NTNU
Norwegian University of Science and Technology
Department of Marine Technology

MASTER THESIS

Title: ***Performance assessment of the energy system onboard an LNG FPSO*** *Vurdering av ytelsen til energisystemet på en LNG FPSO*	Delivered: *08.06.2011* Availability: *Limited*
Student: *Stud.techn. Kristian Føring Devik*	Number of pages: 123

Abstract:
LNG power and heat systems, the balance of demand is dependent of the feed gas and on the location of the production vessel. Further, the power and heat balance is dependent on the gas processing efficiencies and the air and water temperatures. Space and weight requirements and environmental factors may impact the concept selection.

Floating LNG liquefaction process may use different driver concepts. The main concepts are:
- *Electrical power produced by onboard power module producing power necessary for all topside and marine systems. The electrical supply could be combined with electrical power from shore.*
- *Gas turbine direct drive of electrical power generators and gas turbine direct drive of liquefaction compressors.*
- *Steam turbine direct drive of electrical power generators and steam turbine direct drive of liquefaction compressors.*
- *Combinations of alternative 2 and 3 in order to achieve optimum use of fuel.*

This master thesis has used the last point as a basis to achieve better fuel utilization and lower the carbon footprint.

Keywords:
FPSO
LNG
COGES

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Problem Description

The main objective of this master thesis is to design and analyze two combined cycle power systems for Höegh LNG's LNG FPSO. The two power systems will differ in complexity and thermal efficiency. The work done in this master thesis is based on the thoughts and conclusions made in the previous project assignment. Both systems consist of gas turbines and a steam cycle for exhaust energy recovery. To keep the thermal efficiency as high as possible in both proposed systems has been emphasized.

The master thesis also includes a discussion and analyzes related to off-design situations (start-up, reduced site rating, unfavorable ambient conditions and so forth). Finally a brief HAZOP study of the proposed systems is evaluated to identify possible hazards, operational difficulties and areas of improvements.

NTNU
The Norwegian University of Science and Technology

Faculty of Engineering Science and Technology
Department of Marine Technology

MASTER THESIS
for
Stud.techn. Kristian Føring Devik
Spring semester 2011

Performance assessment of energy system on board an LNG FPSO

Vurdering av ytelsen til energisystemet på en LNG FPSO

Background
Production of LNG is a very energy intense process. Reduction of fuel consumption is important both for cost and environmental considerations. This project is concerned with exhaust gas energy recovery in a combined gas turbine and steam power system.

Overall Aim and Focus
Preliminary design of combined cycle power systems for an LNG FPSO, with special emphasis on steam systems for exhaust energy recovery. Assessment of transient and off-design operation of the energy system.

The assignment should be prepared based on following points:

Part 1
Design and analysis of the combined cycle power system consisting of gas turbines and steam cycle for exhaust energy recovery. The study should emphasise system capacity and total energy utilization.

Part 2
Discuss and analyse special considerations related to start-up phase and operational states deviating from normal steady-state.

The assignment text must be included as a part of the MSc-report.

The report should be written like a research report, with an abstract, conclusions, contents list, reference list, etc. During preparation of the report it is important that the candidate emphasizes easily understood and well written text. For ease of reading the report should contain adequate references at appropriate places to related text, tables and figures. On evaluation, a lot of weight is put on thorough preparation of results, their clear presentation in the form of tables and/or graphs, and on comprehensive discussion.

All used sources must be completely documented. For textbooks and periodicals, author, title, year, page number and eventually figure number must be specified.

It is assumed that the student should take initiative for establishing satisfactory contact with his teacher and eventual advisors.

In accordance with current regulations NTNU reserves the right to use any results from the project work in connection with teaching, research, publication, or other activities.

Two -2- copies of the report are required. A complete copy of all material on digital form on CD-ROM in Word-format or other relevant format should be handed in together with the written material.

The MSc-report must be delivered no later than June 14, 2011.

Department of Marine Technology, 2011-02-16

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Harald Valland
Professor

Preface

This master thesis is a continuation of the project assignment written during the autumn of 2010. The project assignment gave the writer better insight on different solutions and the overall business, in short acted as a pre study. This study has further revealed several possibilities but also limitations of what are feasible for an offshore power system. Several of the challenges/problems highlighted in the project assignment are addressed in this master thesis and are outlined in the problem description on the former page.

The work has been carried out at the Norwegian University of Science and Technology (NTNU) in collaboration with Höegh LNG. The main perspective has been on off-design situations on proposed energy systems (Heat and Power) for Höegh LNG's first LNG FPSO design.

This work could not get the quality as is it now possess without the help from my supervisor at NTNU, Professor Harald Valland. And not least my industrial contacts at Höegh LNG, Roy Scott Heiersted and Lars Petter Revheim. Valuable and regular feedback has kept me on track, revealing new possibilities and also limitations in the LNG business.

Trondheim, 08.06.2011

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Kristian Føring Devik

Summary

During this master thesis two combined cycle power systems has been designed and analyzed where high energy utilization through recovery of exhaust gas energy has been utilized. Simulation of off-design under ambient and non-ambient conditions has been done to evaluate if a COGES system suitable for an LNG FPSO.

Both power plants consist of four GE LM2500+G4 gas turbines, each directly connected to a compressor and a HRSG (Heat Recovery Steam Generator). Further the produced hot water and steam is utilized in the LNG process trains and to power production through a steam turbine.

The two different combined steam- and gas turbine plants were designed through use of GT PRO software from Thermoflow. The first system is a 3 pressure level system with a max pressure of 69 bar, and condensing steam turbine. The second system is larger and more complex with a 3 pressure level system utilizing a reheat cycle, with a max pressure of 97 bar. Consequently the latter power plant is more powerful and also obtains a higher thermal efficiency.

Simulations have been done in GT PRO and GT MASTER. The results show that the power output from the steam turbine varies little when the gas turbine site rating is from 100 – 60%. Reduced thermal efficiency in the gas turbines is regained in the steam turbine because the gas turbines have to burn more fuel per kWh produced at part load. This is advantageous in an operational setting since no extra generators have to be started at part load operation down to 60% load.

The power plant without reheat has enough power to maintain 100% production and thrusters operating under ambient conditions. Whereas the power plant with reheat can power both the above and also deliver enough power under offloading. From the simulation it is also clear that in unfavorable conditions the power output can vary with 5425 kW (no reheat) and 6076 kW (with reheat). This is considerable, and for the power plant without reheat, extra generation is needed to maintain 100% production under these conditions. Changes in ambient pressure and temperature and cooling water in different combinations have been evaluated.

A basic calculation done by GT PRO estimates an equipment cost of 84 724 000 USD (no reheat) and 86 756 000 USD (with reheat). The power system with reheat is somewhat more expensive but offer better off-design performance. At first eye-sight the difference seems rather small, but it is important to keep in mind that this calculation is only indicative since GT PRO cost estimates are for land based facilities only.

A HAZOP analysis to reveal hazards and challenges with implementation of a COGES system on a LNG FPSO is included. Higher steam pressures introduce new safety hazards and design issues of how to secure personnel. To implement a steam cycle increases the pressure and volume of steam handled. A boiler explosion or leakage could therefore have severe consequences if the system is not designed in a thought-through way. Secondly, implementation of direct drive for the compressors in the process trains raises safety issues in case of gas leakages (Explosion risk if gas gets into inlet when gas turbine is running).

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Nomenclature

Abbreviations

BOG	Boil Off Gas
CC	Combined Cycle
CHP	Combined Heat and Power
COGES	Combined Gas turbine and Electric Steam turbine
CW	Cooling Water
DA	Deaerator
DF	Dual Fuel
DLE	Dry Low Emission (combustion system)
DNV	Det Norske Veritas
FEED	Front End Engineering Design
FLNG	Floating Liquefied Natural Gas
FPSO	Floating Production Storage and Offloading
FW	Feed Water
GT	Gas Turbine
GTL	Gas To Liquid
HAZOP	Hazard and Operability study
HPB	High Pressure Boiler
HPE	High Pressure Economizer
HPS	High Pressure Superheater
HPT	High Pressure Turbine
HRSG	Heat Recovery Steam Generator (Same as WHRU)
HLNG	Höegh LNG
IPB	Intermediate Pressure Boiler
IPE	Intermediate Pressure Economizer
IPS	Intermediate Pressure Superheater
ISO	International Standards Organization
kWe	kilowatts electric
kWth	kilowatts thermal (kJ/s)
LHV	Lower Heating Value
LNG	Liquefied Natural Gas
LPB	Low Pressure Boiler
LPG	Liquefied Petroleum Gas
LPS	Low Pressure Superheater
LTE	Low Temperature Economizer
MMTPA	Million tonnes per annum
NO_x	Nitrous Oxides
Pa	Pascal [N/m^2]
ppm	parts per million
PT	Power Turbine
PURPA	The U.S. “Public Utility Regulatory Policy Act”
RPM	Rotations Per Minute
SAC	Single Annular Combustor

SI	Spark Ignition
ST	Steam Turbine
TET	Turbine Exhaust Temperature
Thermoflow	Thermal Engineering Software
VSD	Variable Speed Drive
WHRU	Waste Heat Recovery Unit

Prefixes

f	femto	10^{-15}
p	pico	10^{-12}
n	nano	10^{-9}
µ	micro	10^{-6}
m	milli	10^{-3}
h	hekto	10^{2}
k	kilo	10^{3}
M	mega	10^{6}
G	giga	10^{9}
T	tera	10^{12}
P	peta	10^{15}

1 Introduction

The assignment for this master thesis is outlined in the start of this document. The project assignment written in the fall semester of 2010 can be regarded as preliminary work to obtain knowledge on the subject, and this master thesis is the continuation where some of the problems mentioned under the master thesis objective will be addressed (stated in full text in Problem description). Part 1 in the master thesis objective has been emphasized in this thesis, as it has been time consuming to learn and utilize the whole Thermoflow program package from scratch to do data simulations on the suggested COGES systems. Help from different skilled persons within Thermoflow and NTNU has saved time, and uncertainties have been clarified faster than what would have been possible without discussion.

The whole simulation process and summary to the results are described in the following chapters. Enclosed are two Excel sheets with all raw materials from the simulations listed.

Part 2 investigates the suggested power systems in a safety- and operational perspective. Increased complexity and in combination with superheated steam could have some safety challenges which will be addressed.

2 Simulation model

To be able to get a precise result what amount of power to expect under stationary production and also at off design situations, the proposed system from the project assignment (Figure A-1) has been modeled in GT PRO from Thermoflow. GT PRO is known to be versatile software for designing gas turbine plants with or without waste heat recovery. Also pure steam plants and a combination of both are possible. The wide range of possibilities for implementation of different process streams made this software a valuable tool for the simulation done in this master thesis.

