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Power Plant with CO2 Capture based on Absorption – Part-load Performance

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Submission date: July 2012
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Department of Energy
and Process Engineering

EPT-M-2012-37

MASTER THESIS

for

Student Bjørn Jordheim Halvorsen

Spring 2012

Power plant with CO_2 capture based on absorption – part-load performance

Kraftverk med CO_2-fangst basert på absorpsjonsprosess – dellastoppførsel

Background and objective

CO_2 capture, transport and storage in connection with power generation is globally potentially one of the most important technologies to limit emissions of CO_2 to the atmosphere. CO_2 capture requires the use of energy. The most promising technology for CO_2 capture in the short term is chemical absorption using amines or other solvents. Energy use for separating CO_2 from the exhaust gas increases the power plant fuel consumption approx. 15-40%.

The objective of this project is to find how a natural gas-fired power plant with an absorption plant for capture of CO_2 should be operated at part-load.

The following tasks are to be considered:

1. Provide an overview of literature that deals with the part-load and off-design performance of natural gas-fired Combined Cycles, both with and without the use of post-combustion CO_2 capture technologies.
2. Computational models (both design and off-design) shall be developed and documented for a natural gas-fired combined gas and steam turbine power plants, as well as for process for CO_2 capture using absorption with amines.
3. The impact of various part-load control methods shall be investigated; e.g. sliding vs. fixed steam pressure in the steam cycle, use of IGV for gas turbine air flow rate, fixed or variable liquid flow rate in the absorption process.

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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.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

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, January 31, 2012.

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Supervisor

Preface

This thesis, *Power plant with CO_2 capture based on absorption – part-load performance*, was written as the Master Thesis of Bjørn Jordheim Halvorsen. The Master Thesis was the final work of the 5-year Master of Science Degree. The thesis was written in Trondheim, for the Department of Energy and Process Engineering at the Norwegian University of Science and Technology.

The aim of the work was to give a detailed analysis of the part-load behavior of a power plant with an integrated CO_2 capture process. Based on the investigation, a recommendation on how the total plant should be operated at part-load has been given.

This study may be helpful as a part of the ongoing research within the field CO_2 capture and storage.

July 1th 2012, Trondheim

Bjørn Jordheim Halvorsen

Acknowledgements

I would like to thank my supervisor, Professor Olav Bolland, for his help and advices throughout this year. I would also like to thank my fellow students at office 430, for the endless discussions and coffee breaks. Especially thanks to Kjetil Vinjerui Ekre for his help and participation on some of the topics in this thesis.

Finally, I would like to thank my girlfriend, Nina, for her patience (and cooking) at the intensive finish of this work.

Abstract

This thesis gives a detailed evaluation of the part-load operation of a natural gas-fired combined cycle with an absorption plant for capture of CO_2. The study looks into each of the processes related to the plant. Both the combined cycle and the absorption process are investigated separately, in terms of their part-load behavior, and a recommendation on how the total plant should be operated at part-load is given. The first part of the current work was a theoretical study of combined cycles, absorption plants and the integration between those. Both design and off-design models have been looked into. Based on the theory, a reference plant was designed and considered as a starting point for the part-load investigation. By means of simulation models and the theory, several parameter changes have been analyzed for each of the processes.

The investigation of the part-load operation of the power plant indicated a significant net plant efficiency saving if inlet guide vanes were used to reduce the air flow into the gas turbine compressor, in combination with fuel reduction. The most recommended control strategy of the inlet guide vanes regulation was an almost constant target exhaust gas temperature relative to the design point. A higher target exhaust gas temperature obtained marginally better combined cycle efficiency, but problems could occur related to very high temperature gradients in the heat recovery steam generator.

Analysis of the absorption process showed a dramatic reduction in the liquid circulation rate that provided the lowest reboiler duty, as the gas turbine load was reduced. The reduction in liquid flow rate into the absorber was about 30% relative to the flow rate in the design point, for a gas turbine load of 60% with an almost constant exhaust gas temperature. Regarding problems due to insufficient wetting of the packing material in the absorber, a restriction on the liquid flow rate at part-load operation could be profitable. A relative increase in total reboiler duty of 5% was detected from the simulations if a constant liquid flow rate restriction was used, compared to 30% reduction of liquid flow rate, at 60% gas turbine load.

For the integrated power plant and absorption process, steam was preferable extracted from the crossover between the intermediate-pressure- and low-pressure turbine at 3,5 bar. This extraction pressure was independent of the part-load operation, and the low-pressure turbine should be throttled in order to meet the required steam extraction pressure at part-load. The design power plant with CO_2 capture obtained a total plant efficiency of 53%, disregarded mechanical losses- and compressor work in the capture plant. At 60% gas turbine load with almost constant exhaust gas temperature, the respective net plant efficiency was about 49% dependent of the liquid flow rate in the absorber. A efficiency loss of 0,3% percent points were detected if a constant liquid flow rate restriction was used, compared to 30% reduction of liquid flow rate at 60% gas turbine load.

Sammendrag

Denne oppgaven gir en detaljert evaluering av dellast drift av en naturlig gass-kombinert syklus med et absorpsjon anlegg for fangst av CO_2. Studien ser på hver av prosessene knyttet til anlegget. Både den kombinerte syklusen og absorpsjons prosessen blir undersøkt separat, med tanke på deres oppførsel ved dellast, og en anbefaling om hvordan det totale anlegget skal drives på dellast er gitt. Den første delen av arbeidet var en teoretisk studie av kombinerte sykluser, absorpsjonsanlegg og integrasjonen mellom dem. Både design og utenfor-design modeller har blitt sett på. Basert på teorien, ble et referanse anlegg utformet og brukt som utgangspunkt for dellast analysen. Ved hjelp av simuleringsmodeller og teorien, har flere parameterendringer blitt vurdert for hver av prosessene.

Undersøkelsen av dellast driften av kraftverket indikerte en betydelig bedre netto virkningsgrad for anlegget hvis luftstrømmen inn på kompressoren ble redusert, i kombinasjon med drivstoffreduksjon. Den mest anbefalte strategien for dellast kontroll i gassturbinen, var å holde en nesten konstant eksostemperatur i forhold til design punktet. En høyere eksostemperatur vil oppnå en marginalt bedre virkningsgrad på anlegget, men problemer kan oppstå knyttet til svært høye temperatur gradienter i røykgass varmegjenvinneren.

Analysen av absorpsjon prosessen viste en dramatisk reduksjon i den sirkulerte væskemengde som tilsvarte lavest energi behov i dampkjelen, ved redusert gassturbinen last. Reduksjonen i væskemengden inn på absorberen var på ca 30% i forhold til design punktet, for en gassturbin belastning på 60% med en nesten konstant eksos temperatur. Det kan bli problemer relatert til utilstrekkelig fukting av pakningsmaterialet i absorberen, dersom væskemengden blir for liten. En restriksjon på væskemengden ved kjøring av dellast kan derfor være lønnsomt. En relativ økning i totalt behov i dampkjelen på 5% ble påvist fra simuleringene, hvis en konstant væskemengde fra design punktet ble brukt, sammenlignet med 30% reduksjon av væskemengden, for en gassturbin på 60% dellast.

For den integrerte kraftverk- og absorpsjons prosessen var det foretrukket å hente ut damp fra mellomsteget mellom lavtrykksturbinen og mellomtrykksturbinen, ved 3,5 bar. Dette trykket var uavhengig av dellast driften, og lavtrykksturbins bør strupes for å opprettholde ønsket damptrykk.. Naturgass kraftverket med CO_2-fangst oppnådde en total virkningsgrad på 53%, sett bort fra mekaniske tap- og kompressor arbeid i fangstanlegget. Ved 60% gass turbin last med nesten konstant eksos temperatur, var det respektive anleggets virkningsgrad redusert til 49%, avhengig av væskemengden i absorpsjonsprosessen. For en konstant væskemengde i absorberen ble tapet i anleggets totale virkningsgrad 0,3% prosent-poeng, sammenlignet med en 30% reduksjon av væskemengden, for en gassturbin som kjørte på 60% last.

Abbreviations

AF	Air-Fuel ratio
CCS	Carbon Capture and Storage
EGT	Exhaust Gas Temperature
el	Electrical power
GT	Gas Turbine
HETP	Height Equivalent to a Theoretical Plate
HPB	High-Pressure Boiler
HPT	High-Pressure Turbine
HRSG	Heat Recovery Steam Generator
HSS	Heat Stable Salt
IGV	Inlet Guide Vanes
IPB	Intermediate-Pressure Boiler
IPT	Intermediate-Pressure Turbine
L/G	Liquid/gas ratio
LHV	Lower Heating Value
LPB	Low-Pressure Boiler
LPS	Low-Pressure Superheater
LPT	Low-Pressure Turbine
LSP	Live Steam Pressure
LST	Live Steam Temperature
MDEA	Methyldiethanolamine
MEA	Monoethanolamine
NGCC	Natural Gas-fired Combined Cycle
SC	Steam Cycle
SHT	Steam Superheating Temperature
SOT	Stator Outlet Temperature
SRD	Specific Reboiler Duty
ST	Steam Turbine
th	Thermal power
TET	Turbine Exit Temperature
TIT	Turbine Inlet Temperature
VSV	Variable Stator Vanes
wt.%	Weight percentage

Chemical Symbols

CO	Carbon monoxide
CO_2	Carbon dioxide
H_2O	Water
H_2	Hydrogen
N_2	Nitrogen
NO_x	Nitrogen oxide
O_2	Oxygen
SO_2	Sulfur dioxide

Nomenclature

Parameters

α	Ratio between heat potential and power reduction	$[MW_{heat}/MW_{work}]$
c_p	Specific heat capacity, constant pressure	[kJ/kg·K]
c_v	Specific heat capacity, constant volume	[kJ/kg·K]
C	Ratio of formed CO_2 and fuel	[kg/kmol]
$E_{rem,mech}^{CO_2}$	Mechanical work consumption in capture process	[MJ/kg CO_2]
$E_{rem,heat}^{CO_2}$	Heat consumption in stripper process	[MJ/kg CO_2]
$E_{comp}^{CO_2}$	Work requirement for compression of CO_2	[MJ/kg CO_2]
ΔE_{aux}	Work difference with no capture and reference NGCC	[MJ/kg CO_2]
f	CO_2 capture rate	
G'_M	Solute free gas flow rate	[kgmole/s]
H	Absorber height	[m]
$h_{reb\ exit}$	Enthalpy at reboiler exit	[kJ/kg]
h_{cross}	Enthalpy at crossover, IPT/LPT	[kJ/kg]
$h_{ST\ inlet}$	Enthalpy at steam turbine inlet	[kJ/kg]
$h_{ST\ outlet}$	Enthalpy at steam turbine outlet	[kJ/kg]
h_{exit}	Enthalpy at heat exchanger outlet, water side	[kJ/kg]
h_{inlet}	Enthalpy at heat exchanger inlet, water side	[kJ/kg]
h_{ext}	Enthalpy at extraction point, prior to the reboiler	[kJ/kg]
$K_G a$	Overall mass transfer coefficient	[kmol/m^{3}hr·kPa]
κ	Specific heat ratio	
L'_M	Solute free liquid flow rate	[kgmole/s]
$LMTD$	Log mean temperature difference	[K]
$\dot{m}$	Mass flow	[kg/s]
$\dot{m}_{steam}$	Mass flow rate of steam	[kg/s]
$\dot{m}_{water}$	Mass flow rate of water	[kg/s]
$\dot{m}_{CO_2}$	Mass flow CO_2 captured	[kg/s]
$\dot{m}_{cross}$	Mass flow rate of steam at crossover, IPT/LPT	[kg/s]
$\dot{m}_{ext}$	Mass flow rate of steam at extraction point	[kg/s]
MW_{tot}	Molecular weight, total liquid flow	[kg/kmol]
MW_{MEA}	Molecular weight, MEA	[kg/kmol]
η_{NGCC}	Net plant efficiency of natural gas-fired combined cycle	[%]
η_{GT}	Gas turbine gross efficiency	[%]
η_{SC}	Steam cycle gross efficiency	[%]
η_{HRSG}	HRSG efficiency	[%]
η_{ST}	Steam turbine efficiency	[%]
η_{is}	Isentropic efficiency	[%]
η_p	Polytrophic efficiency	[%]
P_{ext}	Pressure at the extraction point, prior to the reboiler	[bar]

P_{cross}	Pressure at the IPT/LPT crossover	[bar]
P_{LPB}	Pressure in low-pressure boiler	[bar]
p	Pressure	[bar]
ρ	Density	[kg/m^3]
$\dot{Q}$	Heat energy	[MW]
$\dot{Q}_{Fuel}$	Lower heating value	[MW]
$\dot{Q}_{GT,Exh}$	Flue gas heat into HRSG	[MW]
$\dot{Q}_{reb}$	Heat requirement in reboiler	[MW]
$\dot{Q}_{sensible}$	Sensible heat	[MW]
$\dot{Q}_{latent}$	Latent heat	[MW]
$\dot{Q}_{des}$	Heat of absorption	[MW]
R	Gas constant	[kJ/kg·K]
T	Temperature	[K]
ΔT	Temperature difference	[K]
u	Velocity	[m/s]
U	Overall heat transfer coefficient	[W/m^{2}K]
v	Specific volume	[m^3/kg]
$\bar{V}$	Swallowing capacity	
$\dot{W}_{ext}$	Steam turbine work in NGCC with steam extraction	[MW]
$\dot{W}_{GT}$	Gas turbine gross power output	[MW]
$\dot{W}_{ST}$	Steam turbine gross power output	[MW]
$\dot{W}_{Aux}$	Auxiliary consumptions in NGCC	[MW]
$\dot{W}_{lost}$	Lost steam turbine work due to steam extraction	[MW]
$\dot{W}_{no\ ext}$	Steam turbine work in NGCC without steam extraction	[MW]
$\dot{W}_{tur,exp}$	Gas turbine expansion work	[MW]
$\dot{W}_{ST,exp}$	Steam turbine expansion work	[MW]
W_{comp}	Energy of CO_2 compression	[MJ/kg CO_2]
X_1	Mole ratio CO_2 in liquid stream, absorber exit	
X_2	Mole ratio CO_2 in liquid stream, absorber inlet	
x_{CO_2}	Lean CO_2 concentration	
Y_1	Mole ratio CO_2 in gas stream, absorber inlet	
Y_2	Mole ratio CO_2 in gas stream, absorber exit	
y	Mole fraction CO_2 in gas stream	
y_e	Mole fraction CO_2 in gas stream in equilibrium	
γ_{CO_2}	Lean CO_2 loading	

Subscript

0	Design point
1	Compressor inlet
2	Compressor outlet

3	Turbine inlet (gas turbine)
4	Turbine outlet (gas turbine)
α	Section of turbine inlet
w	Section of turbine outlet
s	Steam
h	Hot side of heat exchanger

1. Introduction

The climate is changing and the temperature in the atmosphere is increasing. The Intergovernmental Panel for Climate Change have indicated a possible global surface temperature increase from 1,1 to 2,9 °C for their lowest emissions scenario, and from 2.4 to 6.4°C for their highest, during the 21st century [1]. These temperature changes are somewhat caused by human emission and CO_2 have the greatest impact on the greenhouse effect and represents approximately 5% of the global warming[1]. The amount of CO_2 in the atmosphere has lately been measured at the highest level ever. From Figure 1.1, it could be seen how CO_2 gradually has increased over the last years. In addition, the energy demand increases proportional to the increase in population. Fossil fuel represents over 60% of the total electricity production, however, the use of fossil fuel will still increase due to the high energy demand. One option in order to reduce the emissions of CO_2 into the atmosphere is to capture and store the CO_2.

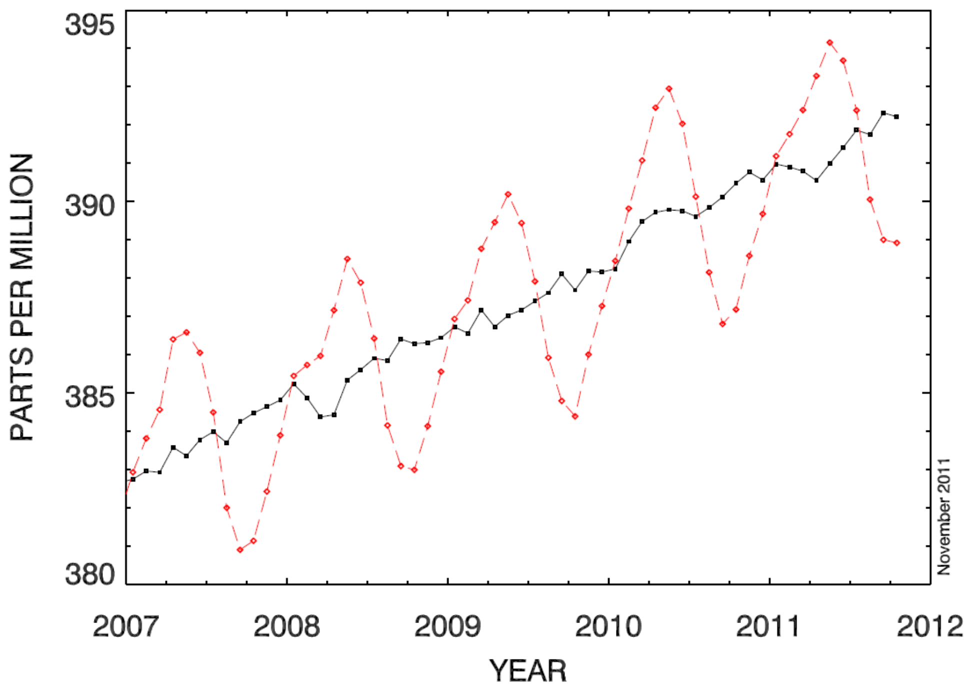

Figure 1.1 - Trends in Atmospheric Carbon dioxide[2].

Prior to the finance crisis in 2008 a lot of money was invested in research on Carbon Capture and Storage (CCS), and the building of CO_2 capture pilot plants from power generation. After the crisis, most of the pilot plants projects have been postponed or canceled due to the lack of further funding. Capturing and storage of CO_2 requires a lot of energy and is very expensive. The energy demand of the capture process is too high at the current state of development. The capture technology needs improvements to get the attention of investors.

1.1 CO_2 Capture

There are three main principles for capturing carbon from a power plant. As seen in Figure 1.2, these are pre-, post- and oxy-fuel combustion capture. Within each of these capture stages membranes can be used in order to separate CO_2. This report will focus on post-combustion based on chemical absorption with monoethanolamine (MEA).

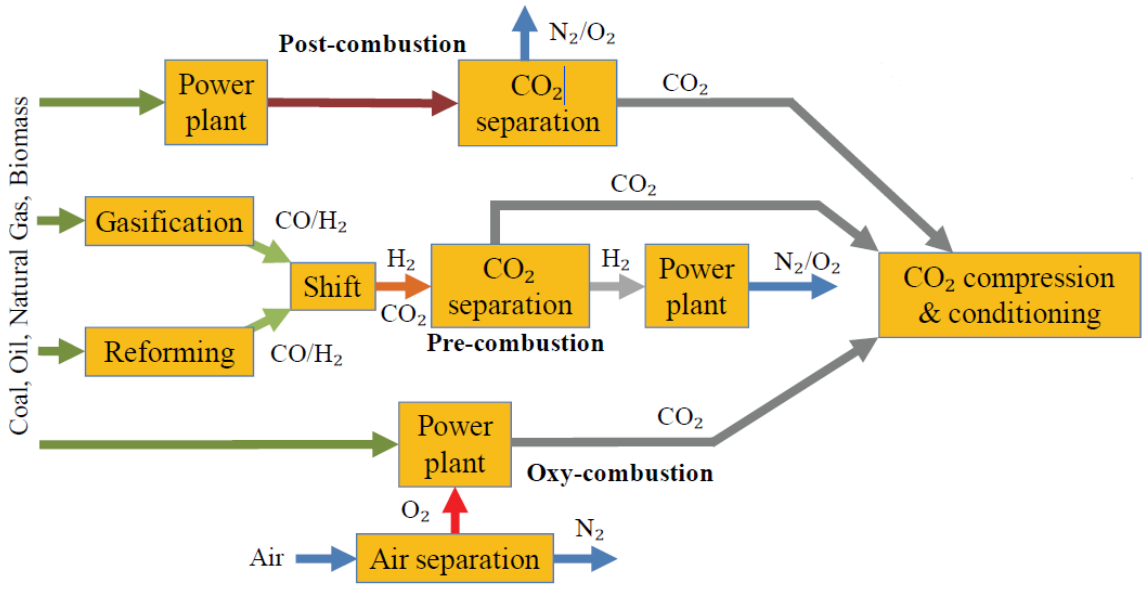

Figure 1.2 - Principle methods for CO_2 capture[3].

1.1.1 Pre-combustion

In pre-combustion, the process is divided into two main reactions. In the first reaction the fuel is mixed with either pure oxygen, equation 1.1, often called partially oxidation or steam, equation 1.2, called steam reforming. These reactions transform the fuel into carbon monoxide and hydrogen.

$$C_xH_y + \frac{x}{2}H_2O \leftrightarrow xCO + \left(\frac{y}{2}\right)H_2 \qquad \Delta H - ve \tag{1.1}$$

$$C_xH_y + xH_2O \leftrightarrow xCO + \left(\frac{x+y}{2}\right)H_2 \qquad \Delta H + ve \tag{1.2}$$

After the steam reforming, the temperature is lowered and a shift reaction transforms the CO into CO_2 by additional steam. The reaction is called a Water Gas Shift Reaction, and is given in equation 1.3.

$$CO + H_2O \leftrightarrow CO_2 + H_2 \qquad \Delta H = -41\ kJ/mol \tag{1.3}$$

The CO_2 is then removed from the process, and the concentration of CO_2 before the separation is in the range of 15-60% (dry basis)[1].

1.1.2 Oxy-fuel Combustion

In oxy-fuel combustion, the fuel is burned with pure oxygen, or a blend of oxygen and carbon dioxide, instead of air. By eliminating the nitrogen from the combustion the products is mostly a mixture of CO_2 and water vapor. The reaction is given by equation 1.4

$$C_xH_y + (x + \frac{y}{4})O_2 \leftrightarrow xCO_2 + \frac{y}{2}H_2O \qquad 1.4$$

After the combustion the flue gas is cooled and water is condensed. This results in a product of almost pure CO_2. However, the flue gas often requires further cleaning. When natural gas is burned with pure oxygen, the temperature in the combustion can reach up to 3500°C. The main problem with oxy-fuel combustion, is that the process of separating oxygen form the air is very energy demanding[1].

1.1.3 Post-combustion

There are many methods for capturing CO_2 after the combustion. Since CO_2 is a rather weak acid, alkaline solvents can be used in order to absorb it. Also, membrane can be used to separate the CO_2, and micro porous structures may be used to adsorb it. The most common process is absorption and the absorption process will be explained in depth in this thesis. With adsorption, the process takes place on the surface of the adsorbent. The most used adsorbent is a solid, but liquids are also used. The micro porous structure that is used, is often activated carbon or molecular sieves[1]. In this process, the CO_2 is accumulated in the pores of the adsorbent, as can be seen in Figure 1.3[4].

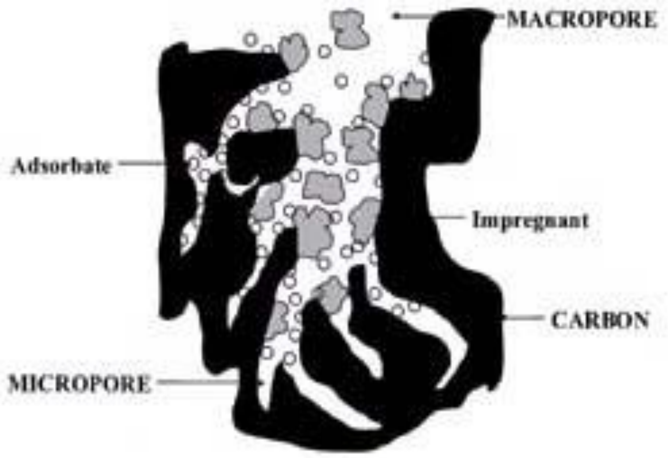

Figure 1.3 - Adsorption with activated carbon.

1.1.4 Membrane

Membrane is a technology that selectively transfers certain chemical components over other components through a wall. A schematic of a membrane is presented in Figure 1.4. The feed gas, containing e.g. CO_2 or O_2, is transported past the membrane wall, and some of the CO_2 or O_2 is transported through. The amount of CO_2 depends on the membranes' permeability and selectivity. On the other side of the wall, sweep gas is used to transport the CO_2 away from the wall. This is done in order to keep the partial pressure low on the permeate side. A low partial pressure on the permeate side results in more diffused CO_2. This technology can be used in pre-, post- and oxy-fuel combustion. It can be used to separate O_2 from air or separate CO_2 from flue gas. The main problem with membrane, at the present time, is its low tolerance to high and varied temperatures.

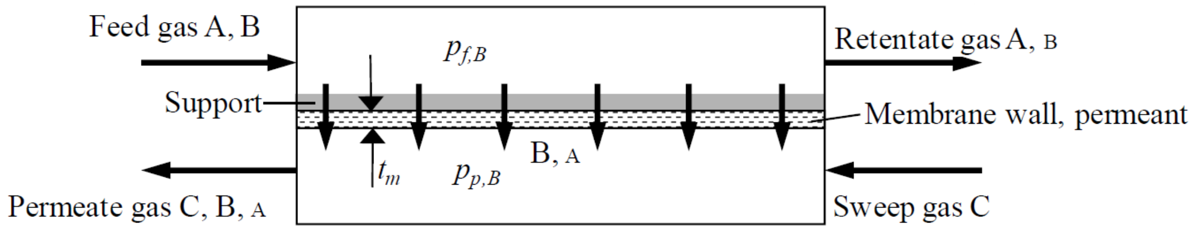

Figure 1.4 - A typical membrane[3].

1.2 Risk Assessment

In this thesis, there has been no laboratorial work in any form. It has neither been any excursions. Therefore, it has not been performed any risk assessment for the work regarding this thesis.

1.3 Scope of the Report and Outline

The aim of the current work was to investigate the influence of a reduced power plant load on the performance of a Natural Gas-fired Combined Cycle (NGCC) and the belonging CO_2 capture process. A recommendation should be provided on how the total plant should be operated at part-load. Different control methods in the combined cycle, such as use of Inlet Guide Vanes (IGV) in the gas turbine compressor, and sliding pressure mode related to the steam extraction from the steam cycle, were to be considered. Also the CO_2 capture plant with the different regulations of the liquid flow rate was an object for analysis. The investigation of the total plant performance was to be done with regards to the total plant efficiency, and the main attention in the CO_2 capture plant should be on the energy demand in the reboiler.

In order to explore the part-load behavior of the total plant, simulation tools had to be used. GT PRO should be used for the design of the power plant, while GT MASTER, which is connected to the design in GT PRO, was to be used for the off-design[1] calculations of the power plant performance. The simulation tool Hysys should be used in order to investigate the CO_2 capture plants behavior, both for the design model and the off-design problems.

The part-load behavior of the total power plant with CO_2 capture depends on the design of the plant. An optimal design of the plant was to be investigated and created prior to the off-design study. The thesis was divided into multiple parts.

- A theoretical part that describes the different parameters and components in the plant
- A literature study that gives a brief summary of past work related to this thesis
- The creation of an optimal design of a NGCC plant with CO_2 capture
- The part-load behavior of the power plant and its affection of IGV regulations in the gas turbine compressor.

[1] The off-design and part-load concept are often interchanged. Off-design could be used for processes that differ from the design point, while part-load is often related to a change in power output in a power plant.

- The off-design performances of the CO_2 capture plant for different flue gases from the power plant. The influence of variations in the liquid flow rate on the energy demand in the reboiler was to be studied.
- A part-load behavior of the total power plant with CO_2 capture.

2. Power Plant

In order to evaluate the possibilities for integration of a power plant and the accompanying CO_2 capture process, and to study the part-load performance, it is important to analyze the power generation process. In particular the steam cycle in a combined cycle is essential. In this chapter, a brief overview of the most important components in a NGCC is given.

2.1 Combined Cycle

A combined cycle could be defined as a combination of two thermal cycles in one plant[5]. This combination makes it possible to extract more energy from the fuel, and contributes to a higher total efficiency in the power plant. In a NGCC these two cycles consists of a gas- and a steam cycle. The gas cycle is operating on a higher temperature level than the steam cycle, and is called the topping cycle[5]. Because of the high gas Turbine Exit Temperature (TET) it is possible to utilize the energy remained in the exhaust stream. This heat is used to generate steam in a second process, and the steam will be expanded in a steam turbine. The net electric power efficiency of a NGCC could be close to 60%[3].

2.1.1 Gas Turbine

A gas turbine consists of a compressor, a burner and a turbine. The gas turbine cycle is called a Brayton cycle, and is shown in Figure 2.1. In the compressor, ambient air is compressed to a pressure of 14 to 30 bar, depending on the gas turbine[5]. The air is used to burn the fuel, producing a hot gas in the combustion chamber. In this report natural gas is the only fuel type considered. The fuel could be pre-heated before the combustion. An important property of the fuel is the fuels Lower Heating Value (LHV). It defines the mass flow of the fuel, which must be supplied to the gas turbine. After the burning, the temperature of the hot gas may be up to 1500°C, and the gas is expanded in a turbine, producing power[6]. The power is produced by conversion of pressure energy to kinetic energy with stators, and then the kinetic energy is converted to power because of rotation of the shaft at the rotors that is connected to a generator.

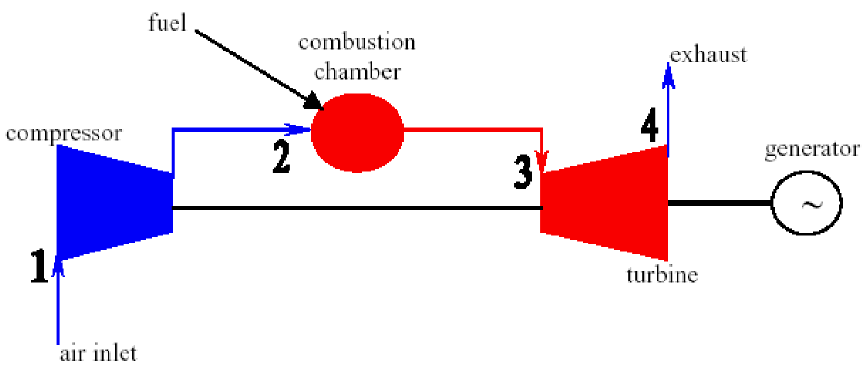

Figure 2.1 - Brayton cycle[7].

When increasing the Turbine Inlet Temperature (TIT), a higher useful enthalpy drop is produced, thus more power is generated. The TIT is limited by the materials of the blades in the gas turbine. Turbine cooling is necessary for high temperatures. The most common cooling system is to inject excess air from the air compressor into small bleeds in the turbine blades.

The efficiency of the gas turbine is defined as the conversion of the fuels LHV to electric power. For large gas turbines in power plants the efficiency is in the range 35-40%[6]. The flue gas out of the turbine is slightly above atmospheric pressure, and has a temperature from 450-650°C. The exhaust stream contains almost all the not converted energy from the fuels lower heating value. As seen from the efficiency, this energy is significant. By connecting the gas turbine to a steam cycle, some of this energy could be recovered. The connection takes place in a Heat Recovery Steam Generator (HRSG).

2.1.2 Heat Recovery Steam Generator

The second cycle is the steam cycle, where steam is produced in a HRSG before it is expanded through a steam turbine for power generation. The HRSG consists of three heat exchanger sections that transport the heat from the hot exhaust gas to a water cycle.

First, the water in the steam cycle is pre-heated in an economizer from a subcooled condition to an almost saturated condition. The saturation temperature depends on the pressure of the water. It is possible for the HRSG to operate at more than one pressure level. This will be explained in chapter 2.1.2.1. The reason why the water is not at saturated conditions at the economizer exit is to avoid evaporation in the economizer at off-design conditions[5]. After the economizer the water is fed into a boiler. In the boiler, the water is heat exchanged with the exhaust gas and evaporated at constant pressure and temperature. At the end, the steam is superheated in a superheater. Superheating of the steam increases the enthalpy in the stream. The expansion work in the steam turbine, except mechanical and generation losses, is given by equation 2.1

$$\dot{W}_{ST,exp} = \dot{m}_{steam}(h_{ST\ inlet} - h_{ST\ outlet}) \qquad 2.1$$

As seen from equation 2.1, an increase in the inlet enthalpy contributes to an increase of the steam turbine power output. The inlet enthalpy depends very much on the inlet temperature, for high pressures. Another reason for superheating of the steam is to avoid liquid in the turbine. Liquid in the turbine could reduce the power output by slowing down the turbine blades. High moisture content at the end of the steam turbine also increases the risk of erosion. It is recommended a moisture content limit of 16% at the steam turbine exit[5].

2.1.2.1 Pressure levels

One of the most important parameters in the HRSG design is the minimum temperature difference between the exhaust gas and the water within a given pressure level. This temperature difference is called the pinch point. The location of the pinch point depends on the HRSG inlet temperature of the flue gas and the pressure in the water cycle. For low flue gas temperature, as is the case for natural gas as fuel, the pinch point is located between the economizer and the evaporator.

The heat transfer on the water side in a heat exchanger is given by equation 2.2

$$\dot{Q} = \dot{m}_{water}(h_{inlet} - h_{exit}) \qquad 2.2$$

The equation could be rewritten by assuming a constant specific heat capacity for water:

$$\dot{Q} = \dot{m}_{water} c_p (T_{inlet} - T_{exit}) \quad 2.3$$

The slope of the water lines in the TQ diagram is given by equation 2.4.

$$\frac{\Delta T}{\dot{Q}} = \frac{1}{\dot{m}_{water} c_p} \quad 2.4$$

In order to achieve a high power generation in the steam cycle, in relation to equation 2.1, the amount of steam produced is of importance. A moderate slope of the water in the TQ diagram is preferred, giving a high steam mass flow. A typical TQ diagram is presented in Figure 2.2.

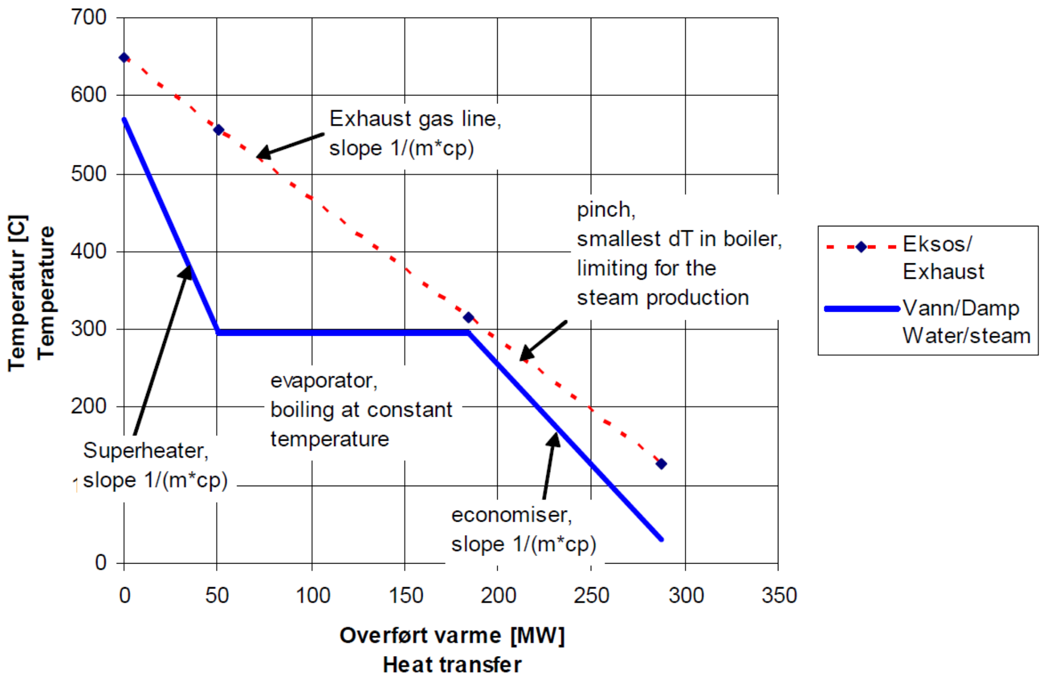

Figure 2.2 – TQ-diagram for heat recovery process[6].

The high heat of evaporation of water is a disadvantage for the steam generation. The effect of a high heat of evaporation on the TQ diagram is a large temperature difference between the water and the flue gas at the steam inlet of the superheater. As seen in Figure 2.2, the low flue gas temperature contributes to a steep slope of the superheater line, in order to reach the live steam temperature. That is a drawback for the steam generation. An option in order to increase the steam generation is to introduce multiple pressure levels.

By introducing multiple pressure levels, it is possible to utilize the lower temperatures in the flue gas because of reduced evaporation temperature at the lowest pressure levels. The mean temperature difference will be reduced through the HRSG, and more steam is produced. In a combined cycle with natural gas as fuel, the inlet temperature of the HRSG is about 450-650°C, and for this low temperature range multiple pressure levels will increase the steam cycle efficiency[3].

The pinch point refers to the smallest temperature difference through a HRSG. With higher temperature difference less heat exchanger area is needed to evaporate the water, compared to a lower pinch point. The surface of the evaporator increases exponentially when the pinch point is reduced. But with lower pinch point, the exhaust heat is utilized better and more steam is generated. How to design the pinch point is an optimization problem, but if the goal is to maximize the efficiency of the plant the lowest pinch point is optimal. There has to be a temperature difference for heat exchanging between the mediums to occur. Pinch points are often in the range of 8-15K[5].

2.1.2.2 Pressure drop

There will be a pressure drop through the HRSG. The size of the pressure drop of the flue gas is a design problem of the heat exchangers. The biggest disadvantage related to the flue gas pressure drop, is the backpressure of the gas turbine. To overcome the pressure drop, the exit pressure out of the gas turbine has to increase. This increase leads to less work generated in the gas turbine. The lost work is proportional to the pressure drop and the temperature close to the turbine exit among other factors [3]. Therefore, the power generation in the gas turbine is very sensitive to a change in the pressure drop.

2.1.3 Steam Turbine

After the superheater the steam enters the steam turbine. The steam expands through the turbine and there will be generated power. Depending upon the number of pressure levels in the HRSG, there will be multiple inlets at the turbine. In the case of three pressure levels with reheat, there will be three inlets, two outlets and one crossover for the steam turbines. The moisture content at the steam turbines exit is a limiting factor for the highest pressure levels. As the steam is expanded the superheating of the steam is decreasing. By introducing reheat of the steam expanded in the High-Pressure Turbine (HPT), the pressures could be higher in relation to the moisture content. The high-pressure steam, normally in the range 100-130 bar and a temperature of 450-560°C, enters the HPT[3]. At the outlet of the HPT, the steam exits and mixes with the intermediate pressure steam in the HRSG. The mixed steam is superheated in a re-heater and enters the Intermediate-Pressure Turbine (IPT) with conditions about 30 bar and temperatures of 450-560°C. There is a crossover between the exit of the IPT and the inlet of the Low-Pressure turbine (LPT), where the steam flows. In addition, steam at low pressure conditions is transferred to the LPT from the HRSG. The steam conditions at the LPT inlet are about 3-5 bar and 200-300°C. Finally the expanded steam exits the steam turbine at a low pressure, around 0,04-0,05 bar. The dry isentropic efficiency is different for each of the three turbine sections in a three pressure level cycle, and the dry efficiency needs to be corrected for moisture when the vapor fraction is below 100% at the turbine outlet.

2.1.4 Cooling System

When the steam exits the steam turbine the pressure is about 0,04-0,05 bar. Low pressure out of the turbine gives the highest power output, related to equation (2.1). At the same time, low pressure causes larger dimension of the condenser and steam turbine[6]. There is a limit of the exit pressure, and it varies with type of cooling system for the exit steam. The low pressure steam out of the steam turbine is condensed before it is pumped up to a higher pressure. To reject the heat of condensation from the steam, a cooling system is needed. In the cooling system, a heat exchanger is used for condensing the steam by heat transferring from the steam to a coolant. The coolant is heated, and before it is returned to the heat exchanger it has to be cooled.

There are basically three different types of cooling systems[5].

- Direct/once-through water cooling
- Wet cooling tower with evaporative cooling
- Direct/indirect air cooling in an air cooled condenser

A wet cooling tower is the cooling system used in the current work.

