario and similar to current year average fuel price [119]— is used as a reference in the following analysis. The annual cost of machinery, the CAPEX and the OPEX of both power

³annual saving = $AK_{\text{Diesel}} - AK_{\text{HAT}}$

Table 5.7: Two-stroke diesel engine and HAT power plant annual costs of machinery assuming 8 round trips per year.

HFO price	MDO price	Two-stroke AK	r HAT AK	Payback time
[\$/t]	[\$/t]	[k\$/year]	[k\$/year]	[years]
<i>discount rate: 4%</i>				
257	360	2 796	2 623	7
386	540	3 850	3 689	7
514	720	4 903	4 754	7
643	900	5 957	5 820	8
771	1080	7 010	6 885	8
<i>discount rate: 10%</i>				
257	360	2 987	2 827	9
386	540	4 040	3 892	10
514	720	5 094	4 957	10
643	900	6 148	6 023	11
771	1080	7 201	7 088	11
<i>discount rate: 18%</i>				
257	360	3 289	3 148	12
386	540	4 342	4 213	13
514	720	5 396	5 279	13
643	900	6 450	6 344	14
771	1080	7 503	7 410	14

Table 5.8: Two-stroke diesel engine and HAT power plant annual costs of machinery assuming 11 round trips per year. RED: reference scenario.

HFO price	MDO price	Two-stroke AK	r HAT AK	Payback time
[\$/t]	[\$/t]	[k\$/year]	[k\$/year]	[years]
<i>discount rate: 4%</i>				
257	360	3 724	3 444	5
386	540	5 173	4 909	5
514	720	6 621	6 375	5
643	900	8 070	7 840	6
771	1080	9 519	9 305	6
<i>discount rate: 10%</i>				
257	360	3 915	3 648	7
386	540	5 364	5 113	7
514	720	6 812	6 578	7
643	900	8 261	8 043	8
771	1080	9 710	9 508	8
<i>discount rate: 18%</i>				
257	360	4 217	3 969	9
386	540	5 666	5 434	10
514	720	7 114	6 900	10
643	900	8 563	8 365	11
771	1080	10 012	9 830	11

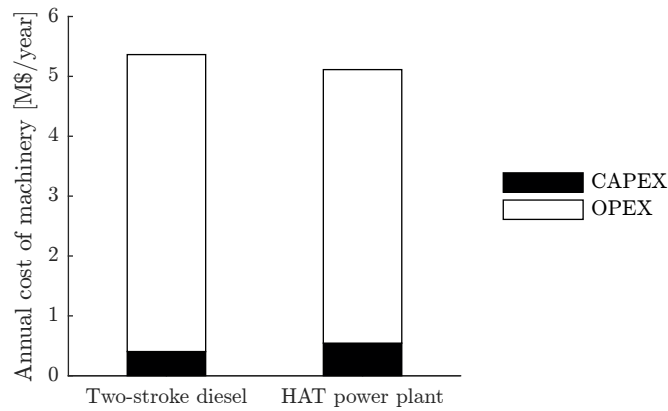


Figure 5.8: Annual cost of machinery of the Tier II compliant two-stroke diesel engine and the *reheated* HAT cycle for the reference scenario. 11 round trips/year, 10% discount rate, HFO price = 386 \$/t and MDO price = 540 \$/t.

plants within the reference scenario are shown in figure 5.8 and highlighted in red in table 5.8. The capital expenditure of the HAT power plant is 35% higher than for the CAPEX of the two-stroke diesel engine and contributes by 11% to the annual machinery cost, whereas the CAPEX of the diesel engine is only 7% of the AK. Therefore, the annual cost of machinery of the HAT power plant is more sensitivity to the variation of discount rate than the two-stroke diesel —i.e., for high discount rates the difference in annual cost of machinery between the two power plants reduces (see table 5.7 & 5.8).

The lack of experience in operating a HAT cycle results in a high uncertainty in the evaluation of the maintenance cost when installing the latter power plant on a vessel. Within the reference scenario, figure 5.9 illustrates the impact of increased annual maintenance cost on the payback time and overall savings after 20 years of investment. An increase of the annual maintenance cost, calculated as shown in eq. 5.7, by 25% and 50% result respectively in a reduction in annual savings by 17.3% and 35.3%, whereas the payback time increases to 8 and 9 years respectively.

In table 5.7 and table 5.8 it can be noticed that a same percentage increase in fuel price of both HFO and MDO results in an increase in the payback time and a decrease in the difference of AK of the two power plants. Although the fuel consumption of the HAT power plant is significantly lower than for the diesel engine, the annual savings decrease when the price of fuel rises, due to the increase in fuel price difference. The variation in payback period and cumulative operating savings when only the price of the MDO fuel is varied is shown in figure 5.11. Within the reference scenario the HFO price is kept constant at 386 \$/t whereas

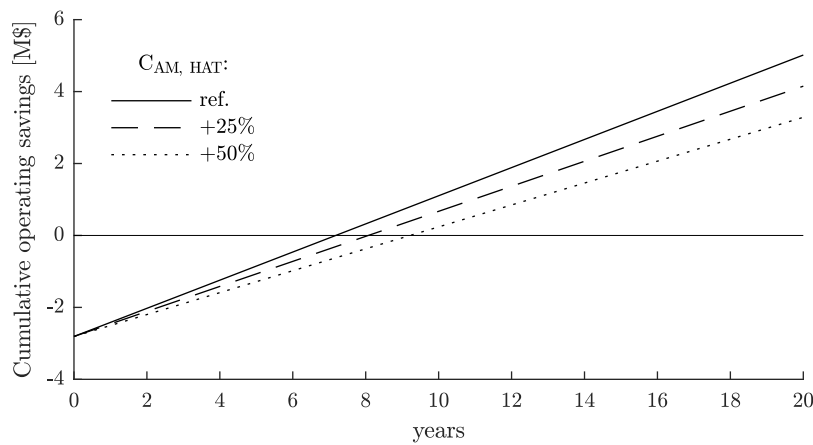


Figure 5.9: Cumulative operating savings in installing the HAT cycle over the two-stroke diesel engine for increased annual maintenance costs for the reference scenario.

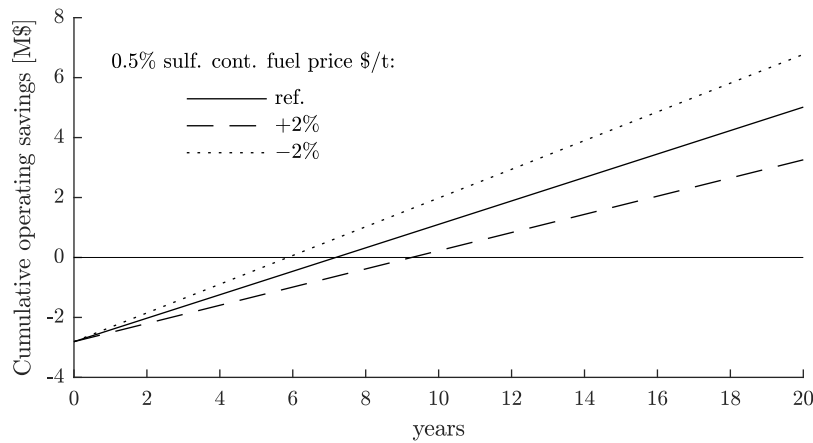


Figure 5.10: Cumulative operating savings in installing the HAT cycle over the two-stroke diesel engine for different MDO fuel prices for the reference scenario.
HFO price = 386 \$/t and MDO reference price = 540 \$/t.

the the price of MDO from 540 \$/t is increased and reduced by 2%. An increase in MDO fuel price by 2% results in an increases in the payback time from 7 to 9 years and the annual savings drop by 35% whereas a reduction in MDO price by 2% results in a reduction of payback time to 6 years and an increase in annual saving by 35%. This highlights the severe impact fuel price fluctuations can have on the profitability of the investment. An increase of MDO price by 5.7% compared to HFO is sufficient to even the annual cost of machinery of the two power plants.

5.3.2 IMO Tier III 2-stroke Diesel vs. HAT Power Plant

The application of Tier III [12] new limitations, ratified by IMO, for NO_x and SO_x emissions has driven the development of technologies for the diesel engines to meet the new environmental norms. Exhaust Gas Recirculation (EGR) and Selective Catalytic Reduction (SCR) are two measures diesel engines can adopt to withstand the NO_x emission limitations, whereas Exhaust Gas Cleaning (EGC) system is necessary for diesel engines to run on HFO with 3.5% sulphur content and still meet the Tier III limitations in SO_x emissions both in and out of the ECAs.

In this section a comparison between the operation and capital expenditure of the 9 MW *reheated* HAT previously described (see section 5.3.1) and a Tier III diesel engines is presented. The calculation of the penalties in CAPEX and OPEX of a new Tier III diesel engine compared to Tier II diesel engines, are derived from the results of the techno-economic study performed by Hansen et al. [17] for the Danish Environmental Protection Agency. However, no assumptions have been made on the possible gains in terms of thermal efficiency Diesel engines could experience by future technology development, and the consequent future reductions in OPEX or/and CAPEX for these prime movers. This study is therefore based on current values of efficiency and purchase costs of the diesel engines.

The same marine route, vessel and diesel engine specifications (see table 5.6) analysed in section 5.3.1 are used within this study. It is assumed that the handymacx bulk carrier completes eleven round trips per year from Rio de Janeiro to Rotterdam, and that the discount rate is 10%. For the purpose of this analysis it assumed that the ship operates 10% of the time in ECAs areas. The investments cost of the two-stroke diesel engine without EGR, EGC and water treatment systems is derived from Theotokatos et al. [120], whereas the PEC of the HAT power plant is calculated as described in section 3.3. The lifetime of the investments are assumed to be 20 years.

Two solutions are analysed for the diesel engine to comply with the Tier III criteria. In the first case the prime mover runs on 0.1% sulphur content fuels when in navigation inside the ECAs, whereas runs on 0.5% sulphur content fuel outside the ECAs. In the second case an EGC systems installed and 3% sulphur content (HFO)fuel is used inside and outside of the ECAs. In both cases an EGR system is installed to floor the NO_x emissions for navigation within the ECAs, outside the latter areas the EGR does not operate as current diesel engines comply with global NO_x emissions limits [17]. To comply with the SO_x emissions regulation, the *reheated* HAT power plant runs on 0.1% sulphur content fuels when in navigation inside the ECAs, whereas runs on 0.5% sulphur content fuel outside the ECAs.

The specific purchase cost of only the EGR and the water treatment system is assumed to be 85.5 \$/kW, whereas the SPEC of the combined EGR and EGC systems with the water treatment system is assumed to be 203.7 \$/kW [17]. These costs do not account for the additional investment that the installation of these systems would require.

The penalties in operation expenditure are due to the NaOH consumption by both the EGR and EGC systems, the maintenance cost these systems require and the power necessary for operating the auxiliary systems. Table 5.9 reports the penalties, derived from Hansen et al. [17], assumed within this analysis, for navigation within and outside ECAs. The price of NaOH, and fuel with 3%, 0.5% and 0.1% of sulphur content are assumed to be respectively 400 \$/t, 386 \$/t, 540 \$/t and 617 \$/t.

Table 5.9: Operation expenditure penalties for Tier III diesel engines.

	EGR		EGR & EGC	
	not ECA	ECA	not ECA	ECA
Fuel sulphur content [%]	0.5	0.1	3	3
Fuel price [\$/t]	540	617	386	386
SFOC penalty [\$/MWh]	–	2.47	0.15	1.56
NaOH [\$/MWh]	–	0.05	–	1.54
Power aux [\$/MWh]	–	3.28	1.76	4.5
Maintenance [\$/MWh]	–	0.18	0.52	0.61

The variation in CAPEX, OPEX and AK for the analysed cycles is shown in figure 5.11 where the diesel engine with the EGR system and running on IMO fuels is used as reference. The CAPEX of the diesel engine with the additional EGC system and of the HAT power plant is respectively 24.9% and 10.6% higher than for the reference engine. The OPEX of the diesel engine with the EGC systems and of the HAT power plant is respectively 22.5% and 30.1% lower than for the reference engine. Therefore, the additional investment in the EGR system or in the HAT results in an overall reduction of annual cost of machinery by 19.2% or 27.3%.

Figure 5.12 shows the cumulative operating savings for the investment in the HAT power plant or the additional EGR system for the diesel engine. The payback period of the diesel engine with EGR and EGC systems and of the HAT over the reference diesel engine with only the EGR system are respectively 1.5 years and 0.5 years.

The impact of the variation of the price of all fuels —i.e., 3%, 0.5% and 0.1% sulphur content fuels— on the cumulative operating savings and payback period

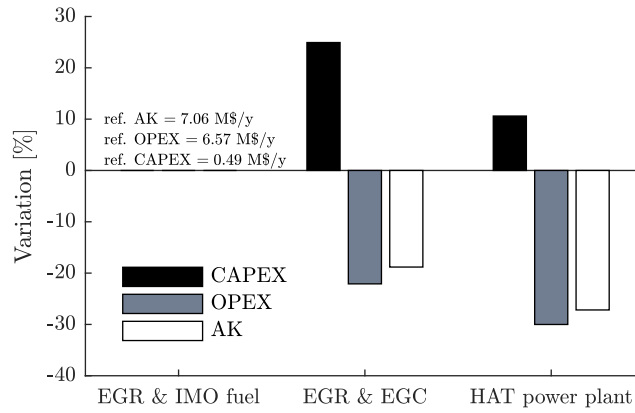


Figure 5.11: Annual cost of machinery of the Tier III compliant two-stroke diesel engines and the *reheated* HAT cycle for the reference scenario. 11 round trips/year, 10% discount rate, 3% sulph. fuel = 386 \$/t, 0.5% sulph. fuel = 540 \$/t and 0.1% sulph. fuel = 617 \$/t.

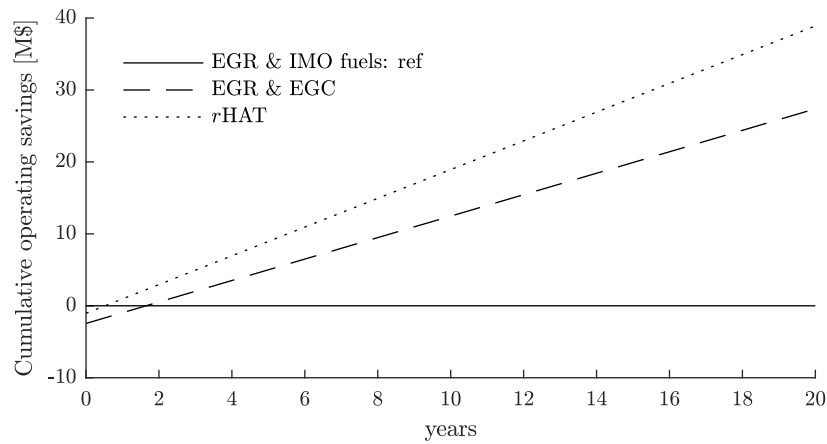


Figure 5.12: Cumulative operating savings in installing the diesel engine with EGR and EGC systems and the *reheated* HAT cycle over the reference diesel engine with EGR system and running on IMO fuels. 11 round trips/year, 10% discount rate, 3% sulph. fuel = 386 \$/t, 0.5% sulph. fuel = 540 \$/t and 0.1% sulph. fuel = 617 \$/t.

of the investment in the HAT power plant and the diesel engine with both EGR and EGC systems are shown in figure 5.13. The payback period for the additional investment of the EGR system drop from 1.5 years to 1 and 0.8 year when the fuel price increase respectively by 33% and 66%. Whereas, for the same fuel price variation the payback period for the HAT power plant over the reference diesel engine is reduced from 0.5 years to 0.4 and 0.3 year.

When the fuel price increase by 33%, the annual cost of machinery on the diesel engine with both EGR and EGC systems and of the HAT cycle rise by respectively 25% and 35%, and as a result the AK of the diesel engine with both EGR and EGC

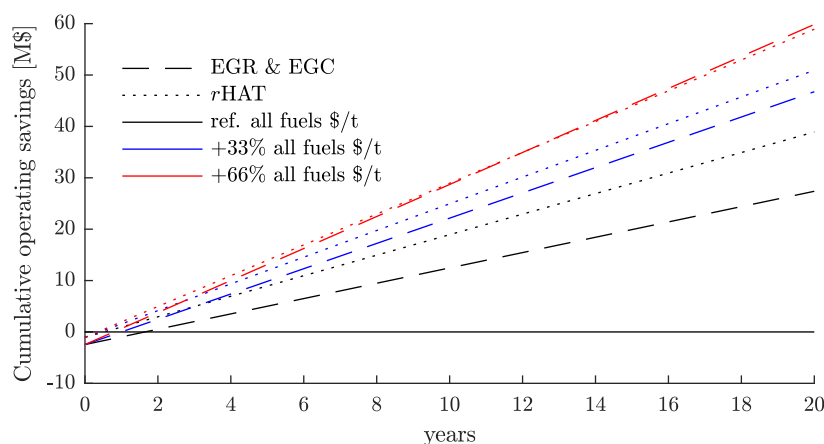


Figure 5.13: Cumulative operating savings in installing the diesel engine with EGR and EGC systems and the *reheated* HAT cycle over the reference diesel engine with EGR system and running on IMO fuels for variations in all fuels price.

11 round trips/year, 10% discount rate, 3% sulph. fuel ref. = 386 \$/t, 0.5% sulph. fuel ref. = 540 \$/t and 0.1% sulph. fuel ref. = 617 \$/t.

systems is only 3% higher than for the HAT power plant. With a 66% increase in fuel price the AK of the diesel engine with both EGR and EGC systems is 1% lower than for the HAT cycle.

The impact of variations in price of only the 0.1% and 0.5% sulphur content fuels on cumulative operating savings and payback period are shown in figure 5.14. The 3% sulphur content fuel price is kept constant whereas the 0.1% and 0.5% sulphur content fuel price are varied by +10% and -10%. The payback period for the HAT cycle does not vary significantly when the price of IMO fuels fluctuate —i.e., the HAT cycle and the reference diesel engine both run on IMO fuels. With a 10% reduction in IMO fuel price the payback period for the diesel engine with EGR and EGC systems increases from 1.5 to 2.6 years, whereas with a 10% increase in IMO fuel price the payback period reduces to 1.2 years.

Moreover, when the IMO fuel price increases by 10% the annual cost of machinery of the HAT cycle increase by 8.6%, and as a result the AK of the diesel engine with both EGR and EGC systems running on 3% sulphur content fuels is only 2.3% higher than the AK of the HAT cycle. On the contrary, when the price of the IMO fuels drops by 10% the AK of the HAT cycle reduces by 8.6%, and the AK of the diesel engine with EGR and EGC systems is 21.6% higher than for the HAT cycle. An increase in IMO fuel price by 13% would result in the annual machinery cost of the diesel engine and of the HAT power plant to even out.

Figure 5.15 illustrates how the cumulative operating savings are affected by the share of navigation within the ECAs. The annual cost of machinery of the HAT

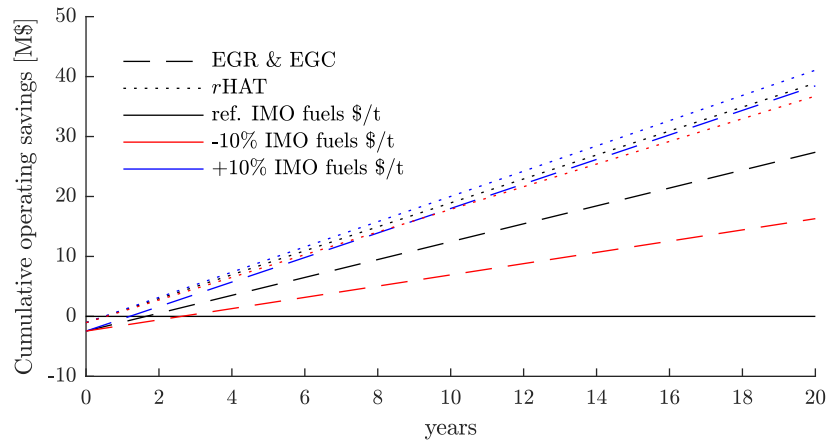


Figure 5.14: Cumulative operating savings in installing the diesel engine with EGR and EGC systems and the *reheated* HAT cycle over the reference diesel engine with EGR system and running on IMO fuels for variations in IMO fuels price.

11 round trips/year, 10% discount rate, 3% sulph. fuel = 386 \$/t, 0.5% sulph. fuel ref. = 540 \$/t and 0.1% sulph. fuel ref. = 617 \$/t.

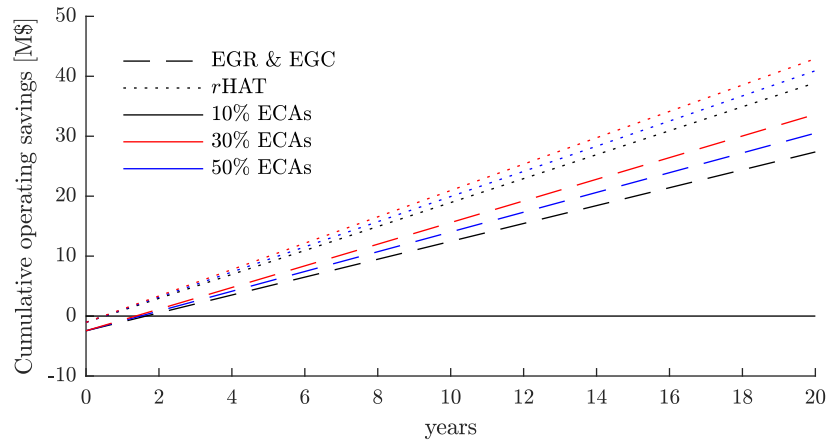


Figure 5.15: Cumulative operating savings in installing the diesel engine with EGR and EGC systems and the *reheated* HAT cycle over the reference diesel engine with EGR system and running on IMO fuels for variations in share of navigation time in ECAs.

11 round trips/year, 10% discount rate, 3% sulph. fuel = 386 \$/t, 0.5% sulph. fuel = 540 \$/t and 0.1% sulph. fuel = 617 \$/t.

cycle and of the diesel engine with EGR and EGC systems increase respectively by 4.8% and 2.4% when the navigation time share within the ECAs is increase from 10% to 50%. For the same increase navigation time share in ECAs, the payback period of the two prime movers is not significantly affected.

5.3.3 Conclusions

A direct comparison in terms of annual cost of machinery between a two-stroke diesel engine, compliant with IMO tier II regulations, and a HAT power plant has been examined, when powering a Handymax bulk carrier with a schedule of eight and eleven round trips from Rio de Janeiro to Rotterdam. The annual cost of machinery of the two power plants has been analysed for a range of discount rates and fuel prices and the HAT cycle has presented the lowest AK across the whole range. Within the reference scenario, the AK of the HAT cycle is 5% lower than for the diesel engine and payback period for the higher capital cost of the HAT cycle is 7 years, due to the lower operation expenditure.

The economic benefits of the investment, however are severely dependent on the relative change in MDO and HFO prices. Within the reference scenario, just a 5.7% increase in the relative MDO is necessary to even out the annual cost of machinery of the two power plants. Moreover, the use of MGO fuel during cruise for the HAT engine is not economically viable if compared with a two-stroke diesel engine running on HFO.

A further analysis has been performed to compare the annual costs of machinery of the *reheated* HAT and the diesel engine when compliant with the norms of the IMO Tier III regarding NO_x and SO_x emission regulation. The *reheated* HAT and the diesel engine with the EGR system run on low sulphur content fuels, whereas the diesel engine with both EGR and EGC systems runs on high sulphur content fuel.

From this analysis the HAT cycle results an economic viable solution for powering the vessel within the frame of IMO Tier III emissions regulation. Within the reference scenario the *reheated* HAT features an annual cost of machinery 11% lower than the diesel with both EGR and EGC systems and 38% lower than for the diesel engine with only the EGR system. The CAPEX of the HAT power plant is 10.6% higher than the CAPEX of diesel engine with only the EGR system whereas it is 11.5% lower than the CAPEX of the diesel engine with both EGR and EGC systems. With the diesel engine with the EGR system as reference the payback period for the investment in the HAT cycle is of half a year, whereas for the diesel engine with both EGR and EGC systems is 1.5 years.

The economic analysis of the HAT and the diesel engines, here presented, has demonstrated the potential of the HAT technology of the powering of commercial marine vessels. However, the immaturity of the HAT technology and lack of experience in operating and manufacturing this power plant are a significant source of uncertainty within the economic analysis. Moreover, it could be expected that, in

the future, further developments of diesel engines would also result in performance improvements.

5.4 Summary

In this chapter the methodology used for the integration of the previously described part-load and off-design performance methodology of the HAT cycle (see chapter 4) within Poseidon, a software for the simulation of a vessel performance, has been described. This method allows the estimation of the fuel consumption and cruise speed of a vessel in integrated full electric propulsion system (IFEP) configuration power by a HAT power plant for a given marine route. As a result the analysis of the annual cost of machinery, capital expenditure and operation expenditure of the *reheated* HAT cycle is made possible. The focus of this research is on the *power plant* and, therefore, no cost of installation and

With the latter methodology the annual cost of machinery of five selected *reheated* HAT cycle designs, from the Pareto front generated from the design space exploration in section 3.4.2, were analysed when powering a 4500 teu Panamax vessel operating a 11 000 nm per round Transatlantic route. The share of operation expenditure ($\sim 90\%$) dominates the annual cost of machinery of all the analysed cycles. The cycle A_5^{40} with the maximum thermal efficiency (61.2%) and the highest SPEC (353.9 \$/kW) has the lowest AK, 5.5% lower than the AK of the reference cycle J_5^{40} with the lowest thermal efficiency (58%) and lowest SPEC (316.7 \$/kW). However, cycle A_5^{40} is also the cycle with the longest payback period (3.4 years) and therefore subject to the highest financial risk.

Data available in literature has been used to estimate the annual cost of machinery of a diesel engine compliant with IMO Tier II emission regulations —i.e., proven and well known technology— and compare it with the annual cost of machinery of a 9 MW down-scaled version of the *reheated* HAT cycle A_5^{40} . Both engines were evaluated when powering a Handymax bulk vessel from Rio de Janeiro to Rotterdam. Within the reference scenario the AK of the HAT cycle is 5% lower than for the diesel engine and the payback period is 7 years. However, the economic benefits in investing in the HAT cycle over a Tier II diesel engine strongly depend on the price ratio between HFO and MDO fuels. Just a +5.7% variation in MDO fuel price is necessary to even out the AK of the two power plants.

Similar analysis is also performed for two diesel engine and the HAT power plant compliant with IMO Tier III norms for emission regulation. The annual cost of machinery has been estimated for a diesel engine with EGR system and the HAT power plant, both running on low sulphur content fuel, and a diesel engine with

both EGR and EGC systems and running in 3% sulphur fuel. From this analysis the HAT cycle results an economic viable solution for powering the vessel within the frame of IMO Tier III emissions regulation. The HAT cycle presents the lowest AK, 11% and 38% lower than for the diesel engine with only the EGR system and the diesel engine with both EGR and EGC systems. Moreover, the CAPEX of the HAT cycle is 11.5% lower than for the diesel engine with both EGR and EGC system, and 10.6% higher than for the diesel engine with only the EGR system. The payback period of the addition investment of HAT cycle over the diesel engine with the EGR system is of 0.5 years in the reference scenario.

The immaturity of the HAT technology and lack of experience in operating and manufacturing this power plant are a significant source of uncertainty within the economic analysis here presented. At this stage, a significant further development of the HAT cycle is still required to improve the accuracy of the economic comparison between HAT and diesel prime movers.

This chapter has covered and fulfilled the objective of evaluating the impact different cycle design vectors have on the economics of HAT power plant. Moreover, the objective of comparing the annual cost of machinery of a HAT cycle and diesel engine over a give marine route was achieved, both in the frame of the Tier II and Tier III norms of the IMO emission regulations.

Chapter 6

Conclusions

To summarise, this thesis is intended to explore the feasibility of the HAT cycle as prime mover of a commercial marine vessel. The design space of both *reheated* and *non-reheated* layouts of HAT cycle were explored to define the impact of the cycle main design parameters on the thermal efficiency and specific purchase cost of the power plant. The off-design and part-load performance of the *reheated* HAT were analysed to determine the sensitivity of the cycle performance to variations in power settings. Moreover, selected HAT cycle designs powering a Panamax vessel on a Transatlantic route were compared, to analyse the impact of the design vector on the capital and operation cost of the power plant. Finally, a comparison between a *reheated* HAT power plant and both IMO Tier II and Tier III compliant diesel engines was presented.

The exploration of the design space confirmed the high thermal efficiency humid air turbines are able to achieve. The maximum thermal efficiency in the design space of 61.2% and 58.3% were respectively obtained for the *reheated* and *non-reheated* HAT, both as a result of the maximum technology level of all heat exchangers. The SPEC of the *non-reheated* HAT with maximum thermal efficiency is 6.3% lower than for the *reheated* cycle with maximum thermal efficiency.

The main outcomes from the analysis of the trade-off between the high thermal efficiency and the low SPEC, when reducing the effectiveness of the heat exchangers, can be summarised as following:

- The reduction in effectiveness of the recuperator results in both a reduction in the thermal efficiency and overall SPEC of the power plant. For the *reheated* HAT, a 6.87% reduction in SPEC can be achieved with a penalty of 2.28 pp in thermal efficiency when the effectiveness of the recuperator is reduced from 0.9 to 0.8. For the *non-reheated* HAT, the same variation in

effectiveness results in a 11.5% reduction in SPEC and in a 1.17 pp drop in thermal efficiency.

- The reduction in effectiveness within the interval $0.95 > \varepsilon_{AC} > 0.85$ for the after-cooler and $0.95 > \varepsilon_{AC} > 0.75$ for the inter-cooler result in a reduction of the power plant SPEC. For value of effectiveness below this interval the SPEC of the power plant increases.
- The reduction in effectiveness of the economiser and overall pressure ratio of the *reheated* HAT, and increase in the pinch temperature difference of the saturator result in penalties in terms of both thermal efficiency and SPEC of the power plant.

The design of the *reheated* power plant for sea and ambient tropical conditions (298.15 K) achieves a maximum thermal efficiency 0.74 pp lower than the cycle designed for temperate conditions. The SPEC also is reduced by 2.17% as a result of the reduction of the heat exchanger purchase cost.

The analysis of the part-load performance of the *reheated* HAT cycle that features maximum thermal efficiency at design point has demonstrated the operability of the cycle at high thermal efficiency across a wide range of power settings. The thermal efficiency is higher than 60% down to 47% of power load, and for power setting of 25% the cycle can be operated with thermal efficiency of 55.6%. The comparison between the part-load performance curves of selected cycle designs has shown that the reduction in effectiveness of the recuperator results in higher penalties in term of thermal efficiency. At 50% power load the thermal efficiency of cycle A_5^{40} drops by 1.14% from the design value whereas in cycle E_5^{40} it reduces by 2.5%.