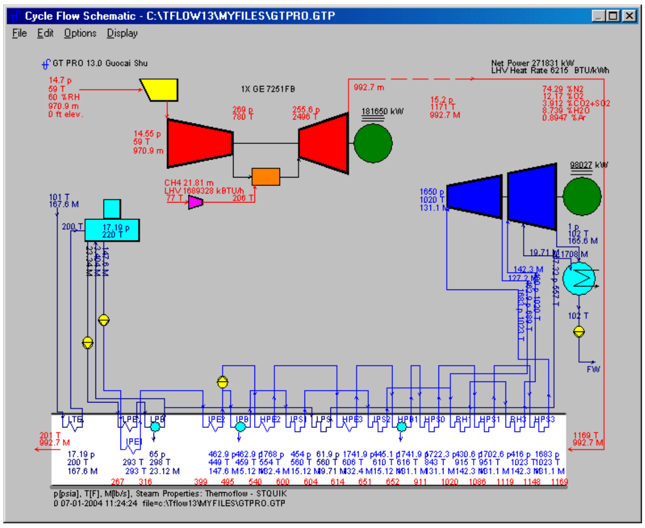

Figure 2-1 Screen picture of GT PRO from Thermoflow. Source: Thermoflow.com

As steam systems have increased efficiency at higher pressures and complexity, two different steam cycles has been designed and proposed for further analysis. Both steam cycles have three different pressure levels, high pressure (HP), intermediate pressure (IP) and low pressure (LP). In addition the steam cycle with the highest power output (and consequently also thermal efficiency) have a dedicated reheat cycle. Beyond that, the cycles have the same characteristics which will be described in detail in the following. To keep the efficiency high, the maximum pressure is automatically set by GT PRO to 69 and 97 bar (for the reheat cycle). This is higher than CB&I have originally proposed in the process trains. The original thought during the project assignment was to implement the process steam and the power generation at the same high pressure cycle. This however proves to be difficult without heavy modification on the process trains (top sides).

In this analysis the problem is avoided by extracting steam to the process from the steam turbine at the correct pressure (Figure 2-2). Extracting process steam directly from the steam turbine is a more energy effective way of reducing the pressure, compared to throttling down the pressure directly before the steam turbine (increased enthalpy loss).

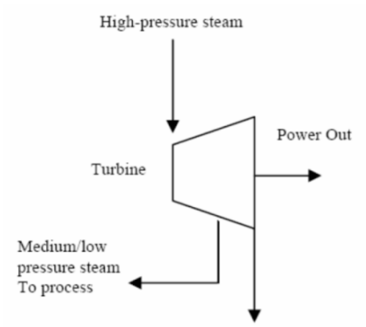

Figure 2-2 Possible way of extracting steam from steam a turbine

Hot process water is taken from the economizer in the HRSGs (previously called WHRU) which has the temperature and pressure closest to the process demand. The same issue exists for the process water as the pressure has to be reduced before it can be used in the process cycle. Fortunately when throttling down the pressure of liquid streams the enthalpy loss is far less compared to throttling gas streams. The main energy waste in this context lies in the pressure rise necessary to get the return flow up to correct pressure again.

2.1 Simplification and assumptions

To model the proposed system, some assumptions and simplifications had to be made to be able to simulate the proposed systems in an effective manner. The assumptions and simplifications done are described in a stepwise manner below. Assumptions regarding generator efficiencies, exhaust pressure loss (inlet pressure loss is set according to Höegh LNG), piping heat- and pressure loss suggested by GT PRO is not changed or modified in any way as these values seem reasonable (and no better data is available). A more detailed analysis of piping distances and heat loss should be done at a later stage, since these distances will be longer than in a regular power plant. The steam turbine will be positioned behind the compartments together with the standby generators.

2.1.1 Cooling water

The LNG FPSO has the possibility to be stationed at different places around the world. As reappearance we can see (Figure 2-3) that the suitable places Höegh LNG have chosen are all in more or less the southern hemisphere. These locations have high relative air temperature and subsequently also high surface water temperature which is unfavorable for the cooling cycle. High temperature of the cooling medium results in larger flow rates, higher condenser pressure, and subsequently lowers the efficiency of the steam cycle.

Figure 2-3 Possible locations for the Höegh LNG FPSO. Source: hoeghlng.com

However by designing a suction pipe to extract cold water from 100 – 400 meters of water depth, this problem could be solved. It is assumed that the water temperature will lie around +4ºC where water has its highest density. This temperature however would of course not be the same at the different locations as Höegh LNG has suggested, but will be basis for the analysis in this master thesis. Careful analysis of the respective deep sea water could be done by collecting updated data from Argo (part of the integrated global observation strategy[1]) which currently have 3214 floats free drifting on the world oceans collecting temperature at different depths.

The allowable cooling water temperature rise is set to max 10ºC. This ensures that the condenser doesn't need a very large cooling surface (compact size) and also keeping the condenser pressure as low as possible (boosts steam turbine output). A power steam cycle could of course also be built without a condenser after the steam turbine, but this would heavily influence the power output. As a result not enough electric power would be produced to operate the LNG FPSO. Even so, as we will establish later on, there is not enough power produced by the steam turbine even with a condenser at certain operational modes (concerns the cycle without reheat).

2.1.2 Steam- and process water

Large quantities of hot water and HP steam are needed to sweeten the natural gas from the well stream. The steam generation system from the process flow diagram to the LNG FPSO (ref [29], page 7) shows in total five streams in and out of the steam drum. To simplify, the streams from the blowdown drum (stream 4415) and the blowdown cooler (stream 4416) are neglected. These flows are very small; 130 and 390 kg/h which amount to 0.58% of the total flowrate (37370 kg/h + 52570 kg/h). The mass flows implemented in Thermoflow are listed in Table 2-1 below.

[1] Argo: http://www.argo.ucsd.edu/

Table 2-1 Process streams in GT PRO

Direction	*Stream*	*Temperature [^oC]*	*Pressure [bara]*	*Flowrate [kg/h]*	*Phase [-]*
To process	Hot water	252	45	37370	Liquid
	Steam	252	41	52570	Steam
Return	Hot water	166	44	37370	Liquid
	Condensate	141	55.6	52570	Liquid

Table 2-1 shows that both return streams have very high return pressures. By default when implementing process streams in GT PRO the process streams are returned to the feed water tank which operates just above ambient pressure. This results in a pressure loss and consequently more pump work has to be added to get the flows up to correct pressure when returned to the HRSGs. The extra pump is not significant (compression of incompressible liquid is not very power consuming compared to compression of gas) but should be revised in collaboration with CB&I (process train designer) when tweaking the system for best performance.

By returning the process flows to the feed water tank, GT PRO ensures that the process water is cleaned and deaerated for each cycle in the process. This is ensured since all fluid from the feed water tank goes through the deaerator before further heating and evaporation again.

The main purpose of a deaerator is to remove air and other dissolved gases in the feed water (and in this case also the return process streams). The deaerator is an important component in the overall design, since external process water could contaminate the return streams during regularly maintenance (when/ if the steam system is opened), by poor material choice in pipes etc. If the dissolved oxygen in the feed water is not removed, serious corrosion damage will occur; rust (oxides) will be formed and further contaminating the feed water. Dissolved CO_2 (Carbon dioxide) can also combine with water and form H_2CO_3 (Carbonic acid) causing increased corrosion. Two basic types of deaerators are commonly used:

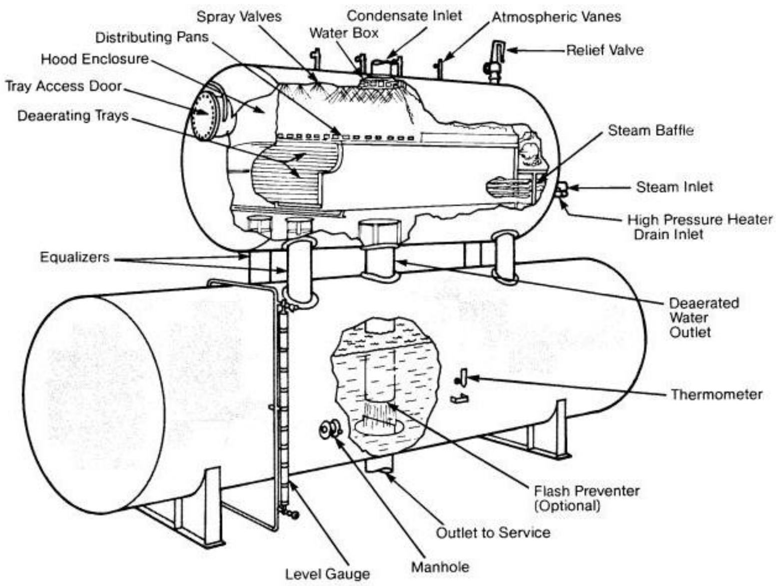

Figure 2-4 Example on a deaerator

- Tray-type (cascade-type): A vertical deaeration section mounted on top of a horizontal cylindrical vessel which serves as the deaerated boiler feed water storage tank.
- Spray-type: A horizontal (possibly vertical) cylindrical vessel which serves both as deaeration section and boiler feed water tank.

It is not made any decision in which type of deaerator would suit the LNG FPSO best at this point.

To see where the deaerator is situated in the steam cycle, please see the high resolution A3 outlines in Appendix E and F for the two proposed systems.

Currently the process system from CB&I and the steam power cycle does not operate at the same max pressures. As mentioned this problem has been avoided by extracting steam at the correct pressure directly from the steam turbine. At a later stage it should be checked if it could be advantageous to have an independent water/steam cycle in the HRSGs for the process cycle, and let the steam turbine operate independently.

2.1.3 Pinch temperature

Pinch temperature is defined as the point of closest approach between the hot and cold side in a heat exchanger. A low pinch temperature needs a large surface area (increased size and cost increases exponentially) but also gives the best heat utilization. Therefore there is always a question of which property to emphasize. Even though the LNG FPSO is a very large vessel, there is still limited space onboard. Therefore the pinch temperature of the HRSGs in GT PRO is set to 20^{o}C for the HP and IP cycle and 10^{o}C for the LP cycle which is rational when low cost and size is emphasized.

Large land based power plants where high energy efficiency are prioritized and with no space limitations can have pinch temperatures of 13^{o}C (HP and IP cycle) and 8.33^{o}C (LP cycle) respectively.

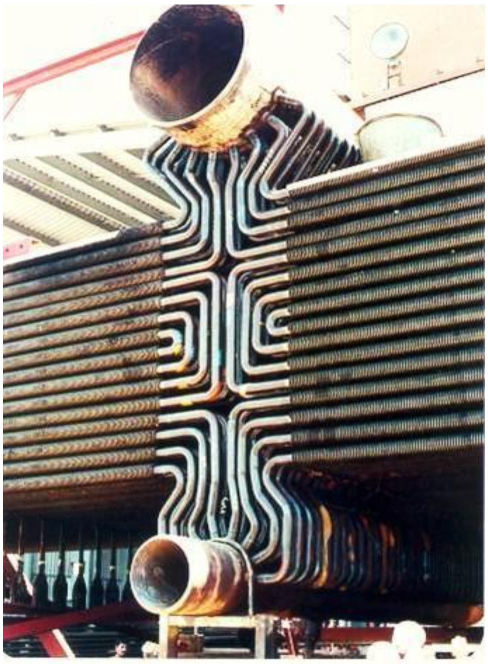

Figure 2-5 Inside of a large HRSG

2.1.4 Condenser pressure

By utilizing extra cold water extracted from deep sea locations, it enables us to reduce the pressure even further compared to using surface water. By default GT PRO has condenser pressure of 0.0689 bar. This pressure can be reduced even further, to 0.015 bar when a use of 4°C cooling water and a maximum temperature increase of 10°C. This considerable pressure reduction in the steam condenser gives an extra boost to the steam turbine, thereby enabling it to produce more power.