2.1.5 Feedwater Tank and Deaerator

The condensed water is pumped to a higher pressure and enters the feedwater tank. In the feedwater tank, the condensate is mixed with makeup-water. Makeup-water is added because of small leakages through the steam cycle. In the case of process steam extraction, returned process condensate is also fed into the feedwater tank. The water mix is often heat exchanged with the flue gas in the last part of the HRSG, and enters the deaerator. In the feedwater, dissolved oxygen and CO_2 are presented and causes corrosion and acid attack in the boilers. By introducing a deaerator, the level of O_2 and CO_2 in the feedwater can be controlled. The deaerator utilizes the principle of how gas solubility in a solution decreases when the temperature increases.

2.1.6 Steam Extraction

Some of the energy in the steam expanded in the steam turbine is converted to power, but not all of the energy is converted. The work produced in the steam turbine depends on the enthalpy difference between the inlet and exit stream of the turbine, see equation 2.1. At the exit, the steam is at saturated conditions at the given condensation pressure. The steam/water is slightly condensed, with a vapor fraction of about 0,84-1[5]. Because of the high heat of evaporation for water, the enthalpy at the turbine exit is high compared to the condensed liquid at the same pressure. This is illustrated as $\Delta h_{1,2}$ in Figure 2.3. A typical condensation pressure of 0,05 bar is indicated as point 3.

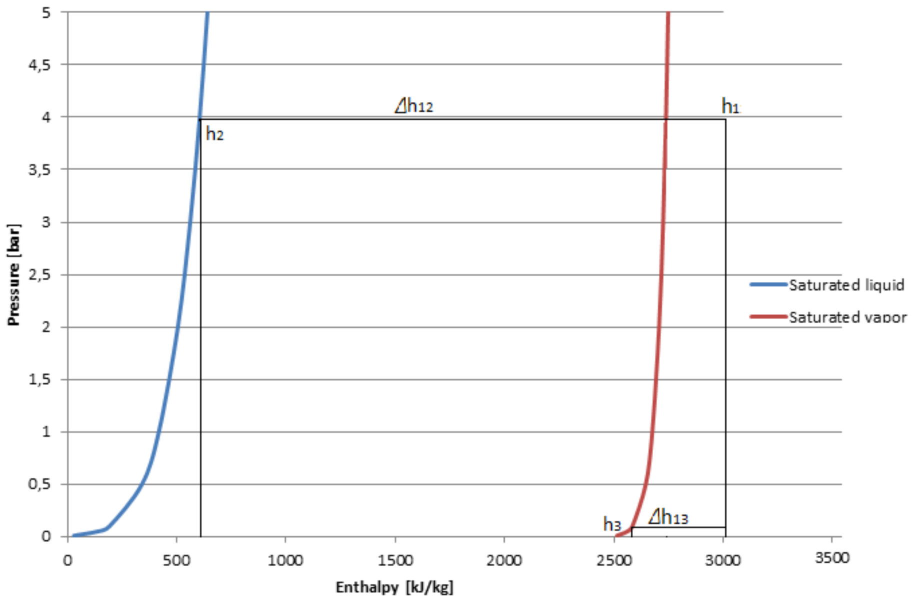

Figure 2.3 – Enthalpy curves for saturated water liquid (blue line) and saturated water vapor (red line) against pressure[9].

The relative high enthalpy at the turbine exit leads to a low enthalpy difference through the turbine, and therefore a low power production.

By extracting steam from the steam turbine, the steam could be used for heating purposes instead. The high heat of evaporation could be utilized in a heat exchanger process. For a given stream, the heat potential is given by equation 2.5 if the water is returned as saturated liquid:

$$\dot{Q} = \dot{m}(h_1 - h_2) \qquad 2.5$$

The enthalpies are indicated in Figure 2.3 for a constant inlet/extraction pressure of 3,9 bar, together with the enthalpy differences $\Delta h_{1,2}$ and $\Delta h_{1,3}$. If the stream is not extracted, it could have been further expanded in the steam turbine.

From Figure 2.3, it is clearly that the heat potential is higher than the turbine work. This tendency is also the case for higher inlet/extraction pressure with a fixed condenser pressure. However, the ratio between the heat potential and the power generation is decreasing as the pressure increases. In Figure 2.4, it is illustrated how the saturated liquid enthalpy increase with pressure. If the condenser pressure is fixed, the enthalpy difference in equation 2.1 is growing more rapidly with higher pressures than the enthalpy difference in equation 2.5. This is due to the increased saturated liquid enthalpy.

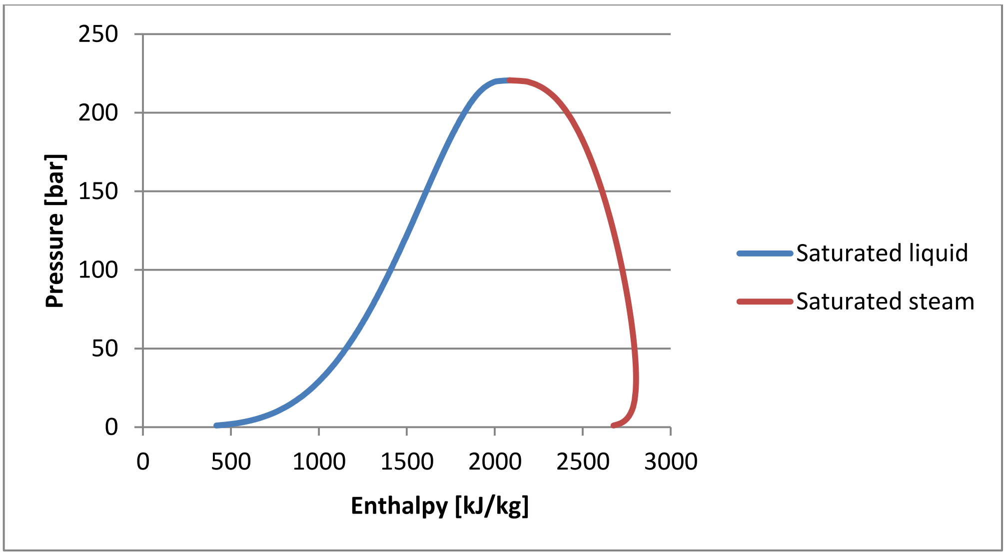

Figure 2.4 - Enthalpy curves for saturated water liquid (blue line) and saturated water vapor (red line) against pressure[9].

The lost turbine work because of steam extraction is defined as the change in generator terminal power output, with and without steam extraction[3]. This lost expansion work is the difference in power output between two steam cycles with the exact same design parameters, apart from the extracted steam. The change in auxiliaries and losses is neglected. The lost power output in the steam cycle is

$$\dot{W}_{lost} = \dot{W}_{no\ ext} - \dot{W}_{ext} \qquad 2.6$$

The ratio between the extracted process heat and the lost power production is defined as the steam quality, α.

$$\alpha = \frac{\dot{Q}}{\dot{W}_{lost}} \tag{2.7}$$

The α-value is preferred as high as possible, indicating high heat potential in the extracted steam compared to the lost power generation in the steam turbine.

2.1.7 Efficiency

The efficiency of a power plant can be defined in several ways. In the current work, the electrical efficiency of the combined cycle plant is defined as:

$$\eta_{NGCC} = \frac{\dot{W}_{GT} + \dot{W}_{ST} - \dot{W}_{Aux}}{\dot{Q}_{Fuel}} \tag{2.8}$$

where $\dot{W}_{GT}$ is the gas turbine gross output, $\dot{W}_{ST}$ is the steam turbine gross output, $\dot{W}_{Aux}$ is the auxiliary consumptions and $\dot{Q}_{Fuel}$ is representing LHV of the fuel. The gas turbine- and steam turbine gross output is the expansion work, or enthalpy change, corrected for mechanical- and generator losses. The gas turbine- and steam cycle efficiency are defined as

$$\eta_{GT} = \frac{\dot{W}_{GT}}{\dot{Q}_{Fuel}} \tag{2.9}$$

and

$$\eta_{SC} = \frac{\dot{W}_{ST}}{\dot{Q}_{GT,Exh}} \tag{2.10}$$

where $\dot{Q}_{GT,Exh}$ is the remaining heat in the flue gas into the HRSG. For convenience, this heat is in the current work defined as

$$\dot{Q}_{GT,Exh} = \dot{Q}_{Fuel} - \dot{W}_{GT} \tag{2.11}$$

By combining equation 2.9 and 2.11

$$\dot{Q}_{GT,Exh} = \dot{Q}_{Fuel} - \eta_{GT}\dot{Q}_{Fuel} = \dot{Q}_{Fuel}(1 - \eta_{GT}) \tag{2.12}$$

is derived. An expression of the total plant efficiency based on the gas turbine- and steam cycle efficiencies could be formed

$$\eta_{NGCC} = \eta_{GT} + \eta_{SC}(1 - \eta_{GT}) - \frac{\dot{W}_{Aux}}{\dot{Q}_{Fuel}} \tag{2.13}$$

A more accurate definition would have included the mechanical and generator losses related to the gas turbine gross power output, thus the remaining heat in the flue gas would decreased marginally. The efficiency of a gas turbine is driven by the pressure ratio in the turbine[5].

2.2 Part-load

Generally, the term off-design describes a plant which operates at conditions away from its design conditions. An off-design analysis investigates the plant behavior for such conditions. It could be changes in ambient temperature, ambient pressure and ambient humidity. More importantly for this thesis, there are variations in the power demanded from a power plant with time. In order to meet the required demand, the net power output of the plant must be regulated. A power plant running at a different power output than the design output is operating at part-load.

The flue gas flow from the power plant is treated in the absorption process. Both the terms, off-design and part-load, are in the present text used to describe the absorption process with a variation in the flue gas conditions, on the basis of a power plant operating at part-load.

The part-load operation of a combined cycle is regulated by means of the gas turbine, while the steam cycle operation is optimized with respect to the available heat in the HRSG. The gas turbine has a short reaction time, and it could follow quick load changes. This property is an advantage in the electricity market, where the power demand is varied with time. As the gas turbine changes its load, the steam turbine will be adjusted automatically as a slave. However, there is a delay because of the response time in the HRSG.

Part-load is in the current work defined as the gas turbine gross power output per gas turbine gross power output at design conditions.

2.2.1 Gas Turbine

For a simple gas turbine, the most efficient part-load control is to maintain a constant air flow related to the design point, and only reduce the fuel flow. In that way, the expansion work in the turbine is relative high because of a high total mass flow and pressure ratio. The drawback is the low TIT. However, the positive effect of a high mass flow and pressure ratio is more dominating than the negative contribution from the low temperature.

On the other hand, if a combined cycle is subject to an optimized part-load operation the exhaust gas temperature is of importance. While the simple cycle makes a profit on a high air mass flow, it causes a low exhaust temperature. The low exhaust temperature results in a poor utilization of the second cycle. Therefore, the most efficient part-load operation of the combined cycle is to control the TIT by simultaneously reduce the air- and fuel flow.

There are two control methods of the gas turbine. Either the power output is varied by TIT control, or the power generated in the gas turbine is controlled by regulation of the inlet guide vanes in the compressor and fuel supplied to the combustion chamber. By use of IGVs, the air mass flow may be reduced to about 60-70% of the air flow in the design point[3]. With this limit of the air reduction, the exhaust gas temperature may remain high for loads down to about 40%[5]. This limit will vary between the different gas turbines and their producers. For loads below that level, the air flow cannot be further reduced and the exhaust gas temperature decreases. This contributes to a dramatic drop in the power plant efficiency.

2.2.1.1 Compressor

Industrial gas turbines are usually single-shaft axial engines that are coupled to a generator running at a constant speed. Figure 2.5 illustrates the compressor characteristic with pressure ratio against a non-dimensional reduced air flow.

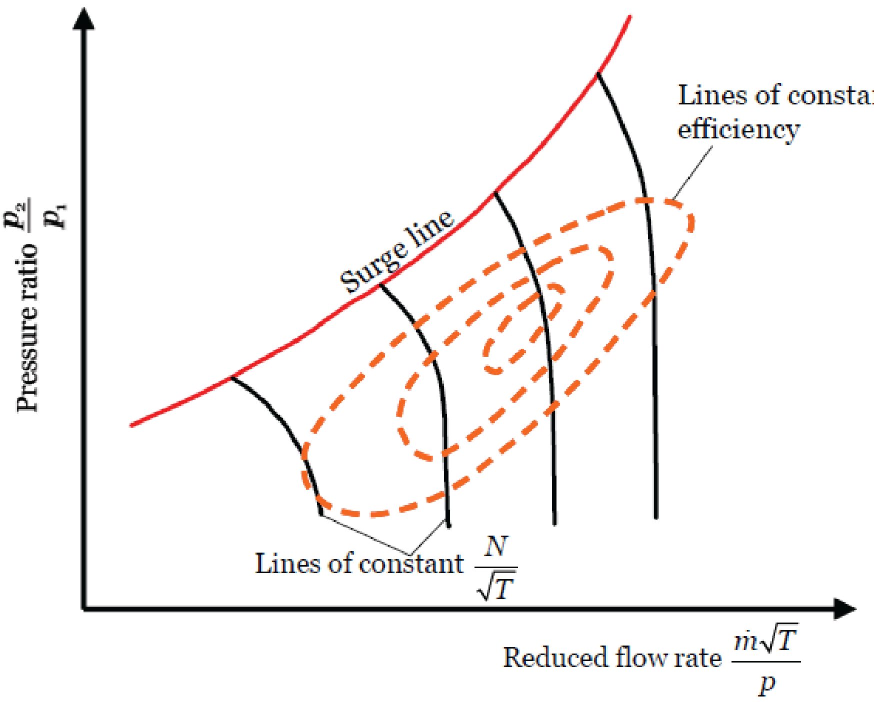

Figure 2.5 - Compressor charecteristic[3].

If the compressor is running at constant speed, it is assumed a constant volumetric sucking capacity of the inlet of the compressor with respect to velocity[3]. From the ideal gas law

$$pv = \frac{p}{\rho} = RT \tag{2.14}$$

and the equation of continuity

$$\dot{m} = \rho u A_c \tag{2.15}$$

the following derivation is obtained

$$u = \frac{\dot{m}RT}{pA_c} \tag{2.16}$$

For a constant speed the following equation is true, for the compressor inlet

$$\left(\frac{\dot{m}RT}{pA_c}\right)_0 = \left(\frac{\dot{m}RT}{pA_c}\right) \tag{2.17}$$

where the index 0 represents the design point. Equation 2.17 may be rewritten to

$$\frac{\dot{m}}{\dot{m}_0} = \frac{p}{p_0}\frac{R_0}{R}\frac{T_0}{T}\frac{A_c}{A_{c,o}} \tag{2.18}$$

If the ambient conditions are kept constant, it follows from equation 2.18 that the inlet air flow is proportional to the cross-sectional inlet area. Thus, for a constant inlet area the mass flow is constant. In order to reduce the air flow into the compressor, inlet guide vanes are used. The air flow into the compressor could be controlled by regulation of the position of the IGVs. Without the use of IGVs, the entry velocity at the first stage in the compressor is purely axial[10] As the IGVs are changing their position, the air flow is guided to the next stage. The flow could be guided in an angle that reduces its axial velocity. With the use of IGVs it is possible to reduce the air flow at a fixed rotational speed. The IGVs are mainly used in the first stage of the compressor, however, Variable Stator Vanes (VSV) could be used at more stages in order to improve the air flow reduction[11]. If IGVs are the only variable vanes, the air reduction is limited to about 15% of the air flow in the design point, while if VSVs are used as well, the air flow could be reduced about 30%[11].

The Air-Fuel ratio (AF) in the combustor depends on the part-load control method. If the AF is constant for reduced load, the turbine inlet temperature remains constant[12]. A constant AF contributes to an unchanged CO_2 mol% in the flue gas. As the TIT decreases, the AF is increased and, though, the CO_2 mol% in the flue gas is decreased.

A compression is an irreversible process, thus an isentropic efficiency or polytropic efficiency is often used in order to correct the compression work for the irreversible losses that occur. The following equation describes the compressor work

$$\dot{W}_{comp} = \frac{1}{\eta_{is}} \dot{m} c_p T_1 \left(\left(\frac{p_2}{p_1} \right)^{\frac{\kappa - 1}{\kappa}} - 1 \right) \tag{2.19}$$

where index 1 is the inlet and index 2 is the outlet of the compressor. κ is equal to c_p/c_v. The isentropic efficiency could be assumed constant for part-load operation[3]. A decrease in the air mass flow because of a regulation in the IGVs is positive for the compressor work, as less work is requested.

2.2.1.2 Turbine

The expansion work in the turbine is given by equation 2.20.

$$\dot{W}_{tur,exp} = \eta_{is} \dot{m} c_p T_3 \left(1 - \left(\frac{p_4}{p_3} \right)^{\frac{\kappa - 1}{\kappa}} \right) \tag{2.20}$$

As for the compressor, also the isentropic efficiency for the turbine could be assumed constant at part-load operation[3]. At reduced load in the gas turbine, either the fuel is regulated exclusively or in a combination with a reduced air flow. The air flow is reduced by IGVs. In order to explain the parameter changes in equation 2.20, an important operating condition must be considered. The gas turbine is very often operating as choked. In a choked condition, the mass flow rate will not increase when the upstream pressure is constant and the downstream pressure is further decreased. In other words, an increase in the pressure drop through the turbine will not increase the mass flow rate. Based on non-dimensional numbers for the reduced flow rate, $\frac{\dot{m}\sqrt{T}}{p}$, and constant rotational speed, $\frac{N}{\sqrt{T}}$, the Figure 2.6 illustrates the choked condition in the turbine

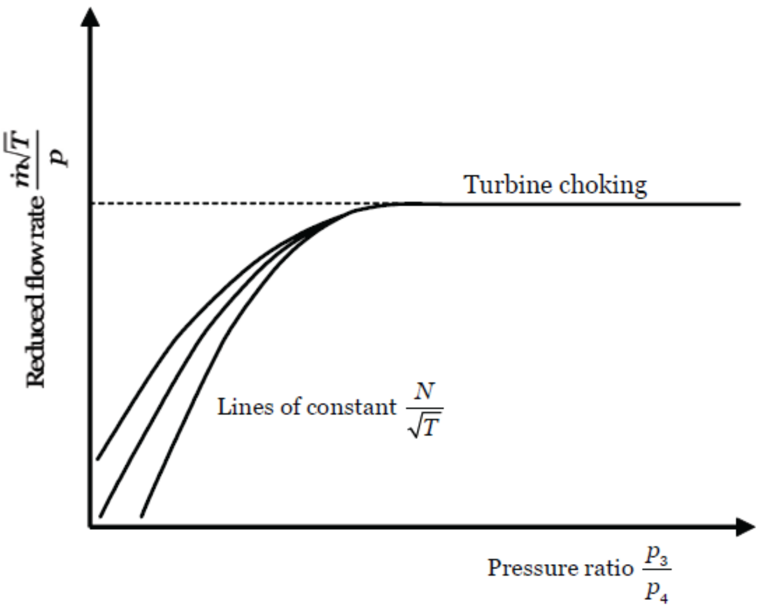

Figure 2.6 – Turbine characteristic[3].

As the pressure ratio is changed, the reduced mass flow rate is constant until very low pressure ratios are obtained. Thus the following relation is derived

$$\frac{\dot{m}\sqrt{T}}{p} = constant \qquad 2.21$$

$$\left(\frac{\dot{m}\sqrt{T}}{p}\right)_3 = \left(\frac{\dot{m}\sqrt{T}}{p}\right)_{3,0} \qquad 2.22$$

$$\frac{p_3}{p_{3,0}} = \frac{\dot{m}_3}{\dot{m}_{3,0}}\sqrt{\frac{T_3}{T_{3,0}}} \qquad 2.23$$

where suffix 3 is the turbine inlet and 0 is the design point.

Equation 2.23 could be used when the turbine is assumed to operate in the choked area, and the working fluid is air. It indicates a close connection between the pressure, mass flow and temperature at the turbine inlet. If the gas turbine is operating at reduced load, and the air flow is kept constant, the inlet turbine mass flow is almost constant. Based on equation 2.20 and 2.23, the TIT is then controlling the expansion work in the turbine. The TIT is controlled by regulation of the fuel flow. A reduced fuel flow at constant air mass flow, gives a lower TIT and therefore a lower expansion work in the turbine. In addition, the compressor will be affected of the changes in the turbine parameters. The calculation of the necessary fuel flow in order to obtain a specific power output is an iterative process.

If the gas turbine is operating with regulated IGVs at part-load, the mass flow in equation 2.23 is reduced. In that case, the TIT is controlled in combination with the regulation of the air flow in order to obtain the specific power output. The changes in compressor work needs to be regarded, and also this control method is an iterative process. The main difference between these two control methods is the possibility to regulate the exhaust gas temperature. For the TIT control method, with constant air flow, there is only one option in parameter regulations for each power output and the exhaust gas temperature decreases for reduced load.

For the part-load control with a simultaneously reduce of the air- and fuel flow, there are several combinations that gives the demanded power output and the TIT or TET could be controlled. However, there are limitations in the use of IGVs. The angles of the IGVs have a maximum, and the lower limit of the air flow rate is in the range 60-70% of the air flow in the design point[3].

The flame temperature in the combustion chamber is sensitive to a regulation in the fuel or-/and air flow. There are restriction regarded the emissions of NO_x and CO in the power plant, and the emissions are dependent of the flame temperature. A high flame temperature increases the NO_x emissions, while a low flame temperature increases the CO emissions. Thus, in order to meet the requirements the flame temperature must be kept within a certain temperature range.

2.2.2 Steam Cycle

The gross efficiency of the steam cycle depends upon the effectiveness of the heat recovery of the exhaust gas from the gas turbine and the power generation in the steam turbine from the heat recovered. The relation appears in equation 2.24

$$\eta_{SC} = \eta_{HRSG}\eta_{ST} \qquad 2.24$$

where η_{SC} is the overall gross steam cycle efficiency, η_{HRSG} is the actual heat recovered in the HRSG divided by the remaining heat in the gas turbine exhaust gas and η_{ST} is the gross steam turbine power output divided by the actual heat recovered in HRSG. The remaining heat in the gas turbine exhaust gas is in the current work defined as

$$\dot{Q}_{GT,Exh} = \dot{Q}_{Fuel} - \dot{W}_{GT} \qquad 2.11$$

This definition is not theoretically correct, since the actual exhaust gas heat will be slightly higher than the heat in equation 2.11. The gross gas turbine power output, $\dot{W}_{GT}$, is lower than the gas turbine enthalpy change due to mechanical and generator losses. However, in order to simplify the current work, the definition in equation 2.11 is used.

A parameter change in the combined cycle could affect the HRSG- and steam turbine efficiency in opposite directions[13]. A high live steam pressure is advantageous for the steam turbine, while a low live steam pressure is profitable for the heat recovery in the HRSG.

In the three pressure level steam cycle process described in chapter 2.1.2.1, the benefits related to the efficiencies discussed above are utilized. The highest pressure level provides good steam turbine efficiency, while the lowest pressure level results in an effective heat recovery. The effect of the intermediate pressure is a reduced loss in exergy related to the heat transfer between the exhaust gas and the water in the HRSG. The reduced loss in exergy is due to reduced temperature differences in the heat exchangers.

The steam turbine efficiency will gain in a high pressure level and temperature in the steam cycle at part load, while the HRSG efficiency is highest when most of the exhaust gas heat from the gas turbine could be utilized. A high utilization of the exhaust gas is equal to a low stack temperature. In order to utilize the heat, the pressure levels must be sufficient low. In addition, a low steam mass flow could result in a low condenser pressure.

2.2.2.1 Steam turbine

The steam cycle should be operated in sliding-pressure mode in order to improve its part-load efficiency. The swallowing capacity for the steam turbine is given by the Law of Cones[5]

$$\frac{\dot{m}_S}{\dot{m}_{S,0}} = \frac{\bar{V} \cdot p_\alpha}{\bar{V}_0 \cdot p_{\alpha,0}} \sqrt{\frac{p_{\alpha,0} \cdot v_{\alpha,0}}{p_\alpha \cdot v_\alpha}} \sqrt{\frac{1 - \left(\frac{p_w}{p_\alpha}\right)^{\frac{n+1}{n}}}{1 - \left(\frac{p_{w,0}}{p_{\alpha,0}}\right)^{\frac{n+1}{n}}}} \tag{2.25}$$

where $\dot{m}$ is mass flow, $\bar{V}$ is the average swallowing capacity, p is pressure, v is specific volume and n is the polytrophic efficiency. The suffixes used are: S is steam, 0 is design point, α is section of a turbine inlet and w is section of turbine outlet. The pressure ratio between the inlet and outlet on the turbine is very small due to low condensation pressure, thus the square term could be replaced by 1. It could also be assumed an average swallowing capacity ratio of 1[5]. The equation 2.25 is then replaced by the simplified equation 2.26

$$\frac{\dot{m}_S}{\dot{m}_{S0}} = \sqrt{\frac{p_\alpha \cdot v_{\alpha 0}}{p_{\alpha 0} \cdot v_\alpha}} \tag{2.26}$$

When the steam turbine operates off-design, the steam mass flow is proportional to the square root of the pressure. Thus, the relative pressures in the steam cycle are reduced when the gas turbine operates at a lower power output, due to reduced steam generation.

There might be problems related to the steam cycle at very low live steam pressures. It is recommended to control the throttle valves at the HPT inlet in order to fix the Live Steam Pressure (LSP) if the pressure is less than 50% of the pressure in the design point[5].

The dry step isentropic efficiency in the steam turbine could be assumed approximately constant at part-load operation[5]. When a sliding pressure mode is used, the overall isentropic efficiency is therefore approximately constant for the HPT and IPT in a three pressure level cycle with reheat. When the load is reduced, and the LST is kept constant, the vapor fraction at the turbine exit will increase due to a lower live steam pressure. Thus the overall isentropic efficiency will increase in the LPT for a three pressure level cycle with reheat.

If a throttle control at the turbine inlet is used in order to fix the live steam pressure, the steam will expand isenthalpic through the throttle valve before it is further expanded in the turbine. Because of the expansion in the throttle, the overall isentropic efficiency will be reduced. This is illustrated in Figure 2.7. The dry isentropic efficiency for the expansion from B-C in Figure 2.7 is equal to the efficiency at the design point, but the overall isentropic efficiency will be reduced because of the expansion from A-B. The result is a reduced enthalpy drop. In the current work, a sliding pressure mode at the HPT is used at part-load.

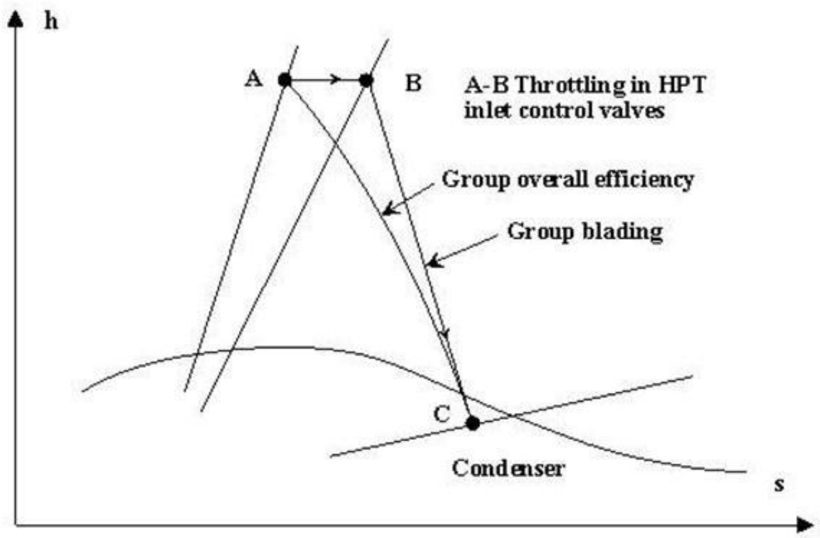

Figure 2.7 – Steam turbine expansion path with partially closed inlet control valves [14].

Regarding a combined cycle with CO_2 capture, steam needs to be extracted from the steam turbine. The extraction pressure may be restricted, and throttle valve control may be necessary. This will be explained more in chapter 7.3.

2.2.2.2 Heat recovery steam generator

In the design of the HRSG the heat exchanger surfaces are calculated by means of the desired steam pressures and temperatures, the geometry of the tubing decided and the desired flue gas pressure drop through the HRSG[14]. At part-load operation of the gas turbine, the flue gas stream is entering the HRSG in another condition relative to the design point. As the heat exchanger surfaces are fixed, obviously, the calculations within the HRSG are turned around. The flue gas inlet conditions and the heat exchanger hardware are used in order to compute the pressure drops for both the water/steam side and the flue gas side, the temperature and pressure levels, the heat transfer and the plant's heat balance[14]. The calculations are restricted by the other components in the plant, such as the steam turbine and the condenser.

At part-load operation, the overall heat transfer coefficient in the heat exchangers will be changed[15]. A simplified expression of the relation between the design- and the off-design overall heat transfer coefficient is given in equation 2.27[15].

$$\frac{UA}{(UA)_0} = \left(\frac{\dot{m}_h}{\dot{m}_{h,0}}\right)^m \qquad 2.27$$

where U is the overall heat transfer coefficient, A is the heat transfer area and $\dot{m}$ is the flue gas mass flow. The index h refers to the hot side of a heat exchanger and the index 0 is the design point. The exponent m is a constant and depends on the geometry of the heat exchanger[15]. For staggered tubes, the value of the exponent m is 0,56-0,58[6].

At part-load operation in the gas turbine, the flue gas mass flow is reduced, while the heat exchanger area is constant. Hence, the overall heat transfer coefficient is reduced. The total heat transfer in a counter-current heat exchanger is given by equation 2.28.

$$Q = U \cdot A \cdot LMTD \tag{2.28}$$

where $LMTD$ is the logarithmic mean temperature difference between hot and cold side in a heat exchanger. The total heat transfer Q is proportional to the flue gas mass flow. Thus, a lower mass flow will result in a relative greater reduction in Q compared to U, because of the exponent m in equation 2.27. That means, a reduction in the flue gas mass flow leads to a lower $LMTD$. In an evaporator, a lower $LMTD$ would probably result in a reduced pinch.

Also, the pressure drop through the HRSG will decrease as the exhaust gas mass flow is reduced.

2.2.2.3 Condenser

There are several options for off-design modeling of the condenser. One option is simply to fix the cooling water circulation rate, regardless of the reduced load. In that way, the cooling water pumps are running at an almost constant speed. As the gas turbine load is reduced, the steam generation in the HRSG is reduced and less steam needs to be condensed in the condenser for a given condenser pressure. Figure 2.8 shows a T-Q diagram of a water-cooled condenser. If the inlet cold end temperature of the cooling water is kept constant, the slope of the cooling water line will be equal to the slope in the design point. As the steam mass flow is reduced, the required heat transfer from the steam to the cooling water is reduced. Thus a higher temperature difference at the hot end outlet of the cooling water will occur, if the condenser pressure is kept constant.

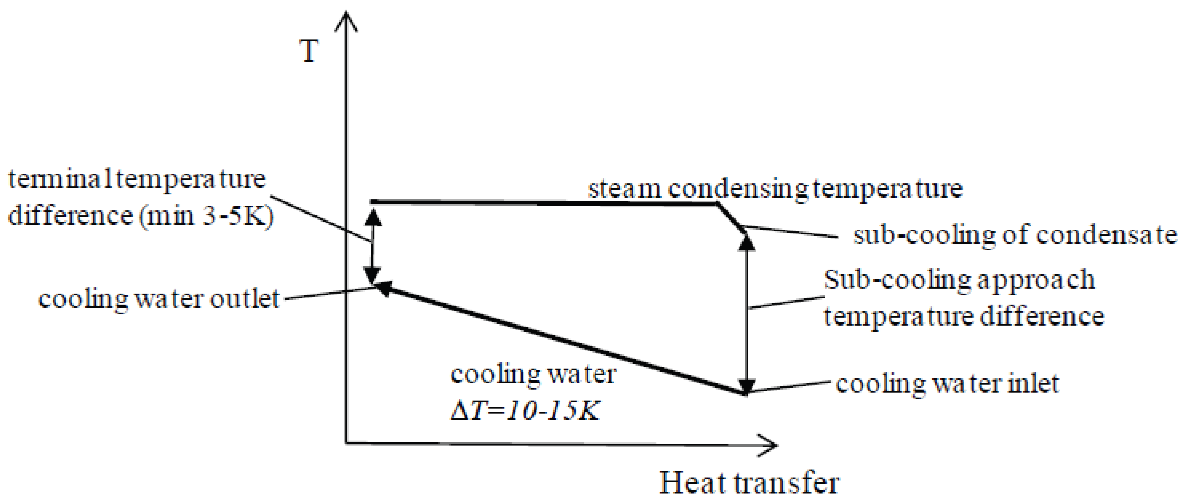

Figure 2.8 – T-Q diagram for a water-cooled condenser[3].

A lower steam condensing pressure may be obtained. If the condenser pressure is reduced, the operating line of the steam in Figure 2.8 is moved further down. The heat of evaporation increases as the pressure is reduced, as seen in Figure 2.9, thus the total heat transfer between the streams in the condenser increases compared to a constant condenser pressure. In order to find the lowest possible condenser pressure for the given mass flows and inlet cooling water temperature, an iteration process is required. Also the pinch temperature difference at the condenser hot end will change at part-load operation, due to a change in the overall condensing heat transfer coefficient and the total heat transfer. The variations in the pinch need to be considered in the iteration process.

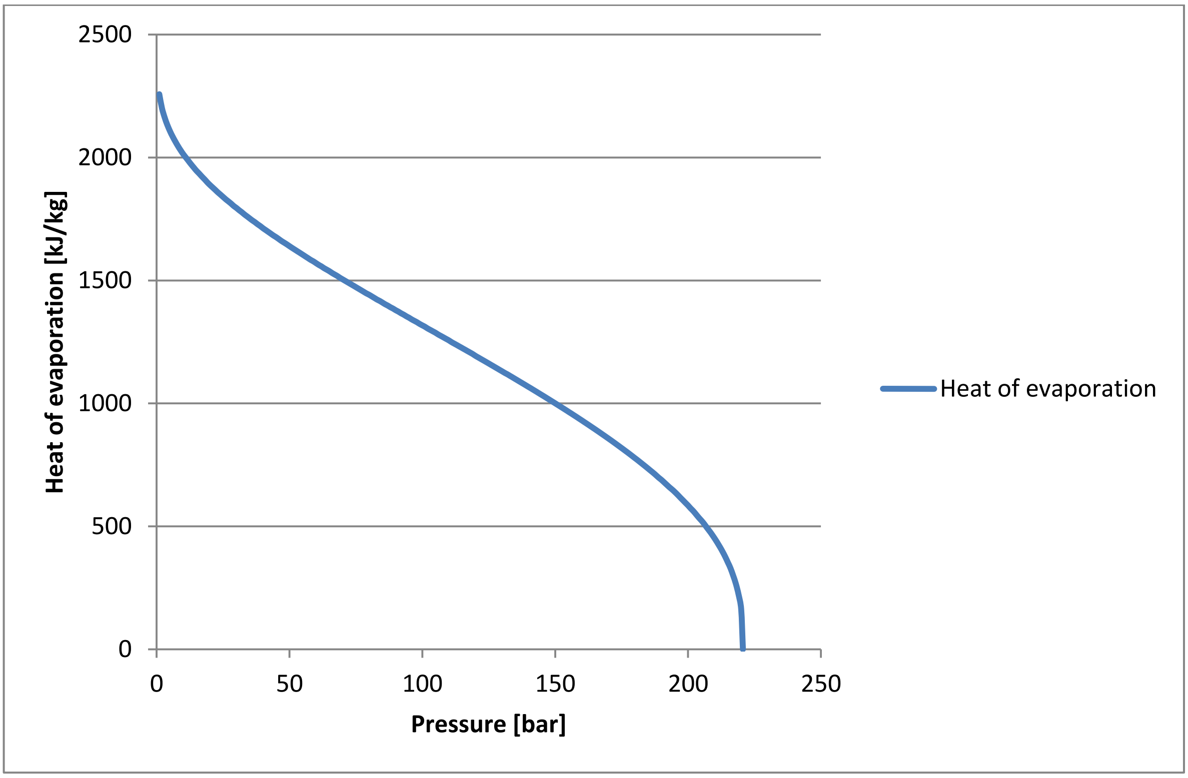

Figure 2.9 – Heat of evaporation for water at different saturation pressures.

As the condensation pressure is reduced, more power is generated in the turbine. The drawback with a very low condensing pressure is a high volumetric flow at the condenser outlet. As the pipe area is constant, due to the design, the velocity of the condensed steam will increase.

It is also possible to regulate the cooling water flow by a regulation in the pump capacity or by shut-off one or more pumps, dependent of the number of pumps in the system. In that case, the mechanical energy in the pump is reduced. However, a lower cooling water flow rate is circulating and the condenser pressure will be higher compared to the case where the pumps are running at full capacity. Thus, the steam turbine output is reduced. The volumetric flow of the condensed steam will be lower by reducing the circulated cooling water.

In the current work, the cooling water is circulating in a constant rate at part-load operations.

3. CO_2 Capture Plant

The absorption process is a well-established process in the gas processing industry; it is used in the natural gas sweetening process for removal of CO_2 and other acidic gases. The main difference from the gas production to the flue gas cleaning is the pressure level. Natural gas sweetening takes place at a pressure level of around 60 bar, and in post-combustion capture the pressure is atmospheric 1.013 bar[16].

A simple flow sheet of the absorption process is illustrated in Figure 3.1. The flue gas enters the absorber after a treatment in the flue gas cooler. In the absorber, most of the CO_2 in the flue gas is reacting with the liquid solution. The treated flue gas is released to the atmosphere, while the liquid stream is treated in a stripper at a higher temperature. The gas flow out of the stripper contains a high fraction of steam, thus before the CO_2 is compressed to its storage pressure the steam is condensed and sent back to the stripper. The lean liquid stream is flowing back to the absorber.

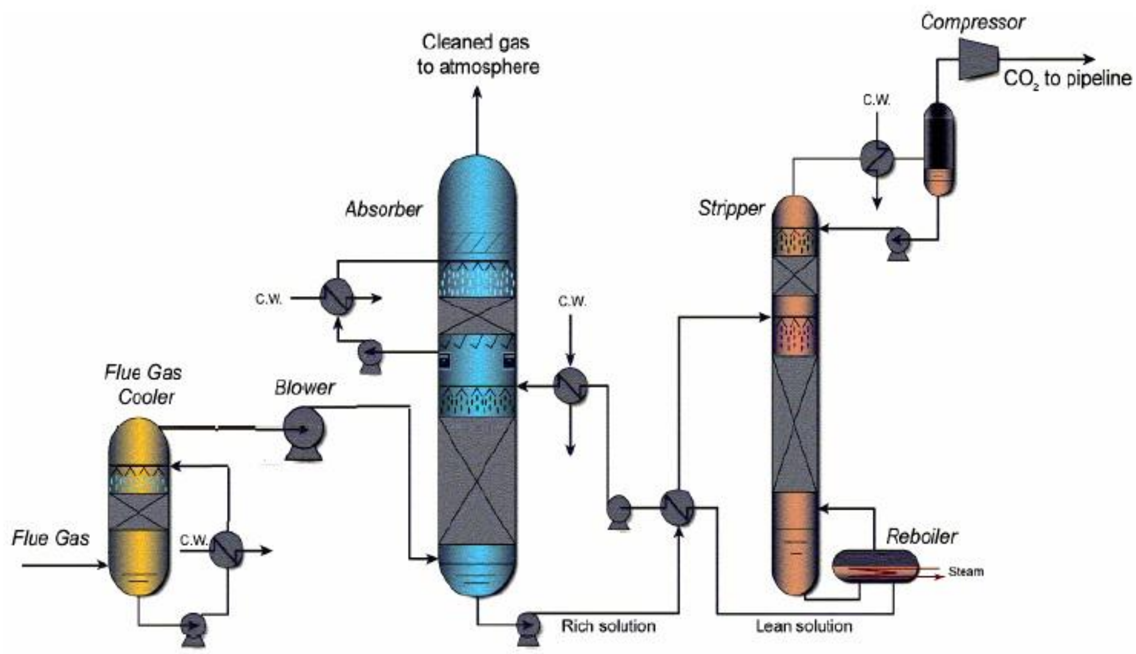

Figure 3.1 - Flow sheet of post-combustion CO2 capture plant[17].

