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Off-design Simulations of Offshore Combined Cycles

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MASTER THESIS

for

Student Øystein Flatebø

Spring 2012

Off-design simulations of offshore combined cycles

Off-designsimuleringer av kombinerte kraftprosesser på offshoreinstallasjoner

Background and objective

Offshore installations are focused on the oil and gas operations and less on the power plant necessary to supply electricity, heat, or mechanical drive to the processes. In a land based power plant the focus is usually on operating the plant as efficient as possible for the given power demand. Offshore, the power plant has to adjust to whatever is needed for the oil and gas operations. Also, backup capacity if often wanted to ensure continuous operation even if one unit, like a gas or a steam turbine, falls off.

All this often leads to operation at off-design conditions. That is, operation away from the design point of the plant.

The objective of the thesis is to design and characterize different combined cycle configurations. The characterization should be with respect to part load performance, changes in ambient conditions, and defining an operating window for the combined cycle plant. A comparison between different technologies and configurations should be conducted to find out how different a compact, low-efficiency design is compared to a more complex high-efficiency design.

The tasks for the Master thesis include:

1) Design of one configuration for an offshore combined cycle. The focus should be directed towards a new installation rather than a retrofit solution.
2) Design a more complex high-efficient combined cycle, which should be compared to the offshore combined cycle design.
3) Off-design simulations for different scenarios. This would include:
   a) Part load operation of gas turbines
   b) Operation at a range of different ambient conditions
   c) Off-design operation of steam cycle
4) The simulations should preferably be carried out in the GTPRO and GTMASTER simulation tools.

The focus of the power plant should be to supply the required power to the offshore installation and not on the plant efficiency. Also, the selected designs should be compact due to the restrictions and high cost of weight and volume on offshore installations.

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

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Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report.

Pursuant to "Regulations concerning the supplementary provisions to the technology study program/Master of Science" at NTNU §20, the Department reserves the permission to utilize all the results and data for teaching and research purposes as well as in future publications.

The final report is to be submitted digitally in DAIM. An executive summary of the thesis including title, student's name, supervisor's name, year, department name, and NTNU's logo and name, shall be submitted to the department as a separate pdf file. Based on an agreement with the supervisor, the final report and other material and documents may be given to the supervisor in digital format.

Department for Energy and Process Engineering, February 1, 2012.

L. Bakken.

Lars E Bakken
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Preface

This is a master thesis written spring 2012 at NTNU, Department of Energy and Process Engineering. I would like to thank my two supervisors Olav Bolland (Professor, NTNU) and Lars O. Nord (Postdoc.) for guidelines and given thoughts.

Trondheim, July 1st 2012

Øystein Flatebø

Abstract

This thesis presents an off-design simulation of offshore combined cycles. Offshore installations have a substantial power demand to facilitate the oil and gas production. To cover this need of power almost all the platforms use one or several gas turbines, often described as a simple cycle. However, because of high taxes on emissions, and increasing gas prices, more efficient technologies have been reviewed. One solution has been installing combined cycles (CC) offshore. Between 1999 and 2000 three combined cycles were installed on the Norwegian continental shelf and are still in operation. A combined plant may operate for prolonged time at off-design conditions, depending on power demand, ambient condition offshore.

First, this thesis gives a description of combined cycles from a thermodynamic and technical point of view. A study of existing offshore combined cycles is performed, and some of the implications of using combined cycles offshore are discussed. In the study, also off-design performance regarding the gas turbine and steam cycle is presented.

Further, the simulation tool GTPRO is used to model two CC plants, one designed for offshore installations, and one designed to achieve high efficiency. As part of the design process a sensitivity analysis is performed to find a good trade-off between efficiency and weight for the offshore plant. The model showed good agreements compared with the existing offshore plants, with a power output of 50.3MW, plant efficiency of 50.3%, and similar weight of the skids. The high efficient plant, based on the same gas turbine, and the same assumptions produced 53.1MW. This model gained 2.4MW more in power output, however with a penalty of 209 ton in extra weight.

To review the plants performance and operability, off-design simulations were performed in GTMASTER. Both part load and changing ambient temperature were investigated. The results showed that both plants had similar behavior in performance at off-design, and that the GT strongly dictates the behavior of the steam cycle. At part load the relative SC efficiency increases, resulting in general high plant efficiency. At 60% GT load, the relative gas turbine efficiency is 81% compared to the relative plant efficiencies of about 90%. The difference in efficiency between the high efficient plant and the offshore plant remains constant at part load.

The result from the simulations of ambient temperature is that none of plants will achieve higher plant gross efficiency at changing ambient temperature. The best plant efficiency occurs at design point. However, both plants have a long interval with approximately 100 % plant efficiency. From 15 to 0°C, the relative SC gross efficiency drops with 5 %, and the relative GT efficiency increase with 2%. However, the power output changes for both the GT and ST. From 28°C to about 0°C the power output increase almost linearly for the SC and GT.

Sammendrag

Denne masteroppgaven omhandler off-design simulering av offshore kombinerte kraftanlegg. Offshoreinstallasjoner har et betydelig et energibehov til produksjon av olje-og gass. For å dekke dette behovet har nesten alle plattformene en eller flere gassturbiner, ofte beskrevet som simple cycles. På grunn av høye skatter på utslipp og økende gasspriser, har mer effektiv teknologi blitt interessant. En løsning som har blitt vurdert er installasjon av kombinerte kraftanlegg offshore. Mellom 1999 og 2000 ble tre kombinerte kraftanlegg installert på norsk sokkel. Disse er fortsatt i drift. Et kombinert anlegg vil med stor sansynlighet operere under off-design i en lengre periode, avhengig av kraftbehovet, omgivelesene offshore.

I denne avhandlingen vil kombinerte anlegg først bli beskrevet ut i fra termodynamikk og et teknisk synspunkt. Et studie av eksisterende offshore kombinerte anlegg er utført, og noen av konsekvensene av å bruke kombinerte anlegg offshore blir diskutert. I avhandlingen blir også teori knyttet til off-design av gassturbin og damp syklus presentert.

Videre er simuleringsverktøy GTPRO brukes til å modellere to CC anlegg, ett designet for offshore installasjoner, og er designet for å oppnå høy virkningsgrad. Som en del av designprosessen for offshore anlegget, er en sensitivitetsanalyse er utført for å finne en god avveining mellom effektivitet og vekt. Modellen viste gode resultater sammenlignet med de eksisterende offshore anleggene. Kraft produksjonen var 50.3MW, den total effektiviteten var 50.3%, og vekten tilsvarte den for offshore anleggene. Det anlegget med høye effektivitet, produserte 53.1MW, og var basert på den samme gassturbinen, og de samme forutsetningene. Denne modellen fikk 2,4 MW mer i effekt, men samtidig en vektøkning på 209 tonn.

I GTMASTER ble simuleringer utført for å beskrive ytelsen og karakterestikken til anleggene. Både dellast og skiftende omgivelses temperatur ble undersøkt. Resultatene viste at begge anleggene hadde veldig lik oppførsel ved off-design, og at det i hovedsak var gass turbinen som bestemte oppførselen til damp syklus. På dellaste øker den relative effektivitet til damp syklusen som medfører i en høy total effektivitet. På 60% gas turbin last, er den relative effektivitet til gass turbinen 81%, samenlignet med den relativt totale effektiviteten til anlegget på ca 90%. Forskjellen i effektivitet mellom offshore anlegget og anlegget med høy virkningsgrad forblir konstant del belastning.

Resultatet fra simuleringene av omgivelsestemperaturen viste at ingen av anleggene vil oppnå høyere total virkningsgrad ved endret omgivelses temperatur. Den beste totale effektiviteten oppnåes på design punkt. Likevel har begge anleggen et langt intervall med tilnærmet 100% effektivitet. Fra 15 til 0 ° C, vil den relativ effektivitet til damp syklusen synker med 5%, og den relative gas turbin effektiviteten øker med 2%. Likevell vil det være en endringen i effekt for både gass turbin og damp turbin. Fra 28 ° C til ca 0 ° C vil effekten øke nesten lineært for både damp turbin og gass turbin.

Acronyms and Abbreviations

CC	Combined cycle
CH_4	Methane
CHP	Combined heat and power
CO_2	Carbon dioxide
GG	Gas generator
GT	Gas turbine
HECC	High efficient combined cycle
HRSG	Heat recovery steam generator
LSP	Live stream pressure
LST	Live stream temperature
LTE	Low temperature economizer
nmVOC	non-methane volatile organic compounds
NOx	Nitrogen oxides
OCC	Offshore combined cycle
PT	Power Turbine
SC	Simple cycle
SFC	Specific fuel consumption
ST	Steam turbine
VIGV	Variable inlet guide vane
WHRU	Waste heat recovery unit

Nomenclature

η_C	Carnot efficiency	[%]
T_H	Temperature of hear addition	[K]
T_C	Temperature of heat rejection	[K]
$\dot{W}_{GT}$	Gross specific power output gas turbine	[MW/kg]
$\dot{W}_{ST}$	Gross specific power output steam turbine	[MW/kg]
$\eta_{CC,gross}$	Gross efficiency	[%]
$\dot{m}_{fuel}$	Mass flow rate of fuel	[kg/s]
LHV	Lower heating value	[KJ/kg]
$\eta_{CC,net}$	Net efficiency combined cycle	[%]
$\dot{W}_{aux}$	Specific auxiliary power	[MW/kg]
$\eta_{GT,ideal}$	Ideal GT efficiency	[%]
γ	Ratio of specific heat	[-]
p_1/p_2	Pressure ratio	[-]
T_3/T_1	Temperature ratio	[-]
$\dot{Q}_{HRSG}$	Specific heat recovery in HRSG	[MW/kg]
U	Overall heat transfer coefficient	[W/m^2 K]
A	Area	[m^2]
ΔT_{LM}	Logarithmic minimum temperature difference	[K]
$\dot{Q}_{exh}$	Specific exhaust heat	[MW/kg]
η_{GT}	Gas turbine efficiency	[%]
$\dot{m}_{steam}$	Mass flow rate of steam	[kg/s]

h	Specific enthalpy	[kJ/kg]
$\dot{m}_{exh}$	Mass flow rate exhaust	[kg/s]
$C_{p,exh}$	Specific heat capacity	[kJ/kgK]
$\bar{R}$	Specific gas constant	[kJ/kgK]
ρ_{α}	Density	[kg/m^3]
$\bar{V}$	Average swallowing capacity	[-]
n	Polytropic exponent	[-]

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1 Introduction

1.1 Background

Norway has since the beginning of the 1970s produced oil and gas, and today the petroleum sector is the largest industry in Norway. It is expected that the petroleum production will remain relatively stable over the next forthcoming years, but with a relative higher percentage of gas production [1]. The majority of offshore installations use gas turbines for mechanical drive and power generation. On the Norwegian continental shelf, there exists roughly 250 gas turbines in total [2]. The primary reasons for this dominance are due to the high power to weight ratio, reliability, and availability of fuel gas offshore. Nevertheless, a negative consequence of these turbines is low thermal efficiency and flue gas emission.

The emission from the petroleum sector is mainly flue gas from combustion of natural gas and flaring. In 2010, the petroleum sector was the biggest source of CO_2 emissions with 29% of the total CO_2 emissions in Norway. Of these emissions, offshore gas turbines accounted for 78.9% alone. Also the contribution of NOx emissions from the petroleum sector is significant, and accounted for 27.2% of the total NOx emission in 2009[1]. This illustrates the greate potential for improving emission and energy conservation within the petroleum sector.

Following the growing awareness of emissions and environmental concerns, legislative actions have been taken out on international and national levels. According to the Kyoto and Gothenburg protocols, Norway is committed to reduce emissions of carbon dioxide (CO_2), methane (CH_4), nitrogen oxides (NO_x), non-methane volatile organic compounds (mmVOC), and sulfur dioxide (SO_2). The most important emissions to air from the petroleum sector are CO_2 and NO_x [3]. As a consequence of these commitments and the increased focus on the environment, the Norwegian government has implemented tax on CO_2 and NO_x emissions[1]. This represents a significant cost for the oil and gas industry, and has led to a motivation to find novel technologies which may reduce emissions. Two approaches have been investigated, cleansing and deposition of emissions, and improved energy consumption. However, the most preferred solution so far is energy consumption and installing offshore combined cycles. Installing a combined cycle would result in higher efficiency, and thereby large savings in fuel and reduced cost of emissions. Also, alternative fuel sale would be beneficial[2].

Between 1999 and 2000 three combined cycles where installed offshore: Oseberg D, Snorre B, and Eldfisk 2/7-E. Today, these are still the only three in the world. In the future, several new offshore combined cycles may be installed due to changes in gas prices, taxes and available technology.

Off-design operations are very common offshore. Combined cycle plant may operate far from the point, which the equipment were designed for. This is due to changes in oil and gas production, demand for electricity, and ambient conditions. These changes in performance are of paramount importance to obtain reliable operation over a wide range[4].

are the only three combined cycles used offshore in the world. The first offshore combined cycle was installed at the Oseberg field in 1999 on the Norwegian sector of the North Sea.

1.2 Objective

Few studies on offshore combined cycles have been conducted in the recent past, which make this an interesting topic for investigation. The primary objective of this thesis is to review the off-design performance of an offshore combined cycle, including some of the following co-objectives:

- Theoretical description of combined cycles and general off-design performance.
- A literature study of existing offshore CC
- Learn how to use the simulation program GTPRO, GTMASTER and PEACE
- Develop a sensitivity analysis concerning weight and efficiency, and design an offshore combined cycle from the results.
- Develop a high efficiency plant as a reference plant.
- Simulate off-design behavior of designed plants
- Compare and evaluate the results concerning off-design behavior

1.3 Report Structure

The thesis is divided in seven chapters. Following the Introduction Chapter, Chapter 2 contains a description of combined cycles from a thermodynamic and technical point of view.

In Chapter 3, combined cycles applied for offshore installations are described. Further, in Chapter 4, off-design theory regarding the main components in a combined cycle is carried out

Chapter 5 describes the methodology for which the simulation is carried out. Further the different cases are established on the basis of theory from Chapter 2 and 3, and assumptions. The different Off-design simulations and the design procedure are defined.

In Chapter 6, the results are presented. A discussion and comparison of the results concerning off-design performance is also completed.

A conclusion is presented in Chapter 7.

2 Combined Cycle

An understanding of a combined cycle plant from a thermodynamic and technical point of view is important to be able to describe the performance of a system and to verify simulations. The fundamentals of each individual component must be pronounced and their operating relations when joined in one plant. Also the factors which are important for the offshore installations must be cleared out in order to understand choices which may seem arbitrary from a thermodynamic point of view. A review of existing literature reveals that few combined cycles for power generation offshore has been studied in the past. In this section a brief overview of previous work is outlined and the platforms, including their requirements, power demand and concept design are described.

2.1 Combined Cycle

When two power cycles are connected in one plant it is referred to as a combined cycle. Heat energy is discharged from one cycle and used as energy input to the other cycle. The most common combined cycle consist of a gas turbine (Brayton-cycle) and steam cycle (Rankine-cycle). The gas turbine burn fuel and operates at high average temperatures, it is therefore called the topping cycle. The second cycle utilizes the exhaust energy from the gas turbine, which also is quite high, to produce steam. This cycle is called the bottoming cycle. The topping and bottoming cycles is coupled with a waste heat recovery steam generator that transfers heat[5].

A schematic diagram of a simplified combined cycle (CC) power plant is shown in Figure 2.1. The gas turbine (GT) is the main component that transforms gaseous or liquid fuel into mechanical work and exhaust energy. The flue gas is transported through a heat recovery steam generator (HRSG) where the exhaust enthalpy is used to produce steam at high pressure and temperature levels. The steam is expanded in a steam turbine (ST), consequently producing additional mechanical work. The GT and ST may be coupled to a generator, which converts mechanical work to electricity. At design condition about 60% of the power is developed by GT and 40% of the power by the ST. The steam is further condensed in a condenser and pumped back to a feedwater tank/deaerator[5].

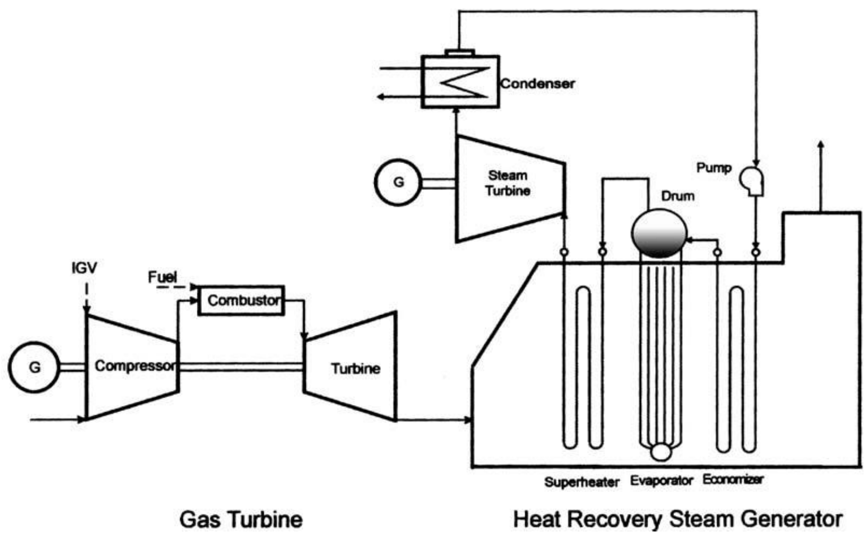

Figure 2.1, Simplified combined cycle

The main reason for choosing a CC is to achieve more power output for a given fuel supply compared to a simple cycle (SC). A large GT used in power plants have typically efficiency in the range of 35-40%, however for a large CC plants the efficiency may be as high as 60%. This increase in efficiency may simply be explained by the Carnot efficiency:

$$\eta_C = 1 - \frac{T_C}{T_H} \tag{2.1}$$

Where $\eta_C$is the Carnot efficiency in percent, and T_C and T_H are respectively the temperatures of heat rejected and supplied. This equation describes the maximum efficiency of an ideal thermal process between two reservoirs at temperatures T_C and T_H. Since the heat input and output occurs at gliding temperatures in a CC, average temperatures may be used.

When reviewing eq. 2.1, the best solution is approached by increasing the mean temperature of heat supplied and/or reducing the temperature of heat rejected. The top temperature is normally restricted by metallurgical properties and the bottom temperature by the surrounding atmosphere. From Figure 2.2, A and B, describing a gas turbine and a steam cycle respectively, the processes has either high temperature for heat rejection (GT) or low temperature for heat input (steam cycle), both leading to low efficiencies. When these processes instead are combined, Figure 2.2 C, the advantages from each cycle are obtained, both a high temperature for heat input and a low temperature for heat rejection.

The efficiency for a real process is lower than the Carnot efficiency mainly due to exergetic losses between the gas and steam cycle, but still the Carnot efficiency describes the quality of a thermal process. The Carnot efficiency of a combined cycle plant may be in the range of 65-78%, but actual plant efficiencies are only 75% of the Carnot efficiency [5].

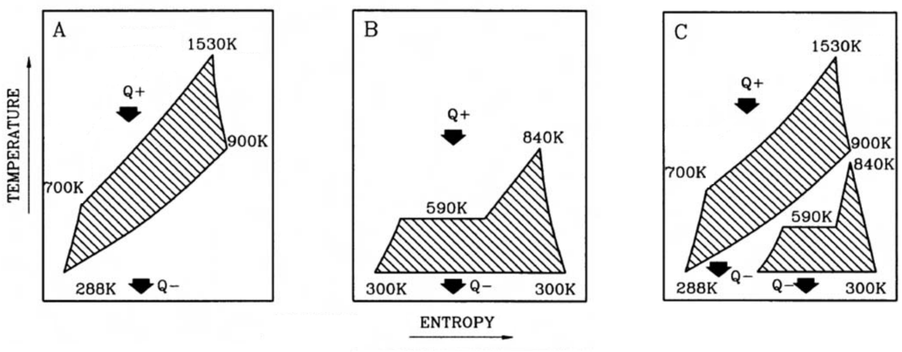

Figure 2.2, Temperature/entropy diagrams for various cycles, A is a simple cycle, B is a steam cycle and C is a combined cycle [5]

A more practical definition is the electrical efficiency of a combined cycle is:

$$\eta_{CC,gross} = \frac{\dot{W}_{GT} + \dot{W}_{ST}}{\dot{m}_{fuel} \cdot LHV} \tag{2.2}$$

Where $\dot{W}_{GT}$ is the gas turbine power output, $\dot{W}_{ST}$ is the steam turbine power output, $\dot{m}_{fuel}$ is the fuel consumption, and LHV is the lower heating value of the fuel. Eq. 2.2 does not consider auxiliary power consumption, which in that case would give the net efficiency

$$\eta_{CC,net} = \frac{\dot{W}_{GT} + \dot{W}_{ST} - \dot{W}_{aux}}{\dot{m}_{fuel} \cdot LHV} \tag{2.3}$$

Where $\dot{W}_{aux}$ is the power consumption from auxiliary equipment, such as pumps.

2.2 Gas Turbine

The gas turbine is the key component of a combined cycle power plant. It produces about 60% of the power and is the heat source for the bottoming cycle [5]. A single shaft, simple cycle gas turbine is illustrated in Figure 2.3. The GT consist of three major parts; a compressor, a combustion chamber and a turbine.