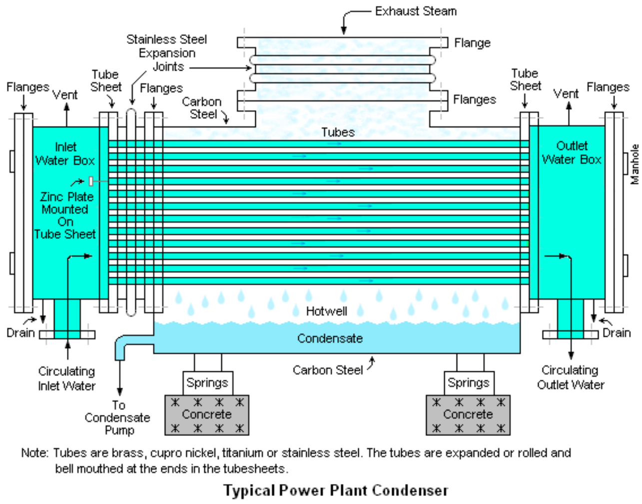

Figure 2-6 Typical layout for stationary condenser

There is a drawback however. Restraining the temperature increase means that more water needs to be pumped through the condenser, increasing the pump work considerable. Even though the pump work increases significantly the gain in power output from the steam turbine is much larger, giving an overall positive effect.

2.1.5 Gas turbine choice

Höegh LNG has currently chosen the LM2500+G4 SAC (Single Annular Combustor) from General Electric to drive the compressors. The notation +G4 tell us that the gas turbine is the fourth generation of the LM2500 series, see ref [31]. Höegh LNG have however expressed that the future may bring stricter discharge levels of pollution, and in that setting DLE (Dry Low Emission) combustion technology can be mandatory. In that context, the LM2500+G4 DLE is the choice for Höegh LNG if necessary in the future. The DLE version of the LM2500+G4 series has slightly lower turbine power output, higher heat rate and therefore also slightly lower thermal efficiency.

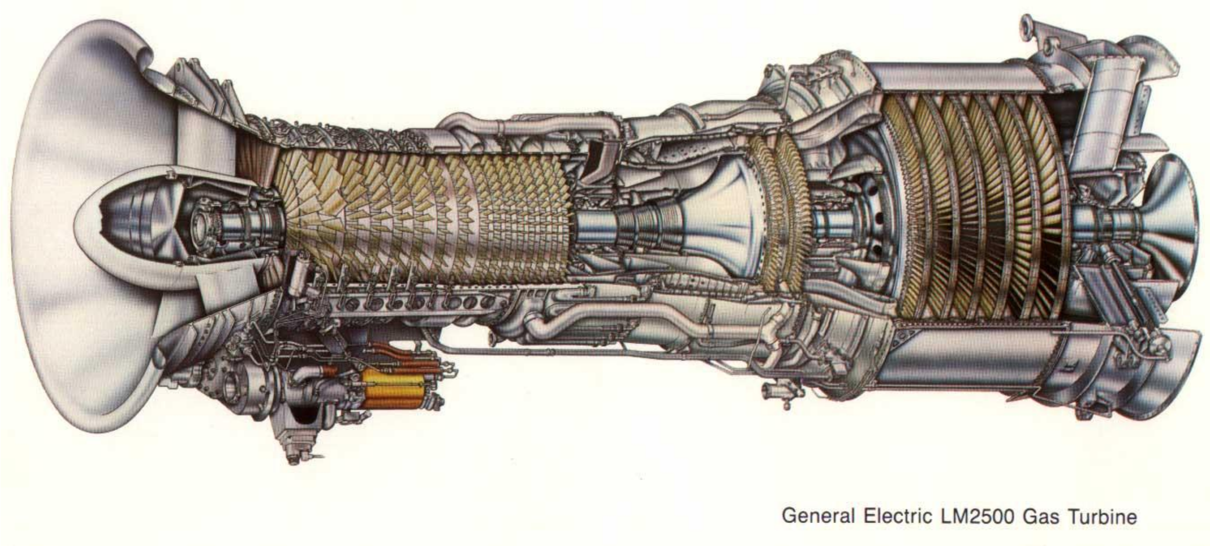

Figure 2-7 Inside view of the General Electric LM2500 Gas turbine

In this master thesis the LM2500+G4 SAC has been chosen as a basis for the simulation. GT PRO has a database with over three hundred GT models from the largest manufacturers. Even so the database is not complete, and the exactly same model as Höegh LNG has chosen has not been found. Virtually all engines in the database are for power generation, and the nominal power output and heat rate shown in the tabulation are defined at the generator terminals, after gearbox loss, and generator loss.

GT PRO assumes the following nominal conditions efficiencies when their nominal power and heat rate is calculated (from chapter 5.2 in the GT PRO help file):

Table 2-2 Assumed nominal condition efficiencies in GT PRO

ISO Rating [MWe]	***Combustion [%]***	***G. T. Mechanical [%]***	***Generator [%]***	***Gear Box [%]***
<2	99.00	98.00	95.00	98.50
2 – 5	99.00	98.00	96.25	98.50
5 – 10	99.15	98.50	97.00	98.50
10 – 20	99.30	98.75	97.50	98.50
20 – 40	99.45	99.00	98.00	98.50
40 – 100	99.60	99.20	98.50	98.50
>100	99.70	99.40	98.60	98.50

Data for Höegh LNG's choice of gas turbine is collected from the Excel sheet "LM2500+G4 SAC and DLE" supplied by Höegh LNG. The data has been adjusted according to the generator efficiency shown in Table 2-3 to be able to compare with GT PRO gas turbine data.

The most resembling gas turbine to Höegh LNG's choice has been chosen from the GT PRO database. Comparison of the turbines has also been done at ISO conditions:

- New, clean engine condition
- At base load, burning pure methane supplied at 25°C
- With ambient air at 15°C at sea level, with 60% relative humidity
- With no pressure losses at the inlet or exhaust

Table 2-3 Performance comparison between GE gas turbines

Performance	***[-]***	***GE LM2500+G4 DLE***	***GE LM2500+G4 SAC***	***GE LM2500+RD (G4)***
Generator efficiency	*%*	0,98	0,98	0,98
PT shaft speed	*RPM*	3600	3600	3600
Turbine power output	*kW*	32541	32771	-
Generator power output	*kWe*	31890	32116	33104
Heat rate	*kJ/kWh*	9288	9266	-
HR at gen. Terminals	*kJ/kWh*	9478	9455	9257
Guaranteed heat rate	*kJ/kWh*	9430	9407	-
Thermal efficiency	*%*	38,77	38,86	38,9
Air flow	*kg/s*	88,74	89,16	90

The General Electric LM2500+RD (+G4) is the best gas turbine comparable to Höegh LNG's option. These turbines have almost the same thermal efficiency and air flow as the Höegh LNG option, but slightly higher generator power output. The combustor is of the DLE type and the power turbine consists of 6 stages (Figure 2-7 Inside view of the General Electric LM2500 Gas turbine).

2.1.6 Inlet- and exhaust pressure loss

Gas turbines are often required to operate in very harsh and unforgiving environments at various installations, where especially offshore structures like Semisubmersibles, FPSOs etc. are common examples in the marine segment.

To operate in these conditions a gas turbine needs state of the art air inlet filter. The ocean atmosphere contains salt aerosols, which is formed at rough seas when small sea water droplets are expelled into the atmosphere. Direct sea spray from severe storms would also reach the inlet of a gas turbine if no air filter is installed.

Salt aerosols (like other particles) can damage a gas turbine through erosion, cooling-path blockage, fouling and corrosion. It is normally hot corrosion that is of most concern, associated with post-combustion sections in the turbine. Molten sulfates are considered to accelerate the attack on turbine materials (also known as sulfidation).

To put it in perspective; salt levels in a typical land-based location are less than 0.008ppm, whereas the offshore equivalent is 0.1ppm, and rising to over 10 ppm during severe storms[2].

Figure 2-8 Up and coming daunting storm offshore. Source: GE ref [33]

Clearly it is essential with a superior air inlet filter. These filters do always have a pressure loss, which is implemented in GT PRO. Höegh LNG has assumed both inlet and exhaust pressure loss which almost agrees with GT PRO's assumption. In the simulation Höegh LNG's estimate is used for the inlet filter, and GT PRO's assumption for the exhaust pressure loss. This is because GT PRO has a more accurate estimate regarding the pressure loss over the HRSGs. However Höegh LNG and GT PRO's estimates are almost similar. Please see Table 2-4 for details (1 mmH_2O = 0.09806648572 mbar).

Table 2-4 Pressure loss estimate GT PRO

Pressure loss	*Höegh [mmH_2O]*	*Höegh [mbar]*	*GT PRO [mbar]*
Inlet	100	9.80665	10
Exhaust	250	24.51662	25

[2] The book of salt from GE, see ref [33]

2.1.7 Combustion machinery on top of LNG storage

As discussed in the project assignment, current regulations for LNG FPSOs from the classification authorities (DNV, BV etc.) clearly states that combustion machinery on top of storage with flammable goods is not allowed. After discussion with Höegh LNG they have expressed that gas turbines over LNG tanks will be allowed in the future (when necessary preventive measures are done).

With this in mind, this master thesis will continue the proposed design with gas turbines located mid ship in the process trains area (over LNG storage).

2.1.8 Properties of HP Fuel Gas

GT PRO allows the user to implement specific what kind of fuel the gas turbines should use. Höegh LNG has four different design cases for different operational situations (ref [19], page 38). The fuel mixture changes slightly when the operational situations deviate because the fuel gas is a mix of BOG and reservoir gas. In this thesis the case 1DLS will be used as basis. Under case 1DLS in short, the liquefaction is 100% and max LPG production. Details on the different holding mode operations (1DLS, 2DLE, 3DLL, 4VHS) is stated in ref [19], page 67.

GT PRO's list of composition of the different alkanes stops at hexane, whereas Höegh LNG's composition also includes heptane and octane. To obtain a total mass distribution of 100% in GT PRO's list, heptane and octane has been summarized as hexane in GT PRO. The error remains small since the mol percentages of the respective substances is very small, and the heating value almost the same.

Table 2-5 Properties of HP Fuel Gas for design case 1DLS

Design Case:		***1DLS***	
Molecular weight		[kg/kmol]	20,371
LHV (ISO-6976[1999]		[MJ/kg]	40,404
HHV (ISO-6976[1999])		[MJ/kg]	44,652
Composition	***Formula***	***[kg/kmol]***	***[mol%]***
Methane	CH_4	16,043	78,046
Ethane	C_2H_6	30,070	4,249
Propane	C_3H_8	44,097	3,590
i-Butane	C_4H_{10}	58,124	0,969
n-Butane	C_4H_{10}	58,124	0,460
i-Pentane	C_5H_{12}	72,151	0,317
n-Pentane	C_5H_{12}	72,151	0,206
n-Hexane	C_6H_{14}	86,178	0,117
n-Heptane	C_7H_{16}	100,205	0,045
n-Octane	C_8H_{18}	114,232	0,019
Nitrogen	N_2	28,013	10,313
Carbon dioxide	CO_2	44,010	1,632
Water	H_20	18,015	0,037
Total			**100,00**

The standard solution in GT PRO is an implemented fuel compressor to be able to deliver fuel gas at the correct pressure. According to Höegh LNG (ref [19], page 38) the fuel pressure from the reservoir is 50 bar. The high pressure makes the fuel compressor unnecessary and we will get a reduction in the auxiliary power losses. The LNG FPSO should however be designed with a standby fuel compressor if the reservoir pressure drops at the end of its lifetime, or if the LNG FPSO is relocated to a low pressure reservoir in the future. Also at transit a fuel compressor is necessary, when fuel gas is taken from storage tanks at ambient pressure.