The NO_x emissions in the HAT cycle are lower than 0.15 g/kWh when operating between 25% and 100% power load. MARPOL AnnexVI sets the limit at 1.96 g/kWh. The low NO_x emissions are a result of the high humidity in the combustion chambers. The NO_x emission estimation is based on semi-empirical correlations, and due to the lack of experimental data for a 40 MW HAT cycle, the empirical data used in the model is based on the WR-21 Rolls Royce marine gas turbine. However, the low NO_x emissions estimated in this thesis are within the range of emissions observed in experimental campaigns on pilot plant and available in literature [21, 22, 24, 30]. Ultra low sulphur content fuels will be necessary for the HAT cycle to operate within the ECAs due to the limits in SO_x emissions.

Selected *reheated* HAT cycles, from the Pareto front obtained as a result of high thermal efficiency and low SPEC, were compared in terms of annual cost of machinery when powering a Panamax carrier operating on a Transatlantic route.

This analysis has shown that the cycle with the highest thermal efficiency has the lowest AK, 5.5% lower than the AK of the reference cycle with the lowest thermal efficiency (58%) and lowest SPEC (316.7 \$/kW). However, the cycle with the highest thermal efficiency is also the cycle with the longest payback period (3.4 years) and therefore subject to the highest financial risk.

The outcomes of the comparison between a down-scaled 9 MW *reheated* HAT cycle and IMO Tier II and Tier III compliant diesel engines powering a Handymax bulk vessel from Rio de Janeiro to Rotterdam can be summarised as following:

- Within the frame of Tier II norms of the IMO emission regulation, the *reheated* HAT features 5% lower annual cost of machinery within the reference scenario. The payback period for the higher capital expenditure of the HAT is of 7 years. Moreover, the economical viability of the HAT cycle strongly depends on the variation of the MDO and HFO fuel price. An increase higher than 5.7% in MDO fuel price, and constant HFO fuel price, would result in a the payback period to exceed the lifetime of the investment.
- Within the frame of Tier III norms of IMO emission regulations the HAT cycle competitiveness increases as the diesel engine capital and operation expenditure rise, as a result of the curbing of NO_x and SO_x emissions. The annual cost of machinery of the HAT cycle is 38% lower than for the diesel engine with a EGR system and running on IMO sulphur content compliant fuels. The payback period for the higher capital investment is of 0.5 years. The capital investment of the HAT cycle is instead 11.5% lower than for the diesel engine with both EGR and EGC systems and running on 3% sulphur content fuels. The annual cost of machinery of the HAT cycle, in this case, is 11% lower than for the diesel engine. However, the immaturity of the HAT technology, the potential further development of diesel engines and the possible introduction of new fuels in the market all represent significant uncertainties in the comparison of the analysed prime movers.

One of the major drawbacks of the HAT cycle remains certainly its low technology readiness level (TRL) compared to the mature and well established diesel engines. Although, pilot plants of simplified versions of the EvGT and AHAT layout of the cycle have run in Sweden and Japan, no full size with complete cycle layout has yet been tested. This represents certainly a significant source of uncertainty in the results here presented, and further research and development of the HAT technology would be necessary to better determine the possible role of the HAT power plant in the market.

Overall the outcomes of this research are a step forward towards the assessing of the feasibility of *reheated* humid air turbines as prime movers of commercial marine applications.

6.1 Way Forward

Based on the research here presented some possible guidelines for future research on this topic have been identified.

- Study the impact of the reheater and of the consequent cooling flows required by the power turbine stages on the polytropic efficiency achievable by the power turbine. In this research the polytropic efficiency of the turbines was fixed and a sensitivity analysis of the cycle performance to this parameters is presented in appendix B.1. However, further detailed research could increase the confidence in the assumptions and increase the understanding of the impact the reheater could have on the design of the cycle components.
- Further analyse the impact of the OPR of the *reheated* HAT on the thermal efficiency of the cycle. In this research the OPR of the *reheated* HAT was limited to 40:1 to reflect the current technology level available today. However, it could be of interest to evaluate the gains in thermal efficiency the cycle could achieve as the technology required for higher OPR becomes available.
- Investigate the impact of the mass imbalance between the cold and hot section of the gas turbine on the design and cost of the devices. Previous research has focused on the utilisation of a combination aero-derivative gas turbine and industrial turbomachinery—studied under the name of CHAT by Nakhamkin et al. [20]—to overcome the issue. However, to fully exploit the potential of the humid air turbine modifications of the gas turbine design will be necessary.
- Analyse the impact the heat exchangers and the turbomachinery deterioration on the performance of the cycle across a range of power settings. This will improve the understanding of the operation limitations of the cycle and increase the fidelity of operation expenditure estimation for the power plant.
- Identify the optimised Pareto front for the cycle design when trading off the thermal efficiency for lower purchase equipment cost of the power plant. This could be included a robust design process of the power plant that would

include the off-design performance of the cycle and the profile of the mission the cycle is expected to operate. This would most probably require the development of surrogate models for the design of components and design space of the thermodynamic cycle to reduce the computational time.

- Investigate the integration of the HAT cycle with a vessel design. In this thesis a integrated full electric propulsion system (IFEP) configuration was considered. However, the potential benefits or drawbacks of integration of a HAT power plant with the vessel for commercial marine applications remain unexplored. This would certainly increase the understanding of the viability of HAT power plants as prime movers of commercial vessel as a more comprehensive economical analysis would be possible.

Part II

Appendixes

Appendix A

Thermodynamic Models

In this appendix the developed models for the thermodynamic simulations of components or systems of the power cycles are reported.

A.1 Design Point Saturator Model

The saturator, used for evaporating a certain amount of water into compressed dry air, is modelled according to Sanchez de Leon's methodology [18] based on Aramayo-Prudencio's and Young's theory of pressurised saturators [31].

A.1.1 Model Description

The methodology is designed for only design point performance calculations, for reduced required computational time and with a minimum amount of input information from the user to calculate the outlet conditions of the two streams:

- Inlet conditions of the two flows (water & dry-air)
- Pinch temperature difference ΔT_{ps} between the operating and the saturated air enthalpy lines in the saturator.

The following assumptions are made within the model:

- The air-vapour mixture is treated as an ideal mixture of ideal gases [31].
- The air at the outlet of the saturator is completely saturated [42, 43]

From the equations of conservation of mass of the dry air and water, and the equation of conservation energy it is possible to derive two invariants of the system (M^+ , H^+):

$$\frac{m_w}{m_a} - \omega = M^+ \quad (\text{A.1})$$

$$h_a(T_g) - \omega[h_v(T_g) - h_w(T_w)] - M^+ h_w(T_w) = H^+ \quad (\text{A.2})$$

Where m_a is the dry air mass flow, m_w is the water mass flow, $h_a(T_g)$ is the enthalpy of the dry air evaluated at the temperature of the gas T_g , $h_v(T_g)$ is the enthalpy of the vapour evaluated at the temperature of the gas T_g , and $h_w(T_w)$ is the enthalpy of the liquid phase water evaluated at the temperature of the liquid water.

In the assumption of ideal mixture of ideal gas specific humidity is defined as:

$$\omega = \frac{\Phi p_{sat}(T_g) R_a}{p - \Phi p_{sat}(T_g) R_v} \quad (\text{A.3})$$

Where $p_{sat}(T_g)$ is the saturation pressure at the temperature T_g , Φ is the relative humidity, R_a is the specific gas constant for air and R_v is the specific gas constant for vapour.

In order to find a solution to the saturation problem, the above mentioned system of equations needs to be solved for three sections of the saturator:

Station "0", corresponds to the pinch condition between the operating and saturated enthalpy lines of the saturator. The temperature difference between the two above mentioned lines is characteristic of the *available technology level* or *size* of the saturator. The thermodynamic proprieties of the saturated gas (temperature $T_{g,0}$ and specific humidity ω_0) in this section are the unknowns of the system. The relative humidity of the gas on the saturation line is $\Phi = 1$, and $T_{w,0}$ in the operating line is determined as $T_{w,0} = T_{g,0} + \Delta T_{ps}$ once the pinch point temperature difference is defined, therefore the invariant and specific humidity equations can be rewritten as following in agreement with Aramayo-Prudencio and Young [31]:

$$\omega_0 - \frac{p_{sat}(T_{g,0})}{p_1 - \Phi p_{sat}(T_{g,0})} \frac{R_a}{R_v} = 0 \quad (\text{A.4})$$

$$M^+ - \frac{cp_a}{cp_w} - \omega_0 \left(\frac{cp_v - cp_w}{cp_w} + \frac{p_1}{p_1 - p_{sat}(T_{g,0})} \frac{R_v}{cp_w} \left(\frac{h_{fg}(T_{g,0})}{R_v T_{g,0}} \right)^2 \right) = 0 \quad (\text{A.5})$$

$$H^+ - h_a(T_{g,0}) - \omega_0 h_{fg}(T_{g,0}) + \omega_0 [h_w(T_{w,0}) - h_w(T_{g,0})] + M^+ h_w(T_{w,0}) = 0 \quad (\text{A.6})$$

Station "1", corresponds to the bottom section of the saturator —i.e. air inlet/water outlet—, for which the unknowns are the water temperature $T_{w,1}$ and the water mass flow $m_{w,1}$ whereas the gas proprieties are known. Therefore, the invariant equations can be written as follows:

$$M^+ + \omega_1 - \frac{m_{w,1}}{m_a} = 0 \quad (\text{A.7})$$

$$H^+ - h_a(T_{g,1}) - \omega_1 h_{fg}(T_{g,1}) + \omega_1 [h_w(T_{w,1}) - h_w(T_{g,1})] + M^+ h_w(T_{w,1}) = 0 \quad (\text{A.8})$$

Station "2", corresponds to the top section of the saturator —i.e. water inlet/air outlet—, for which the unknowns are the outlet gas conditions, namely: temperature $T_{g,2}$ and specific humidity ω_2 , hence, the gas mass flow. Air in this section is assumed to be saturated therefore:

$$\omega_2 - \frac{p_{sat}(T_{g,2})}{p_1 - \Phi p_{sat}(T_{g,2})} \frac{R_a}{R_v} = 0 \quad (\text{A.9})$$

$$M^+ + \omega_2 - \frac{m_{w,2}}{m_a} = 0 \quad (\text{A.10})$$

$$H^+ - h_a(T_{g,2}) - \omega_2 h_{fg}(T_{g,2}) + \omega_2 [h_w(T_{w,2}) - h_w(T_{g,2})] + M^+ h_w(T_{w,2}) = 0 \quad (\text{A.11})$$

The equations of station 0 (eq. A.4, A.5, A.6) and station 2 (eq. A.9, A.10, A.11) form a system six equations and six unknowns that resolve the pinch conditions $(T_{g,0}, \omega_0)$, the gas outlet conditions $(T_{g,2}, \omega_2)$ and the two invariants M^+ and H^+ . Once the invariants are calculated the equations of station 1 (eq. A.7, A.8) allow the solution for the water outlet conditions.

A.1.2 Model Validation

The design point saturator model has been validated against experimental results of a HAT pilot cycle by Lindquist et al. [54]. The inlet conditions of the water and gas, and the pinch temperature difference in the tower were imposed. The outlet conditions of the gas and water from the experimental data and from the module were therefore used to evaluate the accuracy of thermodynamic model. The maximum discrepancy, between the experimental data and the module results in the outlet proprieties of the two streams is of 1.5% as reported in table A.1. The discrepancy is evaluated as $e = (i_{exp} - i)/i_{exp}$, where i_{exp} and i are respectively the outlet proprieties according to experimental data and the model results.

Table A.1: Validation of the design performance saturator model, with experimental data by Lindquist et al. [54].

Inlet/Input conditions			Outlet conditions			
	Unit	Data [54]		Unit	Model	Data [54] Discrepancy (%)
$m_{g,1}$	(kg/s)	2.17	$m_{g,2}$	(kg/s)	2.57	2.55 0.7
$T_{g,1}$	(K)	346.75	$T_{g,2}$	(K)	389.97	389.15 0.2
p_1	(bar)	7.88	ω_2	(-)	0.1837	— —
$m_{w,2}$	(kg/s)	3.48	$m_{w,1}$	(kg/s)	3.08	3.10 0.6
$T_{w,2}$	(K)	419.35	$T_{w,2}$	(K)	347.52	352.82 1.5
ΔT_{ps}	(K)	10.				

A.2 Off-Design Saturator Model

The model for the off-design performance calculations of the saturator is based on the Aramayo-Prudencio et al.[32] and Lindquist et al. [54] for the heat and mass transfer physics of the system, and is based on Dalili [51] for the calculation of the cross-section of the packed-bed.

A.2.1 Model Description

Some general assumptions are made in the model the off-design performance off the saturator:

- all fluid proprieties vary only with the height z
- constant operating pressure throughout the height of the saturator
—ie. neglecting frictional losses—
- no heat exchange to or from the ambient

The saturator is divided in sections with constant height dz , described in figure A.1. Each section (CV3) is composed by two control volumes for the bulk conditions of the gas (CV2) —ie. air and vapour— and for the bulk conditions of the water (CV1), and an interface region.

In the assumption that potential and kinetic energy variation is negligible and that the vapour in contact with the interface is saturated at the temperature T_f —ie. the difference between the specific enthalpies of the gas and the vapour evaluated at the interface temperature T_f is the specific enthalpy of evaporation evaluated at the same temperature ($h_{vf}(T_f) - h_{wf}(T_f) = h_{fg}(T_f)$)—, it is possible to write the *steady-flow energy equations* as following (full derivation in Aramayo-Prudencio et al.[32]):

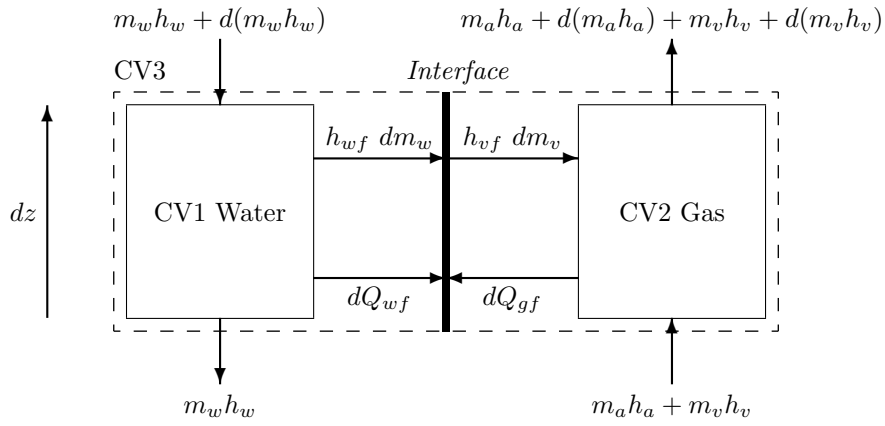


Figure A.1: Differential element of the saturator

$$dQ_{wf} + dQ_{gf} = h_{fg}(T_f) dm_v \quad (\text{A.12})$$

The heat transfer from the water dQ_{wf} and to the gas dQ_{gf} is assumed to be driven only by the temperature gradient, the heat transfer coefficient α_w (between the the bulk liquid and the interface) and α_g (between the the bulk gas and the interface). The following equations define the relationships:

$$dQ_{wf} = \alpha_w(T_w - T_f)A_f dz \quad (\text{A.13})$$

$$dQ_{gf} = \alpha_g(T_g - T_f)A_f dz \quad (\text{A.14})$$

where A_f is the total liquid-gas contact area per unit height.

The vapour mass transfer is assumed to be driven only by the density gradient and the mass transfer coefficient β_g^* (between the interface and the bulk gas defined in eq. A.31) and the relationship is defined as:

$$dm_v = \beta_g^*(\rho_{vf} - \rho_f)A_f dz \quad (\text{A.15})$$

The equations A.12, A.13, A.14 and A.15 create a systems with four unknowns (dQ_{wf} , dQ_{gf} , dm_v and T_f) and can be therefore solved for each section CV3.

The boundary conditions of the system are set by the inlet conditions of the air in the bottom section and the inlet conditions of the water in the top section. The sections are solved from the bottom to the top and an initial guess of the distribution of the temperature of the gas and the water, and of the specific

humidity are necessary. In order to update the temperature profiles between iterations a pseudo transit coefficient is added to the energy equations as described in Aramayo-Prudencio et al.[32]:

$$\begin{aligned} \bar{T}_w^{n+1} = \bar{T}_w^n + \left(\frac{RX_w}{m_a c p_w} \right) [\bar{m}_w \Delta h_w^n \\ - \bar{\alpha}_w (\bar{T}_w^n - \bar{T}_f^n) A_f \Delta z + (\bar{h}_w^n - \bar{h}_{wf}^n) \Delta m_v^n] \end{aligned} \quad (\text{A.16})$$

$$\begin{aligned} \bar{T}_g^{n+1} = \bar{T}_g^n + \left(\frac{RX_g}{m_a c p_g} \right) [m_a \Delta h_a^n - \bar{m}_v \Delta h_v^n \\ - \bar{\alpha}_g (\bar{T}_g^n - \bar{T}_f^n) A_f \Delta z + (\bar{h}_v^n - \bar{h}_{vf}^n) \Delta m_v^n] \end{aligned} \quad (\text{A.17})$$

$(RX_w/m_a c p_w)$ and $(RX_g/m_a c p_a)$ are treated in eq. A.16 and A.17 simply as relaxation factors chosen to facilitate convergence and n is the time step.

Aramayo-Prudencio et al.[32] suggests values between 1×10^{-4} and 8×10^{-4} for RX_w and RX_g , nevertheless, in this model it was found that best convergency is obtained with $RX_w = 2.4 \times 10^{-2}$ and $RX_g = 4.8 \times 10^{-2}$.

The quality of the saturator model results is strongly dependant on accuracy of the heat and mass transfer coefficients calculations.

Water heat transfer coefficient

The water heat transfer coefficient is calculated according to Lindquist et al. [54]:

$$\alpha_w = \frac{\lambda_w}{\delta_w} \quad (\text{A.18})$$

where λ_w is the water conductivity and δ_w is the water film thickness defined as:

$$\delta_w = \left(\frac{3 \mu_w \Gamma}{\rho_w^2 g} \right)^{0.333} \quad (\text{A.19})$$

where μ_w is the viscosity of the water, ρ_w is the density of the water, g is the gravitational acceleration and Γ is the mass flow per unit length defined as:

$$\Gamma = \frac{m_w}{A_{cross} Per} \quad (\text{A.20})$$

where A_{cross} is the cross-sectional area of the packed-bed and Per is a function of the packed-bed geometry:

$$Per = \frac{4 S}{B H} \quad (\text{A.21})$$

where S , H , and B are geometric parameters of the packed-bed as shown in figure A.2.

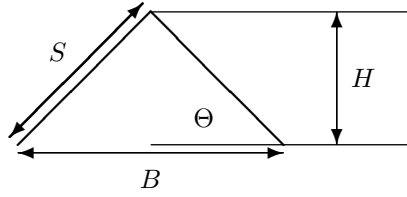


Figure A.2: Geometry of the packed-bed material [54]

Mass transfer coefficient

The mass transfer coefficient is calculated using the procedure, for triangular flow channels in counter-current absorption columns, described in Lindquist et al. [54] and previously determined experimentally by Bravo et al. [123, 124].

The mass transfer coefficient is determined by the Sherwood number (Sh) as:

$$\beta_g = \frac{Sh D_{ab}}{d_{eq}} \quad (\text{A.22})$$

and

$$Sh = 0.00338 Re^{0.8} Sc^{0.333} \quad (\text{A.23})$$

where d_{eq} is the equivalent diameter (eq. A.24), Re is the Reynolds number (eq. A.25), Sc is the Schmidt number (eq. A.28) and D_{ab} is the diffusion coefficient (eq. A.29), and are defined as following:

$$d_{eq} = B H \left(\frac{1}{B + 2 S} + \frac{1}{2 S} \right) \quad (\text{A.24})$$

where B , H and S have been previously defined in figure A.2

$$Re = \frac{\rho_g (u_{g,eff} + u_{w,eff}) d_{eq}}{\mu_g} \quad (\text{A.25})$$

where ρ_g is the density of the gas, $u_{g,eff}$ and $u_{w,eff}$ are respectively the effective velocities of the gas (eq. A.26) and water (eq. A.27) and μ_g is the dynamic viscosity of the gas.

$$u_{g,eff} = \frac{m_g}{\rho_g A_{cross} \varepsilon \sin \Theta} \quad (\text{A.26})$$

where ε is the void fraction (it is an input to the problem) and $\Theta = \arcsin(H/S)$ is the angle specified in figure A.1.

$$u_{w,eff} = \frac{3}{2} \frac{\Gamma}{\rho_w} \left(\frac{\rho_w^2 g}{3 \mu_g \Gamma} \right)^{0.333} \quad (\text{A.27})$$

The Schmidt number Sc is defined as:

$$Sc = \frac{\mu_g}{\rho_g D_{ab}} \quad (\text{A.28})$$

The diffusion coefficient D_{ab} is determined using Fuller-Schettler-Giddings diffusion coefficient with an error estimation of 4.5% [54]

$$D_{ab} = \frac{1.0 \cdot 10^{-7} T_g^{1.75} [(M_g + M_w)/(M_g M_w)]^{1/2}}{p (v_g^{1/3} + v_w^{1/3})} \quad (\text{A.29})$$

where M_g and M_w are the molecular weights of gas and water respectively, p is the absolute total pressure and v_g and v_w are the diffusion volumes for gas and water respectively. The following values of $v_g = 19.7$ and $v_w = 13.1$ are used, taken from Heat Atlas chapter D1 [80].

Mass transfer coefficient *corrections*

The mass transfer coefficient calculated in equation A.22 accounts only for the diffusion contribution. In order to account also for the convective contribution to mass transfer the following correction is made as mentioned in Aramayo-Prudencio et al. [32]:

$$\beta_g^{tot} = \frac{\beta_g}{1 - \omega_{vf}(T_f)} \quad (\text{A.30})$$

Ackerman's Correction is also introduced to include the distortion effects that mass transfer has in concentration, temperature and velocity profiles as suggested in Parente et al. [55].

$$\beta_g^* = \beta_g^{tot} \frac{\Phi_M}{e^{\Phi_M} - 1} \quad (\text{A.31})$$

where

$$\Phi_M = \frac{\frac{dm_v}{M_w}}{\frac{\beta_g^{tot} \rho_g}{M_g}} \quad (\text{A.32})$$

Gas heat transfer coefficient

The heat transfer coefficient of the gas is calculated with the Chilton-Colburn analogy, which relates the heat transfer coefficient with the mass transfer ratio [54] as following:

$$\alpha_g = \rho_g \beta_g^* c_p \left(\frac{Pr}{Sc} \right)^{-2/3} \quad (\text{A.33})$$

Where Sc is the Schmidt number (eq. A.28) and the Pr is the Prandtl number (eq. A.34)

$$Pr = \frac{c_p \mu_g}{\lambda_g} \quad (\text{A.34})$$

A.2.2 Packed-bed geometric parameters & dimensions

In order to be able to simulated off-design performance of the saturator several geometrical values are needed along with the height and cross-sectional area of the packed-bed of the tower.

Packed-bed geometric parameters

The values of the dimensions shown in figure A.2 and the void fraction ε are assumed in agreement with Lindquist et al. [54] and shown in table A.2.

Table A.2: Packed-bed geometric parameters [54].

B	(mm)	25
H	(mm)	7.9
S	(mm)	14.8
ε	(-)	0.95

Cross-sectional Area of the Packed-Bed

The cross-sectional area of the packed-bed determines the pressure losses in the saturator. The area is calculated so that the velocity of the gas is 60% of the flooding velocity and so that the pressure losses are never higher than 300 Pa/m as suggested by Dalili [51].

To find the gas mass flow per unit area for either a given pressure drop or percentage of flooding velocity the generalised pressure drop correlation from Richardson et al. [125] (figure A.3) is used:

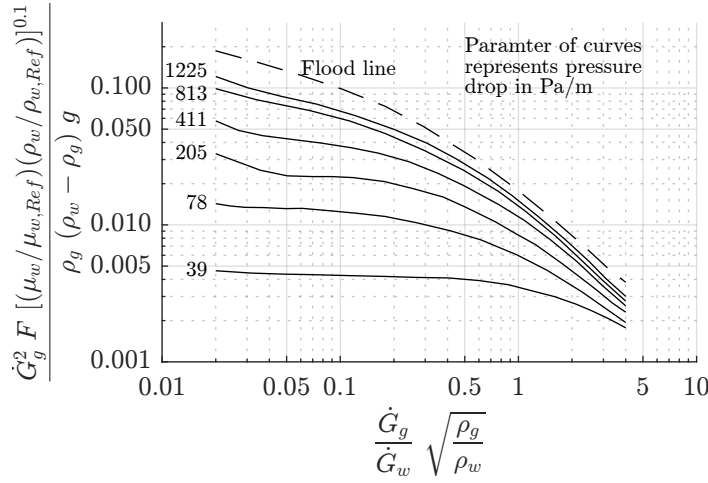


Figure A.3: Generalised pressure drop correlation from Richardson et al. [125].

Where $\dot{G}$ is the mass flow per cross sectional area, F is a packing geometry factor with a constant value of 50 [51], g is the standard gravity, ρ is the density, μ is the dynamic viscosity and the reference proprieties $_{ref}$ are evaluated at the temperature of 298.15 and at a pressure of 1 atm.

The value of $\dot{G}_g / \dot{G}_w \sqrt{\rho_g / \rho_w}$ is calculated for the top and bottom section of the packed-bed. The y axes in figure A.3 for the two section are evaluated both for 60% of the flooding velocity and 300 Pa/m pressure loss. The minimum y value is then used, knowing the required mass flow to calculate the required cross-sectional area.

Height of the Packed-Bed

The height of the packed-bed is calculated iteratively. Knowing the cross-sectional area, the inlet and outlet conditions of both the air and the water, a Broyden method scheme is used to guess the height until outlet conditions of the air and water converge to the values of the design performance.

A.2.3 Model Validation

Table A.3: Validation of the off-design performance saturator model, with experimental data by Lindquist et al. [54].

Inlet/Input conditions			Outlet conditions			
	Unit	Data [54]		Unit	Model	Data [54] Discrepancy (%)
$m_{g,1}$	(kg/s)	2.17	$m_{g,2}$	(kg/s)	2.54	2.55 0.4
$T_{g,1}$	(K)	346.75	$T_{g,2}$	(K)	389.60	389.15 0.14
p_1	(bar)	7.88	ω_2	(-)	0.177	— —
$m_{w,2}$	(kg/s)	3.48	$m_{w,1}$	(kg/s)	3.11	3.10 0.4
$T_{w,2}$	(K)	419.35	$T_{w,2}$	(K)	359.1	352.82 1.7
z	(m)	0.57				
A	(m ²)	0.3848				

The off-design saturator model has also been validated against experimental results of a HAT pilot cycle by Lindquist et al. [54]. The inlet conditions of both the gas and water in the tower were imposed along with the dimensions of the packed-bed, height and cross-sectional area. The experimental and calculated outlet conditions of the gas and water from the tower were used to evaluate the accuracy of the thermodynamic module. The maximum discrepancy in the outlet proprieties of the two streams is of 1.7% as reported in table A.3. The discrepancy is evaluated as $e = (i_{exp} - i)/i_{exp}$, where i_{exp} and i are respectively the outlet proprieties according to experimental data and the model results.

A.3 Heat Exchanger kxA Variation in Off-Design

The performance model of the heat exchangers in off-design relies on the variation of kxA of the heat exchanger. In off-design, when the mass flow in the heat exchanger decreases, the kxA also reduces and vice-versa. This is due to the variation in the heat transfer coefficient. The variation in kxA in part-load is derived as following.

The kxA for a general heat exchanger can be defined as following:

$$\frac{1}{kxA} = \frac{1}{\alpha_c A_c} + R + \frac{1}{\alpha_h A_h} \quad (\text{A.35})$$

Where α_c and α_h are the convection heat transfer coefficient of the cold and hot streams, and A_c and A_h are the heat transfer surface areas for the cold and hot streams. R is a resistance that accounts for the thermal resistance through the tubes or plates and the resistance due to fouling of the heat transfer surfaces.

In this method it is assumed that fouling and the thermal resistance of the metal can be neglected. Therefore the eq. A.35 can be simplified as shown in eq. A.36, thus kxA is only function of the heat transfer areas and the convective heat transfer coefficient of the two flows.

$$\frac{1}{kxA} = \frac{1}{\alpha_c A_c} + \frac{1}{\alpha_h A_h} \quad (\text{A.36})$$

The convection heat transfer coefficient is defined as following:

$$\alpha = \frac{Nu \lambda_{fluid}}{D} \quad (\text{A.37})$$

Where Nu is the Nusselt number, λ_{fluid} is the conductivity of the fluid and D is the characteristic dimension of the flow passage.

In the assumption that the flow in both the hot and cold streams are fully developed —i.e., turbulent flow in the passages—, the Dittus-Boelter formulation for the Nusselt number is adopted.

$$Nu = 0.023 Re^{0.8} Pr^n \quad (\text{A.38})$$

Where Re is the Reynolds number (see eq. A.40), Pr is the Prandtl number (see eq. A.39) and n is either 0.4 for the fluid being heated or 0.3 for the fluid heating.

$$Pr = \frac{cp \mu}{\lambda_{fluid}} \quad (\text{A.39})$$

cp and μ are respectively the specific heat and the dynamic viscosity.

The Reynolds number of the flow in a tube or channel can be defined as following:

$$Re = \frac{4 m}{\pi D \mu} \quad (\text{A.40})$$

Where m is the mass flow and D is the hydraulic diameter of the channel, passage or tube.

In the assumption that in off-design the specific heat, dynamic viscosity and conductivity of the fluids do not vary significantly, the Prandtl number can be considered constant. Within the same assumption the Reynolds number, for a fixed geometry, is only function of the mass flow. It is, therefore, possible to relate the convection heat transfer coefficient to the mass flow as following:

$$\alpha \propto Nu \propto Re^{0.8} \propto m^{0.8} \Rightarrow h \propto m^{0.8} \quad (\text{A.41})$$

Thus it is possible to rewrite eq. A.36 for the off-design (eq. A.42) and design (eq. A.43) conditions as following:

$$\frac{1}{kxA} \propto \frac{1}{m_c^{0.8}} + \frac{1}{m_h^{0.8}} \quad (\text{A.42})$$

$$\frac{1}{kxA_{ref}} \propto \frac{1}{m_{c,ref}^{0.8}} + \frac{1}{m_{h,ref}^{0.8}} \quad (\text{A.43})$$

The combination of eq. A.42 and eq. A.43 results in eq. A.44, where the variation in kxA of the heat exchanger on off-design is function of the mass flow variation of the two streams in the heat exchanger.

$$kxA = kxA_{ref} \left(\frac{m_c^{0.8} m_h^{0.8}}{m_{c,ref}^{0.8} m_{h,ref}^{0.8}} \right) \left(\frac{m_{c,ref}^{0.8} + m_{h,ref}^{0.8}}{m_c^{0.8} + m_h^{0.8}} \right) \quad (\text{A.44})$$

A.4 Turbine Rows Cooling Model

The method for calculating the cooling mass flow required for the cooling of the turbine stages is based on the procedure developed by Harlock et al. [68] and Young et al. [67]. This method is specific for turbine rows that feature:

- Internal convection cooling
- Film cooling
- Thermal barrier coating (TBC)

It calculates the required coolant mass flow for each stator and rotor of each turbine stage separately.