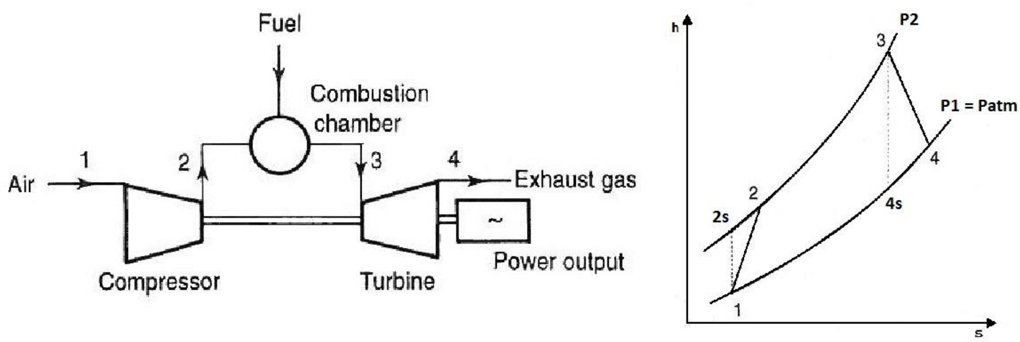

Figure 2.3, Simple cycle and h-s diagram [6]

In point 1, ambient air enters the compressor. Gas turbines are normally rated at ISO-conditions which refer to 15°C, 1.013bar and 60% humidity at ambient conditions. However there are typically inlet filters, and consequently a pressure loss before point 1. The compressor raises the pressure to about 10-35 bars, varying with type of gas turbine. The temperature also is raised because of the temperature-pressure relation. From point 2, the air supplied from the compressor is mixed with fuel in the combustor and burned. In theory this process occurs at constant pressure, but a pressure drop of 2-8% in the combustor is typical. Most GT`s run on natural gas, but other liquid petroleum distillates may also be used. At point 3, after combustion, the temperature of the gas may be as high as 1550°C in aircraft applications. The high temperature flue gas is expanded through the turbine to a pressure slightly higher than ambient. The compressor and turbine is divided in stages, where each stage consists of a row of rotors coupled to the shaft followed by row stators fixed to the casing. In the turbine, the stators convert pressure energy to kinetic energy, and the rotor converts kinetic energy to mechanical work. A fraction of the mechanical work produced in the turbine is used in the compressor for the opposite energy conversion, pressure increase. The exhaust gas has a high temperature in the range of 450-650 °C in point 4. This represents the remaining fuel energy (LHV) not extracted in the GT[7].

The thermodynamic process for the ideal simple cycle (1-2s-3-4s) is illustrated in Figure 2.3. This process may be explained by isentropic compression (1-2s) and expansion (3-4s) between two isobars, heat input (2s-3) at constant pressure and heat rejection (3-4s) at constant pressure. Because of diverging isobars in the enthalpy-entropy diagram, the turbine produces more work than the compression demands, thus power output. This is a closed cycle, but for the actual open process the exhaust is led to the atmosphere.

The thermal efficiency of an ideal cycle depends only on the pressure ratio and the properties of the working fluid, and is defined as

$$\eta_{GT,ideal} = 1 - \left(\frac{p_1}{p_2}\right)^{\gamma-1/\gamma} \qquad 2.4$$

Where the pressure ratio is ${p_1}/{p_2}$, which ideally is equal to ${p_4}/{p_3}$. γ is the isentropic exponent, typically 1.4 for air. From this equation a higher pressure ratio is advantageous and independent of other parameters. However the specific work output is not only a function of pressure ratio but also the maximum cycle temperature T_3. In this case, a combination of both high pressure and temperature is beneficial, and the maximum power output occurs when $T_2 = T_4$, thus at a lower pressure ratio than for maximum efficiency. A real cycle is different from an ideal cycle in terms of performance, but both high pressure ratio and high turbine inlet temperature (TIT) remains important[6].

There are typical losses in all the individual process of compression, expansion and combustion. Compared to the ideal case, the expansion and compression is not isentropic, and consequently the thermal efficiency is also dependent on the cycle temperature T_3/T_1. If considering a compression to pressure p_2 and no heat input from combustion, the irreversibility's implies a negative net work output when expanding back to p_1. The opposite, if heat is added after compression, T_3 increase and net work output becomes positive. Therefore, the efficiency increases with increased T_3 or reduced T_1 .

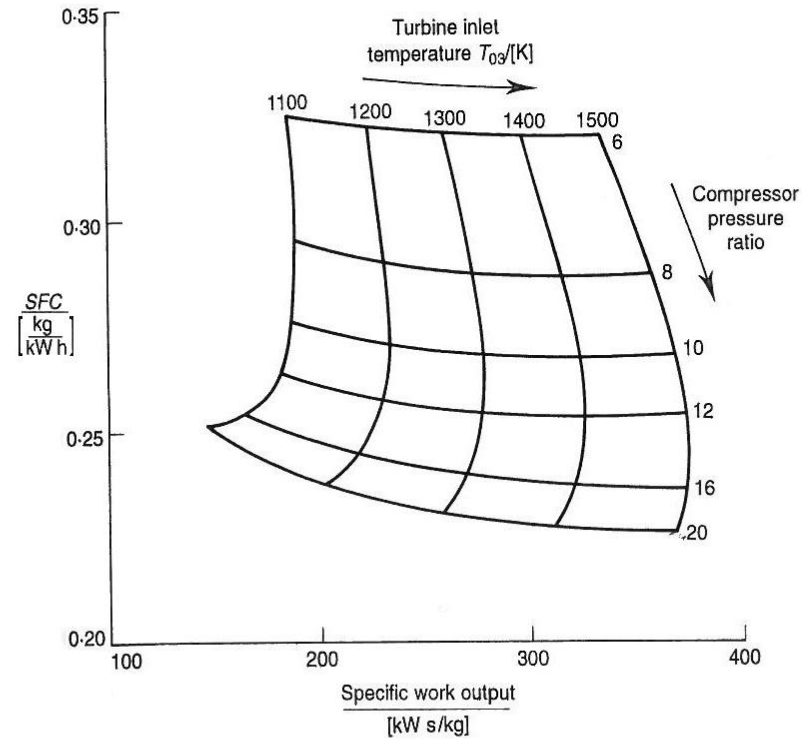

Figure 2.4, Cycle performance curve [6]

Figure 2.4 shows a plot of the specific work output and the specific fuel consumption (SFC). The plot is based on realistic values of polytropic efficiencies and pressure losses. SFC is a more common definition rather than efficiency. This value is inversely proportional to the efficiency and also describes the fuel consumption. Output per kg airflow is important because it connects the size of the gas turbine with the power output, and SFC is important because it directly relates to the operating cost. The plot shows the high effect on increasing TIT with respect to specific output, and that the SFC is mainly driven by pressure ratio. At a high specific work output, less mass flow air is needed to obtain a given net output, resulting in a smaller plant size[6]. For economic reasons, gas turbines are generally optimized with respect to specific work output rather than efficiency. Coincidentally often this optimum corresponds fairly accurate with the optimum efficiency of combined cycles. Exhaust gas temperature (EGT) is proportional to TIT, and a higher value corresponds to a more efficient combined cycle. When designing a combined cycle plant, the gas turbine concept and TIT are therefore important factors. The gas turbine is generally a standard engine designed by the manufacturer, therefore specific engines should be chosen for a CC plant[5].

It is normal to distinguish between industrial and aeroderivative gas turbines. Aeroderivatives are gas turbines derived from aircraft applications, where weight and size are primary drivers. Aeroderivatives are typically twin-shaft arrangement consisting of a gas generator (GG) and a free power turbine (PT). The GG and PT are only aerodynamically coupled, making aeroderivatives desirable for driving variable speed loads, such as pipeline compressors and marine propeller.

Industrial gas turbines are commonly used in large baseload power plants with an output of 100-350MW. They are designed in a single-shaft configuration and are heavier. These single-shaft units are suitable for running generators at 3000 or 3600 rpm directly without gearbox. An aeroderivative engine is illustrated in Figure 2.5.

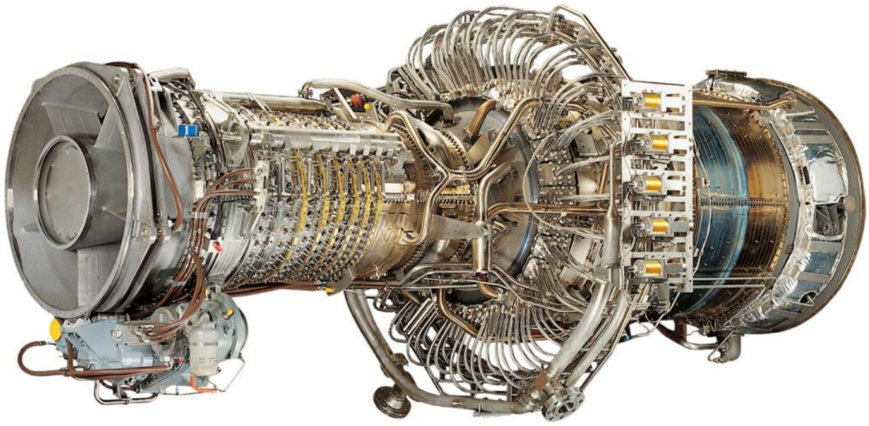

Figure 2.5, GE LM2500+G4 [8]

From Table 2.1, the differences between aeroderivatives and industrial gas turbines are summarized. It follows that industrial gas turbines have higher EGT than aeroderivatives, and consequently would be the most efficient for combined cycle plants. Aeroderivatives have a higher pressure ratio, hence lower values of EGT and are less efficient for steam cycle. However, in this thesis aeroderivatives are only considered further because of reasons described later in Chapter 3.

Table 2.1, Differences between aeroderivative and industrial gas turbines [6, 7]

Aeroderivative Gas Turbines	Industrial Gas Turbines
Based on jet engine	Not designed to fly
High power to weight	Low power to weight
Compact in volume	Medium compact to bulky
High pressure ratio of 25-35	Moderate pressure ratio (10-18)
Low values of EGT of 450°C	High values of EGT 550-600°C

The importance of a high TIT has been described both with respect of high specific work and high combined cycle efficiency. However, there are practical design limitations; the TIT is limited by metallurgic restrictions of the turbine blades. To obtain highest possible temperatures, film blade cooling is used on modern turbines.

2.3 HRSG

The gas turbine is connected with the steam cycle by the HRSG. Available exhaust heat from the GT is used to raise superheated steam for driving a steam turbine. In a combined cycle, the HRSG is the only unit which is tailored specially for each plant and gas turbine. The simplest form of an HRSG is one steam pressure level. It typically consist of three heat transfer sections;

economizer, evaporator and a superheater. The basic idea of heat transfer in a single pressure HRSG is shown in Figure 2.6. After the condenser, cold feedwater is pumped into the economizer. In this zone the feedwater is heated to a temperature close to saturated condition. Further the feedwater is evaporated at constant pressure and temperature in the evaporator. In the last zone, the superheater, the steam is superheated to a required live stream temperature. The temperature of the exhaust gas drops simultaneously in each stage before it reaches the stack[5].

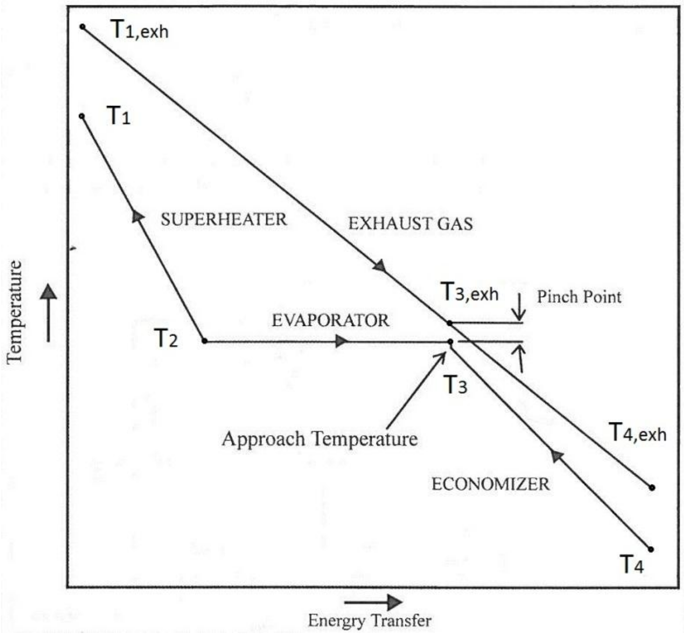

Figure 2.6, T-Q diagram, single pressure[5]

From the figure, two important parameters are outlined; the pinch point temperature and the approach temperature. There must be a temperature difference between the exhaust gas and water/steam through the entire HRSG to ensure heat transfer. Ideally this difference should be constant to minimize the exergy loss, however the because of constant temperature of evaporation this difference vary .The minimum temperature difference, called the pinch point, occurs between the inlet of the evaporator and the gas side. This property is important for the HRSG design because it is direct linked to heat transfer area and steam production. The surface area required for a heat exchange is given by

$$A = \frac{\dot{Q}_{HRSG}}{U\Delta T_{LM}} \tag{2.5}$$

$\dot{Q}_{HRSG}$ is the desired heat exchange, U is the overall heat coefficient and ΔT_{LM} is the log mean temperature difference. ΔT_{LM} is related to the pinch point temperature.

A reduction in pinch point would lead to more heat recovery, hence steam production. However, the surface area, which is inversely proportional to the temperature difference would be larger and at one point impractical[5]. The pinch point is typically between 8-12K in high efficient plants and up to 35K in plants where capital cost or weight are primary drivers, such as units used in peak load and at offshore installations[7].

The approach temperature is the temperature difference between the economizer outlet and the saturation temperature. If the flue gas exiting the evaporator has a higher temperature at off-design, this could lead to steaming in the economizer, with the further consequence of blocking of the flow and additional evaporation in the economizer. Problems such as overheating, water hammering may occur. To prevent this, the economizer heat transfer area may be reduced, with the effect of creating an approach temperature. This value varies from 5 to 12K and will together with the pinch point temperature difference determine the steam production.

The most important design parameter of the HRSG is the heat load, which determines the amount of steam generated. About two thirds of the total energy is available as energy in the exhaust gas, this heat may be expressed as the amount of unused energy from the GT

$$\dot{Q}_{exh} \cong \dot{m}_{fuel} \cdot LHV(1 - \eta_{GT}) \qquad 2.6$$

From a heat balance between exhaust gas and steam/water over pinch, including evaporator and superheater, the steam production may be determined.

$$\dot{Q}_{HRSG} = \dot{m}_{steam}(h_1 - h_3) = \dot{m}_{exh}C_{p,exh}\left(T_{1,exh} - T_{3,exh}\right) \qquad 2.7$$

$$\Rightarrow \dot{m}_{steam} = \frac{\dot{m}_{exh}C_{p,exh}\left(T_{1,exh} - T_{3,exh}\right)}{(h_1 - h_3)} \qquad 2.8$$

The HRSG is generally restricted in both ends, the GT on the exhaust side, and ST and condenser on the water/steam side. However the main design parameters, live stream pressure (LSP) and live stream temperature (LST) should be optimized to obtain the best plant solution. The steam turbine work output is defined by

$$\dot{W}_{ST} = \dot{m}_{steam}(h_{steam,inlet} - h_{steam,exit}) \qquad 2.9$$

From Equation 2.9, the work output is function of steam production and steam enthalpy at the inlet and exit of the ST. The effect of a higher live stream pressure will make the steam turbine achieve a greater power output due to higher enthalpy difference in the expansion. However because the evaporation temperature also increase, less steam will be produced in the HRSG. Furthermore, this results in higher stack temperature and a lower HRSG efficiency.

$$\eta_{HRSG} = \frac{\dot{m}_{exh}c_{p,exh}\left(T_{1,exh} - T_{3,exh}\right)}{\dot{m}_{fuel} \cdot LHV(1 - \eta_{GT})} \qquad 2.10$$

The best solution may be found between these two effects. Another interesting behavior is the high power output does not correspond to a high HRSG efficiency. If lowering the live stream pressure, the exhaust stack temperature increase and more heat energy is recovered. However, as illustrated in Figure 2.7, reduced pressure gives a higher exergy loss in the HRSG. From this one may conclude that high exergy efficiency is the dominant factor for ST power output compared to energy efficiency. It is common to design larger plants with two or three pressure levels. In this way, the temperature difference between the exhaust gas and steam/water side is reduced and hence reduced irreversibility[5]. The dual pressure increases the steam cycle efficiency with about 4% from the single pressure and another 1% with a triple pressure.

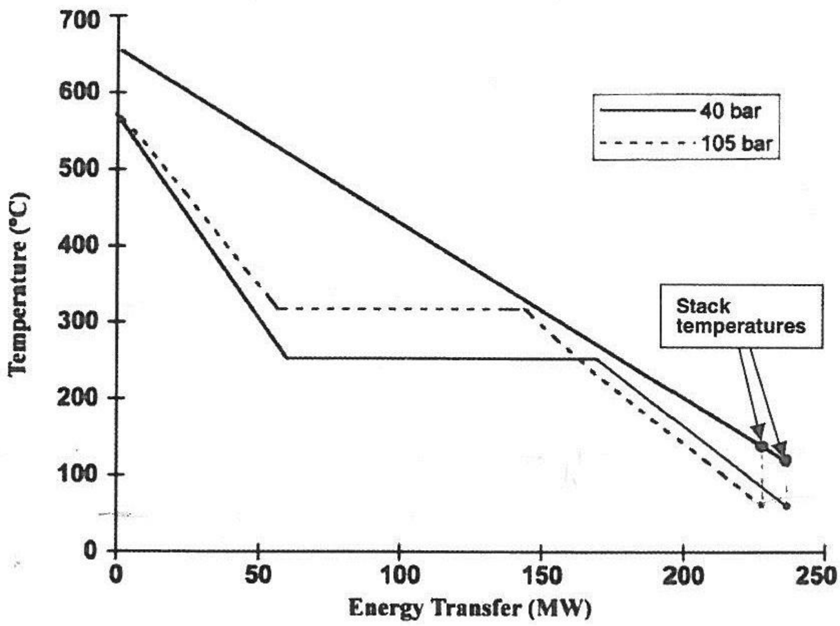

Figure 2.7, T-Q diagram, live stream pressure at 40bar and 105bar[5]

The effect of increasing live stream temperature in a single pressure HRSG, also results in a higher steam enthalpy drop in the ST. However in this case, the reduced mass flow because of additional superheating is more significant, thus reduced power output. One should bear in mind that this changes when several pressure levels are used, and higher live stream temperature may be beneficial[5].

HRSG`s are typically of fin-tube designs with water flowing inside the tubes and the exhaust gas around, and there is consequently a pressure drop on both sides. Emphasis should be placed on designing a HRSG with as low as possible pressure loss on the flu gas side. The gas turbine has a lower expansion because of increase in the backpressure, and thus a reduced efficiency. Unfortunately a low pressure drop across the HRSG would lead to a bigger and more expensive unit. The exhaust pressure loss is normally between 25 to 30mbar[5].

When the feedwater enters the economizer, the temperature is low. The heat transfer on the water side is superior to the gas side, and therefore the tube wall temperature is approximately the same as the water temperature. This may cause a problem if the feedwater temperature drops below 40-45°C in situations where fuel containing no sulfur. A temperature below this limit may cause acids to condensate on the tube wall, resulting in corrosion. The sulfuric acid dew point is proportional to the partial pressure of sulfur in the exhaust, if fuel with sulfur is burned this would increase the minimum temperature limit of condensation[5].

There are several configurations of HRSGs, however two basic configuration described by the exhaust flow direction is common. In a vertical HRSG the tube bundles are horizontal and circulation pumps maintain constant flow through the evaporator. In a horizontal HRSG the tubes are oriented vertical and natural circulation ensures constant circulation. In the past vertical HRSG where advantageous because of less footprint, smaller boiler volumes due to smaller tube diameter and less sensitive to steaming in evaporator during startup. Today the differences in performance are minor: Natural circulation boilers have reduced the tube diameter, hence the water volume, and steaming and blockage problems are partly solved. Startup time and space requirements are almost the same and they may achieve the same pinch point temperature difference[5].

An important decision concerning the design and operation is whether to associate more than one gas turbine with a single HRSG and steam turbine. In such a design, each gas turbine should be mounted with independent diverter and bypass-stack in case one GT should shut down and should need maintenance. For better regulation and availability, also supplementary firing should be reviewed as a second option. Supplementary firing is burning of additional fuel after the GT, increasing the peak load but also the capital cost.

2.4 Steam Turbine

Steam turbines utilize the high enthalpy steam which is at high pressure and temperature in an expansion process. The steam enthalpy is converted into mechanical energy as it passes through a turbine stage. Each stage consists of nozzles and rotor blades. In the nozzles, the steam is accelerated and transformed into kinetic energy with a reduction in potential energy (pressure and temperature). The flow is directed onto the rotor blades which convert the kinetic energy to mechanical energy, thus work output[9].

Typically for large plants with several pressure levels in the HRSG, the turbine section consist of a high pressure turbine stage (HP), intermediate pressure turbine stage (IP) and a low pressure stage (LP). The first HP stage may have a pressure between 100-310bar and a temperature of 656°C. After the HP, the steam may be reheated to a higher temperature before expanding in the IP stage. The IP and LP stage have values between 20-40bar and 3-6bar, respectively. The steam from IP exit is mixed with the LP steam before the last expansion[9]. The expansion process may be illustrated in an enthalpy/entropy -diagram, se Figure 2.8.