2.1.9 Steam turbine

GT PRO allows the user to decide how many steam turbines the electric power production should be divided on. The LNG FPSO will for most of its life time operate at 100% stationary production, and therefore should be optimized for that case. As one larger steam turbine is more efficient than two smaller ones, the choice has fallen on one steam turbine (higher efficiency from large scaling). In terms of simplicity and maintenance one steam turbine is also favorable compared to two.

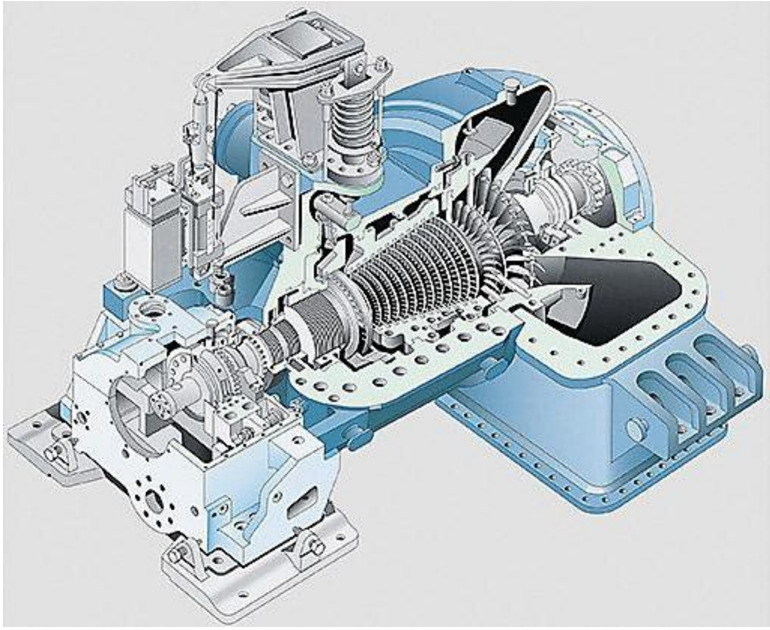

Figure 2-9 Inside view of condensing steam turbine. Source: greenesolpower.com

3 Design point

With the assumptions and simplifications established above the information was implemented into GT PRO and the simulation model developed further. In this chapter the two different models and the simulated data at design point will be presented.

At the design point the process facility operates under stationary conditions at 100% under ambient conditions. This is the case GT PRO calculates first with the given inputs. Under stationary production the steam turbines would produce the largest amount of electric power.

If we look back to the project assignment we had a graph from the Driver Study (ref [16]) which showed the different electric demands for the LNG FPSO at start-up and production situations. The graph is redisplayed again in Figure 3-1.

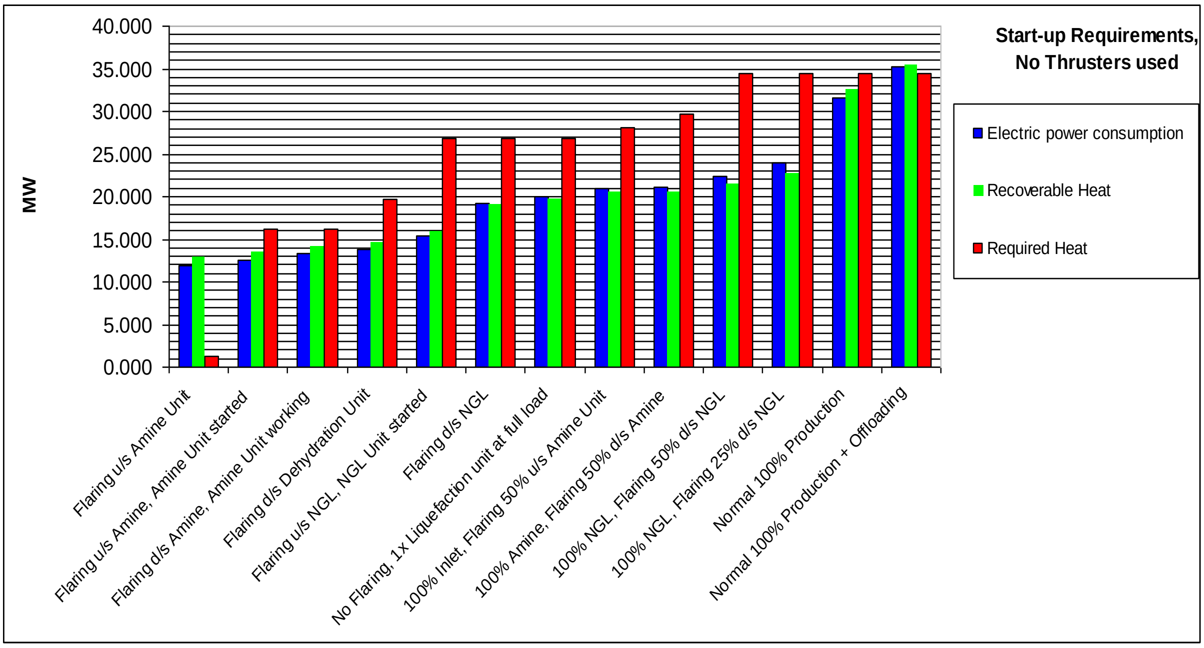

Figure 3-1 Heating and electric power requirements (thrusters not operating) from Driver Study

The highest power demand in this situation is 35.272 MW at “Normal 100% Production + Offloading” and 31.570 MW at “Normal 100% Production” which is the stationary point where the plant would operate most of the time.

With thrusters working we have the following supplementary graph also from the same driver study:

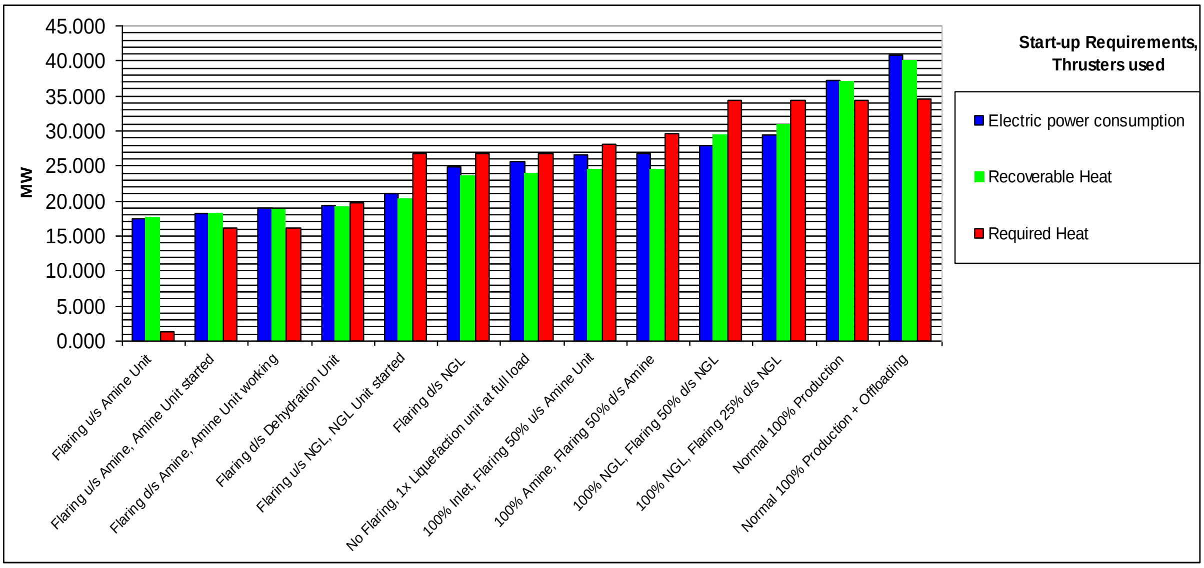

Figure 3-2 Heating and electric power requirements (thrusters operating) from Driver Study

Figure 3-2 shows a maximum electric power consumption of 40.884 MW at “Normal 100% Production + Offloading” and 37.182 MW at “Normal 100% Production”. We can from Figure 3-2 state that with thrusters operating considerable more power has to be produced. The power demands for with and without thrusters working are summarized in the table below:

Table 3-1 Electric power requirements from Driver Study

Situation	***Thrusters operating [MW]***	***Thrusters not operating [MW]***
Normal 100% Production	37,182	31,570
Normal 100% Production + Offloading	40,884	35,272

Power demands when the thrusters are in operation are never stationary due to the continuously changing wind and currents. There are several ways to cover up for the extra power needed when the thrusters are working. Solutions for this issue will be discussed under: Power production at deviating situations and also under: Possible changes or improvements; Steam drum.

In the next subchapters the two proposed steam cycles will be described in more detail.

3.1 Steam cycle with 3 pressures without reheat

A basic view of the steam cycle without reheat generated by GT PRO can be viewed below.