A.4.1 Heat transfer calculations

The dimensionless coolant mass flow rate m_{c+} is calculated with the following expression, fully derived in Young et al. [67], by specifying the values of the film cooling effectiveness ε_f , the internal cooling efficiency η_c and the metal Biot number Bi_m that takes into account the thermal barrier coating.

$$(1 + Bi_m)m_{c+} = \frac{\varepsilon_0 - (1 - \eta_c)\varepsilon_f - \varepsilon_0\varepsilon_f\eta_c}{\eta_c(1 - \varepsilon_0)} \quad (\text{A.45})$$

The blade cooling effectiveness ε_0 is defined by the external metal temperature $T_{m,ext}$, mainstream temperature of the gas path T_{0g} and the mainstream temperature of the coolant T_{0c} and is calculated as following:

$$\varepsilon_0 = \frac{T_{0g} - T_{m,ext}}{T_{0g} - T_{0c}} \quad (A.46)$$

It is then possible to calculate the actual coolant/mainstream mass flow ratio, neglecting upstream cooling flows:

$$\frac{m_c}{m_g} = \frac{A_{surf}}{A_{g*}} \frac{cp_g}{cp_c} St_g m_{c+} \quad (A.47)$$

Where A_{surf} is the total cooled external area, A_{g*} is the gas flow cross-sectional area at the blade throat and St_g a the Stanton number calculated with the mean external heat transfer coefficient and the gas flow proprieties at the inlet of the blade throat.

In absence of a stage geometry, the surface ratio A_{surf}/A_{g*} and the Stanton number St_g are specified, whereas the cp_g and cp_c are calculate from the gas and coolant flow proprieties to account for the water and fuel content of the two streams.

The specified parameters values used for the heat transfer calculations are reported in table A.4 and are in agreement with Harlock et al. [68] and Young et al. [67].

Table A.4: Specified parameters values for heat transfer calculations.

Film cooling effectiveness	ε_f	=	0.4
Internal cooling efficiency	η_c	=	0.7
Metal Biot number	Bi_m	=	0.3
Area ratio	A_{surf}/A_{g*}	=	27.27
Stanton number	St_g	=	0.0015

A.4.2 Stage required coolant mass flow calculations

Due to the nature of the cycle thermodynamic studies, the turbine design is not required.

The approximate number of stages per turbine is calculated by specifying the blade Mach number at mid-span of the blade M_{bl} and the stage loading coefficient ψ .

$$u_{bl} = \sqrt{\gamma R t_{g,1}} M_{bl} \quad (A.48)$$

$$n_{stg} \simeq \frac{h_{0g,in} - h_{0g,out}}{u_{bl}^2 \psi} \quad (\text{A.49})$$

Where u_{bl} is the blade speed at the mid-span section, R is the specific gas constant for the humid air and $h_{0g,in}$ and $h_{0g,out}$ are the total enthalpy of the gas at the inlet and outlet of the turbine.

The calculated number of stages n_{stg} is rounded to the nearest higher natural number and the enthalpy drop per stage is calculated as following, in the assumption that it is constant for each stage of the turbine.

$$\Delta h_{stg} = \frac{h_{0g,in} - h_{0g,out}}{n_{stg}} \quad (\text{A.50})$$

The coolant required mass flow required to cool the stator is calculated with equations A.46, A.45 and A.47, where $T_{0g} = T_{0g,1}$ as in figure A.4.

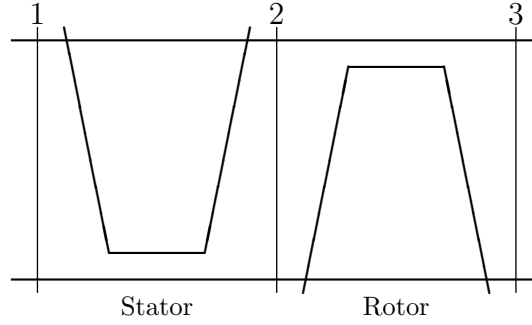


Figure A.4: Turbine stage notation.

In order to calculate the coolant mass flow required the heat transfer model need the *relative* total enthalpy of the rotor.

The total enthalpy at the inlet of the rotor is calculated within the hypothesis of full mixing of the coolant flow of the stator with the main-stream gas flow.

$$h_{0g,2} = \frac{h_{0g,1} + (m_c/m_g)_S h_{c,S}}{1 + (m_c/m_g)_S} \quad (\text{A.51})$$

Whereas the relative total enthalpy of the rotor is calculated assuming zero swirl at the stage outlet and therefore:

$$h_{0g,2}^{rel} = \frac{1}{2\psi} h_{0g,2} + \left(1 - \frac{1}{2\psi}\right) h_{0g,3} \quad (\text{A.52})$$

The coolant required mass flow required to cool the stator is then calculated with equations A.46, A.45 and A.47, where $T_{0g} = T_{0g,2}^{rel}$

In the first stage of the turbine down stream of the combustion chamber or reheater T_{0g} is replaced with an estimated maximum temperature, to account for

the possible hot spots in the gas main-stream.

$$T_{0g}^{max} = T_{0g} + K_{comb} \Delta T_{comb} \quad (\text{A.53})$$

Where ΔT_{comb} is the temperature rise in the combustion chamber or reheater and K_{comb} is the combustion pattern factor that depends on the type of combustion and the position of the blade rows with respect of the outlet of the combustion chamber.

The specified parameters values used for the required stage coolant mass flow calculations are reported in table A.5

Table A.5: Specified parameters values for required stage coolant mass flow calculations.

Core stage turbine loading coefficient	$\psi_{\text{HPT, LPT}}$	=	1.5
Power stage turbine loading coefficient	ψ_{PT}	=	2
Blade mid-span Mach number	M_{bl}	=	0.7
Combustion pattern factor (stator)	K_{comb}	=	0.15
Combustion pattern factor (rotor)	K_{comb}	=	0.05
Metal temperature [K]	T_m	=	1300

The cooling flows affect the temperature and mass flow at the outlet of the turbine and inlet of the down stream turbines. Therefore the coolant flow calculations and the thermodynamic cycle calculations are iterated until convergence is achieved.

A.5 Cooling configurations

The cooling required cooling mass flow is bled from the outlet of the high pressure compressor as shown in figure A.5.

Other positions for bleeding the coolant, such as at the outlet of the aftercooler or at the outlet of the saturator, were initially considered but then discarded due to the low temperatures of the fluid that would generate unsustainable thermal stress in the turbine stages.

The temperature at the outlet of the high pressure compressor is low enough to achieve low blade cooling effectiveness —i.e. low cooling mass flows— but high enough to limit the thermal stress in the turbine blades.

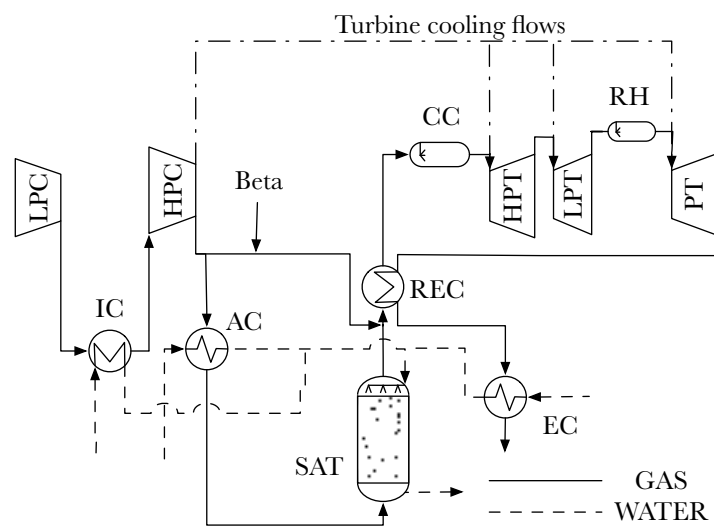


Figure A.5: Layout of the cooling flows for the HAT cycle.

Appendix B

Performance Sensitivity to Design Assumptions

In the exploration of the design space in chapter 3 a number of design variable are fixed according to values that reflect the current component available technology. In this section the impact of the variation of these parameters on the thermodynamic performance of the cycle are shown. The impact of the reduction of the maximum heat capacity ratio on the optimised performance of the cycle is also analysed.

B.1 Impact of Gas Turbine Design Parameters

B.1.1 Impact of the Turbomachinery Polytropic Efficiency

In this section, the *reheated* cycle A_5^{40} (see table 3.5) is used as reference. When the design assumption of interest is varied the remaining design parameters remain unchanged as show in table 3.5 for exception of the air mass flow that is adjusted to achieve a power output of 40 MW.

In section 3.2 it is assumed that the polytropic efficiency of the compressors and core turbines is 90% whereas the polytropic efficiency of the power turbine is 92%. Figure B.1 shows the impact in terms of the power plant thermal efficiency of the variation in polytropic efficiency of the compressors and core turbines from 90% to 86%, and the variation in power turbine polytropic efficiency from 92% to 88%. The variation in thermal efficiency of the power plant due to the reduction in polytropic efficiency of the turbomachinery is linear. The polytropic efficiency of the compressors has the most sever impact on the thermal efficiency. With a reduction from 90% to 86% of the polytropic efficiency of the compressors —i.e.,

both LPC and HPC—the cycle thermal efficiency drops by 2.44%. The reduction in HPT and LPT polytropic efficiency from 90% to 86% results in a 0.8% reduction in thermal efficiency, whereas the reduction in the polytropic efficiency of the power turbine from 92% to 88% results in a 1.6% reduction in thermal efficiency.

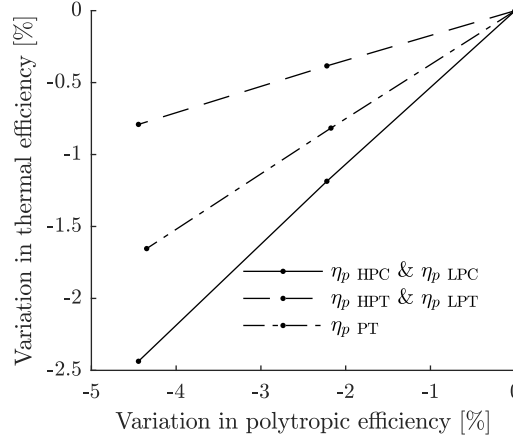


Figure B.1: Impact of turbomachinery polytropic efficiency on the thermal efficiency of the HAT cycle.

B.1.2 Impact of the Turbine Inlet Temperature

In all analysis of the *reheated* HAT cycle it is assumed than the maximum temperature at the outlet of the combustion chambers is 1600 K. Figure B.2 shows the impact of the maximum design TIT of both the main and reheater combustion chamber on the thermal efficiency of the power cycle.

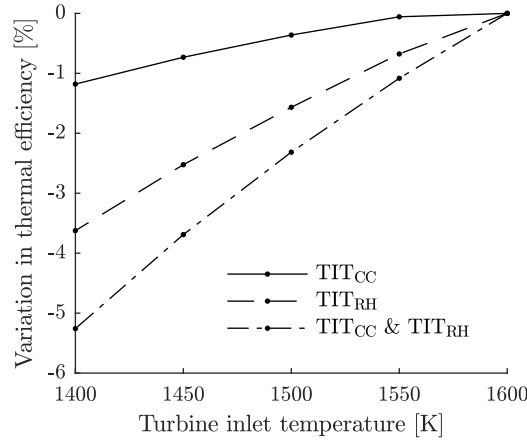


Figure B.2: Impact of design turbine inlet temperature on the thermal efficiency of the HAT cycle.

The reduction in TIT of the main combustion chamber least affects the thermal efficiency of the power plant. The reduction of the maximum design TIT_{CC} from 1600 K to 1400 K and constant TIT_{RH} at 1600 K results in a drop in thermal efficiency by 1.2%, whereas the reduction of the maximum design TIT_{RH} from 1600 K to 1400 K results in a drop in thermal efficiency by 3.6%. The reduction of the maximum outlet temperature of both combustion chambers results in a drop in thermal efficiency by 5.3%.

B.2 Impact of the Heat Capacity Ratio

This section focuses on the impact of the heat capacity ratio on the optimised performance and design variables of the *reheated* HAT. The thermodynamic drivers that promote the variation in performance are also analysed.

B.2.1 Methodology

In the analysis of the design space of the *reheated* HAT cycle in section 3.4 the heat capacity ratio in the inter-cooler, after-cooler and economiser is optimised for maximum thermal efficiency. The minimum and maximum bound for the heat capacity ratio used in the optimisation process are respectively 0.3 and 0.95. The results reported in table 3.5 show the maximum thermal efficiency is achieved with the maximum value of heat capacity ratio of both the inter-cooler and after-cooler.

Figure B.3 shows the variation of number of transfer units (NTU) as function of the effectiveness and the C^* of a counter current heat exchanger. For a constant effectiveness of the heat exchanger, the reduction of the heat capacity ratio results in a drop of NTU and therefore of required heat transfer area. Thus, the reduction of the heat capacity ratio can result in cheaper and more compact design. Moreover, in the water-air heat exchangers with heat capacity ratio of 0.95, the volumetric flow rate of the air is three order of magnitude greater than the volumetric flow rate of the water. This results in significant challenges in the design of the heat exchanger.

The impact of the heat capacity ratio on the performance of the *reheated* HAT A_5^{40} is analysed by reducing the C^* maximum value in the optimisation process described in section 3.4.1. Thus, for each new maximum value of the heat capacity ratio, the design variables of the cycle (see table 3.3) are optimised for maximum thermal efficiency.

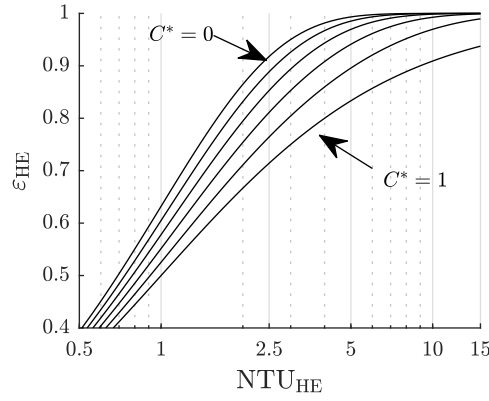


Figure B.3: Number of heat transfer units for a counter current heat exchangers as a function of the effectiveness of the heat exchanger and heat capacity ratio.

B.2.2 Results

Figure B.4a illustrates the impact of the the reduction of maximum heat capacity ratio on the thermal efficiency and humidity of the gas at the inlet of the main combustion chamber. The reduction of the maximum heat capacity ratio of the inter-cooler, after-cooler and economiser from 0.95 to 0.75 results in a drop in the thermal efficiency by 0.94 pp, a reduction in specific power by 4.9% and an increase in β by 2.5% —i.e., higher fraction of air flows though the after-cooler and saturator. When the heat capacity ratio is reduced from 0.75 to 0.55 the thermal efficiency reduces by 4.33 pp, the specific power drops by 17.5% and β increase by 24% .

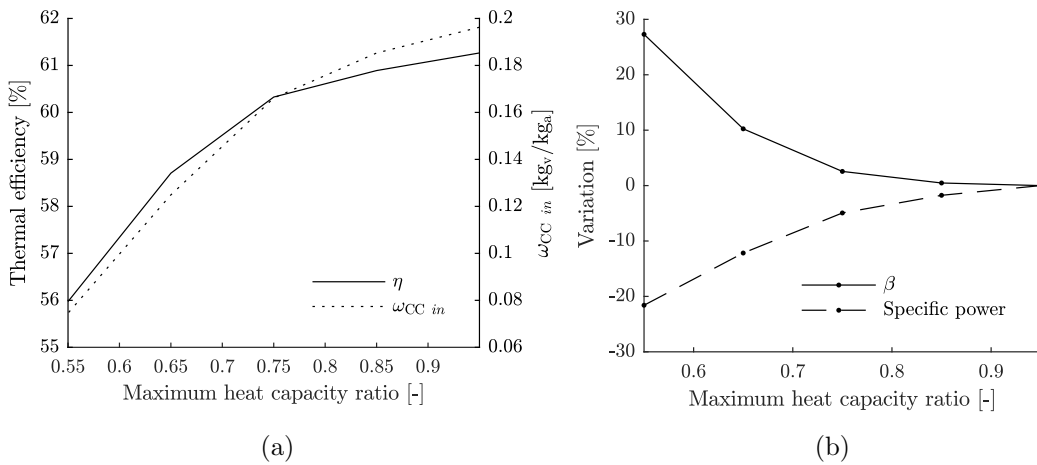


Figure B.4: Impact of the heat capacity on the (a) thermal efficiency and absolute humidity at the combustion chamber inlet, and (b) β and the specific power.

The reduction in thermal efficiency is a consequence of the reduction in humidity in the hot section of the gas turbine. The humidity at the inlet of the combustion chamber drops by 15.4% and 62% respectively when the maximum heat capacity ratio is changed from 0.95 to 0.75 and from 0.75 to 0.55. This different in impact of the maximum heat capacity ratio on the humidity in the hot section of the gas turbine when changed within the two ranges —i.e., $0.95 > C^* > 0.75$ and $0.75 > C^* > 0.55$ — is due to the variation in heat capacity ratio of the economiser. When the maximum heat capacity ratio is reduced from 0.95 to 0.75 the heat capacity of the inter-cooler and after-cooler are reduced by the same amount, whereas the heat capacity in the economiser increases from 0.72 to 0.75. For maximum heat capacity ratios below 0.75 the heat capacity in economiser is also the maximum value allowed.

The reduction in heat capacity in the water-air heat exchangers results in an increase in generation of hot water by the latter. Figure B.5a shows the variation in water mass flow through the inter-cooler, after-cooler, economiser and saturator. The water mass flow in the after-cooler is the most affected, with an increase of 183% when the maximum value of C^* is reduced by 0.95 to 0.55. The water mass flow in the inter-cooler also increases by 120%, whereas the water mass flow in the economiser increases by 40%. Overall the hot water mass flow generated by the heat exchangers and injected in the saturator increases by 84%. However, due to the increase in β and in the reduction in specific power —i.e., higher air inlet mass flow in the gas turbine— the ratio between the water and air mass flow in the saturator increases by only 12%.

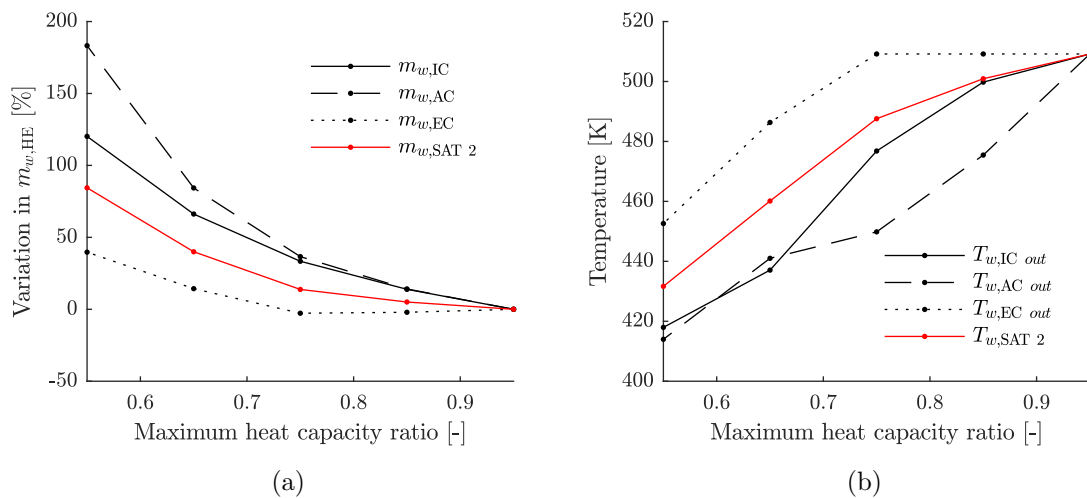


Figure B.5: Impact of the heat capacity on the (a) water mass flow and (b) temperature of the water at the outlet of the inter-cooler, after-cooler and economiser.

Figure B.5b shows the impact of the maximum heat capacity ratio on the temperature of the water at the outlet of the inter-cooler, after-cooler and economiser, and on the water temperature injected in the saturator. When the maximum heat capacity ratio is reduced from 0.95 to 0.75, the temperature of the water at the outlet of the inter-cooler and after-cooler reduces respectively by 6.4% and 11.6%, whereas the temperature of the water at the outlet of the economiser remains constant. Hence, the temperature of the water at the inlet of the saturation drops by 3%. When the maximum heat capacity ratio is decreased below 0.75 to 0.55 the temperature of the water at the outlet of the economiser drops by 11% due to the reduction in C_{EC}^* .

B.2.3 Conclusions

The impact of the variation in the maximum value of the heat capacity ratio of the water-gas heat exchanger on the performance of the *reheated* HAT cycle has been analysed.

Both the thermal efficiency and the specific power of the power plant drop by respectively 8.6% and 21.6% when the maximum value of C^* is reduced from 0.95 to 0.55. Although the mass flow of hot water generated in the heat exchangers and injected in the saturator increases the humidity in the hot section of the gas turbine drops from 0.196 to 0.074, due to the reduction in temperature of the water injected in the saturator.

Appendix C

Journal Publications

The research presented in this thesis has contributed to the publication of the following journal articles:

- Brighenti, G. D., Orts-Gonzalez, P. L., Sanchez-de-Leon, L., and Zachos, P. K., Design Point Performance and Optimization of Humid Air Turbine Power Plants, *Applied Sciences*, vol. 7, Apr. 2017, p. 413.
- Brighenti, G. D., Zachos, P. K., and Orts-Gonzalez, P. L., Part-load performance modelling of a reheated humid air turbine power cycle, *Applied Thermal Engineering*, vol. 138, 2018, pp. 365373.
- Orts-Gonzalez, P. L., Zachos, P. K., and Brighenti, G. D., Techno-economic analysis of a reheated humid air turbine, *Applied Thermal Engineering*, vol. 137, 2018, pp. 545554.

Article

Design Point Performance and Optimization of Humid Air Turbine Power Plants

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Abstract: With the recent drive towards higher thermal efficiencies and lower emission levels in the power generation market, advanced cycle power plants have become an increasingly appealing option. Among these systems, humid air turbines have been previously identified as promising candidates to deliver high efficiency and power output with notably low overall system volume, weight and emissions footprint. This paper investigates the performance of an advanced humid air turbine power cycle and aims to identify the dependencies between key cycle design variables, thermal performance, weight and cost by means of a parametric design optimization approach. Designs of the main heat exchangers are generated, aiming to ascertain the relationship between their technology level and the total weight and acquisition cost of them. The research outcomes show that the recuperator and the intercooler are the two components with the largest influence on the thermal efficiency and the total cost. The total weight of the power system is driven by the technology level of the recuperator and the economizer. Finally, the effectiveness of the aftercooler seems to have the greatest impact in reducing the total acquisition cost of the system with minimum penalty on its thermal efficiency.

Keywords: humid air turbine; power generation; design optimization

1. Introduction

Clean and cost effective power generation is a key factor to respond to the challenges arising nowadays in the power generation market. Thermal efficiency enhancements by advanced cycle power plants significantly impacts pollutant emissions and consequently cost of electricity. Therefore, it can be argued that advanced cycle plants with low environmental footprint increase their competitiveness in a market dominated by increasingly stringent emission regulations. Techno-economic assessments of such technologies should be, therefore, undertaken to identify the economic viability of such systems.

Humid Air Turbine (HAT) or Evaporative Gas Turbine (EvGT) cycles were previously found to offer lower specific investment costs than combined-cycle gas turbines (CCGTs) and notably higher thermal efficiencies, especially in the range of small to medium-sized power plants [1,2]. Moreover, HAT cycles can also offer greater specific power (higher even than the specific power of the ‘dry’ gas turbines), faster start-up times (comparable to the typical start-up times of ‘dry’ gas turbine packages), potential to reduce even further the NO_x emissions due to the presence of high levels of humidity within the combustion chamber of the gas turbine, better part-load performances, and low susceptibility to ambient conditions compared to the CCGTs [3].

Since the introduction of HAT cycles in the late 80s [4], the potential of these systems has been studied extensively. Chiesa et al. [5] performed a comparison of three humid cycles showing that the HAT cycle achieved the highest thermodynamic performance, enhancing the thermal efficiency more

than 10 percentage points (pp) compared to a simple cycle. The enhancement was mainly attributed to the reduction in the discharge losses and the recuperation of the heat rejected in the intercooler within the saturator. The authors also pointed that the thermal efficiency could be further increased if a reheater was placed. Lazzaretto and Segato [6,7] performed a thermodynamic optimization of a HAT cycle where they considered the heat exchanger network as a “black box” to be later optimized. The study established that the intercooling temperature for optimum thermal efficiency was the one minimizing the compression work, and that the pressure ratio does not affect the thermal efficiency but only the specific power. When optimizing the heat exchanger network independently, they discover that the best configuration only included one intercooler, one aftercooler, and one economizer. Ågren and Westermarck analysed the potential of part-flow humidification whereby a percentage of the air flow bypasses the humidifier, showing that, in some cases, it can imply a lower initial investment and even higher cycle thermal efficiencies [8,9]. Moreover, Jonsson and Yan [1] presented a detailed economic analysis proving the advantages of part-flow humidification, as lower specific investment cost for a similar cost of electricity compared to the combined cycle or the fully humidified version. They also proved, that for larger overall pressure ratios, the benefits of the humidification process increase. In addition, Thern et al. [10] tested the idea of including an aftercooler downstream the compressor, proving that the addition of this component boosts the performance of the whole plant by enhancing the amount of heat recovered in the humidifier. Further studies have been carried out on the part load performance of HAT cycles. Wang et al. [11] and Kim et al. [12] concluded that HAT systems are less susceptible to ambient conditions than simple cycle gas turbines. Takahashi et al. [13] showed that humid turbines may demonstrate a better part-load performance than combined cycles. Additionally, optimizations performed by Kavanagh et al. [14] and Moller et al. [15] showed that the variables that drive the cycle’s thermal efficiency are the overall pressure ratio (OPR), the relative pressure ratio, and the recuperator effectiveness. Additionally, Nyberg and Thern [16] studied the influence of each of the heat exchangers in thermal efficiency and specific power of the cycle, proving that the intercooler and the recuperator are the components with the largest impact on the efficiency, and the economizer is the one that drives the most significant increment in the specific work.

Although no HAT power plants are yet in commercial service, a few experimental facilities have been built [17–20] to demonstrate this concept in reduced-scale or even micro gas turbines and identify experimentally the potential of such systems in terms of performance, part-load operation and emissions. Only one full-scale 40 MW power plant has been built aiming to investigate potential commercialisation of such systems on the power generation field [21].

Although simple humid air turbine systems have been thoroughly researched previously, no emphasis was given on the identification of the maximum thermal efficiency potential of such system as highlighted by Chiesa et al. in [5] by means of a reheated humid cycle. As pointed out by Chiesa et al. [5], such a cycle may feature notably better performance relatively to a single burner system that may promote the exploitation of humid cycle power plants in applications where high thermal efficiency and high power-to-plant size ratios are of importance. This paper presents the design and optimization of an advanced humid air turbine reheated power plant and a parametric analysis across a range of key design parameters. The relationship among the key design parameters of the main heat exchangers of the system and its thermal efficiency is established. Finally, design specifications of the key heat exchangers are produced enabling correlations between cycle efficiency and heat exchanger weight and acquisition cost.

2. Methodology

Figure 1 shows the layout of the HAT system studied herein. It comprises a dual-shaft gas generator with a free power turbine. Water absorption into the compressed air takes place in the saturator. The dry air entering the saturator is previously cooled down in the aftercooler, which increases the performance of the saturator tower as it permits to recover larger amounts of low-quality—i.e., low temperature—heat [10]. Once inside the tower, both heat and mass are

transferred from the stream of hot water circulating counter-currently into the air, increasing its temperature and mass flow. Due to the relatively low temperatures of the humid air exiting the saturator, the HAT cycle benefits from a recuperator that recovers the high-quality—i.e., high-temperature waste heat from the exhaust gases before streaming the humid air into the combustion chamber. An additional bypass is also present between the high-pressure compressor exit and the inlet of the recuperator, as suggested by Ågren and Westermark [8,9], so that a certain fraction of dry air can bypass the humidification process and is mixed with the rest of the humid air between the saturator and the recuperator inlet. The amount of air that bypasses the humidifier is represented by the parameter β , which represents the ratio of air flowing through the aftercooler and saturator to the inlet mass flow. Therefore, $\beta = 1$ when all the air goes through the aftercooler and saturator, whereas, for $\beta = 0$, all of the air bypasses them.

In this study, an open loop is used to feed water in the heat exchangers, on the assumption that a water source is available. The evaporation process in the saturator and an efficient droplet eliminator at the gas outlet of the saturator acts as a water-treatment device per se, allowing the use of non-demineralized water without undermining the life of the components.

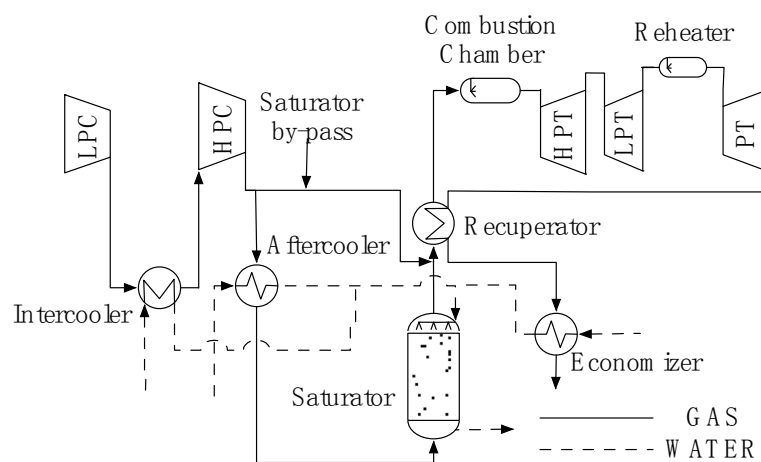


Figure 1. Schematic layout of the Humid Air Turbine.

The thermal performance of the system is resolved using Turbomatch© [22,23], an in-house 0D performance software developed at Cranfield University, with external modules for the heat exchangers and saturator performance, and blade cooling mass flow requirements calculations. For the design point performance prediction, the power output is set constant at 40 MW with no mechanical losses considered. In terms of fuel, Marine Diesel chemical properties were used for combustion calculations [24]. With regards to turbomachinery performance, a 90% polytropic efficiency is assumed for the compressors and the core turbines, whereas 92% is assumed for the power turbine, to reflect the current technology levels of these components. The turbine inlet temperature (TIT) at the outlet of the main combustion chamber as well as at the outlet of the reheater was set constant at 1600 K. Turbine cooling flows are calculated using the methodology proposed by Young and Wilcock [25]. This method accounts for the cooling requirement of the disks, and the stationary and rotating blades when cooled with internal convection and film cooling technology. Typical values for maximum metal temperature ($T_{m,max} = 1300$ K), film cooling effectiveness ($\epsilon_f = 40\%$) and internal flow cooling efficiency ($\eta_c = 70\%$) are used as discussed by Horlock et al. in [26]. Turbine cooling flows are extracted downstream of the high pressure compressor and upstream of the aftercooler.