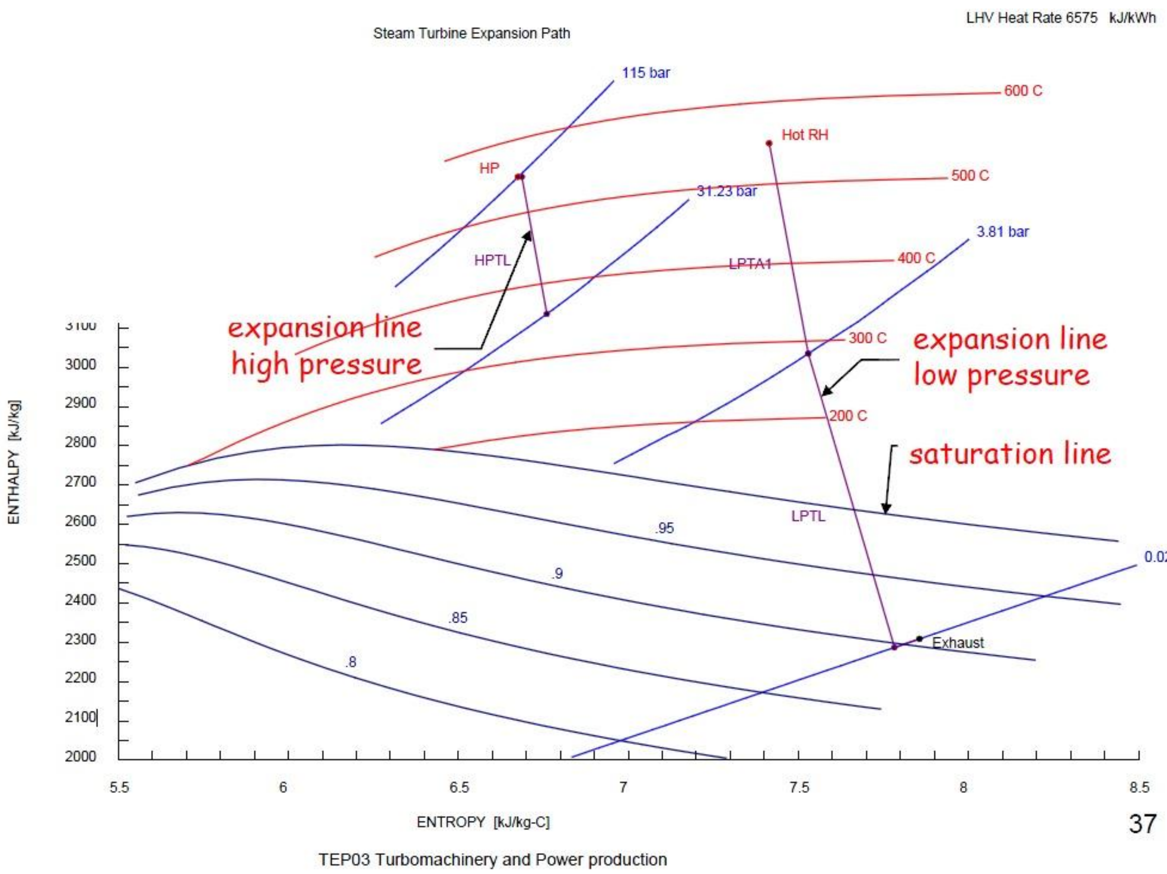

Figure 2.8, Expansion process in enthalpy/entropy-diagram [5]

The specific work done by the turbine is sum of the enthalpy difference between inlet and exit at each pressure stage. Hence, inlet enthalpy, which is a function of temperature and pressure, and exit vacuum pressure are important parameters of a steam turbine output. From Figure 2.8, one may see that the LP turbine expansion crosses the saturation line and end in the two-phase area. A lower backpressure is advantageous, resulting in increased output. Because of high latent heat of water, expansion further than the saturation line, results in larger enthalpy difference. However, a high moisture content in the steam may lead to erosion of the last LP blading[9]. The steam quality should not be lower than 84% when exiting the ST. The steam quality is both determined by ST input values, such as pressure and temperate, and condenser pressure[5].

There are two types of steam turbines, back-pressure and condensing. A back-pressure steam turbine is designed for steam extraction for process heat, and therefore the ST exit pressure is higher. A condensing turbine is designed for high power output and therefore must be larger in size to handle the large exit steam volume.

2.5 Cooling System

The exhaust steam from the ST needs to be condensed to water before it is pumped back to a higher pressure. The cooling system rejects this heat from the condenser to some cooler medium. As previously discussed, low condenser pressures are associated with higher turbine power output. At the same time, a low condenser pressure causes larger dimension of the condenser and steam turbine. The cooling medium and its characteristics, such as temperature, possible heat load, and environmental regulations are the main parameters that affect the selection of cooling system[7]. There are three types of cooling system used in combined cycle plants classified by their cooling medium[5]:

- Once-through water cooling
- Evaporation cooling
- Dry cooling system

Water is the primary choice of cooling medium because of its high heat capacity and heat transfer properties. In a once-through water cooled condenser water is taken directly from a water source like the ocean, a river or a cooling pond, and used as a heat sink for process cooling, before it is returned to the water source. This is the most economic type of cooling system and allows the lowest condenser pressure in the range of 0.02-0.06bar.

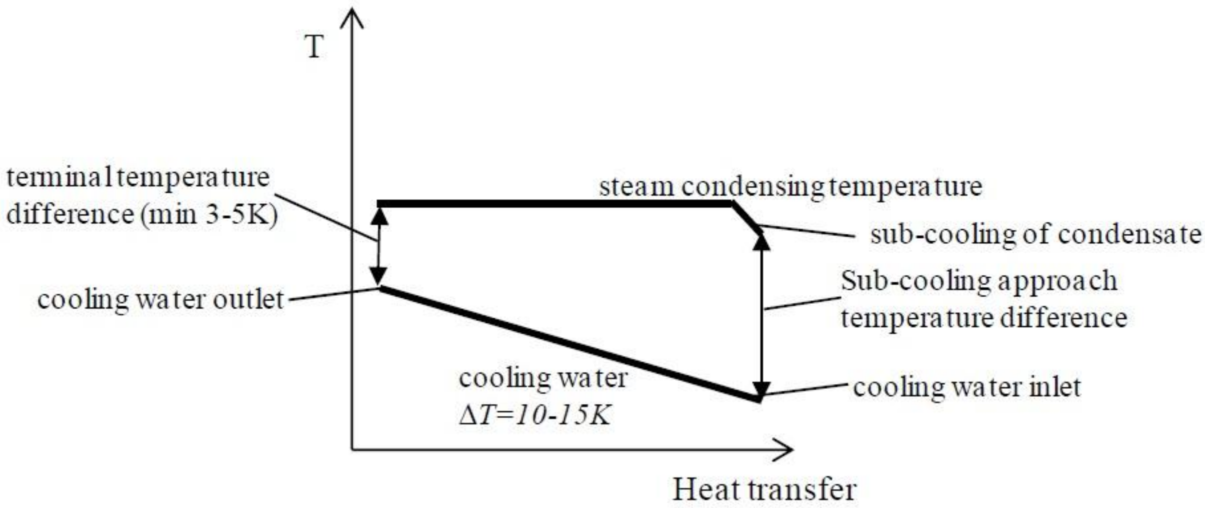

Figure 2.9, T-Q diagram of a condenser[7]

The cooling water typical has a temperature rise of 10K and a minimum temperature difference to the feedwater of 3-5K. A T-Q diagram of the water cooled condenser is showed in Figure 2.9. If assuming the cooling water has a temperature of 15°C at the inlet, a cooling water temperature rise is 10K and a minimum temperature difference of 5K, the steam condensing temperature must be 30°C, this result in a condensing pressure of 0.042bar. Offshore there is an unlimited amount of cooling water, hence once-through water cooling is chosen in this thesis. However, in situations where water is not available or limited, dry cooling or evaporative cooling is used.

Dry cooling may be of the type direct air-cooled condenser or indirect dry cooling tower. Because of the heat transfer coefficient of air is low, the heat transfer area becomes larger. The condensing pressure is limited by the dry bulb temperature of air. In an evaporative cooling process, the hot cooling water from the condenser is sprayed in droplets in a cooling tower. The raising ambient air makes the surface of the droplets evaporate, and therefore cooling the remaining droplet. The wet bulb temperature of the air determines in this case the condenser pressure[7].

The difference in condenser pressure is shown in Table 2.2. The temperature of the cooling medium is directly linked to the condenser pressure. One may see that water is the best option.

Table 2.2 Condenser pressure intervals for different cooling systems [7]

Cooling System	Condenser Pressure
Indirect dry cooling tower	0.10-0.20 bar
Direct air-cooled condenser	0.07-0.15 bar
Evaporation cooling with wet cooling tower	0.05-0.12 bar
Direct water cooling of condenser	0.02-0.06 bar

2.6 Feedwater Tank/Deaerator

Most power plants with a steam cycle have a deaerator prior to the HRSG. If high amount of oxygen is presence in the feedwater this may cause corrosion of equipment and piping. If additional carbon dioxide also is presence, then the ratio of corrosion will increase. Therefore removal of these noncondensing gases is important. Deaeration is an important process that removes dissolved gases from the feedwater before it enters the HRSG. Deaeration is a continuous process due to water losses and air leakages into the system. The concentration of oxygen should not exceed 20ppm in the feedwater. The feedwater tank is placed under the deaerator, and function as a buffer in feedwater. In Figure 4.2, the deaeration process is illustrated. Boiler feedwater is sprayed in a thin film at upper part of the deaerator and is heated by hot deaeration steam injected at lower level. This reduces the solubility of the dissolved gasses, and the amount of the gases in the feedwater is reduced. The gases are vented out at the top, and the deaerated water is pumped to the HRSG. [7]

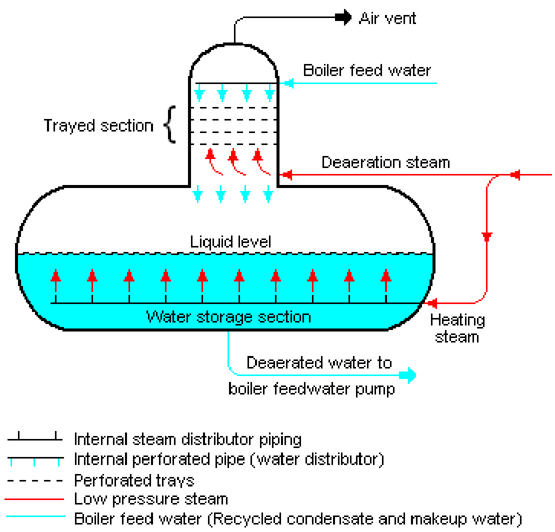

Figure 2.10, Deaerator [7]

3 Offshore Power Generation

When designing a combine cycle plant concept it is important to understand which processes that need energy and which requirements that has to be fulfilled. The oil and gas installations are in many ways different from onshore CC, both with respect to requirements, operating philosophy, site related factors and plant concept solution. The different requirements may sometimes be conflicting making the design process more complicated, therefore compromises have to be made. Relatively little literature about offshore CC exist. Instead more common power generating systems offshore may be used to describe requirements to offshore combined cycles.

3.1 Energy Demand on Offshore Installations

Offshore installations have a substantial power demand to support the oil and gas extraction and processing. The energy demand on installations varies with field and location, production plan, pressure development in reservoir, composition of well stream, methods used for oil and gas recovery, and time of year. In 2005 the total mechanical and electrical power demand on the Norwegian continental shelf was about 15.75 TWh, and the heat demand was approximately 5 TWh.

The main functions of an installation are processing of well stream, enhanced oil and gas recovery, export compression and drilling of wells. The oil and gas needs to be processed to meet the transport specifications, and this include processes such as separation and stabilization of the phases, dehydration, sour gas absorption, regeneration of MEG and water treatment. An oil field is different from a gas field with respect to which processes that are the most energy consuming. For a typical oil field that produces mainly oil, but also some gas, the most energy demanding process is injection of water. This is a recovery method that starts early in the production phase to maintain high reservoir pressure and oil extraction. In addition, oil pumping for export and gas compression for export or re-injection represents a major source of energy consumption. In the case of a gas field, the initial gas pressure is high and the energy demand is primary related to gas compression before pipeline transportation. In a later phase, the wellhead pressure decrease and pre-compressors are used for gas lift [10]. Other energy demanding processes are, sea water lifting, drilling of new fields, and utility systems. The energy demanding processes may be summarized as followed:

- Water injection
- Gas lift
- Oil pumping
- Gas compression for transport/injection
- Sea water lift
- Drilling operations
- Utility systems
- Processing of oil and gas

The different processes on an offshore installation may be categorized in three forms of energy, mechanical power, heat and electricity:

- Mechanical power is needed to drive large compressors, pumps and generators
- Electricity to drive compressors, pumps, utility systems coupled to an electric motor, and for living quarters.
- Heat is needed in the processing of oil and gas.

Currently the oil and gas sector is developing towards a more energy demanding production. A higher share of gas production and more activities further north increase energy of gas compression and transport. There is also a development towards more mature fields, which need improved recovery methods. In addition, technology development and subsea operation at greater depths increase pumping, artificial lift and heat energy[10]. A higher consumption implies higher costs and emissions. This may result in higher interest around combined cycle offshore

3.2 Offshore Power and Heat Systems

The layout of power and heat system varies from every offshore plant due to different requirements. Likewise, most offshore installations are self-sufficient with energy with the exception of Gjøa, Valhall, Ormen Lange and Troll A which is supplied with electricity from shore [11]. Depending on installation the required power may vary from a few to several hundred megawatts. To cover this need of power almost all the platforms use one or several gas turbines, often described as simple cycle gas turbines (SCGT). This represents the most preferred technology offshore due to many years of optimizations and is implemented in several different systems. On the Norwegian continental shelf there are about 167 gas turbines with an installed power of 3000MW. The actual power consumption is lower due to backup capacity and part-load performance. It is normal to have additional gas turbines or diesel engines in backup to ensure spare capacity in case of shutdown. Of the actual utilized power, 45% is used for electricity generation and 55% for mechanical drive applications[10].

The heat demand is either covered by combined heat and power systems (CHP) or with direct fired heaters. The most common is CHP and use of waste heat recovery units (WHRU) to utilize the exhaust energy. A CHP system increase the efficiency compared to stand alone gas turbines without heat recovery. In the end of 2005, 61 gas turbines were equipped with WHRU and 6 fired heaters were in operation[12]. Three platforms, Ekofisk, Oseberg D and Snorre B have installed combined cycle with gas turbines as topping cycle. A combined cycle increase the efficiency compared to a simple cycle. The gas turbines in a combined cycle are either used for mechanical drive or generator drive, and the steam turbines are only used for generator drive. The high number of gas turbines represents a potential for more combined cycles offshore, though many plants use WHRU`s to recover heat.

3.3 Requirements on Offshore Installations

The oil and gas companies have special requirements for a offshore combined cycle which in many ways are different to an onshore plant:

- Availability/Reliability
- High Power to Weight ratio
- Size and area requirements
- Hostile environment offshore
- Easy maintenance and repair

The economy to an oil and gas field is directly linked to extraction and production of oil and gas. As a result, high production and high availability is of primary importance. However the production is limited by drivers and compressors capacity, and their reliability. Downtime would result in losses of production and revenues. When designing an offshore combined cycle, simple and reliable system, which is easy to maintain and repair, is therefore a priority. For large onshore combined cycles the most important factor is the cost of a unit electricity delivered, and consequently optimization between capital cost and efficiency is important. Offshore, fuel has traditionally been available from the oil and gas processing, and has been considered as free energy. More recently fuel gas is reviewed as production loss and fuel savings could increase income and reduce emissions.

Offshore installations have limitations of both weight and space. It is therefore important that the equipment is compact and have a high power/weight ratio. However, combined cycles introduce a higher weight than gas turbines and this is mainly because of the HRSG. Most HRSG are developed for onshore operation where there are few limitations of weight and size. Offshore single pressure and multiple gas turbines per HRSG are common to save weight. The GT`s used offshore are normally aeroderivative engines because of their high power/weight ratio. Aeroderivative GT`s are developed from aircraft engines, and they are usually lighter than industrial engines. Coincidentally they may also have higher efficiency and high reliability [13].

The offshore environment on are considerably different from onshore. Strong wind gusts, high sea waves, and high salt content and humidity in the air involve special requirements to the equipment. The ambient temperature varies between -10°C to +23°C, and anti-icing systems are used to prevent icing on inlet and ducting when temperature drops below zero degrees. Filters and enclosures are used to prevent salt and hydrocarbons to reach internal parts. The severe environment must be considered together with the high cost of maintenance on an offshore platform and the extreme difficulty in reaching many of the parts located outside the platform [14].

3.4 Weight and Skids

Combined cycles offshore are normally designed and divided in three single lift skids; GT skid, ST skid and HRSG skid. The skids include all equipment including auxiliaries and control systems. It is a modular approach where each unit are preassembled and pretested before installation. This approach eases installations by having all interconnections, such as piping and signal cables ready in the interface. Another advantage is protection of the equipment from the harsh and corrosive environment [14].

Table 3.1 Equipment on different skids

THE GAS TURBINE SKID [15]:	THE HRSG SKID [2]:	THE STEAM TURBINE SKID [2]:
• gas turbine (GG and PT)	• economizer, boiler bank and superheater with casings	• steam turbine
• driven equipment	• inlet and outlet transitions, main and by pass stacks	• admission system
• fuel system	• single blade diverter(s)	• turbine bypass system
• bearing lube oil system including tank, filters, pumps and coolers	• steam drum	• if applicable; steam extraction valves
• starter	• blow down tank	• generator
• controls (on-skid, off-skid)	• boiler circulation pumps	• speed reduction gear
• seal gas system (compressors)	• make-up water mixed bed filter	• condenser
• single lift skid structure	• make-up water pumps	• condenser evacuation system
• air intake filters with anti-icing equip.	• chemical dosing station	• gland steam condenser
• outlet duct and silencers	• instruments and instrument valves incl. water monitoring	• lubrication oil system incl. pumps
	• instrument junction boxes on skid edge	• hydraulic systems incl. pumps
	• piping incl. piping components to skid edge	• instruments and instrument valves
	• single lift skid structure	• instrument junction boxes on skid edge
		• turbine and WHRU-SG automation
		• turbine governor
		• voltage controller
		• generator synchronizer
		• piping incl. piping components to skid edge
		• enclosure incl. ventilation system
		• single lift skid structure with three point support
		• special erection tools

If one consider a low-pressure steam turbine between 15-20MW, the weight of the steam turbine skid will be about 150-175tons. This is approximately the same as for a 20MW gas turbine skid. For example the steam turbine skid on Snorre B, including a steam turbine generator set, condenser and auxiliary equipment weights 190 tons dry. However the extra weight is concerning a combined cycle is related to the HRSG. The weight of a single inlet HRSG which is designed for the LM2500+ gas turbine is about 125 tons. This is about 30-50% less than standard onshore HRSG which weights 200-250tonnes [2]. The dry weight of a 30MW gas turbine skid (SGT-700) without equipment weights 78tons[16].

In a combined cycle a steam turbine skid may replace a gas turbine skid without additional area requirements. It is normal to place the HRSG skid is on top of the gas turbine skid and therefore the footprint of a combined cycle is typical the same as for a traditional gas turbine plant offshore. However one gas turbine does not supply the HRSG with enough exhaust energy to operate a steam turbine with similar power rating. The solution may be several gas turbines connected to a multiple inlet HRSG. This has proven to be weight efficient[2].

3.5 Offshore Combined Cycles in Operation

Oseberg was the first offshore platform to install a combined cycle which was in operation in 1999. Later the same year, the combined cycle on Eldfisk was in operation. The Snorre B combined cycle was planned to be in operation in 2000. The following description of these three CC is based on design data from 1999:

The combined cycle on Oseberg D uses two gas turbine packages (PGT25+) to drive two gas compressors for reinjection. The package consists of a LM2500+ gas generator and a power turbine designed by Nuovo Pigone. The power turbines are rated to 30MW each and have 40.3% efficiency at ISO condition. The steam turbine is located on the neighbor platform Oseberg A, and a 400m steam pipe connects the HRSG with the steam turbine. The steam turbine is rated to about 19MW with the gas turbines running on full load, this correspond to total plant efficiency of 50%. The HRSG is of a double inlet module and recover heat from both gas turbines. The exhaust has a temperature of 480°C and the steam is produced at one pressure level. The HRSG package is placed over the two gas turbines and the design is of a vertical gas flow arrangement with forced circulation. The gas turbines run independently of the steam cycle, and the steam turbine will produce electricity from whatever steam is produced in the HRSG. This ensures simple regulation. The CC is designed to run continuously and variations in electricity are covered by existing turbine gensets. Another operating possibility is to extract steam from the steam turbine to utilize as process heat. One by-pass stack and one diverter are fitted to allow simple regulation[2, 17]. The Oseberg flow diagram is illustrated in Figure 3.1

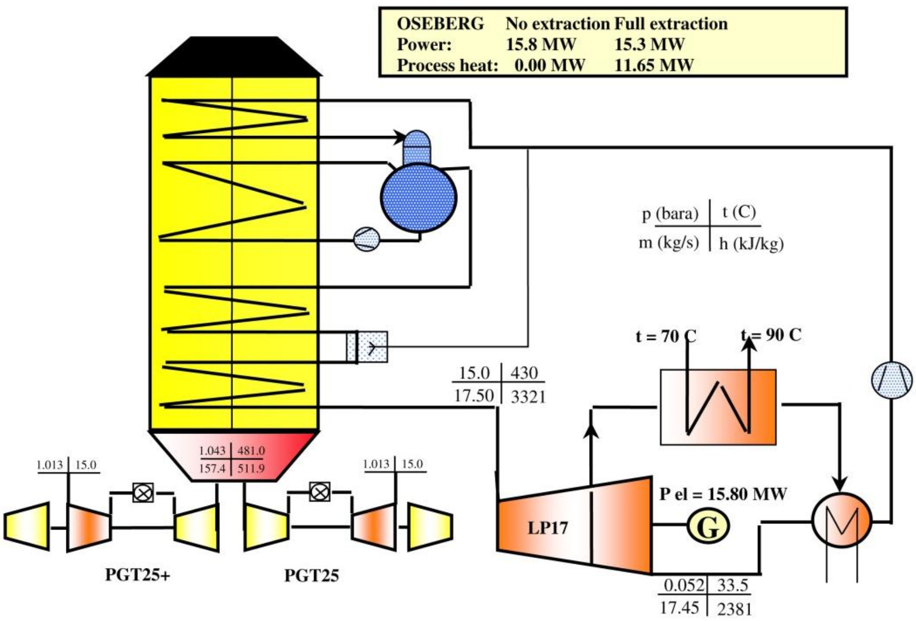

Figure 3.1, Oseberg flow diagram [2]

On the Eldfisk combined cycle plant, one HRSG with triple-inlets will utilize energy from one PGT25 and two LM1600 gas turbines. The steam produced will be used for driving a 10.3MW steam turbine which is the only electric power source on the platform. The PGT25, rated to 23.3MW is used for gas compression and the two LM1600 of each 12.2MW are used for water injection. Because of both varying electricity demand and power demand the steam cycle is designed for 10% more steam production than required, and a bypass steam valve is used to reject surplus steam to the condenser. There is no steam extraction from the steam turbine, but the in the cold end of the HRSG there is additional heat recovery for production of freshwater. Another special concept is the use of injection water instead of seawater in the condenser. This saves energy related to seawater lift. The steam process flow in the HRSG is identical with the one at Oseberg D, single pressure with four heat transfer zones; economizer, evaporator and two superheating zones[2, 17]. The Eldfisk flow diagram is illustrated in Figure 3.2.