3.1 Absorption Process

3.1.1 Inlet Condition

The flue gas from the power plant has a temperature in the range of 80-100 °C[3] and an atmospheric pressure. The temperature should not be too low because of condensation of water in the pipes prior to the capture plant. This gas only contains 3-4 mol% of CO_2 which gives a very low partial pressure for the CO_2, 0,03-0,04 bar[20]. The rest of the gas is a variation e.g. O_2, N_2 and H_2O. In some flue gases, traces of NO_x and SO_2 may be found, these gases need to be pretreated before the gas enters the absorption column because of degradation of the solvent.

The flue gas temperature is reduced in a flue gas cooler, and in the cooler liquid is condensed out of the gas. The liquid is mostly water, but it is possible to find traces of CO_2 in it as well.

3.1.2 Absorption Column

In the absorption column the flue gas is mixed with water and the absorbent. An absorber column is illustrated in Figure 3.2. The absorption column is divided into different trays. On the top tray the solvent is sprayed over structured packing or random packing[21], which is designed to have a very high area per volume, typical 150 m2/m3[22]. Because of the high area per volume the contact area between the liquid stream and the gas stream is great. The gas flows into the bottom of the absorber and the solvent at the top.

The absorption process is exothermic, thus the reactions that occur in the absorber column releases heat. At the bottom of the column the CO_2-rich solution leaves the absorber at a temperature of 40-50°C. The treated gas leaves the absorber in a vent at the top of the column. This type of absorption column can obtain a removal efficiency of over 90% with the correct operation condition and design[23].

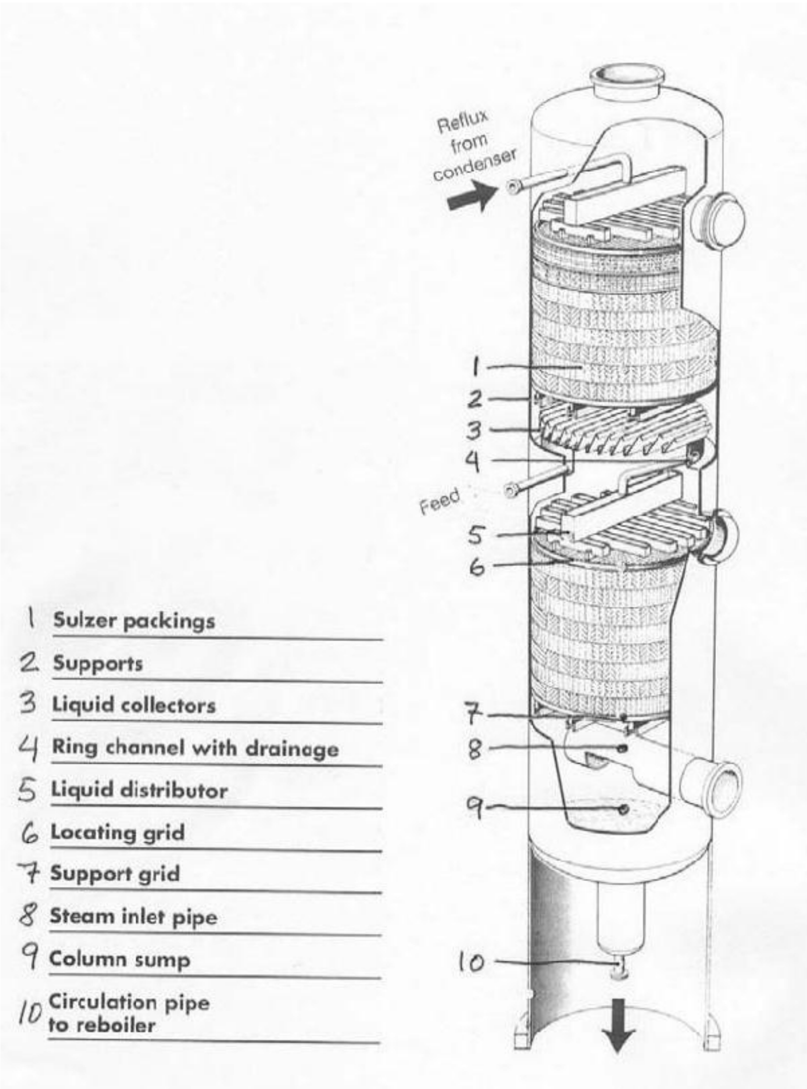

Figure 3.2 - Absorption column schematic[24].

3.1.3 Heat Exchanger

The rich solution is pumped through a cross-flow heat exchanger before it enters the stripper column. In the heat exchanger, heat is transferred from the warm lean solution that is leaving the stripper column to the rich solution. By using this rich/lean heat exchanger the rich solution can be heated up to about 110°C before it is entering the stripper. Thus, a lot of "free" heat is utilized. Another benefit from having this rich/lean heat exchanger is that the necessity of cooling water for cooling of the lean solution is reduced. The heat transfer coefficient in the heat exchanger is reduced when two phase flow occur. In order to minimize the two phase flow in the heat exchanger, the pressure in the rich liquid flow is increased prior to the heat exchanger by a pump. There will be a fraction of gas phase after the heat exchanger. The rich flow is throttled down to stripper pressure and the gas fraction increases.

3.1.4 Stripper

The entering stripper pressure of the rich liquid flow is about 2 bar. In the stripper, a desorption process occur. This process is opposite to the absorption process. Heat is added to release the CO_2 from the amine. The separator works as a distillation column and a great contact area between the liquid- and gas flow is important. In the stripper, the main concept is the heating of the rich solution. However, some of the CO_2 in the liquid flow will be released to the gas stream. The rich solution flows downward in the column and the gas flow rises up. The heat is produced in the reboiler. Most of the stripping of the CO_2 from the takes place in the reboiler. The boilup from the reboiler is sent back into the separator to heat up the rich solution. The remaining liquid stream is transported back to the absorption column. The temperature of the lean solution leaving the desorber is typically around 120°C, but it should not exceed 122°C because of degradation and corrosion issues[26].

3.1.5 CO_2 Compression

CO_2 should be transported and stored in liquid form. The CO_2 leaving the stripper has a pressure of 2 bar and a temperature of 30°C. For the CO_2 to be in liquid form at ambient temperature the pressure needs to be above 80 bar; see Figure 3.3. When transporting CO_2 in pipes, there are always pressure losses. To prevent two phase flow in the pipes, the CO_2 is normally compressed to 100-150 bar[3]. In the current work, the CO_2 compression has not been analyzed. However, the energy required in a compression process has been calculated by means of the following expression[27]. It is assumed an end pressure of 100 bar.

$$W_{comp} = 0{,}33 \frac{MJ}{kg\ CO_2} \qquad (3.1)$$

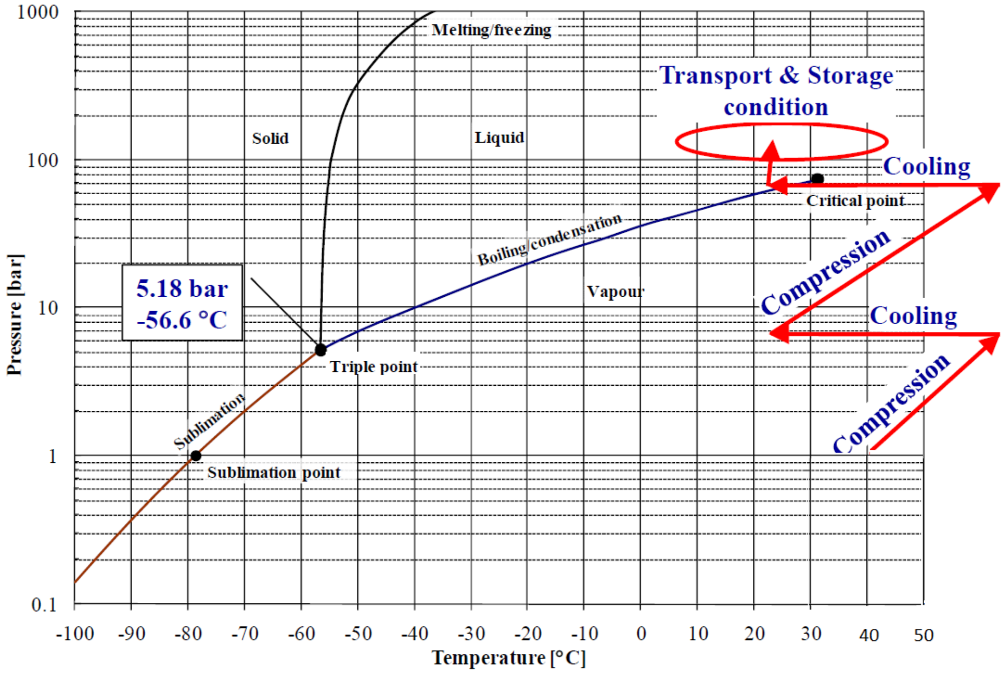

Figure 3.3 - Gas-phase separation[3].

3.2 Loading Capacity and Choice of Solvent

An important parameter in the absorption process is the CO_2 loading. The loading is defined as mole CO_2 per mole solvent. The equilibrium CO_2 loading refers to the minimum number of moles solvent required to absorb one mole of CO_2. The actual number of moles solvent is higher than this minimum value due to deviation from equilibrium in the absorber. To obtain equilibrium in the absorber, an infinity large packing is required. In addition, the liquid stream into the absorber will contain a fraction of CO_2 because of a not complete stripping. The heat required in the reboiler in order to regenerate the CO_2 from the liquid stream depends on the purity of the stripping. This will be explained in details in chapter 3.3. The actual circulation loading is defined as

$$Loading_{circ} = Loading_{rich} - Loading_{lean} \tag{3.2}$$

As seen from the definition above, the total circulation loading decreases when the lean loading increases or-/and the rich loading decreases. Since the loading is defined as moles CO_2 per moles absorbent, the required circulation of absorbent increases for decreasing circulation loading.

Figure 3.4 illustrates two typical equilibrium loading curves for CO_2 in an aqueous alkanolamine solution against the partial pressure of the CO_2 in the gas phase. The equilibrium loading is the highest loading the CO_2 could obtain in the liquid phase[3].

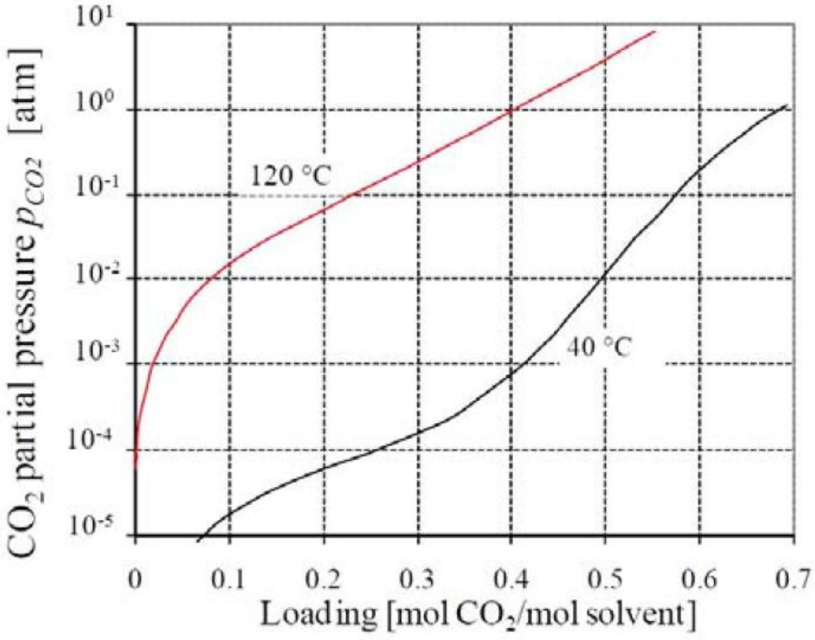

Figure 3.4 - Solubility of CO2 in aqueous alkanolamine[3].

Figure 3.4 clearly illustrates how the loading is affected of the temperature. In the absorber it is preferable with a low temperature in order to obtain a high loading. In the desorption process, however, a high temperature is advantageous in order to reduce the loading.

It could also be seen how the loading, at 40°C, increases rapidly with increased CO_2 partial pressure in the low pressure range. As the partial pressure increases further, the loading curve is steeper. For primary amines, such as MEA, the capacity of CO_2 in the solution is restricted to about 0,5 mole CO_2/mole amine even at high partial pressures. The reason is the high stability of the formed carbamate. The carbamate ion ties up a an akanolammonium ion, thus the CO_2 capacity in the solution is reduced[19].

Tertiary amines, such as methyldiethanolamine (MDEA), are unable to form carbamate and a higher CO_2 capacity in the solution may be achieved. However, the reaction is very slow. The reaction between CO_2 and primary amines are faster. As the reaction is faster, the contactor sizes could be reduced compared to slower reactions.

MEA is chosen as the preferred amine in the current work, due to its fast reaction time and its relative high absorption capacity at atmospheric pressure. However, one of the most serious disadvantages related to MEA is its high heat of absorption.

The amount of MEA being used may vary from 15-30 wt.%[19]. Due to the high corrosiveness of MEA only low wt.% can be used, but with effective corrosion inhibitors in the system the wt.% of MEA may come up to 30. If the flue gas contains SO_2, NO_x or high levels of O_2 degradation of the MEA may occur. Degradation of the solvent reduces its loading ability, and may even destroy it completely.

3.3 Energy Demand

The energy demand in the CO_2 capture plant is high. Mechanical energy is required in the pumps related to the lean- and rich liquid flow and the cooling water. In addition, the blower in front of the absorber needs mechanical energy. This blower is the second most energy consuming unit in the capture plant. The most demanding part is the reboiler.

The heat requirement for the reboiler could be divided into three parts

$$\dot{Q}_{reb} = \dot{Q}_{sensible} + \dot{Q}_{latent} + \dot{Q}_{des} \tag{3.3}$$

where $\dot{Q}_{reb}$ = Total heat demand in the reboiler to regenerate the solvent

$\dot{Q}_{sensible}$ = Heat to raise the solvent from inlet stripper temperature to reboiler temperature

$\dot{Q}_{latent}$ = The heat required to evaporate the water in the stripper

$\dot{Q}_{des}$ = The heat of absorption of the solvent with CO_2

3.3.1 Heat of Desorption

In contrast to the absorption, the desorption process is endothermic. The same amount of heat released in the absorption process needs to be supplied back to the desorption process in order to break the chemical bonds between the CO_2 and the MEA.

The specific heat of absorption for MEA is almost constant, at a temperature of 40°C, for different CO_2 loading until the saturation loading point is reached[28]. At the same time, the heat of absorption is strongly dependent on temperature and increases for increasing temperatures[28]. The heat of absorption is about 85 kJ/mole CO_2 for CO_2 loading up to 0,5 mole CO_2/mole MEA[19]. This corresponds to almost 2 MJ/kg CO_2.

3.3.2 Sensible Heat

The CO2-rich amine solution that enters the stripper has a temperature below the boiling temperature of the solution. The energy that is required to heat the solution up to reboiler temperature is called sensible heat. This heat is proportional to the liquid mass flow in the stripper. For simplification this mass flow, in addition to the specific heat capacity, are often assumed constant over the stripper range[29].

An increased CO_2 concentration in the inlet gas leads to a higher loading capacity, and therefore a lower circulation rate of the solute. When the circulation rate of the solute is reduced, the liquid mass flow into the stripper decreases and the sensible heat is lower.

3.3.3 Latent Heat

The latent heat is the heat required to evaporate the water in the reboiler. The amount of vaporized water is given by the steam demand in the stripper, and the latent heat is proportional to this water flow. When the liquid is flowing down the stripper column, it is heated by the hot gas stream, and some of the CO_2 is released to the gas stream. In order to make this transport possible, the partial pressure of the CO_2 in the gas must be lower than the saturation pressure in the liquid solution. In addition, to ensure that the CO_2 in the gas stream is not condensed, the partial pressure must be lower than the saturation pressure. In order to reduce the partial pressure of the CO_2 in the gas stream, steam must be generated. The saturation pressure increases when the CO_2 concentration in the liquid stream increases and vice versa.

An increased CO_2 mol% in the inlet gas leads to decreasing latent heat[30]. This is due to a higher rich concentration, and thus the saturation pressure in the top of the stripper is increased.

3.3.4 CO_2 Concentration

The specific reboiler duty increases as the CO_2 concentration in the flue gas decreases. This could be seen from the equation of minimum thermodynamic work required to compress the CO_2 from the inlet of the absorber to the desorber outlet at isothermal conditions[29].

$$\frac{W_{min}}{n_{CO_2}} = \Delta G_{T,P} = RTln\frac{P_{CO_2,out}}{P_{CO_2,in}} \qquad 3.4$$

Thus, a lower CO_2 concentration in the flue gas gives a higher minimum compression work.

3.4 Process Parameters and Restrictions

3.4.1 Flooding and Minimum Liquid Load

In the absorption column there are two mechanisms limiting the design, flooding and minimum liquid load. This is illustrated in Figure 3.5.

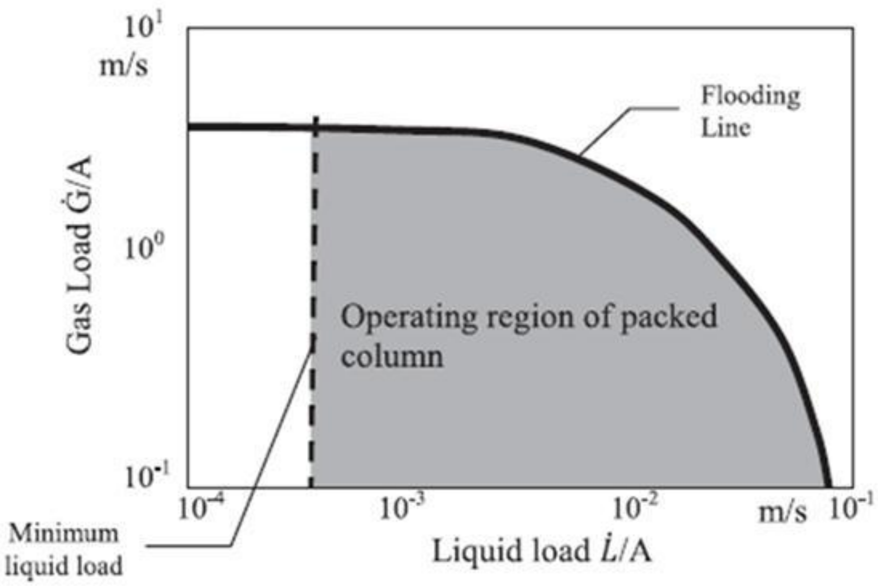

Figure 3.5 – Operating region of a packed column[31].

Liquid load is the liquid mass flow divided by the cross-sectional area of the column. Flooding occurs when the liquid is no longer able to flow downwards, and fills the entire column. This is due to a high gas velocity and liquid flow. That combination will cause a great resistance from the liquid stream and high pressure drop across the packing. At flooding the absorption process is difficult, and the pressure drop increases dramatically and may destroy the packing material. It is important to design the absorber in a way that reduces the risk of flooding. The gas flow is given by the power plant, thus the only parameters to regulate is the diameter of the column and the liquid flow rate. A low gas load and-/or a low liquid load could be obtained by increasing the diameter of the absorber.

For part-load, with lower flue gas flow, the gas load will decrease. As the liquid flow does not increase by a reduction in the gas flow, there will be no problems related to flooding at part-load operation of the power plant with lower power output. Thus in this thesis, flooding has not been taken into account and the diameter is set to 10m.

To ensure a great contact between the liquid flow and the gas flow in the absorber, the liquid load must exceed a certain level. If the load is below this value, the entire packing surface is not wetted and the effective contact area is reduced. Since the liquid load is the liquid mass flow divided by the cross sectional area of the column, an increase of the diameter contributes to a lower liquid load. As could be seen from Figure 3.5, the limiting liquid load is independent of the gas flow.

There might be problems related to minimum liquid flow when the power plant operates at part-load. In order to optimize the reboiler duty, the liquid flow may be reduced when the gas flow is reduced and its composition is changed. For absorption processes with a solvent with fast reaction rate, and with a reaction that is equilibrium limited, the required size of the column depends strongly on the contact effectiveness between the gas and liquid[32]. Thus, the importance of avoiding incomplete humidification of the packing material is central in order to reduce the risk of a dramatic decrease in absorption efficiency. A large diameter in addition to a reduced liquid flow increases the risk of a not completely wetted packing material.

3.4.2 Pressure Drop

Pressure drop occurs at the inlet of the absorption column. To prevent this from affecting the turbine in the power plant a blower is installed to maintain an even pressure level. The pressure drop depends on the gas velocity, the liquid flow and the height of the absorber among others, and it is a result of the friction forces between the gas and liquid flow. A high gas velocity gives a high pressure drop. In order to regulate the gas velocity, the diameter of the column may be varied. A high pressure drop ensures a good liquid and gas distribution in the absorber[31]. At the same time, the mechanical energy required by the blower increases with the pressure drop. The design of the column pressure drop is an optimization problem with costs, mechanical energy and capture efficiency as parameters.

At part-load, the flue gas flow is reduced and the pressure drop in the absorber will be decreased compared to the design point. Also, changes in the liquid flow will affect the pressure drop. The changes in pressure drop will have consequences for the off-design operation of the capture plant. However, for simplifications the pressure drop is assumed constant in the part-load simulations in the current work.

3.5 Material Balance

In Figure 3.6, the material balance of an absorber is illustrated. Because of a mass transfer between the gas and liquid phase, the gas and liquid flow rates will be changed from inlet to outlet of the column. Thus, to obtain constant flow rates through the absorber solute-free flow rates and mole ratios are used to express the material balance[19].

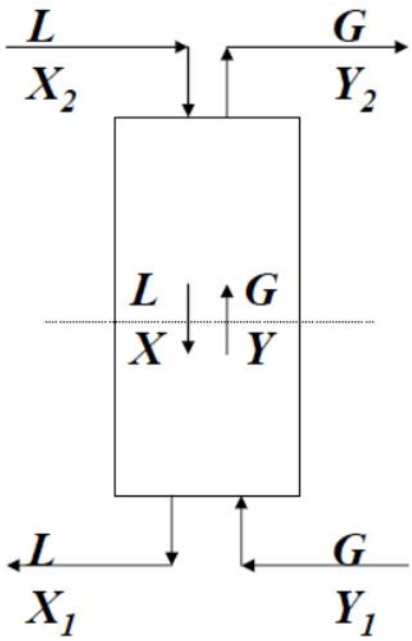

Figure 3.6 – Material balance diagram of a counter-current absorber[20].

The material balance between inlet and outlet of the column is given by

$$G'_M(Y_1 - Y_2) = L'_M(X_1 - X_2) \qquad 3.5$$

where G'_M is the solute free gas flow rate, L'_M is the solute free liquid flow rate, X is the mole ratio solute in the liquid phase and Y is the mole ratio solute in the gas phase. By redirecting equation 3.5

$$\frac{L'_M}{G'_M} = \frac{(Y_1 - Y_2)}{(X_1 - X_2)} \qquad 3.6$$

Equation 3.6 represents the slope of the operating line. This will be further explained in chapter 6.2.2.

3.6 Equilibrium Model

There are two ways of modeling a packed distillation and absorption column; the equilibrium stage model, and non-equilibrium or rate-based model. The equilibrium model makes use of an assumption that the streams leaving an ideal stage or section of packed columns are in equilibrium with each other. To deal with the deviation from equilibrium for a real stage, stage efficiencies or Height Equivalent to a Theoretical Plate (HETP) are introduced.

3.6.1 Height

The HETP is defined as

$$HETP = \frac{Height\ of\ packed\ zone}{Number\ of\ theoretical\ plates\ achieved\ in\ packed\ zone} \qquad 3.7$$

From equation 3.7, the total height of the packing is obtained by multiply the HETP with the number of theoretical plates required. The prediction of the HETP is difficult, and the results are often based on empirical data.

The height of the absorber column could also be calculated by means of the overall mass transfer coefficient concept. If it is assumed that the equilibrium curve is straight, the partial pressures of the inert gases are constant through the column and the solute concentration in the liquid- and gas phase are low the following expression could be derived

$$H = \frac{G_M}{K_G a P} \int_{y_2}^{y_1} \frac{dy}{y - y_e} \qquad 3.8$$

where y refers to the mole fraction of solute in the gas stream, H is the height, G_M is the molar flux of the gas, P is the pressure and K_G is the overall mass transfer coefficient. The subscript e refers to the equilibrium conditions in the opposite phase's bulk stream. If the operating- and equilibrium lines are straight throughout the absorber column, equation 3.8 may be rewritten[19]

$$H = \frac{G_M(y_1 - y_2)}{K_G a P (y - y_e)_{LM}} \qquad 3.9$$

$(y\text{-}ye)_{LM}$ is the logarithmic mean of the driving forces at the top and bottom of the column, and is often used to correlate the $K_G a$ values for the systems in cases where the equilibrium curve is not a straight line[19]. This simplification could result in calculation errors.

A major disadvantage related to the equilibrium model is the uncertainty in the determination of the packing height. This is a big problem in off-design simulations, when the height is kept constant. However, in the current work an equilibrium model is used as simulation tool. In order to calculate the height by use of these methods, knowledge of $K_G a$ and/or the HETP is required. The calculation of these is the most demanding part. Empirical correlations or experimental correlations are frequently used.

3.6.2 Overall Mass Transfer Coefficient

In the overall coefficients concept it is assumed a linear equilibrium curve. Even though the equilibrium line is not straight throughout the column, these overall coefficients are often used in absorber design problems because of its practicability[19].

The liquid flow rate into the absorber has a great influence on the absorption efficiency for structured packings. An increasing liquid flow rate results in a higher absorption efficiency[33]. It is two reasons for the increase; the degree of humidified packing material grows as the liquid load increases and the change in concentration in the liquid solution is lower at high liquid flow rates[33]. An increase in the absorption efficiency means a higher overall mass transfer coefficient. As the absorption efficiency increases, the CO_2 removal rate in the absorber increases unless the reboiler duty is regulated.

The degree of wetted packing surface and its impact on a change in the liquid load depends on the packing used and its geometry. The difference in liquid load dependency for different packings can be observed in Figure 3.7.

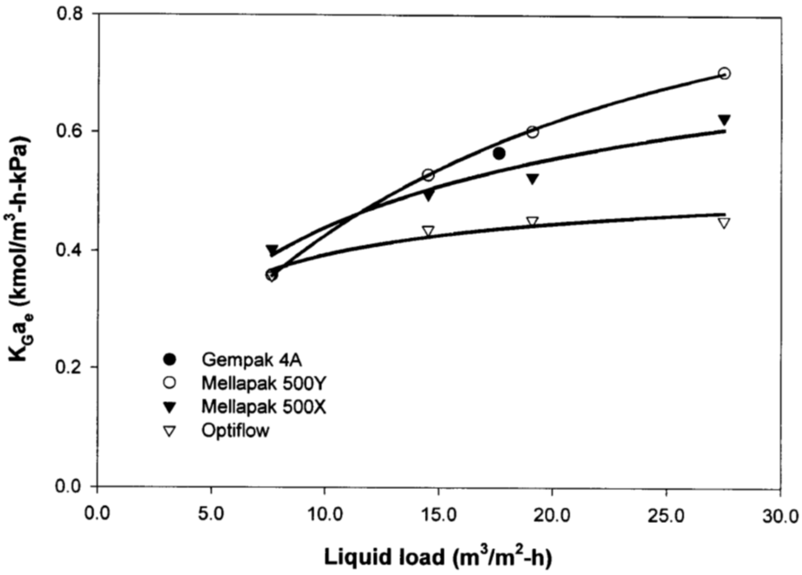

Figure 3.7 – Overall mass transfer coefficient of different structured packings at varied liquid load[33].

In the case of the Mellapak 500Y structured packing, the overall mass transfer coefficient is highly dependent of the liquid load. The packing geometry is allowing the liquid to spread in a high extent over the packing surface. Thus, an increase in the liquid flow rate improves the wetting of the packing surface and the overall mass transfer coefficient increases.

The overall mass transfer coefficient also depends on the CO_2 loading in the liquid flow. A high CO_2 loading results in a low absorption efficiency and a low overall mass transfer coefficient. This is illustrated in Figure 3.8. When the CO_2 loading is high, there is a reduced concentration of free amines. Thus, the reaction rate is slower and the mass transfer rate is reduced[29]

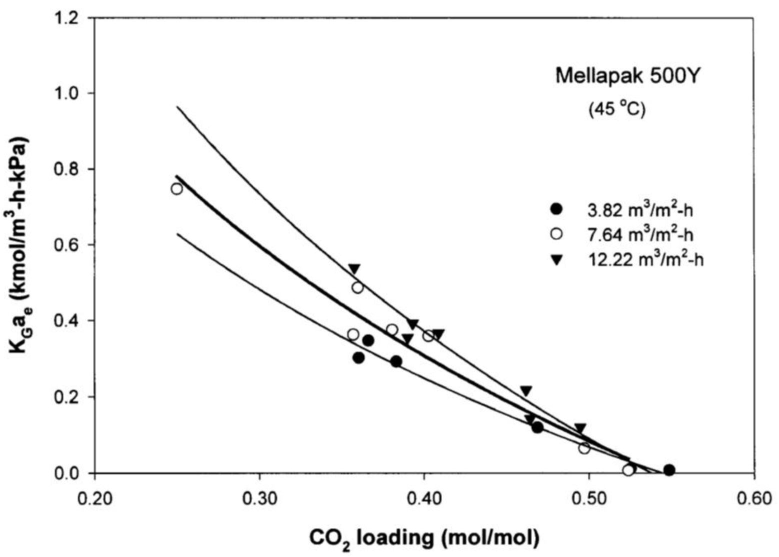

Figure 3.8 – Overall mass transfer coeficcient for three different liquid loads at varied CO_2 loading[33]

3.7 Specific Reboiler Duty

In order to compare the operations and efficiencies of the capture plants for different external- (flue gas conditions) and internal inputs (L/G ratios, lean loading etc) a specific energy demand in the reboiler may be defined

$$SRD = \frac{\dot{Q}_{reb}}{\dot{m}_{CO_2}} \qquad 3.10$$

where SRD is the Specific Reboiler Duty, $\dot{Q}_{reb}$ is the total energy demand in reboiler and $\dot{m}_{CO_2}$ is the mass flow of CO_2 captured. This expression turns out to be a helpful term in the optimization problems of the capture plant. It gives a good indication on how different parameters affect the effectiveness of the capture plant. However, when the effectiveness of the total power plant is calculated, the total energy demand is used.

4. Methodology

In this section, a short description is given for the different modeling cases and methods that are used in the present work, included the simulation programs.

4.1 Process Design and Modeling

4.1.1 Simulation Program

For designing and simulation of the CO_2 capture process the program Aspen Hysys 7.3 from AspenTech[34] has been used. This is a straight forward program with no programming needed. The simulation environment in Hysys is graphical. The parameters are set inside each process block. Color codes are used to systematically provide information on whether or not components are sufficiently defined or within the boundaries of the thermodynamic fluid package. Regarding the absorption process, Hysys uses a tray by tray, or stage by stage equilibrium model. Thus, the program has a limitation regarding the off-design modeling, since it is not possible to specify the height of the absorber. The height is neither calculated in Hysys. However, the number of stages could be specified and fixed for the off-design modeling. The property package that is used in the simulations is Kent-Eisenberg. This property package is recommended for systems containing water, CO_2, one of four ethanolamines such as MEA and also other components typically present in gas-sweetening processes. This package does not include argon, thus the remaining components in the flue gas will obtain a slightly higher concentration in Hysys.

In order to calculate the NGCC design model, with and without steam extraction, the simulation program GT PRO version 21 is used[35]. GT PRO is a simulation program from Thermoflow. The program computes all necessary heat balances and equipment designs for the given plant. The user types the input criteria and other assumptions, while GT PRO computes the heat and mass balances, system performance and component sizing[36]. In addition, GT PRO is automatically suggesting several input values. The user has the choice of utilizing these values or to make adjustments. It is possible to either optimize the economy or the net efficiency of the plant. In the current work, GT PRO is used in order to optimize the efficiency.

For off-design calculations, GT MASTER version 21 is used as simulation program. GT MASTER is the off-design version of GT PRO. There is a direct link between GT PRO and GT MASTER, and the design specifications of a power plant in GT PRO are used in GT MASTER. The simulation in GT MASTER is an iteration process.

4.1.2 Process Design and Specifications, NGCC

A reference NGCC plant is designed without CO_2 capture in order to create a starting point for the CO_2 capture study. The output values of the flue gas conditions are used as input values in the capture process. The output values from the capture plant, in addition to the reference NGCC plant, create the basis for the design of the NGCC plant with CO_2 capture. Also, the plant performance with and without capture is analyzed and the losses in the NGCC plant by capturing CO_2 are discussed. The effect of part-load operation on the power plant is compared with the design NGCC plant without capture. The basic design parameters and specifications of the NGCC plant is taken from the European guidelines for assessment of CO_2 capture technologies, published by DECARBit[37]. One of the challenges in compering simulation results between different studies is the variation in input values of the plants. With the guidelines from DECARBit, it could be easier to compare the simulations with other results.

A summary of the most important input values used in the reference NGCC plant is given in Table 4.1.

Fuel composition		
Methane	Volume %	89
Ethane	Volume %	7
Propane	Volume %	1
I-Butane	Volume %	0,05
N-Butane	Volume %	0,05
N-Pentane	Volume %	0,004
Pentene	Volume %	0,005
Hexane	Volume %	0,001
Carbon dioxide	Volume %	2
Nitrogen	Volume %	0,89
Plant design		
HPT pressure and temperature	bar/°C	125/566
IPT pressure and temperature	bar/°C	30/566
Crossover IPT/LPT pressure	bar	3,92
LPB pressure	bar	4,52
LPS exit temperature	°C	290
Dearator pressure and temperature	bar/°C	1,1/102,3
Condenser pressure and temperature	bar/°C	0,048/32,2
Pinch LPB/IPB/HPB	K	5/10/10
Isentropic efficiency HPT/IPT/LPT	%	92/94/90

Table 4.1 – Design parameters, NGCC plant.

A three pressure level steam cycle with reheat of the HPT exit flow is used. The gas turbine installed is a General Electric 9371FB. As cooling system in the condenser, a natural draft wet cooling tower is used, with the ambient conditions described in the DECARBit report[37]. The temperature increase of the cooling water is assumed to be 11K through the condenser. A temperature difference of 3K is used between the cooling water and the condensed steam/water. It is assumed no bleeds or leakages in the steam turbine.

4.1.3 Process Design and Specifications, CO_2 Capture Plant

The process is defined with the input parameters given in Table 4.2. The removal rate of CO_2 is 90%. Since the amine absorption program in Hysys uses a stage by stage model, the number of equilibrium stages in the absorber and desorber must be decided by the user instead of the height. As described in the theory, there is a close link between the number of equilibrium stages and the height of the absorber. The number of stages in the absorber is fixed at 11, while the number of stages in the desorber is 20. 50 mbar is chosen as the pressure drop in the absorber. The approach temperature in the lean/rich cross-flow heat exchanger is fixed at 5 K, while the pressure drop is 20 mbar. All of these parameters are constant throughout the simulations, and the design values are given in Table 4.2.

Absorber		
Lean solvent temperature into absorber	°C	40
CO_2 removal	%	90
Number of stages		11
Diameter	m	10
Pressure drop	mbar	50
Lean/rich cross-flow heat exchanger		
Pressure drop	mbar	20
Temperature approach, hot end	K	5
Stripper		
Number of stages		25
Diameter	m	8
Pressure drop	mbar	20
Bottom pressure	bar	2
Overhead condenser temperature	°C	30

Table 4.2 – Design parameters, CO_2 capture plant.

4.2 Part-Load Methodology

4.2.1 NGCC without CO_2 Capture

In order to operate a combined cycle at part-load, the power output of the gas turbine is regulated. There are two different control methods of the gas turbine power output; reduction in fuel exclusively and reduction in fuel and air flow simultaneously. In the present work, both of these methods are investigated. Every gas turbine manufactures have their own control strategy for the part-load operation. The default control method for the gas turbine used in this thesis is pre-programmed in GT MASTER. This control method is compared to the fuel reduction exclusively strategy, for a part-load range of 40-100% of the initial power output.

There are several combinations of fuel- and air flow reduction for a specified gas turbine power output. In GT MASTER, several control strategies could be chosen and analyzed. The impact of some of these strategies is studied at a fixed power output of 60% of the design output.

The steam cycle works as a slave for the gas turbine, and the cycle is optimized based on the available heat in the gas turbine exhaust stream. This optimization is done in GT MASTER, within the limitations from the design and the given gas turbine control method. The behavior of the steam cycle and the gas turbine are analyzed individually and as a total combined cycle.

4.2.2 CO_2 Capture Plant

When process parameters are changed from their values in the design point, a lot of things in the capture plant are affected. For instance, the mechanical work in the process is changed. As seen from the flow sheet in Figure 3.1, there are several pumps in the system. The pump work related to the flue gas cooler depends on the flue gas mass flow, and temperature. In the off-design model of the capture plant, it is assumed a constant inlet temperature. The pump work related to the rich- and lean liquid flow depends mainly on the liquid circulation mass flow. Also, the pump work connected to the cooling of the lean liquid flow upstream of the absorber depends on the liquid flow. In addition, it is pump work linked to the condensation of the reflux at the overhead condenser.

The far most energy demanding component related to the mechanical work is the flue gas blower upstream of the absorber. This blower is mainly influenced by the flue gas mass flow, but also the pressure drop over the absorber and the temperature in front of the blower.

In the design and off-design models of the capture plant used in the present work, the mechanical work is calculated. Thus, the mechanical work's dependence of parameter changes is investigated. However, for simplifications the pressure drop over the absorber is assumed constant. Therefore, the results are not exactly, but they give a good indication on the affection of changes in the capture plant conditions.

In the study of the capture plant behavior in the next sections, the main focus is connected to the energy demand in the reboiler and how this is influenced of parameter changes. The particular parameter changes related to an off-design model are listed below.

Flue gas flow

Due to the part-load operation of the power plant, the exhaust gas flow will vary. The total mass flow and the composition of the flue gas flow are highly dependent on the power plants part-load control methods. Thus, both the effect of a change in total flue gas flow and the CO_2 mol% in the flow needs to be investigated

Liquid circulation flow

The energy demand in the reboiler is connected to the L/G ratio, or the liquid circulation flow rate in the absorption process. As the flue gas flow is changed from the design point, the liquid circulation flow related to the lowest reboiler duty will differ from its flow in the design point. However, if the liquid flow rate is below a certain value there may be problems related to the wetting of the packing material in the absorber. Therefore, it may be relevant to fix the liquid flow in order to reduce the risk of incomplete humidification. The effect of a change in the liquid flow rate needs to be investigated.

Lean CO_2 concentration

The energy demand in the reboiler is sensitive to a change in the purity of the lean solution leaving the stripper column. The latent heat demand is highly dependent on the saturation pressure of the CO_2 in the liquid stream through the stripper. As the CO_2 concentration in the liquid stream at the stripper bottom is reduced, the saturation pressure decreases and, thus, the steam request in the up-flowing gas stream increases. The liquid flow rate and the lean CO_2 concentration are strongly connected. Therefore, in order to analyze the liquid flow rates influence on the energy demand in the reboiler, the lean CO_2 concentration is of importance. Also, the lowest reboiler duty for a given gas stream is determined by the lean CO_2 concentration.

4.2.3 NGCC with CO_2 Capture

When the NGCC plant with capture is investigated at part-load operation, the most important difference from the NGCC plant without capture operation is the restriction on the steam extraction pressure. The different part-load control methods that is used on the gas turbine in the study of the combined cycle behavior, and the belonging steam demand from the CO_2 capture plant, is utilized in the NGCC with capture section. In addition the mechanical work from the capture plant and the CO_2 compression work are added as lost power output in the plant.

5. Litterature study

A lot of work and research have been done in the field of combined cycles with CO_2 capture from the post-combustion gas stream. The combined cycle with natural gas as the fuel in the gas turbine, a HRSG with one or more pressure levels and a connecting steam turbine is a well-known technology. Also the post combustion CO_2 capture process with use of MEA as the absorbent is covered in past work. There are fewer studies on the integration of the capture plant into the power plant, however, the work are increasing.

Regarding off-design operation of the plant, there are some studies on the modeling of a gas turbine, however, the past work on part-load operation of a combined cycle are less. The CO_2 capture plant has lacking information on the off-design behavior in the literature. Neither the part-load behavior of an integrated CO_2 capture plant into a combined cycle has been extensively studied before. In this section, a brief overview of previous work is presented.