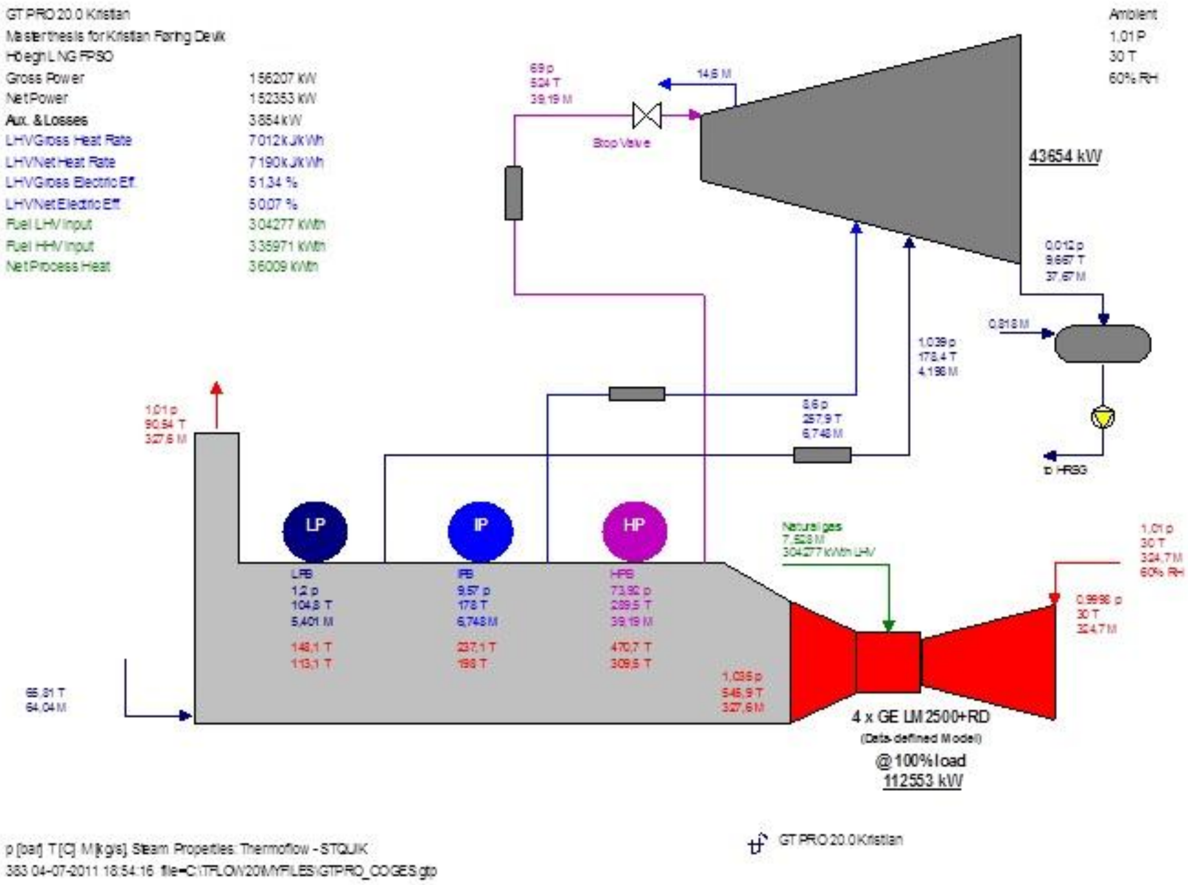

Figure 3-3 Basic outline of steam cycle with 3 pressures and without reheat

For a detailed view of the power cycle, please refer to Appendix E for a high resolution A3 outline. The figure above shows a gross electrical production from the steam turbine of 43654 kW, but net electrical sums up to 39800 kW. The plant has an auxiliary loss of 3854 kW. The four gas turbines in this diagram are in real life directly coupled to gas compressors and would not contribute to any electricity production.

It may look from Figure 3-3 that GT PRO combines the exhaust heat from the four gas turbines in one single HRSG, but this is not the case. Each gas turbine is connected to its own HRSG.

The basic outline tells us that this proposed solution does not produce enough electricity when offloading occurs. In this situation there is an electrical power gap between available and requested power of approx. 1.1 MW (40884 kW – 39800 kW = 1084 kW) which has to be supplied by a supplementary generator or extra firing in the HRSGs.

The extracted process steam flow is shown as a blue line on upper left part of the steam turbine on Figure 3-3. On the high resolution outline (Appendix E) it is further shown that GT PRO has included a water stream from IPE2 (Intermediate Pressure Economizer number 2) for desuperheating to be able to deliver steam to the process at the correct temperature.

Hot water is extracted after the HPE2 (High Pressure Economizer number 2), this can only be seen in the high resolution outline.

Inlet filter, feed water tank, deaerator, condenser, pumps etc. is displaced in the full plant summary in the high resolution outline.

Dedicated A3 outlines for the preliminary HRSG design for the hot- and cold end can be viewed in Appendix G and Appendix I.

GT PRO files which show all details for the different systems (for both without and with reheat described in the next section) are enclosed in a separate CD. Keep in mind that these files only can be viewed with a Thermoflow program package installed on the respective computer. The program package can be borrowed from the university[3] or a 30-day trial version can be requested on Thermoflow's homepage[4]. The enclosed A3 outline files (also displayed the appendixes) can be viewed by GT MASTER and other programs as Autodesk Design Review 2012 which can display DWFX-files.

Table 3-2 displayed below shows a shortened version of the Excel summary produced by GT PRO for the power plant. GT MASTER also produces a similar summary when the simulation is done. Small differences in figures at design point can be displayed, due to different calculation methods for GT PRO and GT MASTER. GT PRO figures are used as a basis throughout this master thesis.

The full versions of the summaries from GT PRO and GT MASTER is displayed in the first work sheet "COGES Summary" in the enclosed Excel Sheet: Off_Design_Calculation.xlsx

Table 3-2 Shortened GT PRO summary for steam cycle without reheat

GT PRO 20.0 Kristian						
383 04-07-2011 18:15:19 file=C:\TFLOW20\MYFILES\GTPRO_COGES_1.GTP						
Master thesis for Kristian Føring Devik						
Höegh LNG FPSO						
Plant Configuration: GT, HRSG, and condensing non-reheat ST						
4 GE LM2500+RD Engines (Data-defined Model), One Steam Turbine, GT PRO Type 8, Subtype 9						
Steam Property Formulation: Thermoflow - STQUIK						
SYSTEM SUMMARY						
	Power Output kW		**LHV Heat Rate kJ/kWh**		**Elect. Eff. LHV%**	
	@ gen. term.	**net**	**@ gen. term.**	**net**	**@ gen. term.**	**net**
Gas Turbine(s)	**112553**		**9732**		**36,99**	
Steam Turbine(s)	**43654**					
Plant Total	**156207**	**152353**	**7012**	**7190**	**51,34**	**50,07**

PLANT EFFICIENCIES			
PURPA efficiency	**CHP (Total) efficiency**	**Power gen. eff. on**	**Canadian Class 43**
%	**%**	**chargeable energy, %**	**Heat Rate, kJ/kWh**
55,99	**61,9**	**57,37**	**6292**

[3] NTNU, Department of Energy and Process Engineering
[4] http://www.thermoflow.com/FreeTrialRequest.asp

GT fuel HHV/LHV ratio =			1,104		
DB fuel HHV/LHV ratio =			1,104		
Total plant fuel HHV heat input / LHV heat input =			1,104		
Fuel HHV chemical energy input (77F/25C) =			335971	kW	
Fuel LHV chemical energy input (77F/25C) =			304277	kW	
Total energy input (chemical LHV + ext. addn.) =			304277	kW	
Energy chargeable to power (93,0% LHV alt. boiler) =			265558	kW	
GAS TURBINE PERFORMANCE - GE LM2500+RD (Data-defined Model)					
	Gross power	Gross LHV	Gross LHV Heat Rate	Exh. flow	Exh. temp.
	output, kW	efficiency, %	kJ/kWh	kg/s	C
per unit	28138	36,99	9732	82	546
Total	112553			328	
Number of gas turbine unit(s) =			4		
Gas turbine load [%] =			100	%	
Fuel chemical HHV (77F/25C) per gas turbine =			83993	kW	
Fuel chemical LHV (77F/25C) per gas turbine =			76069	kW	
STEAM CYCLE PERFORMANCE					
HRSG eff.	Gross power output	Internal gross	Overall	Net process heat output	
%	kW	elect. eff., %	elect. eff., %	kW	
87,71	43654	26,7	23,42	36009	
Number of steam turbine unit(s) =			1		
Fuel chemical HHV (77F/25C) to duct burners =			0	kW	
Fuel chemical LHV (77F/25C) to duct burners =			0	kW	
DB fuel chemical LHV + HRSG inlet sens. heat =			186431	kW	
Net process heat output as % of total output =			19,12	%	

The GT PRO summary shows that over 19% of the heat recovered from the exhaust streams is utilized for different purposes in the process trains (last row in Table 3-2). The summary lists a plant total of 156 207 kW at the generator terminal. Note again that the gas turbines do not in reality deliver electric power but mechanical power directly to compressors.

GT PRO automatically implements an electric loss, so the mechanical output is somewhat higher. It is not possible to do any changes in the program in the current version to set the electric efficiency to 100%. The default value is in GT PRO 97.92% as shown for the gas turbine generator package in Appendix C and D (ref Figure C-1 and Figure D-1).

The overall efficiency of the power plant is 51.34%, a considerable increase from 36.99% for a standalone GE LM2500+RD gas turbine. Keep in mind that the mechanical efficiency will be somewhat higher.

The GT PRO summary also produces a simplified cost estimate for the major equipment costs in k USD = 1000 USD. This estimate has to be regarded as a guideline only, since this is by default for land based facilities. However the cost estimate gives us a clue where the major expenses are located. The specific cost [USD/kW] gives us a good indication of how advanced the systems are compared to another.

Table 3-3 Very simplified cost estimate for steam cycle without reheat

VERY SIMPLIFIED ESTIMATE OF MAJOR EQUIPMENT COSTS (k USD)							
	GT/Gen.	HRSG	ST/Gen.	Condenser	CT	Others*	Total
Cost modifier	1	1	1	1	1		
Unit cost	11000	5070	15245	2877,6	0	2320,1	
No of units	4	4	1	1	0	1	
Sums	44000	20281	15245	2877,6	0	2320,1	84724
* Others include fuel handling system, and water treatment & makeup system.							
Specific cost of above components (USD per net kW) =					556,1	USD/kW	
Estimate of total project costs (based on user-defined capital cost multiplier = 2,25)					190628	kUSD	
Estimate of project specific cost (USD per net kW) =					1251,2	USD/kW	

3.2 Steam cycle with 3 pressures and reheat

By introducing reheat in the steam turbine we get in total four different steam cycles. This increases the power output, but also the space, complexity and the overall price tag of the system. In addition this system does also have more comprehensive HRSGs with increased number of economizers, pressure boilers and superheaters. Further will this demand a considerable amount of space, and could pose a problem for the overall design. In short; this option looks and operates more like a land based facility.