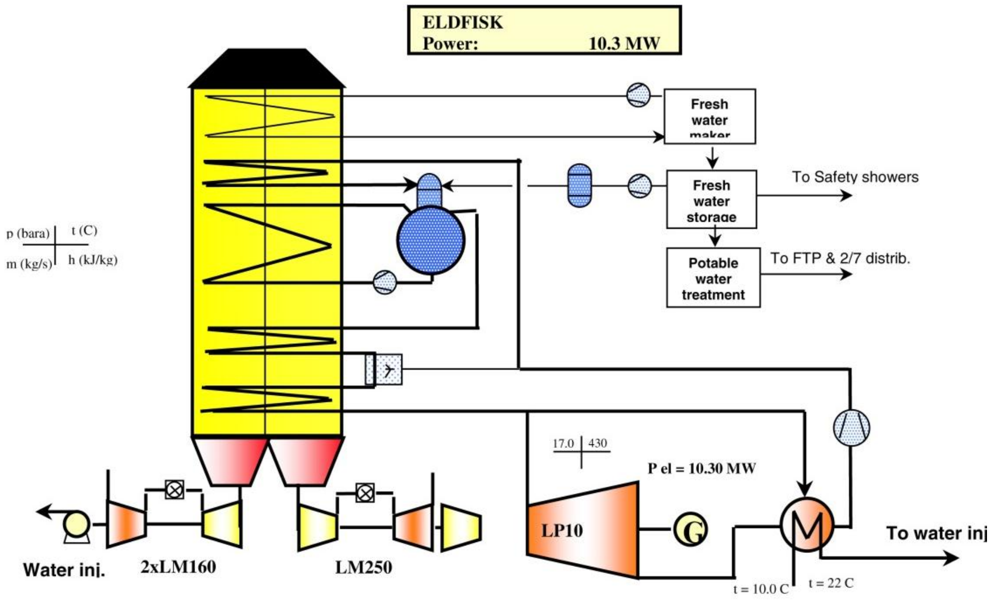

Figure 3.2, Eldfisk flow diagram [2]

The Snorre B platform uses another combined cycle concept. The combined cycle produce only electricity and run continuously at 100% load to maximize the efficiency and cut the payback time. Both the two DR63P (LM2500+) gas turbine and the LP17 steam turbine package are used for generator drive. The HRSG is of a double inlet type and has incorporated supplementary firing in case of a gas turbine shut down. The Snorre B platform export surplus electricity to Snorre TLP. This inter-platform power distribution makes better flexibility and utilization of the electricity. At design point and 100% load the gas turbines produce about 30MW each and the steam cycle produce 17.3MW electricity. There is also a possibility for extraction of steam with a total energy of 8.0 MW, and then the ST production will be 15.2MW[2].

This three cases show how different the configuration of power generation may be offshore and how each combined cycle are designed especially for each platform. Because the platforms are self-sufficient in energy and the energy demand varies, it is important to have a system which is reliable and easy to regulate. All the combined cycles have backup capacity and god regulations.

The Oseberg D platform has a bypass stack with diverters and backup capacity is covered by gas turbine genset. The Eldfisk platforms ensure reliability by using a dual fueled 5.25MW gas turbine in backup and god regulation of steam to the ST. On the Snorre platform surplus energy is delivered to a neighboring platform and deficit exhaust energy is covered by supplementary firing.

Table 3.2 summarizes the most important configurations of the offshore CCs presented above.

Table 3.2 Summary of the combined cycles data [2, 17]

INFO	OSEBERG D	ELDFISK	SNORRE B
CC application	Electricity (ST), gas compression (GT), steam extraction	Electricity (ST), gas compression(GT), water injection(GT)	Electricity (ST,GT), steam extraction
Gas turbine power	2x25.9MW (LM2500+, 88% load)	2x12,2MW(LM1600) 1x14,4MW(LM2500)	2x 30MW (LM2500+)
Exhaust temperature	481°C	504°C(LM1600), 577°C(LM2500)	-
Exhaust mass flow	157.4kg/s	2x43.7kg/s(LM1600), 52kg/s (LM2500)	-
HRSG	Vertical gas flow, single pressure, forced circulation, four zones, by-pass stack,	Vertical gas flow, single pressure, forced circulation, four zones, heat exchanger for water treatment	Vertical gas flow, single pressure, forced circulation, supplementary firing
Live steam temp.	430°C	430°C	-
Live stream pressure	15.0 bar	17.0 bar	-
Steam mass flow	17.5 kg/s	-	-
Steam turbine	15.8MW at 88% load (19MW at 100% load) 15.3MW (11.65MW steam extraction)	10.3MW	17.3MW 15.2MW (8.0MW steam extraction)
Plant efficiency	47% (88% load) 50% (100% load) 56% CHP	~50%	~50%

When the CC`s where installed they represented the best solution compared with simple cycle offshore with respect of cutting emissions and fuel consumption. The three CC`s where calculated to give 10 million EUR/year in savings of CO2 tax and up to 7 million EUR/year in alternative gas values[2].

4 Off-design Theory

The focus so far has been on describing the thermodynamic principles of combined cycles at design point and practical design considerations offshore. However it must be realized the operating conditions change, and the system must be able to operate at conditions far from design point.

Off design theory is about predicting how the system reacts to changing parameters. As compared to design, the actual geometry of the components or hardware remains constant during off-design. A combined plant may operate for prolonged time at off-design conditions, depending on power demand, ambient condition and other considerations offshore. This will have significant impact on the plant performance, and consequently, it is very important to ensure the system performs satisfactory not only at design conditions, but also at off-design conditions.

Off-design performance is generally affected by the following parameters[5]:

- Plant load
- Ambient air temperature
- Ambient air pressure
- Ambient relative humidity
- Cooling water temperature
- Frequency
- Power factor and voltage of generators
- Process energy extraction
- Fuel type and quality

Off all the ambient conditions affecting the gas turbine, the most important are ambient air temperature and pressure; however pressure is more a consideration at design than at off-design. The cooling water temperature has a major effect on the steam cycle; however at certain depths this is constant during the year. In this thesis two off-design conditions are studied, varying plant load and varying ambient air temperature. Varying plant load, referred to as part-load, may be required as a result of changes in electricity demand and/or mechanical work on the platforms. The ambient temperature typically changes from winter to summer, and may change with several tens of degrees.

Accurate calculations of the overall combined cycle and the interaction between the components, namely the GT, HRSG, ST and condenser is complicated. It is obvious that the system is much interconnected and changes of one component may lead to off-design operation of other components. The operating behaviour will also differ from plant to plant, depending on concept solution and actual design point. However, analytical methods to calculate off-design performance of the CC power plants are available. This will be presented in this chapter.

4.1 Gas Turbine

The gas turbine is the key component in a CC plant and defines the overall plant performance. All off-design calculations depend on satisfying the essential condition of compatibility of mass flow, work and rotational speed between the various components. This interaction is referred to as component matching. The performance of a gas turbine involves an excessive number of variables, and consequently non-dimensional parameters are used instead. These non-dimensional parameters are used to describe the GT characteristics in a manageable manner, and are defined as

Table 4.1 Non-dimensional parameters

Parameter	Compressor	Turbine
Non-dimensional mass flow, 4.1	$\frac{m\sqrt{T_{01}}}{p_{01}}$	$\frac{m\sqrt{T_{03}}}{p_{03}}$
Non-dimensional speed, 4.2	$\frac{N}{\sqrt{T_{01}}}$	$\frac{N}{\sqrt{T_{03}}}$
Pressure ratio, 4.3	$\frac{p_{02}}{p_{01}}$	$\frac{p_{03}}{p_{04}}$

Where m is the mass flow, T is stagnation temperature, p is pressure, and N is rotational speed. The suffix, 01 and 02, is respectively the inlet and outlet of the compressor, and the suffix 03 and 04 is respectively the inlet and outlet of the turbine. The temperatures ratios are not considered because they are proportional to the pressure ratios. The characteristics of compressor and turbine are normally obtained experimentally and are rarely available for the publics. However, the operation at design point may be used to predict off-design performance by simplified methods.

The compressor characteristics, sometimes called compressor map is showed in Figure 4.1. The overall performance is plotted on the basis of non-dimensional mass flow $m\sqrt{T_{01}}/p_{01}$, versus pressure ratio p_{02}/p_{01} for constant values of non-dimensional speed $N/\sqrt{T_{01}}$.

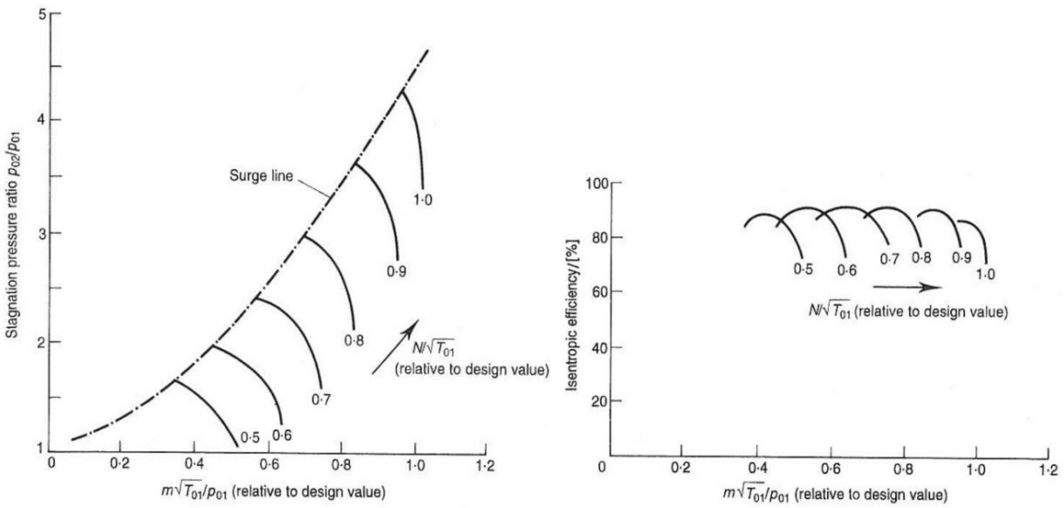

Figure 4.1, Axial compressor characteristics [6]

The operation area is restricted by surge and choking conditions. Surge is associated with a sudden drop in delivery pressure and may lead to violent aerodynamic pulsations. Rotating stall is another instability that also may occur, and can cause drop in performance and blade vibrations. It is difficult to distinguish surge from stall, and one phenomenon may lead to another, however operation in this area must be avoided. The compressor normally operates close to the surge line where the efficiency is high. At high speeds, the lines of non-dimensional speed, becomes vertical, and choked condition occur. At chocked condition, the mass flow cannot increase with a further decrease in downstream pressure[6]. This may be seen from the vertical lines in Figure 4.1. It is common that large axial compressor operate at choked flow, this decouples the dimensionless flow from the pressure ratio. Gas turbines are volumetric engines, and one may assume constant volume flow and axial velocity. With this assumption, an important relation for the compressor inlet may be defined

$$\frac{\dot{m}}{\dot{m}_{design}} = \frac{p}{p_{design}} \frac{\bar{R}_{design}}{\bar{R}} \frac{T_{design}}{T} \frac{A}{A_{design}} \qquad 4.4$$

This equation is based on the ideal gas law and the continuity equation, and is used to relate the actual condition with the known design point. The specific gas constant $\bar{R}$ may be replaced with universal gas constant divided by the molecular weight (MW). The gas turbine performance is proportional to the mass flow; hence equation 4.4 is important when performing a sensitivity analysis. A reduced pressure p, for example at high altitudes will lead to less mass flow and consequently less power output than at design. The same consequence occurs when the ambient air temperature increase.

The turbine characteristics are represented in a similar way as the compressor, illustrated in Figure 4.2. The performance is described by plotting non-dimensional mass flow $m\sqrt{T_{03}}/p_{03}$, versus pressure ratio p_{03}/p_{04} for constant values of non-dimensional speed $N/\sqrt{T_{03}}$. At a given pressure ratio, the turbine operate as choked, and the maximum non-dimensional mass flow is obtained. When choking occurs in the nozzles, the non-dimensional speed lines merges into one constant line. This is the most common choking condition, however if the rotors are choked, the non-dimensional mass flow increase slightly with increasing speed lines. When the turbine is connected to the other components, the operation area becomes more limited. From the Figure 4.2, and the efficiency plot, it is apparent that the overall efficiency η_t is fairly constant over a wide range of non-dimensional speeds and pressure ratio.

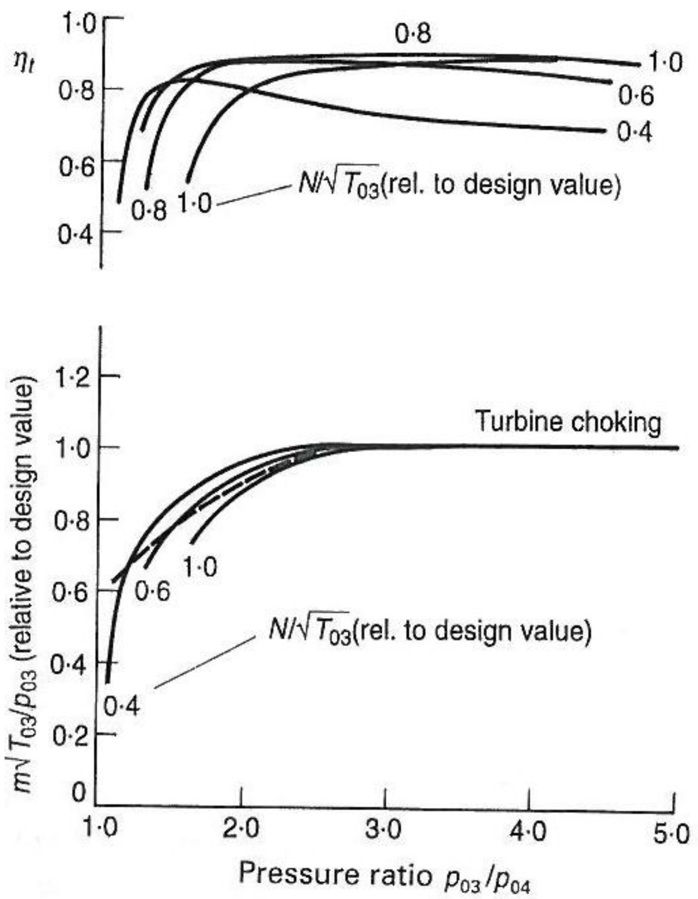

Figure 4.2, Turbine flow characteristics [6]

If considering a single shaft gas turbine used in generator drive as illustrated in Figure 2.3, component matching may be used to describe the performance at off-design. The compressor is matched to the turbine by the fact that the mass flow leaving the compressor equals the entry of the turbine. Bleeds from the compressor may be approximated equal to the fuel input, hence constant mass flow. As earlier stated, the gas turbine operates as choked, and the choked-nozzle equation is commonly used

$$\frac{p_3}{p_{3,design}} = \frac{m_3}{m_{3,design}}\sqrt{\frac{T_3}{T_{3,design}}} \qquad 4.5$$

This eases calculations, and relates the off-design parameters with design point. The turbine inlet pressure p_3 is from equation 4.5 proportional to the square of the turbine inlet temperature T_3, which may be used to determine the compressor pressure ratio[7].

The compressor and turbine are also coupled by the shaft, and run at constant speed due to the synchronous speed of the generator. If one neglect the pressure losses in the combustor, the pressure ratio in the compressor and turbine are also equal matched. At off-design, the task is to find an operation point where all these criteria are fulfilled, hence equilibrium. This is an iterative procedure, which will not be described.

The operating characteristics will vary significant with type of engine configurations at off-design. Compared to the single shaft gas turbine described previously, where power changes at constant speed and approximately at constant airflow, an engine with a free power turbine must operate at different compressor speeds, hence airflows as the power setting is changed. In modern engines, control systems plays a major role in part load[6].

4.1.1 Part Load

The power output in a combined cycle is dictated by the gas turbine performance. There are two methods of reducing the load of gas turbines[5]:

- Fuel control
- Variable inlet variable guide vanes and fuel control

The simplest method of adjusting the load is by lowering the fuel flow in the combustion chamber while keeping the air flow rate constant. This is preferred in simple cycle plants and gives the highest efficiency at part load[7]. When the fuel is reduced, the turbine inlet temperature drops, hence the exhaust temperature also decreases. This is a negative consequence for the steam cycle which is dependent on high exhaust temperature to achieve high efficiency. From the chocked nozzle equation, it follows that a reduction in turbine inlet temperature results in a reduction in pressure, and consequently less work output from the GT.

A more common configuration is reduction of load with VIGV`s in combination with fuel reduction. A row of VIGV`s may change the inlet angel of the flow into the first stage in compressor, and consequently reduce the mass flow. In combined cycles, VIGV`s are preferred at part load because of better steam cycle efficiency. When combining reduction of mass flow and fuel flow, it is possible to maintain a high TIT, and consequently high EGT. The VIGV`s may typically reduce the mass flow down to 40% GT load, where maximum position occur. At loads below this level, the TIT is reduced by reduction of fuel only, and the efficiency drops quicker.

The controlling of part load becomes more complicated when also considering emissions. The control system is designed to keep emissions within the limits over the complete operating range. The emissions of CO and NO_x are both a function of flame temperature, where NO_x set the upper limit and CO the lower limit of the temperature. This may be illustrated in Figure 4.3, and the operating range is showed.

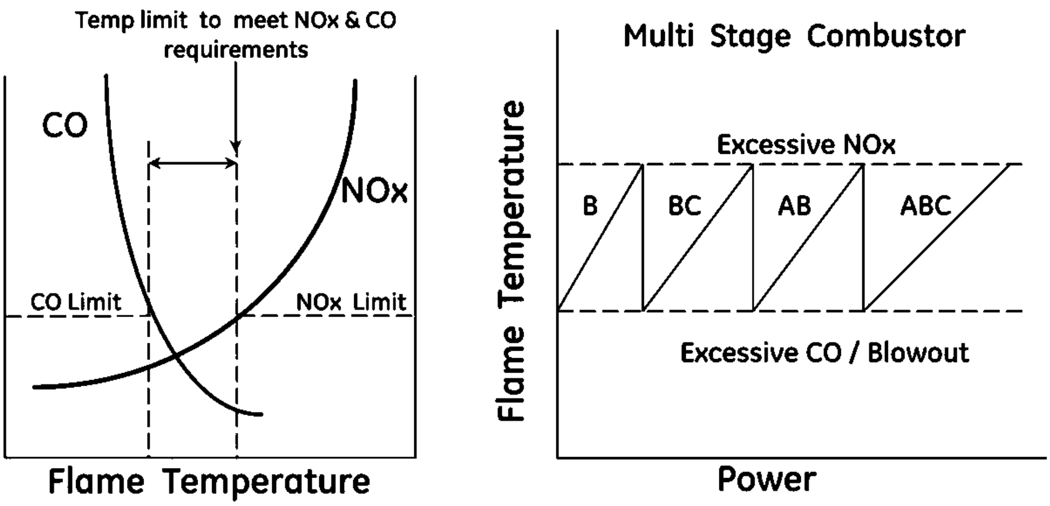

Figure 4.3, Flame temperature and stage combustor [18]

The LM2500+G4 aeroderivative gas turbine has a dry low emission (DLE) combustion system, illustrated in Figure 4.4. The system involves supplying fuel to different zones (Fuel Staging) of the combustor at different operating conditions to maintain the narrow flame temperature window. When the load is reduced, different configurations of the three premixers A, B and C are used. This may affect the exhaust temperature, hence the steam cycle performance.