The outlet conditions from all gas-water heat exchangers, namely the intercooler, the aftercooler and economizer, are resolved by imposing the effectiveness, which is viewed as the heat exchanger's technology level, ($\epsilon = Q/Q_{max}$) and the heat capacity ratio ($C^* = C_{min}/C_{max}$) between the two streams, assuming that $C_w = C_{max}$.

Water condensation in the air stream is possible within the intercooler and aftercooler due to the high pressure and low temperature of the gas at the outlet of the heat exchangers ($T_{g, out} < T_{g, dew}$). In the economizer, condensation may also occur due to high humidity levels in the exhaust gas of the recuperator and the low temperature required at the economizer's exit. The latent heat of condensation of the water in the humid air varies the heat capacity of the gas C_g during condensation and generates a pinch point between the two streams within the heat exchanger (Figure 2). Therefore, this process is considered when simulating the performance of the heat exchangers.

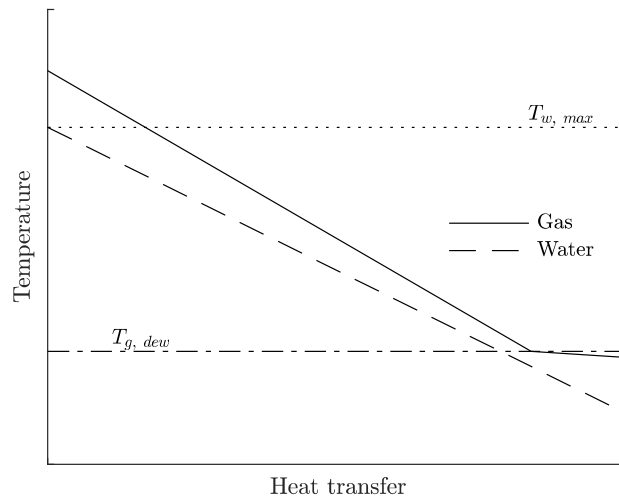


Figure 2. Heat transfer process in the Economizer.

Liquid water is injected into the saturator to facilitate the humidification process. The temperature of the water at the outlet of the heat exchangers is, therefore, limited by the water saturation temperature. A safety coefficient of 0.9 is introduced into the performance calculations to avoid any steaming in the saturator as follows:

$$T_{w, out} \leq T_{w, max} = T_{sat}(0.9 \times p_{SAT}), \quad (1)$$

where $T_{w, out}$ is the water temperature at the outlet of the heat exchangers, T_{sat} is the saturation temperature and p_{SAT} is the pressure in the saturator. When the water temperature at the outlet of the heat exchangers exceeds the limit temperature $T_{w, max}$, a new value of C^* is calculated, overwriting the imposed value. In the recuperator, the inlet properties of both streams are defined; therefore, the outlet conditions of the two streams are calculated from the prescribed recuperator effectiveness. Total pressure losses of the heat exchanger are fixed at a target value of 5% in both sides of all heat exchangers as recommended previously by Kavanagh et al. [14] and Nyberg et al. [16].

The conditions at the outlet of the saturator are calculated by imposing the pinch point temperature difference between the operating line of the water and the saturation line. The saturator is modelled according to the approach discussed by Sanchez de Leon [27] and originally developed by Aramayo-Prudencio [28]. The detailed description of the thermodynamic model of the saturator and its validation against experimental data is provided in Appendix A.

The impact of the key heat exchanger design parameters on the cycle thermal efficiency, plant weight and cost is assessed via a parametric study. The heat exchanger technology level is represented by their effectiveness. In terms of saturator design, the pinch temperature difference between the operating and the saturated air enthalpy lines [28] is used as a design parameter, as it represents the driver for the heat and mass transfer process within the saturator.

Cycle analysis has shown that the overall pressure ratio (OPR) that maximizes the thermal efficiency for the imposed TIT of 1600 K is beyond the technology today available for stationary or

aero gas turbines ($OPR > 50$). Therefore, two values of OPR are investigated, namely 30 and 40, as representatives of a standard and an advanced compression system technologies currently available in the market.

The design envelopes of the above-mentioned variables are summarized in Table 1 and represent the range across which the parametric campaign was conducted for a total of 486 cycle designs.

Table 1. Range of parameters across which the parametric study was conducted.

Parameter		Units	Range
Intercooler, aftercooler, economizer effectiveness	$\varepsilon = Q/Q_{max}$	(-)	[0.75, 0.8, 0.95]
Recuperator effectiveness	$\varepsilon = Q/Q_{max}$	(-)	[0.8, 0.85, 0.9]
Saturator pinch temperature difference	ΔT_{ps}	(K)	[5, 10, 15]
Overall pressure ratio	OPR	(-)	[30, 40]

For each of the 486 cycles calculated, the rest of the design variables (heat capacity ratio of the water-air heat exchangers, relative pressure ratio, and saturator bypass) are optimized to maximize the thermal efficiency. This optimization is performed by means of the single objective genetic algorithm in MATLAB® (R2016a, MathWorks, Natick, MA, USA) [29]. The upper and lower bounds of these variables are reflected in Table 2. Therefore, the thermodynamic “design vector” comprises the effectiveness of the heat exchangers, the OPR and the humidifier pinch point as the free design variables, and the heat capacity ratios of the water-air heat exchangers, the relative pressure ratio and the saturator bypass as the optimized design variables.

Table 2. Heat exchanger design parameters and range specification.

Design Variable		Range
Intercooler, Aftercooler, Economizer heat capacity ratio	$C^* = C_{min}/C_{max}$	[0.6–0.95]
Relative pressure ratio	$relPR = \log_{OPR} PR_{LPC}$	[0.25–0.75]
Saturator bypass	β	[0–1]

Detailed designs of the heat exchangers and the saturator tower are produced to estimate the weight and acquisition of these components. The geometries selected for the heat exchangers are: counter-current plate-fin for the intercooler and the aftercooler, multi-pass cross-flow plate-fin for the recuperator, and finned-tube for the economizer. Lastly, for the saturator, a structured packing-bed tower is selected. The models employed for the detailed design and optimization of these components and the correlations used to estimate the acquisition are summarized in Appendix B.

3. Results and Discussion

A number of cycles have been selected to describe the effect of the “design vector” on the performance, cost and size of the power plant. The heat exchanger “design vectors” of the selected cycles ($A \rightarrow G$) are shown in Table 3 and Figure 3. When referring to the cycles in the text, the overall pressure ratio is defined by the superscript L^{OPR} and the pinch temperature difference in the saturator by the subscript $L_{\Delta T_{ps}}$.

Table 3. Cycle nomenclature definition.

Parameter	A	B	C	D	E	F	G
ε_{IC} (-)	0.95	0.85	0.75	0.95	0.95	0.75	0.95
ε_{AC} (-)	0.95	0.85	0.75	0.75	0.95	0.95	0.95
ε_{EC} (-)	0.95	0.85	0.75	0.95	0.75	0.95	0.95
ε_{REC} (-)	0.90	0.85	0.8	0.90	0.90	0.90	0.80

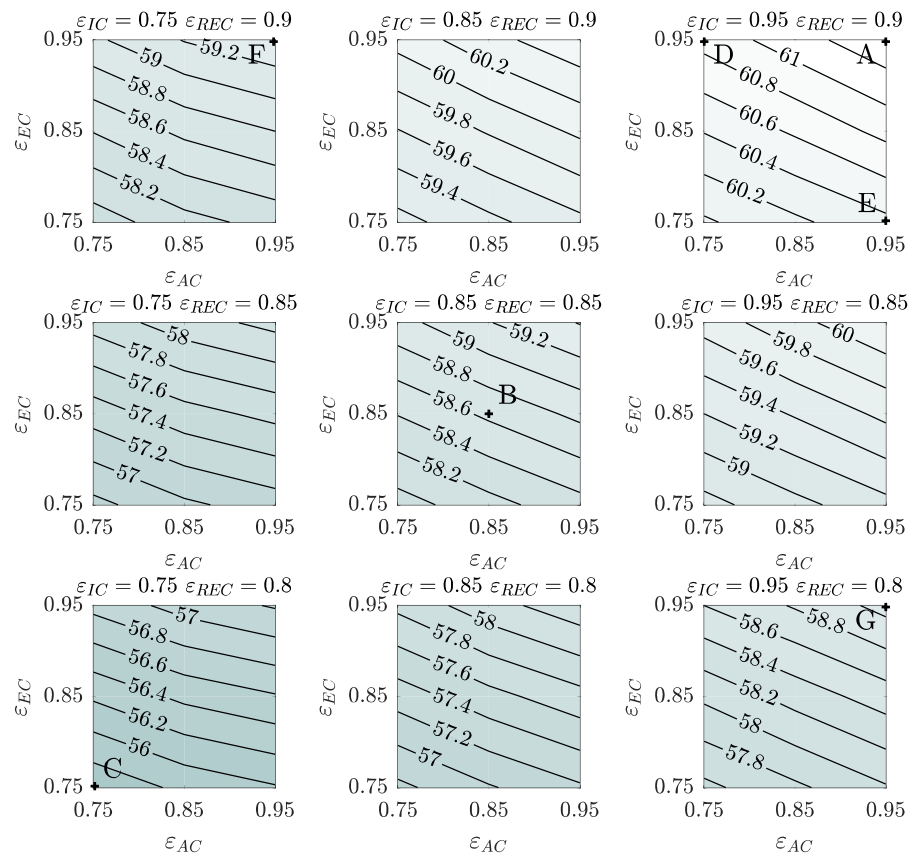


Figure 3. Cycle thermal efficiency across the design space as a function of the heat exchanger effectiveness— $OPR = 40$ & $\Delta T_{ps} = 5$ K.

The outcomes of the parametric cycle performance studies are analyzed using cycle A_5^{40} as a reference, which represents the system with maximum thermal efficiency, maximum effectiveness of all heat exchangers, maximum OPR and minimum pinch temperature difference in the humidifier. The design vector of the baseline cycle as well as its performance metrics are shown in Table 4. All values in subsequent plots are normalized against the cycle with maximum thermal efficiency A_5^{40} .

Table 4. Design variables values of the baseline cycle A_5^{40} with maximum thermal efficiency.

Optimized Design Variable		Units	Value
Thermal efficiency	η_{HAT}	(%)	61.4
Overall pressure ratio	OPR	(-)	40
Relative pressure ratio	$relPR$	(-)	0.53
Intercooler effectiveness	ε_{IC}	(-)	0.95
Aftercooler effectiveness	ε_{AC}	(-)	0.95
Economizer effectiveness	ε_{EC}	(-)	0.95
Saturator pinch temperature difference	ΔT_{ps}	(K)	5
Recuperator effectiveness	ε_{REC}	(-)	0.9
Intercooler heat capacity ratio	C_{IC}^*	(-)	0.95
Aftercooler heat capacity ratio	C_{AC}^*	(-)	0.95
Economizer heat capacity ratio	C_{EC}^*	(-)	0.82
Saturator bypass	β	(-)	0.43
GT inlet gas mass flow	$\dot{m}_{g,GT inlet}$	(kg/s)	40.6

3.1. Impact of Heat Exchanger Effectiveness on Cycle Performance

The impact of heat exchanger effectiveness on the cycle thermal efficiency is shown in Figure 3, for constant OPR of 40 and pinch temperature difference in the saturator of 5 K. The x -axis and y -axis of the contour plots are, respectively, the aftercooler and economizer effectiveness, and the contour lines show the estimated cycle thermal efficiency. Moving vertically and horizontally through the contour plots, the effectiveness of the recuperator and intercooler vary respectively. Therefore, the top-right point of the top-right contour corresponds to the cycle A_5^{40} shown in Table 4.

In all contours with intercooler effectiveness of 0.75 (left column of contours in Figure 3), there is a change in the variation rate of the thermal efficiency when the ε_{AC} falls below 0.85. This is due to the notably high value of aftercooler heat capacity ratio $C_{AC}^* = 0.95$ for aftercooler effectiveness of 0.95 and 0.85, which it decreases to $C_{AC}^* = 0.64$ when the effectiveness of the aftercooler is 0.75, in order for the outlet water temperature to remain within the limits imposed to avoid steaming in the saturator ($T_{w,out} = T_{sat}(0.9 \times p_{SAT})$).

The dependency between the thermal efficiency and the heat exchanger effectiveness remains linear when changing the effectiveness of all heat exchangers together. The comparison between cycles A_5^{40} and G_5^{40} highlights the impact of the recuperator effectiveness and results in a reduction of 2.28 pp in cycle thermal efficiency when recuperator effectiveness becomes 0.8 from 0.9. The impact of intercooler effectiveness is shown by comparing cycle A_5^{40} against F_5^{40} . A reduction in ε_{IC} from 0.95 to 0.75 causes a reduction of 2 pp in the cycle thermal efficiency. The impact of the aftercooler and economizer is shown by the comparison of the cycle A_5^{40} with the cycle D_5^{40} , and of A_5^{40} with the cycle E_5^{40} , with, respectively, a reduction of 0.48 pp and 1 pp on the thermal efficiency. The combined effect of the reduction in effectiveness of all the heat exchangers is shown by the comparison of the cycle A_5^{40} with the cycle B_5^{40} and C_5^{40} with, respectively, a drop of 2.7 pp and 5.5 pp in thermal efficiency.

Figure 4 shows the effect of the heat exchanger effectiveness on the absolute humidity at the inlet of the combustion chamber for constant values of OPR (40) and saturator pinch temperature difference (5 K). The maximum absolute humidity in the design space is 24.6%, which ensures stable combustion in a modern combustion chamber as discussed by Göke in [30].

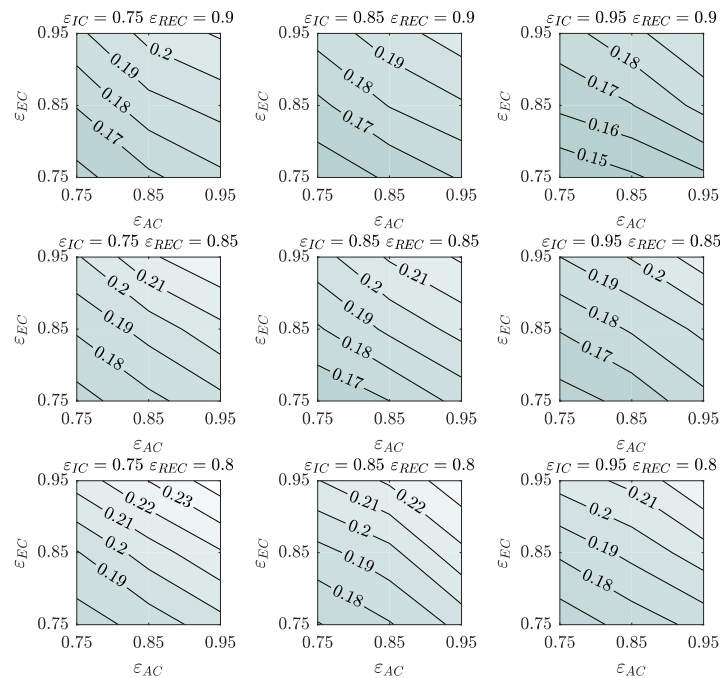


Figure 4. Absolute humidity at the inlet of the combustion chamber across the design space as a function of the heat exchanger effectiveness— $OPR = 40$ & $\Delta T_{ps} = 5$ K.

The impact of heat exchanger effectiveness on the optimized saturator bypass ratio, on the consequent absolute humidity (ω) at the inlet of the combustion chamber and on cycle thermal efficiency is illustrated in Figure 5 for constant $OPR = 40$ and $\Delta T_{ps} = 5$ K.

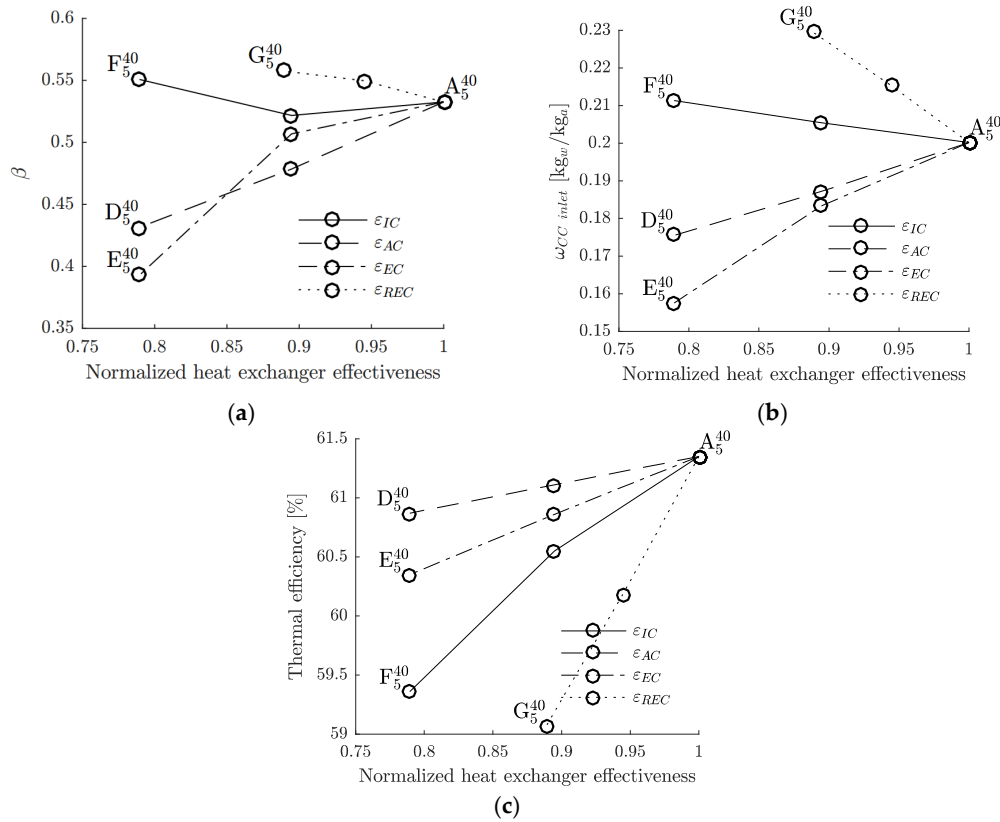


Figure 5. (a) normalized heat exchanger effectiveness vs. mass flow fraction through the saturator and aftercooler; (b) normalized heat exchanger effectiveness vs. absolute humidity at the inlet of the combustion chamber; and (c) normalized heat exchanger effectiveness vs. cycle thermal efficiency— $OPR = 40$ & $\Delta T_{ps} = 5$ K.

The reduction in effectiveness of the aftercooler and economizer has an impact on the amount of heat and mass transfer that occurs in the saturator and, therefore, on the thermal efficiency of the cycle. For an aftercooler effectiveness of 0.75 ($A_5^{40} \rightarrow D_5^{40}$ —see Figure 5), the water mass flow in the saturator is reduced by 0.5% while a 1.2% reduction is observed in its temperature ($C_{AC}^* = 0.95$ across the range). The temperature of the gas at the inlet of the saturator increases by 13.8%, reducing the temperature leap in the saturator and, therefore, the heat and mass transfer that occurs within the device. Hence, the fraction of gas that bypasses the saturator (β) decreases by 19% (Figure 5a) and the humidity at the inlet of the combustion chamber also decreases from 20% to 18% (Figure 5b). The reduction in heat recuperated at low temperature in the cycle, due to the aftercooler, affects the thermal efficiency of the cycle, with a 0.48 pp reduction between cycles A_5^{40} and D_5^{40} (Figure 5c).

When the economizer effectiveness reduces from 0.95 to 0.75 ($A_5^{40} \rightarrow E_5^{40}$ —see Figure 5), the mass flow of the water heated in the heat exchanger falls by 22%, whereas the water outlet temperature falls only by 0.3%—i.e., C_{EC}^* increases. As for the aftercooler, the saturator bypass β decreases by 26% as shown in (Figure 5a) and the humidity at the inlet of the combustion chamber decreases from 20% to 15.7% (Figure 5b). This reduction in heat recuperated in the cycle at low temperature, as for the aftercooler, entails a decrease in thermal efficiency of 1 pp between cycles A_5^{40} and E_5^{40} (Figure 5c).

The effectiveness of the intercooler affects the amount of waste heat that is recuperated both at high and low temperatures within the cycle. When ϵ_{IC} becomes 0.75 from 0.95 ($A_5^{40} \rightarrow F_5^{40}$ —see

Figure 5), the total amount of water injected in the saturator increases by 10% due to the increase of heat available in the gas at the inlet of the economizer—i.e., the temperature at the outlet of the recuperator on the hot side increases due to the increase in temperature of the gas at the inlet of the cold side. Therefore, the mass flow ratio of gas that bypasses the saturator decreases by 2% (Figure 5a) and the humidity at the inlet of the combustion chamber increases by 5% (Figure 5b). This highlights the importance of the intercooling process, since it not only reduces the compression work required but also the heat extracted from the gas between the two compressors is partly recovered in the saturator and not rejected entirely as in most common intercooled gas turbines. Thus, the reduction of the effectiveness of the intercooler from 0.95 to 0.75 entails a reduction of 2 pp in the cycle thermal efficiency ($A_5^{40} \rightarrow F_5^{40}$ —see Figure 5c).

The recuperator is the heat exchanger whose effectiveness change has the most severe effects on the thermal efficiency of the cycle, with a reduction of 2.28 pp in thermal efficiency when it is reduced to 0.8 from 0.9 ($A_5^{40} \rightarrow G_5^{40}$ —see Figure 5), due to the reduction of high quality heat recovery—i.e., heat recovery at high temperature—within the cycle. Nevertheless, this reduction in recuperator effectiveness results in higher enthalpy gas at the inlet of the economizer and entails 22% higher amount of heat transfer in the latter heat exchanger. Therefore, the water mass flow injected into the saturator increases by 10%, the mass flow ratio of gas that bypasses the saturator decreases—i.e., β increases by 4.7% as shown in Figure 5a—and the humidity at the inlet of the combustion chamber increases from 20% to 23% (Figure 5b).

Figure 6 shows the impact of the heat exchanger effectiveness on the heat transfer coefficient times the heat transfer area ($k \times A$) of each heat exchanger for constant OPR (40) and saturator pinch temperature difference (5 K). For similar inlet conditions, the variation in $k \times A$ of a heat exchanger is a useful parameter to evaluate the impact of the of the cycle “design vector” on the variation in size independently from the design choices in sizing the heat exchanger. Nonetheless, $k \times A$ should not be used to evaluate the differences in size between intercooler, aftercooler, economizer and recuperator due to the different inlet conditions of the two flows—i.e., different heat transfer coefficients. The values of $k \times A$ for each heat exchanger are normalized using the values of the reference cycle A_5^{40} which is reported in Table 5.

Table 5. $k \times A$ of the heat exchangers for the reference cycle A_5^{40} (see also Table 4).

Component	Units	$k \times A$ for Reference Cycle A_5^{40}
Intercooler	(kW/K)	545.5
Aftercooler	(kW/K)	267.7
Economizer	(kW/K)	389.1
Recuperator	(kW/K)	379.4

The heat exchangers that are mostly affected by the reduction in effectiveness are the intercooler and aftercooler, with a reduction of $k \times A$ respectively of 79% ($A_5^{40} \rightarrow F_5^{40}$ —see Figure 6a) and of 82% ($A_5^{40} \rightarrow E_5^{40}$ —see Figure 6b) in relative terms when reducing the effectiveness of the latter from 0.95 to 0.75.

The variation of effectiveness of a heat exchanger, however, has a knock-on effect on the $k \times A$ of the remaining heat exchangers. The aftercooler $k \times A$ is strongly affected by the variation of the intercooler and economizer effectiveness with a 38.5% and 22% reduction when reducing ε_{IC} and ε_{EC} become 0.75 from 0.95 ($A_5^{40} \rightarrow F_5^{40}$ in Figure 6a and $A_5^{40} \rightarrow D_5^{40}$ Figure 6c). In addition, the $k \times A$ of the economizer is strongly affected by the variation of the effectiveness of the recuperator ($A_5^{40} \rightarrow G_5^{40}$) with a reduction of 21%. No knock-on effect on the recuperator and intercooler $k \times A$ was found by effectiveness changes of the rest of the heat exchangers.

The increase of the saturator pinch temperature difference ΔT_{ps} from 5 K to 15 K ($A_5^{40} \rightarrow A_{15}^{40}$) has a linear effect on the thermal efficiency and on the absolute humidity of the air at the inlet of the combustion chamber (Figure 7a), with a reduction in thermal efficiency of 0.64 pp from the baseline

cycle. The $k \times A$ of all heat exchangers are not significantly affected by the pinch temperature difference in the saturator (Figure 7b) with a maximum increase of 4% in intercooler $k \times A$ for $\Delta T_{ps} = 15$ K. The increases in $k \times A$ (cycle $A_5^{40} \rightarrow A_{15}^{40}$) are attributed to the 3.2% higher gas mass flow requirement to produce the prescribed power output of 40 MW triggered by the lower water content in the hot section of the power plant.

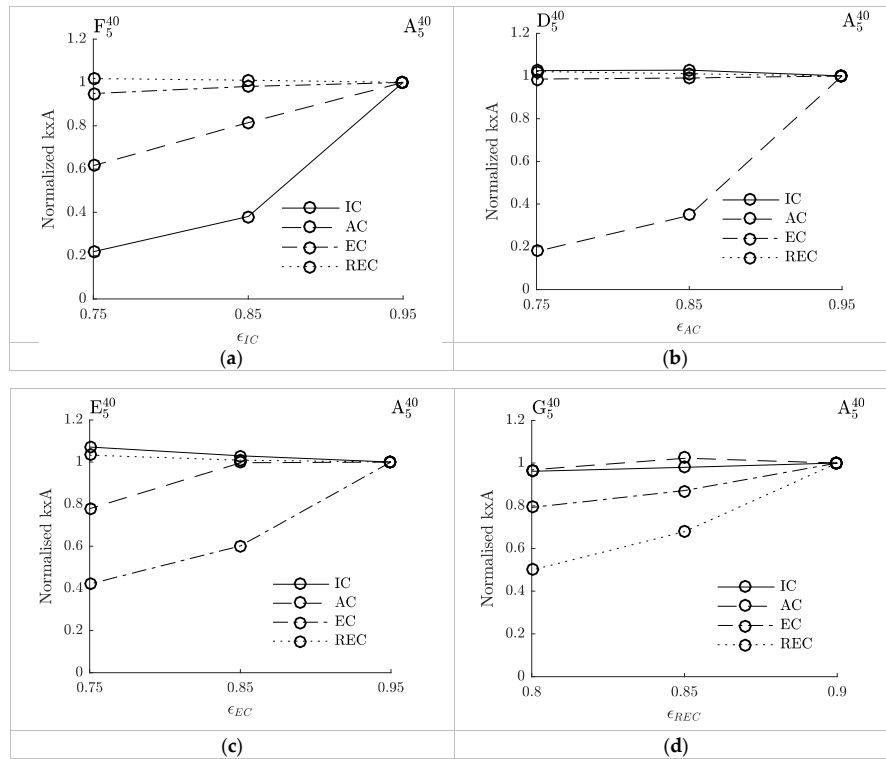


Figure 6. (a) normalized inverse total resistance, $k \times A$ of each heat exchanger vs. intercooler effectiveness ($A_5^{40} \rightarrow F_5^{40}$); (b) normalized $k \times A$ of each heat exchanger vs. aftercooler effectiveness ($A_5^{40} \rightarrow E_5^{40}$); (c) normalized $k \times A$ of each heat exchanger vs. economizer effectiveness ($A_5^{40} \rightarrow D_5^{40}$); and (d) normalized $k \times A$ of each heat exchanger vs. recuperator effectiveness ($A_5^{40} \rightarrow G_5^{40}$)—OPR = 40 & $\Delta T_{ps} = 5$ K.

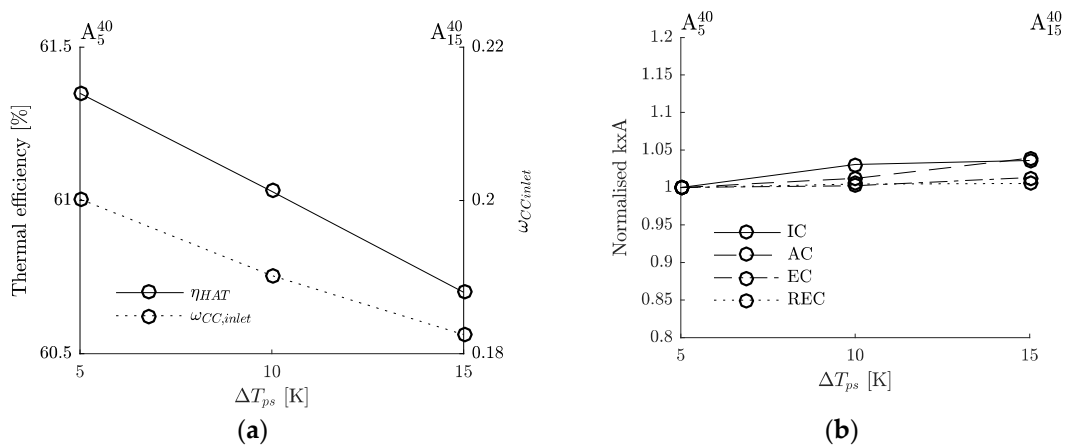


Figure 7. (a) saturator pinch temperature difference vs. cycle thermal efficiency and absolute humidity at the inlet of the combustion chamber ($A_5^{40} \rightarrow A_{15}^{40}$); (b) saturator pinch temperature difference vs. normalized $k \times A$ of each heat exchanger ($A_5^{40} \rightarrow A_{15}^{40}$)—OPR = 40 & $\epsilon_{IC} = 0.95$ & $\epsilon_{AC} = 0.95$ & $\epsilon_{EC} = 0.95$ & $\epsilon_{REC} = 0.9$.

3.2. Impact of Overall Pressure Ratio

The parametric design point performance analysis was conducted at an OPR of 30 that represents a typical technology level for current power generation applications. An optimum cycle configuration with thermal efficiency of 60.7% is achieved when reducing the overall pressure ratio from 40 to 30 ($A_5^{40} \rightarrow A_5^{30}$)—i.e., a reduction of 0.7 pp with regards to the baseline cycle with $OPR = 40$. The inlet gas mass flow to deliver the prescribed power output of 40 MW increases by 9.8%. This is achieved with the maximum specified effectiveness of all heat exchanger and minimum pinch temperature difference in the humidifier as shown in Table 6. The humidity levels at the inlet of the main burner is reduced by 10.2% while saturator bypass β increases by 4 pp when reducing the overall pressure ratio from 40 to 30. This is attributed to the lower water temperature at the outlet of the intercooler, aftercooler, economizer and the reduction in temperature of the gas at the outlet of the high pressure compressor.