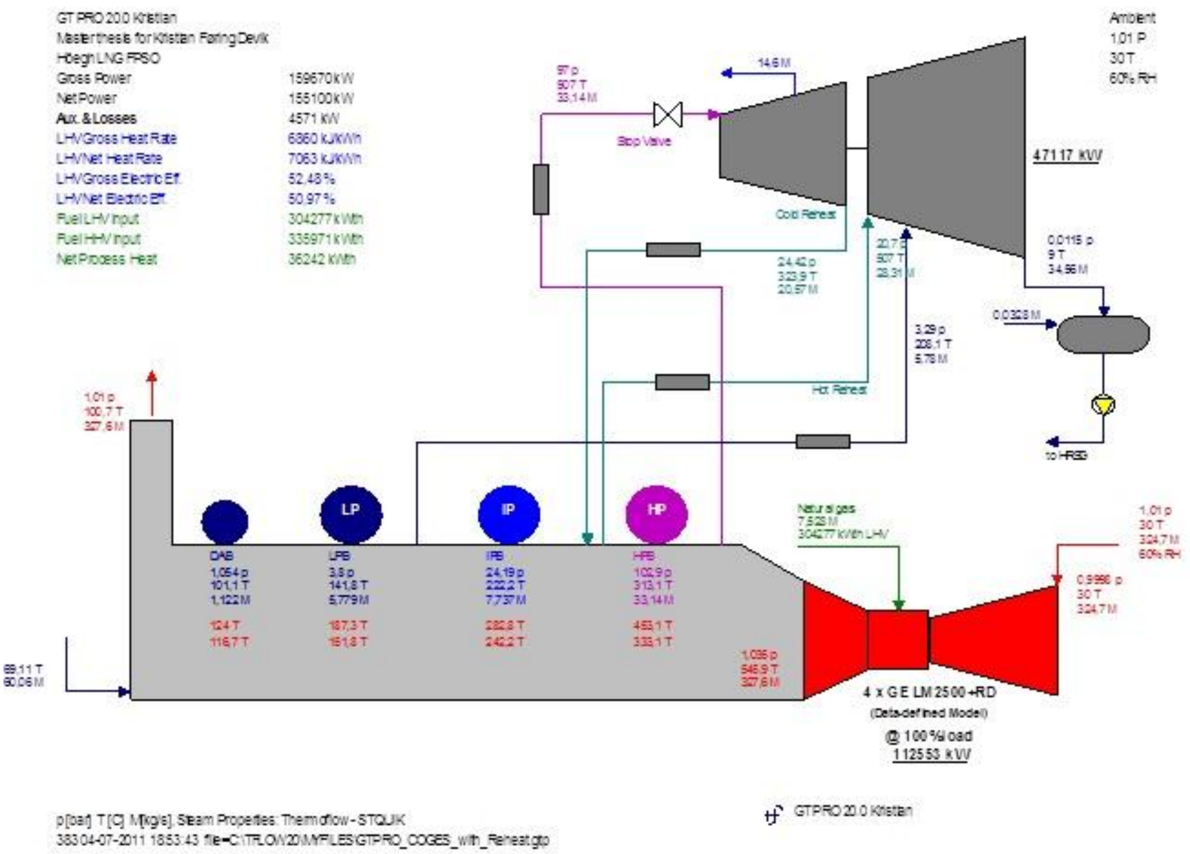

Figure 3-4 Basic outline of steam cycle with 3 pressures and reheat

The reheat steam cycle in Figure 3-4 have an output of gross 47117 kW, with the auxiliary losses of 4571 kW accounted for the cycle delivers net 42546 kW. This is more than enough to power the LNG FPSO at all production scenarios. An excess power delivered of 1662 kW could be used to boost one of the LNG compressors or other rotating equipment. Another solution is to reduce power output on the gas turbines and further achieve equilibrium between power demand and production. Reduced LNG production is however unfavorable in many situations since this directly reduces the income (less LNG to sell to customers). An evaluation between the current LNG price and reduced fuel costs due to reduced engine load has to be done in such a case.

A detailed view of the power cycle is shown in Appendix F as a high resolution A3 outline. In addition as for the steam cycle without reheat, A3 outlines of the preliminary HRSG design for the hot- and cold end can be viewed in Appendix H and Appendix J.

A shortened summary from GT PRO for the reheat cycle is shown in Table 3-4. It is evident that this cycle is more fuel efficient and in addition delivers a higher power output. Still this configuration are only 1.14% (52.48% - 51.34%) more efficient if we compare the electric efficiency. However, this overall improvement as stated previously is enough to deliver power for all requirements.

It should be noted that the steam cycle improvement is larger; 1.85%.

The full versions of the summaries from GT PRO (and also from GT MASTER) is displayed in the first work sheet "COGES Summary" in the enclosed Excel Sheet: Off_Design_Calculation_REHEAT.xlsx

Table 3-4 Shortened GT PRO summary for steam cycle with reheat

GT PRO 20.0 Kristian						
383 04-07-2011 19:59:55 file=C:\TFLOW20\MYFILES\GTPRO_COGES_WITH_REHEAT.GTP						
Master thesis for Kristian Føring Devik						
Höegh LNG FPSO						
Plant Configuration: GT, HRSG, and condensing reheat ST						
4 GE LM2500+RD Engines (Data-defined Model), One Steam Turbine, GT PRO Type 9, Subtype 3						
Steam Property Formulation: Thermoflow - STQUIK						
SYSTEM SUMMARY						
	Power Output kW		**LHV Heat Rate kJ/kWh**		**Elect. Eff. LHV%**	
	@ gen. term.	**net**	**@ gen. term.**	**net**	**@ gen. term.**	**net**
Gas Turbine(s)	**112553**		**9732**		**36,99**	
Steam Turbine(s)	**47117**					
Plant Total	**159671**	**155100**	**6860**	**7063**	**52,48**	**50,97**
PLANT EFFICIENCIES						
PURPA efficiency	**CHP (Total) efficiency**		**Power gen. eff. on**		**Canadian Class 43**	
%	**%**		**chargeable energy, %**		**Heat Rate, kJ/kWh**	
56,93	**62,88**		**58,46**		**6174**	
GT fuel HHV/LHV ratio =			1,104			
DB fuel HHV/LHV ratio =			1,104			
Total plant fuel HHV heat input / LHV heat input =			1,104			
Fuel HHV chemical energy input (77F/25C) =			335971	kW		
Fuel LHV chemical energy input (77F/25C) =			304277	kW		
Total energy input (chemical LHV + ext. addn.) =			304277	kW		
Energy chargeable to power (93,0% LHV alt. boiler) =			265307	kW		
GAS TURBINE PERFORMANCE - GE LM2500+RD (Data-defined Model)						
	Gross power	**Gross LHV**	**Gross LHV Heat Rate**	**Exh. flow**	**Exh. temp.**	
	output, kW	**efficiency, %**	**kJ/kWh**	**kg/s**	**C**	
per unit	**28138**	**36,99**	**9732**	**82**	**546**	
Total	**112553**			**328**		
Number of gas turbine unit(s) =			4			

Gas turbine load [%] =	100	%
Fuel chemical HHV (77F/25C) per gas turbine =	83993	kW
Fuel chemical LHV (77F/25C) per gas turbine =	76069	kW

STEAM CYCLE PERFORMANCE				
HRSG eff.	Gross power output	Internal gross	Overall	Net process heat output
%	kW	elect. eff., %	elect. eff., %	kW
86,24	47117	29,31	25,27	36242

Number of steam turbine unit(s) =	1	
Fuel chemical HHV (77F/25C) to duct burners =	0	kW
Fuel chemical LHV (77F/25C) to duct burners =	0	kW
DB fuel chemical LHV + HRSG inlet sens. heat =	186431	kW
Net process heat output as % of total output =	18,94	%

Table 3-4 shows at first eyesight a more efficient system and also higher output. This is reasonable when looking at the modifications done in comparison with the system without reheat cycle.

Table 3-5 tells us that the price tag of this power plant compared to the one above without reheat is somewhat more expensive in components cost. If we look on the specific cost value this has also increased, but not considerable. The additional cost by implementation of reheat and more comprehensive HRSGs is not considerable. However space requirements and complexity could be decisive factors when a choice between the two proposed systems has to be taken.

Table 3-5 Very simplified cost estimate for steam cycle with reheat

VERY SIMPLIFIED ESTIMATE OF MAJOR EQUIPMENT COSTS (k USD)							
	GT/Gen.	HRSG	ST/Gen.	Condenser	CT	Others*	Total
Cost modifier	1	1	1	1	1		
Unit cost	11000	5582	16359	1771,8	0	2297	
No of units	4	4	1	1	0	1	
Sums	44000	22328	16359	1771,8	0	2297	86756

* Others include fuel handling system, and water treatment & makeup system.		
Specific cost of above components (USD per net kW) =	559,4	USD/kW
Estimate of total project costs (based on user-defined capital cost multiplier = 2,25)	195200	kUSD
Estimate of project specific cost (USD per net kW) =	1258,5	USD/kW

3.3 Figures and graphs

The Appendix C and D list several figures produced by GT PRO with relevant data describing the power plant for both the power plant without- (Appendix C) and with reheat (Appendix D). A general description of the different figures in the appendixes is outlined below. The same figures are displayed in both appendixes with different numerical values.

3.3.1 Gas turbine model

After simulation completion GT PRO lists all the relevant data for the selected gas turbine and the relevant inputs given. Pressure loss in inlet filter and the HRSG, fuel gas pressure etc. are also listed in these figures (ref Figure C-1 and Figure D-1).

The gas turbine model also displays that the generator side has an electric loss. As mentioned, there has been no success in setting this value to zero in GT PRO, simulating that this in real life is linked mechanically to a compressor.

3.3.2 HRSG temperature profile

The HRSG temperature profile displays how efficient the exhaust gas energy is transferred to the heating medium. From the profile figures (ref Figure C-2 and Figure D-2) we can see that the temperature profile for the reheat cycle is located closer to the exhaust gas temperature (red line) as it's going thru the HRSGs. This property has to do with larger surface areas exposed to the exhaust heat and water cycle. The location of the different economizers, boilers and superheaters is also shown in the profiles. The HRSGs to the steam cycle with reheat are more complex and consequently containing an increased number of elements.

3.3.3 Steam turbine group data

GT PRO designs the steam turbine in the best possible way which is practically feasible in a commercial sense. Detailed figures (ref Figure C-3 and Figure D-3) for the different casings with changing number of stages, efficiencies and exit pressure is outlined.

It is also shown where the HP extraction to the process requirement takes place. Note that GT PRO also designs the steam turbine in such a way that it will fit with the process needs (process steam is extracted between the casings). In addition GT PRO implements bleeds/leakages which is used to correct temperature of the extraction of the hot process water.

The difference between the steam cycle with and without reheat is clearly visible. This can be seen as the cycle with reheat redirect steam after two HP casings out and sends it back to the HRSG for reheating.

3.3.4 Steam turbine expansion path

To show how the steam travels in the steam turbine, GT PRO utilizes an enthalpy-entropy diagram (also called a Mollier diagram). Lines of constant temperature and pressure are shown in the diagram (ref Figure C-4 and Figure D-4). Further GT PRO has drawn vertical lines which show the turbine expansion path. The lines showing the steam paths are not completely vertical, due to entropy losses which are inevitable (heat transfer between the turbine and its surroundings and kinetic and potential energy effects).

It can be seen in the diagrams that the reheat cycle has a “fracture” in the steam path. This is due to the reheating of the steam.

3.3.5 Steam turbine exhaust loss

Steam leaving the last expansion stage of a condensing steam turbine can carry a considerable amount of useful power to the condenser as kinetic energy. Dependent on how big the exhaust area is, some of this energy is lost. The size of the exhaust area is a balance between exhaust loss and investments cost in steam turbine equipment.

The lowest point on the unbroken line is the thermodynamic optimum. Historically it has not been economically justifiable invest in sufficient exhaust area to operate at 100% rating at the bottom to the unbroken line. Further by sizing the turbine in such a way can result in excessively low V_{AN} (and consequently high exhaust loss) at part load. But as today's society demands higher thermal efficiencies, the trend has been to move the economic optimum towards the thermodynamic optimum.

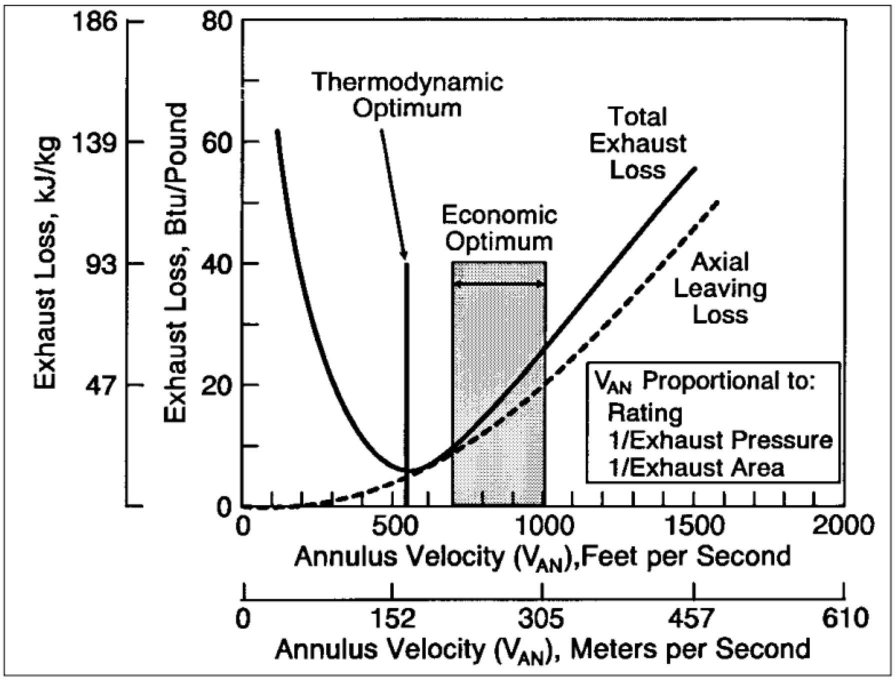

Figure 3-5 Illustrative exhaust loss curve. Source: gepower.com, ref [37]

Figure 3-5 shows exhaust loss (in specific energy) plotted versus the steam velocity passing through the exhaust annulus V_{AN}. Leaving loss (kinetic energy carried by the exhaust flow) is shown by the dashed curve. During low velocities the total exhaust loss is far greater than the axial leaving loss part (this is due to internal off-design inefficiency and off-angle effects).