5.1 Part-load Behavior of a Combined Cycle

In the study [11] '*Comparative analysis on the part load performance of combined cycle plants considering design performance and power control strategy*' the connection between the part-load- and design performance of a combined cycle is analyzed. Different part-load control strategies are used. The performance analysis program, for both design and off-design calculations, used in the work is developed be the author, T.S.Kim.

For the part-load operation of a combined cycle, three cases are established:

1. Control C1: Maximum air flow control. The air flow is constant related to the design point, and the fuel is reduced in order to reduce the load.
2. Control C2: IGV control until the air flow is 70% of the air flow in the design point. The air flow and fuel is reduced simultaneously to obtain a constant TET. When the load is further reduced and the IGVs are in their minimum positions, the air flow is constant and the fuel is reduced.
3. Control C3: IGV control until the air flow is 85% of the air flow in the design point. The air and fuel is reduced simultaneously to obtain a constant TIT. When the load is further reduced and the IGVs are in their minimum positions, the air flow is constant and the fuel is reduced.

The gas turbine used is a state-of-the-art high performance gas turbine, with a design pressure ratio of 15. For the steam cycle, the design live steam temperature is 560°C. It is only used one pressure level, and the live steam pressure is 50 bar at the design point. The corresponding combined cycle efficiency at design point is 54,6%. This efficiency is lower than the state-of-the-art efficiency of a combined cycle with three pressure levels and reheat of the high pressure steam. The results from the performance calculations of the power plant at part-load are given in Figure 5.1. The three control cases defined above is plotted against the load and the total plant-, gas turbine- and steam cycle efficiencies.

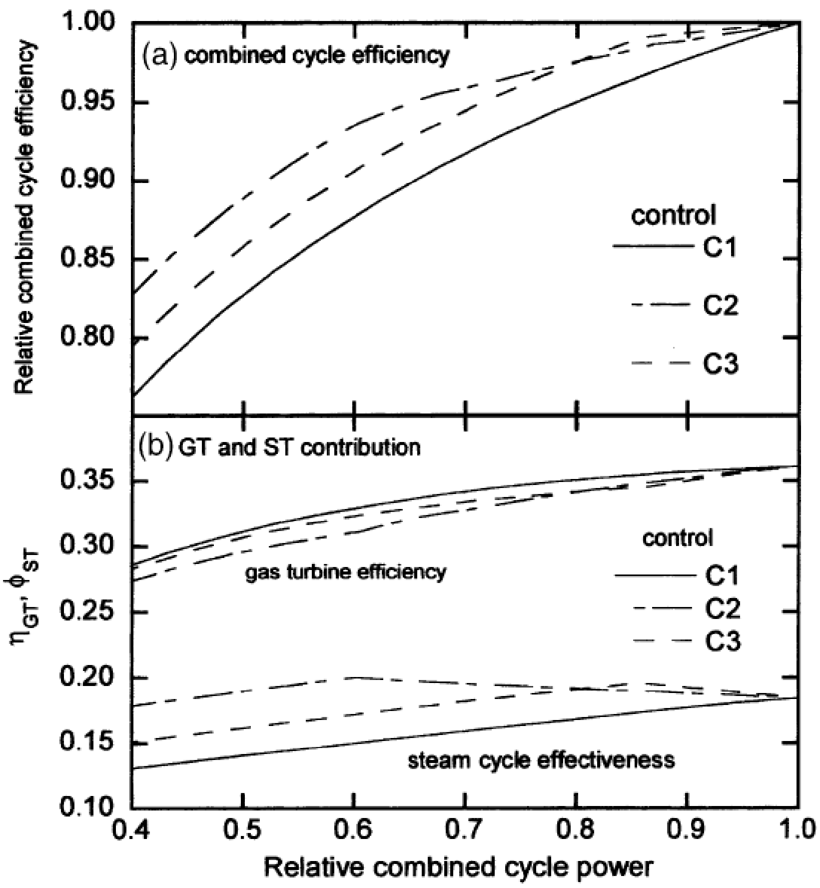

Figure 5.1 – Comparison of part-load performances between three control methods[11].

If control method C1 and C2 are compared, the control method C2 gives the highest combined cycle efficiency at part-load. The efficiency difference between the two methods increases as the load is reduced. When the load approaches 60%, the difference in the combined cycle efficiency becomes stable. At this load, the air flow could not be further reduced and the exit temperature is decreasing. The gas turbine efficiency is highest for the control strategy C1, however, the exit temperature is low and the respective steam cycle efficiency is much lower for C1 than C2. This results in a poor total efficiency for C1 at part-load operations.

Control method C3 has the highest combined cycle efficiency of the three methods for a load reduction in the range 0-20%. From that point, its combined cycle efficiency is lower than the efficiency of C2 in the remaining load range. The reason is the reduced TET. A reduced TET contributes to low steam cycle efficiency. It could be seen from Figure 5.1 that for a load below 80%, the steam cycle efficiency of the C2 control method is higher than C3.

5.2 Part-load Behavior of a CO_2 Capture Plant

The part-load operation of a CO_2 capture plant is rarely covered in the literature. However, in order to analyze the part-load operation in the capture plant, typical parameter changes that occur related to the part-load operation in the power plant could be investigated. Also, the optimization process related to the energy demand in the design absorption plant could be useful in the determination of the off-design behavior.

In the study[29] 'Modeling of CO_2 Capture by Aqueous Monoethanolamine', sensitivity analyses have been done on different process variable in order to find the operating conditions at the lowest steam requirements. The absorber and stripper were modeled with Rate Frac. Rate Frac is a rate-based model in Aspen Plus. MEA was chosen as the solvent, and the model also includes Heat Stable Salt (HSS). The wt.% of MEA was fixed at 33,5%, while the HSS loading was 0,1 mol/mol MEA. At the bottom of the stripper, the pressure was fixed at 1,7 atm and the temperature of the liquid flow entering the absorber was kept at 40 °C.

Figure 5.2 illustrates the optimization of the solvent flow rate.

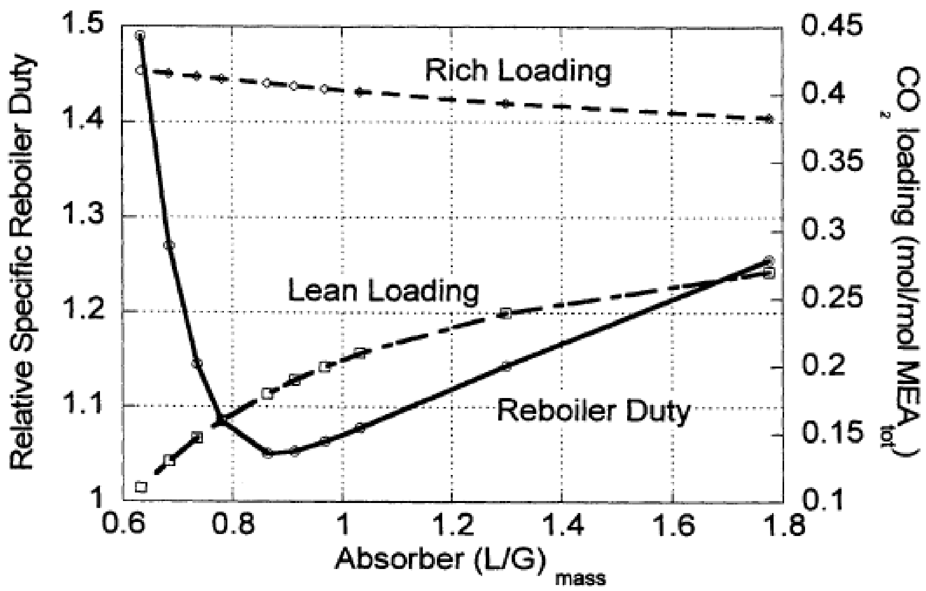

Figure 5.2 – Optimization of solvent flow rate at 85% removal and 3% mole CO_2 in flue gas[29].

In order to optimize the solvent rate, the lean loading was varied. The removal was kept constant, together with the other inputs. From Figure 5.2 it is clearly how the specific reboiler duty varies with the solvent rate or the mass based L/G ratio. At very low L/G ratio the belonging lean loading is low. This results in a high energy demand because of the large steam requirement in the stripper. As the L/G ratio increases, the SRD decreases until a minimum is obtained. When the solvent rate is further increased, the energy required to heat the solvent to reboiler temperature dominates the reboiler duty. In Figure 5.3 the SRD is divided into three energy sinks; the heat of desorption, the latent heat and the sensible heat.

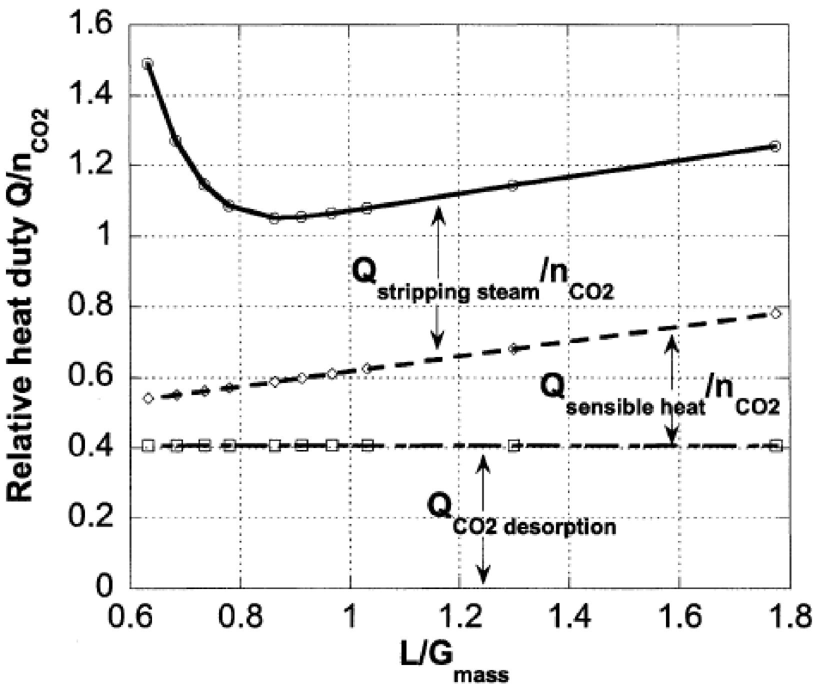

Figure 5.3 – SRD divided into three heat sinks[29].

The heat of desorption is unaffected of the L/G ratio, while the sensible heat increases linearly with the solvent rate. The reason for linear increase is because the sensible heat is directly proportional to the liquid flow, and the liquid flow is assumed constant through the stripper. On the other hand, the latent heat is highly sensible to a change in L/G ratio in the low L/G ratio range. The stripping steam increases rapidly with decreased solvent rate in that region. At high L/G ratios, the latent heat tends to an asymptote.

In the same study, also the effect of a change in absorber height, CO_2 concentration in flue gas and removal rate was investigated. Figure 5.4 shows the effect of the absorber height on the SRD for three different cases.

1. 3% mole CO_2 in flue gas, 85% removal
2. 3% mole CO_2 in flue gas, 90% removal
3. 10% mole CO_2 in flue gas, 90% removal

The absorber diameter and the stripper size were kept equal from the design case in Figure 5.2, while the L/G ratio was optimized for each case.

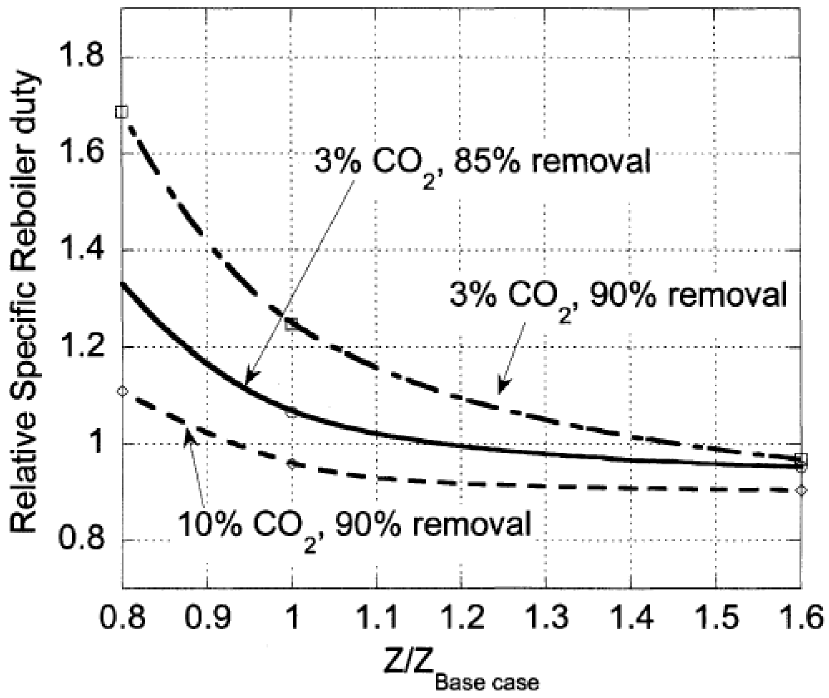

Figure 5.4 – Effect of absorber height on specific reboiler duty[29].

At a constant removal of 90%, the SRD increases as the mol% CO_2 in the flue gas decreases. The difference is more significant for low absorber heights.

5.3 Part-load Behavior of a Power Plant with Steam Extraction

In the study[38] "*On the off-design of a natural gas-fired combined cycle with CO_2 capture*", the authors have investigated the behavior of an integrated steam production for regeneration of the amines operated at off-design conditions. The integration is modeled at design load, and further studied for different part-load strategies. The off-design operation of the combined cycle is modeled by means of a gas turbine performance deck, whereas the boiler is considered by simplified correlations regarded the heat-transfer and pressure drop for off-design. Models for the flow-pressure-efficiency connection are used in the steam turbine calculations, while the HEI method is utilized in the steam condenser.

The gas turbine used in the study is the gas turbine GE PG9351A, and it is used in combination with a triple pressure steam cycle with reheat of the HPT exit steam (111/27/3,2 bar). It is assumed a condensation temperature of 135 °C in the reboiler in the capture plant. The steam is extracted from the low pressure section in the steam turbine. In the CO_2 capture plant it is assumed a constant specific reboiler duty of 4,9 MJ/kg CO_2, whereas the amine over-circulation rate is varied. The pressure drop in the absorber and desorber is corrected for changes in mass flow and velocities. Also, the compressor work related to the compression of CO_2 toward storage pressure is based on the following equation, given in kJ/kg CO_2

$$W_{CO_2} = -120 ln(p_{CO_2}) + 369 \qquad 5.1$$

Three different control strategies are used for the part-load operation in the gas turbine:

1. 'SOT decrease' strategy. This is a simple decrease in firing temperature, or Stator Outlet Temperature (SOT).
2. 'High EGT' strategy. The 'High EGT' strategy involves three different steps. First, the inlet guide vanes are regulated in combination with the fuel in order to achieve a constant SOT. The SOT is kept constant as the load is reduced until the Exhaust Gas Temperature (EGT) reaches 630 °C. The design EGT is 617 °C. Then, the IGVs are regulated in order to obtain a constant EGT and there will be a reduction in SOT. This control method continues until the IGVs have reached their minimum position, and a further reduction in load will be controlled by the fuel with the IGVs in their minimum position
3. 'Constant SHT' strategy. For this strategy, the same method as for the 'High EGT' strategy is used in the first step. However, the next step is beginning when the Steam Superheating Temperature (SHT) reaches a maximum value of 565 °C. The design SHT is 560 °C. The SHT is kept constant as the load is reduced, until the IGVs reach their minimum position. A further reduce in load will be controlled by a fuel reduction with the IGVs in their minimum position.

The 'High EGT' strategy will obtain a higher EGT and SHT in the load range of about 55-95%, while the two strategies will obtain equal temperatures for higher- and lower loads.

In order to keep the temperature in the reboiler constant, variable guide vane stagger are used on the lowest pressure level in the steam turbine. Thus, a constant pressure is achieved. The results from the study are given in Figure 5.5.

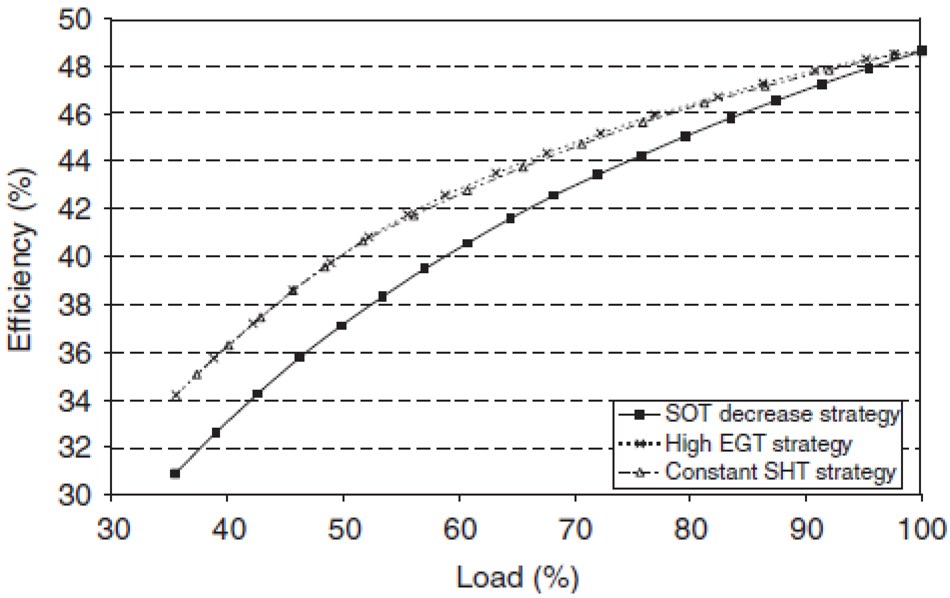

Figure 5.5 – Total plant efficiency for different part-load operation controls of a NGCC with CO_2 capture[38].

The design plant obtained a net electrical efficiency of 48,6%. As the load is reduced, the net efficiency of the 'SOT decrease' control method is falling quickly compared to the two other strategies. The difference in efficiency between the 'High EGT' and the 'Constant SHT' methods are very small, with a maximum of 0,3 percentage points. On the other hand, the difference in SOT is about 20 °C at 75% load and 30 °C at 60% load. Thus, the 'constant SHT' strategy could result in a longer life for the gas turbine blades compared to the 'High EGT' strategy with a marginal difference in net efficiency.

The study is highly relevant for the current thesis, however, changes in specific reboiler duty and operation control of the capture plant are not taken into account.

6. Design

6.1 Power Plant without CO_2 Capture

The most important results from the simulation of the reference NGCC plant are presented in Table 6.1. A net efficiency of 58% is in the medium range compared to the state of the art NGCC plants. The temperature of the flue gas leaving the HRSG is 82 °C, and is higher than the required temperature in order to avoid condensation of water in the pipes. A moisture content of 8,5% is within the limit of 16% suggested in the theory. In GT PRO the steam generation is dictated by the pinch points, and as seen from Table 6.1, all the pinches are at their limitations.

Flue gas composition		
Nitrogen	Mole %	74,2
Oxygen	Mole %	11,85
Carbon dioxide	Mole %	4,22
Water	Mole %	8,84
Argon	Mole %	0,89
Flue gas		
Temperature	°C	82,2
Pressure	Bar	1,013
Flow rate	kg/s	658,3
Steam flow rates		
LPB	kg/s	12
IPB	kg/s	13,2
HPB	kg/s	86
LPT	kg/s	111,2
HRSG temperatures		
LPB temperature, steam/flue gas	°C/°C	148,1/153,1
IPB temperature, steam/flue gas	°C/°C	242,0/252,0
HPB temperature, steam/flue gas	°C/°C	335,4/345,4
Plant performance		
Gross gas turbine output	MW_{el}	285,2
Gross steam turbine output	MW_{el}	157,2
Auxiliaries & losses	MW	6,8
Net power output	MW_{el}	435,6
Fuel LHV	MW_{th}	751
LHV net electric efficiency	%	58
Vapor fraction at LPT exit	%	91,5

Table 6.1 - Results from NGCC reference case.

6.2 CO_2 Capture Plant

6.2.1 Explanation of Typical Expressions

CO_2 mol% in flue gas

CO_2 mol% in the flue gas refers to the number of kgmole CO_2 per total amount of kgmole in the flue gas. Other terms that are used with the same meaning are CO_2 concentration and CO_2 mole fraction. The partial pressure of CO_2 in the flue gas is the product of this mol% and the total pressure of the flue gas.

L/G ratio

The L/G ratio is in the next section referred to the ratio of kgmole/s liquid and kgmole/s gas at absorber inlet.

Liquid circulation flow

The liquid circulation flow is in the current work defined as the liquid mass flow at the absorber inlet. This flow is almost equal to the liquid mass flow at the absorber exit/stripper inlet. The liquid mass flow at the stripper exit is a bit smaller, and make-up water and make-up MEA is supplied upstream of the absorber. Similar expressions used for the liquid mass flow at the absorber inlet, are liquid flow rate and liquid load. The liquid load is the liquid mass flow divided by the cross-sectional area of the absorber. Unless anything else is specified, these expressions refer to the flow at the absorber inlet.

Lean CO_2 concentration

The lean CO_2 concentration is the ratio of kgmole/s CO_2 and the total kgmole/s liquid at the stripper outlet. This value is almost equal to the CO_2 concentration in the liquid flow at the absorber inlet. However, it will differ marginally because of the make-up water and the make-up MEA.

Lean loading

Lean loading refers to the ratio of kgmole/s CO_2 and kgmole/s MEA at the stripper outlet. The lean loading is closely related to the lean CO_2 concentration. The wt.% of MEA is supposed to be approximately constant at 30% (kg/s MEA per kg/s total liquid) at the absorber inlet. Because of the make-up water and make-up MEA, the wt.% of MEA will be a little higher at the stripper outlet. However, the wt.% is almost constant at the stripper outlet. The following relationship could be derived

$$x_{CO_2} = \gamma_{CO_2} \cdot wt.\%_{MEA} \cdot \left(\frac{MW_{tot}}{MW_{MEA}}\right) \qquad 6.1$$

where x_{CO_2} is the lean CO_2 concentration, γ_{CO_2} is the lean loading, MW_{tot} is the total molecular weight of the liquid flow and MW_{MEA} is the molecular weight of the MEA. The derivation is given in Appendix. For low concentrations of CO_2, MW_{tot} could be assumed constant. Also, MW_{MEA} is constant. If the wt.% of MEA is fixed, the lean concentration of CO_2 is directly proportional to the lean loading. However, the wt.% MEA will differ marginally in the simulations.

Rich CO_2 concentration

Rich CO_2 concentration is defined as the ratio of kgmole/s CO_2 and the total kgmole/s liquid at the absorber outlet/stripper inlet.

Rich loading

The rich loading refers to the ratio of kgmole/s CO_2 and kgmole/s MEA at the absorber outlet/stripper inlet. There is a similar relation between the rich CO_2 concentration and rich loading, as for the lean CO_2 concentration and the lean loading in equation 6.1.

Specific reboiler duty

See description in chapter 3.7.

Total energy demand in reboiler

The total energy demand in the reboiler is the product of the SRD and the total mass flow CO_2 captured.

6.2.2 Optimal Design, CO_2 Capture Plant

The parameters given in Table 6.1 are used as input in the absorption process. As the theory has described, there is a relation between the liquid- and gas flow parameters in the absorber.

$$\frac{L'_M}{G'_M} = \frac{(Y_1 - Y_2)}{(X_1 - X_2)} \tag{3.10}$$

Even though the L/G ratio in the equation above is the solute free molar L/G ratio, the total molar L/G ratio will not differ much from this relation. The L/G ratio is in the current work defined as the number of moles in liquid stream into absorber divided by the number of moles in gas stream into absorber. In the design of the capture plant, the gas flow conditions are given. The mole fraction of CO_2 in the gas stream at the outlet of the absorber is 10% of the CO_2 mol% in the inlet flue gas. However, the CO_2 concentration in the liquid flow at the inlet and outlet of the absorber depends on the design of the absorber. This could be seen from an operation-/equilibrium line diagram.

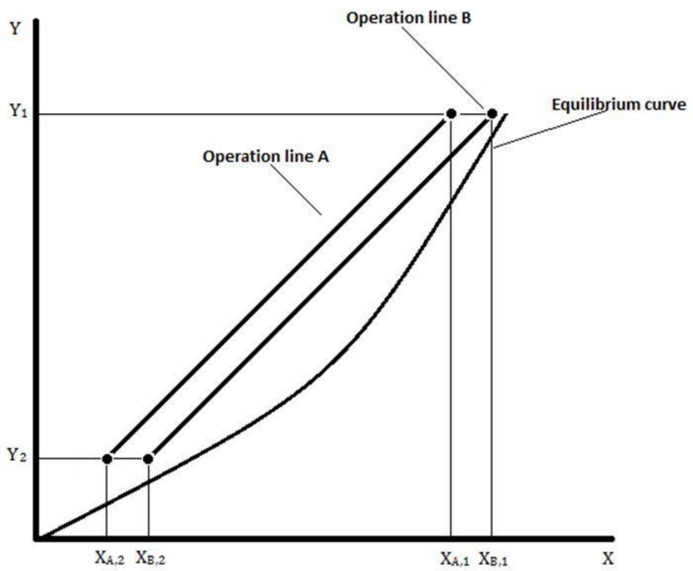

Figure 6.1 - Operation-/equilibrium line diagram.

Figure 6.1 illustrates two fictional absorber designs with their respective operation lines, A and B. As stated in the theory, the L/G ratio is the slope of the operation line. A real operation line will not be a straight line, however, for a simplified illustration it is assumed linear. The gas flow conditions, the removal rate and the L/G ratio are fixed. In addition, the equilibrium line depends mostly on the temperature and mol% of the CO_2 in the gas stream. The temperature is assumed constant in the absorbers, thus the equilibrium curve is approximately equal.

The difference between operation line A and B is the gap between their operation line and equilibrium line. A small difference between the operation line and the equilibrium line represents a low driving force for CO_2 mass transfer in the streams. In addition, a high liquid CO_2 concentration is equal to a high CO_2 loading. From Figure 3.8, a high CO_2 loading results in a low overall mass transfer coefficient. Thus, the absorption in design B requires a longer contact period than design A in order to remove the same amount of CO_2. A longer contact period means a higher absorber. This could be seen from equation 3.9. Since Hysys does not use the height, more equilibrium stages are needed in the simulation of design B.

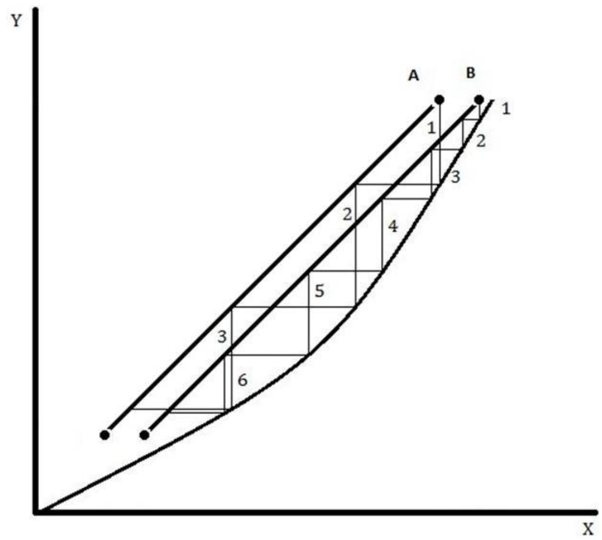

Figure 6.2 - Operation-/equilibrium line diagram. Different equilibrium stages.

Figure 6.2 demonstrates the required number of equilibrium stages for each operation line. The energy demand in the stripper related to design A will be higher than the belonging energy demand for design B. As the L/G ratio is equal and the heat of absorption is about constant for the given temperature, the difference in reboiler duty is related to the latent heat. From the theory, the latent heat is the heat required in order to generate steam in the reboiler. As both the lean- and rich CO_2 concentration in design A is lower than B, the saturation pressure in the stripper is low. The belonging partial pressure of the CO_2 in the gas stream in the stripper has to be lower than the saturation pressure, and more steam is required to reduce the CO_2 partial pressure. Therefore the latent heat demand in design A is higher than B.

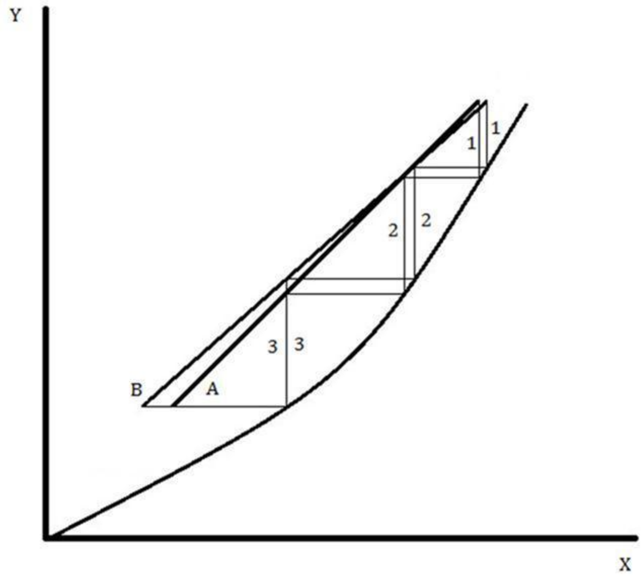

Figure 6.3 - Operation-/equilibrium line diagram. Equal equilibrium stages.

Figure 6.3 illustrates two operation conditions for an absorber with the same number equilibrium stages and equal gas flow conditions. As the L/G ratio decreases in process B, the rich concentration increases and the lean concentration decreases. Whether the reboiler duty benefits on this change depends on the magnitude of the lean concentration. The sensible heat decreases when the L/G ratio is reduced, but the latent heat demand increases. At high lean concentrations the sensible heat dominates, but as the lean concentration decreases the required stripping steam is more pronounced.

When the capture plant is designed the optimal L/G ratio, with its respective liquid CO_2 concentrations, must be calculated. In order to find this optimal design, the simulation program Hysys has been used with the inputs given in Table 4.2. The L/G ratio has been varied with an adjuster, while the lean CO_2 concentration has been regulated manually for each L/G ratio in order to attain 90% removal efficiency. In a practical approach the lean concentration in the capture plant cannot be regulated manually, but is given by the heat transfer in the reboiler and the L/G ratio. However, by doing the regulation backwards the results should be theoretically correct.

Figure 6.3 illustrates the simulation process in Hysys. As the equilibrium stages are equal, the operation line is changed in accordance to the figure. For each operation line, the energy demand in the reboiler is calculated.

The other process parameters are kept constant for every step, and the results are illustrated in Figure 6.4.

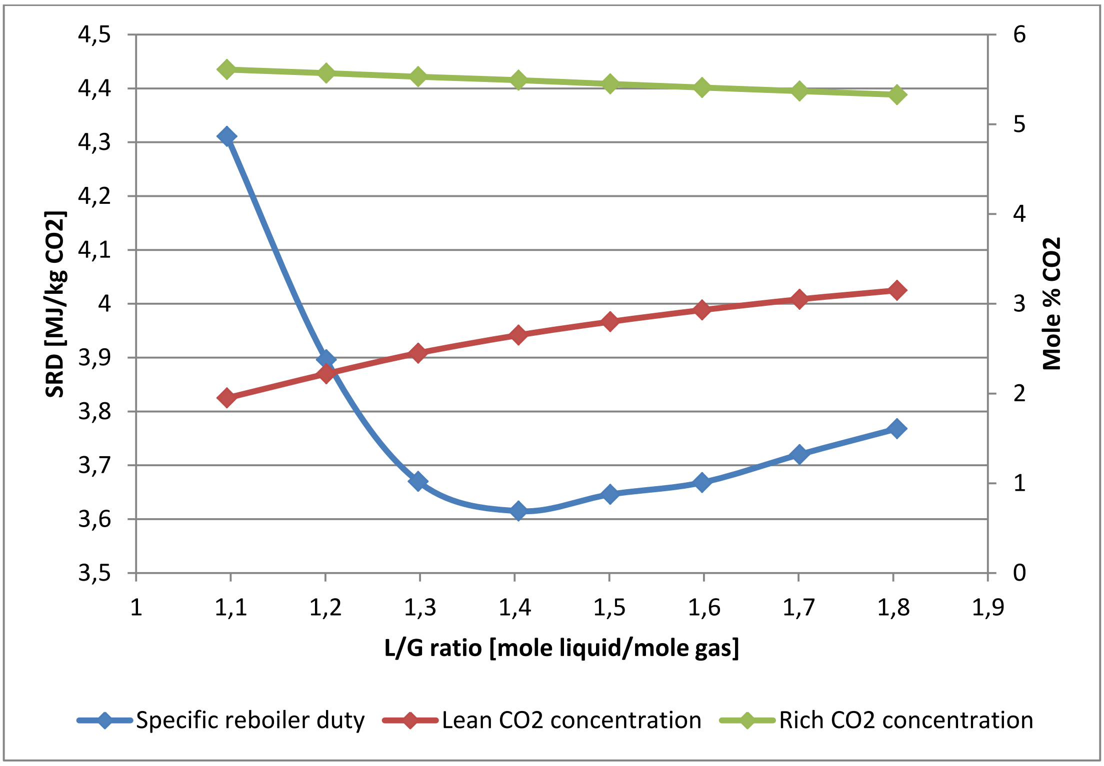

Figure 6.4 - Specific reboiler duty versus L/G ratio and lean/rich CO_2 concentration.

The results clearly indicate the dependence of the liquid/gas ratio on the energy demand in the reboiler. The results correspond to the previous theory. The optimal L/G ratio with respect to the lowest SRD is about 1,4. As the liquid flow is increased further, the reboiler duty increases in an almost constant rate. This increase is due to the sensible heat demand. As described in the theory, the sensible heat demand is proportional to the liquid mass flow into the stripper. In addition, the steam demand is getting negligible at high lean concentrations. The heat of absorption is approximately constant. Thus, the heat demand in the reboiler is increasing nearly constant at high L/G ratios.

When the L/G ratio is decreased and approaching its optimal value, related to the SRD, the latent heat demand is getting more dominating. The lean concentration is low, and thus, the saturated CO_2 pressure in the liquid flow in the bottom of the stripper is low. Steam needs to be generated to reduce the CO_2 partial pressure in the gas stream in the stripper. As the L/G ratio is further reduced, the lean concentration is decreasing even more rapidly and the steam demand is increasing quickly. The sensible heat demand is small compared to the latent heat, and the reboiler duty is increasing at the same rate as the latent heat demand. It could also be seen how the rich CO_2 concentration is increasing slightly when the L/G ratio is reduced.

An interesting observation is the relative small change in reboiler duty for high L/G ratios. Figure 6.5 indicates less than 5% increase of the reboiler duty when the L/G ratio increases from its optimum at 1,4 to a L/G ratio of 1,8. This observation is relevant and may influence the part-load regulation of the plant. As stated in the theory, there might be problems in terms of adequate humidification of the packing material when the liquid flow is low.

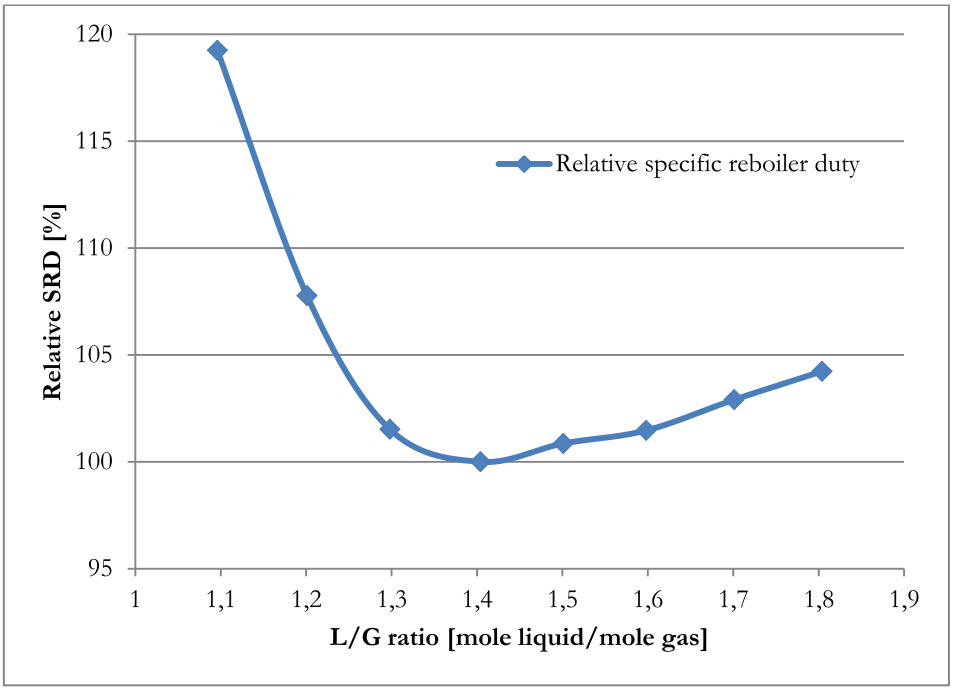

Figure 6.5 - Relative specific reboiler duty related to the lowest reboiler duty.

The corresponding operation lines for a L/G ratio of 1,2, 1,4 and 1,6 are illustrated in Figure 6.6, and it confirms the trend in Figure 6.3. The change in operation line for constant equilibrium stages is illustrated.

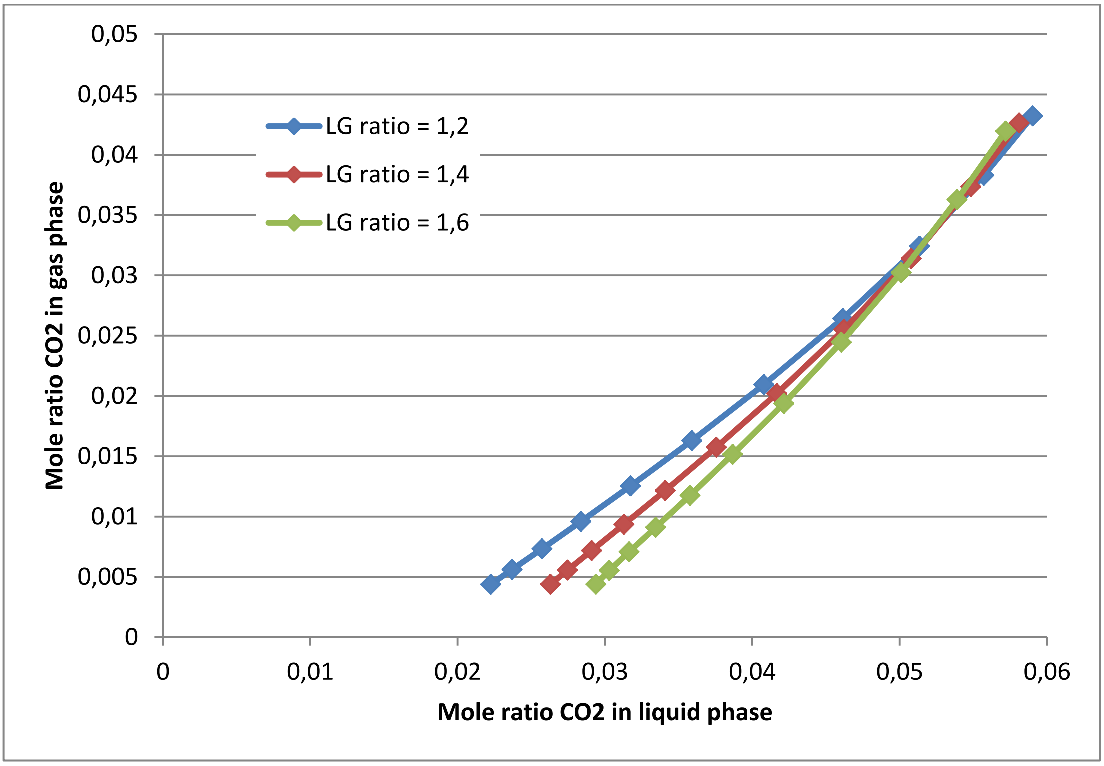

Figure 6.6 - Real operation lines for the process. The mole ratios are described in chapter 3.5.