Table 6. Design variables values of the baseline cycle A_5^{30} with maximum thermal efficiency.

Optimized Design Variable		Units	Value
Thermal efficiency	η_{HAT}	(%)	60.7
Overall pressure ratio	OPR	(-)	30
Relative pressure ratio	$relPR$	(-)	0.53
Intercooler effectiveness	ε_{IC}	(-)	0.95
Aftercooler effectiveness	ε_{AC}	(-)	0.95
Economizer effectiveness	ε_{EC}	(-)	0.95
Saturator pinch temperature difference	ΔT_{ps}	(K)	5
Recuperator effectiveness	ε_{REC}	(-)	0.9
Intercooler heat capacity ratio	C_{IC}^*	(-)	0.95
Aftercooler heat capacity ratio	C_{AC}^*	(-)	0.95
Economizer heat capacity ratio	C_{EC}^*	(-)	0.70
Saturator bypass	β	(-)	0.47
GT inlet gas mass flow	$\dot{m}_{g,GT\ inlet}$	(kg/s)	44.86

Although the reduction in overall pressure ratio implies cheaper and more compact turbomachinery, the decrease in specific power results in a 11.2% and 10.3% increase in $k \times A$ (Figure 8) of the intercooler and recuperator, respectively, whereas the aftercooler and economizer $k \times A$ decrease by approximately 4%. This yields more expensive designs of the intercooler and recuperator with an increase in acquisition cost index of, respectively, 25.6% and 7% (Figure 9).

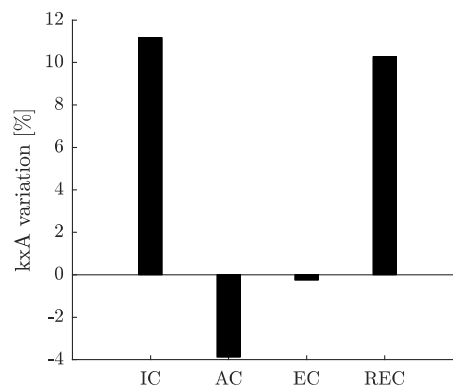


Figure 8. Variation of the $k \times A$ of the heat exchangers when varying the overall pressure ratio from 40 to 30 ($A_5^{40} \rightarrow A_5^{30}$)— $\Delta T_{ps} = 5$ K, $\varepsilon_{IC} = 0.95$ & $\varepsilon_{AC} = 0.95$ & $\varepsilon_{EC} = 0.95$ & $\varepsilon_{REC} = 0.9$.

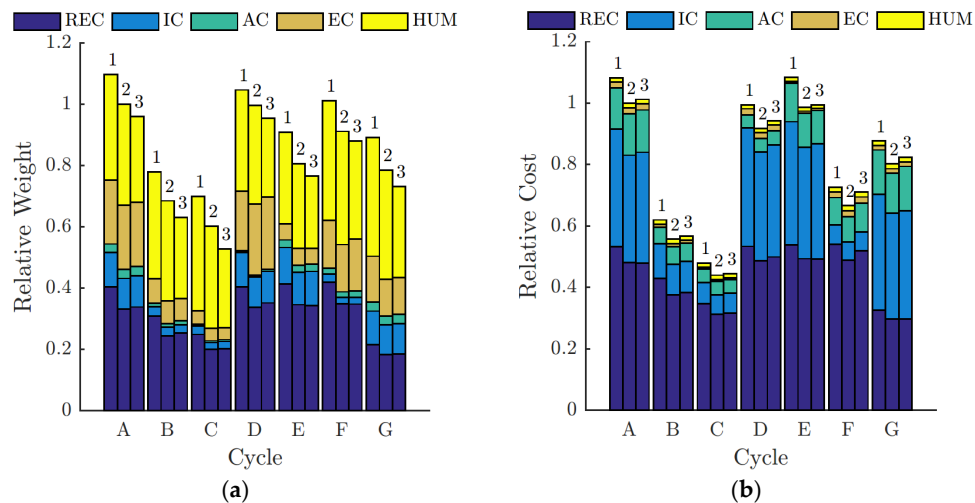


Figure 9. (a) break down of heat exchanger total relative weight; and (b) break down of heat exchanger total relative cost. (Bar 1) $OPR = 30$ & $\Delta T_{ps} = 5$ K; (Bar 2) $OPR = 40$ & $\Delta T_{ps} = 5$ K; (Bar 3) $OPR = 40$ & $\Delta T_{ps} = 15$ K.

3.3. Heat Exchanger Size and Cost Analysis

The breakdown of the total weight and cost for the heat exchanger configurations are highlighted in Figure 9. Cycles A, B, and C (see Table 3) show the variation when the effectiveness of all heat exchangers vary together. On the other hand, cycles D, E, F, and G show the effect of reducing the technology level of one heat exchanger at a time. The reference cycle (A_5^{40}) has been used to normalize the weight and cost of the components. The total reference weight of all heat exchangers is estimated up to 47.76 tones, whereas the reference cost estimate is 4.2 m USD. These estimates are broadly in accordance with the cost estimates previously presented by Jonsson and Yan [1] for a power plant of 55.1 MW (3.8 m USD updated to 2015). It is important to highlight that the thermal efficiency achieved by the reference cycle (61.4%) is 11.9 pp higher than the one reported by Jonsson and Yan in [1], which partially justifies the discrepancy in the heat exchanger acquisition cost estimate.

For cycle A_5^{40} , the recuperator is the heaviest component—33.12% of the total weight—followed by the humidifier, the economizer, the intercooler, and finally the aftercooler—with relative contributions to the overall power plant weight of 33.01%, 20.96%, 9.98%, 2.94%, respectively. Although the economizer and the humidifier are among the heaviest components, their contribution to the overall cost of the system was found to be approximately 3% (Figure 9). In terms of heat exchanger cost, the recuperator is the most expensive heat exchanger—48.11% of the total cost, followed by the intercooler, the aftercooler, the economizer, and finally the saturator—with relative contributions to the overall cycle cost of 34.95%, 13.43%, 1.99%, and 1.52%, respectively.

Reducing the OPR from 40 to 30 increases the total weight across a range between 5% and 10% depending upon the cycle technology level (Figure 9a), and the total cost by 8% in all cases (Figure 9b). The observed increment in heat exchanger weight is primarily driven by the recuperator and the economizer due to the excess of available heat downstream the power turbine that needs to be managed by these two components whose size is further increased by the higher flow rate requirement of this low OPR cycle. As a result, the size increase of the recuperator primarily drives the increase in the estimated acquisition cost of the system. It is worth noting that when the OPR becomes 30 from 40, the humidifier is 7% less expensive, driven by the need for thinner shell walls. However, an increase in weight is observed as the required material for packing increases by 12% compared to the baseline case to facilitate the larger enthalpy flux. Finally, although the reduction of the OPR to 30 increases the weight and the cost of the components, it is associated with a reduction in the overall thermal efficiency of the system by 0.7 pp.

When the ΔT_{ps} becomes 15 K from 5 K, the size, and, consequently, the weight and cost of all the heat exchangers increases due to the demand of a higher mass flow to achieve the required power output. The saturator is the only component whose weight and cost reduce as it becomes shorter. Except for A_{15}^{40} , which increases its total weight compared with A_5^{40} , the reduction in the weight of the humidifier outbalances the increment in the weight of the heat exchangers, but not the increment in the cost. When the pinch point temperature is increased to 15 K, the total acquisition cost of the system marginally increases by around 3% while the thermal efficiency falls by approximately 0.45 pp.

The comparison of cycles A_5^{40} , B_5^{40} , and C_5^{40} (Figure 9), where the effectiveness of all the heat exchangers is reduced at the same time, shows the exponential relationship between the technology level of the components, and their weight and cost. When the effectiveness of the heat exchangers increases from 0.75 to 0.85 ($C_5^{40} \rightarrow B_5^{40}$), the total weight increases by 14% and the total cost by 28%. For a further effectiveness increase to 0.95 ($B_5^{40} \rightarrow A_5^{40}$), the weight rises by 46% and the cost by 80% with respect to the baseline cycle. On the other hand, the thermal efficiency shows a quasi-linear correlation with the cycle design parameters. From C_5^{40} to B_5^{40} , the efficiency rises by 2.8 pp, and from B_5^{40} to A_5^{40} , an increase of 2.7 pp is observed (Figure 3).

Considering each heat exchanger separately, between cycles A_5^{40} and C_5^{40} , the intercooler is the component that undergoes the largest variation in weight, manifesting a reduction of 77%. The economizer shows a reduction of 65% with regards to its reference cost, followed by the aftercooler with a reduction of 60%, and, finally, the recuperator with a reduction of 40%. On the other hand, the intercooler is the component that suffers the largest reduction in cost—82% of its reference cost, followed by the economizer, the aftercooler, and the recuperator—70%, 67%, and 35%, respectively. This demonstrates that the variations in the total weight of the heat exchangers and the variations in cost are linearly related, as the changes in cost are directly correlated with the variations in weight of each component.

Moreover, when comparing cycles D_5^{40} , E_5^{40} , F_5^{40} , and G_5^{40} with A_5^{40} , it can be seen that the reduction of the technology level of a heat exchanger has a knock-on effect on the remaining heat exchangers of the system (see Figure 9). The aftercooler is the heat exchanger that experiences the largest reduction in weight and cost—88% and 87%, respectively. In addition, both the weight and cost of the aftercooler are cut by 39% when the effectiveness of the intercooler is reduced from 0.95 to 0.75 ($A_5^{40} \rightarrow F_5^{40}$). However, both the weight and cost of the aftercooler are cut by 39% and 17%, respectively, when the effectiveness of the economizer is reduced from 0.95 to 0.75 ($A_5^{40} \rightarrow D_5^{40}$).

The weight and cost of the economizer is reduced by 43% and 14%, respectively, when the recuperator effectiveness is changed from 0.9 to 0.8 ($A_5^{40} \rightarrow G_5^{40}$). Hence, the design of the power plant must account for the impact of the technology level of one of the components on the design, cost, and weight of the remaining heat exchangers.

The overall system estimated acquisition cost and weight is shown in Figure 10 normalized against the baseline cycle A_5^{40} metrics. In terms of total weight, the change produced by the economizer and the recuperator—20% and 22% of the reference weight, respectively—is an order of magnitude higher than the changes produced by the intercooler, or the aftercooler—9%, and 1%, respectively. On the other hand, the changes produced by the intercooler, the recuperator, and the aftercooler—34%, 20%, and 8% of the reference cost, respectively—are one order of magnitude higher than the ones the economizer can create which is around 2%. In addition, from the contours illustrated in Figures 3 and 10b, it can be demonstrated that the most efficient way to reduce the total cost of the heat exchangers is to reduce the effectiveness of the aftercooler. The ratio of the percentage variation in cost to the variation in thermal efficiency $\Delta c(\%) / \Delta \eta(\%)$ is 17—i.e., the cost drops by 17% per percentage point drop of thermal efficiency, whereas, when reducing the effectiveness of the intercooler, recuperator and economizer, these ratios become 16, 9 and 1, respectively.

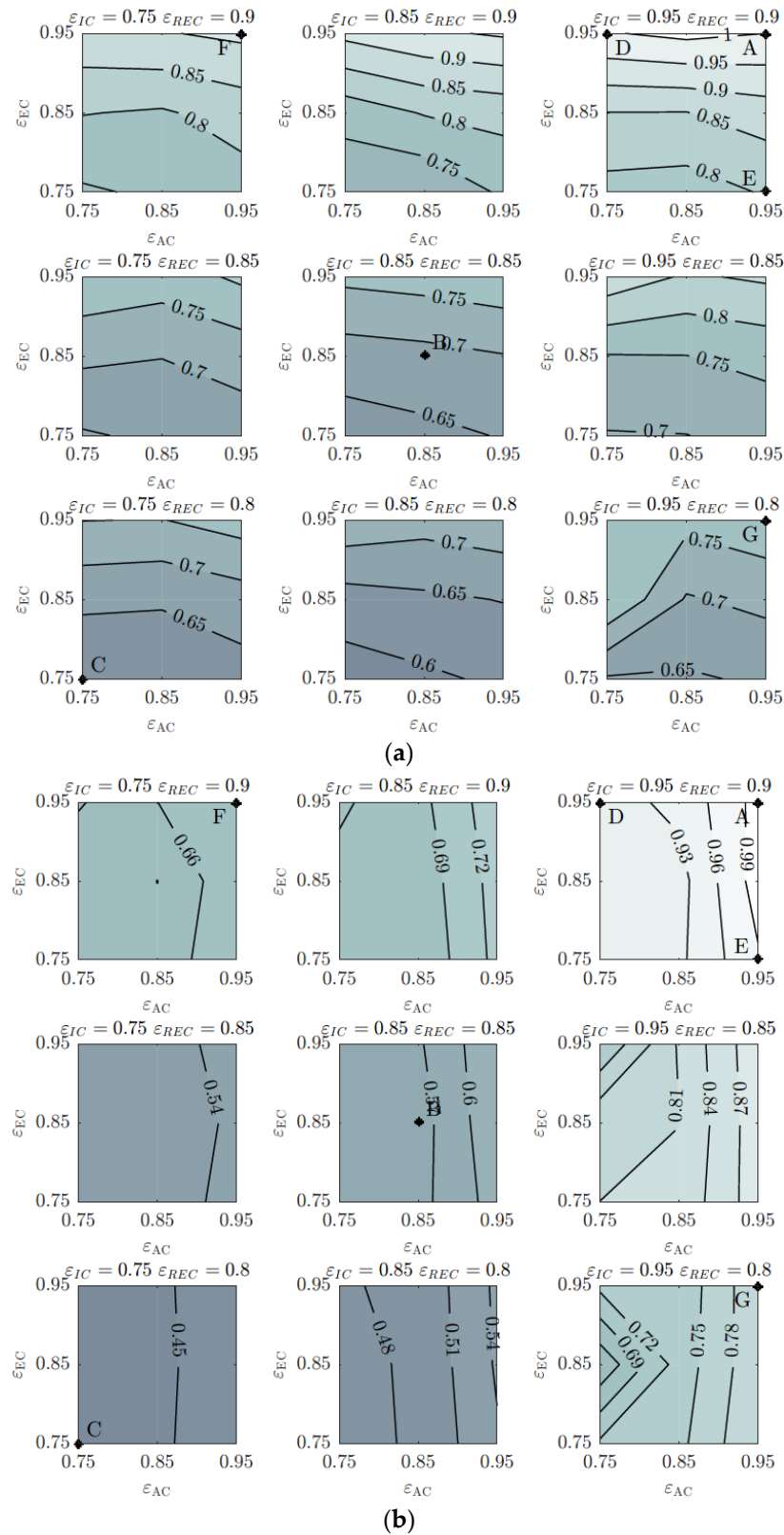


Figure 10. (a) heat exchanger total normalized weight and (b) total normalized acquisition cost index across the range of heat exchanger effectiveness for $OPR = 40$ & $\Delta T_{ps} = 5$ K.

4. Conclusions

The parametric study of a humid air turbine system conducted in the current work has shown the high efficiency potential of this cycle across the design space, with a maximum thermal efficiency of 61.4%, achieved with maximum technology level of the heat exchangers and saturator, and maximum overall pressure ratio of at the gas generator. The absolute humidity at the inlet of the combustion chamber across the design space is within the limits for flame stability, never exceeding 0.25 kg_v/kg_a.

The final selection of the configuration of the power plant will depend on the levelized cost of the power production, which would require performing a full economic analysis. Depending on the market where the power plant is intended, an investment analysis should be conducted in order for the most appropriate candidate power system to be identified.

The parameter that was found to primarily drive the thermal efficiency is the intercooler and recuperator effectiveness. For a change in intercooler effectiveness from 0.95 to 0.75, the thermal efficiency falls by 2 pp, whereas, for the recuperator, a change from 0.9 to 0.8 causes a reduction in thermal efficiency of 2.28 pp. Cycle efficiency is not significantly affected by aftercooler and economizer effectiveness.

The parametric studies also showed that the technology level of each heat exchanger affects the thermal efficiency, total cost, and total weight of the power system differently: the recuperator was found to have the greatest impact on thermal efficiency, whereas the estimated acquisition cost is found to be mainly driven by the intercooler, while the total weight is mostly affected by the economizer and recuperator.

Overall pressure ratio was found to have great impact over the cycle metrics; when the OPR becomes 30 from 40, the total cost of the heat exchangers increases by 10%, total weight by 5% while thermal efficiency is penalized by 0.67 pp. On the other hand, an increase of the saturator's pinch temperature difference ΔT_{ps} to 15 K penalizes the thermal efficiency by 0.65 pp, while the full system's acquisition cost index was found to increase by 3%.

Overall, a typical design space of a complex humid air turbine power plant was systematically explored and the dependencies between key cycle parameters and component design including weight and acquisition cost were identified. The impact of heat exchanger effectiveness on cycle metrics was demonstrated, revealing their influence on thermal efficiency, cost and weight that was not previously known. The outcomes of this research constitute a step forward in understanding the performance of a complex power system across its design space and appreciating the potential of such systems for applications where high efficiency combined with low overall plant volume and weight are of importance.

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Author Contributions: L.S.-d.-L. developed and implemented the heat exchanger and cycle design methodologies, G.D.B. developed the cycle thermal design and optimization platform, performed the simulations and analyzed the cycle performance data, P.L.O.-G. developed the heat exchanger detailed design and optimization framework, conducted the simulations and analyzed the design outcomes. P.K.Z. put together the manuscript.

Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

Symbols

C	kJ/K	Heat capacity
c	$\text{\$}$	Cost
C^*	$(-)$	Heat capacity ratio
C_i	$(-)$	Weighting coefficient
d_t	m	Saturator packing bed diameter
F_i	$(-)$	Optimization fitness
h	kJ/kg	Specific enthalpy
h_t	m	Height of the packing bed of the saturator
H^+	MJ/kg	Enthalpy invariant
$k \times A$	kW/K	Overall heat transfer coefficient $\times$ Heat transfer area (Inverse of total resistance)
M^+	$(-)$	Flow invariant
p	MPa	Pressure
P	m	Perimeter
Q	MW	Heat transfer
SP	$(-)$	Selection parameter
T	K	Temperature
ΔT_{ps}	$(-)$	Saturator pinch temperature difference

Abbreviations

AC	Aftercooler
CC	Combustion chamber
CCGTs	Combined cycle gas turbines
EC	Economizer
EvGT	Evaporative gas turbine
GT	Gas turbine
HAT	Humid air turbine
IC	Intercooler
OPR	Overall pressure ratio
pp	Percentage point
REC	Recuperator
relPR	Relative pressure ratio
SAT	Saturator
TIT	Turbine inlet temperature

Subscripts

0	Pinch conditions
1	Bottom of the saturator
2	Top of the saturator
a	Dry air
dew	Dew
g	Gas
sat	Saturation
v	Vapor
w	Water

Greek Symbols

α	Weighting coefficient
β	Saturator bypass
ε	Effectiveness
ϕ	Relative humidity
ω	Absolute humidity

Appendix A. Humidifier Design

The methodology, discussed by Sanchez de Leon [27] and originally developed by Aramayo-Prudencio [28], defines the design point saturator performance using the inlet pressure and temperature of the two streams (water and dry air) at the inlet and the pinch temperature difference ΔT_{ps} . The pinch point temperature difference is defined as the minimum temperature difference between the operating line of the water and the saturated air enthalpy line across the saturator [28]. For this modelling process, the air–vapor mixture is treated as an ideal mixture of ideal gases, while the air at the outlet of the saturator is assumed to be fully saturated [9]. From the conservation of the dry air and water mass as well as conservation of energy two invariants of the system, (M^+, H^+) can be defined as:

$$\frac{m_w}{m_a} - \omega = M^+, \quad (A1)$$

$$h_a(T_g) + \omega[h_v(T_g) - h_w(T_w)] - M^+h_w(T_w) = H^+. \quad (A2)$$

Under the assumption of ideal mixture of ideal gas, absolute humidity is defined as:

$$\omega = \frac{\Phi p_{sat}(T_g)}{p - \Phi p_{sat}(T_g)} \frac{R_a}{R_v}. \quad (A3)$$

For the calculation of the saturated gas thermodynamic properties, the above-defined system of equations needs to be solved for three sections along the saturator as follows:

Station “0” corresponds to the pinch condition between the operating and saturated enthalpy lines of the saturator. The temperature difference between the two abovementioned lines is characteristic of the available technology level, which corresponds to the height of the saturation tower. The thermodynamic properties of the saturated gas (temperature T_0 and absolute humidity ω_0) in this section are the unknowns of the system. The relative humidity of the gas on the saturation line is $\Phi = 1$, and $T_{w,0}$ in the operating line is determined as $T_{w,0} = T_0 + \Delta T_{ps}$ once the pinch point temperature difference is defined; therefore, the invariant and absolute humidity equations can be rewritten as the following in agreement with [28] as follows:

$$\omega_0 - \frac{p_{sat}(T_0)}{p_1 - \Phi p_{sat}(T_0)} \frac{R_a}{R_v} = 0, \quad (A4)$$

$$M^+ - \frac{cp_a}{cp_w} - \omega_0 \left(\frac{cp_v - cp_w}{cp_w} + \frac{p_1}{p_1 - p_{sat}(T_0)} \frac{R_v}{cp_w} \left(\frac{h_{fg}(T_0)}{R_v T_0} \right)^2 \right) = 0, \quad (A5)$$

$$H^+ - h_a(T_0) - \omega_0 h_{fg}(T_0) + \omega_0 [h_w(T_0 + \Delta T_{ps}) - h_w(T_0)] + M^+ h_w(T_0 + \Delta T_{ps}) = 0. \quad (A6)$$

Station “1” corresponds to the bottom section of the saturator—i.e., air inlet/water outlet, for which the unknowns are the water temperature $T_{w,1}$ and the water mass flow $m_{w,1}$ whereas the gas properties are known. Therefore, the invariant equations can be written as follows:

$$M^+ + \omega_1 - \frac{m_{w,1}}{m_a} = 0, \quad (A7)$$

$$H^+ - h_a(T_{g,1}) - \omega_1 h_{fg}(T_{g,1}) + \omega_1 [h_w(T_{w,1}) - h_w(T_{g,1})] + M^+ h_w(T_{w,1}) = 0. \quad (A8)$$

Station “2” corresponds to the top section of the saturator—i.e., water inlet/air outlet—for which the unknowns are the outlet gas conditions, namely temperature $T_{g,2}$ and absolute humidity ω_2 , hence the gas mass flow. Air in this section is assumed to be saturated. The system of equations can be written as follows:

$$\omega_2 - \frac{p_{\text{sat}}(T_{g,2})}{p_1 - \Phi p_{\text{sat}}(T_{g,2})} \frac{R_a}{R_v} = 0, \quad (\text{A9})$$

$$M^+ + \omega_2 - \frac{m_{w,2}}{m_a} = 0, \quad (\text{A10})$$

$$H^+ - h_a(T_{g,2}) - \omega_2 h_{fg}(T_{g,2}) + \omega_2 [h_w(T_{w,2}) - h_w(T_{g,2})] + M^+ h_w(T_{w,2}) = 0. \quad (\text{A11})$$

The equations of station “0” (Equations (A4)–(A6)) and station “2” (Equations (A9)–(A11)) form a system of six equations and six unknowns that resolve the pinch conditions (T_0, ω_0), the gas outlet conditions ($T_{g,2}, \omega_2$), and the two invariants M^+ and H^+ . Once the invariants are calculated, the equations of station “1” (Equations (A7) and (A8)) allow the solution for the water outlet conditions.

This model has been validated against experimental results of a HAT pilot cycle obtained by Lindquist et al. [17] and the maximum error in the outlet proprieties of the two streams is 1.5% as reported in Table A1.

Table A1. Validation of the saturator model against experimental data from Lindquist et al. [17].

Inlet/Input Conditions			Outlet Conditions				
Parameter	Units	Data	Parameter	Units	Model	Experimental Data	Discrepancy (%)
$m_{g,1}$	(kg/s)	2.17	$m_{g,2}$	(kg/s)	2.5685	2.55	0.7
$T_{g,1}$	(K)	346.75	$T_{g,2}$	(K)	389.97	389.15	0.2
p_1	(bar)	7.88	ω_2	(-)	0.1837	-	-
$m_{w,2}$	(kg/s)	3.48	$m_{w,1}$	(kg/s)	3.0815	3.10	0.6
$T_{w,2}$	(K)	419.35	$T_{w,1}$	(K)	347.52	352.85	1.5
ΔT_{ps}	(K)	10.	-	-	-	-	-

Appendix B. Heat Exchanger Design

A counter-current plate-fin architecture was chosen for the intercooler and aftercooler design. For the airside, a serrated fin arrangement was selected based on its heat transfer and mechanical properties as discussed in [31]. On the waterside, stainless steel plain fins were employed to minimize pumping work, decrease the risk of fouling or passage blocking and avoid potential corrosion problems. The effectiveness-Number of Transfer Units (ϵ -NTU) method was followed to estimate the heat transfer area as described by Kays and London [32]. The number of transfer units was obtained by means of the correlations provided by the Engineering Society Data Unit (ESDU) in [33]. To obtain the main heat exchanger dimensions, the heat transfer coefficient of the airside was estimated using the expressions provided by Yang and Li [34]. For the waterside, the equations suggested in [35] were employed to evaluate the heat transfer coefficient for turbulent flow, and the equations in [36] were utilized for the laminar flow.

The recuperator is an air-air heat exchanger and, as such, a serrated-fin configuration was considered for both sides. In addition, the recuperator was arranged in a counter-current multi-pass cross-flow configuration. Due to the presence of the reheater, the maximum temperatures that the recuperator has to be able to stand can go up to 900 °C. Previous research by McDonald [37], Maziasz [38], and Min [39] has stated that Inconel 625 presents excellent creep resistance properties. Therefore, the passes with a temperature higher than 800 °C were assumed to be made of Inconel 625, whereas the rest of the heat exchanger material was assumed to be stainless steel. For the calculation of the heat transfer area, the ϵ -NTU method was followed [32]. The number of transfer units was obtained by means of the ESDU correlations [33] and the heat transfer coefficient was estimated with the equations purposed by Yang and Li [34].

In the economizer, up to 25% percent of the moisture in the air condenses during design point operation. Hence, to avoid issues associated with passage blocking due to the presence of contaminated air and water inside its passages, a mixture of finned-tube and plain-tube configuration was selected. The first part of the heat exchanger (dry part), where no condensation takes place, was designed using a finned-tube configuration. The later part (wet part), where the condensation begins to occur, was designed in plain-tube to ease the flow of the condensing moisture. The tubes are arranged in a counter-current multi-pass cross-flow configuration, with the air flowing over the tubes and the water through them in an effort to minimize passage blocking and fouling and ease maintenance procedures. The ϵ -NTU method [32] was followed to calculate the number of transfer units. Then, to calculate the heat transfer coefficient of the dry airside flow, the correlations given in the ESDU items [40] were utilized. However, in the wet part, the heat and mass transfer process was simulated following the model developed in [41]. The heat transfer coefficient of the water was evaluated as dictated in the ESDU items [35].

An index of the acquisition cost of the four heat exchangers specified above was estimated using heat transfer area based correlations. For the plate-fin geometries, the expressions proposed by Traverso and Massardo [42] were used. These correlations were originally derived for heat exchangers made of aluminum alloys; consequently, the material correction factors suggested by Towler and Sinnott [43] were applied to consider the different alloys employed. For the economizer, the correlation presented by Casarosa [44] was utilized for estimating its acquisition price. All cost estimates were updated to 2015 by means of the Chemical Engineering Plant Cost Index (CEPCI) [45].

For a prescribed thermodynamic “design vector” of the system, several different heat exchangers designs can be identified. A genetic algorithm based multi-objective optimization approach was applied to identify the individual that features the best compromise between cost and compactness [29]. The compactness was evaluated by means of the outlet perimeter, defined as the sum of the three main dimensions—i.e., height, length, and width. The result of each optimization was a Pareto front. In principle, any point within this Pareto front is an optimal candidate, and the balance between compactness and cost will depend on the considered application. To select the traded configuration from the Pareto front, a selection parameter SP (see Equation A12) of each configuration in the Pareto is calculated and finally the one with the lowest SP is selected. $F_i(x)$ represents the fitness of each individual in the Pareto—i.e., cost and perimeter, and α_i represents the weighting coefficients. These coefficients allow the designer to place more emphasis on any of the objectives when selecting the heat exchanger from the Pareto front. In the current case, their values were set to 0.5, in order to select the configuration that trades equally both objectives:

$$SP(x) = \sqrt{\sum_{i=0}^{i=n} \alpha_i \left(\frac{F_i(x)}{\max(F_i)} \right)^2}; \sum \alpha_i = 1. \quad (\text{A12})$$

For the saturator design, a structured packing-bed tower was selected. For packing sizing, the models presented by Coulson and Richardson [46], and Aramayo-Prudencio and Young [47] were used. This method calculates the thermodynamic performance of the saturator tower using as input the geometrical data and the inlet conditions. Thus, it allows the derivation of the tower’s key dimensions—i.e., the height and diameter of the packing—on a single iterative process, as the thermodynamic performance is defined by the cycle. The model was validated against experimental data presented by Lindquist et al. [17] and the results are presented in Table A2. The cost of the tower was calculated based on the shell weight and the packing volume. For the packing material a cost of 3800 USD/m³ was assumed while a material cost of 8800 USD/ton was used for the vessel, both made in stainless steel based on [48]. To account for the costs of the droplet eliminators, water sprays, and the rest of subcomponents, an additional 14,000 USD was added on top of the baseline value of the saturator [48].

Table A2. Validation of tower packing height calculation against experimental data from Lindquist et al. [17].

Inlet/Input Conditions			Outlet Conditions				
Parameter	Units	Data	Parameter	Units	Model	Experimental Data	Discrepancy (%)
$m_{g,1}$	(kg/s)	2.17	$m_{g,2}$	(kg/s)	2.54	2.55	0.4
$T_{g,1}$	(K)	346.75	$T_{g,2}$	(K)	390.64	389.15	0.4
p_1	(bar)	7.88	ω_2	(-)	0.176	-	-
h_t	(m)	0.57	-	-	-	-	-
d_t	(m)	0.70	-	-	-	-	-
$m_{w,2}$	(kg/s)	3.48	$m_{w,1}$	(kg/s)	3.11	3.10	0.4
$T_{w,2}$	(K)	419.35	$T_{w,1}$	(K)	358.67	352.85	1.6

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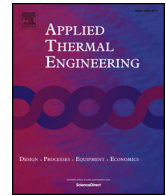
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Part-load performance modelling of a reheated humid air turbine power cycle

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HIGHLIGHTS

- A novel method for the part-load performance simulation of a reheated HAT system is described.
- Predicted performance is shown across a range of turbine inlet temperatures settings.
- The impact of sea water and ambient air temperature on the performance of the cycle is evaluated.