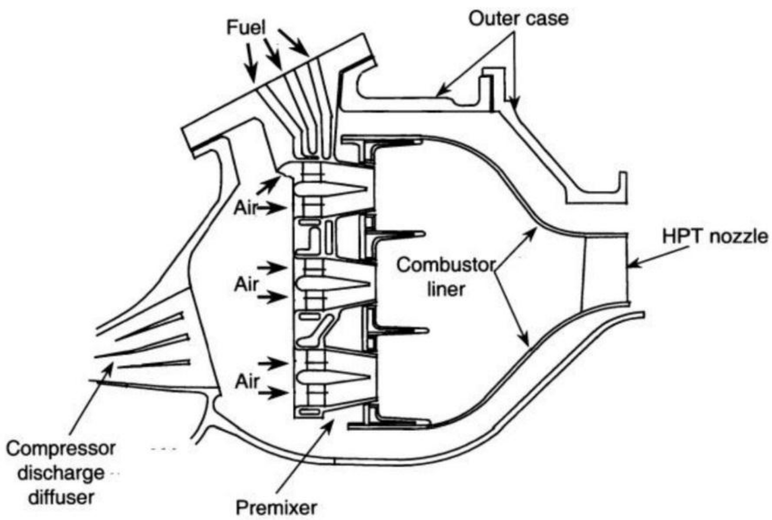

Figure 4.4, DLE annular combustor[18]

4.1.2 Ambient Temperature

The ambient temperature has a large effect of the gas turbine, both with respect of power output and efficiency. There are several reasons why the performance changes. The primary effect is related to the density of the air that changes with ambient temperature. For a constant volumetric engine, such as a gas turbine, reduction in density implies a reduction in mass flow and consequently reduced power output [5]. The relation between ambient temperature and inlet compressor mass flow is described in Equation 4.4. If assuming constant turbine inlet temperature, Equation 4.5 (Chocked nozzle) specifies that the turbine inlet pressure must increase with increasing mass flow. This change between two different temperatures is showed in Figure 4.5 in a temperature entropy diagram. A higher pressure ratio in both compressor and turbine leads to a higher efficiency.

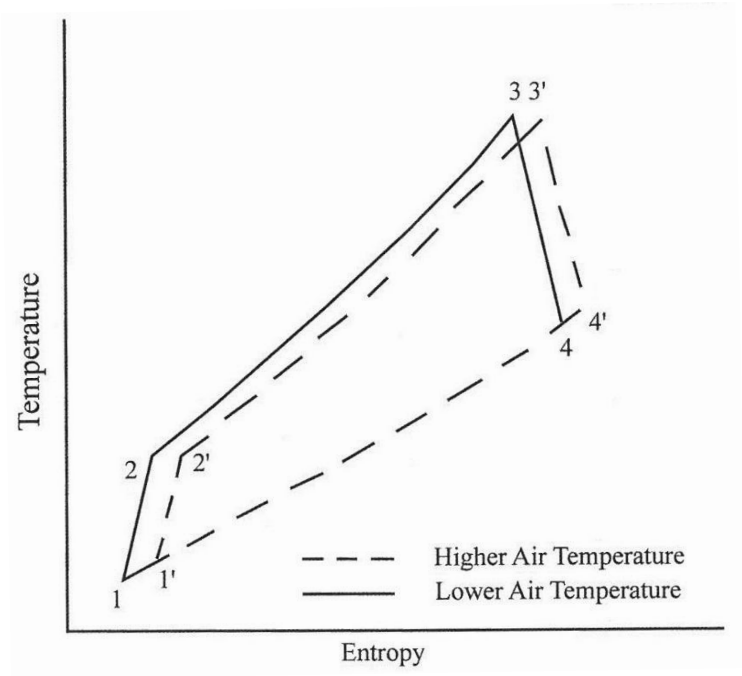

Figure 4.5, Temperature entropy diagram [5]

Another influence of lower ambient temperature is reduction of specific work at compressor. The specific compressor work is proportional to the inlet temperature (in K), and in the same manner both affect efficiency and power output. however this effect does not occur in the turbine[5].

The effect of changing ambient temperature on the exhaust gas temperature is of mayor concern considering the steam cycle. From Figure 4.5, higher ambient temperature gives a higher exhaust gas temperature, favoring the steam cycle performance. At the same time, higher ambient temperature cause a slightly reduction in average temperature of heat input from 2`to 3`. This behavior will according to the Carnot-efficiency, equation 2.1, result in a lower thermal efficiency for the total combined cycle at constant condenser temperature.

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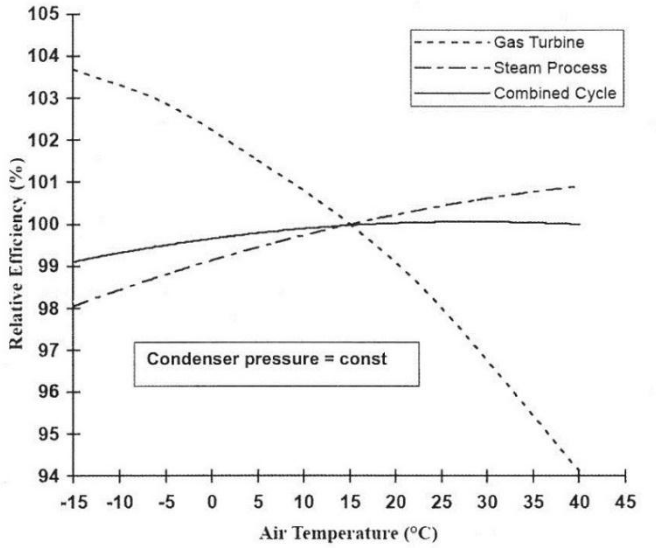

Figure 4.6, Relative efficiency of gas turbine, steam turbine and combined cycle with changing ambient temperature. Condenser pressure is constant [5]

In Figure 4.6, the relative change in efficiency for the GT, ST, and CC are presented with varying ambient air temperature and fixed condenser pressure. This figure supports the previous description with decreasing GT efficiency at increasing air temperature. The efficiency of the steam process increase due to higher exhaust temperature, hence the CC efficiency remains more unchanged. The relative change in power output differs from the trends in efficiency due to the effect of changing exhaust mass flow.

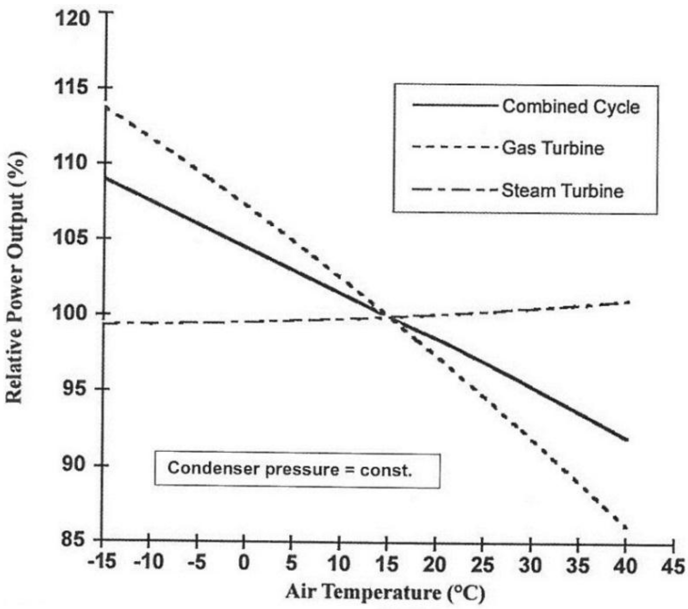

Figure 4.7, Relative power output of gas turbine, steam turbine and combined cycle with changing ambient temperature. Condenser pressure is constant [5]

This may be described in Figure 4.7. The behavior of the CC is more obvious in this plot, and is different from the efficiency plot due to the large energy developed by the gas turbine compared to the steam turbine.

4.2 Steam Turbine

At part load and off-design conditions the exhaust heat energy may change, which affect the steam production in the HRSG, and consequently the ST. The ST turbine is designed to follow the GT without control of the power output. Most steam cycle's s in combined cycle plants use sliding pressure operation down to 50% load. This ensures good utilization of the exhaust energy and high efficiency. Below 50% load, the live stream pressure is held constant by a valve at the steam turbine inlet. This introduces throttling losses, and increasing stack losses[5]. The sliding pressure operation is illustrated in Figure 4.8

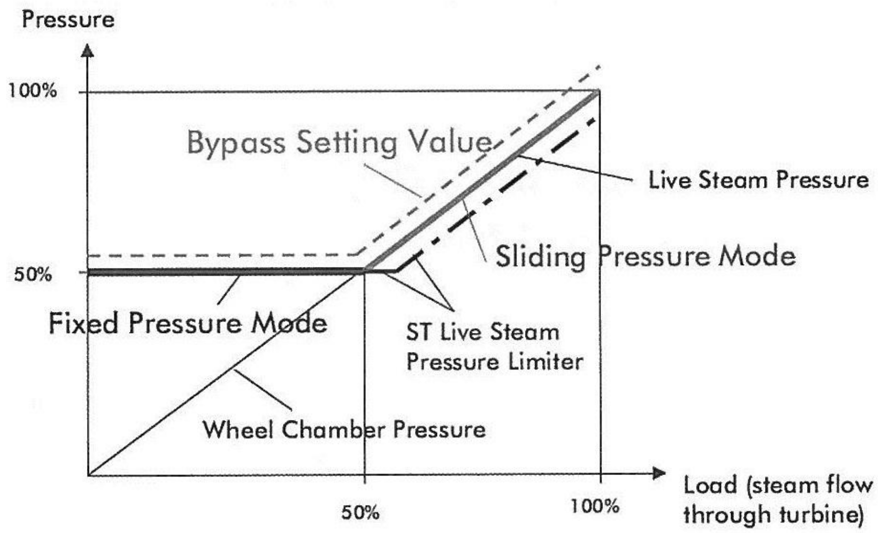

Figure 4.8, Sliding pressure operation [5]

At part load the ST have approximately constant volume flow. This implies that the velocity vectors remain unchanged, hence the efficiency is constant[7].

The steam mass flow through the ST at off-design may be calculated using the Law of Cones

$$\frac{\dot{m}_s}{\dot{m}_{s,0}} = \frac{\bar{V} \cdot p_\alpha}{\bar{V}_0 \cdot p_{\alpha,0}} \sqrt{\frac{p_{\alpha,0} \cdot v_{\alpha,0}}{p_\alpha \cdot v_\alpha}} \sqrt{\frac{1 - \left[\frac{p_\omega}{p_\alpha}\right]^{\frac{n+1}{n}}}{1 - \left[\frac{p_{\omega,0}}{p_{\alpha,0}}\right]^{\frac{n+1}{n}}}} \qquad 4.6$$

Where $\dot{m}_s$ is the steam mass flow, p is the pressure, v is the specific volume, $\bar{V}$is the average swallowing capacity, and n is the polytropic exponent. The suffix 0 is design point, α at ST inlet and ω at ST outlet.

For condensing turbines, where the pressure ratio is low and the ratio of swallowing capacity is almost 1, this equation may be simplified to

$$\frac{\dot{m}_s}{\dot{m}_{s,0}} = \sqrt{\frac{p_\alpha \cdot \rho_\alpha}{p_{\alpha,0} \cdot \rho_{\alpha,0}}} \Rightarrow \frac{p_\alpha}{p_{\alpha,0}} = \left[\frac{\dot{m}_s}{\dot{m}_{s,0}}\right]^2 \frac{\rho_{\alpha,0}}{\rho_\alpha} \qquad 4.7$$

Where ρ is the density. This equation may be used to determine the relation between live stream pressure and steam mass flow.

4.3 HRSG

At off-design, gas side pressure loss and heat transfer coefficient are important in the HRSG. The gas side pressure loss is proportional to the velocity squared, and consequently the exhaust mass flow. Higher pressure loss results in higher backpressure at the gas turbine, hence lower gas turbine efficiency.

5 Methodology

This chapter describes the methodology of the simulation process, including the associated simulation tools. A presentation of the two different CC configurations is given with the necessary input parameters and assumptions. Further, the off-design cases are described.

5.1 Simulation Software

Three computer programs from Thermoflow were used in this thesis, GT PRO, GT MASTER and PEACE (version 21). The combined cycle design cases were developed in GT PRO from user inputs and/or reasonable defaults set by the program. GT PRO computes heat and mass balances, system performance, and physical design of equipment.

GT MASTER is a separate program, used to simulate steady-state off-design performance. The calculation is based on a fixed plant developed in GT PRO, and inputs such as loads, and ambient conditions. The simulations in GT MASTER is connected to Microsoft excel with Thermoflow E-link. PEACE is a separate module that defines equipment designs, specifications and cost estimates for equipment [19].

5.2 Definition of CC Plants

The main task of the simulation process is to develop an offshore combined cycle configuration and run off-design simulations. The designed plant, called Offshore CC plant (OCC), is based on theory and literature about existing offshore CC configurations, which were described in Chapter 2 and 3 respectively. However, a reference plant is also developed for comparison and validation of the OCC. This plant is designed to achieve higher efficiency, and is referred to as High efficiency CC plant (HECC).

- **OCC:** The Offshore CC plant is designed for operation on offshore installations, consequently with the stringent requirements this involves. Because of practical design limitations in the simulation programs, not all the requirements can be fulfilled. A sensitivity analysis is performed to achieve the desired design, this is described in Chapter 6.1
- **HECC:** The High efficiency CC plant is designed to achieve high efficiency using the defaults set by the program. This plant was based on the same gas turbine and assumptions as in the Offshore CC plant. The final plant design is described in Chapter 6.1

5.3 Assumptions at Design Point

The assumptions at design point is summarized in the Table 5.1

Table 5.1 Assumptions at design point

Ambient temperature (°C)	15
Ambient pressure (bar)	1.013
Ambient relative humidity (%)	60
Altitude (m)	0
Line frequency (Hz)	60
Fuel type	CH_4
Fuel LHV at 298K (kJ/kg)	50047
Cooling water	Sea water
Cooling water temperature(°C)	10
Cooling water temperature rise ΔT (K)	10

5.4 Off-design Cases

Two off-design cases were simulated and reviewed in this thesis, part load and changing ambient temperature:

- **Part load:** This case considers how reduction from 100% to 40% GT load affects the performance of the CC plant. Part load is of major importance in applications where there may be required to run at low power settings. This could be the case of mechanical drive of water injection pumps where the load changes during the field's lifetime.
- **Ambient temperature:** This case reviews changes in ambient temperature from -10 to 28°C. Ambient temperature has a major impact on performance and is therefore considered.

5.5 Methodology and Simulation Process

The methodology and simulation process may be described as following:

- Inputs, such as ambient conditions and plant configuration is defined in GTPRO
- With the GTPRO simulation program, a sensitivity analysis is performed to develop the OCC plant design.
- The HECC plant is designed in GTPRO
- The two plants are compared and evaluated from results from GTPRO and PEACE.
- GTMASTER is used to develop off-design simulation files on background of the fixed plants developed in GTPRO. These files are linked to Microsoft Excel with E-LINK.
- E-LINK receives off-design inputs and simulates the plants performance.
- The results from Excel are used to review the off-design performance of the two concepts. Results and discussion is presented in Chapter 6

The process is illustrated in Figure 5.1.

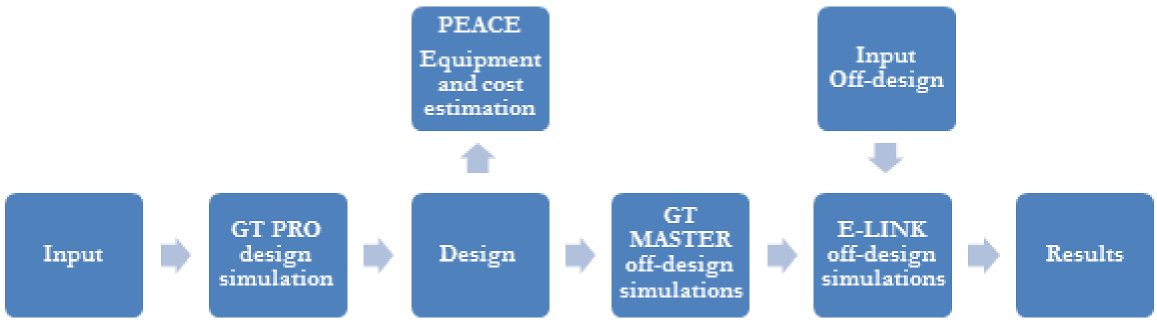

Figure 5.1, Simulation process

6 Results and Discussion

In this chapter the two CC plants developed in GT PRO are presented and discussed. The OCC plant is developed from a sensitivity analysis which is explained, and the HECC plant is developed from realistic defaults values in GTPRO. Further, the results from the off-design simulation is presented and evaluated. The parameters investigated were variations in gas turbine load and changes in ambient temperature. The last part of the chapter defines a practical and reliable operation area where the CC should operate.

6.1 OCC Plant Design

The design is chosen based on theory about combined cycles and how existing offshore plants are designed. As described earlier the requirements offshore are mainly reliability and availability, high power to weight ratio and small footprint. However, in GTPRO these requirements are not an option for optimization, whereas one may choose between either optimization of capital cost or plant efficiency. Consequently, the study is somewhat limited by constrains in the simulation program. As far as the simulating program allows, components and input properties are selected with emphasis to the requirements by offshore CC.

6.1.1 Plant Configuration

The design of an offshore CC plant differs somewhat from the GTPRO configuration for several reasons. Offshore, two or three gas turbines is typically connected to one HRSG with multiple inlets producing steam to one ST. In GTPRO the only option is one GT, one HRSG and one ST. This affects the operation of the plant, and the weight of the steam cycle. One may assume that larger steam cycle have higher power to weight ratio.

Gas turbines are used in both mechanical and generator drive on offshore installations, however in GTPRO generator drive is standard for both the GT and ST. This configuration is similar to the Snorre B CC, thus no steam extraction is considered in the simulation. 60Hz frequency has been chosen. For practical reasons the ST is not used in mechanical drive offshore. Any regulation concerning the GT will affect the steam turbine, making it difficult to regulate the ST independently.

With a complete electrical drive system, i.e. both ST and GT, in electric drive, the power distribution is flexible. This ensures better regulation and utilization of the energy. A disadvantage of this configuration can result in a lower efficiency, because of additional stages of energy conversion. Typically, GT`s are used in mechanical drive of large compressors. Such systems have high transmission efficiencies, but lack the flexibility to redirect or share power, which is important offshore. The choice of either of these systems is determined independently by the requirements on each individual platform.

The following configuration is used in GTPRO:

- One GT in generator drive
- One single inlet HRSG
- One ST in generator drive

6.1.2 Choice of GT

GTPRO consist of a large selection of gas turbines, both aeroderivatives and industrial. As previous described an offshore CC should use an aeroderivative GT type because of high power to weight ratio. The gas turbine selected for the OCC plant was the LM2500+G4 which are manufactured by General Electric. This engine may be delivered in a small lightweight package for offshore generation [20]. All three platforms, Oseberg D, Eldfisk and Snorre B use the previous version LM2500+ which the LM2500+G4 is developed from. The LM2500+ is widely used offshore because it is known to be reliable; hence this is also the case for the LM2500+G4. The LM2500+G4 compared to the LM2500+, has higher efficiency, pressure ratio and increased flow capacity. The engine is equipped with dry low emission (DLE) combustor which is important to meet the stringent emissions requirement in Norway. To achieve high availability of oil and gas production, the selected engine is capable of both liquid and gaseous fuel, so called dual-fuel. The engine use VIGV on several of the first stages.

GTPRO use a data-defined model to describe the performance of the LM2500+G4. The data is based on vendor's information and is listed in a Thermoflow data library. The model describes both design point and part load operation, which is important for the off-design simulations.

The gas turbine selection was based on the following:

- Aeroderivative and high power to weight
- High reliability and availability
- Dry low emission (DLE) combustion
- Dual-fuel
- Part load capability

The LM2500+G4 specifications is given in Table 6.1

Table 6.1 Full-load performance of the LM2500+G4 from GTPRO

Model	GE LM2500+RD(G4)
Generated power (MW)	32.6
Heat rate LHV (kJ/kWh)	9398
Efficiency LHV (%)	38.3
Pressure ratio	23.0
Air flow (kg/s)	90
Turbine speed (rpm)	3000
TIT (°C)	1288
EGT(°C)	526

6.1.3 Steam Cycle

In GTPRO, the normal simulation procedure is to design the steam cycle for high efficiency or low capital cost. On offshore installations, none of these are primary's. The main requirements concern weight, size and footprint. The steam cycle should be of a single pressure type to keep the weight and size low, however there is a trade-off between efficiency and weight. Therefore a sensitivity analysis is performed in order to find an optimum configuration. The following parameters are varied,

- Live stream pressure
- Live stream temperature
- Minimum pinch point temperature difference
- Condenser pressure

The plant design should achieve following criteria according to current offshore CC plants:

- Approximately 50% plant efficiency according to the Oseberg B platform
- The weight of a single inlet HRSG skid design for a LM2500+ should be around 120 ton
- According to the Snorre B platform, the steam turbine skid, including a 17.3 MW steam turbine generator set, condenser and auxiliary equipment should weigh 190 tons dry

All the offshore CC`s described previously use a single pressure HRSG with vertical gas flow. They are mounted directly over the gas turbine stack, which minimizes the footprint and space requirements. A vertical HRSG`s may be the most practical because of less bends and ducting are required. One may also assume that a single pressure HRSG is less exposed to failure than a more complex HRSG with three pressure levels and reheat.

To obtain high availability of the GT, the steam cycle is mounted with by-pass stack. This allows continuously operation of the gas turbine, independent of the steam cycle. This equipment is used on the Oseberg D platform, and is mounted on the HRSG skid.

The steam turbine is a condensing type, with no steam extraction. This ensures the highest power output. As described in Chapter 2, the steam quality should not be lower than 84% when exiting the ST.

The cooling system used offshore is typically a condenser (Once-through water cooling) where seawater is used as cooling medium, this is also selected in GTPRO. Assume that the seawater is 10 °C and the allowable cooling water temperature rise is 10K.