As the removal rate is fixed in the simulation, the total energy demand in the reboiler is proportional to the specific reboiler duty graph. At part-load, the CO_2 flow will vary, and the total energy demand must be evaluated individually. The optimal absorber conditions have been located, and the design parameters are displayed in Table 6.2. The CO_2 compression work is calculated with the expression

$$W_{comp} = 0{,}33 \frac{MJ}{kg\ CO_2} \tag{3.4}$$

Liquid flow		
Lean CO_2 concentration	mol%	2,65
Lean loading	kgmol CO_2/kgmol MEA	0,219
Lean solvent temperature out of desorber	°C	121,4
Rich CO_2 concentration	mol%	5,49
Rich loading	kgmol CO_2/kgmol MEA	0,468
Liquid/gas ratio	kgmol liquid/kgmol gas	1,4
Liquid flow rate	kg/s	735
Performance		
CO_2 removal	%	90
Specific reboiler duty	MJ/kg CO_2	3,62
Total energy demand in reboiler	MW	141,9
Mechanical energy	MW	7,8
CO_2 compression work	MW	12,9

Table 6.2 - Basic design data for the CO_2 capture plant.

A flow sheet from Hysys is given in Appendix.

6.3 Power Plant with CO_2 Capture

The reboiler duty related to the capture process is high. By integrating the heat demand into the power plant, a lot of energy could be saved. The steam in the steam cycle could be used for other purposes than generating power in the steam turbine. Because of the high heat of evaporation for water, steam is superb as a heating source. For that reason it is preferable to use the steam in the steam cycle to deliver the heat demand to the reboiler. There are several options for integration. In the current work it is focused on steam extraction from the crossover between the IPT/LPT.

6.3.1 Extraction from Crossover

In a three pressure level steam cycle, there is a crossover pipe that connects the IPT and the LPT. The steam expanded in the IPT flows through the crossover, is mixed with steam produced in the low pressure cycle and expands further in the LPT. At this crossover it is possible to extract steam. The steam is extracted upstream of the low pressure steam mixing, see Figure 6.7.

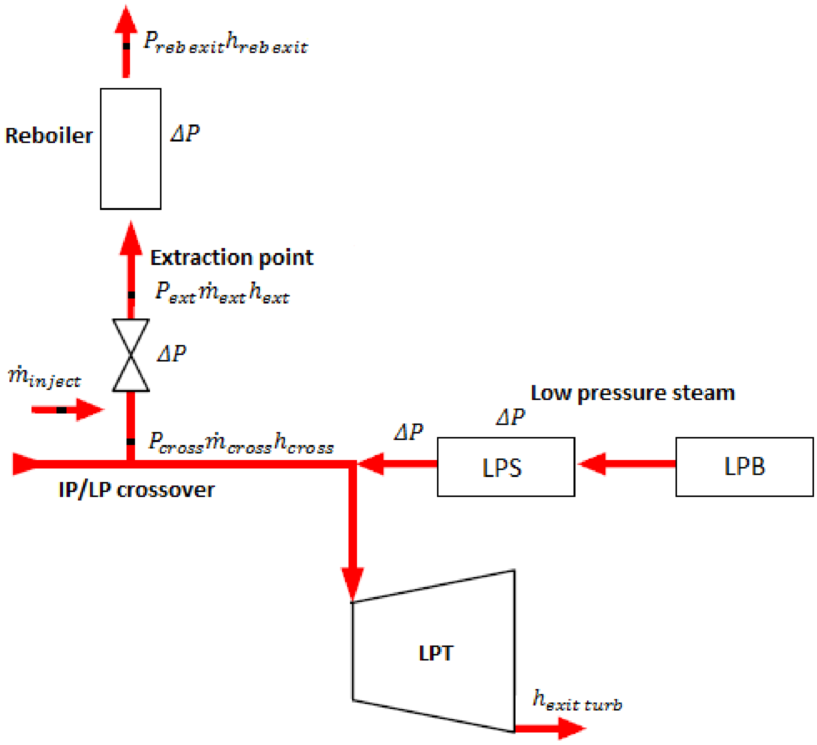

Figure 6.7 – Illustration of steam extraction from IPT/LPT crossover.

The steam at the crossover is superheated, and in the current work, additional water from the deaerator is injected in order to saturate the steam. In the piping between the crossover and the reboiler, pressure loss occurs. There will also be heat losses in the piping, however, they are neglected in the present study. The extraction point defined in this report is just in front of the reboiler, with saturated steam conditions, see Figure 6.7

When extracting steam from a crossover, it is possible to design the crossover to exactly match the preferred extraction pressure.

6.3.2 Preferred Extraction Pressure

The heat demand is significant in the reboiler in a CO_2 capture process. If steam from the IPT/LPT crossover is extracted and saturated with water, the necessary steam mass flow in the extraction point is given by equation 6.2

$$\dot{m}_{ext} = \frac{\dot{Q}_{reb}}{(h_{ext} - h_{reb\ exit})} \qquad 6.2$$

and the lost turbine work, without mechanical/generator losses

$$\dot{W}_{lost} = \dot{m}_{cross}(h_{cross} - h_{ST\ outlet}) \qquad 6.3$$

The definition of the α-value is stated in equation 2.7

$$\alpha = \frac{\dot{Q}_{reb}}{\dot{W}_{lost}} = \frac{\dot{Q}_{reb}}{\dot{m}_{cross}(h_{cross} - h_{ST\ outlet})} \qquad 2.7$$

In Figure 6.8, the α-value is plotted against the extraction pressure for a constant reboiler duty.

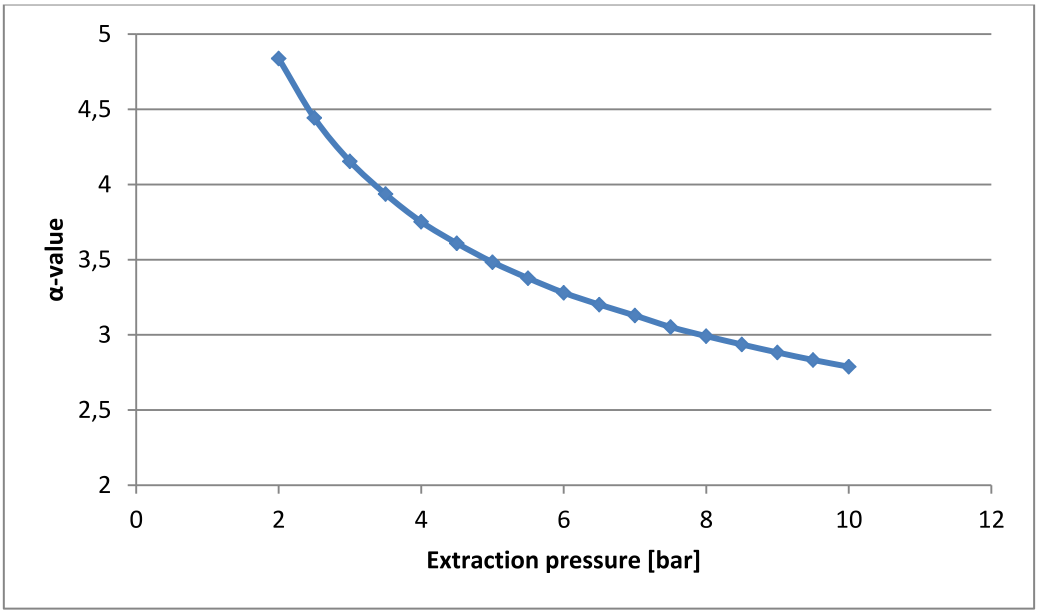

Figure 6.8 - Variation of α-value against extraction pressure. The heat demand in the reboiler is 142 MW.

As seen from the graph; in case of low extraction pressure the α-value is high, and thus, the power generation penalty of extracting steam from the turbine is small. The steam is therefore preferably extracted at lowest pressures possible within the conditions for the simulation.

It is important to distinguish between α-value and total plant efficiency. The α-value is only taken into account the lost turbine power output between a plant with steam extraction and a plant without steam extraction, at the same design conditions. It does not say anything about how a different extraction pressure, and thereby different crossover pressure, affects the steam generation in the HRSG. Therefore, it is not possible to compare the total efficiency of a power plant between two different steam extraction pressure designs with only regard to the α-value. However, the α-value gives a significant guiding principle for steam extraction designs.

The CO_2 capture processes described in chapter 6.2.2 is integrated into the power plant by steam extraction from the IPT/LPT crossover. The steam is preferable extracted at its lowest possible pressure, restricted by the temperature limit of the solvent. The regeneration temperature of the absorbent MEA is about 120°C. A minimum temperature difference of 10 K is required in the reboiler. It is assumed saturated steam/vapor at the inlet and saturated liquid at the outlet of the reboiler. The pressure loss through the reboiler is fixed at 0,5 bar. The minimum extraction pressure is:

$$P_{ext} = P_{sat}(130°C) + 0{,}5 = 2{,}7 + 0{,}5 = 3{,}2\ bar \qquad 6.4$$

2,7 bar is an absolute minimum pressure at the reboiler exit. Included a safety margin, 3 bar is chosen as the reboiler exit pressure. The extraction pressure is thus 3,5 bar. The corresponding crossover and Low-Pressure Boiler (LPB) pressures, regarding pressure losses in the piping and heat exchangers, becomes

$$P_{cross} = 1{,}12 * 3{,}5 = 3{,}92\ bar \qquad 6.5$$

and

$$P_{LPB} = 1{,}03 * 1{,}12 * 3{,}92 = 4{,}52\ bar \qquad 6.6$$

It should be observed that the crossover pressure and the LPB pressure are equal to the design NGCC plant without steam extraction. The reboiler heat demands are given from Table 6.2. For the CO_2 capture base case, the heat demand is 141,9 MW. The requested mass flow in the extraction point is given from equation 6.2. The enthalpy difference through the reboiler is calculated based on the saturated steam/liquid conditions at the given pressures determined.

$$h_{ext} - h_{reb\ exit} = 2732 - 561{,}4 = 2170{,}6\ kJ/kg \qquad 6.7$$

The corresponding mass flow at the extraction point in the base case becomes

$$\dot{m}_{ext} = \frac{141900}{2170{,}6} = 65{,}37\ kg/s \qquad 6.2$$

6.3.3 Efficiency Penalty

The efficiency of the power plant including the capture process can be calculated using equation 6.8[3]. This equation includes all the losses that consist with capturing CO_2.

$$\eta_{Post-Comb} = (1)\eta_{no\ capture} - (2)\frac{E^{CO_2}_{rem,mech}C}{LHV} - (3)\frac{E^{CO_2}_{rem,heat}\alpha_{str,atm}Cf}{LHV} - (4)\frac{E^{CO_2}_{comp}Cf}{LHV} - (5)\frac{\Delta E_{aux}}{LHV} \qquad 6.8$$

1) Efficiency of a standard plant with no capture.
2) Efficiency penalty by use of mechanical work or electricity in the CO_2 absorption process. The work includes fans and pumps. The C gives the ratio between formed CO_2 and consumed fuel.
3) It is the largest penalty and it consists of the loss by extracting steam from the steam cycle. f is the percentage of CO_2 captured in the absorption process. α express the loss of steam turbine power output compared to the potential heat in the steam.
4) This penalty is the energy required to compress and condense the CO_2 after it has been captured.
5) The α value is only taking into account the loss of the steam turbines power output and neglects the change in auxiliaries and losses in the cycle. This term represents the difference in auxiliary work in case of steam extraction.

6.3.4 Optimal Design

A summary of the main results from the simulation are presented in Table 6.3. The water at the reboiler exit is pumped up to a pressure of 4 bar and a temperature of 130°C, before it is returned to the feed water tank in the steam cycle.

Steam/water flow rates		
LPB	kg/s	12
IPB	kg/s	13
HPB	kg/s	86
LPT	kg/s	53
Extraction from crossover	kg/s	58
Water injection from deaerator	kg/s	7
Flue gas		
Flue gas temperature	°C	118
Plant performance		
Gross gas turbine output	MW_{el}	285,2
Gross steam turbine output	MW_{el}	121,1
Auxiliaries & losses	MW	5,7
Net power output	MW_{el}	400,7
Fuel LHV	MW_{th}	751
LHV net electric efficiency	%	53,4
Efficiency loss compared to no capture plant	%-point	4,6
Net plant efficiency, included mech/comp work	%	50,6

Table 6.3 - Results from design case, NGCC with CO_2 capture.

The efficiency loss in the power plant is 4,6 percent-points disregarded the CO_2 compression work and the mechanical energy in the capture plant, while the total efficiency loss is 7,4 percent-points, compared to the reference NGCC without CO_2 capture. The NGCC flow sheets from GT PRO are given in Appendix.

7. Part-load Results

The flue gas conditions from the power plant are the input values in the capture process. Thus, the part-load operation of the NGCC plant is initially examined. To simplify the work, the part-load behavior of the reference NGCC plant without capture is explored. The flue gas composition is not affected of the steam extraction, thus, it is correct to use the results from the NGCC plant without capture. Accordingly, the capture plant behavior for different flue gas conditions is considered. The design capture plant is used as a reference, and the physical parameters should remain constant. However, Hysys is mainly applicable for the design of a plant, not for off-design evaluations. Most of the physical parameters are kept constant, but the height is difficult to control. The number of equilibrium stages is fixed and thus the height will not depart that much. Nevertheless, the HEPT may vary between the simulations and actual values from this study should be used with caution. The study, however, illustrates a lot of tendencies for a capture plants part-load behavior.

The results from the two processes are collected and used in the part-load simulations of the NGCC plant with CO_2 capture.

7.1 Power Plant without CO_2 Capture

The study of the NGCC part-load behavior is split into two parts. First, the default part-load control method for the particular gas turbine in GT MASTER is examined against a part-load control with constant air flow. The load is in the range 100-40%.

The next part explores the effect of different IGV control methods on the power plant for a specified gas turbine power output.

7.1.1 Default Part-load Method against Constant Air Flow

The reference NGCC plant designed in chapter 6.1 is used as the design power plant in the current section. The design parameters are given in Table 4.1 and Table 6.1.

IGVs in the compressor are used in order to reduce the air flow into the turbine. The gas turbine operates as choked, thus the turbine inlet pressure is highly dependent of the air flow. The exit temperature depends on the TIT and the pressure ratio. At part load, the exit pressure is reduced due to a lower pressure drop in the HRSG. Also the inlet pressure is reduced, because of choked conditions.

In the simulations below it is assumed a sliding-pressure mode unless anything else is stated. The live steam temperature has a maximum value of 566 °C, by means of a desuperheater upstream of the turbine. The ambient conditions are kept constant.

The different gas turbine producers have their own part-load control procedure[3]. In the next simulation, the pre-programmed part-load operation control of the current gas turbine is used. This is the gas turbine manufactures own control of the IGV regulation and the fuel supply reduction in order to obtain a target exhaust gas temperature at the given gas turbine power output.

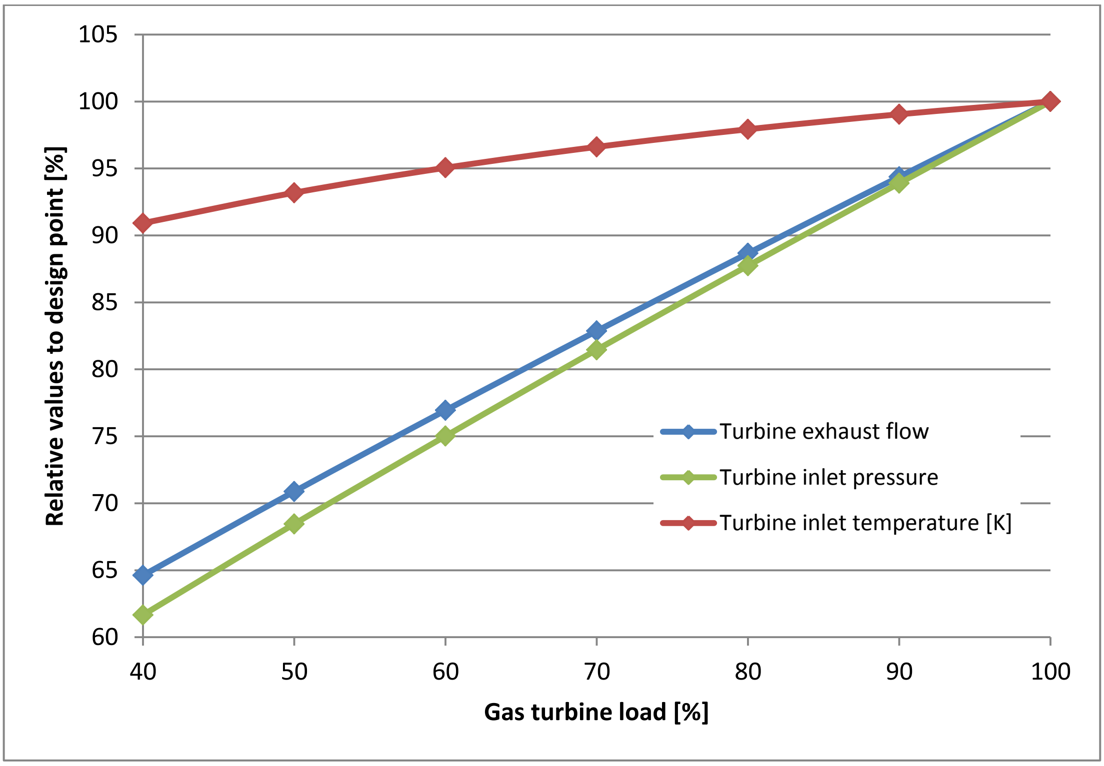

Figure 7.1 – The relation between the turbine exhaust flow, inlet pressure and inlet temperature as function of the gas turbine load. The relative inlet temperature is given in Kelvin/Kelvin. The default part-load control is used.

Figure 7.1 above illustrates the relation between the turbine exhaust mass flow, the TIT and the TIP. As stated in the theory, the TIP is proportional to the inlet mass flow and the square root of the turbine inlet temperature. The relative change in inlet turbine mass flow could be assumed equal to the relative change in exhaust mass. The trend in the figure is in accordance to the relation. The relative reduction in TIP is slightly more rapid than the relative reduction in the exhaust flow. This is due to the reduced TIT.

Also, the part-load operation control of the power plant with no regulation of the IGVs has been simulated. The air flow is kept constant, and only the fuel supply is reduced in order to approach the reduced power output. In Figure 7.2, the default part-load control is compared to a part-load control with no regulation of the IGVs.

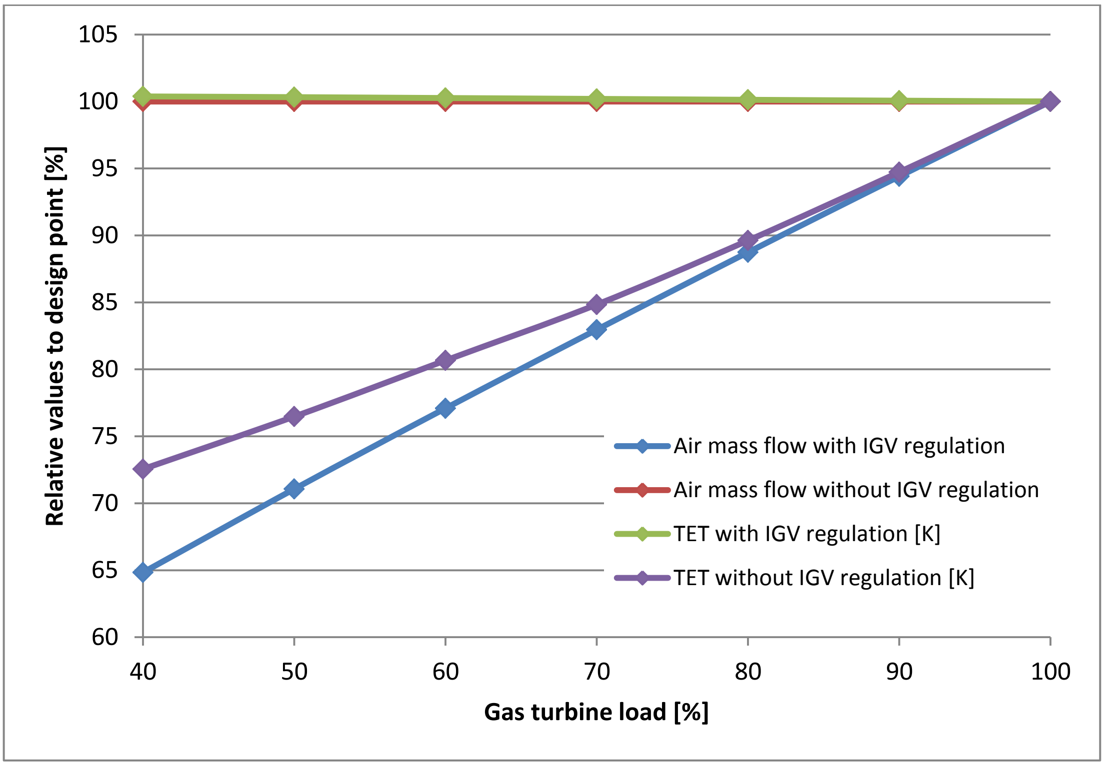

Figure 7.2 – Different parameters for two different part-load control methods. The temperature ratio is given in Kelvin/Kelvin.

In Figure 7.2, the relative TET and air mass flow is plotted against the load reduction for two different part-load operation methods. The exhaust gas temperature is given by equation [6]

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{R\eta_p}{C_p}} \qquad 7.1$$

where η_p is the polytrophic efficiency, and index 3 and 4 refers to outlet and inlet of the turbine respectively. If the efficiency is assumed constant, the exhaust gas temperature depends on the inlet temperature and the pressure ratio. It could be seen from Figure 7.2 that the part-load control strategy from the gas turbine manufacture, is to keep an almost constant exhaust gas temperature. Actually, the exhaust gas temperature is increasing slightly for reduced load in the load range simulated. This is possible because of the large reduction in air flow. The relative air flow reduction is 35% at 40% load. The limiting value for air flow reduction by use of IGVs varies between the types of compressors. As stated in the theory, it is possible for some compressors to reduce the air flow to about 60-70% of its design flow. The reduction illustrated in the Figure 7.2 is at this limit.

When the position of the IGV is constant, the air flow into the turbine is constant throughout the part-load operation. As Figure 7.2 illustrates, the respective exhaust gas temperature decreases rapidly.

The effect of the two different part-load operations on the plant efficiency is indicated in Figure 7.3

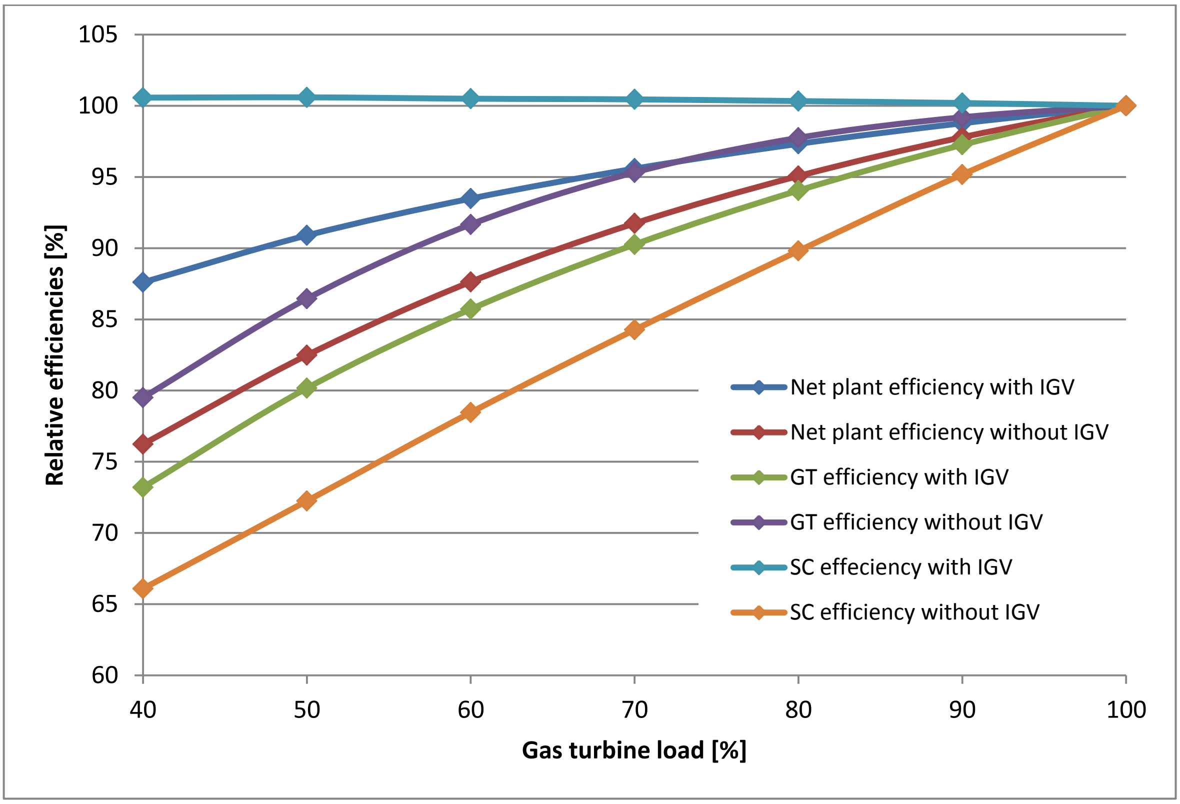

Figure 7.3 – Relative efficiencies for two different part-load control strategies.

Figure 7.3 indicates greater net plant efficiency for the part-load operation with reduced air flow. The steam cycle efficiency is the main difference. As the air flow is reduced and the exhaust temperature is kept about constant, the steam cycle efficiency is approximately constant. When there is no reduction in the air flow, the SC efficiency decreases rapidly for reduced load.

It is a higher drop in GT efficiency for the default IGV control method compared to the constant air flow strategy. However, the high loss in SC efficiency is dominating for the total plant efficiency, for the constant air flow method. As described in the theory, the net plant efficiency is a function of those two efficiencies and the auxiliary loss.

$$\eta_{NGCC} = \eta_{GT} + \eta_{SC}(1 - \eta_{GT}) - \frac{\dot{W}_{Aux}}{\dot{Q}_{Fuel}} \qquad 2.13$$

In order to describe the behavior of the steam cycle efficiency, the efficiency could be related to heat recovery efficiency and steam turbine efficiency.

$$\eta_{SC} = \eta_{HRSG}\eta_{ST} \qquad 2.24$$

The efficiency variations could be seen in Figure 7.4.

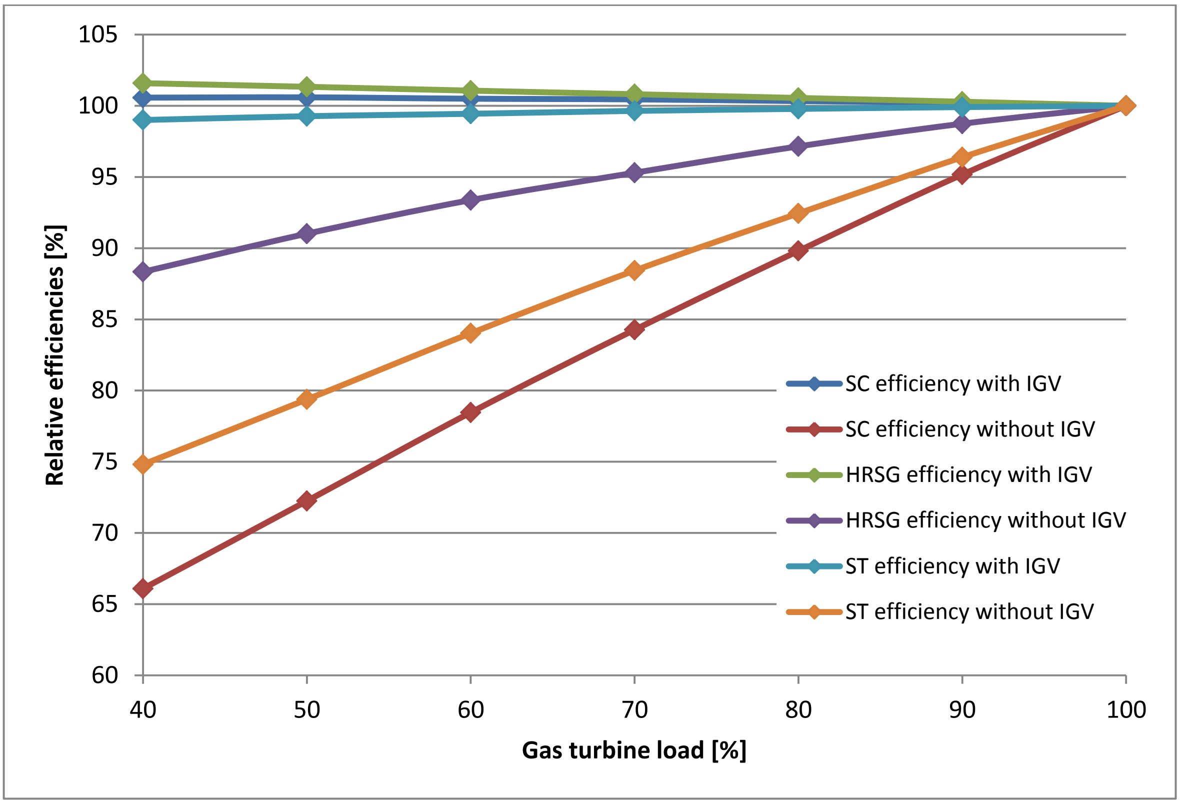

Figure 7.4 - Relative efficiencies for two different part-load control strategies.

The HRSG efficiency is increasing marginally for reduced load, when the default IGV control method is used. This efficiency is high when most of the exhaust gas heat from the gas turbine is utilized. As the flue gas mass flow is reduced, the pinch temperatures in the heat exchangers will decrease according to equation 2.27 and 2.28. A reduced pinch is an advantage for the heat utilization. The pressure levels will decrease. Also, the condenser pressure will decrease. When the condenser pressure is reduced, the temperature of the feed water decreases, and more heat may be utilized in the HRSG.

The live steam temperature is constant because of the high TET, thus the change in ST efficiency will not be significant. From Figure 7.4, the ST efficiency is reduced very slightly.

If the air flow is kept constant, the pressure levels in the steam cycle are low. This will result in a low steam mass flow, according to equation 2.26. As both the pressure levels and the steam generation are low, the heat transfer in the HRSG is small.

The ST efficiency is decreasing rapidly for reduced load, when the inlet air flow is kept constant from the design point. From Figure 7.4, this is the main reason for the low SC efficiency. As the exhaust gas temperature is reduced, the LST and the reheat temperature are decreasing. The steam turbine efficiency is very dependent on the LST. Also, when the LST and the reheat temperature are reduced the vapor fraction at the steam turbine exit will decrease. This results in a lower overall isentropic efficiency in the low-pressure turbine.

Due to the significant loss in the net plant efficiency, the part-load operation without regulation of the IGVs is rejected, and not further inspected.

Since the flue gas is threated in the capture plant, the variations in the flue gas conditions at part-load operation are of importance. Only the default part-load operation, is inspected.

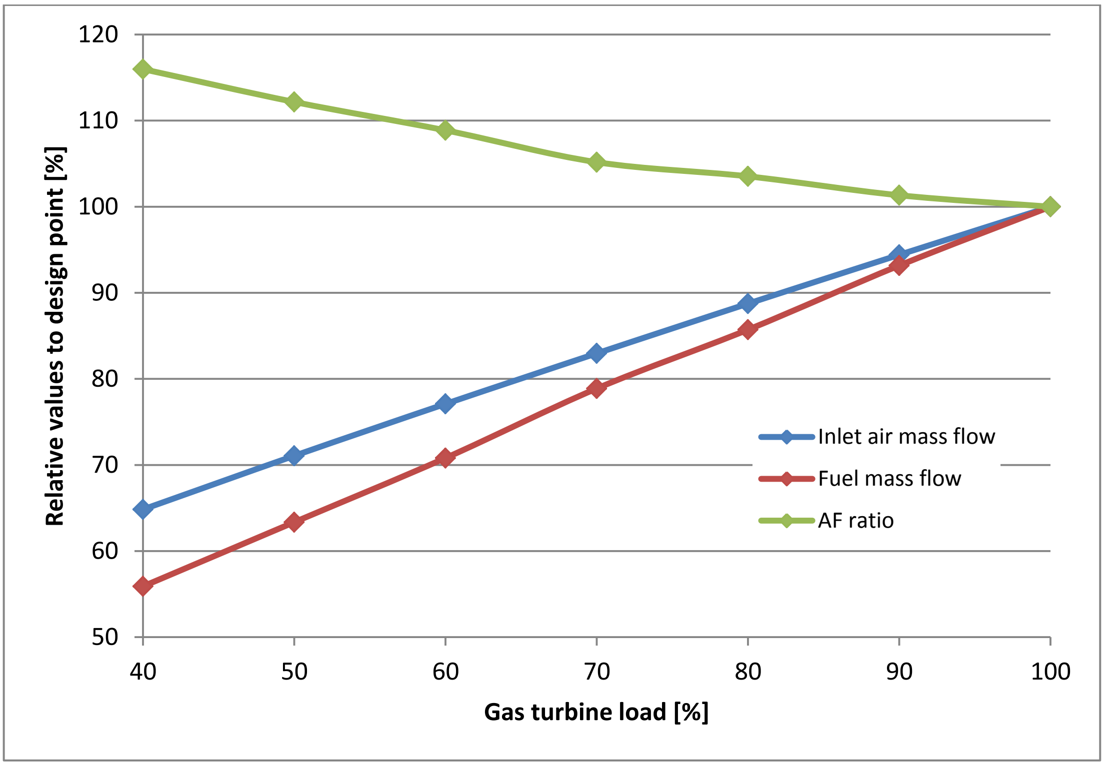

Figure 7.5 – Changes in flow compositions against the gas turbine load. The part-load control method used is the default strategy in GT MASTER.

From Figure 7.5 it could be seen that the relative decrease in the fuel supply is greater than the relative decrease in air flow for reduced load. Thus, the air-fuel ratio increases. An increased AF results in a reduced mole fraction of CO_2 in the flue gas. This could be seen in Figure 7.6.

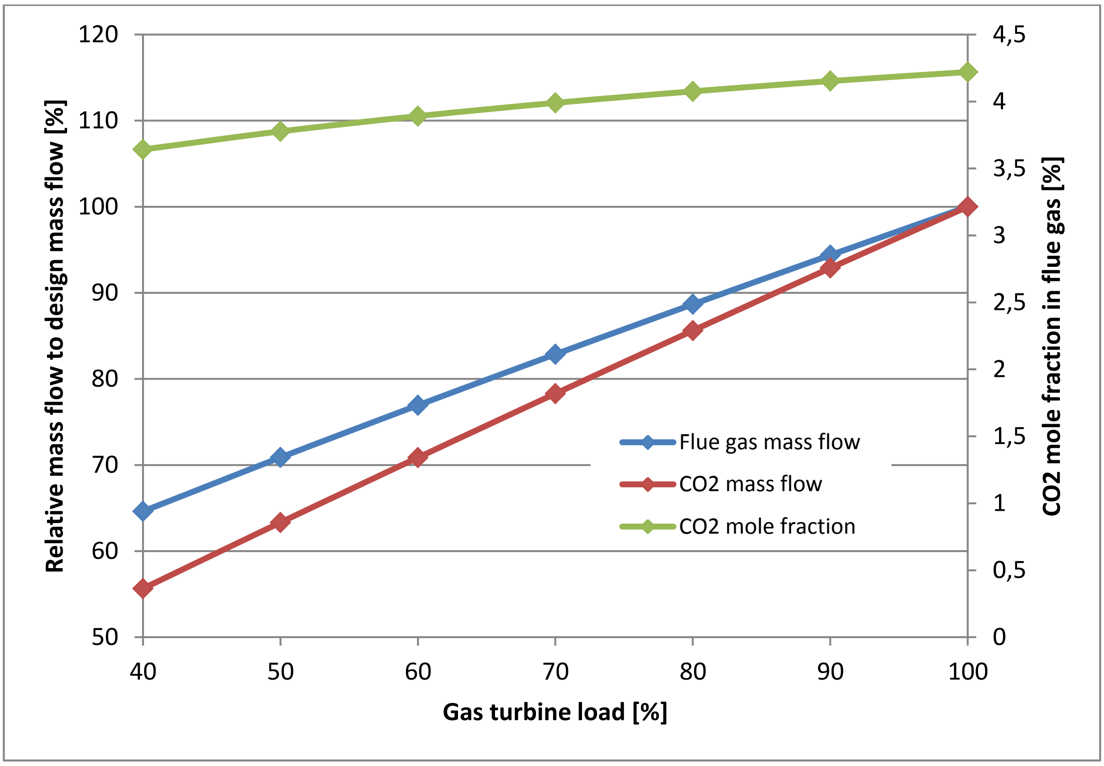

Figure 7.6 - Changes in flue gas compositions against the gas turbine load. The part-load control method used is the default strategy in GT MASTER.

The mole fraction is reduced from about 4,2% to almost 3,6% at 40% load. As the mole fraction of the CO_2 is reduced, the exhaust CO_2 mass flow is reduced relative more than the total flue gas flow. A lower CO_2 mole fraction in the flue gas means a lower partial pressure. As the partial pressure is reduced, the theoretical separation work increases, according to equation 3.4. At the same time, the total CO_2 flow is lower and less CO_2 needs to be captured in order to obtain 90% removal rate.

A further reduction in load for the pre-programmed part-load mode and a comparison with a different gas turbine is done in order to verify the reliability of the results. A gas turbine from Siemens is chosen; Siemens SGT5-4000F. This gas turbine is also recommended in the DECARBit report[37]. The load was simulated in the range 100-20%, and the results are indicated in Figure 7.7.

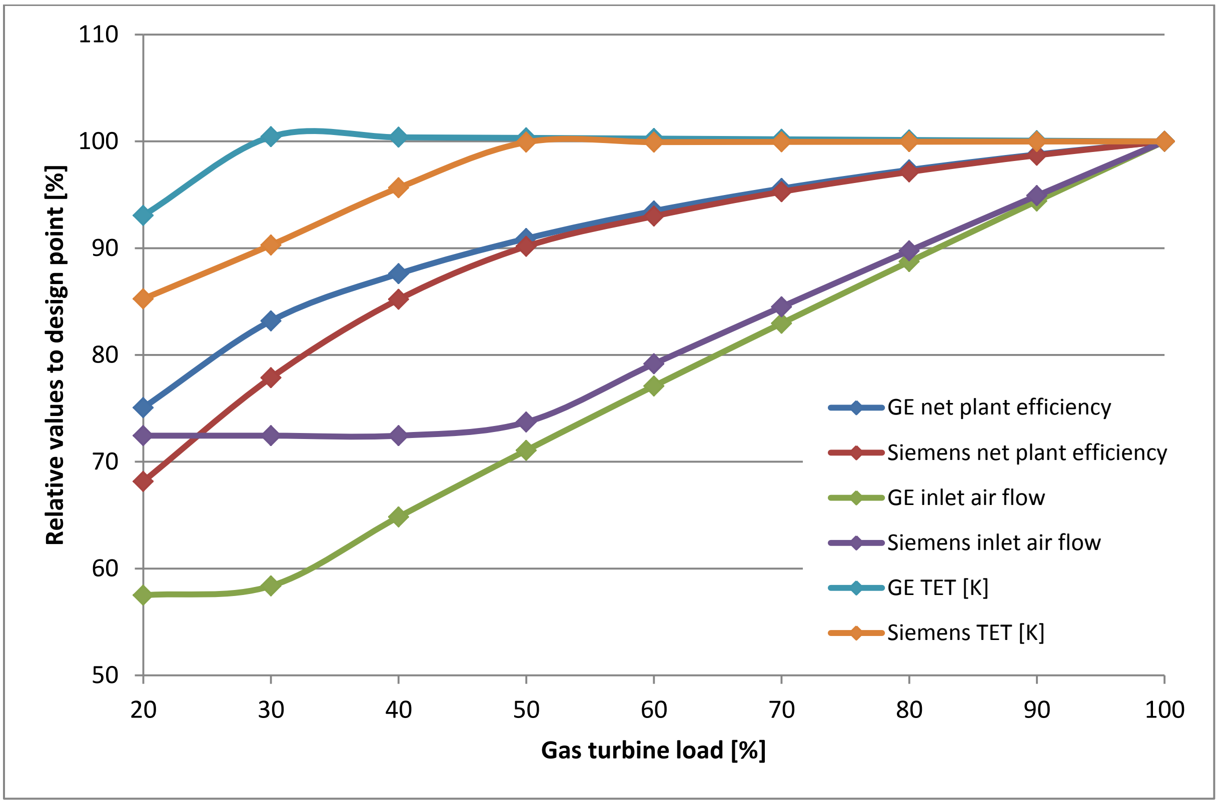

Figure 7.7 – Comparison of the part-load behavior between two different gas turbines. The design parameters for the steam cycle are exactly equal for both of the gas turbines.