Common steam turbine designs choose to lay the operation at intermediate annulus velocities (150-300 m/s). Above 300 m/s other losses come into play. GT PRO has chosen in excess 250 m/s to keep the exhaust area relatively small (ref Figure C-5 and Figure D-5). The operating point is marked with a cross sign, and is more or less the same for both the suggested steam cycles.

To calculate the annulus velocity the continuity equation is used $V_{AN}=Q/A$. A is the exhaust annulus area and Q is the volume steam flow.

3.3.6 Cooling system

By default the GT PRO user can choose between several different types of cooling equipment. Naturally, water cooling has been chosen on the LNG FPSO since there are infinite amounts nearby. In addition, as mentioned previously, extraction of cold deep water will be utilized. This makes the cooling process more effective, and also lower condenser pressures are attainable.

Temperature rise of the cooling water is also restrained to be able to keep the condenser pressure to a minimum. This further result in rather large pump work, respectively 1030 kW (without reheat) and 1272 kW (with reheat). The extra auxiliary loss is regained in the increased steam turbine power output.

All relevant data for pressures, flows and temperatures etc. are stated in Figure C-6 and Figure D-6 in Appendix C and D. Figure 3-6 shows two different types of uptake systems. The solution on the left is an inboard caisson type, whereas the right one is the outboard version.

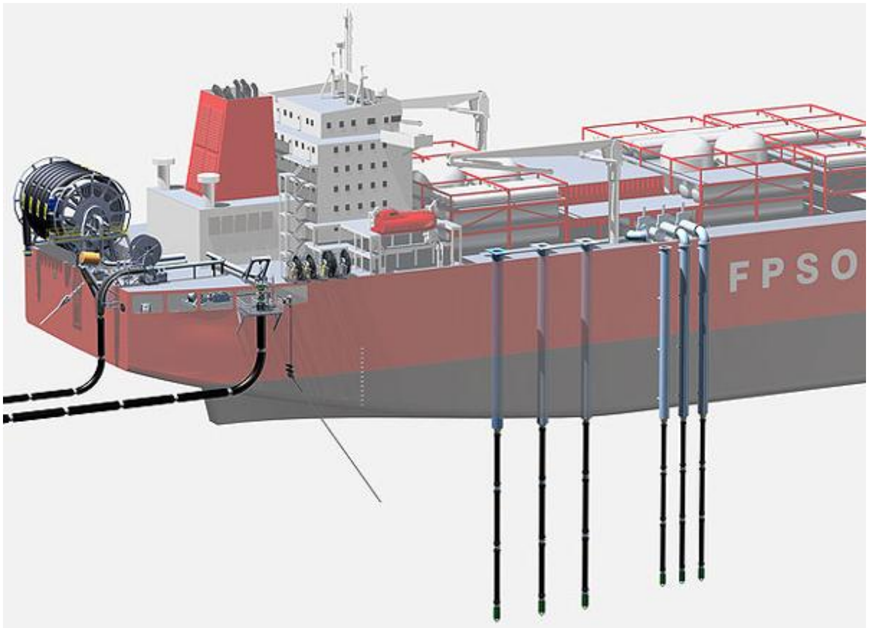

Figure 3-6 Deep seawater uptake systems. Source: emstec.net

3.3.7 Water cooled condenser T-Q diagram

The T-Q diagrams display the heat transfer from the condensing steam to the cooling water versus the water and steam temperature. Inbound and outgoing cooling temperature, and also the heat transfer at the respective temperature. The actual temperature rise is governed by GT PRO, but the user has given through input in the software a maximum temperature rise and desired condensing pressure (kept as low as possible in these systems).

By comparison of the two T-Q diagrams we can see that the steam cycle with reheat has lower condenser pressure, and consequently the cooling water temperature rise is lower. This contributes to a larger cooling water flow. In addition more heat that has to be removed because the HRSGs are more effective. The respective condenser pressures are displayed in Figure C-7 and Figure D-7 in Appendix C and D where further details are displayed.

3.3.8 Plant energy in–out

GT PRO produces easy to read pie charts which displays what channels the energy comes into the power plant, and how it is distributed out of the power plant. This enables us too easily see where available heat and power goes (ref Figure C-8 and Figure D-8).

It is evident that most of the energy in comes from the fuel as we would expect. Further it is interesting to see that almost $^1/_8$ of the energy in comes from different sources, such as ambient air latent, ambient air sensible and make up & process return.

When comparing the steam cycles with and without reheat, we see that a larger part of the energy out for the cycle with reheat is delivered as power. This is reasonable since this cycle is more efficient. Also the condenser work is bigger for the reheat cycle since the HRSG is more efficient, and therefore also more heat has to be removed in the condenser.

3.3.9 Steam cycle energy in–out

Likewise as for the plant energy diagrams, GT PRO also produces pie chart dedicated for the steam cycle. These diagrams (ref Figure C-9 and Figure D-9) give more thoroughgoing information of where the produced steam and hot water goes. The steam energy out diagram shows how much of the steam produced actually goes to electricity production, whereas the plant energy out above only shows us the net output.

By inspection we can see that the process requirements for the top side demand a considerable amount of heat. In excess of 20% for both steam cycles of which the steam requirement clearly demands the main part.

4 Off design simulations

In this section we will look on how the power plant will react at different loads on the gas turbines, ambient temperature and cooling water temperature. Also deviation in ambient pressure is tested, owing the fact that Asian territories could have large differences in high- and low barometric pressure during upcoming storms and foul weather.

According to ref [34 and 35][5] a tropical low barometric pressure could go down to 980 hPa (0.98 bar) and maximum high barometric pressure is set to be 1050 hPa. This is a considerable deviation from the standard barometric pressure of 1010 hPa at ambient conditions. Deviations in barometric pressure results in changing needs to compress the inlet air of the gas turbine, and therefore influence the power output of the gas turbines.

GT PRO offers several methods to improve performance at high ambient temperature (not addressed in this thesis), as water injection which could be alternative on days with high ambient temperatures to keep production rate of LNG at a desired rate. It should be mentioned that extensive use of water injection could result in challenges in delivering enough fresh water.

GT MASTER, which is part of the Thermoflow package is used for the off design simulations, and further GT MACRO (part of GT PRO) for multiple off designs. It emphasized to present the results in diagrams, which makes it easy to follow the trends when reducing loads, changing ambient temperature and pressure and so forth. The two Excel sheets enclosed contains in total of over 240 pages of numbers for the different situations produced by GT MACRO.

It is assumed that the requirements for steam- and process water reduces linearly with the given part load of the gas turbines when both process trains are running. This is also assumed when only one process train is running (2 GT & HRSG Operating in Table 4-1). It should at a later stage be revised to check if the process requirements are slightly exponential due to insulation losses e.g. (increased ratio between surface area and flow).

[5] Store norske leksikon, from the web based edition.

Table 4-1 Steam and hot water process requirements for part load operation

	4 GT & HRSG Operating		2 GT & HRSG Operating	
[%]	*Steam [kg/s]*	*Hot Water [kg/s]*	*Steam [kg/s]*	*Hot Water [kg/s]*
100	14,6	10,38	7,3	5,19
95	13,87	9,861	6,935	4,9305
90	13,14	9,342	6,57	4,671
85	12,41	8,823	6,205	4,4115
80	11,68	8,304	5,84	4,152
75	10,95	7,785	5,475	3,8925
70	10,22	7,266	5,11	3,633
65	9,49	6,747	4,745	3,3735
60	8,76	6,228	4,38	3,114
55	8,03	5,709	4,015	2,8545
50	7,3	5,19	3,65	2,595
45	6,57	4,671	3,285	2,3355
40	5,84	4,152	2,92	2,076
35	5,11	3,633	2,555	1,8165
30	4,38	3,114	2,19	1,557
25	3,65	2,595	1,825	1,2975
20	2,92	2,076	1,46	1,038
15	2,19	1,557	1,095	0,7785
10	1,46	1,038	0,73	0,519
5	0,73	0,519	0,365	0,2595
0	0	0	0	0

4.1 Steam cycle without reheat

The results from the simulation in GT MASTER are outlined below, divided in subchapters between with one and both process trains running (2 or 4 gas turbines running).

4.1.1 Both process trains running at ambient condition

The simulation for all cases shows an interesting trend which is also very useful. The ambient case in Figure 4-1 illustrates the situation. Whereas the plant net power output is reduced linearly as expected (green and purple line), the steam turbine output remains more or less the same over the whole specter (blue and red line).

This behavior is caused by the gas turbine's reduced thermal efficiency at part load operation. At part load the gas turbine's heat rate [kJ/kWh] increases; meaning more heat (i.e. more fuel gas) has to be combusted per kilowatt hour produced. This extra exhaust heat is further utilized in the HRSGs making it possible to maintain the needed steam production.

This feature is one of the most useful strengths to a COGES system together with the considerable higher thermal efficiency compared to a pure gas turbine plant.

Figure 4-1 reveals that the steam turbine produces enough power to keep 100% production and thrusters going. For example when the GT Power is 80% the steam turbine have a net output of 35529 kW, and to maintain 100% production demands 31570 kW (Thrusters not operating). This leaves room to also power the LNG FPSO's thrusters at part load (please see Table 3-1). It is assumed that the power requirement reduces when the production reduces.

Offloading is a power intensive operation and in this situation the steam turbine does not deliver enough power. Under steady 100% production it takes approximately two weeks to fill up the LNG FPSO's storage tanks. The offloading situation itself takes about 12 hours[6]. A very high offloading rate is maintained to reduce the overall BOG production; in addition the general rule of "minimum non-sailing time" for any carrier also governs in this context.

This operational mode applies in only short periods every second week, and therefore it would be most efficient to supply the extra power needed by extra generators instead of overdesigning the overall system.

Instead of extra generators, supplementary firing in the HRSGs could be a cost effective solution for these short periods were extra steam for power production is needed. Please see: Power production at deviating situations for further discussion.

When one process train is running Figure 4-1 (red line) shows that we get approximately half the power output. This is reasonable when also we have half the exhaust energy to produce steam with.