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ABSTRACT

Humid air turbines have previously demonstrated the potential to deliver high efficiency and power output combined with low emissions. This paper investigates the part-load performance of a 40 MW class advanced humid air turbine for power generation applications across a range of operating conditions. The paper investigates the impact of the main burner and reheater burner on the system's part-load power output and thermal efficiency and provides insights into the behavior of the key modules across the power spectrum of operation including the saturator tower which was never reported previously. The impact of the ambient air and sea water temperature on the cycle's performance are also investigated. The outcome of the research shows that the thermal efficiency if the system is less than 0.26% penalized when operating down to 50% of the design power output. Sea water temperature was found to have a more notable impact than ambient air temperature on both power output and thermal efficiency. Overall, this work constitutes a step ahead in understanding the potential benefits of an advanced humid air turbine system for power generation applications across a range of operating conditions which is not previously shown.

1. Introduction

Humid air turbine (HAT) systems are among the most efficient gas turbine based thermal cycles as highlighted by Jonsson and Yan in [1]. Previous studies (Rao et al. [2], Chiesa et al. [3], Jonsson and Yan [4]) have demonstrated effective utilization of low-quality waste heat within these cycles and the high thermal efficiency these systems can achieve. Chiesa et al. [3] performed a comparison between a number of humid cycles whereby HAT was identified as the most efficient, with a maximum thermal efficiency of 54%. Jonsson and Yan [5] and Traverso et al. [6] presented detailed techno-economic studies that showed that HAT cycles can offer lower investment cost and higher thermal efficiencies than combined gas turbine cycles (CCGT). Ågren and Westermarck [7,8] introduced the concept of saturator by-pass while Jonsson and Yan [5] proved that this would not only increase the efficiency of the cycles for high pressure ratios but also reduce the specific

investment cost of the whole system. Among others, Kavanagh and Parks ([9,10]), Moller et al. [11] and Nyberg et al. [12] studied the impact of each heat exchanger on the efficiency of the cycle. The influence of the ambient temperature on the off-design performance of the cycle was studied by Wang et al. [13] and Kim et al. [14] concluding that the cycle is less sensitive to ambient temperature variations than simple gas turbine cycles. Both works investigated the impact of the ambient temperature but since a closed water loop was assumed the impact of the water temperature was never considered. Wei et al. [15] conducted an experimental investigation on the off-design performance of a small-sized HAT cycle to conclude that the thermal efficiency increased by 3.1% compared to a simple cycle configuration. Takahashi [16] investigated the part-load performance of an advanced humid air turbine (AHAT) and the impact of ambient temperature. The off-design performance of the saturator tower was previously studied experimentally by Pedemonte et al. [17,18], whereas a methodology for the

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Nomenclature

Symbols

C	heat capacity (kJ/K)
C^*	heat capacity ratio (-)
kxA	overall heat transfer coefficient x heat transfer area (Inverse of total resistance) (kW/K)
M	mass flow rate (kg/s)
P	pressure (MPa)
$\hat{P}$	power output (MW)
Q	heat transfer (MW)
R_a	specific gas constant of dry air (J/(kg K))
R_v	specific gas constant of dry vapor (J/(kg K))
T	temperature (K)

Abbreviations

AC	aftercooler
CC	combustion chamber
CCGTs	combined cycle gas turbines
CHAT	cascade humidified advanced turbine
EC	economizer
EvGT	evaporative gas turbine
HAT	humid air turbine
HPC	high-pressure compressor
HPT	high-pressure turbine
IC	intercooler

LPC	low-pressure compressor
LPT	low-pressure turbine
OPR	overall pressure ratio
pp	percentage point
PR	pressure ratio
PT	power turbine
REC	recuperator
RH	reheater
relPR	relative pressure ratio
SAT	saturator
TIT	turbine inlet temperature
vIGV	variable inlet guide vanes

Subscripts

a	dry air
g	gas
in	inlet
2	top section of the saturator
sat	saturation
w	water

Greek Symbols

β	saturator by-pass
ε	effectiveness
ω	absolute humidity
Φ	relative humidity

simulation of the saturator tower was developed by Aramayo-Prudencio et al. [19]. Brighenti et al. [20] studied the design point performance of a reheated humid air turbine cycle and the impact of the technology of selected components on the preliminary acquisition cost of the heat exchanger units. As Chiesa et al. [3] highlighted, such a cycle features notably better performance relatively to a single burner system that may promote the exploitation of humid cycle power plants in applications where high thermal efficiency and high power-to-plant size ratios are of importance.

A potential application for a reheated HAT system may be within the power generation or the marine propulsion markets. The high thermal efficiency and low NO_x and SO_x emissions [22] could potentially make such power plants attractive in view of future marine emission regulation [23]. Nevertheless, the part-load performance of open water loop HAT systems remains still unexplored. This paper presents a methodology for the modelling and simulation of the steady state part-load performance of a 40 MW reheated HAT cycle across a range of operating conditions. The impact of the outlet temperature of both combustion chambers on the power output and thermal efficiency of the cycle is examined. The impact of ambient air and water temperature on the cycle performance is also presented.

2. Methodology

2.1. Cycle description

Fig. 1 shows the reheated HAT cycle layout analyzed herein comprising a dual-shaft, and free power turbine gas-turbine arrangement. The cycle features a reheater combustion chamber upstream of the power turbine. A detailed description of the power system is presented by Brighenti et al. in [20]. In this study, an open water loop is used to supply the heat exchangers. The evaporation process in the saturator and an efficient droplet eliminator at the gas outlet of the saturator acts as a water-treatment device, enabling the use of direct sea water [24]. The cycle design is a result of the optimization of the design parameters

for maximum thermal efficiency as shown by Brighenti et al. [20]. No mechanical or heat losses are considered. The cycle design parameters is reported in Table 1.

To capture future trends in recuperator and air/water heat exchanger design and exploit the full thermal efficiency potential of the RHAT cycle, 90% recuperator effectiveness was assumed for this study while an effectiveness of 95% was assumed for all air/water heat exchangers. The assumed effectiveness level for the recuperator and the air/water heat exchangers is slightly higher than the current standards and as such larger units in size will be required with knock-on effects in the overall system's volume, weight and cost. The impact of the effectiveness (defined as $\varepsilon = Q/Q_{max}$) of the recuperator and air/water heat exchangers on the acquisition cost, weight and performance of the cycle layout at design point has been previously analyzed by Brighenti et al. in [20]. For the current study, the assumed heat exchanger effectiveness levels are considered to be representative of near future trends and are educated by the effectiveness levels of the WR21 heat exchangers previously reported by Crisalli and Parker in [21].

Marine diesel fuel properties were used for combustion calculations [25]. Turbine cooling flows are calculated using the methodology proposed by Young and Wilcock [26]. Typical values for maximum metal temperature ($T_{m,max} = 1300$ K), film cooling effectiveness

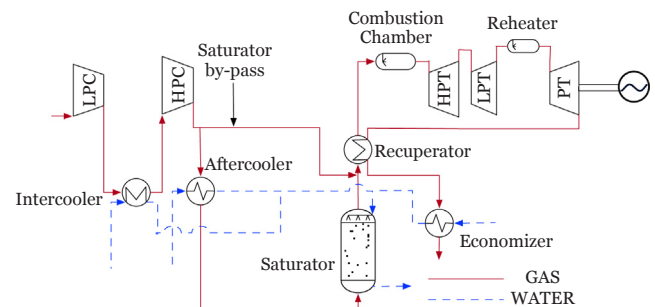


Fig. 1. Reheated HAT system layout.

Table 1

Cycle design parameters.

	Value	Constraints/assumptions
Ambient conditions		
Ambient temperature	288.15 K	
Ambient pressure	101.325 kPa	
Air relative humidity	80%	
Water temperature	288.15	
Compressors		
Overall pressure ratio	40:1	
Relative pressure ratio ($relPR = \log_{OPR} PR_{LPC}$)	0.526	$0.1 < relPR < 0.9$
Polytropic efficiency	90%	
Combustion chambers		
Combustion efficiency	99%	
Pressure loss	5%	
Turbine inlet temperature	1600 K	
Turbines		
High and low-pressure turbine polytropic efficiency	90%	
Power turbine polytropic efficiency	92%	
Recuperator		
Effectiveness	90%	
Hot-side pressure loss	5%	
Cold-side pressure loss	7.5%	
Air/Water heat exchangers		
Effectiveness	95%	
Inter-cooler water mass flow	15.4 kg/s	$C_{IC}^* = C_{min}/C_{max} \leq 0.95$ & $T_{w,out} \leq T_{sat} @ (0.9 * p_{SAT})$
After-cooler water mass flow	4.89 kg/s	$C_{AC}^* \leq 0.95$ & $T_{w,out} \leq T_{sat} @ (0.9 * p_{SAT})$
Economiser water mass flow	19.2 kg/s	$C_{AC}^* \leq 0.95$ & $T_{w,out} \leq T_{sat} @ (0.9 * p_{SAT})$
Pressure loss	7.5%	
Saturator		
Pinch temperature difference	5 K	
Pressure loss	5%	
Global parameters		
By-pass ratio $\beta = m_{g,SAT}/m_{g,GT \text{ inlet}}$	50.2%	$0\% \leq \beta \leq 100\%$
Gas inlet mass flow	40.89 kg/s	

($\epsilon_f = 40\%$) and internal flow cooling efficiency ($\eta_c = 70\%$) are assumed as discussed in [27]. The cooling flows are extracted downstream of the high-pressure compressor and upstream of the aftercooler.

2.2. Off-design modelling

The performance of the gas turbine and recuperator are resolved by TURBOMATCH [28,29] interfaced with MatLab functions for heat exchanger modelling. A Broyden based solver was employed to match the cycle components at part load [30]. The developed part load simulation algorithm is shown in Fig. 2. It requires the design point vector of the cycle and the off-design handles as free variables to control the engine operation. The system of checks and guesses used within TURBOMATCH as well as the integration of component maps within the scheme is further described by Pellegrini [29].

The performance model of the heat exchangers relies on the variation of the kxA coefficient foreach heat exchanging unit. Assuming that the flow properties of the two streams do not vary substantially from design point, Prandtl number and the thermal conductivity of the fluid can be considered fairly constant. Thus, the main driver for the variation of the heat transfer is the variation of the Reynolds number and consequently of the stream mass flow. The Dittu-Boelter heat transfer correlation, therefore, enables scaling of the kxA parameter at off-design conditions as:

$$kxA = kxA_{ref} \left(\frac{m_c^{0.8} m_h^{0.8}}{m_{c,ref}^{0.8} m_{h,ref}^{0.8}} \right) \left(\frac{m_c^{0.8} + m_h^{0.8}}{m_{c,ref}^{0.8} + m_{h,ref}^{0.8}} \right) \quad (1)$$

where c and h indicate the cold and hot streams and ref indicate the design values of the parameter.

The off-design effectiveness of the heat exchanger is calculated under the assumption that the flow arrangement can be approximated as being counter-current.

$$\epsilon = \frac{1 - \exp[-NTU(1-C^*)]}{1 - C^* \exp[-NTU(1-C^*)]} \quad (2)$$

where $NTU = kxA/C_{min}$ are the number of transfer units and $C^* = C_{min}/C_{max}$ is the heat capacity ratio between the two streams. Once the new effectiveness is defined, the outlet conditions of the heat exchanger are calculated with the definition of effectiveness as $\epsilon = Q/Q_{max}$. In order to avoid steaming within the heat exchanger, a threshold at the water outlet temperature is applied based on its saturation temperature at 90% of the pressure in the saturator:

$$T_{w,max} = T_{sat} @ (0.9 * p_{SAT}) \quad (3)$$

The pressure losses in each stream of the heat exchangers are calculated as suggested by Walsh and Fletcher in [30] as follows:

$$\frac{\Delta P}{P_{in}} = \alpha \left(\frac{m \sqrt{T_{in}}}{P_{in}} \right)^2 \quad (4)$$

where α is a pseudo loss coefficient derived at design point and remains constant at part-load calculations as discussed by Walsh and Fletcher in [31].

The model for the off-design performance calculations of the saturator is based on the Aramayo-Prudencio et al. [19] and Lindquist et al. [32]. The saturator's cross-section sizing is performed as shown by Dalili in [33] while the pressure drop across the device is estimated as shown by Coulson et al. in [34]. The model has been validated against experimental data from Lindquist et al. [32] as the maximum discrepancy is of 1.6% as reported by Brighenti et al. in [20].

3. Results and discussion

3.1. Impact of turbine inlet temperature

The $TITs$ of the main burner (TIT_{CC}) and reheater (TIT_{RH}) are used as handles for the part-load performance across the range reported in Table 2. The by-pass ratio of the saturator is kept constant at the design

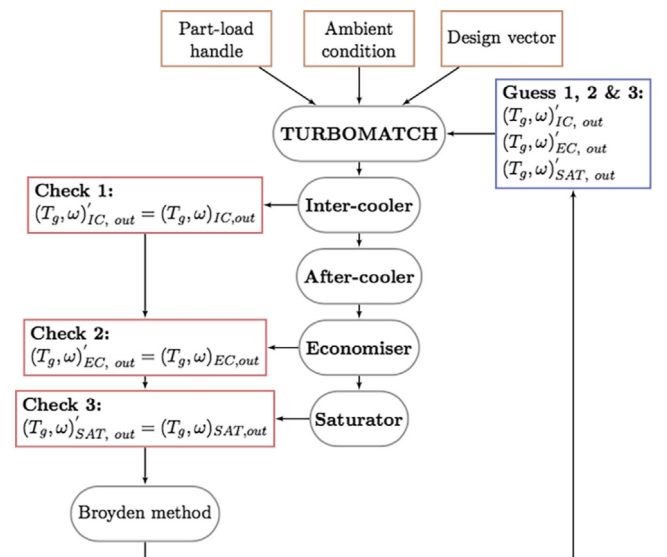


Fig. 2. Schematic layout of the Humid Air Turbine.

Table 2

Variation of the TIT of the main and reheater combustion chamber at part load.

TIT_{CC}	1600: –25:1275 K
TIT_{RH}	1600: –25:1200 K

value.

As long as no additional water is streamed into the heat exchangers to prevent steaming in the saturator, the water mass flow in each heat exchanger is kept constant at the design value. As such, the variation of the heat transfer coefficient of the heat exchanger (see Eq. (1)) becomes a function only of the gas mass flow.

Fig. 3 illustrates the system's running lines across a range of TIT_{CC} between 1600 K (design point) and 1275 K. The reduction in the turbine inlet temperature at the exit of the main burner (TIT_{CC}) results in a 65% reduction in inlet gas mass flow and a reduction in pressure ratio and non-dimensional mass flow at both compressors. The rotational speed of the low-pressure shaft is mostly affected by the reduction in TIT_{CC} from 1600 K to 1275 K and drops to 47% of the design value. For the same reduction in TIT_{CC} the rotational speed of the high-pressure shaft drops to 84% of the design value. Although typically a variation in the rotational speed between the two shafts is normal, this difference is further pronounced by the presence of the intercooler. Due to the presence of the intercooler, the temperature of the gas at inlet of the HPC drops by only 11 K when the TIT_{CC} is reduced from 1600 K to 1275 K which mitigates notable reductions in non-dimensional mass flow. The proximity of the operating point of the LPC to the surge line could raise issues related to the operability of the system in such conditions which will require special treatment of the compression system.

When TIT_{RH} is reduced from 1600 K to 1375 K and TIT_{CC} is kept constant at 1600 K (Fig. 4a–b) a 5.2% increase in the inlet flow of the cycle can be observed. The non-dimensional mass flow between the outlet of the LPT and the PT inlet increases due to the temperature rise in the reheater combustion chamber which, neglecting the fuel mass flow, it can be written as:

$$\left(\frac{m\sqrt{T}}{P}\right)_{PT,in} = \left(\frac{m\sqrt{T}}{P}\right)_{LPT,out} \sqrt{\frac{T_{PT,in}}{T_{LPT,out}}} \quad (5)$$

where $\sqrt{\frac{T_{PT,in}}{T_{LPT,out}}}$ is the temperature ratio across the reheater.

Given that the inlet non-dimensional mass flow does not vary significantly, the non-dimensional mass flow at the outlet of the LPT increases to satisfy Eq. (5) and the expansion ratio in the LPT also increases. This justifies the observed increase in the mass flow at the inlet of the gas turbine. This increase, nonetheless, depends on the variation of the humidity in the hot section of the cycle: if the humidity decreases, the increase in inlet gas mass flow is more significant than if the humidity remains constant or increases. Small variations in the pressure ratio of the high and low pressure compressors are observed as a result of the variations in the non-dimensional flow of the two systems.

Fig. 5 shows the thermal efficiency, power output and humidity at the inlet of the main burner of the HAT cycle across a range of main burner and reheater outlet temperatures. Two discrete regimes can be identified within the part load operating envelope of the cycle (A, B in Fig. 5a). Region A (Fig. 5a) where the system's power output is predominantly affected by the TIT of the main burner (TIT_{CC}), while the reduction in TIT_{RH} at constant TIT_{CC} is found to cause a slight increase the power output. In region B (Fig. 5a) although the TIT_{CC} remains the main driver of the power, a reduction in TIT_{RH} at constant TIT_{CC} causes a slight reduction in the power output. In terms of the system's thermal efficiency (Fig. 5b), there are again two discrete regions following the trend of the power output. Region A where the thermal efficiency is mainly dependent upon the TIT_{CC} and region B where the TIT_{RH} is found to have a detrimental effect on the efficiency of the cycle while

TIT_{CC} is only affecting it weakly.

The behavior of the system's power output across these identified part-load regimes A and B (see Fig. 5), is predominantly driven by the variations in gas humidity and temperature at the inlet of the combustion chamber as shown in Fig. 5c. Gas humidity (ω) depends on the pressure in the saturator tower (p_{SAT}), the relative gas humidity (Φ) and the air saturation pressure $P_{sat}(T_g)$ evaluated at the temperature of the gas T_g according to the following equation:

$$\omega = \frac{\Phi P_{sat}(T_g)}{P_{SAT} - \Phi P_{sat}(T_g)} \frac{R_a}{R_v} \quad (6)$$

Gas temperature (T_g) is in turn dependent upon the mass flows of the water and gas stream as well as upon the water inlet temperature at the saturation tower as described by Brighenti et al. in [20]. In region A of Fig. 5b the reduction in TIT_{CC} results in a reduction in humidity of the gas (Fig. 5c) that combined with the reduction in gas mass flow leads to a notable reduction in power output. Although the pressure in the saturator drops (Fig. 6a), the reduction in humidity is due to the water temperature drop at the inlet of the saturator (Fig. 6c) and a consequent drop in gas temperature at the top section of the saturator tower (Fig. 6b).

In region B TIT_{CC} has only a minor impact on the humidity reduction (Fig. 5c) as the water inlet temperature (Fig. 6c) in the saturator is less affected by TIT_{CC} than in region A. The smaller impact of the TIT_{CC} on the humidity leads to a weaker dependency of the power output

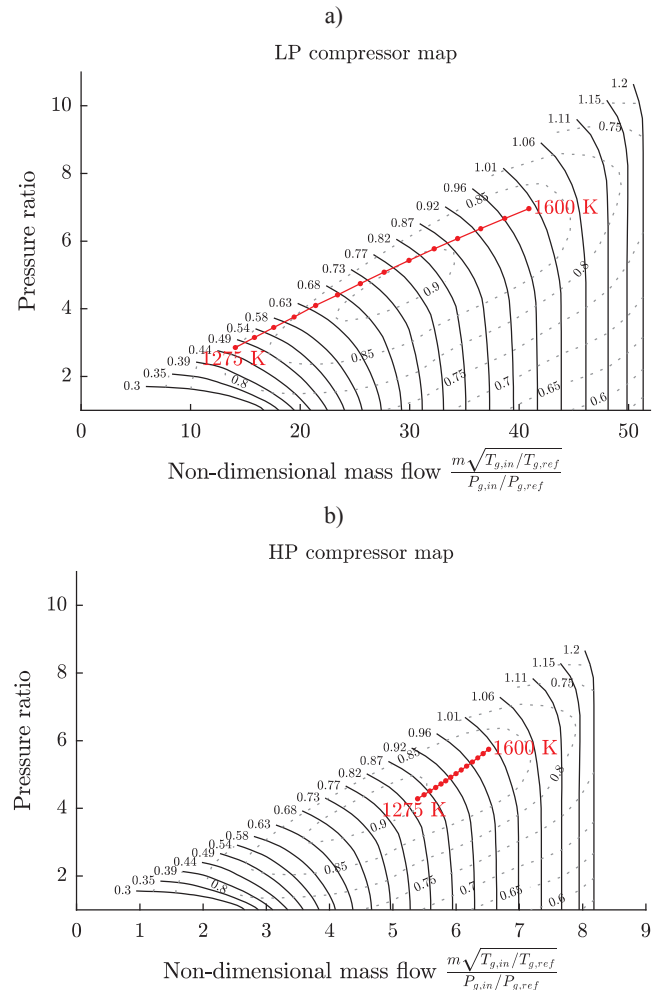


Fig. 3. Running lines with TIT_{CC} as a handle (in red) and $TIT_{RH} = 1600$ K. (a) low-pressure compressor and (b) high-pressure compressor map. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

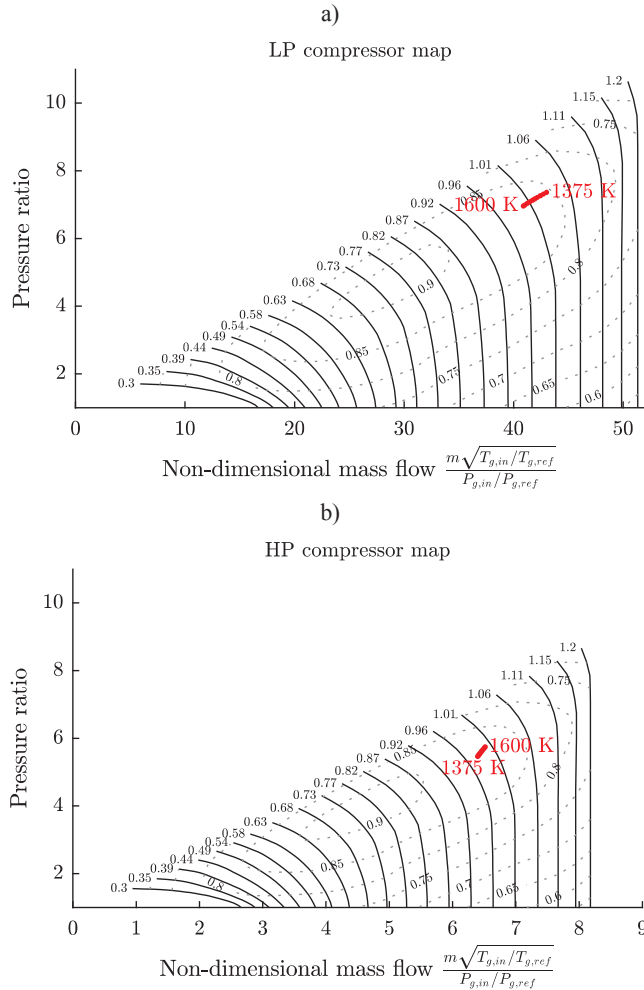


Fig. 4. Running lines with TIT_{RH} as a handle (in red) and $TIT_{CC} = 1600$. (a) low-pressure compressor and (b) high-pressure compressor map. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

upon the variations in TIT_{CC} . TIT_{RH} was found to have a more detrimental effect on the humidity of the gas in region B compared to region A (Fig. 5c). Although for reductions in TIT_{RH} the pressure in the saturator increases in both regions (A, B in Fig. 6a), the water

temperature (Fig. 6c) in region B drops whereas it increases across region A. The behavior of the water temperature at the inlet of the saturator is driven by the economizer performance within region A (see Eq. (3)). In region A the temperature of the water is directly affected by the pressure in the saturator, thus the temperature increases for reductions in TIT_{RH} . In region B, as the water to gas mass flow ratio is constant, the temperature is affected by the heat available in the economizer, thus the temperature drops for reductions in TIT_{RH} .

Cycle part-load control across a range of operating conditions is fundamental to maintain high thermal efficiencies across all power settings. Fig. 7a shows the variation of both power output and thermal efficiency across a range of TIT_{CC} and TIT_{RH} . The bold line with markers in Fig. 7a represents the operating points with maximum thermal efficiency for a power load between 100% and 25% of the design point power output. As shown in Fig. 7b, the highest thermal efficiency between 100% and 29% power output is achieved by a constant TIT_{RH} at its design value and a variable TIT_{CC} to control the load (Fig. 7b). Between 29% and 25% of power load a reduction in TIT_{RH} from 1600 K to 1500 K is also required as well as a reduction in TIT_{CC} to 1295 K. The efficiency of the cycle, between design point and 67% of power load, remains higher or equal to the design value, with a maximum increase of 0.16 pp at 80% of the power load. When at 50% of the power load, the thermal efficiency drops by 0.72 pp from the design value.

The cycle can be therefore operated down to approximately 50% of power load maintaining a thermal efficiency higher than 60%. Although the high thermal efficiency would limit the operating costs related to fuel consumption, the high TIT_{RH} needed across the power range to achieve such efficiencies may impact the life of the power turbine blade rows which may impact the maintenance costs. A full techno-economic analysis of the cycle for a specific operation would be necessary to assess the optimum part-load management of the cycle from a cost point of view.

An alternative common way for part-load control is via compressor variable inlet guide vanes (vIGVs). The performance of the reheated HAT cycle when varying the position of the vIGVs is studied by keeping the water to air mass flow ratio constant at the design values for all water to air heat exchangers. Both TIT_{RH} and TIT_{CC} are kept constant at 1600 K. Fig. 8 illustrates the impact of the vIGVs regulated inlet mass flow on the thermal efficiency and power output. When the cycle operates at 85% of the power output (34 MW) the thermal efficiency drops by 1.13 pp from the design point level. For the same power load of 34 MW, when achieved with the reduction of the TIT_{CC} , the thermal efficiency remains almost constant to the design point value (Fig. 7a). As such, operating the cycle at part load by using TIT_{CC} as handle,

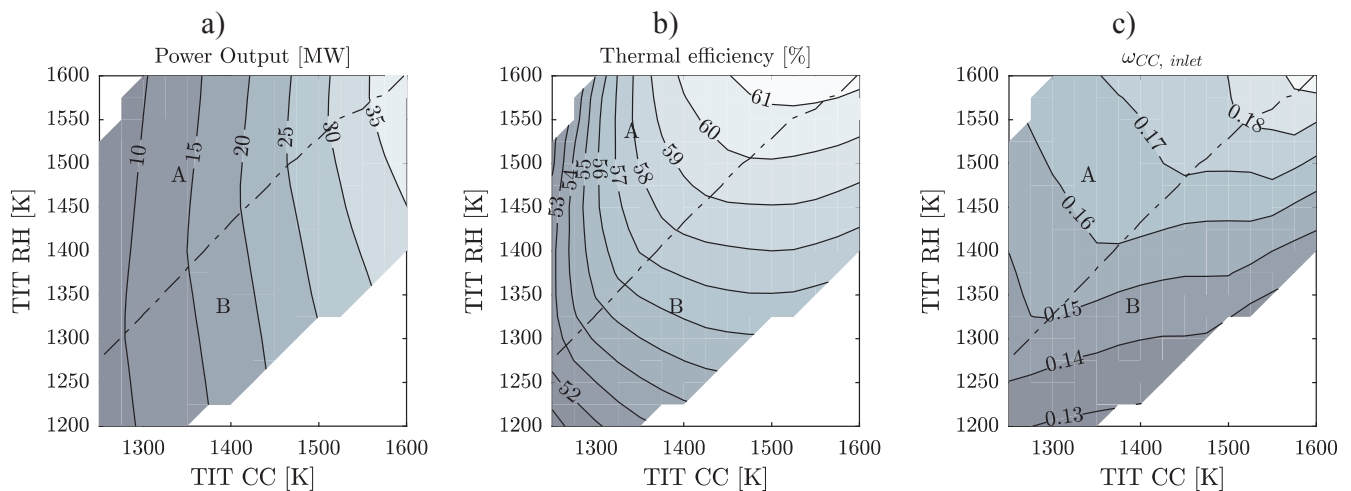


Fig. 5. Part-load performance of the HAT cycle across a range of main burner and reheater outlet temperatures. (a) power output, (b) thermal efficiency and (c) humidity at the inlet of the main burner.

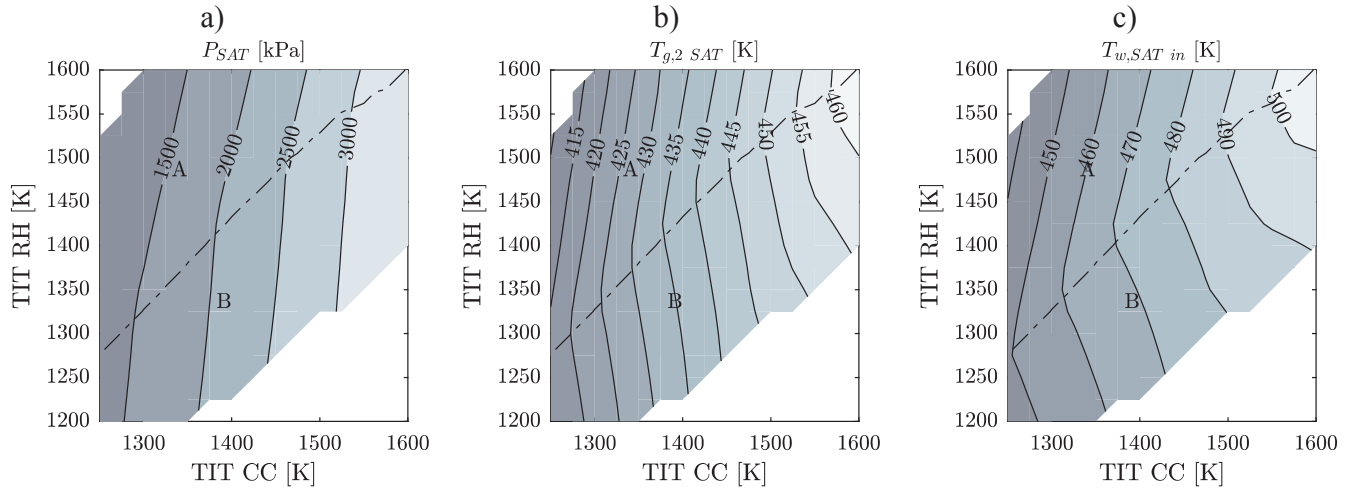


Fig. 6. (a) Saturator pressure, (b) gas outlet temperature and (c) water inlet temperature as a function of TIT_{CC} and TIT_{RH} .