Figure 7.7 gives a very good indication on how the plant efficiency is connected to the reduced air flow. The two different gas turbines are simulated in GT PRO/GT MASTER, under the exact same conditions as in Table 4.1. While the gas turbine from GE is able to regulate the air mass flow to 55-60% relative to the design point, the gas turbine from Siemens could only regulate the air flow to 70-75%. Consequently, the exhaust gas temperature from Siemens gas turbine breaks and is reduced rapidly at about 50% load. This results in a serious decrease in efficiency. At the same time, the exhaust temperature from GE's gas turbine remains almost constant until 30% load is reached. From that point the same tendency as for the gas turbine from Siemens could be seen. The exhaust gas temperature and the correspondingly efficiency are reduced quickly.

The results from the study in this section indicate a preferably part-load operation of the power plant with regulation of the inlet air mass flow. The default part-load control method has been used, with its specific target exhaust gas temperature. Other degrees of IGV regulation with other target exhaust gas temperature may be used. This is explored further in chapter 7.1.2

The most important results from the part-load simulation, with the default IGV control mode, are given in Table 7.1. The results are used further in chapter 7.2.2

Gas turbine								
Load	%	**100**	**90**	**80**	**70**	**60**	**50**	**40**
Flue gas composition								
Nitrogen	mol%	74,2	74,25	74,3	74,37	74,44	74,53	74,63
Oxygen	mol%	11,85	11,99	12,16	12,34	12,55	12,79	13,08
Carbon dioxide	mol%	4,22	4,15	4,08	3,99	3,89	3,78	3,64
Water	mol%	8,84	8,71	8,57	8,41	8,22	8,01	7,76
Argon	mol%	0,89	0,89	0,89	0,9	0,9	0,9	0,9
Compressor								
Inlet air flow	kg/s	642	606	570	533	495	456	416
Exhaust gas parameters								
Flow rate	kg/s	658	621	584	545	506	467	425
Turbine exhaust gas temperature	°C	644,9	645,5	646,1	646,7	647,3	647,9	648,4
Steam turbine								
Crossover IPT/LPT pressure	bar	3,9	3,7	3,5	3,3	3	2,8	2,6
Plant performance								
Gross gas turbine output	MW_{el}	285,2	257,4	229,5	201,4	173,1	144,6	116
Gross steam turbine output	MW_{el}	157,2	148,6	139,8	130,8	121,6	112,1	102,2
Net power output	MW_{el}	435,6	399,4	362,7	325,7	288,3	250,5	212
Fuel LHV	MW_{th}	751	697,1	642,6	587,5	531,7	475,1	417,3
LHV net electric efficiency	%	58,0	57,3	56,5	55,4	54,2	52,7	50,8

Table 7.1 – Part-load behavior for the NGCC without CO_2 capture, with default part-load control method [14].

7.1.2 IGV Control Methods at 60% Gas Turbine Load

In the last section a specific IGV control mode for part-load operation was studied and compared with a fixed air flow rate into the gas turbine. However, there are other degrees of IGV regulation. The IGVs are regulated simultaneously with the fuel flow in order to obtain a target exhaust gas temperature for a given gas turbine power output. Thus, for each load there are multiple combinations of the air- and fuel flow that gives the demanded power output in the gas turbine and a unique TET. In this part, it is looked into different combinations of those two flows and their respective part-load behavior. The study is restricted to different combinations in order to obtain 60% gas turbine power output.

A specific IGV control-tool in GT MASTER is used in order to regulate the air- and fuel mass flow for the given gas turbine power output. The accuracy in the simulations is high, but there are small deviations in the gross gas turbine output within a specified load. In the results presented in this chapter, only the simulations with the most equal gas turbine outputs are chosen. In that way, the parameters related to the different control methods could be compared.

The gas turbine output is fixed at 60% of its design output. The air flow into the gas turbine is changed, and the fuel flow is regulated in GT MASTER in order to obtain the specific gas turbine power output. In Figure 7.8, the relation between the decisive parameters in the gas turbine is given for a gas turbine operating at 60% load.

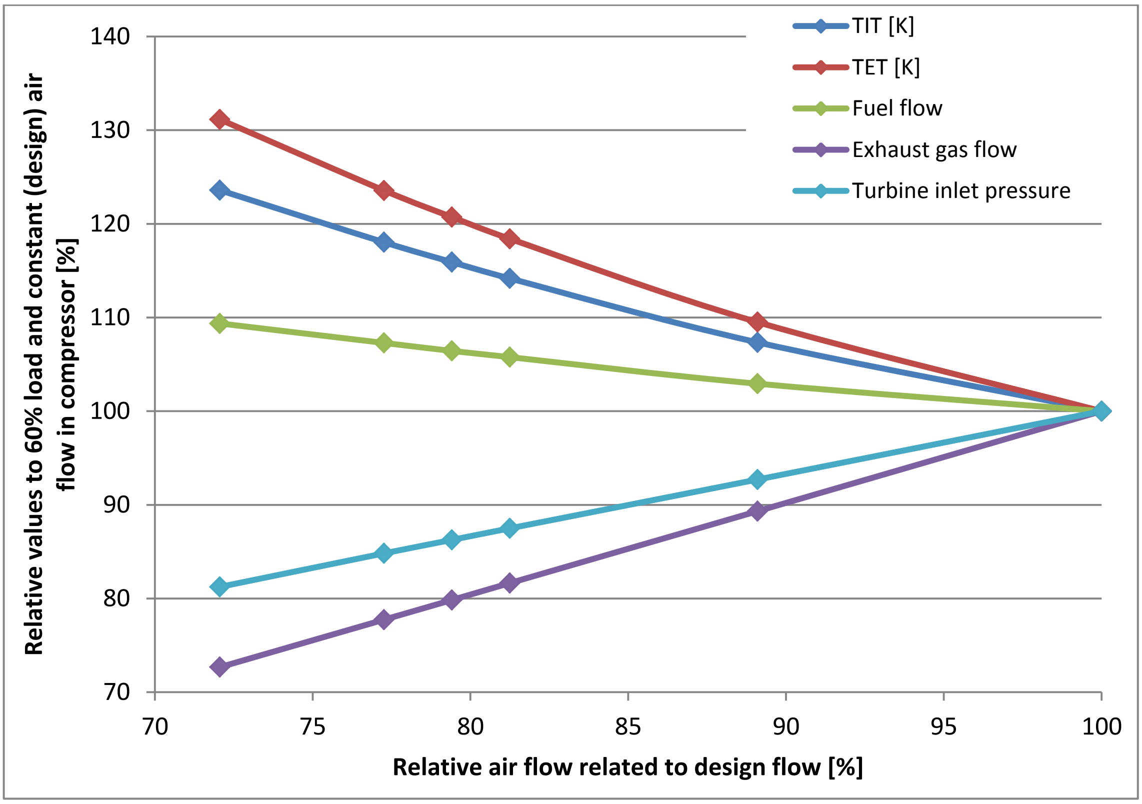

Figure 7.8 – Changes in parameters for different part-load control methods. The gas turbine power output is fixed at 60%. The relative values are related to the constant air flow operation point, at 60% load.

As the turbine operates in the choked area, the turbine inlet pressure is still proportional to the inlet mass flow and the square root of the TIT. The exhaust gas flow decreases due to the decrease in inlet air flow. At the same time, the fuel flow increases and thereby; the TIT increases. The relative reduction in mass flow is higher than the relative increase in TIT. In addition, the pressure is proportional to the square root of the TIT. Thus, the pressure decreases as the air flow into the turbine is reduced.

There are a lot of parameters affecting the gas turbine power output. The compressor work will vary with the inlet air flow and the compressor pressure ratio, while the gas turbine work depends on the turbine inlet pressure, TIT, mass flow, c_p etc. However, in order to restrict the amount of work in this study these parameters and their influence on the gas turbine power output is not examined in detail. Nevertheless, the relation between the TIT and the inlet turbine pressure is important in order to obtain a constant gas turbine power output.

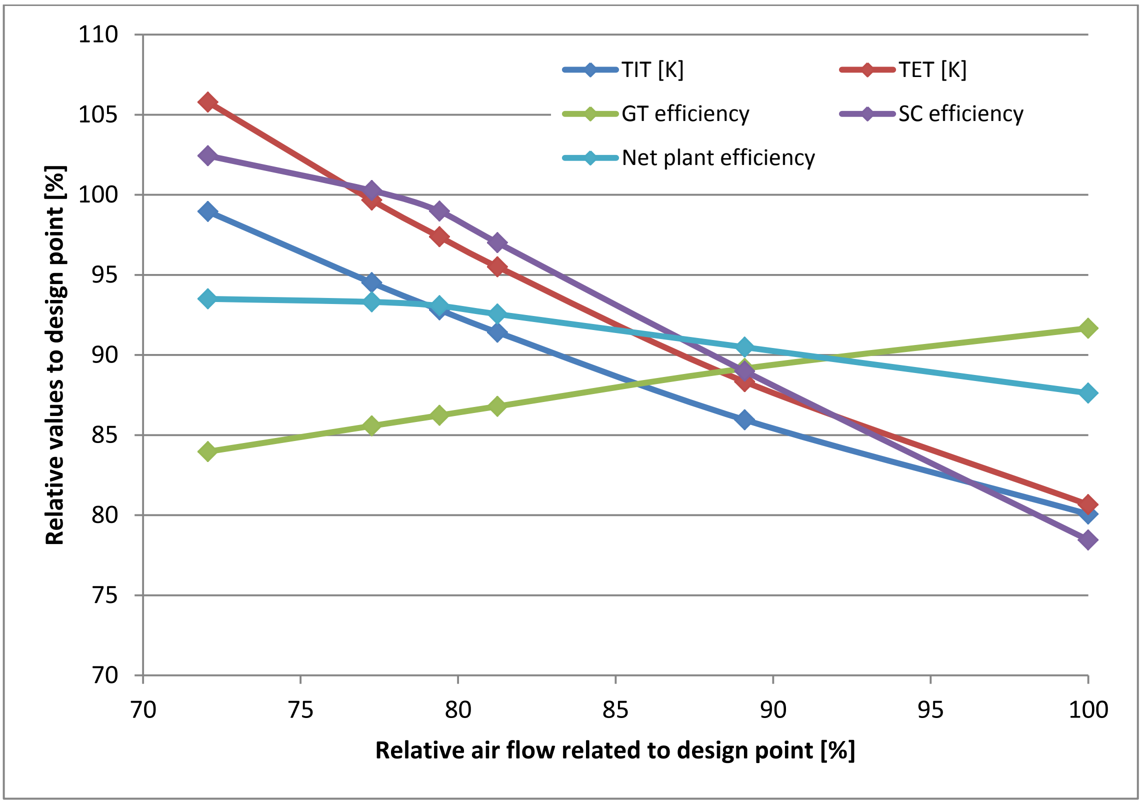

Figure 7.9 - Changes in parameters for different part-load control methods. The gas turbine power output is fixed at 60%. The relative values are related to the design point.

Figure 7.9 demonstrates the relations between the temperature, air flow and efficiencies in the power plant for the same simulation as in Figure 7.8. The parameters in the figure are relative to the design point. The gas turbine efficiency is defined as the gas turbine output per LHV of the fuel. The LHV per kg fuel and the gas turbine power output are constant in Figure 7.9, thus the efficiency is inversely proportional to the fuel flow. This relation is clear when the GT efficiency in Figure 7.9 and fuel flow in Figure 7.8 is compared. Thus, the gas turbine efficiency decreases as the air flow is reduced for a given power output. It seems like the decrease is almost constant for reduced air flow, within the air flow range in the simulations.

As the TIT increases and the exhaust gas mass flow decreases for reduced air flow, the TET will increase. This could be seen in Figure 7.9. The gross power output- and efficiency in the second cycle is highly dependent on the TET. The connection is indicated in Figure 7.9. As the TET increases, the steam cycle efficiency increases rapidly. The incline is reduced as the TET approaches its design temperature.

The net plant efficiency depends on the GT efficiency and the SC efficiency. As the air mass flow is reduced from the design value, the SC efficiency increases rapidly compared to the decrease in GT efficiency. Thus the net plant efficiency increases. When the air flow is further reduced, the incline in SC efficiency is slower. The decrease in GT efficiency is almost constant, thus, the increase in net plant efficiency flattens out. The relative gain in net plant efficiency is small as the air flow is reduced more than 20% from its design.

In order to explain the sudden change in the SC efficiency line, the SC efficiency is divided into two different efficiencies; HRSG efficiency and ST efficiency. The SC efficiency is the product of these two efficiencies. The efficiencies are plotted in Figure 7.10 against the relative air flow.

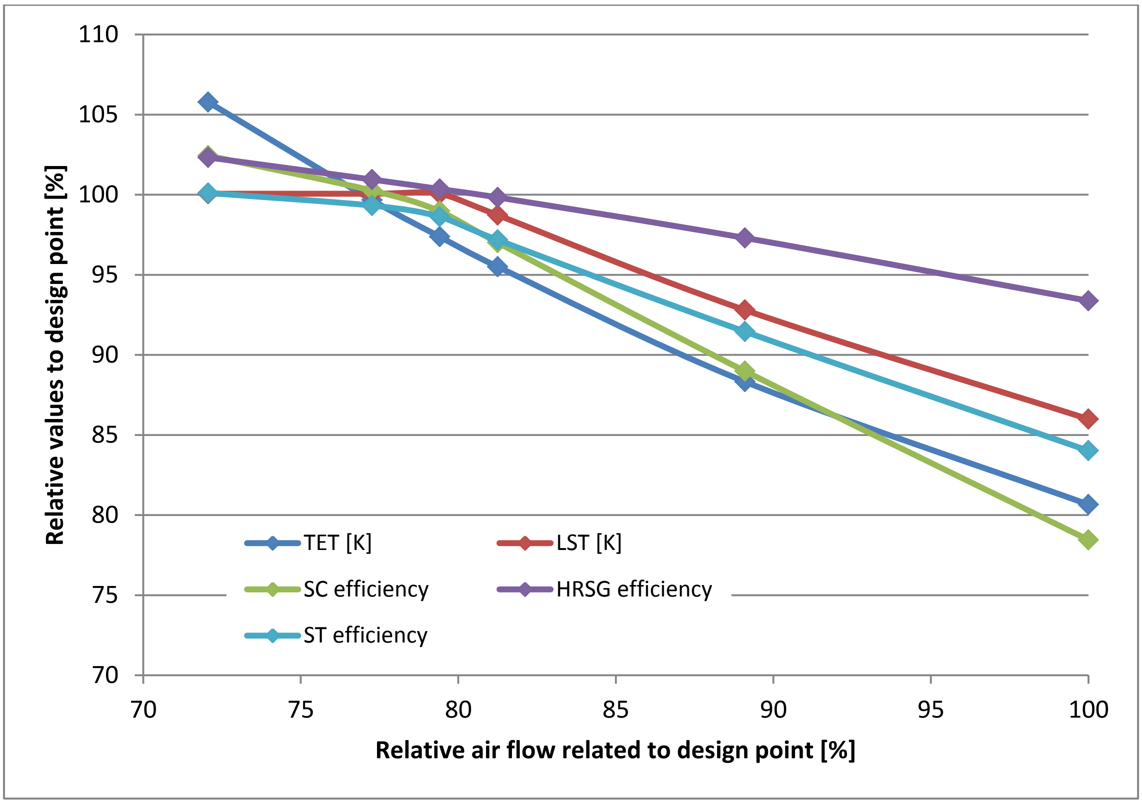

Figure 7.10 - Changes in parameters for different part-load control methods. The gas turbine power output is fixed at 60%. The relative values are related to the design point.

The reason for the sudden change in SC efficiency is a reduced incline in the ST efficiency. From Figure 7.10, an interesting observation is made. The ST efficiency line follows the live steam temperature line almost perfectly. This emphasizes the influence of the live steam temperature on the steam turbine efficiency. When the air flow is high, the TET is low, and the steam cycle could not obtain a constant LST. The effect is a low ST efficiency. As the air flow is reduced, the TET and LST are increasing. When the TET is approaching its design temperature, the LST is flattening out because of the maximum LST restriction. At the same time, the ST efficiency is approaching an asymptotic value.

The increase in HRSG efficiency is almost constant as the air flow is reduced. The HRSG efficiency depends on the heat utilization of the exhaust gas. The heat transfer in the HRSG is influenced of the steam pressure levels and the steam generation, and those parameters are calculated with regards to the exhaust gas temperature.

The steam cycle efficiency is the product of the HRSG- and the ST efficiency. Based on the behavior of those two efficiencies in Figure 7.10, the SC efficiency is increasing rapidly as the air flow in the compressor is reduced from its design value. This increase lasts until the LST is equal to the LST in the design point. From that point, the SC efficiency increases less rapidly with a further reduction in the air flow, and TET.

From a power plant efficiency point of view the optimal operation of the power plant at 60% load is a reduced air mass flow of 80% or less, corresponding to a TET close to the design point or higher. At this point, the LST is constant from its design, and a further increase in TET does not affect the net plant efficiency that much. However, in a NGCC with CO_2 capture the capture plants affection of the different part-load operations in the power plant is of importance. The influence of the IGV regulations on the exhaust gas condition is described in Figure 7.11.

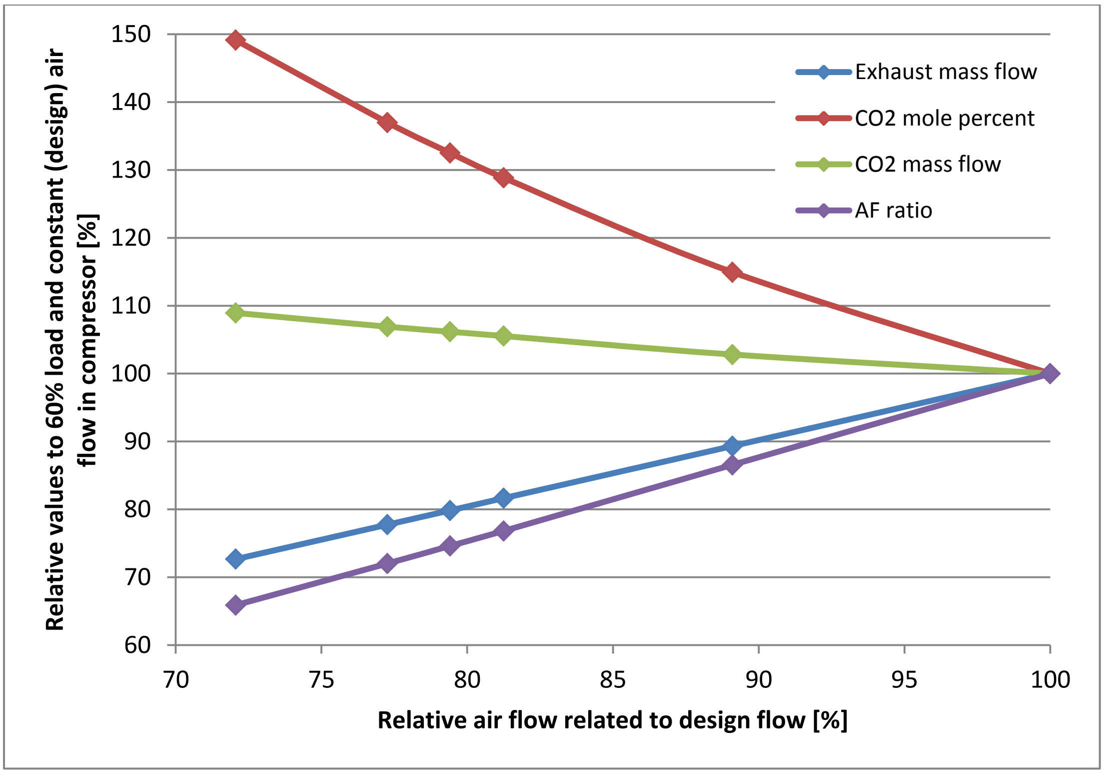

Figure 7.11 – Flue gas conditions for different part-load control methods, at 60% gas turbine power output.

In Figure 7.11, the parameters are relative to the constant air flow operation point. The AF decreases as the air flow into the compressor is reduced. Thus, the mole fraction of the CO_2 in the flue gas grows. The relative increase in CO_2 mole fraction is higher than the relative decrease in total flue gas mass flow. This is an important observation related to the capture process. The total CO_2 mass flow in the exhaust gas increases as the air flow is reduced. Thus, the amount of CO_2 to be removed in the capture plant is higher. As the AF increases, the concentration of O_2 in the flue gas increases. This could be problematic related to corrosion issues in both the NGCC plant and the capture plant.

Four different flue gas flows from the simulations in the current section are chosen, and described in details in Table 7.2. These flue gases, with their conditions and compositions, are used further in an analysis of the off-design behavior of the absorption process in chapter 7.2.3. In order to more easily separate the flue gases from each other, each flue gas stream has its own abbreviation.

Case					
Case		**FG 1**	**FG 2**	**FG 3**	**FG 4**
Flue gas composition					
Nitrogen	mol%	74,64	74,56	74,47	74,22
Oxygen	mol%	13,12	12,9	12,63	11,91
Carbon dioxide	mol%	3,62	3,73	3,85	4,19
Water	mol%	7,72	7,91	8,15	8,79
Argon	mol%	0,9	0,9	0,9	0,89
Compressor					
Inlet air flow	kg/s	522	510	496	463
Flue gas					
Mass flow rate	kg/s	533	521	507	474
Turbine exhaust gas temperature	°C	603,6	620,8	641,8	698
Steam turbine					
Crossover IPT/LPT pressure	bar	2,9	3	3	3,1
Plant performance					
Gross gas turbine output	MW_{el}	171,3	171,3	171,3	171,3
Gross steam turbine output	MW_{el}	114,1	117,5	120,4	126,5
Net power output	MW_{el}	279,1	282,5	285,3	291,5
Fuel LHV	MW_{th}	519,9	523,2	527,2	537,4
LHV net electric efficiency	%	53,7	54	54,1	54,2

Table 7.2 - Part-load behavior for the NGCC without CO_2 capture, at 60% gas turbine power output.

7.2 CO_2 Capture Process

The lean/rich cross-flow heat exchanger in the capture plant is not taken into account in the analysis of the capture plants off-design behavior. The pressure drop is fixed at 20 mbar, and the minimum pinch is 5 K. Either the pressure drop in the absorber is studied. It is fixed at 50 mbar. The change in pressure drop for off-design calculations of the total energy demand in the capture plant is important. As the flue gas mass flow is reduced for part-load operations in the power plant, the pressure drop in the absorber decreases. However, the magnitude of the pressure drop mainly affects the energy demand related to the fan in front of the absorber. The current work is focusing on the change in reboiler heat demand. Nevertheless, the changes in mechanical work are considered even though the variation in the pressure drop is not evaluated.

7.2.1 Parameter Variations

The conditions of the flue gas from a power plant will vary at part-load operation. At reduced load, the total mass of the flue gas decreases and the amount of CO_2 in the gas will vary. This has been explored in the last chapter. In order to give some indications on the behavior of the absorption process for different flue gases in a capture plant, the influence related to varied gas mass flow and CO_2 mole fraction is examined separately. The parameters from the design plant are the basis for the study.

First the influence of a reduced gas mass flow with a constant CO_2 mole fraction in the flue gas is studied, and then the influence of a reduced CO_2 mole fraction for a constant gas mass flow is investigated.

7.2.1.1 Flue gas mass flow

When the mass flow of the flue gas is reduced from its design flow, and the absorber height is kept constant, the size of the absorber is relative greater for the reduced gas flow than the design flow. This may indicate an increased contact period between the gas- and liquid flow in the absorber. As Figure 6.1 in chapter 6.2.2 illustrated, the operation line could be shifted in parallel towards the equilibrium line when the contact period increased. Thus, for a given L/G ratio the lean- and rich CO_2 concentration will increase.

As the results in chapter 6.2.2 and the theory have stated, the reboiler duty varies with the L/G ratio and its respective lean CO_2 concentration. The point where the lowest reboiler occurs depends on the sensible heat and the latent heat. The sensible heat demand increases almost linearly with the L/G ratio, while the latent heat depends highly on the lean CO_2 concentration. In Figure 7.12, the lean CO_2 concentration is varied in Hysys for four different gas mass flows, and plotted against the specific reboiler duty. The L/G ratio is regulated by an adjustor in order to attain 90% removal efficiency. The gas flows used in this simulation are the flue gases from the respective 100-, 90-, 80- and 70% gas turbine loads, with a constant AF.

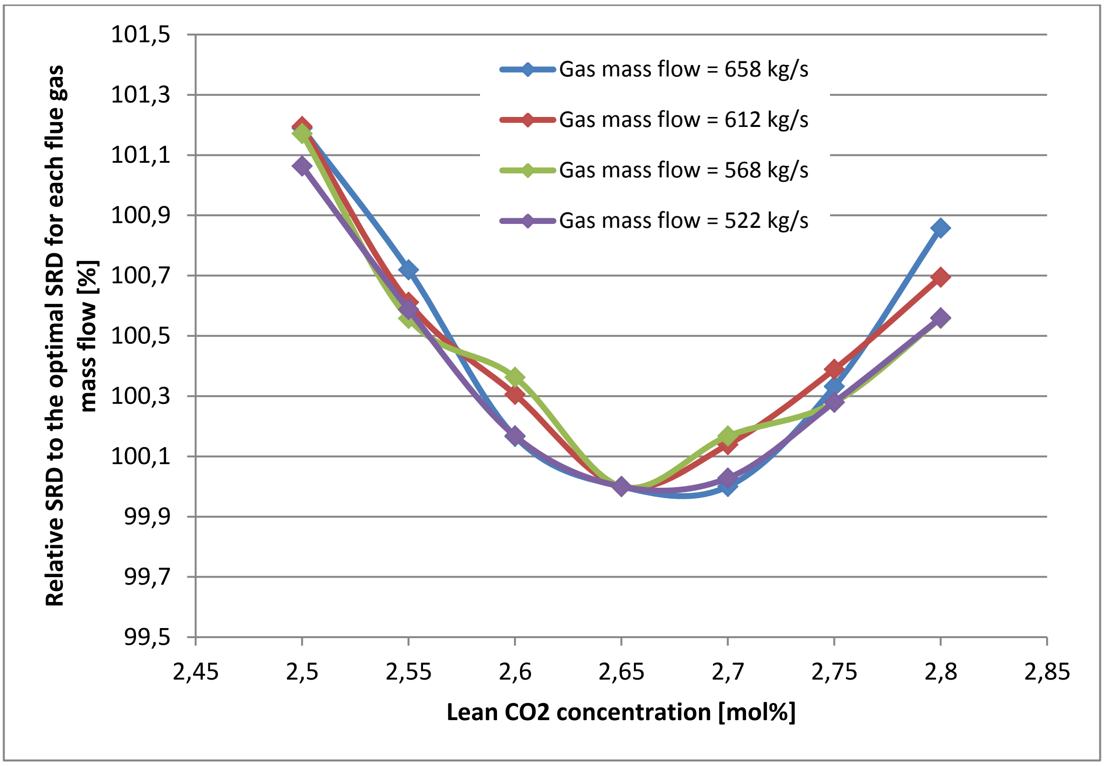

Figure 7.12 – Relative SRD against lean CO_2 concentration. Four different flue gas flows are simulated, with the exact same compositions.

The relative SRD is the SRD divided by the lowest SRD for each gas flow separately. The simulations in Hysys are very sensitive to small changes in the conditions, thus the results may not be exactly. However, there is a clear tendency of an optimal SRD at 2,65%-2,7% lean CO_2 concentration. This is in accordance to the theory about the latent heat, as the saturation pressure for the CO_2 in the liquid stream is low. For simplification, it is in the current work assumed an optimal lean CO_2 concentration of 2,65% with regard to the gas mass flow.

The respective optimal reboiler duty and its belonging L/G ratio and rich concentration is plotted against the reduced gas mass flow in Figure 7.13. The lean CO_2 concentration is fixed at 2,65%, and the L/G ratio is adjusted in order to obtain 90% removal efficiency.

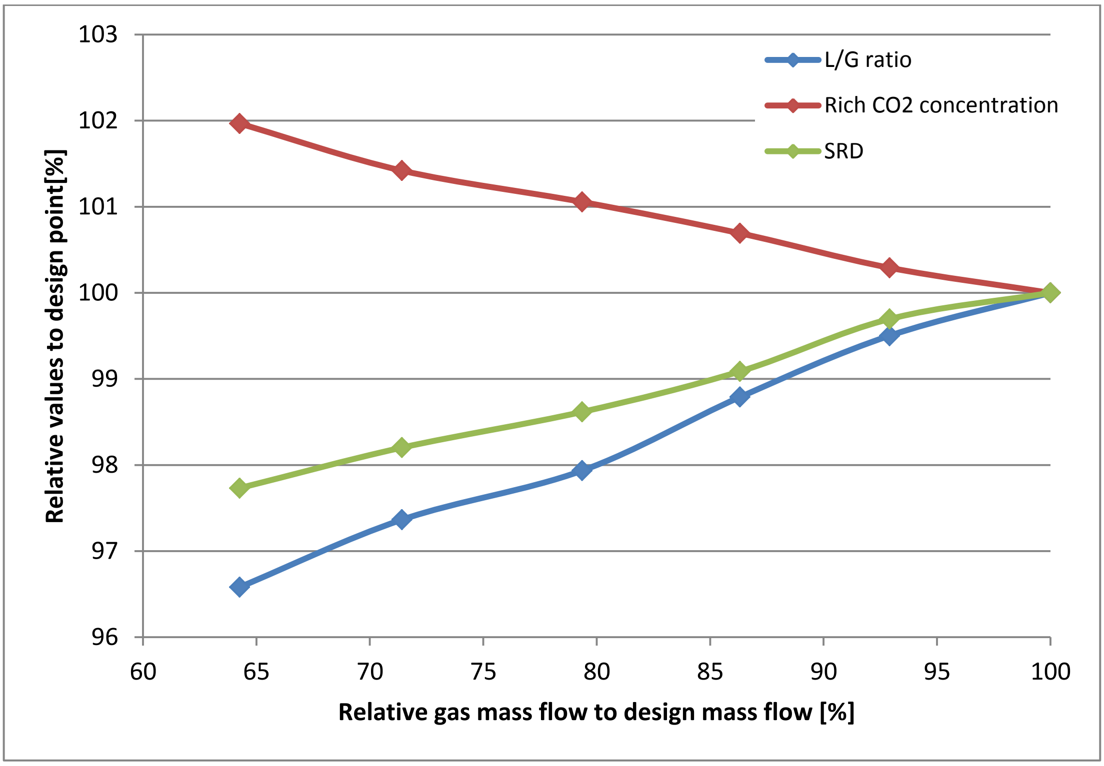

Figure 7.13 – Parameter changes against a reduced flue gas mass flow. The flue gas compositions are constant from the design point.

There is a small change in L/G ratio for reduced gas mass flow. The L/G ratio is about 3,5% lower at 65% gas mass flow compared to the design. The optimal specific reboiler duty is also reduced marginally. There is about 2% reduction in the SRD at 65% flue gas flow. The rich CO_2 concentration increases slightly.

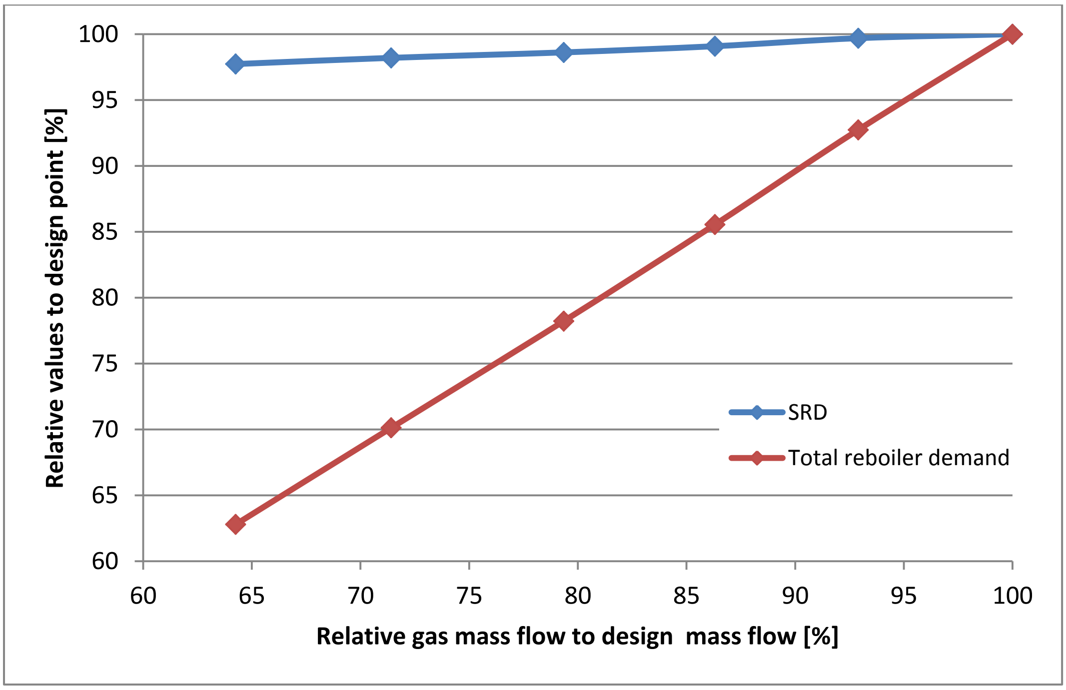

Figure 7.14 – Total energy demand in reboiler and specific reboiler duty plotted againsta reduced flue gas mass flow. The compositions of the flue gas flows are constant.

The total energy demand decreases rapidly because of the reduced CO_2 mass flow captured. As the mol% CO_2 in the flue gas is constant, the CO_2 captured is reduced proportionally to the flue gas mass flow. Since the SRD decreases slightly for reduced gas mass flow, the total energy demand is reduced marginally more than the relative gas mass flow. This is illustrated in Figure 7.14.

From the results in this section, it seems to be a connection between the reboiler duty and the L/G ratio. In these simulations, the mol% of the lean CO_2 stream was constant at 2,65%. The results might indicate that the change in reboiler duty is the result of a reduced sensible heat demand in the stripper, and that the latent heat is nearly constant because of a constant lean CO_2 concentration.

7.2.1.2 Flue gas CO_2 mole percent

In this section, the flue gas mass flow is assumed constant and the CO_2 mole fraction in the flue gas is varied. As the flue gas mass flow is equal to the design, the relative size of the absorber could be assumed equal to the design. From the theory, a reduced partial pressure of the CO_2 into the absorber will increase the minimum compression work.

First, the optimal lean CO_2 concentration has been studied. The flue gas mass flow is fixed at the design mass flow, and the flue gas composition is changed. The actual flue gas compositions from the part-load operation study with constant air flow into the gas turbine are used. For each flue gas composition, the lean CO_2 concentrations are varied. The L/G ratio is adjusted in Hysys in order to reach 90% removal of the CO_2 for the given lean CO_2 concentration. The results are given in Figure 7.15.

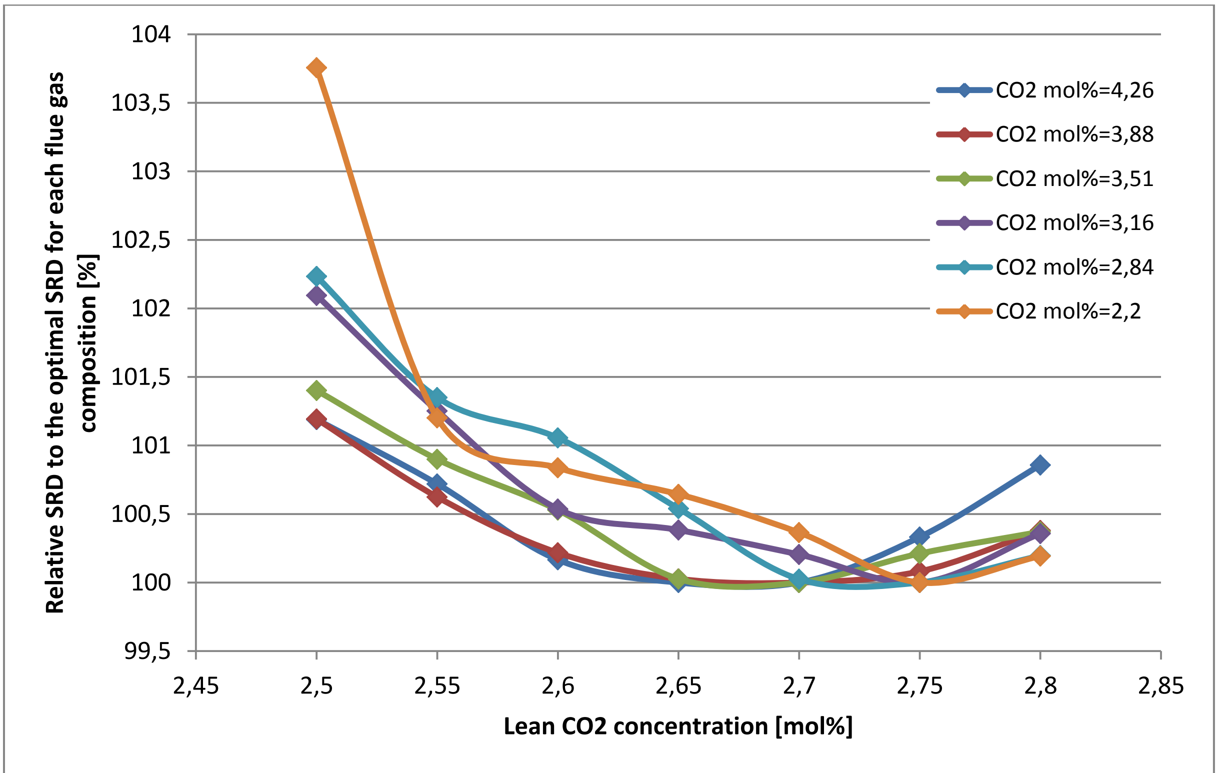

Figure 7.15 - Relative SRD against lean CO_2 concentration. Six different flue gas compositions are simulated, with the exact same mass flow. The mass flow is equal to the design flow.

The relative specific reboiler duties in Figure 7.15 are defined as the SRD divided by the lowest SRD for each CO_2 mol% in the flue gas.

The simulations in Hysys are very sensitive to small changes in the input conditions. Therefore, there may be some inaccurate results when the lean CO_2 concentration is regulated at small steps. Nevertheless, the results may be used as indicators and the tendency in the figure is clear. From Figure 7.15, it seems like the CO_2 mole fraction has a small effect on the optimal lean CO_2 concentration. The tendency is a slightly higher optimal lean CO_2 concentration for reduced CO_2 mol% in the flue gas.

However, according to the Figure 7.15 the difference in optimal lean CO_2 concentration is minimal in the range that is relevant for this thesis, and varies between 2,65% and 2,75%. In order to simplify the simulations of the different part-load conditions from the power plant, the lean CO_2 concentration is fixed at 2,65% in the current work. Slightly better results could have been obtained if the lean CO_2 concentration had been optimized for each case.

In the next simulation, the lean CO_2 concentration is fixed at 2,65%. The L/G ratio is adjusted in Hysys in order to achieve 90% removal efficiency. The mass flow of the flue gas is constant, but the composition is changed and the belonging CO_2 mol% is reduced from its design.