Following configuration is used for the OCC in GTPRO:

- Single pressure
- Vertical HRSG, forced circulation
- By-pass stack
- Condensing ST
- Condenser

6.1.4 Sensitivity analysis

As a preliminary design, conservative values were assumed to obtain a compact design. The pinch temperature was set to 30 K, and the condenser pressure to 0.1bar.

The effects of live stream pressure on the steam cycle gross efficiency are illustrated in Figure 6.1. The efficiency increases with live stream pressure, and reaches a maximum value around 24 bar (450 °C). At this point, changes in efficiency are moderate, and one may assume that pressure between 17 to 28 bar is possible values. In this interval, the improvement in efficiency is mainly driven by LST. At higher pressures, less steam is produced in the HRSG. However, the HRSG dry weight increases slightly. From 10 to 40 bar the weight rises with 6 to 7ton independent of LST, this may be a result of lower average temperature difference, hence more heat transfer area, and/or thicker tube walls. Because of lower steam production, the ST and generator weight is reduced with about 8 ton. From these observations one may conclude that the overall steam cycle weight is not much influenced by the pressure. Further the LSP is set to 24bar.

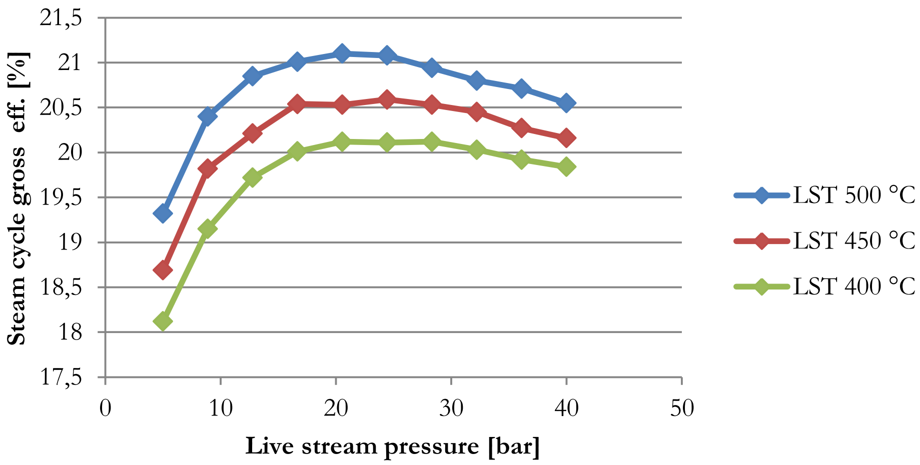

Figure 6.1, Live stream pressure variation with steam cycle efficiency at different live stream temperatures

Live stream temperature has a significant impact on the steam cycle efficiency, this is illustrated in Figure 6.2. The enthalpy at the ST inlet is function of both temperature and pressure, however the temperature is the dominant parameter.

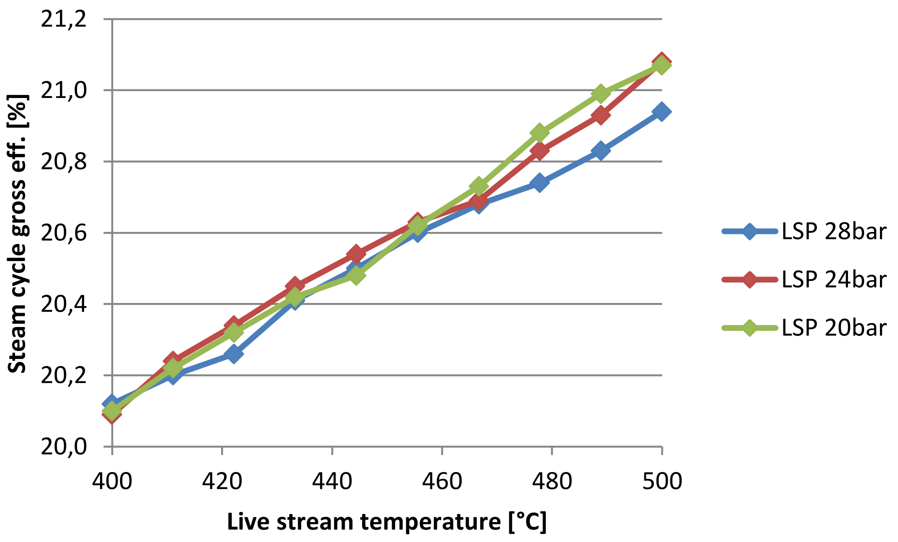

Figure 6.2, Live stream temperature variation with steam cycle efficiency at different live stream pressures

The LST have almost a linear relationship with the efficiency, and a high LST result in a larger enthalpy drop in ST. A negative aspect of higher LST is the exponential increase in superheater heat transfer area due to lower temperature difference between the hot exhaust gas and steam. Consequently, the HRSG dry weight increase, as illustrated in Figure 6.3. The LST is set to 470°C which is the point where the change in weight becomes more linear. This is to avoid too much loss in efficiency.

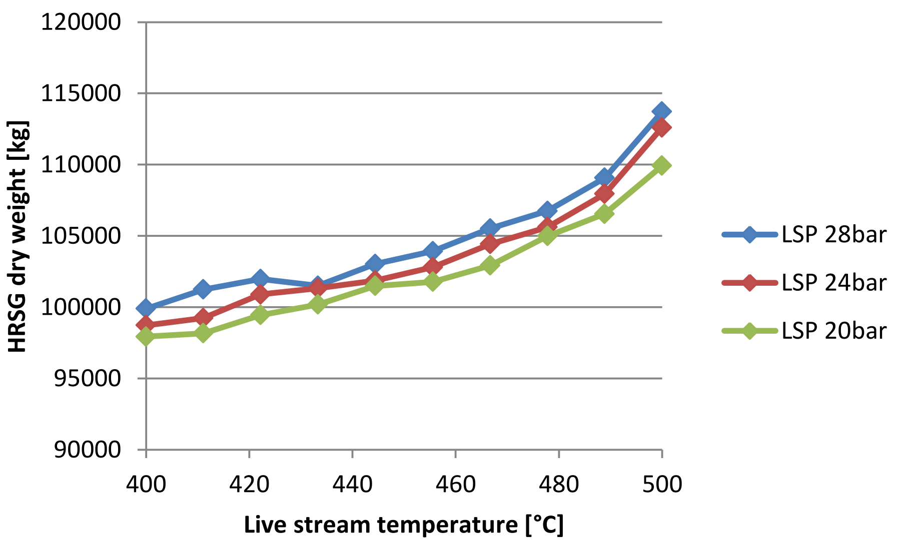

Figure 6.3, Live stream temperature variation with HRSG dry weight at different live stream pressures

Further the approach temperature is set to 12K, which would result in a lower economiser surface. A large approach temperature reduces the possibility of steaming, and gives higher reliability at off-design.

ΔT pinch is the single most important parameter on the HRSG weight because the heat transfer surface area is inversely proportional to the ΔT pinch. This relation may be seen from Figure 6.4. Increasing ΔT pinch from 5K to 28K results in an approximately reduction of 53 ton which is substantial.

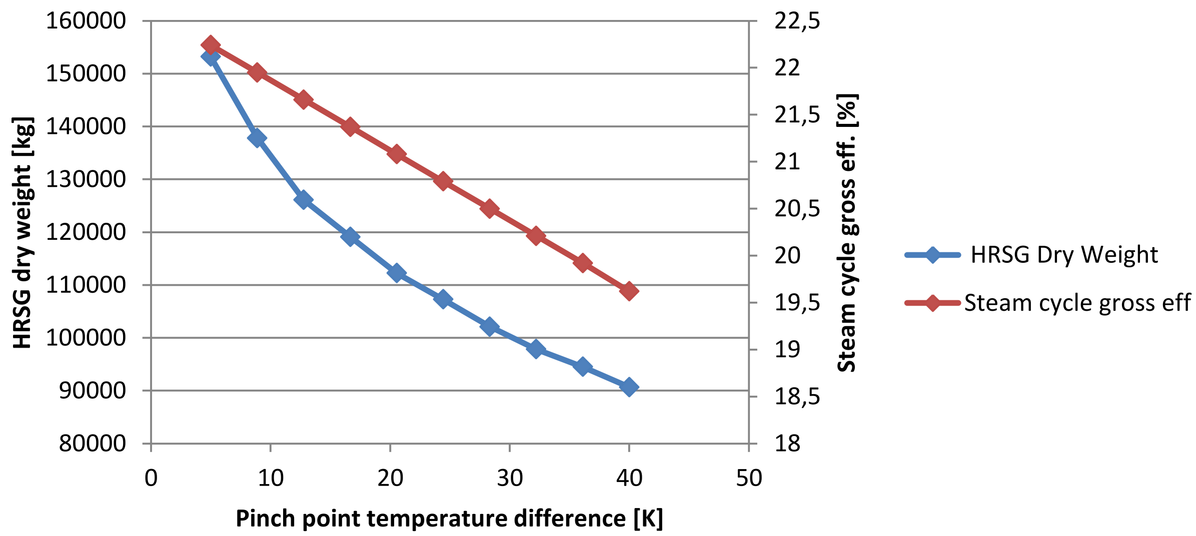

Figure 6.4, Variation of pinch point temp. diff. with HRSG dry weight and steam cycle gross eff. at 24bar/470°C

However when increasing ΔT pinch, less heat is recovered and the steam production drops. The steam cycle efficiency is a linear function of the pinch point difference, shown in Figure 6.4. Below 20K pinch temperature difference the HRSG dry weight decay rapidly, and should be avoided. A value of 25K is assumed to give a good trade-off between efficiency and weight, and is selected for the given design point.

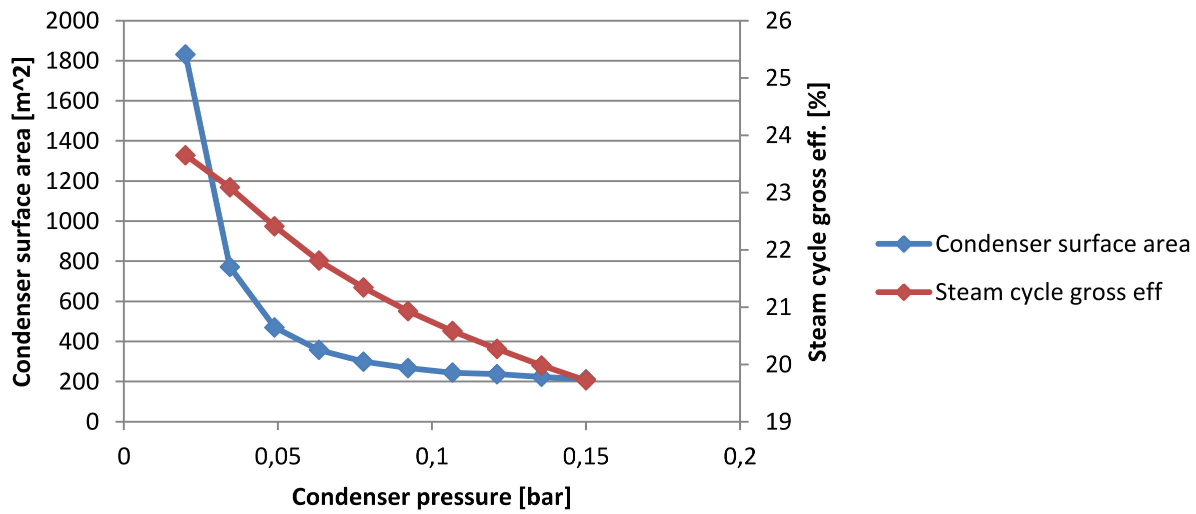

Figure 6.5, Variation condenser pressure on condenser surface area and ST gross efficiency

The effect of varying condenser pressure on the condenser surface area and steam cycle efficiency is presented in Figure 6.5. A low condenser pressure results in a high efficiency, but consequently a high surface area. At condenser pressure below 5 mbar, the change in condenser surface area is excessive. From 0.02 bar to 0.08 bar the change in area is approximately 1500 m^2, which correspond to a change in weight of 18 ton.

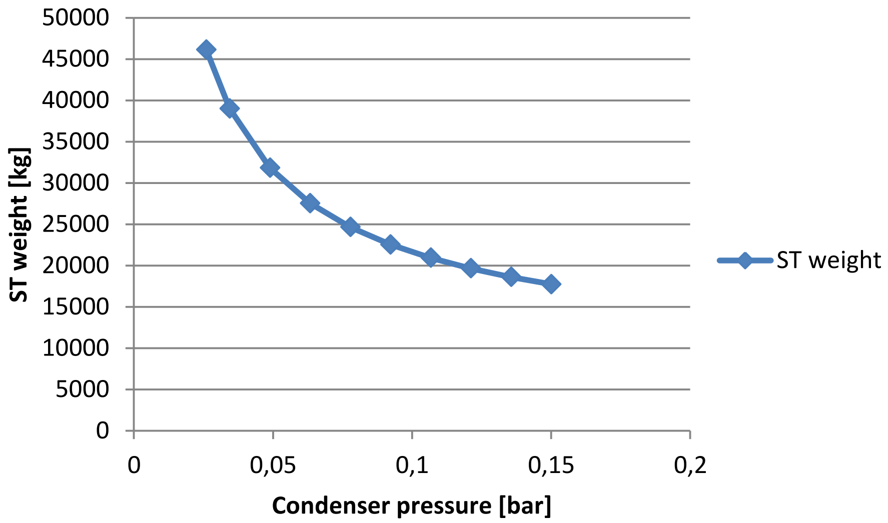

Figure 6.6, Variation of condenser pressure on ST weight

Another effect of changing condenser pressure is on the ST weight. Lower condenser pressure results in a larger ST with higher pressure ratio and power output. In the same interval, 0.02 to 0.08 bar the difference in ST weight is 22 ton. The condenser pressure is set to 0.08bar which results in a steam cycle gross efficiency of 21.3%. The steam quality is 92%. This is within the limit of 84%, which was described in Chapter 2.4.

Ideally, finding the optimum steam cycle configuration is an iterative procedure. However, this sensitivity analysis shows that the trimming of operation parameters is essential for low steam cycle weight. Following values were determined:

- Live stream pressure: 24 bar
- Live stream temperature: 470 °C
- Minimum pinch point temperature difference: 25 °C
- Condenser pressure: 0.08 bar

Compared to the requirements stated earlier, the developed model, OCC plant, show good agreements. The plant efficiency is 50.3%, which is slightly higher than the criteria of 50% plant efficiency. The weight of the OCC plant compared to the weight requirements are illustrated in Figure 6.7

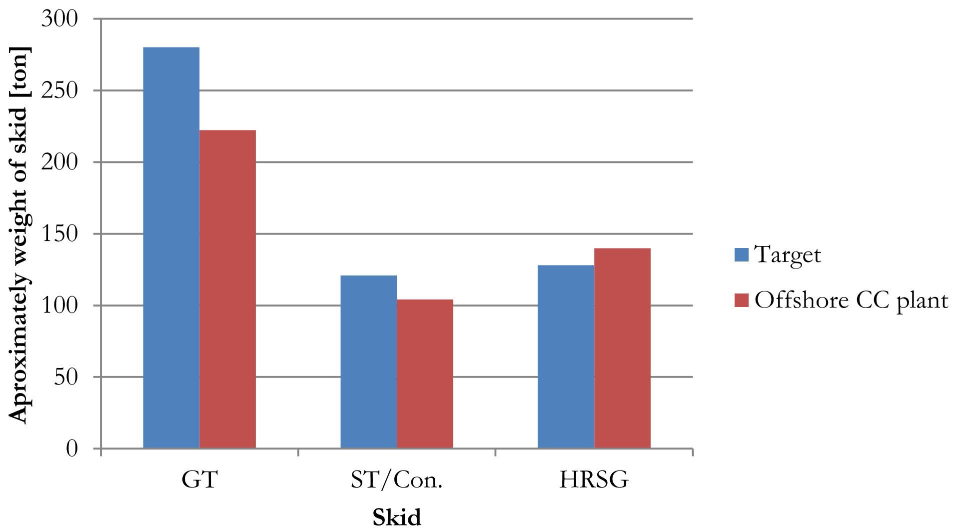

Figure 6.7, Comparison of weight between target and the Offshore CC plant

The Target weight is based on power to weight ratio of the requirements, and is corrected for the power output for the OCC plant. Both the ST/Con. skid and HRSG skid are within reasonable values. The only value that shows disagreement is the weight of the GT skid. However, the gas turbine is of a package type with specifications from the data library in Thermoflow, and should therefore be realistic. The new LM2500+G4 is a updated model of the LM2500+, and weights more.

6.2 HECC Plant Design

The HECC plant design is based on the same gas turbine as the OCC plant, and use the same assumptions listed in Table 5.1. However in this case there were no weight considerations and the plant was designed to have higher efficiency. Instead of one pressure, a dual pressure steam cycle is chosen. The HRSG is horizontal with forced circulation and the steam turbine is of a condensing type. Defaults values were used in GT PRO if not specified.

6.3 Design Point Performance

Schematic illustrations of both plants are given in Figure 6.8 and Figure 6.9. As described earlier, the combined cycles consist of one gas turbine, a HRSG, a ST and a condenser. The Offshore CC plant has four heat exchanger sections while the High efficient plant has four sections per pressure level. Additionally both plants are designed with a low temperature economizer (LTE) before the deaerator. The HECC plant use recirculation of feedwater from the LTE exit to the inlet to obtain higher LTE surface temperature than the exhaust dew point. The exhaust dew point is 41 °C, and recirculation should ideally been used on the OCC plant also. A fraction of the steam produced in the HRSG, is used for deaeration.

The steam cycle parameters are given in Table 6.3, and show the differences between the two simulated plants at design point. The main differences are; pressure levels, live stream temperature, condenser pressure, minimum pinch point, and approach temperature.

Table 6.2 Summary of design parameters

Simulation	Offshore CC plant	High efficiency CC plant
Water/steam cycle		
Cycle type	Single-pressure	Dual-pressure
Pressure level IP/HP (bar)	24	6.9/55
Temperature IP/HP (°C)	470	258/510
Deaerator pressure (bar)	1.054	1.054
Condenser pressure (bar)	0.08	0.048
HRSG		
Minimum pinch point IP/HP (K)	25	8/8
Approach subcooling IP/HP (K)	12	2/2
HRSG pressure drop (mbar)	22	22
Minimum stack T (°C)	88	88
Feedwater preheating	No	Yes

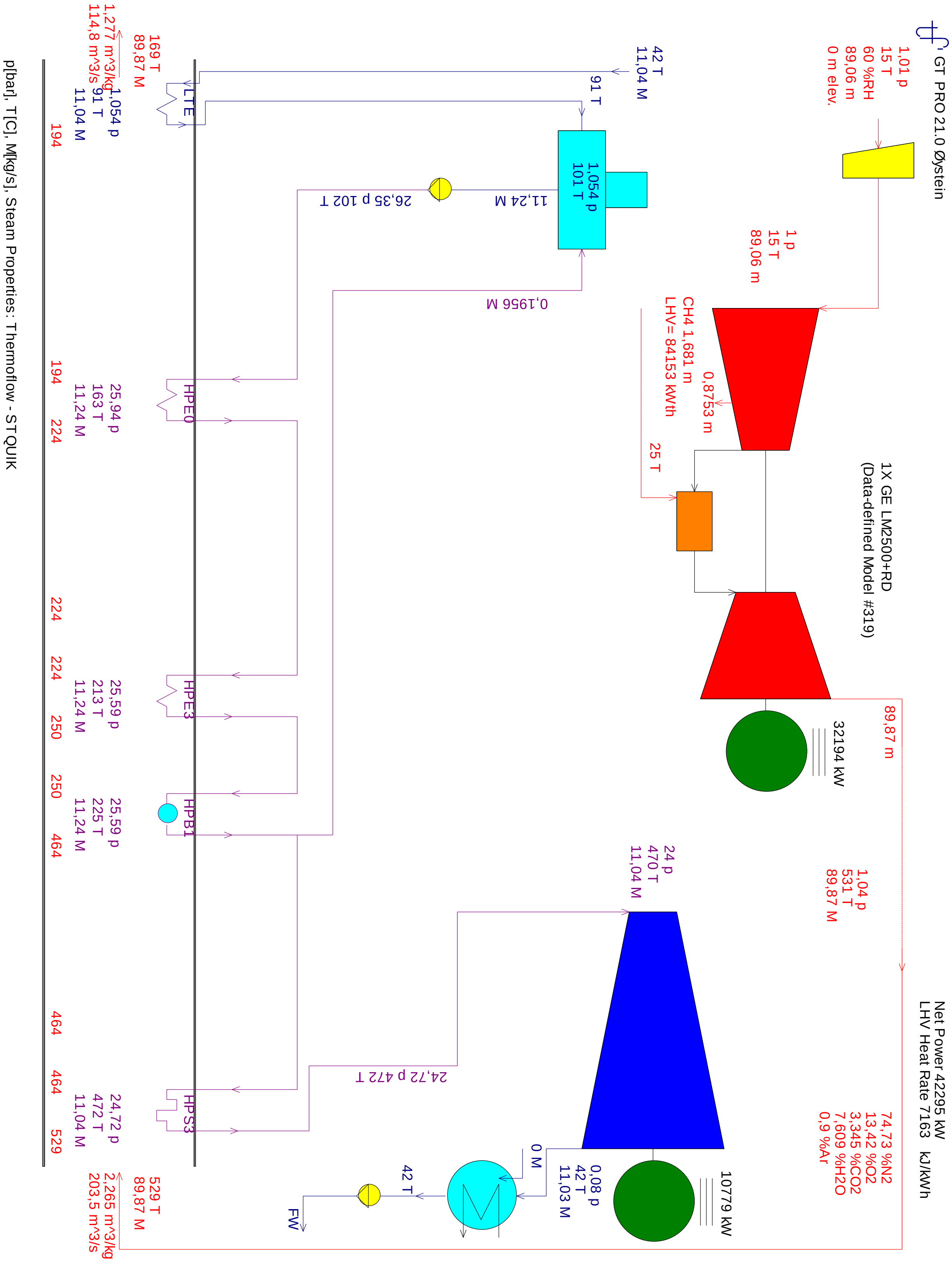

Figure 6.8, Offshore CC plant schematic in GTPRO. Annotations: pressure p [bar], mass flow m/M [kg/s], temperature T [°C]