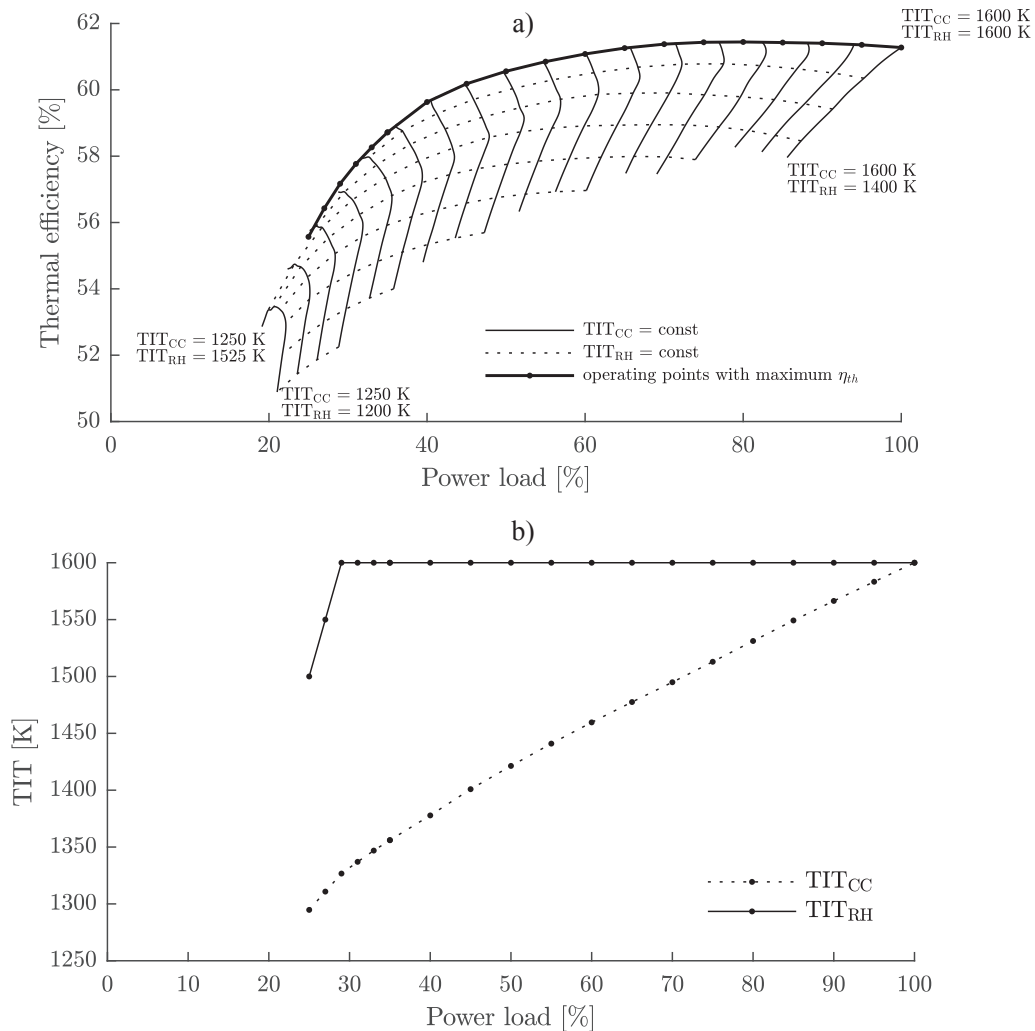


Fig. 7. (a) Thermal efficiency vs. power load in function of TIT_{CC} and TIT_{RH} . (b) TIT_{CC} and TIT_{RH} for maximum thermal efficiency across a range of power output.

rather the viGVs, seems a more viable approach from an efficiency prospective as well as from a maintenance cost viewpoint.

3.2. Impact of ambient conditions

In marine power applications, ambient air and sea water

temperature, in which such a power system can operate, can vary by several degrees depending on the time of day, season and latitude. The impact of ambient air and sea temperature on the cycle performance are studied at constant TIT_{CC} and TIT_{RH} at the nominal value of 1600 K. The water temperature (T_w) is varied parametrically from 280.15 K to 312.15 K whereas the ambient air temperature is varied as a function of

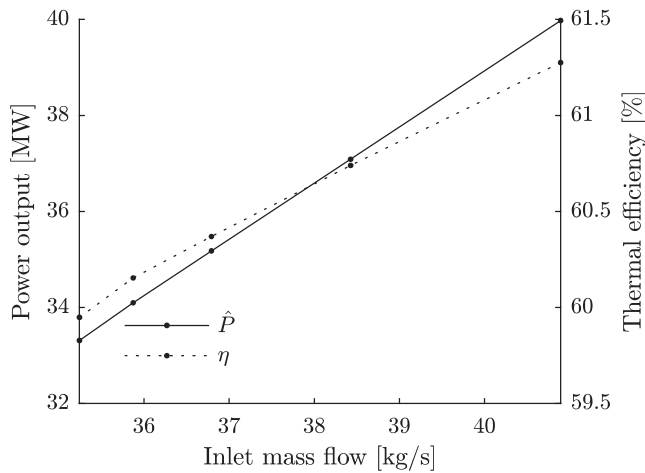


Fig. 8. Power output and thermal efficiency for performance control via vIGVs.

the water temperature from $T_w - 16$ K to $T_w + 16$ K. The water to air mass flow ratio is kept constant and if the temperature of the water at the outlet of the heat exchanger exceeds $T_{w,max} = T_{sat}@ (0.9 \cdot p_{SAT})$ then the water mass flow in that heat exchanger is increased so that $T_{HE,out} = T_{w,max}$.

Fig. 9 shows the variation in thermal efficiency and power output of the cycle as a function of water and air temperature. Thermal efficiency and power output are mostly affected by the ambient temperature for values higher than the design value of 288.15 K. The thermal efficiency and power output drop by 0.96 pp and 5.1 MW (12.7%) respectively for an increase in ambient temperature and sea water temperature of 10 K (308.15 K). Nevertheless, for the same air temperature (308.15 K), the variation of the water temperature from 292.15 K to 312.15 K penalises the thermal efficiency by 0.13 pp and the power output by 0.35 MW.

The reduction in power output and thermal efficiency is driven by the reduction in air compressibility for higher ambient temperatures and less effective inter-cooling for higher sea water temperature. In both cases, the new equilibrium steady state operating point of both HP and LP shafts shifts to lower power whereby gas mass flow (Fig. 10a) and pressure ratio are reduced. The humidity (Fig. 10b) on the other hand increases as a result of the reduction in pressure and the increase in water to air mass flow ratio in the saturator tower. This mitigates the reduction in power due to the reduction in compressibility of the air and pour inter-cooling compared to dry cycles. Part of the reduction in enthalpy available in the turbines due to the decrease in gas mass flow and pressure ratio is compensated by this increase in humidity.

For air temperatures below the design value the variations in thermal efficiency become very small. When both air and water temperatures decrease from the design values (288.15 K) to 280.15 K the thermal efficiency drops by only 0.09 pp whereas the power output

drops to 37.6 MW. In this region, where the air temperature is lower than the design value, the water temperature is found to have greater impact on power output than the air temperature (Fig. 9b).

Although previous research has shown that HAT cycles are able to achieve notably better performance in terms of thermal efficiency than combined steam-gas systems, the work presented herein identified potential shortcomings that may arise from the operation of such a system across a range of conditions which was never explored before. These are related to the operability of the compression system and the susceptibility this showed to the humidification across the operating envelop as well as with the impact of the ambient conditions on the cycle's performance and operability. In addition, humidified cycle technologies still present a number of unexplored areas which need addressing before any large scale deployment of such systems for power generation or propulsion applications. One of the main issues remains the volumetric mismatch in the flow rate between the compressors and turbines as a result of the enhanced humidification within the gas generator. Gas turbines able to accommodate the increased volumetric flow rate, necessary for the HAT to achieve its potential of high thermal efficiency, are yet to be developed. Moreover, HAT systems capable of achieving the high thermal efficiencies indicated in this work, will require the development of high effectiveness heat exchangers. Although the current trends in the heat exchanger market indicate that the required levels of effectiveness will be feasible in the near future, such technologies have yet to be fully demonstrated. At the moment, previous experience in operating heat exchangers as part of a gas turbine cycle for propulsion applications across a range of operating conditions is still limited to the WR-21 marine prime mover [21].

4. Conclusions

A study on the part-load performance of a reheated humid air turbine system is presented herein. The method developed to predict the stand-alone, off-design performance of key components including a validated method for the modelling of the saturator is shown. The integration of the individual models onto a system simulation platform is described. Key equations and iterative engine matching schemes are provided. The developed method was used for the system's performance analysis across a range of power settings in terms of turbine entry temperatures.

Variations in the turbine inlet temperature at the exit of both the main as well the reheater burners primarily affect the humidification process, which in turn drives the output power and thermal efficiency behavior across the range of operating conditions. Two discrete regimes were identified within the part load operating envelope of the cycle. In the first one, the variation in humidity injected into the cycle is driven by variation in the saturator's pressure. In this region the reduction of the TIT of both main and reheater burners results in a reduction in humidity. In the second region the variation in humidity levels is

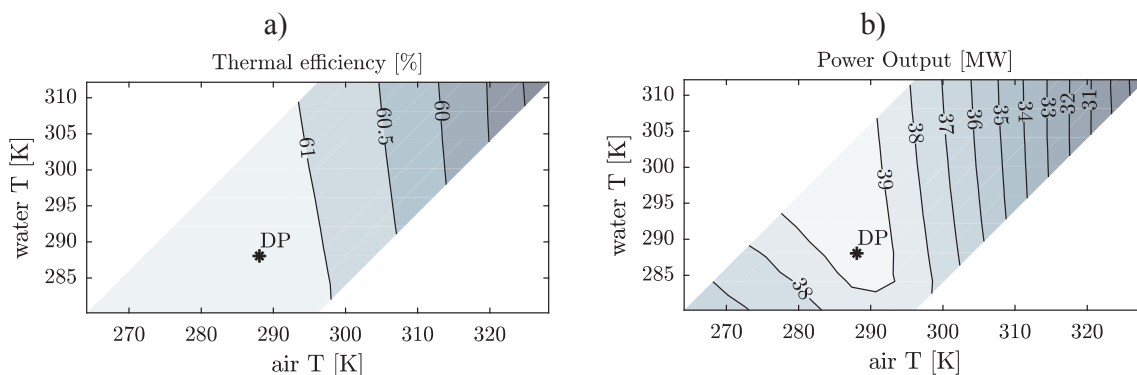


Fig. 9. (a) Thermal efficiency and (b) power output as a function of air and water temperature.

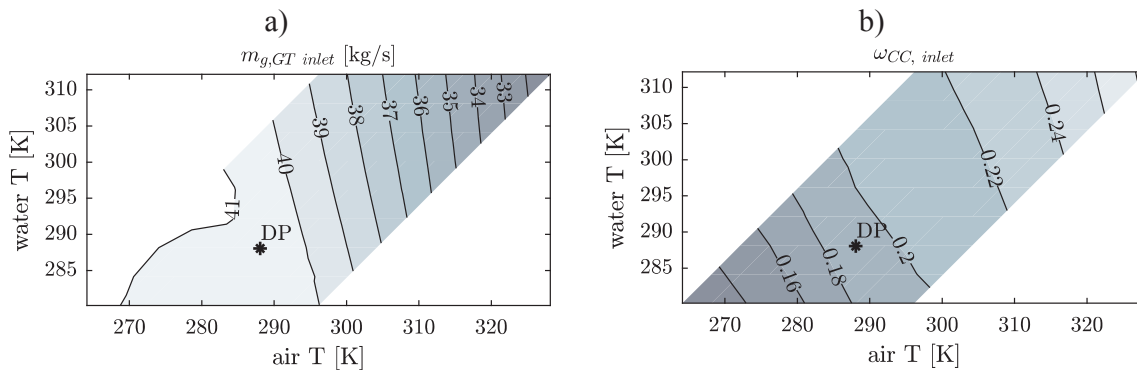


Fig. 10. (a) Gas mass flow and (b) humidity at the inlet of main burner as a function of the air and water temperature.

limited by the available heat in the air/water heat exchangers. In this region the reduction of the TIT of the main burner causes an increase in humidity, whereas the reduction of the TIT of the reheater burner results in a more notable reduction in humidity. The reheated HAT cycle was found able to achieve thermal efficiency beyond 55% across a wide range of power settings down to 25% of its design nominal power output with values above 60% across most of this range. Controlling the power output via the temperature at the exit of the reheater was found to have a detrimental effect on the system's thermal efficiency across the range of operation. Power output control via the main burner's exit temperature (TIT_{CC}) at constant reheater's temperature at its design value was found able to fully exploit the high thermal efficiency potential of the HAT system across the range of operation. However, varying the main burner's temperature (TIT_{CC}) was found to be associated with potential operability issues at the low pressure compression system as its surge margin can be notably penalized by the increase in pressure ratio. Operating point variations using variable geometry as part of the compression system may be able to mitigate some of the operability penalties observed across the range but was found to also have a detrimental effect on the cycle's efficiency at part load. Finally, the ambient air temperature was found to have a larger impact on the thermal efficiency and power output than the water temperature. Any variation in water and ambient air temperature from their design values was found to cause penalties to the power output of the system when the TIT at the outlet of both combustion chambers remains constant. The efficiency of the cycle was found unaffected by reductions in ambient air and water temperatures, whereas a reduction of approximately 0.1 pp per degree K was observed for higher than nominal ambient air temperatures driven by the amount of heat recuperated back into the cycle and the amount of heat and mass exchanged in the saturator tower.

Overall, the current work presents a novel methodology to enable performance predictions of a humid air turbine cycle across a range of operating conditions. The proposed method enables part load investigations of the reheated HAT system which were not previously possible with existing methods. A number of considerations with regards to the maturity of such system for the power generation market was also enabled. These are focused to the design and development of gas generator systems able to handle the notable volumetric discrepancy between the turbine and compression systems which is the result of the enhanced humidification in such a cycle but also on the availability of state of the art heat exchanging units able to deliver the effectiveness levels required to achieve high thermal efficiencies. The outcomes of this work constitute a step forward in understanding the behavior of a humid air turbine across a range of operating conditions and enable performance assessments even from the early stages of the preliminary design process.

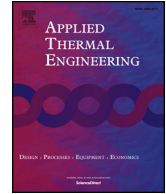
Acknowledgments

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Research Paper

Techno-economic analysis of a reheated humid air turbine

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HIGHLIGHTS

- Economiser and recuperator dictate the economic metrics of a reheated humid air turbine system.
- Fuel and equipment purchase cost drive the average cost of the power produced by the system.
- Humid air turbine shows 14% higher thermal efficiency than a typical combined gas-steam plant.
- Humid air turbine show roughly 62% lower total purchasing cost than a combined gas-steam plant.

ARTICLE INFO

Keywords:

Humid air turbine
Evaporative gas turbine
Economic analysis
Power generation

ABSTRACT

The purpose of this paper is to identify the economic potential of a reheated humid air turbine system for power generation applications. A parametric analysis is performed to correlate the technology level of the system with the required cost of the electricity for economic viability. The effect of fluctuations of the main cost drivers is evaluated via an uncertainty analysis. The performance of the studied reheated humid air turbine is compared against previously studied humid configurations and well established gas-steam combined cycles. The fuel cost is found to be driving the cost of electricity. The uncertainty analysis also shows the dependency of the optimum cycle design parameters upon the market prices. The analysis reveals the capability of the reheated humid air turbine to be an economically viable option for the power generation sector featuring an estimated cost of electricity 2.2% lower than simpler humid cycles, and 28% lower than established combined cycles currently in service. The outcome of the work constitutes a step forward in the understanding of the economic performance of advanced complex cycles and proves the potential of such systems for applications where high efficiency and economic performance is jointly required.

1. Introduction

Over the past decades, thermal efficiency enhancements in gas turbine systems have been a key driver in the development of advanced power plant configurations. Advanced gas turbine configurations previously studied include steam injection, triple-pressure combined cycles, and humid air turbines [1–5]. Although combined gas-steam cycles are currently an established option in terms of thermal efficiency, several studies have postulated that humid air systems could also be attractive in the small to medium-size power generation market [5–10].

Humid Air Turbines (HAT) or Evaporative Gas Turbines (EvGT) were initially introduced by Rao in the late 80's [11]. Jonsson and Yan [9] performed a techno-economic comparison between HATs and combined cycles. This study proved the capability of the HATs to achieve a similar cost of electricity with a lower specific investment cost. Subsequent cost studies performed by Traverso and Massardo [12], and Kavanagh and Parks [10] showed that HATs are capable to

achieve a lower cost of electricity than Combined Cycle Gas Turbines (CCGT), demonstrating the techno-economic potential of this advance cycle for the power generation market. The ability of the HAT systems to recuperate a notable part of the waste heat back into the cycle without the need of a bottoming cycle is the main driver of the observed competence against the CCGTs.

Pedemonte et al. [13,14] experimentally analysed the off-design performance of the air saturator. Wang et al. [15] and Kim et al. [7] studied the effect of the ambient conditions on the performance of the HAT. In both works, it was proved that as the ambient temperature increases the performance of the HAT is less penalised than the 'dry' gas turbines or the CCGTs. During warm days, the humid cycle is capable to evaporate a larger amount of water compensating the negative effect of a poorer compressor performance. In addition, Takashi et al. [16] concluded that humid air turbines show a better efficiency than CCGTs during part-load performance across a similar range of operation. In terms of emissions, Yagi et al. [17] reported that HAT NO_x emissions

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Nomenclature*Symbols*

C_x	[J/K] Heat capacity
C^*	[-] heat capacity ratio
c_p	[J/kg K] specific heat capacity
COE	[\$/kW h] cost of electricity
$\overline{COE}$	[\$/kW h] average cost of electricity
H^+	[J/kg] enthalpy invariant
h	[J/kg] specific enthalpy
h_{fg}	[J/K] specific enthalpy of evaporation
M^+	[-] mass invariant
$\dot{m}$	[kg/s] mass flow
n	[years] years of life of the power plant
p	[Pa] pressure
p_{sat}	[Pa] saturation pressure
PEC	[\$] purchase equipment cost
$\dot{Q}$	[W] heat rate
R_x	[J/kg K] specific gas constant
$SPEC$	[\$/kW] specific purchase equipment cost
ΔT_{sp}	[K] saturator pinch point temperature difference
T	[K] temperature
t	[hours/year] hours of operation per year
T_{dew}	[K] wet bulb temperature
T_{sat}	[K] saturation temperature
$\dot{W}$	[W] plant power output

Abbreviations

AC	aftercooler
AE	annual expenses
CEPCI	chemical engineering plant cost index
DC	direct costs
EC	economiser
EvGT	evaporative gas turbine
FCI	fixed capital investment

HAT	humid air turbine
IC	intercooler
Int	interests
LNG	liquid natural gas
O&M	operation and maintenance
PFI	plant fixed investment
RC	recuperator
RHAT	reheated humid air turbine
SAT	saturator
TCI	Total Cost of Investment

Subscripts

a	dry air
$comb$	combustor
$comp$	compressor
fin	financing
g	humid air
gen	generator
HX	heat exchangers
i	operational year
in	inlet
ini	initial
max	maximum
min	minimum
out	outlet
v	vapour
w	water

Greek Symbols

η_{th}	thermal efficiency
ε	effectiveness
Ξ	[\$] cost
ϕ	relative humidity
ω	water to air ratio

can be as low as roughly 10 ppm due to the high content of water within in the combustion chamber. Moreover, HAT systems are more compact power-units compared with ‘dry’ gas turbine packages and CCGT and present faster start-up times.

Although previous studies have focused on the performance capabilities of HAT systems, little effort has been invested to understand the full techno-economic potential of this cycle. Chiesa et al. [18] suggested that the addition of a reheater in the gas turbine would augment the thermal efficiency and specific work of the power plant. A reheated HAT system was previously studied by Brighenti et al. [19]. This work confirmed the potential of the reheated HAT configuration to achieve thermal efficiencies beyond the threshold of 60%. Nevertheless, no economic study of the reheated HAT system has been presented so far to identify the economic viability of such a system.

This paper presents a techno-economic analysis of a 40 MW class reheated humid air turbine power plant for power generation. A parametric design space exploration is performed to demonstrate the impact of the heat exchanger technology level on the economic metrics. An uncertainty analysis showing the impact of the main cost driver fluctuations on the cost of the electricity is also included. Finally, the economic performance of the investigated cycle is benchmarked against the performance achieved by high efficiency humid and combined cycle systems previously presented.

2. Methodology**2.1. Cycle configuration and modelling approach**

The Reheated Humid Air Turbine (RHAT) analysed in this study, shown in Fig. 1, is based on the configuration previously presented by Brighenti et al. [19]. The cycle layout includes an aftercooler to augment the saturator performance as proposed by Thern et al. [20], an

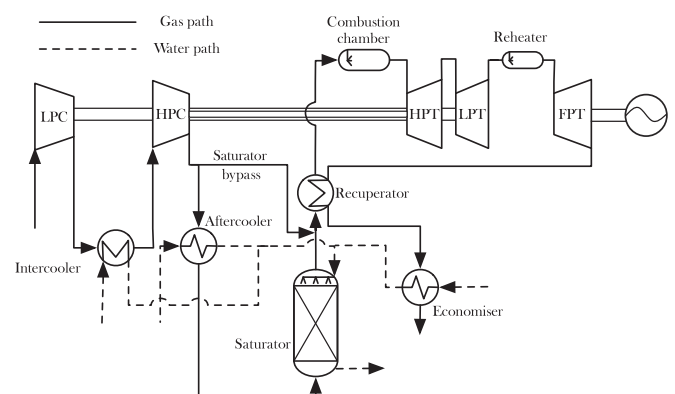


Fig. 1. Reheated humid air turbine system cycle layout.

intercooler to minimise the compression work, and an economiser to raise the temperature of the feeding water, as discussed by Nyberg and Thern [21]. A saturator bypass is included between the exit of the high pressure compressor and the inlet of the combustion chamber as suggested by Ågren and Westermarck [22,23]. An open loop is used to feed water in the heat exchangers as previously discussed by Rosen [24].

The thermal performance of the system is simulated using a computational platform comprising modules that estimate the thermal behaviour of each component of the RHAT system. This platform has been previously described in detail by Brighenti et al. [19]. The performance of the gas turbine and the recuperator is calculated by Turbomatch© [25], a zero-dimensional, modular, component-based simulation framework developed at Cranfield University. The performance of the turbines and compressors are calculated using the polytropic expansion/compression relationships, the efficiency definitions, and the enthalpy balances. The performance of the combustor is estimated the enthalpy balance and the assumed efficiency of the burner. Details about the modelling approach for the main parts of the gas generator, i.e. compressors, turbines and burners, are shown by Walsh and Fletcher [26]. The performance calculation platform has been extensively validated in previous studies [27–30]. The performance of the saturator is calculated using the model based on mass and energy balance, in addition to mass and heat transfer equations, presented by Brighenti et al. [19]. This model has been also validated against experimental data provided by Lindquist et al. [31]. The key equations of the saturator performance calculation method adopted herein as well as the validation of the method is shown in Appendix A. The outlet conditions from all air-water heat exchangers and the recuperator unit are calculated using their imposed effectiveness and the energy balance equations as shown in Appendix B. The simulation platform also includes the calculation of the required turbine blade cooling flows, bled at the outlet of the high pressure compressor. The approach developed by Young and Wilcock in [32] has been employed for the estimation of the turbine cooling requirements.

The required parameters for the definition of the cycle are presented in Table 1, together with their assumed values. The bypass ratio, heat capacity ratio of the heat exchangers, and relative pressure ratio are optimised to maximise the thermal efficiency of the cycle, according to the model presented by Brighenti et al. [19]. The bypass ratio of the saturator is an optimised variable that varies across a range between 0 and 1, the relative pressure ratio from 0.25 to 0.75, and the air-water heat exchanger's heat capacity ratio from 0.3 to 0.95. The effectiveness of the heat exchangers is used as the technology level indicator of the cycle. The heat exchanger technology level scenarios analysed are shown in Table 2 and include a low, an average, and a high technology level case in terms of heat exchanger effectiveness. The optimised cycle parameters for each heat exchanger technology scenario are shown in Table 3.

Table 1
Cycle performance modelling assumptions.

Power output	40 MW
Turbine inlet temperature for both combustion chambers	1600 K
Overall pressure ratio	40
Compressor polytropic efficiency	0.90
Core turbine polytropic efficiency	0.90
Power turbine polytropic efficiency	0.92
Combustion chamber efficiency	0.99
Combustion chamber pressure loss	5%
Saturator pinch point	5 K
Saturator pressure loss	5%
Water-air heat exchangers air-side pressure loss	7.5%
Cold-side recuperator pressure loss	7.5%
Hot-side recuperator pressure loss	5%
Maximum allowed blade metal temperature	1300 K
Film cooling effectiveness	0.40
Internal flow cooling efficiency	0.70

Table 2

Heat exchanger effectiveness envelopes analysed.

Parameter	Technology level		
	Low	Average	High
Intercooler, Aftercooler, Economiser effectiveness	0.75	0.85	0.95
Recuperator effectiveness	0.8	0.85	0.90

Table 3

Optimised parameters for the studied cycle configurations.

Parameter		Technology level		
		Low	Average	High
Saturator bypass ratio	$\dot{m}_{g,AC}/\dot{m}_{g,HPC}$	0.46	0.41	0.53
Relative low pressure ratio	$\frac{\log PR_{LPC}}{\log PR_{HPC}}$	0.57	0.53	0.53
Intercooler's heat capacity ratio	C_g/C_w	0.95	0.95	0.95
Aftercooler's heat capacity ratio	C_g/C_w	0.95	0.95	0.95
Economiser's heat capacity ratio	C_g/C_w	0.88	0.88	0.82

2.2. Cost analysis method

The economic analysis of the system relies on the Purchase Equipment Cost (PEC) as an economic metric to define the acquisition price. This comprises the cost of the gas generator including the power turbine ($\Xi_{bare\ GT}$), the packaging ($\Xi_{packaging}$), the electric generator (Ξ_{gen}), the saturator tower (Ξ_{SAT}), and the heat exchangers ($\sum \Xi_{HX}$), as shown in Eq. (1). The total price of the bare gas turbine includes the compressors, the turbines, the combustion chambers, and the ancillaries, as shown in Eq. (2). The acquisition cost of the turbomachinery and the electric generator are estimated using the correlations presented by Traverso et al. [33]. The cost of the compressors (Ξ_{comp}) and turbines (Ξ_{turb}) is based on the inlet mass flow, pressure ratio, inlet temperature and the assumed polytropic efficiency. The cost of the combustion (Ξ_{comb}) chamber is based on the outlet temperature, the temperature raise, and the relative pressure losses. The cost of the generator (Ξ_{gen}) is correlated with the power generated. The cost of the ancillaries ($\Xi_{ancillaries}$) is assumed to be 40% of the turbomachinery cost, and the cost of the packaging is estimated as 40% the cost of the previous two, as suggested by Kavanagh [10].

$$PEC = \Xi_{bare\ GT} + \Xi_{packaging} + \Xi_{gen} + \Xi_{SAT} + \sum \Xi_{HX} \quad (1)$$

$$\Xi_{bare\ GT} = \sum \Xi_{comp} + \sum \Xi_{turb} + \sum \Xi_{comb} + \Xi_{ancillaries} \quad (2)$$

The acquisition cost of the saturator is estimated based on the weight of the shell (8800 USD/ton) and the packing volume (3800 \$/m³), assuming that both are made in stainless steel as recommended by Lindquist in [34]. To account for the cost of the droplet eliminator, water sprays, and the rest of the subcomponents an additional \$14,000 is added on top of the baseline cost [34]. The main dimensions of the saturator are calculated using the model presented by Brighenti et al. [19] (also in Appendix A of this paper) for a structured packing-bed configuration.

The intercooler, the aftercooler, and the recuperator are plate-fin counter-current cross-flow units, whereas the economiser is designed as a finned-tube counter-current cross-flow unit to avoid passage blocking as a consequence of water condensation, according to the configuration presented by Brighenti et al. [19]. The acquisition cost of the heat exchanger is estimated based on the heat transfer area using the correlations presented by the ESDU [35] for the plate-fin units, and by Casarosa [36] for the finned-tube units. The total heat transfer area of the heat exchangers is calculated using the effectiveness-Number of Transfer Units method developed by Kays and London [37]. Unit costs

Table 4

Economic assumptions for the calculation of the total cost of investment [40].

Total cost of investment	Fixed capital investment	Direct costs	Installation	20% of PEC
			Piping	10% of PEC
			Instruments and control systems	6% of PEC
			Electric equipment and materials	10% of PEC
			Land	5% of PEC
		Indirect costs	Civil, structural, and architectural work	15% of PEC
			Service facilities	30% of PEC
			Engineering and supervision	25% of PEC
			Construction	15% of DC
			Contingency	8% of all the above
	Other outlays	Startup	8% of FCI	
		Working capital	15% of FCI	
		Licensing, and research and development	7% of FCI	

known from previous years are updated to 2015 prices using the Chemical Engineering Plant Cost Index (CEPCI) as described in [38]. The detailed heat exchanger geometry is specified using the methodology previously presented by Brighenti et al. [19].

In the absence of any detailed data about the infrastructure required, the Total Cost of Investment (TCI) is estimated based on the Purchasing Equipment Cost (PEC), the Direct Costs (DC), and the Fixed Capital Investment (FCI), as suggested by Bejan et al. [39]. The assumed values of these costs are based on the work published by Barberis et al. [40] and are represented in Table 4.

For a lifecycle economic analysis, the annual expenses during the plant economic life are required. These are estimated using the model proposed by Bejan et al. [39] which accounts for the payment of the Total Cost of Investment (TCI), the interest generated by the required loan (cost of financing), the taxes, the cost of fuel, and the Operation and Maintenance (O&M) costs to estimate the annual expenses (C_{AEi}) as shown in Eq. (3). The TCI accounts for the plant's sub-system's costs including installation costs, the land acquisition costs, the construction of the access roads, etc. The loan's interests are derived from the credit required to pay off the TCI. In this study the loan is divided in three parts, a part paid with a common equity, other with debt, and the other with preferred stock. The fuel costs include the purchase of fuel required to operate the power plant, which is based on the power and efficiency of the plant. Finally, the O&M costs include the maintime required for normal operation of the power plant as well as the maintenance costs over the lifetime of the system including the costs for replacement parts. In this study, the O&M costs include a fixed part proportional to the PEC and a part proportional to the fuel used. Full details about the cost model can be seen in [39].

The assumed values to enable the cost study are presented in Table 5. The analysis is conducted in current dollars accounting for

inflation and the assumptions made to support it rely on the economy of a developed country. A similar study using economic figures of an emerged economy might be able to identify the potential of such an advanced cycle system as part of the energy market in a developing country where high efficiency plants are also of critical importance.