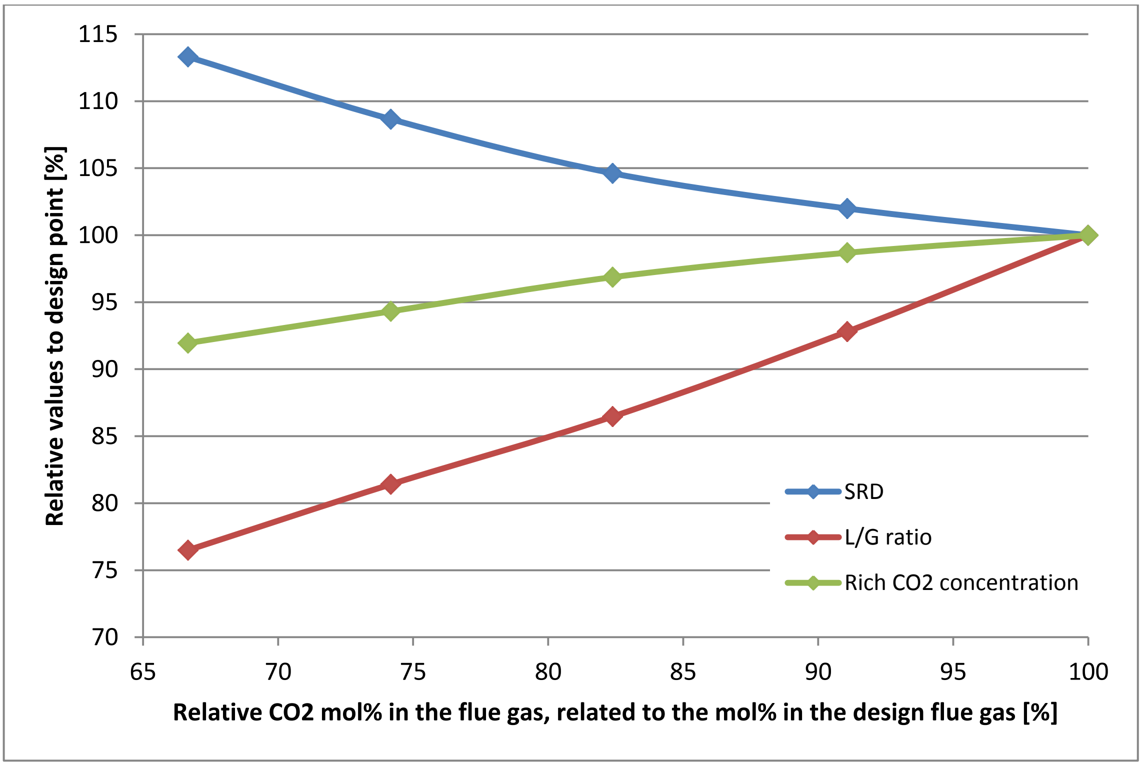

Figure 7.16 – Parameter changes for a reduced CO_2 concentration in the flue gas. The flue gas mass flow is kept constant.

The graph in Figure 7.16 indicates an exponential increase in specific reboiler duty for reduced CO_2 mole fraction in the flue gas. At the same time, the L/G ratio decreases rapidly. A decrease in L/G ratio and a constant lean CO_2 concentration should in theory result in a decreased SRD. A lower L/G ratio leads to less sensible heat required in the stripper per flue gas mass flow. However, the mass flow of CO_2 per total mass flow in the flue gas decreases as the mol% is reduced. This result in an increased sensible heat demand per kg CO_2 captured compared to the sensible heat per kg flue gas flow. For that reason, it is difficult to predict the L/G ratios effect on the specific reboiler duty for this specific case.

From Figure 7.16, the rich CO_2 concentration decreases as well for reduced CO_2 mol% in the flue gas. A decrease in the rich CO_2 concentration represents a reduced saturation pressure of the CO_2 in the liquid stream into the stripper. Thus, a lower partial pressure of the CO_2 in the gas stream in the upper part of the stripper is required, and the latent heat demand increases. The reduction in rich CO_2 concentration is also exponential, and there seems to be a connection between the rich CO_2 concentration and the specific reboiler duty.

The reduction in L/G ratio is about 15% compared to its design value when the mol% CO_2 in the flue gas is 80% of its design. The belonging increase in SRD is about 5% of its design.

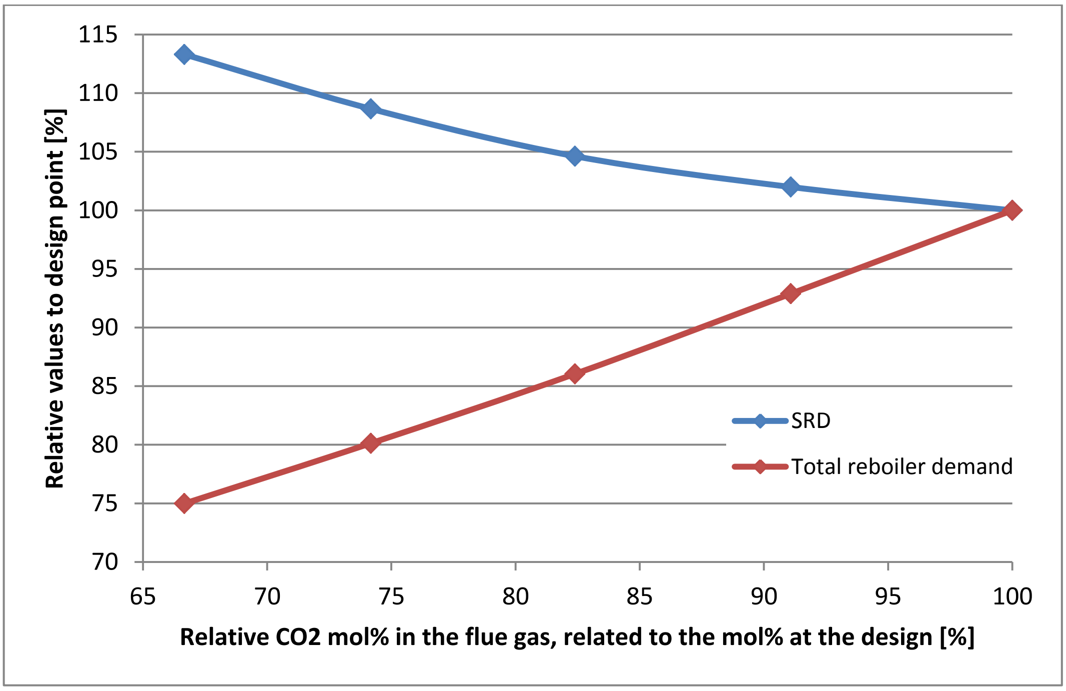

Figure 7.17 – Energy demand in the reboiler, for a reduced CO_2 concentration in the flue gas. The flue gas mass flow is kept constant.

Figure 7.17 illustrates the connection between the specific reboiler duty and the total energy in the reboiler for the example in Figure 7.16. Even though the SRD increases for reduced CO_2 mol% in the flue gas the total energy demand decreases. Thus, the specific separation work is high, but the total energy demand is low for reduced mol% CO_2 in the flue gas. A reduced CO_2 mol% leads to a reduced mass flow of CO_2 captured. The increasing specific reboiler duty counteract with the reduced CO_2 mass flow, thus the decline in total energy demand is slower as the CO_2 mole fraction is reduced.

7.2.2 Default Part-load Method

In this section, the optimal off-design operation of the capture plant with the flue gas conditions from chapter 7.1.1 is studied. First, the operation conditions leading to the lowest energy demand in the reboiler is examined. As the absorber performance is affected of the degree of wetting of the packing material in the column, also the influence of different liquid circulation rates on the energy demand in the reboiler are studied.

7.2.2.1 Optimal operation in capture plant based on lowest specific reboiler duty

The flue gas conditions given in table Table 7.1 are used as input in the part-load simulations in this section. On basis of the study in chapter 7.2.1, the lean CO_2 concentration corresponding to the lowest specific reboiler duty is assumed constant, independently of the flue gas conditions. This concentration is fixed at 2,65%. The L/G ratio is adjusted in order to obtain 90% removal of the incoming CO_2 flow. The most important results are collected in Table 7.3.

Gas turbine								
Load	**[%]**	**100**	**90**	**80**	**70**	**60**	**50**	**40**
Flue gas								
Mass flow	[kg/s]	658	621	584	545	506	467	425
Percent of design flue gas flow	[%]	1	94,4	88,8	82,8	76,9	71	64,6
Mole percent CO_2	[%]	4,26	4,19	4,11	4,03	3,93	3,81	3,67
Mass flow CO_2	[kg/s]	43,6	40,5	37,3	34,1	30,9	27,6	24,3
Absorber								
Liquid/gas ratio	[$kgmole_L/kgmole_G$]	1,4	1,38	1,35	1,32	1,28	1,24	1,19
Inlet liquid flow	[kg/s]	735	682	628	574	518	464	408
Percent of design liquid flow	[%]	100	92,8	85,4	78,0	70,5	63,1	55,5
Stripper								
Rich CO_2 concentration	[%]	5,49	5,49	5,5	5,5	5,5	5,51	5,51
Rich loading	[$kgmole_{CO2}/kgmole_{MEA}$]	0,47	0,47	0,47	0,47	0,47	0,47	0,47
Lean CO_2 concentration	[%]	2,65	2,65	2,65	2,65	2,65	2,65	2,65
Lean loading	[$kgmole_{CO2}/kgmole_{MEA}$]	0,22	0,22	0,22	0,22	0,22	0,22	0,22
Capture plant performance								
Removal efficiency	[%]	90	90	90	90	90	90	90
Specific reboiler duty	[MJ/kg_{CO2}]	3,62	3,62	3,61	3,62	3,62	3,62	3,62
Total reboiler energy demand	[MW]	141,9	131,9	121,5	111,1	100,5	89,9	79,1
Mechanical work	[MW]	7,8	7,4	7	6,6	6,1	5,7	5,2
CO_2 compression work	[MW]	12,9	12	11,1	10,1	9,2	8,2	7,2

Table 7.3 – Capture plant performance, default part-load control strategy in gas turbine.

The specific reboiler duty remains approximately constant for the reduced load. According to the results in chapter 7.2.1, a reduction in flue gas mass flow contributes to a reduced SRD and the reduction in CO_2 mol% contributes to an increased SRD. These effects are opposite, and they results in an almost constant SRD for the flue gas conditions in this section. Thus, the total energy demand in the reboiler depends only on the amount of CO_2 in the flue gas. The reduction of the CO_2 mass flow, from Figure 7.6 in chapter 7.1.1, is almost linearly. At 40% load the CO_2 mass flow in the flue gas is about 55% of its design value. From the values given in table 2.3, the total energy demand is reduced with about 10 MW per 10% load decrease.

The mechanical work in Table 7.3 is reduced with the load. As the flue gas mass flow is reduced, the energy required in the blower in front of the absorber is reduced, since the fan work is proportional to the mass flow. The pressure drop in the absorber is assumed constant. A reduction in the flue gas mass flow in addition to a reduction in the liquid flow results in a lower pressure drop through the column. By implementing this into the model, the mechanical energy given in Table 7.3 would have been even lower for reduced load. The changes in liquid flow do also contribute to the reduced mechanical work at part-load, because of reduced pump work in the cycle. In addition, as the liquid flow is reduced the required cooling duty is lower. A lower cooling duty represents a lower cooling water circulation, and thus the pump work related to the cooling water is reduced.

An interesting observation from Table 7.3 is the reduced liquid mass flows. The optimal liquid mass flow decreases rapidly as the load is reduced. This is illustrated in Figure 7.18.

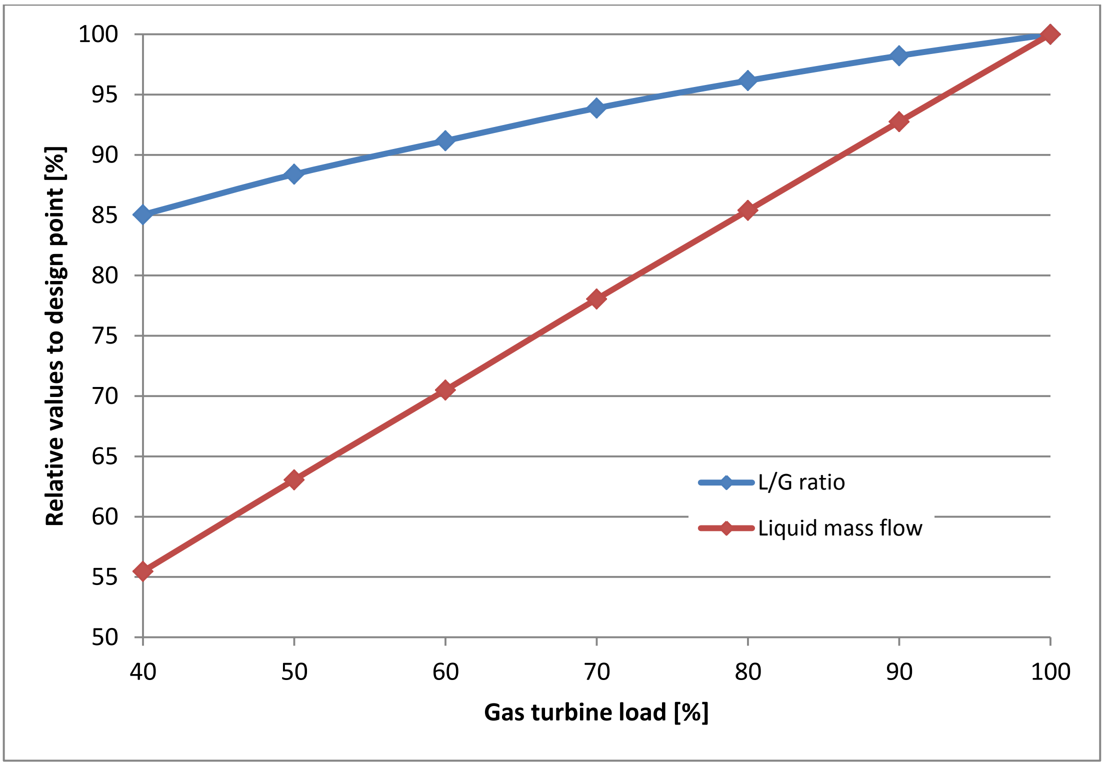

Figure 7.18 – Change in liquid flow for reduced gas turbine load. The L/G ratio is regulated in order to obtain the lowest SRD.

As Figure 7.18 indicates the optimal liquid mass flow into the absorber decreases with an almost constant rate with the reduced gas turbine load. The reduction is enhanced by the reduced optimal L/G ratio. For a 60% gas turbine load, the optimal liquid mass flow is just above 70% of its design flow. A low liquid mass flow could result in incomplete humidification of the packing material in the absorber, and this will result in reduced performance in the absorber. In order to reduce the risk of not enough wetting of the packing material, it may be necessary to fix the liquid mass flow. This will affect the theoretical SRD. To what extent the SRD is affected is explored in the next section.

7.2.2.2 Variation in liquid flow rate

In this section, the results from the simulation of the capture plants operation with a variation in liquid circulation rate are presented. For simplifications, the regulation of the liquid mass flow in the capture plants part-load study is related to the total liquid mass flow and not the solvent flow. The composition of the liquid mass flow will vary with the lean CO_2 concentration. For example, if a fixed liquid mass flow is used as part-load operation control, the solvent flow will decrease for an increased lean CO_2 concentration.

The liquid mass flow into the absorber is regulated in Hysys by an adjuster. The lean CO_2 concentration is varied manually until 90% removal of the CO_2 in the flue gas has been obtained. The shape of the specific reboiler duty curves will be equal to the graphs presented in the previous sections. At low L/G ratios, the lean CO_2 concentration is low and reduced rapidly. Thus a high steam amount is required in the stripper in order to reduce the CO_2 partial pressure in the gas flow. The corresponding reboiler heat demand is high. And opposite, when the L/G ratio is high and increases further the reboiler heat demand will increase almost linearly due to the sensible heat demand. The results are illustrated in Figure 7.19.

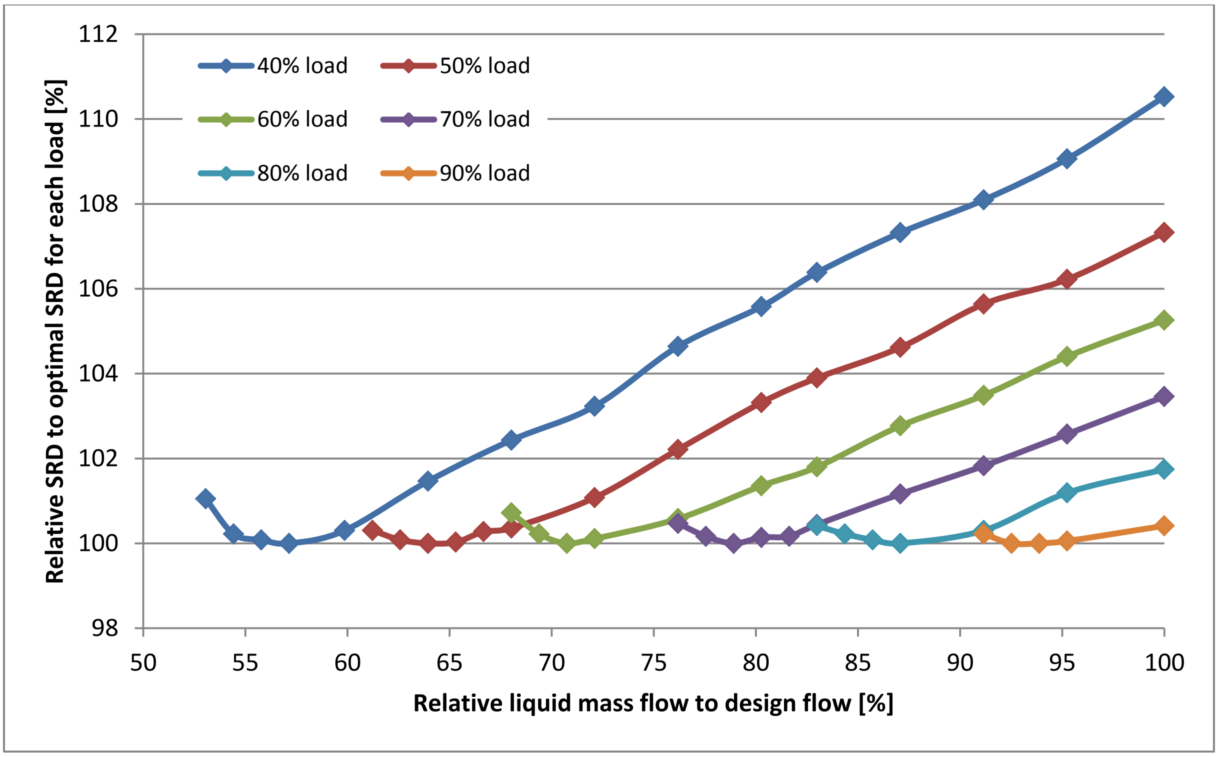

Figure 7.19 – SRD against liquid flow rate into the absorber for six different flue gas streams. The part-load control method in the gas turbine is the default strategy in GT MASTER.

The shape of the curves in Figure 7.19 is recognized from the previous sections. The most interesting part is the increase in SRD for increased liquid mass flow. The increase in SRD depends highly on the growing sensible heat demand. The incline is almost constant as the liquid flow increases. The figure indicates a more rapid increase in SRD per relative increase in the liquid mass flow for the flue gases at the lowest gas turbine loads. This is due to a lower CO_2 mass flow in the gas stream. Thus the liquid flow per kg CO_2 captured is higher.

In order to illustrate the behavior of the absorption process with a control of the liquid mass flow at part-load operation, three cases are established.

Case	Description
Case 1	There are no restrictions on the liquid mass flow into the absorber. The liquid mass flow is determined by the L/G ratio that provides the lowest reboiler duty.
Case 2	The liquid mass flow into the absorber is restricted to be minimum 80% of its design flow.
Case 3	There is no change in the liquid mass flow according to the design. Thus, the liquid mass flow is equal to the design flow.

Table 7.4 – Case descriptions.

Case 1

With free regulation of the liquid mass flow into the absorber, each part-load step will optimize its reboiler duty. The results are given in Table 7.5.

Gas turbine								
Load	**[%]**	**100**	**90**	**80**	**70**	**60**	**50**	**40**
Absorber								
Liquid/gas ratio	[$kgmole_L/kgmole_G$]	1,4	1,38	1,35	1,32	1,28	1,24	1,19
Inlet liquid flow	[kg/s]	735	682	628	574	518	464	408
Percent of design liquid flow	[%]	100	92,8	85,4	78,0	70,5	63,1	55,5
Stripper								
Rich CO_2 concentration	[%]	5,49	5,49	5,5	5,5	5,5	5,51	5,51
Lean CO_2 concentration	[%]	2,65	2,65	2,65	2,65	2,65	2,65	2,65
Capture plant performance								
Removal efficiency	[%]	90	90	90	90	90	90	90
Specific reboiler duty	[MJ/kg_{CO2}]	3,62	3,62	3,61	3,62	3,62	3,62	3,62
Total reboiler energy demand	[MW]	141,9	131,9	121,5	111,1	100,5	89,9	79,1
Mechanical work	[MW]	7,8	7,4	7	6,6	6,1	5,7	5,2
CO_2 compression work	[MW]	12,9	12	11,1	10,1	9,2	8,2	7,2

Table 7.5 - Capture plant performance, default part-load control strategy in gas turbine. Case 1.

Case 2

If the liquid mass flow is restricted to be minimum 80% of its design value, the part-load operation of the capture plant will be equal to Case 1 at 80% gas turbine load and above. For gas turbine loads below 80%, the liquid mass flows in Case 1 are below the limit. The results are given in Table 7.6.

Gas turbine								
Load	**[%]**	**100**	**90**	**80**	**70**	**60**	**50**	**40**
Absorber								
Liquid/gas ratio	$[kgmole_L/kgmole_G]$	1,4	1,38	1,35	1,35	1,45	1,57	1,72
Inlet liquid flow	[kg/s]	735	682	628	590	590	590	590
Percent of design liquid flow	[%]	100	92,8	85,4	80	80	80	80
Stripper								
Rich CO_2 concentration	[%]	5,49	5,49	5,5	5,49	5,43	5,36	5,27
Lean CO_2 concentration	[%]	2,65	2,65	2,65	2,72	2,92	3,13	3,32
Capture plant performance								
Removal efficiency	[%]	90	90	90	90	90	90	90
Specific reboiler duty	$[MJ/kg_{CO2}]$	3,62	3,62	3,61	3,62	3,66	3,74	3,82
Total reboiler energy demand	[MW]	141,9	131,9	121,5	111,2	101,9	92,9	83,5
Mechanical work	[MW]	7,8	7,4	7	6,6	6,2	5,8	5,4
CO_2 compression work	[MW]	12,9	12	11,1	10,1	9,2	8,2	7,2

Table 7.6 - Capture plant performance, default part-load control strategy in gas turbine. Case 2.

Case 3

If the liquid flow rate is restricted to be equal to the design flow, the liquid mass flow is fixed at its design value. All the part-load operations of the capture plant will be restricted by this limit. The results are given in Table 7.7

Gas turbine								
Load	**[%]**	**100**	**90**	**80**	**70**	**60**	**50**	**40**
Absorber								
Liquid/gas ratio	$[kgmole_L/kgmole_G]$	1,4	1,48	1,58	1,68	1,81	1,95	2,13
Inlet liquid flow	[kg/s]	735	735	735	735	735	735	735
Percent of design liquid flow	[%]	100	100	100	100	100	100	100
Stripper								
Rich CO_2 concentration	[%]	5,49	5,45	5,4	5,35	5,28	5,2	5,11
Lean CO_2 concentration	[%]	2,65	2,82	2,98	3,14	3,28	3,42	3,55
Capture plant performance								
Removal efficiency	[%]	90	90	90	90	90	90	90
Specific reboiler duty	$[MJ/kg_{CO2}]$	3,62	3,63	3,68	3,74	3,8	3,88	4
Total reboiler energy demand	[MW]	141,9	132,4	123,5	114,8	105,7	96,5	87,4
Mechanical work	[MW]	7,8	7,5	7,1	6,7	6,4	6	5,6
CO_2 compression work	[MW]	12,9	12	11,1	10,1	9,2	8,2	7,2

Table 7.7 - Capture plant performance, default part-load control strategy in gas turbine. Case 3.

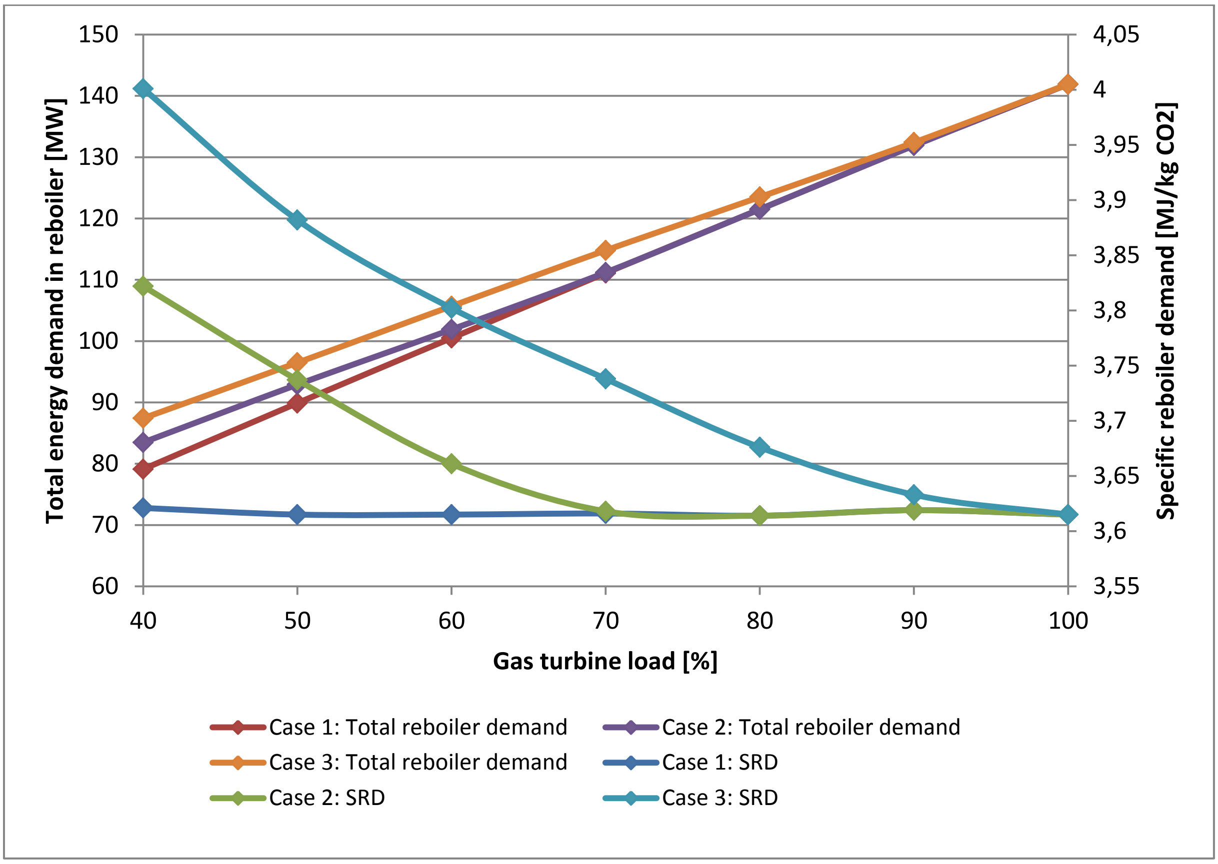

Figure 7.20 – SRD and total energy demand in reboiler against load reduction, for three different control methods of the liquid flow rate.

The difference in total energy demand and SRD for a given load between the three cases is given in Figure 7.20. The difference in total energy demand between the load steps in Case 3 is mainly the heat of absorption. As the liquid flow into the absorber is constant, the total sensible heat is almost constant. There will be some differences due to the inlet stripper temperature, the liquid flow composition and the liquid mass flow into the stripper. However, the differences are negligible compared to the high mass flow. The latent heat is low at high lean CO_2 concentrations. The heat of absorption, on the other hand, is considerable. As described in the theory, the heat of absorption is approximately constant at 2 MJ/kg CO_2.

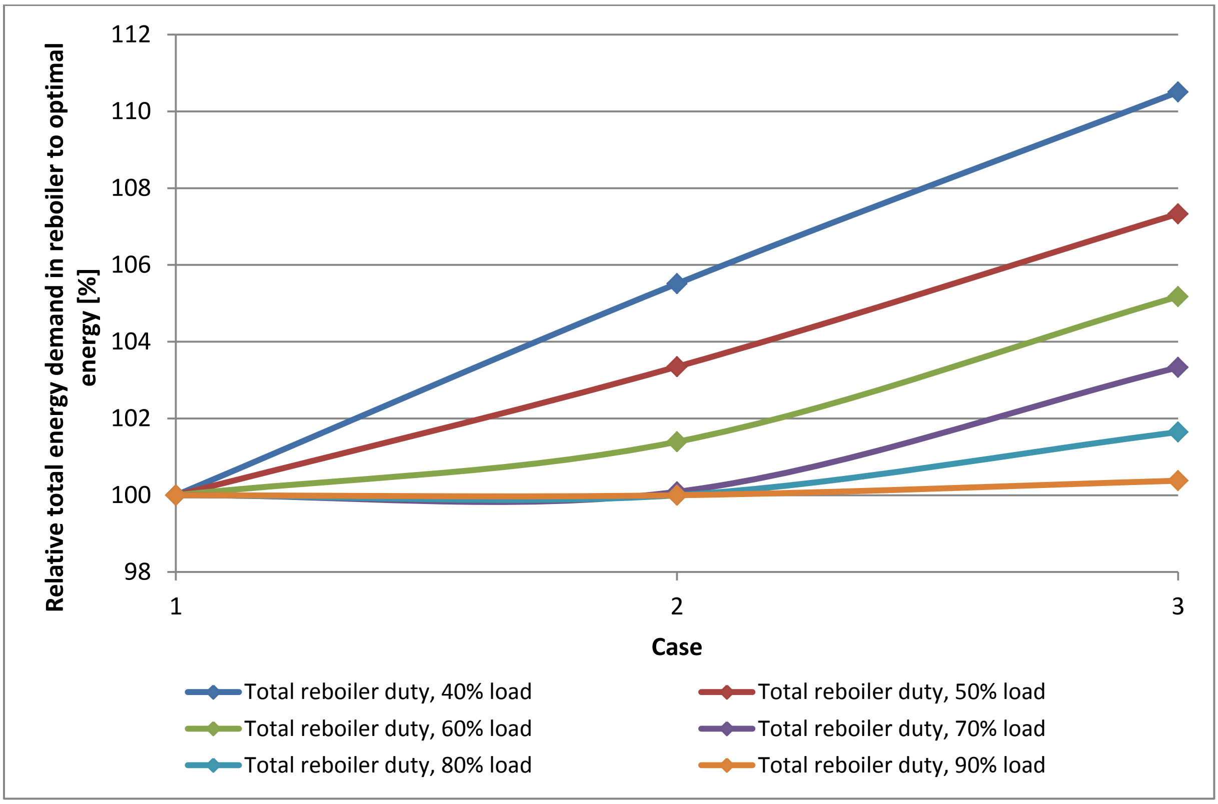

Figure 7.21 – Relative total energy demand in the reboiler compared to the lowest energy demand for each flue gas flow.

Figure 7.21 demonstrates how the total energy demand is influenced by a control of the liquid mass flow into the absorber. The main reason for the increase is the sensible heat. If the liquid mass flow is restricted to be minimum 80% of its design value, Case 2, the energy demand is almost unaffected for part-load operations down to 70% load. For a further decrease in load, the relative total energy demand in the reboiler increases. At 60% gas turbine load, the increase in relative energy demand is below 2%. At 40% load, the increase is almost 6%.

These increments in energy demand must be considered and compared to the risk of insufficient wetting of the absorber packing material.

Figure 7.22 shows the relation between the mechanical work in the absorption plant and the gas turbine load, for the three different cases. The relative decrease in mechanical work is significant. At 40% gas turbine load, the mechanical work is reduced with about 30%, dependent on the amount of liquid flow. However, the dependency of the liquid flow rate is less than the dependency on the gas flow. This is obvious from Figure 7.22. From Table 7.3, the reduction in liquid flow rate from its design flow is about 45% at 40% load. At the same time, the reduction in gas flow is about 35% at 40% load. As the liquid flow rate is kept constant in Case 3, the mechanical work is reduced with almost 30% at 40% load. Thus, the relative reduction in mechanical work is almost equal to the relative reduction in the gas flow. At the same time, the relative reduction in mechanical work is much smaller when the liquid flow is reduced with 45% and the gas flow is constant at 40% load. Thus, the mechanical work depends highly on the gas flow and less on the liquid flow.

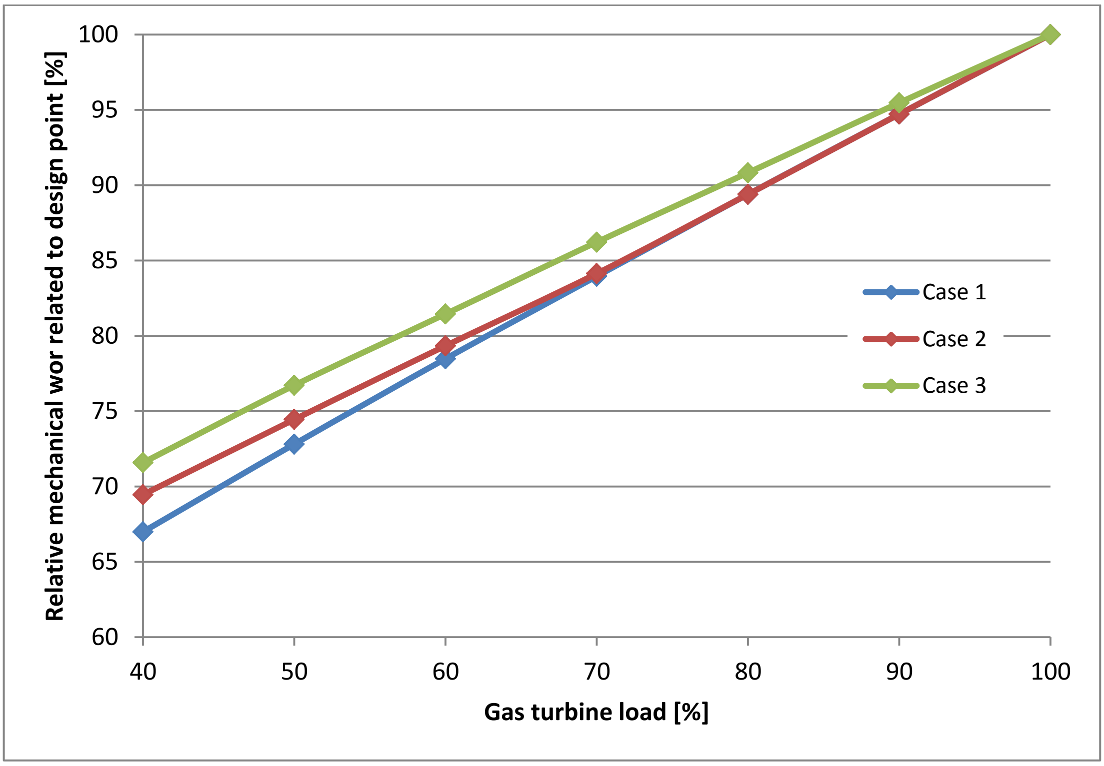

Figure 7.22 – Mechanical work plotted against gas turbine load and different control strategies on the liquid mass flow in the absorber.

7.2.3 IGV Control Methods at 60% Gas Turbine Load

In chapter 7.1.2, the power plant performance at part-load operation with different regulations of the IGVs for a given power output, were studied. The gas turbine load was fixed at 60% and different regulations of the inlet air flow in combination with fuel control were examined. In this section, four of the flue gases from the power plant are used further in the capture plant. The flue gas compositions are given in Table 7.2

The same procedure as in the last section is used. First, the lowest specific reboiler duty for each case is discussed. Then, the capture plant operation for different liquid mass flow rates into the absorber is analyzed.

7.2.3.1 Optimal operation in capture plant based on lowest specific reboiler duty

On basis of the study in chapter 7.2.1, the lean CO_2 concentration corresponding to the lowest specific reboiler duty is assumed constant, independently of the flue gas conditions. This concentration is fixed at 2,65%. The L/G ratio is adjusted in order to obtain 90% removal of the incoming CO_2 flow. The most important results are collected in Table 7.8.

Case					
Case		**FG 1**	**FG 2**	**FG 3**	**FG 4**
Flue gas					
Total mass flow	kg/s	533	521	507	474
CO_2 mole percent	%	3,65	3,76	3,89	4,23
Relative air flow	%	81,2	79,4	77,3	72,1
Absorber					
Liquid/gas ratio	$kgmole_L/kgmole_G$	1,22	1,24	1,27	1,36
Inlet liquid flow	kg/s	520	519	517	513
Percent of design liquid flow	%	70,7	70,6	70,3	69,8
Stripper					
Rich CO_2 concentration	%	5,44	5,46	5,5	5,57
Lean CO_2 concentration	%	2,65	2,65	2,65	2,65
Capture plant performance					
Removal efficiency	%	90	90	90	90
SRD	MJ/kg_{CO2}	3,68	3,65	3,62	3,56
Total reboiler demand	MW	100,2	100,1	100	100
Mechanical work	MW	6,4	6,3	6,1	5,8
CO_2 compression work	MW	9	9,1	9,1	9,3

Table 7.8 – Capture plant performance with four different flue gases from the gas turbine, 60% load. The liquid flow rate is determined based on the lowest SRD, Case 1.

The specific reboiler duty is highest for the flue gas flow corresponding to the highest air flow rate, FG 1. As the inlet air flow is reduced further, the SRD decreases. There are two reasons for the decrease in SRD according to the results in chapter 7.2.1; the total flue gas mass flow is reduced, and the CO_2 mole fraction is increased.

However, the total CO_2 mass flow in the flue gas increases as the inlet air flow is reduced. Therefore, the total amount of CO_2 captured is higher, and the total energy demand in the reboiler is almost constant between the regulation methods. The highest energy demand occurs for FG 1, 100,2 MW, and the lowest energy demand occurs in FG 4, 100 MW. This difference is very small.

The difference in mechanical work is more significant. 0,6 MW differs between FG 1 and FG 4. This is in accordance to the reduced flue gas mass flow, since the mechanical work is highly dependent on that flow.

The respective liquid mass flows, that provide the lowest reboiler duty, are low. According to Table 7.8, the liquid flow rates are about 70% of the design flow for each case. The liquid flow rate is greatest for the flue gases representing the highest air flow rate. In the next section, the influence of a restriction on the liquid mass flow is presented.

7.2.3.2 Variation in liquid flow rate

The liquid mass flow into the absorber is regulated in Hysys by an adjuster. The lean CO_2 concentration is varied manually until 90% removal of the CO_2 in the flue gas has been obtained. In Figure 7.23, the simulation results are illustrated.

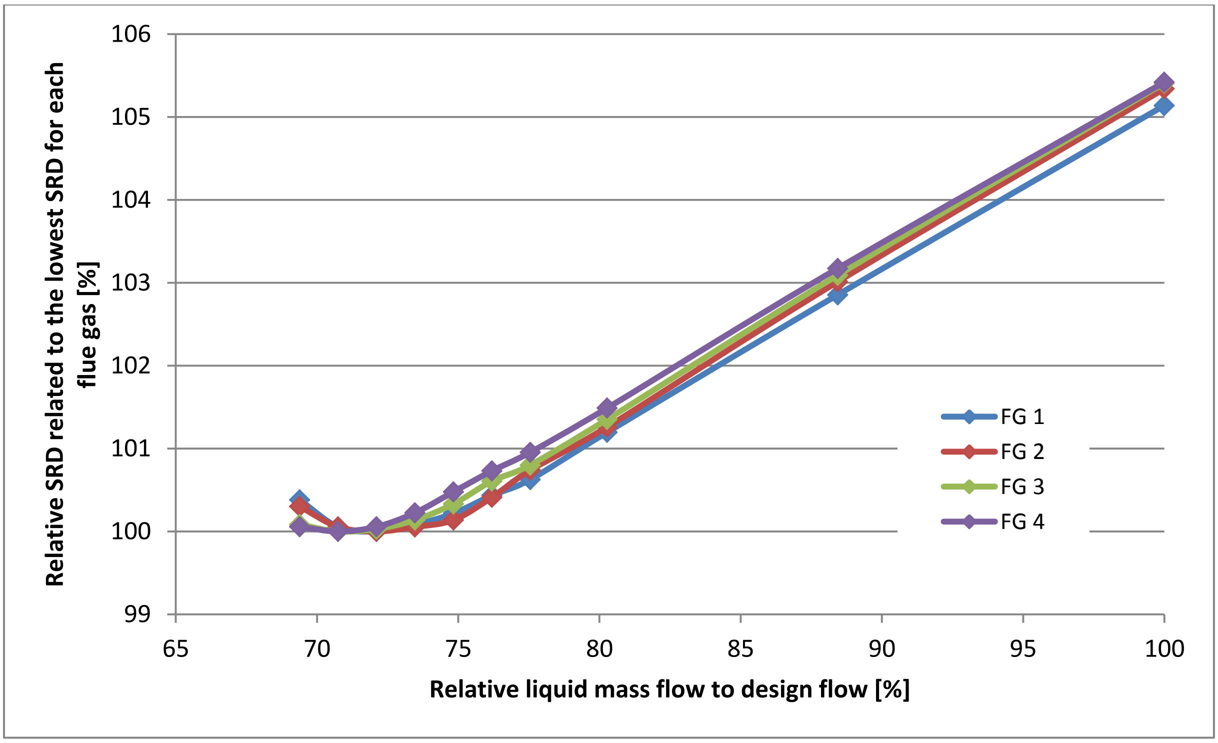

Figure 7.23 – SRD against liquid flow rate in the absorber, for four different flue gas flows.