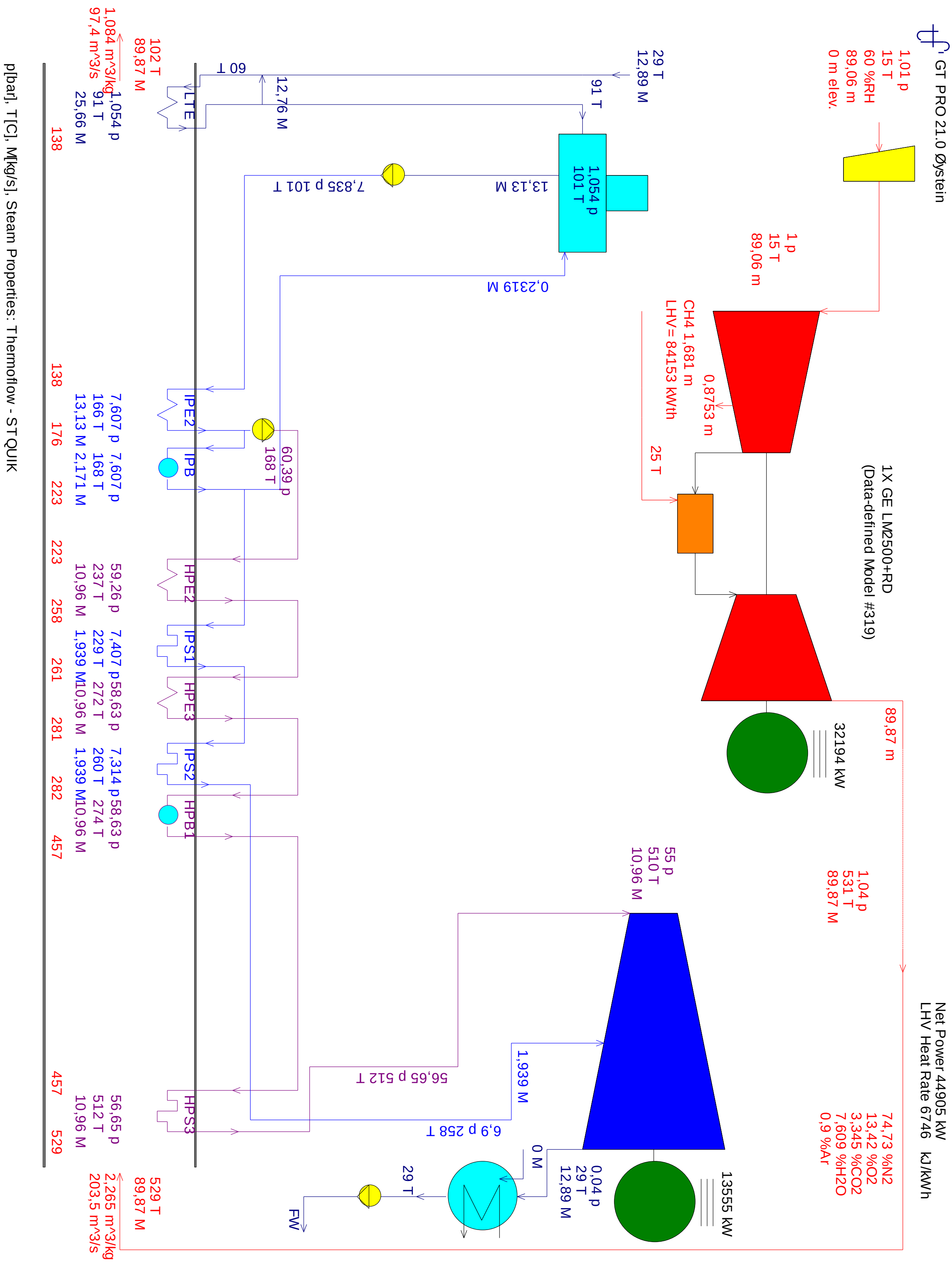

Figure 6.9, High efficient CC plant schematic in GTPRO. Annotations: pressure p [bar], mass flow m/M [kg/s], temperature T [°C]

6.4 Summary

The simulations of the two plants show expected results. With dual pressure and optimizing for high efficiency in GTPRO the steam cycle achieve higher power output. The individual performances of the two designed plants are summarized in Table 6.3. The High efficient CC plant has 2.4 MW more in net power output than the Offshore CC plant, and the plant net efficiency is with 2.8 % higher. However, the weight of the HECC increases with 209 ton compared to the OCC, which is significant.

Table 6.3 Summary of design point performance

Simulation	Offshore CC plant	High efficiency CC plant
Gas turbine		
Natural gas LHV input (MW)	84.2	84.2
Gross power output GT (MW)	32.2	32.2
Gross efficiency (%)	38.3	38.3
Exhaust flow (kg/s)	90	90
EGT(°C)	531	531
HRSG		
HRSG efficiency (%)	69	81.1
Steam mass flow IP/HP (kg/s)	11.04	10.96/1.94
Stack temperature (°C)	170	104
Steam turbine		
Gross power output ST (MW)	10.8	13.3
ST efficiency (%)	30.1	32.0
Steam quality at outlet (%)	91.9	89.5
Plant		
Gross power output (MW)	43.0	45.5
Gross power output (% of LHV input)	51.1	54.1
Net power output (MW)	42.3	44.7
Net power output (% of LHV input)	50.3	53.1
Weight (ton)	426	635

Since both plants use the same gas turbine, the difference in performance is only due to the steam cycle. The dual-pressure cycle utilize more of the exhaust heat, which may be seen from the stack temperature. The temperature is 170 °C for the single pressure, compared to 104°C for the dual pressure. And consequently, the HRSG efficiency is much higher. With two pressure levels and lower pinch temperature, also the exergetic losses are less in the HRSG. This increases the ST power output, and may be one of the reasons for higher ST efficiency. Another important factor concerning the power output is the condenser pressure; this was illustrated in Figure 6.5 in the sensitivity analysis.

One may conclude that the two plant configurations clearly are different with respect to efficiency and weight, hence fulfil their purpose. The problem then remains to find the performance at off-design.

6.5 Off-design Simulations

So far, the performance of the CC has been related to full load. Further the off-design cases are described and discussed.

6.5.1 Part Load Operation

Both plants are using the same GT, and hence the topping cycle behaviour will be the same at part load. Since the gas turbine is a data-defined model, only input and output values, such as inlet air flow and EGT may be obtained. This hardens the evaluation of the GT and questions may arise.

In Figure 6.10, the effect of reduced gas turbine load on the gas turbine performance is shown. The figure describes relative change in fuel and exhaust mass flow, relative change in efficiency and changes in EGT. Ideally data such as TIT, pressure ratio and VIGV angle should be described, but are unavailable.

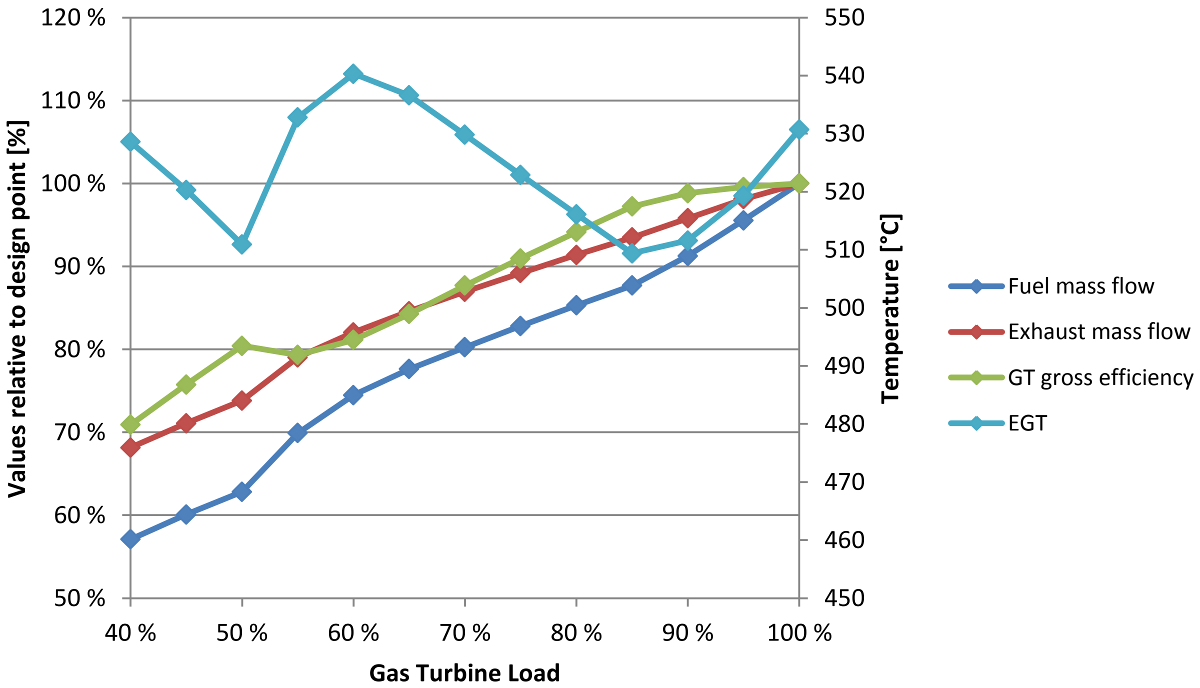

Figure 6.10, Part load performance of the gas turbine, GE LM2500+RD(G4)

The impact of VIGV`s may be seen from the reduced exhaust mass flow, which is varied down to 40% GT load. With a combination of both reduced air flow and reduced fuel flow, the fuel air ratio remains high, hence the TIT and the EGT remains high. Since the pressure ratio decreases with decreasing mass flow, according to theory in Chapter 4, the gas turbine efficiency drops. The strange behaviour in EGT may be related to the DLE combustion system which the GE LM2500+G4 engines use. The flame temperature has an upper and a lower limit, respectively due to NO_x and CO emissions, which must be maintained. From full load, the EGT decreases until the CO emission limit occurs at 85% load. Then a new stage is activated, increasing the EGT until the limit of NO_x emission is reached at about 60% GT load. This staging combustion system was illustrated in Figure 4.4. It is worth noting that the EGT is directly connected to the steam cycle performance, and consequently all the following results are affected by the variations in EGT.

In Figure 6.11, the relative efficiencies of the two plants are plotted against gas turbine load. The relative gas turbine efficiency will be the same in the two cases.

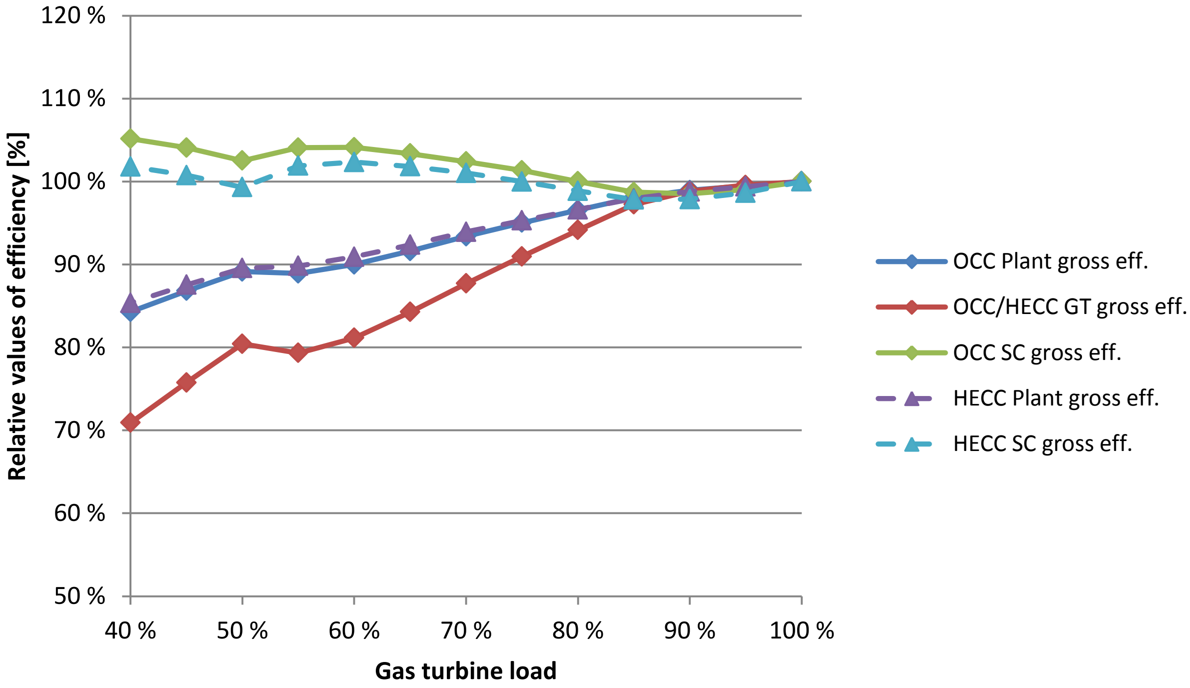

Figure 6.11, Variation of gas turbine load on relative efficiency for the OCC and the HECC plant

The relative steam cycle efficiencies show a clear connection with the variations in EGT from Figure 6.10. An increasing EGT results in increasing SC efficiency, and opposite. In general, a high EGT, in combination whit reduced exhaust gas flow result in higher SC efficiency at part load. As the gas turbine efficiency drop, the increasing SC efficiency contributes to higher plant efficiency. At 60% GT load, the relative gas turbine efficiency is 81% compared to the relative plant efficiencies of about 90%. Originally, the gas turbine delivers about two thirds of the total power output. As the GT load is reduced, the ratios of ST to GT power output changes towards more ST power output. Offshore, this is a positive effect. If a gas turbine in mechanical drive is running at part load, the ST may still offer high availability of electricity production.

The differences between the OCC and HECC relative SC efficiencies are of special interest. The graphs show identical behaviour, thus the relative SC efficiency increase more in OCC plant than in the HECC plant. There is a larger potential for heat recovery in the OCC plant than in the HECC plant, which already has been optimized for high. The main reason is probably because of the drop in condenser pressure is relative higher in the OCC than in the HECC.

For the relative plant efficiency, the HECC plant is conversely higher than for the OCC plant. The may be described from the generally higher ST power output in the HECC, which at all GT loads is higher. In Table 6.4, data from the two plants at 100 and 60% GT load is given. At design the OCC plant has a penalty in net efficiency of 2.8%. This penalty increases marginally to 3% at 60% part load. This is an insignificant increase, and the difference in efficiency between the plants may be reviewed as constant.

Table 6.4 Part load performance at 60% GT load compared to design point

Simulation	Offshore CC plant		High efficiency CC plant	
Gas turbine load (%)	100	60	100	60
Gas turbine				
Natural gas LHV input (MW)	84.2	62.7	84.2	62.7
Gross power output GT (MW)	32.2	19.5	32.2	19.5
Gross efficiency (%)	38.3	31.0	38.3	31.0
Exhaust flow (kg/s)	90	74	90	74
EGT(°C)	531	540	531	540
Water/steam cycle				
Pressure level IP/HP (bar)	24	20.7	6.9/55	5.8/47.6
Temperature IP/HP (°C)	470	470	258/510	254/510
Minimum pinch point IP/HP (K)	25	20	8/8	6.1/5.9
Steam mass flow IP/HP (kg/s)	11.0	9.5	1.9/11.0	1.0/9.5
Condenser pressure (bar)	0.08	0.064	0.048	0,040
Steam quality at outlet (%)	91.9	92	89.5	89.7
Stack temperature (°C)	170	157.2	104	98
HRSG efficiency (%)	69	72	81.1	82.6
Gross power output ST (MW)	10.8	9.3	13.3	11.4
ST efficiency (%)	30.1	30.0	32.0	32.0
Plant				
Gross power output (MW)	43.0	28.8	45.5	30.8
Gross power output (% of LHV input)	51.1	46.0	54.1	49.2
Net power output (MW)	42.3	28.2	44.7	30.1
Net power output (% of LHV input)	50.3	45.0	53.1	48.0

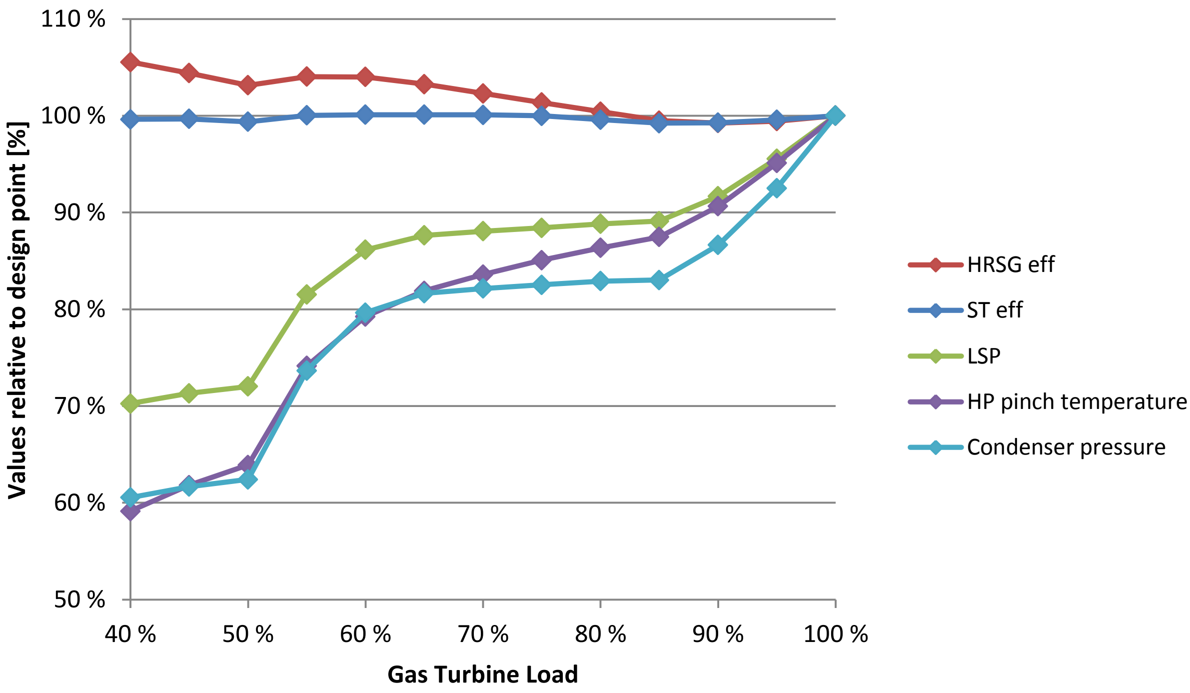

Figure 6.12, Variation of gas turbine load on steam cycle performance, OCC plant

There are several reasons for the increasing SC efficiency. The SC efficiency is a function of both HRSG efficiency and ST efficiency. However the steam turbine efficiency remains almost constant at part load. In Figure 6.12, some of the main parameters affecting the OCC steam cycle performance are shown. Results have shown that both cycles have almost the same behaviour, hence the HECC plot is not shown.

When the exhaust mass flow decreases, less heat is available to the steam cycle. The HRSG is designed with a given heat transfer area, which becomes over dimensioned at lower heat loads. This results in lower pinch point temperature, and better HRSG efficiency. In The relative steam cycle efficiencies show a clear connection with the variations in EGT from Figure 6.10. An increasing EGT results in increasing SC efficiency, and opposite. In general, a high EGT, in combination whit reduced exhaust gas flow result in higher SC efficiency at part load. As the gas turbine efficiency drop, the increasing SC efficiency contributes to higher plant efficiency. At 60% GT load, the relative gas turbine efficiency is 81% compared to the relative plant efficiencies of about 90%. Originally, the gas turbine delivers about two thirds of the total power output. As the GT load is reduced, the ratios of ST to GT power output changes towards more ST power output. Offshore, this is a positive effect. If a gas turbine in mechanical drive is running at part load, the ST may still offer high availability of electricity production.

The differences between the OCC and HECC relative SC efficiencies are of special interest. The graphs show identical behaviour, thus the relative SC efficiency increase more in OCC plant than in the HECC plant. There is a larger potential for heat recovery in the OCC plant than in the HECC plant, which already has been optimized for high. The main reason is probably because of the drop in condenser pressure is relative higher in the OCC than in the HECC.

For the relative plant efficiency, the HECC plant is conversely higher than for the OCC plant. The may be described from the generally higher ST power output in the HECC, which at all GT loads is higher. In Table 6.4, data from the two plants at 100 and 60% GT load is given. At design the OCC plant has a penalty in net efficiency of 2.8%. This penalty increases marginally to 3% at 60% part load. This is an insignificant increase, and the difference in efficiency between the plants may be reviewed as constant.

Table 6.4, 100% GT load is compared to 60% GT load for both plants. The HRSG efficiencies increase from 69 to 72% in the OCC, and from 81.1 to 82.6% in the HECC. In both cases the LST remains constant due desuperheating before the ST.