The Cost of Electricity (COE) is calculated using Eq. (4). The COE_i represents the total cost of generating a kilowatt hour at year i and reflects the minimum sale price to recover the annual expenses. The COE varies during the book life of the power plant following the annual cost variation. As such, the average cost of electricity, $\overline{COE}$, for the book life is used as an economic index and is calculated as shown in Eq. (5). Average cost of electricity represents the total average cost of generating one kilowatt hour accounting for the annual expenses along the whole book life.

$$\Xi_{AEi} = \Xi_{ini} + \Xi_{fini} + \Xi_{taxi} + \Xi_{fueli} + \Xi_{O\&Mi} \quad (3)$$

$$COE_i = \frac{\Xi_{AEi}}{t\dot{W}} \quad (4)$$

$$\overline{COE} = \frac{\sum_{i=1}^n C_{AEi}}{nt\dot{W}} \quad (5)$$

3. Results and discussion

3.1. Effect of heat exchanger technology

The effect of the heat exchanger effectiveness on the thermal efficiency and the economic metrics of the power plant is presented in Fig. 2 for the three scenarios shown in Table 2. In this study the Specific Purchase Equipment Cost ($SPEC$) and the average cost of electricity ($\overline{COE}$) of the average technology scenario (heat exchanger effectiveness at 0.85 for all units, $SPEC = 334.06$ \$/kW, $\overline{COE} = 6.81$ c\$/kW h) have been used as reference to calculate relative changes. Fig. 2 shows that a

Table 5

Assumptions for economic analysis.

Assumed starting year of the project	January 2015
Time until commissioning starts	1 year
Assumed plant commissioning time	2 years
Plant economic life	20 years
Plant life for tax purposes	10 years
Income tax rate	30%
Other taxes and insurances	2% of PFI
Real inflation	2.5%
Real inflation for the fuel	3.0%
Fraction funded by common equity	35%
Fraction funded by preferred stock	15%
Fraction funded by debt	50%
Interest rate of the common equity	6.5%
Interest rate of the preferred stock	6.0%
Interest rate of the debt	5.5%
Fuel price (LNG)	0.25 \$/kg
O&M variable	2% of FCI
O&M fixed	0.83 \$/kg of fuel
Availability	85%

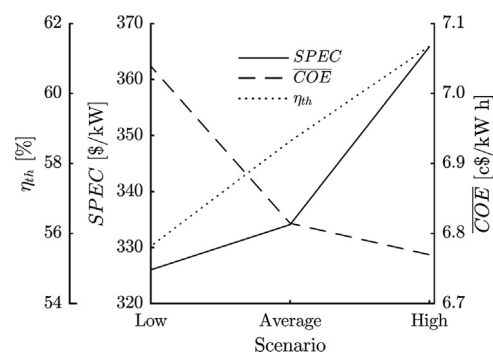


Fig. 2. Impact of the technology level of the heat exchangers on the $\overline{COE}$, $SPEC$, and η_{th} .

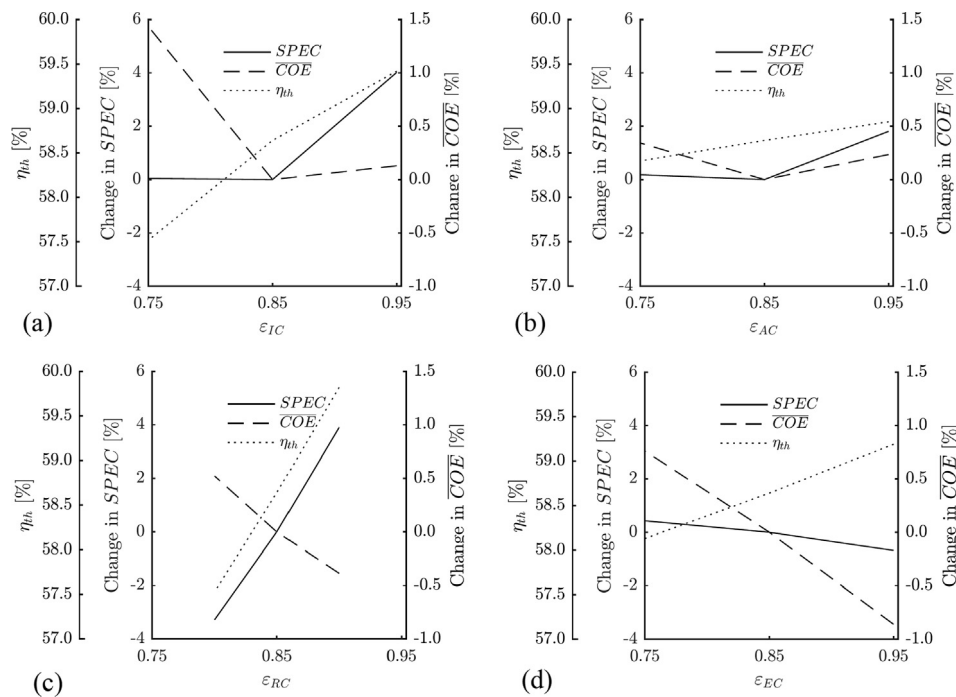


Fig. 3. (a) Effect of the intercooler, (b) effect of the aftercooler, (c) effect of the recuperator, (d) effect of the economiser effectiveness on η_{th} on SPEC and on $\overline{COE}$.

reduction in the heat exchanger effectiveness from the average to the low technology level is associated with a 2.3% reduction in the Specific Purchasing Equipment Cost (SPEC), which when combined with the 3 percentage points (pp) loss in thermal efficiency yields to a 3% increase of the $\overline{COE}$. Improving the heat exchanger technology from the average to the high level scenario (see Table 2) yields to a more aggressive change in SPEC, which increases by 9.5% in this case. This increase combined with the improvement in thermal efficiency yields to a 1% reduction of the $\overline{COE}$ relative to its baseline value.

The impact of the effectiveness of each heat exchanger on the thermal efficiency, SPEC, and $\overline{COE}$ is shown in Fig. 3. The effectiveness of each heat exchanger varies across its specified design range, in order to identify its relative contribution to the overall economic metrics of the system, while the rest of the design variables are kept constant.

Fig. 3a shows that the reduction of the intercooler effectiveness from 0.85 to 0.75 causes an increase in the average cost of electricity by 1.46%. Although this reduction doesn't affect the purchasing costs of the unit, a notable penalty on its thermal efficiency is imposed, which drops by approximately 1 pp from 58.6%. Poor air humidification and the higher required compression work promoted by the less effective intercooling reduces the specific power of the plant causing an increase in size. Therefore, the lower acquisition cost of the less effective intercooler is compensated by the higher price of the rest of the units which keeps the SPEC fairly constant. An intercooler effectiveness of 0.95 drives the system to more efficient but also to higher SPEC configurations, which effectively cancel each other out leaving the average cost of electricity almost unaffected. The aftercooler shows broadly a similar behaviour as the intercooler (see Fig. 3b). However, the changes in SPEC and thermal efficiency triggered by the increase in effectiveness are found to have a secondary effect on the predicted $\overline{COE}$. A 5 pp reduction in recuperator effectiveness (0.8 from 0.85, Fig. 3c) entails a 0.5% increase in the predicted $\overline{COE}$ due to the associated penalties in thermal efficiency (−1.34 pp) which are not compensated by the almost 4% reduction in SPEC. The driver behind the thermal efficiency

Table 6

Heat exchanger effectiveness for minimum average cost of electricity ($\overline{COE}$).

ε_{IC}	[–]	0.85
ε_{AC}	[–]	0.85
ε_{RC}	[–]	0.90
ε_{EC}	[–]	0.95
η_{th}	[%]	60.33
SPEC	[\$/kW]	345.33
$\overline{COE}$	[c\$/kW h]	6.74

deterioration is the reduced capacity of the system to exploit the waste heat. For a 5 pp increase in recuperator effectiveness (0.9 from 0.85), the thermal efficiency gain (+1.18 pp) out-competes the 4% increase in SPEC yielding to 0.5% reduction in the $\overline{COE}$. Finally, the economiser is found to also drive the cost of electricity across a $\pm 1\%$ zone for effectiveness values of 0.75 and 0.95 respectively. This change is primarily dominated by the increase in thermal efficiency of the system produced by a more effective unit.

In conclusion, the analysis shows that the recuperator and the economiser primarily drive the cost metrics of the power plant, while the intercooler has a big impact on cost only for low effectiveness values. The parametric analysis enables the identification of the most cost-efficient system configuration. This features an intercooler and aftercooler effectiveness of 0.85, economiser effectiveness 0.95, and recuperator effectiveness 0.90 (see Table 6). The identified values of heat exchanger effectiveness that produce the cost optimum configuration balance the capacity of the cycle to recuperate heat directly against its capacity to exploit un-recuperated waste heat to raise humidity.

3.2. Uncertainty analysis

The total annual cost of the power plant operation depends on four key parameters: (i) fuel costs, (ii) purchase equipment costs (PEC), (iii)

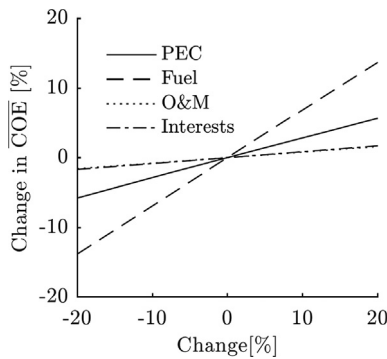


Fig. 4. $\overline{COE}$ sensitivities in cost drivers for the most cost efficient configuration.

interest rates over the loan period to pay off the initial investment, and (iv) operation and maintenance (O&M) costs. The nature of these cost elements is associated with an implicit uncertainty. Fuel prices may undergo severe fluctuations over time following global financial trends. Purchase cost estimates rely on correlations derived statistically from public domain data with an implicit standard deviation. The assumed interest rates may change significantly depending on the risk of the economic project, which is declared by the investor and heavily influenced by the current financial conditions. Finally, the O&M costs can vary across a wide range depending on the technology level of the equipment installed, the location of the power plant, operation mode, ambient conditions as well as other unforeseen circumstances which need to be accounted for as contingency. According to the above considerations, an imposed variation, across a range between -20% to $+20\%$, is applied to each of the four parameters over the lifetime of the plant aiming to assess the sensitivity of the average cost of electricity on these cost drivers. The selected uncertainty range is based on the expected error ranges derived from the calculation methods shown by Traverso et al. [33] and Bejan et al. [39] which justify that such a range is representative for the type of cost studies presented herein. The cycle configuration shown in Table 6 is used as reference to show the variations of the $\overline{COE}$. The reference values of interest rates, baseline fuel cost, and the O&M costs are the presented in Table 5, whereas the baseline purchase equipment cost (PEC) of \$13,813,200 is derived from the SPEC provided in Table 6.

Fig. 4 shows the impact of the four parameters analysed on the $\overline{COE}$. The fuel cost is found to have the strongest influence in the $\overline{COE}$, showing a change of 0.75% per percentage point fuel cost variation. The PEC uncertainty causes small changes in the $\overline{COE}$, of the order of 0.25%, for each PEC percentage point variation. Variations in the O&M costs or the interest rates of the debts have minor effect, producing changes of only 0.05% per percentage point change (note that O&M and Interests lines are collapsing on top of each other in Fig. 4). Therefore, both lines appear overlapped in Fig. 4. The analysis reveals a linear correlation between any of the four cost drivers and the $\overline{COE}$. Therefore, such a study enables the impact of larger fluctuations of any of the four cost drivers by extrapolation using the data presented herein as reference.

The tile-plot shown in Fig. 5 identifies the exchange rates between all cost drivers across the design space. On each plot, the x-axis represents the percentage variation of the interest rates, and the y-axis the percentage variation of the O&M costs from the reference values shown in Table 5. Fuel cost variation from nominal values is shown in the major x-axis while uncertainty in PEC is shown in the major y-axis. The contour lines represent the percentage $\overline{COE}$ departure from the reference value (Table 6). Point A in Fig. 5 represents a pessimistic

scenario whereby the variation of the four parameters is $+20\%$ with regards to their reference values. In that case, the $\overline{COE}$ is estimated to be around 23% higher with respect to the baseline case (point REF in Fig. 5). An optimistic scenario is represented by point B (bottom left corner in Fig. 5), where the variation for the four parameters is -20% in relation to their baseline values. This scenario shows a $\overline{COE}$ reduced by 22% compared to the baseline case (point Ref. in Fig. 5). In addition, as the PEC uncertainty decreases, the rate of change of the $\overline{COE}$ produced by the interest rates and the O&M cost variations diminishes as these two parameters show strong dependence on purchasing costs. Finally, from Fig. 5 it becomes clear that increased relative fuel costs dictate a cost optimum system configuration with high technology level inter-cooler and recuperator (light grey regime $\varepsilon_{IC} = 0.95$, $\varepsilon_{RC} = 0.90$), which better exploits the waste heat within the power cycle. When the anticipated relative fuel costs drop, a low technology level system seems to also be viable ($\varepsilon_{IC} = 0.85$, $\varepsilon_{RC} = 0.85$, dark grey regime in Fig. 5), which combines low operating costs with low purchasing costs due to the relatively more compact size of these heat exchangers.

3.3. Comparison against other advanced cycles

In Table 7 the RHAT configuration shown in Table 6 is compared against the cost metrics of humid cycles previously reported by Kavanagh and Parks [10], and Traverso and Massardo [12]. A CCGT cycle [10] is also included for reference. The cost metrics of the current RHAT system are re-estimated under the assumptions of previous studies (Kavanagh et al. [10], Traverso and Massardo [12]) to enable comparison.

Table 7 shows that the RHAT system features 62% lower predicted cost of investment than an equivalent CCGT. As a result, the average cost of electricity ($\overline{COE}$) is reduced by 28%, also due to the RHAT's 14% higher thermal efficiency. Comparing against previous humid cycles, it is observed that the RHAT cycle features a 15% and roughly 8.5% lower SPEC than STIG and simpler HAT cycles respectively (Kavanagh and Parks [10]). The lower SPEC is a consequence of the higher specific power achieved by the RHAT (130% compared with the STIG and 38% compared with the HAT), as it reaches higher humidification ratios. As such, despite the additional combustion chamber and the requirement for advanced materials, the reduction in size of all the components, as a consequence of the higher specific power, enables a cost reduction that out-competes the above mentioned drawbacks. Nevertheless this reduction is not sufficiently large to achieve a lower SPEC than the TOP Humid Air Turbine [10], which does not require an intercooler, an aftercooler, or a humidifier. From Table 7 is also observed that the RHAT cycle out-competes the rest of the cycles in terms of thermal efficiency due to its increased capability to exploit waste heat and introduce humidity into the cycle. Finally, the enhanced thermal efficiency and the low purchasing costs of the RHAT system yield to a notably lower estimated average cost of electricity against the competitor systems with the $\overline{COE}$ reductions ranging from -28% against the CCGT plant [10] to roughly -2% against the humid air water injected system shown in [12].

4. Conclusions

In this paper, a techno-economic analysis of a reheated humid air turbine system is presented. Lifecycle cost analysis showed that the average cost of electricity is primarily driven by the effectiveness of the recuperator and economiser. These heat exchanger units represent the key components that manage the waste heat recuperation and humidity levels while they also dictate the size, hence the acquisition cost, of the power plant. Increasing the technology level of the recuperator and the

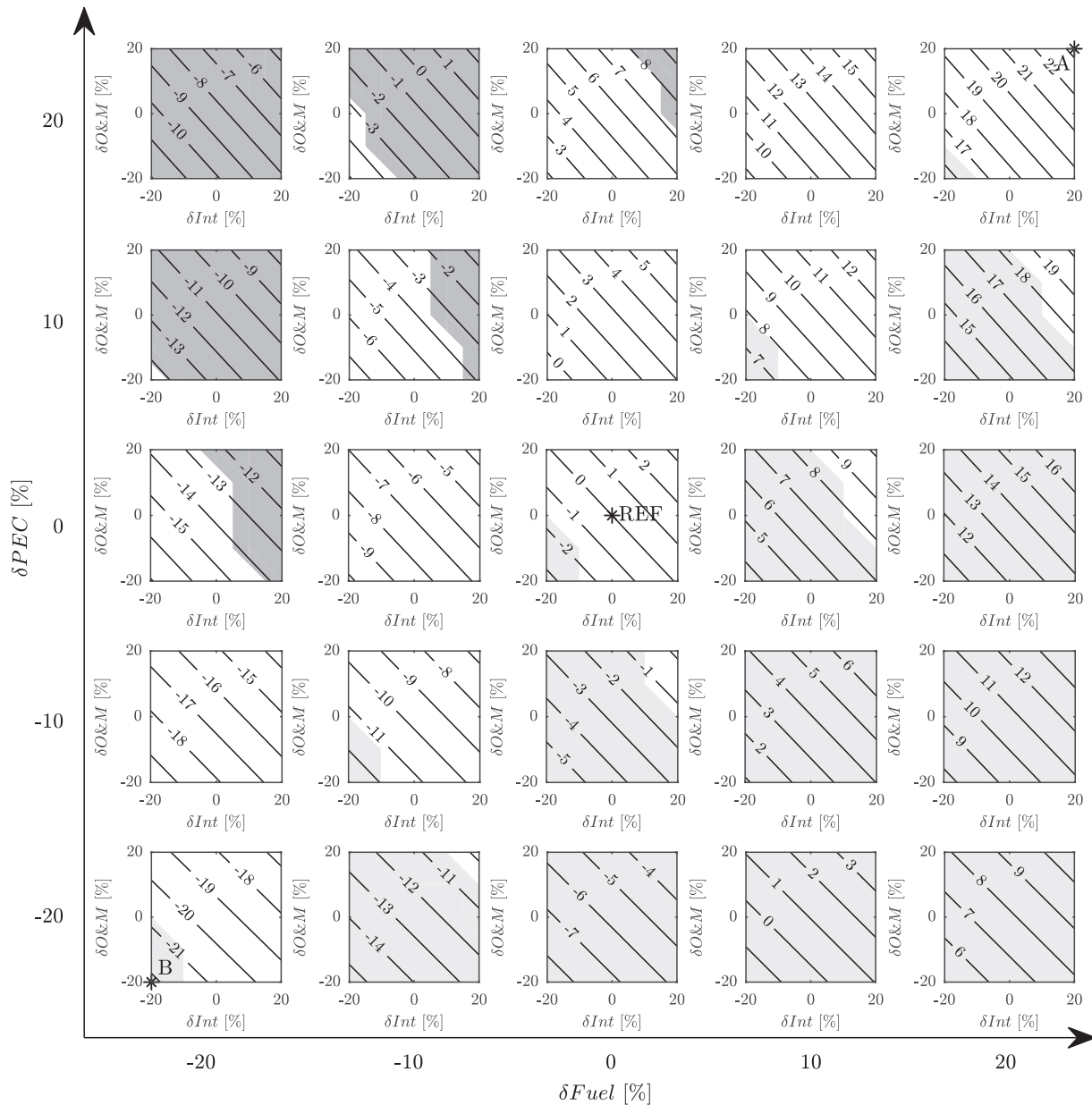


Fig. 5. Percentage mean cost of electricity variation from reference cycle. Cost optimum configuration in dark grey region $\varepsilon_{IC} = 0.85$, $\varepsilon_{AC} = 0.85$, $\varepsilon_{RC} = 0.85$, $\varepsilon_{EC} = 0.95$. Cost optimum configuration in blank region $\varepsilon_{IC} = 0.85$, $\varepsilon_{AC} = 0.85$, $\varepsilon_{RC} = 0.90$, $\varepsilon_{EC} = 0.95$. Cost optimum configuration in light grey region $\varepsilon_{IC} = 0.95$, $\varepsilon_{AC} = 0.85$, $\varepsilon_{RC} = 0.90$, $\varepsilon_{EC} = 0.95$.

Table 7

Comparison of reference reheated humid air turbine (Table 6) performance against previously reported advanced cycles.

Cycle	η_{th} [%]	SPEC [\$/kW]		$\overline{COE}$ [c\$/kW h]	
		RHAT	RHAT	RHAT	RHAT
Steam injected cycle [10]	49.36	+22.2%	323	−14.9%	6.85
Gas-steam combined cycle [10]	53.00	+13.8%	720	−61.8%	7.00
Humid Air Turbine [10]	52.26	+15.4%	300	−8.3%	6.48
TOP Humid Air Turbine [10]	54.12	+11.5%	187	+47.0%	6.00
Humid Air Turbine [12]	51.74	+16.6%	n/a	n/a	4.21
Humid Air Water Injected Turbine [12]	50.04	+20.6%	n/a	n/a	4.07

economiser by 5 and 10 percentage units respectively yields to a reduction in the cost of the electricity of about 0.5% and 1% respectively.

The uncertainty study showed that possible variations in fuel costs and the uncertainty of the purchased equipment cost are the main

parameters driving the fluctuations of the average cost of the electricity. The cost of electricity increases by 0.75% per percentage point of fuel price increase, and by 0.25% per percentage point rise in purchase equipment cost. Comparisons against previously reported studies

on combined and humid cycles show that the reheated humid air turbine features 61.8% lower purchasing costs than a typical CCGT. Moreover, due to its high efficiency (approximately 14% higher than a typical combined cycle), estimated average costs of electricity are approximately 30% lower than these of CCGT system.

Overall a detailed economic analysis of a reheated humid air turbine power plant intended for power generation has been performed in order to explore the potential of this configuration for power generation applications. The impact of the heat exchanger technology on the economic metrics was demonstrated, revealing their influence on the acquisition cost and in the $\overline{COE}$, which was not previously known. The outcome of the work constitutes a step forward in the understanding of the economic performance of advanced complex cycles and proves the

potential of such systems for applications where high efficiency and economic performance is jointly required.

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Appendix A. Saturator tower performance calculation

The model for simulating the design performance of the saturator tower, previously presented by Brighenti et al. [19], relies on the inlet pressure, temperature, and mass flow of the two streams (dry air and water) and the pinch point temperature difference (ΔT_{ps}). The pinch point temperature difference is defined as the minimum temperature difference between the operating line of the water and the saturated air enthalpy line across the saturator [41]. For this modelling process, the air–vapour mixture is treated as an ideal mixture of ideal gases, while the air at the outlet of the saturator is assumed to be fully saturated [22]. From the conservation of the dry air and water mass as well as conservation of energy two invariants of the system (M^+ , H^+) can be defined as:

$$\frac{\dot{m}_w}{\dot{m}_a} - \omega = M^+ \quad (6)$$

$$h_a @ (T_g) + \omega [h_v @ (T_g) - h_w @ (T_w)] - M^+ h_w (T_w) = H^+ \quad (7)$$

Under the assumption of an ideal mixture of ideal gas, absolute humidity is defined as:

$$\omega = \frac{\phi p_{sat} @ (T_g) R_a}{p - \phi p_{sat} @ (T_g) R_v} \quad (8)$$

$\dot{m}$ represents the mass flows of the two streams, M^+ the mass invariant, $h_x @ (T)$ the specific enthalpy evaluated at temperature T , p the pressure, ω the water to air ratio (defined as the amount of vapour in the air divided by the dry air $\dot{m}_v / \dot{m}_a$), H^+ the enthalpy invariant. $p_{sat} @ (T_g)$ indicates the saturation pressure evaluated at the temperature of the gas. ϕ represents the relative humidity, R_x is the specific gas constant of each fluid, and the subscripts a , g , w , and v stand for dry air, humid air, water, and vapour respectively.

For the calculation of the saturated gas thermodynamic properties, the above-defined system of equations is solved at three sections along the saturator gas path. “Station 0” corresponds to the pinch point condition between the operating line and the saturated enthalpy line of the air. The temperature difference between the two above-mentioned lines is defined by ΔT_{ps} . Therefore, it is possible to approximate the water temperature as $T_w = \Delta T_{ps} + T_g$. “Station 1” corresponds to the bottom section of the tower, where the air inlet conditions are known. Last, “Station 2” corresponds to the top section of the saturator, where the water inlet conditions are known and the outlet air is assumed to be saturated. The equations Eq. (6)–(8) are evaluated and solved at “Station 0” and “Station 2”, which enables the calculation of the two invariants (M^+ , H^+) and the conditions of the outlet gas. Once the invariants are known, the system of equations is evaluated at “Station 1”, which corresponds to the bottom section of the saturator, to calculate the conditions of the outlet water. Finally, the air outlet pressure is calculated assuming a 5% pressure loss of the inlet total pressure.

The saturator performance model was previously validated against experimental data obtained by Lindquist et al. [31] using the Lund HAT cycle demonstrator, as shown in Table 8.

Table 8
Validation of the saturator model against experimental data from Lindquist et al. [31].

Inlet/Input conditions			Outlet conditions				
Parameter	Units	Data	Parameter	Units	Model	Experimental data	Discrepancy (%)
$\dot{m}_{g,in}$	(kg/s)	2.17	$\dot{m}_{g,out}$	(kg/s)	2.5685	2.55	0.7
$T_{g,in}$	(K)	346.75	$T_{g,out}$	(K)	389.97	389.15	0.2
p	(bar)	7.88	ω_{out}	(–)	0.1837	–	–
$\dot{m}_{w,in}$	(kg/s)	3.48	$\dot{m}_{w,out}$	(kg/s)	3.0815	3.10	0.6
$T_{w,in}$	(K)	419.35	$T_{w,out}$	(K)	347.52	352.85	1.5
ΔT_{ps}	(K)	10	–	–	–	–	–

The sizing model of the tower enables the estimation of the height and diameter of the saturator’s packing. The diameter is calculated following the correlation presented by Coulson and Richardson [42]. The pressure losses are set to 300 Pa/m and the maximum velocity is 60% of the flooding velocity to ensure stable operation and avoid water over-flood. The packing height is estimated based on a variation of the method previously proposed by Aramayo-Prudencio and Young [43]. This model enables the estimation of the saturator’s height from a given performance. The required inputs are: the characteristic dimensions of the packing-bed, the previously calculated diameter, an initial guess for the height, and the inlet conditions of both streams as inputs. The model iteratively calculates the humidifier’s performance based on the data provided and the initial guessed height, and compares the results with the humidifier thermodynamic data obtained from the cycle design. The packing height is determined

once the above-mentioned iteration reaches convergence.

Appendix B. Heat exchanger thermodynamic model

The outlet conditions from all air-water heat exchangers are calculated using the optimised heat capacity ratio (C^*), defined in Eq. (9), and the imposed effectiveness (ε), defined Eq. (10), where $\dot{Q}$ represents the heat rate. Gas mass flows are defined as part of the cycle analysis, whereas the heat exchanger water mass flows are calculated using the heat capacity ratio of the air and water streams, C^* , under the assumption that the water always presents the higher heat capacity ($C_w = C_{\max}$). As water condensation in the air stream is possible ($T_{g,out} < T_{g,dew}$), the latent heat of condensation of the water in the humid air causes variations in the heat capacity of the gas during the condensation process and generates a pinch point between the two streams within the heat exchanger (see Fig. 6).

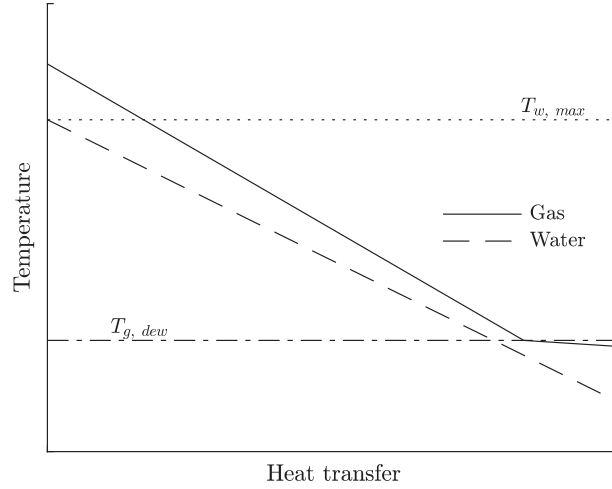


Fig. 6. Heat transfer process in air-water heat exchangers.

$$C^* = \frac{C_g}{C_w} \quad (9)$$

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} \quad (10)$$

In order to calculate the heat exchanger outlet conditions and iterative scheme is used. The procedure begins by guessing the water mass flow. Then the maximum heat transferred ($\dot{Q}_{\max}$) is calculated as the minimum between Eq. (11) and Eq. (12). The subscripts *in* and *out* stand for inlet and outlet conditions respectively, $h(T)$ is the specific enthalpy evaluated at temperature T and h_{fg} is the specific enthalpy of evaporation of the water.

$$\dot{Q}_{\max} = \dot{m}_{g,in} h_g(T_{g,in}) - [\dot{m}_{g,out} h_g(T_{w,in}) + (\dot{m}_{g,in} - \dot{m}_{g,out}) h_{fg}] \quad (11)$$

$$\dot{Q}_{\max} = \dot{m}_w h_w(T_{g,in}) - \dot{m}_w h_w(T_{w,in}) \quad (12)$$

Subsequently, the outlet gas and water temperatures are obtained from Eq. (13) and Eq. (14) respectively, where $\dot{Q}$ is obtained from Eq. (10).

$$\dot{Q} = \dot{m}_{g,in} h_g(T_{g,in}) - [\dot{m}_{g,out} h_g(T_{g,out}) + (\dot{m}_{g,in} - \dot{m}_{g,out}) h_{fg}] \quad (13)$$

$$\dot{Q} = \dot{m}_w h_w(T_{w,out}) - \dot{m}_w h_w(T_{w,in}) \quad (14)$$

The amount of condensed vapour is obtained from Eq. (15). $\omega_{sat}(T_{g,out}, p_g)$ is the saturated water to air ratio evaluated at the outlet temperature of the gas and at the pressure of the gas. When $\omega_{sat} > \omega_{in}$ no condensation is taking place and the amount of water condensed is to zero.

$$\dot{m}_{g,in} - \dot{m}_{g,out} = \dot{m}_a (\omega_{in} - \omega_{out}) = \dot{m}_a [\omega_{in} - \omega_{sat}(T_{g,out}, p_g)] \quad (15)$$

Once the outlet temperature of both flows is obtained, the heat capacity ratio of the air and water side is calculated from Eq. (16). $\Delta T_{x,dry}$ represents the temperature jump of the stream in the dry section, that is from the inlet of the gas up to the dew point. When no condensation occurs, $\Delta T_{x,dry}$ represents the temperature jump across the whole heat exchanger.

$$C_x = \frac{\dot{Q}}{\Delta T_{x,dry}} \quad (16)$$

Finally, a revised heat capacity ratio C^* is re-calculated using the values obtained from Eq. (16). In case that the calculated and the initial design values do not match, the water mass flow guess is revised and the process is repeated until convergence is reached.

The water is injected into the saturator at a liquid phase to facilitate the humidification process. The outlet temperature of the water is limited by the water saturation temperature evaluated at the saturator's pressure. A safety coefficient of 0.9 is introduced to avoid any steaming in the saturator, as shown in Eq. (17), where $T_{sat}(0.9p_{SAT})$ is the saturation temperature of the water evaluated at 0.9 the saturator's pressure. In case the condition was not satisfied, the water mass flow would be increased, a new heat capacity ratio is calculated which overwrites the initial value.

$$T_{w,out} \leq T_{w,max} = T_{sat} @ (0.9p_{SAT}) \quad (17)$$

In the recuperator, the inlet properties of both streams are defined. Therefore, the outlet conditions of the two streams are calculated with the definition of the effectiveness of the heat exchanger. The pressure losses of the heat exchangers are calculated analogously to the method used in the saturator since the pressure losses are imposed as a percentage on the inlet pressure.

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