Figure 7.23 illustrates the relative increase in specific reboiler duty for each flue gas flow, as the liquid mass flow into the absorber is regulated. The shape of the curves is similar to the previous results. The liquid flow rate representing the lowest SRD for each flue gas flow is lowest for FG 4. This is in accordance to Table 7.8.

As the liquid mass flow increases, FG 1 obtains the lowest relative SRD, and FG 4 the highest. Thus, if the liquid mass flow is fixed at a high liquid flow rate, the loss in SRD is most significant for the FG 4. FG 4 is representing the highest exhaust gas temperature for the given gas turbine power output.

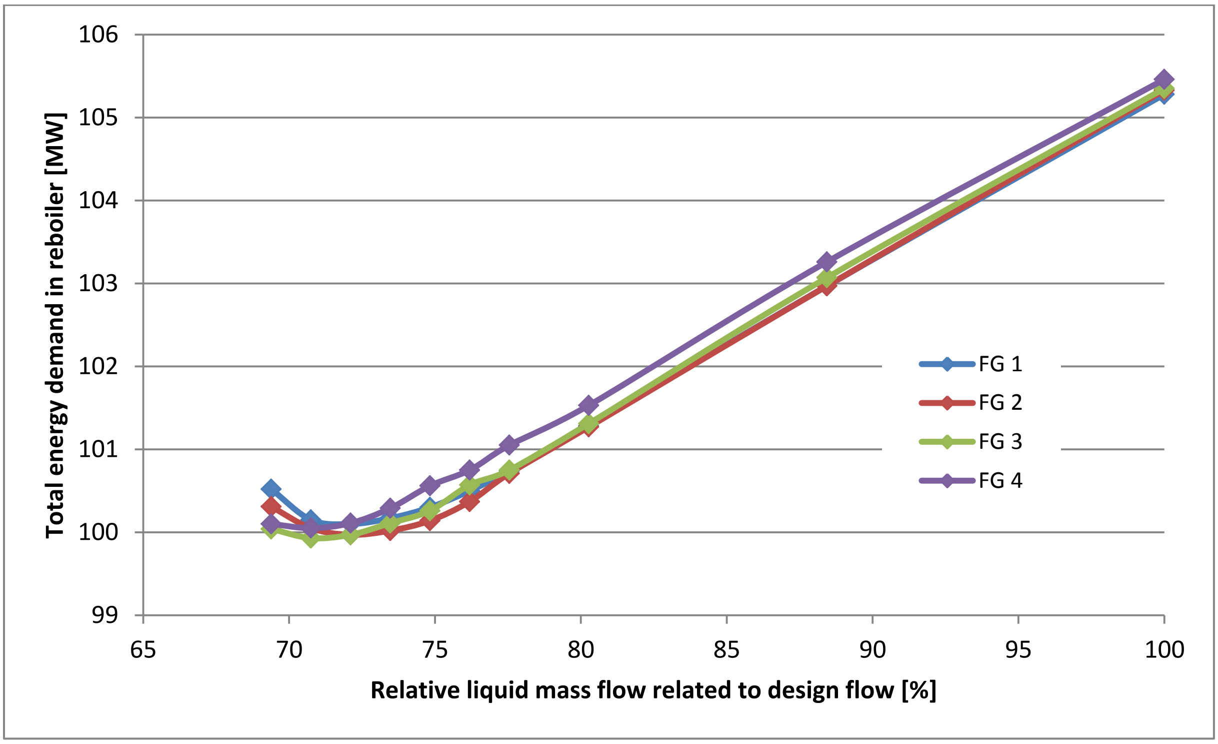

Figure 7.24 – Total energy demand in the reboiler against liquid flow rate in the absorber, for four different flue gases.

In Figure 7.24, the total energy demand in the reboiler is plotted against the liquid mass flow reduction. As the figure illustrates, the difference in total energy demand between the different flue gas flows is marginal for the entire liquid flow range. In Table 7.8, Table 7.9 and Table 7.10, the capture plants operation parameters are given for the three liquid flow rate cases.

Case					
Case		**FG 1**	**FG 2**	**FG 3**	**FG 4**
Flue gas					
Total mass flow	kg/s	533	521	507	474
CO_2 mole percent	%	3,65	3,76	3,89	4,23
Relative air flow	%	81,2	79,4	77,3	72,1
Absorber					
Liquid/gas ratio	$kgmole_L/kgmole_G$	1,38	1,41	1,45	1,56
Inlet liquid flow	kg/s	590	590	590	590
Percent of design liquid flow	%	80	80	80	80
Stripper					
Rich CO_2 concentration	%	5,36	5,38	5,41	5,48
Lean CO_2 concentration	%	2,91	2,92	2,93	2,95
Capture plant performance					
Removal efficiency	%	90	90	90	90
SRD	MJ/kg_{CO2}	3,72	3,7	3,67	3,61
Total reboiler demand	MW	101,3	101,3	101,3	101,5
Mechanical work	MW	6,5	6,3	6,2	5,9
CO_2 compression work	MW	9	9,1	9,1	9,3

Table 7.9 - Capture plant performance with four different flue gases from the gas turbine, 60% load. The liquid flow rate is kept at 80% of the design flow, Case 2.

Case					
Case		**FG 1**	**FG 2**	**FG 3**	**FG 4**
Flue gas					
Total mass flow	kg/s	533	521	507	474
CO_2 mole percent	%	3,65	3,76	3,89	4,23
Relative air flow	%	81,2	79,4	77,3	72,1
Absorber					
Liquid/gas ratio	$kgmole_L/kgmole_G$	1,71	1,75	1,8	1,94
Inlet liquid flow	kg/s	735	735	735	735
Percent of design liquid flow	%	100	100	100	100
Stripper					
Rich CO_2 concentration	%	5,22	5,24	5,27	5,32
Lean CO_2 concentration	%	3,26	3,27	3,29	3,31
Capture plant performance					
Removal efficiency	%	90	90	90	90
SRD	MJ/kg_{CO2}	3,87	3,85	3,82	3,75
Total reboiler demand	MW	105,3	105,3	105,3	105,5
Mechanical work	MW	6,6	6,5	6,4	6,1
CO_2 compression work	MW	9	9,1	9,1	9,3

Table 7.10 - Capture plant performance with four different flue gases from the gas turbine, 60% load. The liquid flow rate is kept constant from the design flow, Case 3.

7.3 Power Plant with CO_2 Capture

Because of the regeneration temperature of the solvent, the reboiler temperature must be fixed. Therefore, the extraction pressure must be constant at 3,5 bar. As the power plant is operated at reduced load, the pressures in the steam cycle are reduced. From Table 7.1 and Table 7.2, the crossover pressure between the IPT and LPT decreases as the load is reduced. The decrease in crossover pressure for the default part-load method in GT MASTER and the part-load control with no regulation of the IGVs are indicated in Figure 7.25

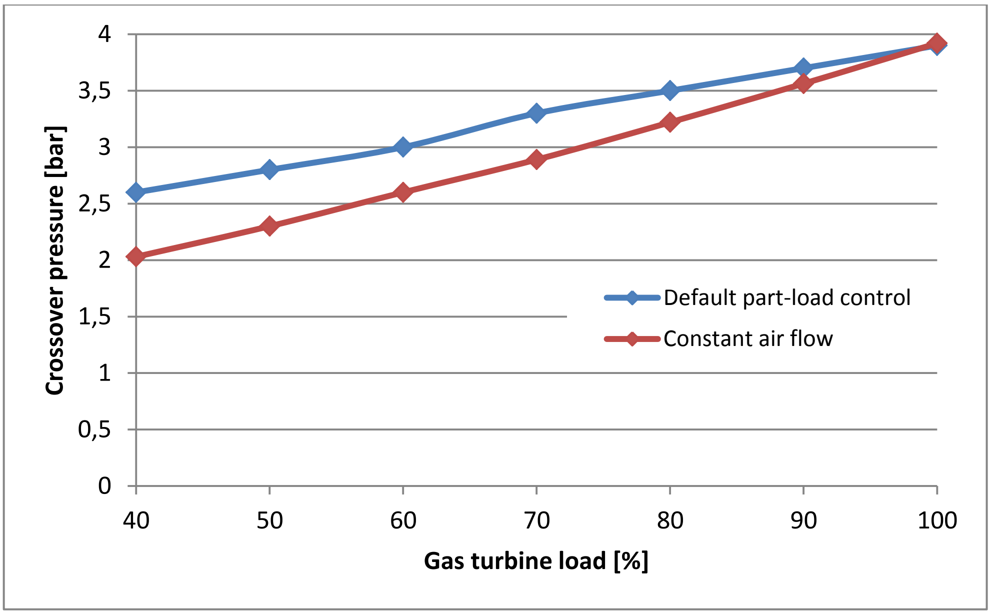

Figure 7.25 – Reduction in crossover pressure for reduced gas turbine load

According to equation 2.26, the pressure is reduced as the mass flow into the turbine is reduced.

$$\frac{\dot{m}_S}{\dot{m}_{S0}} = \sqrt{\frac{p_\alpha \cdot v_{\alpha 0}}{p_{\alpha 0} \cdot v_\alpha}} \qquad 2.26$$

The greatest reduction in crossover pressure occurs for the part-load control with no reduction in inlet air flow.

If a constant extraction pressure of 3,5 bar should be obtained, there must be a throttle valve at the LPT inlet when the load is reduced. Because of pressure losses in the piping, the crossover pressure must be higher than the extraction pressure. At the design point, 12% pressure loss was assumed in the piping between the crossover and the extraction point. When the required steam flow in the reboiler in the capture plant is reduced, the pressure loss in the piping will decrease. GT MASTER is calculating the required crossover pressure in order to deliver the steam at 3,5 bar in the extraction point. According to the theory, the LPT will obtain a lower overall isentropic efficiency because of the isenthalpic expansion in the throttle. Therefore, the gas-turbine control method with no reduction in air flow will obtain more throttle loss compared to the default control method. Because of the extraction pressure requirement there is no other option than to fix the crossover pressure for part-load operation of the total plant.

7.3.1 Default Part-load Method

With a constant extraction pressure the enthalpy difference through the reboiler is assumed equal to the design, and the enthalpy difference is 2170,6 kJ/kg. Thus, the steam extraction mass flow depends only on the total energy demand in the reboiler.

The net plant efficiency from the part-load simulations in GT MASTER are given in Table 7.11.

Load		100	90	80	70	60	50	40
Case 1								
Net plant efficiency, excl. mech/comp loss	%	53,4	52,6	51,7	50,6	49,3	47,6	45,5
Net plant efficiency, incl. mech/comp loss	%	50,6	49,8	48,9	47,8	46,4	44,7	42,5
Case 2								
Net plant efficiency, excl. mech/comp loss	%	53,4	52,6	51,7	50,6	49,2	47,4	45,3
Net plant efficiency, incl. mech/comp loss	%	50,6	49,8	48,9	47,8	46,3	44,5	42,3
Case 3								
Net plant efficiency, excl. mech/comp loss	%	53,4	52,6	51,6	50,4	49	47,3	45
Net plant efficiency, incl. mech/comp loss	%	50,6	49,8	48,8	47,5	46,1	44,3	41,9

Table 7.11 – Net plant efficiency of the NGCC with CO_2 capture at part-load, with default part-load control method. The table includes both net plant efficiency with and without mechanical work and CO_2 compression.

For a given load, there are small differences in the net plant efficiency for different liquid flow rates. At 40% gas turbine load, the efficiency loss is about 0,5 percent points if the liquid mass flow rate is obtained constant from the design, and 0,3 percent points if the liquid flow rate is kept at 80% of its design flow. This is efficiency loss disregarded mechanical and CO_2 compression work in the capture plant. If these are included, the differences in efficiency are almost equal. Figure 7.26 illustrates the variations in efficiency against the gas turbine load and the liquid flow rate constraint.

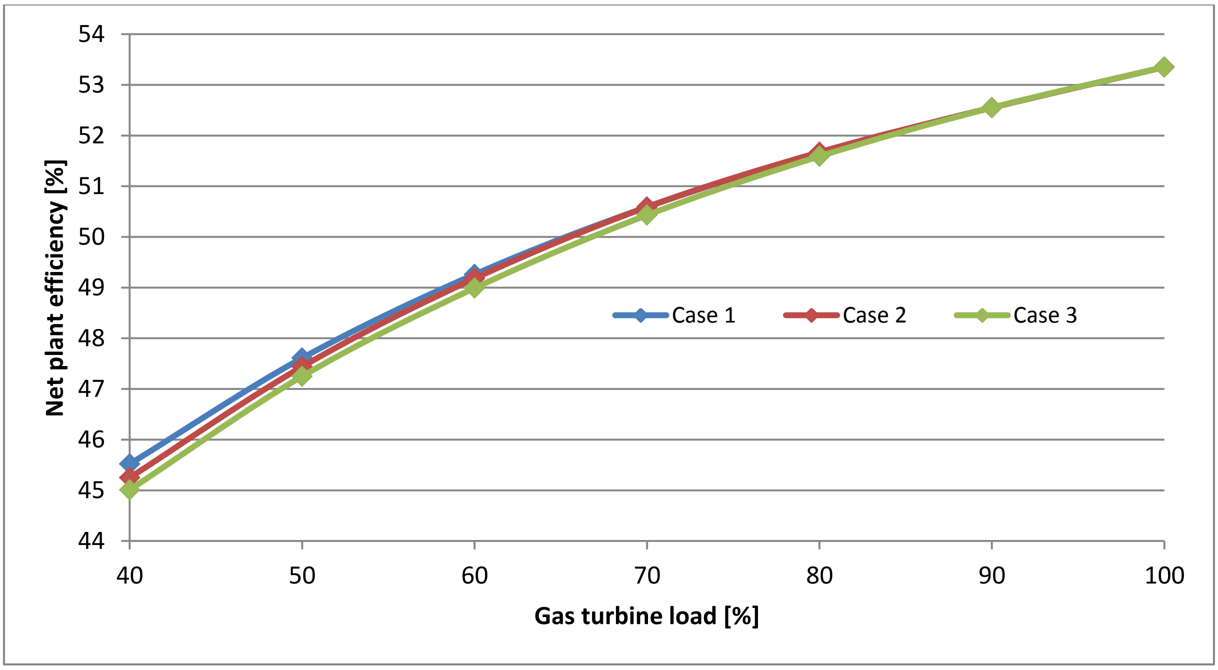

Figure 7.26 – Efficiency of the NGCC plant with CO_2 capture. The mechanical losses and the CO_2 compression are excluded.

7.3.2 IGV Control Methods at 60% Gas Turbine Load

The same restrictions regarding the steam extraction in the last section applies for the different IGV control methods at 60% load. A throttle valve is used at the LPT inlet, in order to obtain a steam extraction pressure of 3,5 bar. The results from the part-load simulations in GT MASTER are given in Table 7.12.

Case		FG 1	FG 2	FG 3	FG 4
Case 1					
Net plant efficiency, excl. mech/comp loss	%	48,6	48,9	49,1	49,5
Net plant efficiency, incl. mech/comp loss	%	45,6	46,0	46,2	46,7
Case 2					
Net plant efficiency, excl. mech/comp loss	%	48,6	48,9	49,1	49,4
Net plant efficiency, incl. mech/comp loss	%	45,6	46,0	46,2	46,6
Case 3					
Net plant efficiency, excl. mech/comp loss	%	48,4	48,7	48,9	49,2
Net plant efficiency, incl. mech/comp loss	%	45,4	45,7	46,0	46,3

Table 7.12 - Net plant efficiency of the NGCC with CO_2 capture. Four different part-load control methods at 60% gas turbine load is used. The table includes both net plant efficiency with and without mechanical work and CO_2 compression.

The loss in net plant efficiency for different liquid flow rates is low for a given flue gas flow, both included- and disregarded the mechanical and CO_2 compression work. If the liquid flow rate is restricted to be minimum 80% of the design flow, there are almost no efficiency losses. However, there are some differences in net plant efficiency between the IGV regulation control methods. In the range simulated, the efficiency is highest for the case with the highest exhaust gas temperature, FG 4.

7.4 Discussion

The results from the part-load operation of the combined cycle were not surprising according to the literature and other studies. It is very important to remain a high exhaust gas temperature when the gas turbine is running at reduced load. Even though the gas turbine efficiency is relative greater for a part-load control method with no, or a little, regulation in the IGVs, the loss in steam cycle efficiency, related to the reduced exhaust gas temperature, is more dominating. For a low TET, the live steam- and reheat temperature is forced down. The effect is a low efficient steam turbine, both because of the inlet temperature and the reduction in vapor fraction at the turbine exit.

There are multiple combinations of fuel reduction and regulation of the IGVs in order to obtain a specific gas turbine power output. Every combinations result in a unique TIT and the subsequent TET. From the present study, it is a tendency of an asymptotic combined cycle efficiency as the TET approaches its design value, for a given gas turbine power output. The gain in the combined cycle efficiency is low as the TET is further increased. However, the efficiency grows slightly. A very high exhaust gas temperature may result in vaporization of water in the economizer. The economizer is designed for a small subcooling at its outlet in order to avoid vaporization. However, as the gas temperature in the HRSG is high, the temperature difference in the economizer is high and vaporization of some of the water may occur. This problem must be regarded in the part-load control method decision.

Also, in order to be within the restrictions regarded emissions of NO_x and CO, the firing temperature should be within a certain range. A high firing temperature enforces the NO_x emissions, while a low temperature could result in emission of CO.

As the results from the present work have indicated, the SRD depends both on the total flue gas mass flow and the CO_2 mole fraction within the flue gas. The SRD decreases with a reduction in the mass flow and with an increase of the CO_2 mole fraction. This relation is in favor of the part-load control method with a high degree of IGV regulations. However, the total CO_2 mass flow increases when the control method with the highest TET is used for a given power output. Therefore, the total energy demand in the reboiler has been shown to be nearly equal regardless of the control method used in the gas turbine. According to the low combined cycle efficiency, the part-load control methods representing very low exhaust gas temperatures have not been considered in the CO_2 capture plant.

The absorber relies on a good liquid/gas distribution in the column in order to work efficiently. When the load is reduced in the gas turbine, and the flue gas flow is reduced, the liquid circulation rate in the capture plant that provides the lowest reboiler duty will decrease from its design rate. The part-load control strategy that provides the highest exhaust gas temperature for a given gas turbine output, are representing the lowest liquid circulation rate. The reduction in liquid mass flow into the absorber is about 30% for a gas turbine load of 60%. This reduction could be destroying for the absorber performance. The packing material in the absorber may not be sufficient wetted, and the liquid/gas contact will be reduced rapidly. In order to avoid these problems, the liquid mass flow should be restricted. According to the results in the present study, the energy demand in the reboiler will grow almost linearly as the liquid flow rate is increased for a given flue gas flow. However, this growth is small.

The increment in reboiler demand is even less significant when steam is extracted from the steam cycle in order to provide the heat to the reboiler. For a gas turbine running at 60% load, with the proposed control method from the manufacture, the lost net plant efficiency is about 0,5 percent points if the liquid flow rate into the absorber is kept constant. This is disregarded th mechanical and CO_2 compression work. The mechanical work in the capture plant has almost no affection on the efficiency loss between the different liquid flow control methods. However, this work depends mostly on the flue gas mass flow and its influence on the efficiency as the load is reduced, is more significant. The efficiency loss regarding a fixed liquid flow rate is decreasing for higher gas turbine loads. With regard to the small efficiency loss, it is recommended to use a restriction on the liquid circulation flow in order to reduce the risk of incomplete wetting in the absorber. The limit will depend on the design capture plant.

It is a trend that for a given gas turbine power output, the part-load control strategy referring to the highest TET will obtain the highest total reboiler duty for a fixed liquid flow rate. However, the difference is marginal and the gain in increased combined cycle efficiency is more dominating for the total plant efficiency.

The steam extracted from the steam turbine is pressure restricted. A throttle valve at the LPT inlet is essential. This throttle valve will result in small losses in the plant efficiency. The losses will increase when the part-load control method related to a low TET is used, because of a reduced crossover pressure in the NGCC plant without steam extraction.

The results in this thesis are based on simulations in Hysys. It is not possible to decide the height of the absorber, something that is essential in the off-design investigation of the capture plant. The number of stages in the column has been set constant. Stage efficiencies are required in order to connect the number of stages to the height of the column. This efficiencies are influenced by factors such as liquid viscosity, gas velocity, gas solubility, contact time, stage design and liquid mixing[19]. A study of the off-design behavior of the CO_2 capture plant with a rate-based model program must be done in order to verify the results in the present work.

However, as the number of stages are kept constant, the results could be used a good indicator on the part-load behavior of the total power plant with CO_2 capture.

8. Conclusion

Based on the results in the current work, the natural gas-fired combined cycle with steam extraction should be operated with an almost constant exhaust gas temperature at part-load mode. The constant exhaust gas temperature is achieved by simultaneously regulation of the inlet guide vanes in the gas turbine compressor, and fuel reduction.

It is recommended to operate with a restriction on the liquid circulation flow rate in the absorber, in order to reduce the risk of insufficient wetting of the packing material. At 40% gas turbine load, the net plant efficiency loss is about 0,5 percent points between a capture plant operated with a constant liquid flow rate, and a capture plant operating with the liquid flow rate that provides the lowest reboiler duty. This is disregarded the mechanical- and CO_2 compression work in the capture plant. The minimum liquid flow rate required will depend on the capture plant design, such as the packing material and the column diameter.

If steam is extracted from the crossover between the low-pressure turbine and the intermediate-pressure turbine, a throttle valve must be included at the low-pressure turbine inlet. The required steam pressure in the extraction point is 3,5 bar, regardless of the part-load operation of the power plant. A throttle valve at the low-pressure turbine inlet must be used in order to fix the extraction pressure.

The mechanical work related to the capture process is strongly dependent of the flue gas mass flow, and less influenced of a change in liquid flow in the absorber.

For a given gas turbine power output, the total energy demand in the reboiler is almost independent of the different part-load control methods. A high turbine exhaust gas temperature is important in order to achieve great combined cycle efficiency. However, as the exhaust gas temperature approaches its design temperature at part-load and the temperature is further increased, the increment of the net combined cycle efficiency is low. In addition, a too high exhaust gas temperature may result in vaporization of water in the economizers in the heat recovery steam generator.

When the gas turbine load is reduced, the liquid circulation rate in the capture plant that provides the lowest reboiler duty will decrease from its design rate. The reduction in liquid mass flow into the absorber is about 30% for a gas turbine load of 60%, with constant exhaust gas temperature. The packing material in the absorber may not be sufficient wetted, and the liquid/gas contact will be reduced rapidly. This could be fatal for the absorber efficiency, especially for MEA as solvent, due to its normally fast reaction rate and equilibrium dependence.

A restriction on the liquid mass flow is recommended based on the results in the previous study.

9. Further work

In the current work, an equilibrium model was used as simulation tool. The equilibrium model is not an off-design program, as the height of the absorber could not be fixed. An analysis on the off-design operation of the absorption process must be done with a rate-based equilibrium model in order to verify the present results.

The removal rate of the CO_2 flow from the flue gas has a great influence on the energy demand in the capture process. A sensitivity analysis on the removal rate should be done for the off-design process. A small reduction in the removal rate may result in significant energy savings in the reboiler. How this removal rate is influenced of the part-load operation should be investigated.

As the total energy demand in the reboiler decreases for reduced gas turbine load, the amount of steam extracted from the steam turbine will be lowered. Thus, the mass flow in the reboiler goes down. A reduced mass flow and a lower heat transfer in the reboiler, will result in a lower pinch temperature. If this temperature difference is significant decreased, the extraction pressure could be reduced. Throttle losses in the steam turbine may be prevented and the steam could be further expanded in the steam turbine. This is an interesting subject for further investigation.

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11. Appendix

11. Appendix

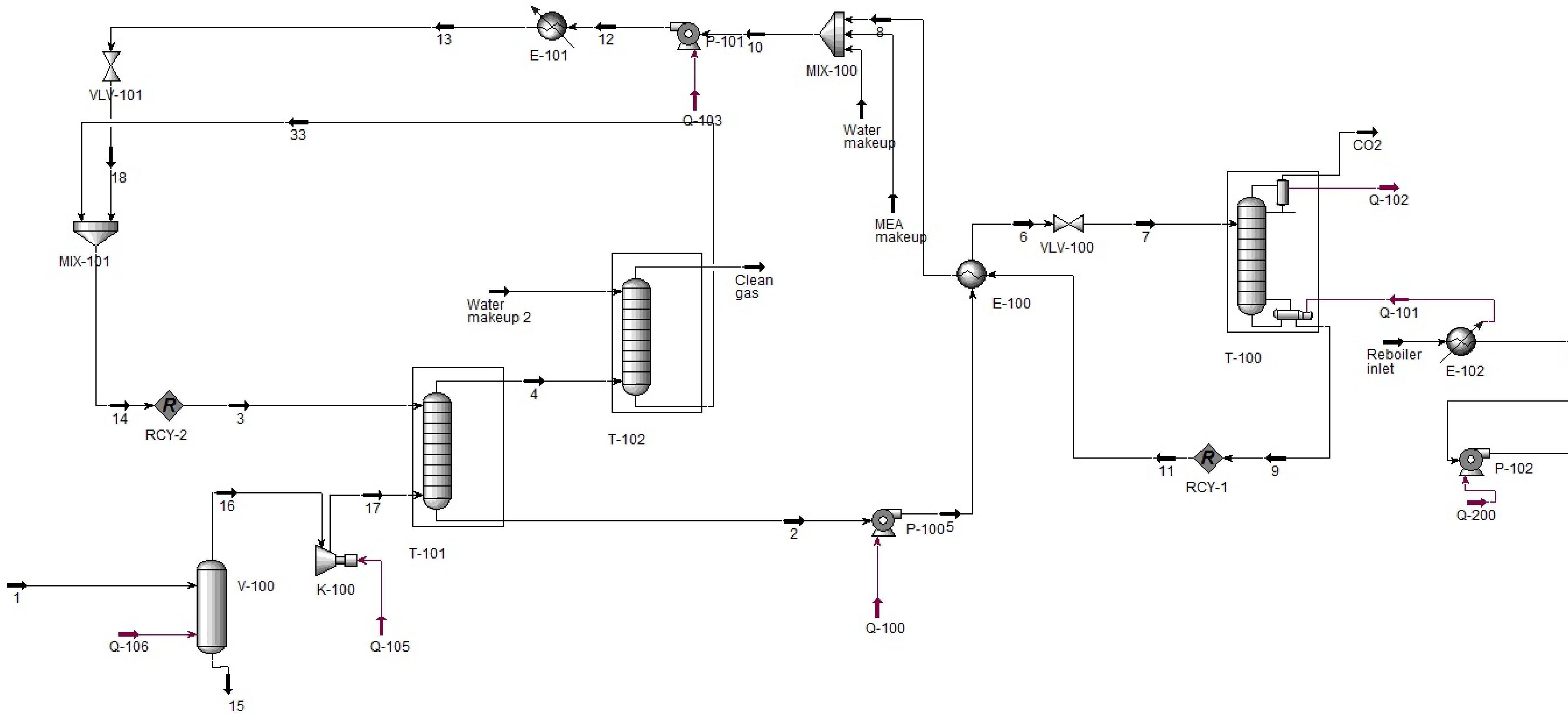

Figure 11.1 – CO_2 capture plant, simulation model in Hysys.

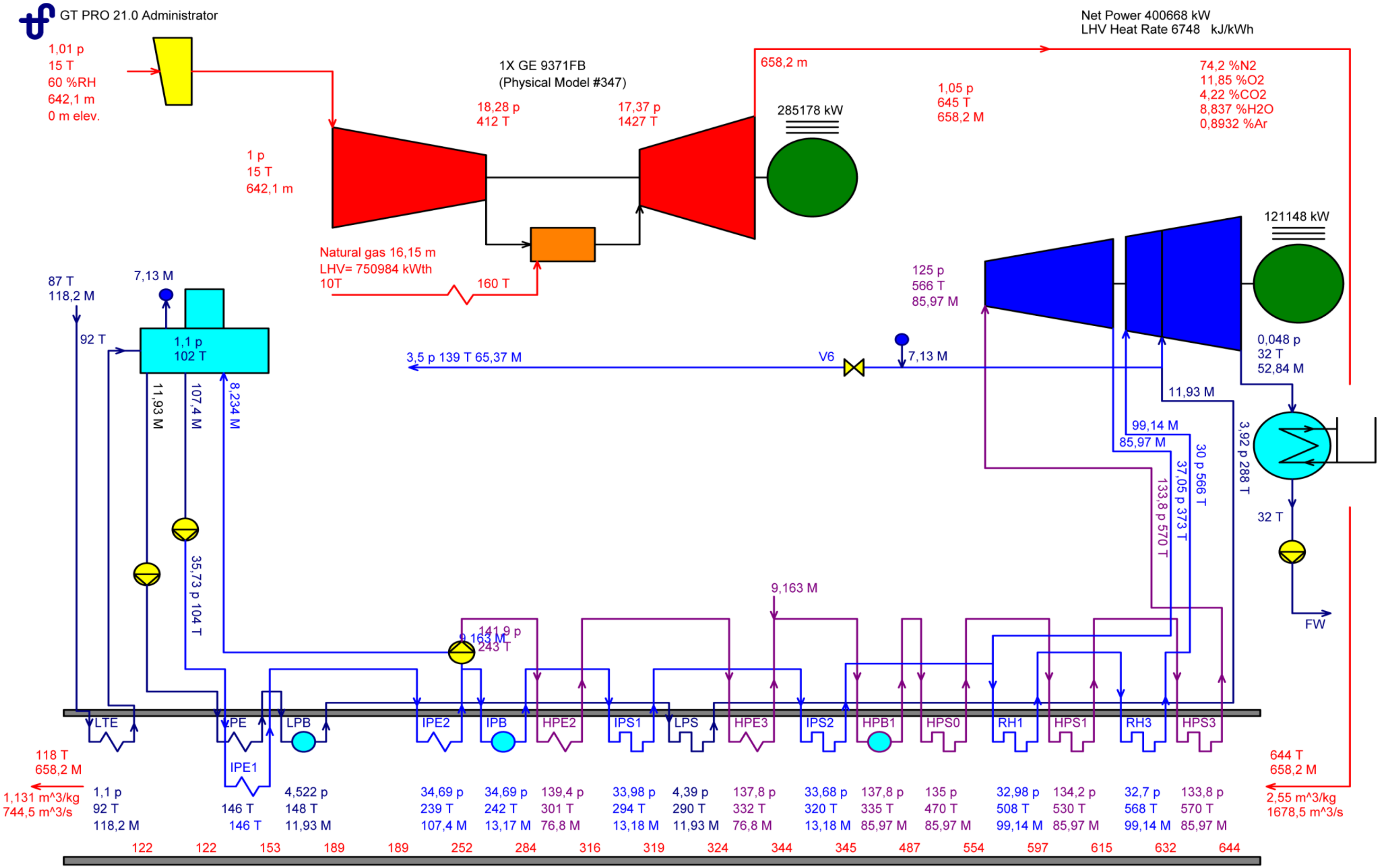

Figure 11.2 – NGCC with CO_2 capture, design case. Cycle flow schematic.

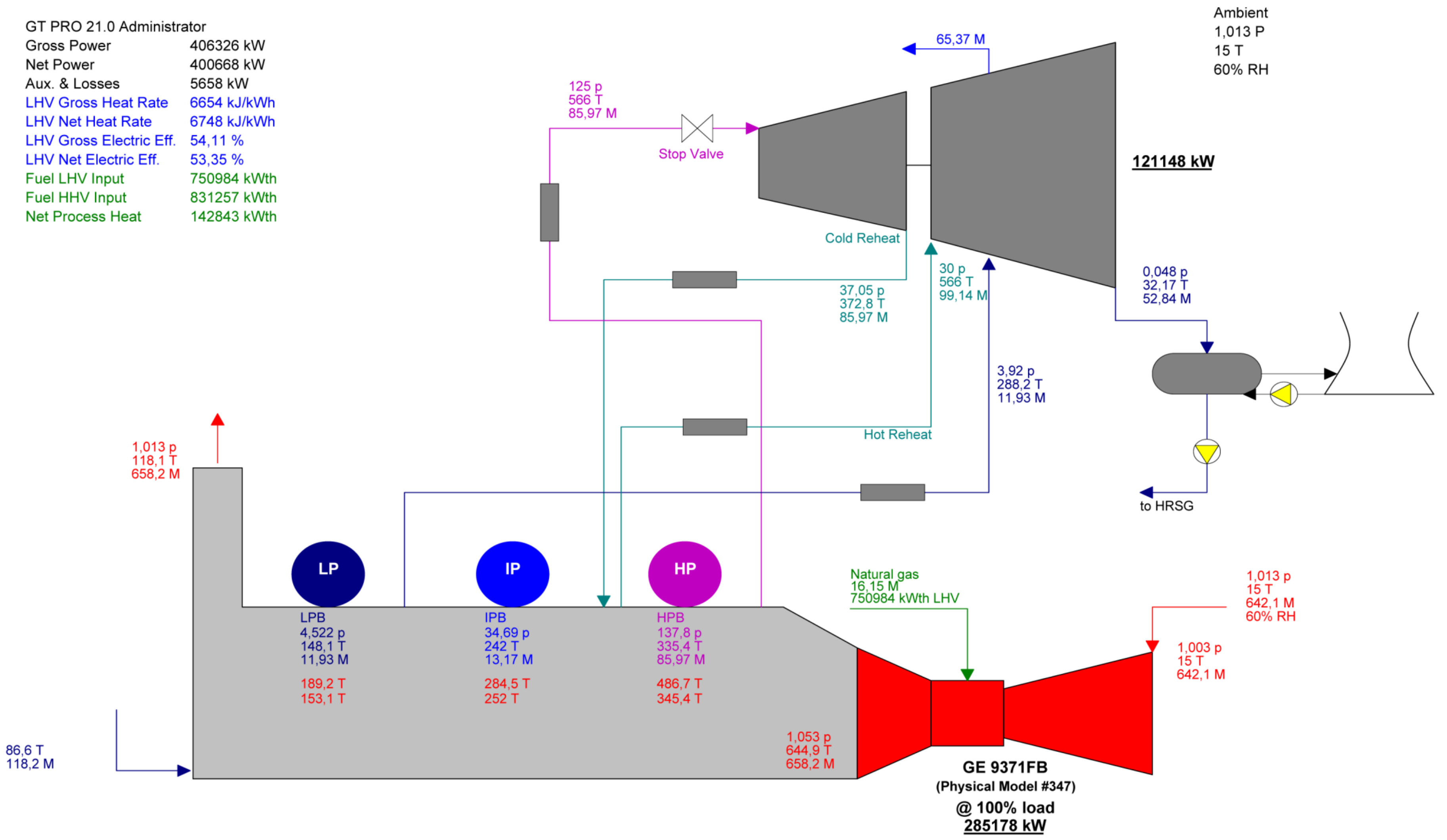

Figure 11.3 - NGCC with CO_2 capture, design case. Plant summary.

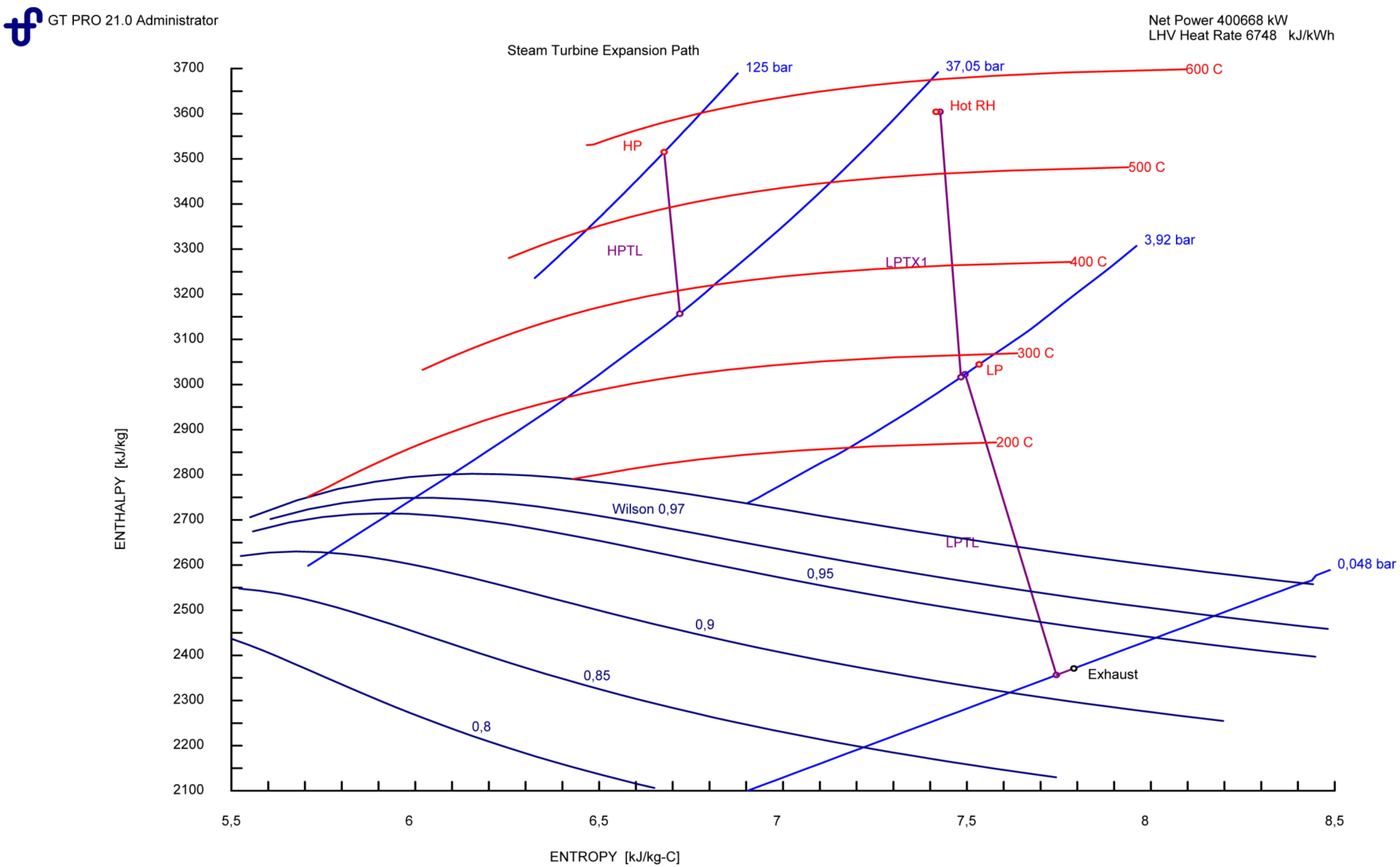

Figure 11.4 - NGCC with CO_2 capture, design case. Steam turbine expansion path.

Derivations

$$x_{MEA} = wt.\%_{MEA}\left(\frac{MW_{tot}}{MW_{MEA}}\right) \tag{11.1}$$

where x_{MEA} is the MEA concentration in the stream

$$\gamma_{MEA} = \frac{x_{CO_2}}{x_{MEA}} \tag{11.2}$$

$$Lean\ CO_2\ concentration = x_{CO_2} \tag{11.3}$$

by combine equation 11.1, equation 11.2 and equation 11.3, the following expression for the lean CO_2 concentration is derived

$$x_{CO_2} = \gamma_{MEA} \cdot wt.\%_{MEA}\left(\frac{MW_{tot}}{MW_{MEA}}\right) \tag{11.4}$$