The relative change in LSP is equal to the relative change in steam mass flow. From design to 60% load the total mass flow is reduced with 14% in the OCC plant and 19% in the HECC plant. Because the steam production drops, the LSP is reduced with sliding pressure regulation to maintain constant volume flow at the ST inlet. At constant volume flow, the velocity vectors remains unchanged from design to part load operation. Hence, the ST efficiency is almost constant. The HECC plant has at all times a higher ST efficiency than the OCC. This may be a result of less exergy loss in the HRSG, and higher LST and LSP, and lower condenser pressure. The ST efficiency is typically 30 and 32% for the OCC and HECC, respectively.

Also, the condenser pressure is reduced with reduced steam mass flow. The ST will expand to a lower pressure because of less steam that needs to be condensed. This result in increased specific power output in the ST, however the effect of reduced steam flow is primary, and the ST output decreases. Lower condenser pressure, result in lower water inlet temperature at LTE. This gives higher energy utilization in the cold end of the HRSG, but even higher risk of flue gas condensation. At 60% load, the OCC has a LTE inlet temperature of 37.4 °C, which is about 4K below the flue gas dew point. This will be even lower at reduced GT load.

Other constrains are typically steam quality at ST outlet. In both plants, this value has increased slightly compared to design point. At 60% load, the steam qualities are 92 and 89.7% for the OCC and HECC plants, respectively. The variation in LSP is dominant over the lower condenser pressure, hence higher steam quality.

6.5.2 Ambient Temperature

Ambient temperature has a major effect on the gas turbine and the combined cycle performance. Because both systems use water cooled condensers, the gas turbine is the only component that is affected directly. In Figure 6.13, the gas turbine performance at changing ambient temperature is presented. The values are relative to design point at 15 °C and apply for both plants.

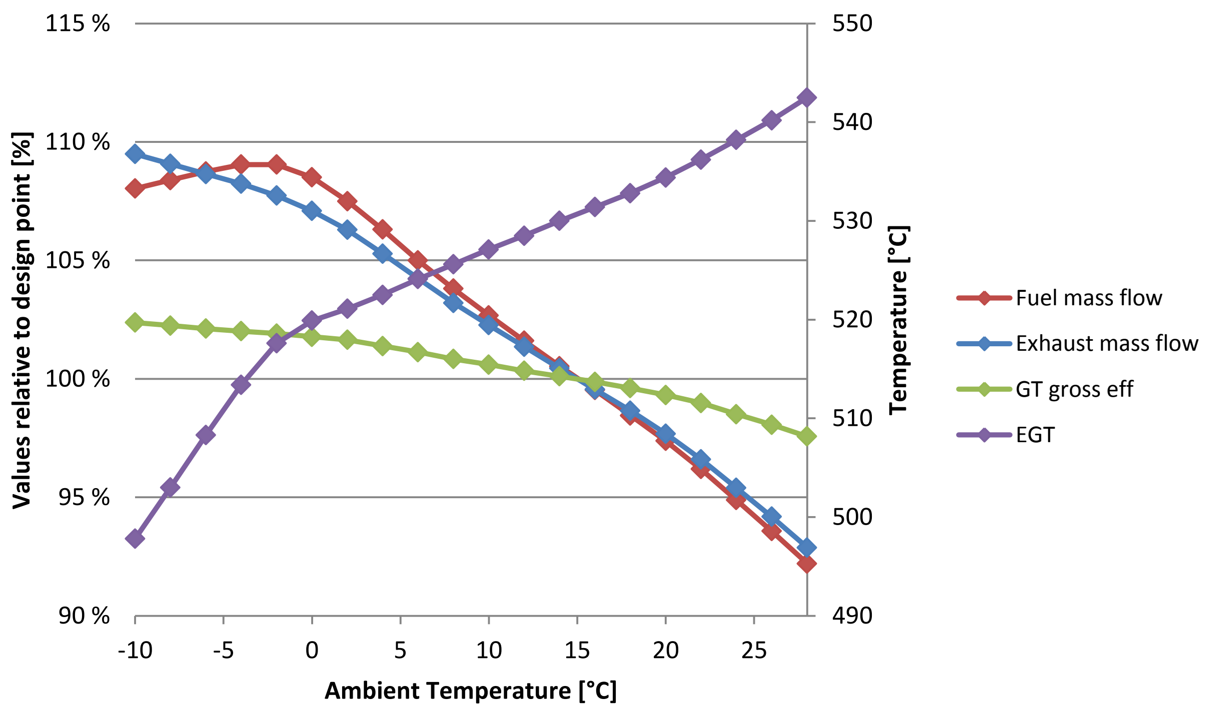

Figure 6.13, Effect on ambient temperature on the gas turbine performance

The effect of the ambient temperature on the gas turbine efficiency is evident from the figure. Lower ambient temperature results in higher mass flow at compressor inlet, and consequently a higher pressure ratio. And the opposite happens at higher ambient temperature. If the ambient temperature changes with 5K in either direction, the mass flow changes with approximately 2 kg/s. The change in exhaust mass flow and EGT corresponds well to the described theory in Chapter 4.1. However, the kink that occurs at 0 °C may not be described from thermodynamics. This is probably a mechanical limitation of the engine. To avoid excessive loads, which may damage the engine, the control system reduces the fuel input. This stabilizes the power, but reduces the EGT even more.

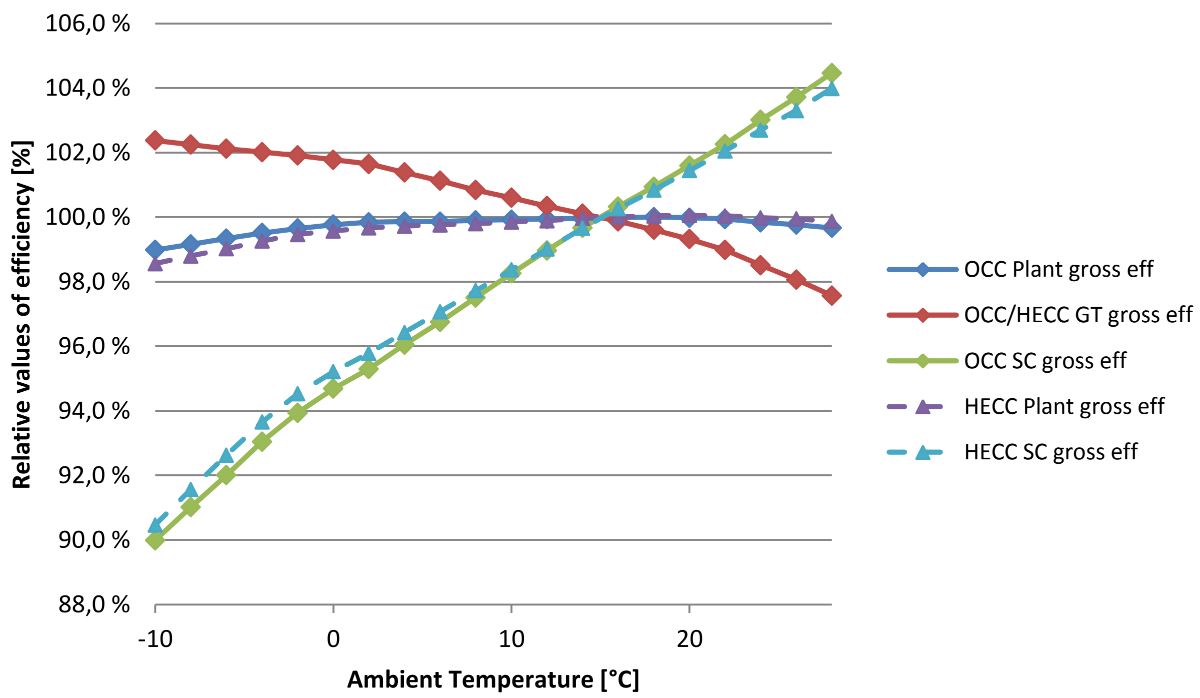

Figure 6.14, Effect on ambient temperature on. Values are relative to design point

Figure 6.14 illustrate the relative efficiency of the gas turbine, steam cycle, and the CC plant whit changing ambient temperature. The SC gross efficiency declines with lower ambient temperature and the GT gross efficiency increases. From 15 to 0°C, the relative SC gross efficiency drops with 5% for the OCC and HECC plant. The ST efficiency decrease because of lower LST, and the HRSG efficiency decreases because a under dimensioned HRSG. With higher exhaust flow rate, the exhaust energy increase, hence a higher steam production rate. However, the LSP must increase to maintain constant volume flow of steam through the ST. This introduces higher pinch point temperature differences and lower condenser pressure. Consequently, this results in a lower HRSG efficiency.

The main finding from the simulations is that none of plants will achieve higher plant gross efficiency at changing ambient temperature. The best plant efficiency occurs at design point. However, both plants have a long interval with approximately 100 % plant efficiency. The HECC plant has slightly lower relative plant gross efficiency than the OCC plant at low ambient temperatures. This is because the SC power output represents a larger share of the total power output in the HECC.

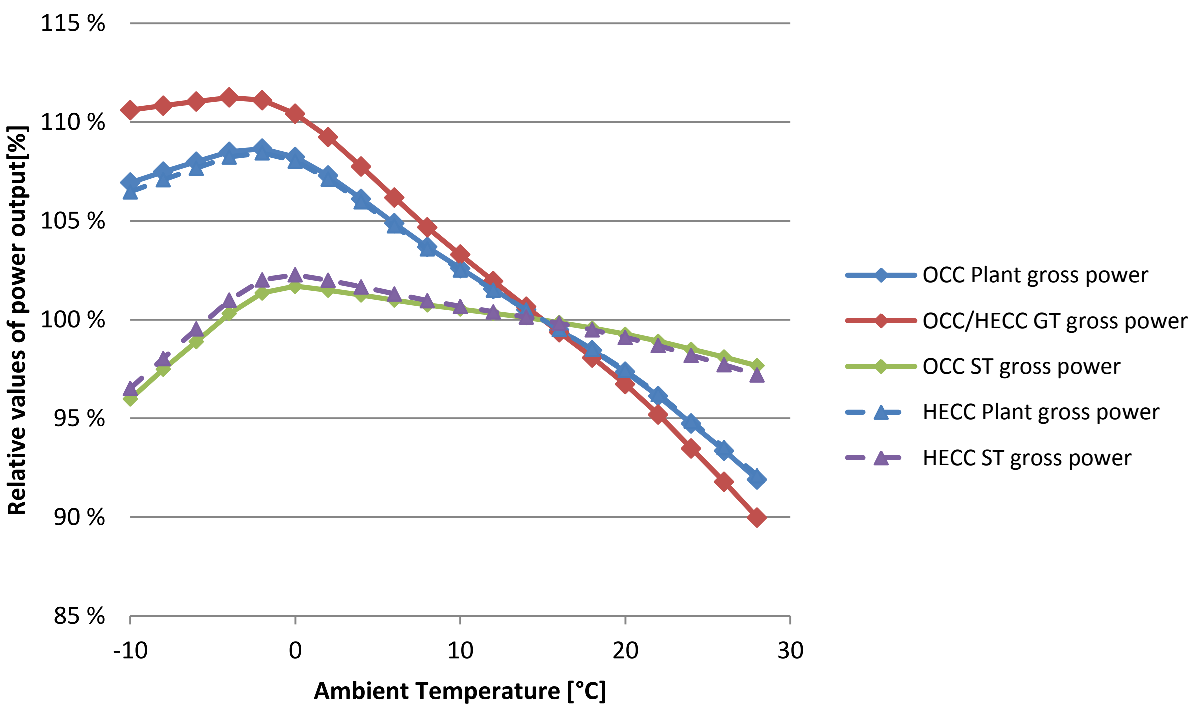

Figure 6.15, Effect on ambient temperature on the CC plant performance

On contrary to the plant efficiency, the plant power output changes with ambient temperature. This is illustrated inFigure 6.15. At lower ambient temperatures, the high exhaust mass flow compensate for the reduced SC efficiency, and consequently the ST archive higher power output. At high ambient temperature the opposite occur. The SC efficiency is high, and the ST power output low due to less exhaust gas.

In Table 6.5, performance values at 0C and 15C ambient temperature is listed for the two plants.

Table 6.5 Off-design performance at 0 °C ambient temperature compared to the design point

Simulation	Offshore CC plant		High efficiency CC plant	
Ambient temperature (°C)	15	0	15	0
Gas turbine				
Natural gas LHV input (MW)	84.2	91.3	84.2	91.3
Gross power output GT (MW)	32.2	35.5	32.2	35.5
Gross efficiency (%)	38.3	38.9	38.3	38.9
Exhaust flow (kg/s)	90	96	90	96
EGT(°C)	531	520	531	520
Water/steam cycle				
Pressure level IP/HP (bar)	24	24.6	6.9/55	7.15/56.12
Temperature IP/HP (°C)	470	461	258/510	258/500
Minimum pinch point IP/HP (K)	25	26.2	8/8	8.8/8.6
Steam mass flow IP/HP (kg/s)	11.0	11.4	11.0/1.9	2.2/11.3
Condenser pressure (bar)	0.080	0.084	0.048	0.051
Steam quality at outlet (%)	91.9	91.6	89.5	89.2
Stack temperature (°C)	170	173	104	107
HRSG efficiency (%)	69	68	81.1	80
Gross power output ST (MW)	10.8	11.0	13.3	13.6
ST efficiency (%)	30.1	30	32.0	31
Plant				
Gross power output (MW)	43.0	46.5	45.5	49.2
Gross power output (% of LHV input)	51.1	51.0	54.1	53.9
Net power output (MW)	42.3	45.8	44.7	48.3
Net power output (% of LHV input)	50.3	50.2	53.1	52.9

From design point to 0°C ambient temperature the net power output increase with 3.5MW for the OCC plant, and with 3.6MW for the HECC plant. This is mainly because of the GT which alone increased with 3.3MW. The net efficiencies decreased marginally with 0.1% and 0.2%, OCC and HECC respectively. From Figure 6.15, a higher ambient temperature, 30°C, will approximately result in the same change in power output and efficiency, but opposite.

Even if the GT is the component which is most affected by changing ambient temperatures, it is shown that SC plant is also sensitive for changes. At design, it is therefore essential to choose the ambient temperature which is most likely to occur. Choosing a wrong ambient temperature one may risk that the equipment becomes unnecessarily oversized. The equipment may also be to under dimensioned, such as the condenser. When the ambient temperature decreases, the steam production increases, hence the heat load on the condenser. The condenser has an upper limit of heat rejection which may limit for the SC.

6.6 Summary

The sensitivity analysis showed that the OCC plant may save 209 tons with a penalty of 2.8% in efficiency and 2.4MW in power output compared to the HECC. The OCC plant showed also good agreements compared with the existing offshore plants, with a plant efficiency of 50.3% and similar weight of the skids.

The CC performance at off-design is strongly dependent on the GT performance. At lower loads the plant efficiency remains high, and the ratio of ST to GT power output changes towards more ST power output. At 60% GT load, the relative gas turbine efficiency is 81% compared to the relative plant efficiencies of about 90%. The relative SC efficiency increase more in OCC plant than in the HECC plant. This is likely to be due to a larger potential for heat recovery in the OCC plant than in the HECC plant, and a larger drop in condenser pressure in the OCC plant. However, the difference is small, and the difference in plant efficiency remains constant at part load. The performance was satisfying at all loads, except for the low temperature on the LTE inlet. This was a design fault, which could be solved with a feedwater preheater.

The relative plant efficiency remains constant and approximately 100% in a long temperature interval around design point. This is due to diverging GT and ST relative efficiencies. From 15 to 0°C, the relative SC gross efficiency drops with 5 %, and the relative GT efficiency increase with 2%. However, the power output changes for both the GT and ST. From 28°C to about 0°C the power output increase almost linearly for the SC and GT. The increase in SC power output is due to the fact that the increased exhaust energy compensate for reduced efficiency. At 0°C, a maximum value is reached for the GT, and the GT power output is held constant.

In a real situation, part load and changes in ambient temperature occur at the same time. In an entire electric CC power production, regulation is simple, and efficient. At a fixed power demand, changes in ambient temperature must be controlled by means of the gas turbine. The simulation has shown that the plant may obtain high efficiency at part load and relative unchanged efficiency at changing ambient temperature.

In a CC with combination of a GT in mechanical drive and a ST in generator drive, the system may not utilize the high plant efficiency at different ambient temperatures. The mechanical power demand is independent of the electricity demand, which makes the system over constrained. If the GT power output correspond to an electricity production which is larger than the demand, special regulations is required. To common approaches are used, a bypass stack diverting parts of the exhaust heat to the atmosphere, or a bypass steam path were a fraction of the steam is routed directly to the condenser. In such configurations, the GT operate independent and may achieve the best possible efficiency, while the SC will experience a dramatic drop in efficiency. From this reason, an electric plant configuration is the most beneficial.

7 Conclusion

An offshore combined cycle model was developed in GTPRO. Offshore installations have special requirements which in many ways are different from onshore CC. Availability and reliability, high power to weight ratio, and size and area requirements are primaries. Based on these requirements and data from existing offshore plants, a sensitivity analysis was performed to find and good trade-off between weight and efficiency. The model showed good agreements compared with the existing offshore plants, with a power output of 50.3MW, plant efficiency of 50.3%, and similar weight of the skids. Also a high efficient plant was developed. This model gained 2.4MW more in power output, however with a penalty of 209 ton in extra weight.

A combined cycle offshore, may operate for prolonged time at off-design conditions, depending on power demand, ambient condition and other considerations offshore. The two models were used in off-design simulations in GTMASTER, concerning part load and changing ambient temperature. The results showed that both plants had similar behavior in performance at off-design, and that the GT strongly dictates the behavior of the steam cycle. At part load the relative SC efficiency increase resulting in general high plant efficiency. At 60% GT load, the relative gas turbine efficiency is 81% compared to the relative plant efficiencies of about 90%. The difference in efficiency between the high efficient plant and the offshore plant remains constant at part load.

The result from the simulations of ambient temperature is that none of plants will achieve higher plant gross efficiency at changing ambient temperature. The best plant efficiency occurs at design point. However, both plants have a long interval with approximately 100 % plant efficiency. From 15 to 0°C, the relative SC gross efficiency drops with 5 %, and the relative GT efficiency increase with 2%. However, the power output changes for both the GT and ST. From 28°C to about 0°C the power output increase almost linearly for the SC and GT.

8 Reference

1. Jon Ødegård Hansen, B.R., *Facts 2012, THE NORWEGIAN PETROLEUM SECTOR*, B.R. Jon Ødegård Hansen, Editor. 2012, Ministry of Petroleum and Energy, Norwegian Petroleum Directorate: http://www.npd.no/Global/Engelsk/3-Publications/Facts/Facts2012/Facts_2012_web.pdf.
2. Kloster, P. *Energy Optimization on Offshore Installations with Emphasis on Offshore Combined Cycle Plants*. 1999.
3. Ministry of Petroleum and Energy, N.G. *Emissions to air from the petroleum sector*. 2007.
4. Gülen, S.C. and J. John. *Combined Cycle Off-Design Performance Estimation: A Second-Law Perspective*. 2011: ASME.
5. Kehlhofer, R., *Combined-cycle gas & steam turbine power plants*. 2009, Tulsa, Okla.: PennWell. xix, 434 s.
6. Saravanamuttoo, H.I.H., et al., *Gas turbine theory*. 2009, Harlow: Prentice Hall. XVII, 590 s.
7. Bolland, O., *Thermal power generation*. 2010: Department of Energy and Process Engineering - NTNU.
8. Badeer, G.H., *GE's LM2500+G4 Aeroderivative Gas Turbine for Marine and Industrial Applications*, in *GE Energy, GER-4250*. 2005.
9. Boyce, M.P., *Handbook for cogeneration and combined cycle power plants*. 2010, New York, N.Y. (ASME, Three Park Avenue. New York, NY 10016): American Society of Mechanical Engineers. 1 electronic text (xvii, 759 s.).
10. Oljedirektoratet, O., Statoil, Hydro, Conoco Phillips, *Utredning av muligheter for mer effektiv energiforsyning på norsk sokkel*. 2004: Norwegian Petroleum Directorate.
11. Silje Lundberg, K.E.K., *Strøm fra land til olje- og gassplattformer*, Z.E.R. Organisation, Editor. 2011.
12. Technology, A.K.E.a., *Elektrifisering: Kraft fra land og samkjøring mellom innretninger*. 2008.
13. Sæther, S., *Combined cycles with compact heat recovery steam generators*. SINTEF rapport. Vol. STF15 A93111. 1993, Trondheim: SINTEF. 52 bl.
14. Elmhed, G., S. Ferm, and S.O. Svensson, *Industrial Type Gas Turbines for Offshore Applications.* Journal of Engineering for Power, 1980. **102**(2): p. 297-302.
15. Martin Wall, R.L., Simon Frost, *Offshore gas turbines (and major driven equipment) integrity and inspection guidance notes*. 2006, HSE, Health and Safety Executive: www.hse.gov.uk/research/rrhtm/rr430.htm.
16. Hellberg, A., *EXPERIENCE OF 29MW SGT-700 GAS TURBINE IN POWER GENERATION APPLICATIONS.* POWER-GEN International 2006 – Orlando, FL, 2006.
17. Farmer, R., *Combined Cycles Offshore, "North Sea platforms are converting mech drives to comb cycle operation".* Gas Turbine World, 1998(November-December 1998).
18. Evulet, A.T., *Innovation Now, Gas Turbine Combustion Technologies*, in *CORE Symposium*. 2011, GE Oil & Gas: Houston.
19. Thermoflow, I. *Products,Combined Cycle,GT PRO*. 2012 [cited 2012 24.06]; Available from: http://www.thermoflow.com/index.htm.
20. Company, G.E. *LM2500+G4 Aeroderivative Gas Turbines*. 2012 [cited 2012 25.06].