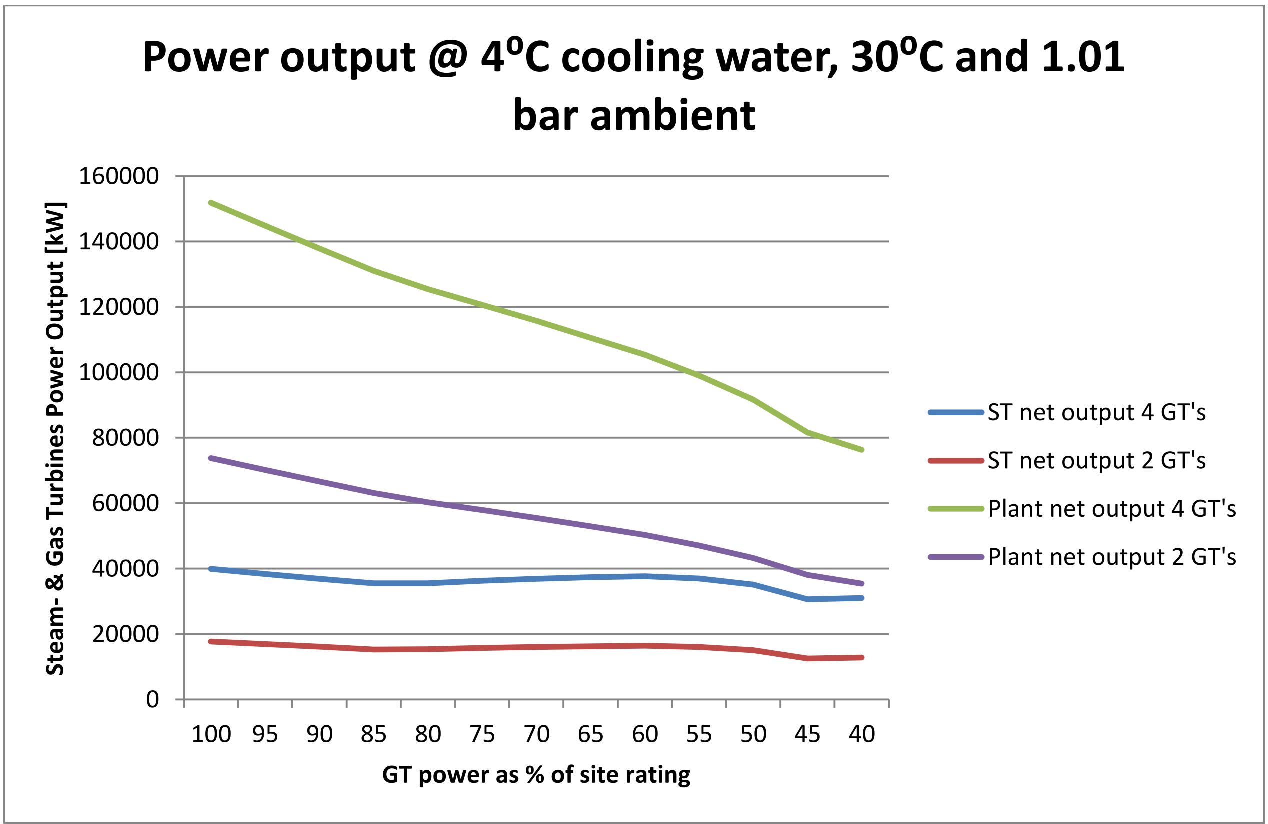

Figure 4-1 Power output without reheat @ 4°C cooling water, 30°C and 1.01 bar ambient

4.1.2 Both process trains running at non-ambient condition

In this part the ambient conditions will be changed to look how the power output deviates at off-design conditions.

[6] CB&I Boil Off and Fuel Gas Report 04301-CLN-T00-00-TR-PR-00018, page 9

The power output from gas turbines is very sensitive for changing ambient conditions. Both air pressure and temperature influence the power output. However, in this context the air- and cooling water temperature is most relevant since these variables often stays high over extended periods of time. Whereas the air pressure tends to drop before a storm, or increase at good weather conditions is more fluctuating. Figure 4-2 shows that changes in air pressure have a considerable effect on the power output (light blue line vs. light red line), whereas differences in the air temperature has less effect when changing.

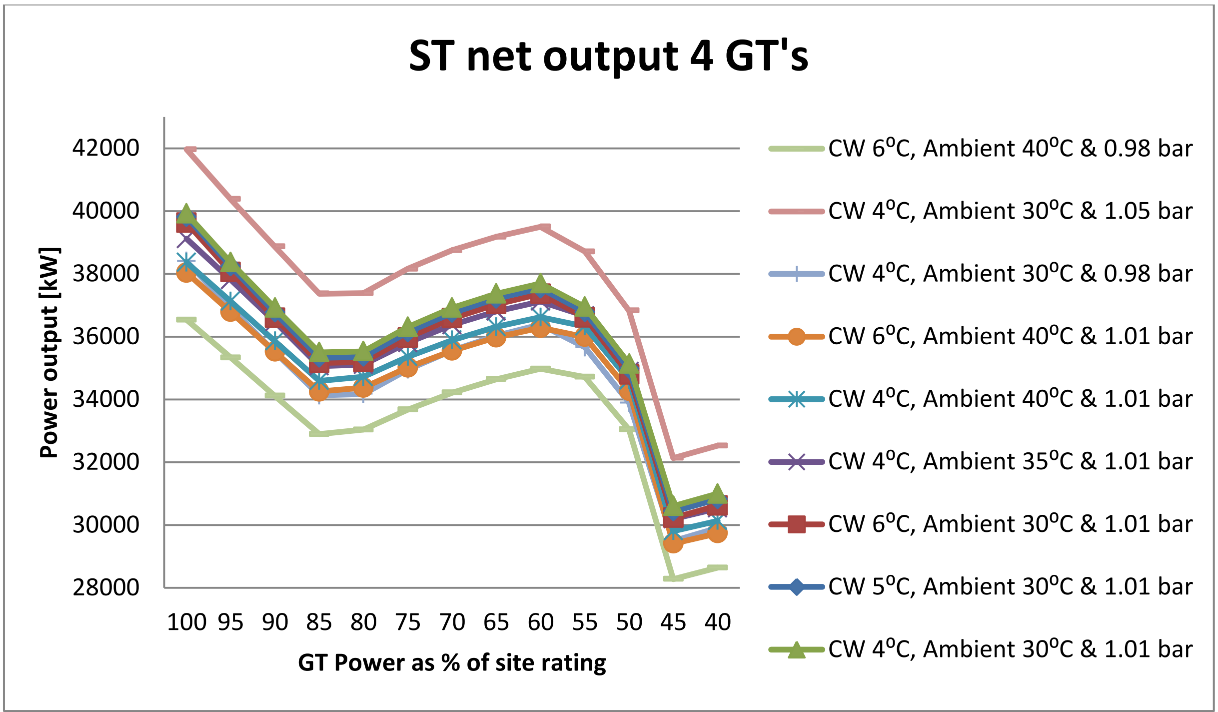

Figure 4-2 Steam turbine net power output without reheat and both process trains running

It is also evident by comparison of the "worst case" light green line (CW 6°C, Ambient 40°C & 0.98 bar) and the "best case" light red line (CW 4°C, Ambient 30°C & 1.05 bar) that difference in available power output is heavily dependent on ambient conditions. At 100% site rating on the gas turbines the net power output difference from the steam turbines is 5425 kW (41960 kW - 36535 kW) which is considerable. Not only has the steam turbine reduced power output at unfavorable ambient conditions, but also the gas turbines. Reduced LNG production and thermal efficiency is the outcome when the LNG FPSO is located in tropical areas.

The lines in Figure 4-2 are equivalent to the blue line in Figure 4-1. At first eyesight they look different, but one should keep in mind that the y-axis on the non-ambient condition diagrams is compressed (starts at minimum power output instead of zero).

Figure 4-2 shows that the steam turbine power output is decreasing steadily from 100% site rating until 85%. From 80% site rating we can again spot increased power of the steam turbine. This interesting trend is due to increasingly worsening efficiency of the gas turbine from approximately 85%.

By looking on Figure 4-3 the "bend" in the gas turbine LHV efficiency is slightly visible at 85% site rating. This trend is the same for all non-ambient conditions for with and without reheat because the same gas turbine model is used in all simulations.

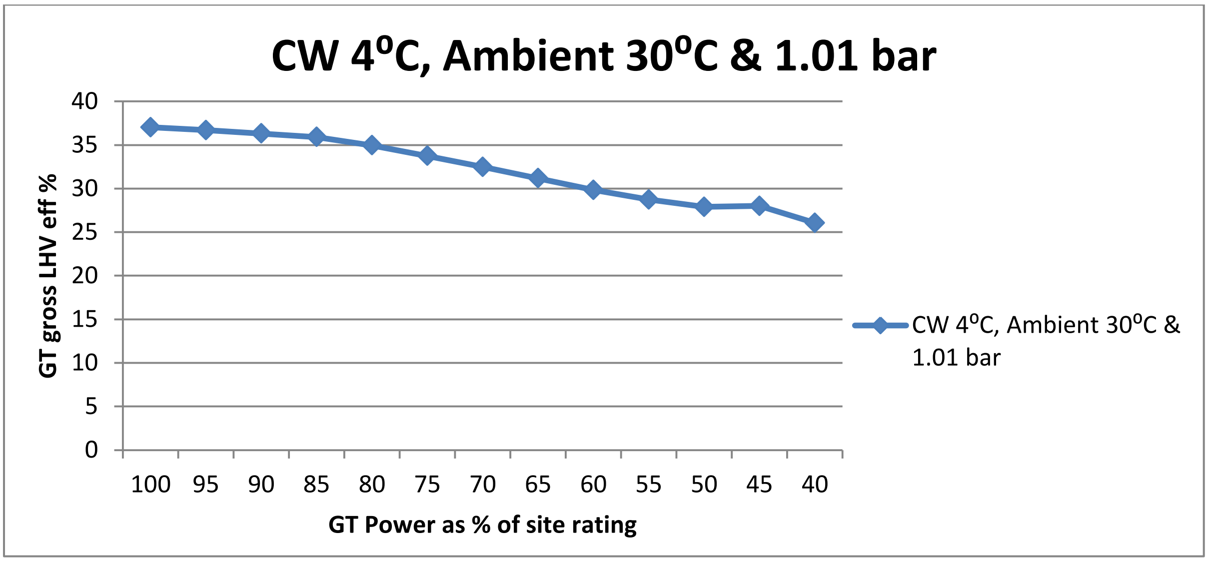

Figure 4-3 GT gross LHV at ambient condition with both process trains running

From 45% to 40% the graph in Figure 4-2 rises again. The reason for this is not clear, but it is believed that it may be due to calculation uncertainties within the software, or decreased efficiency on the gas turbines (same as for the increase from 80% to 60%).

4.1.3 One process train running at non-ambient condition

In this mode we can display from Figure 4-4 a maximum net power output from the steam turbine of 18954 kW. This is under half the amount when both process trains (4 gas turbines) are running. The explanation for this behavior lays in the nature of the steam turbine which has decreasing efficiency at part load (same as for the gas turbines). Even though the two gas turbines are running at maximum, the steam turbine will only be running at 50% rating due to half the amount of steam delivered. In this mode extra generators has to be started to fulfill the power demand of the vessel.

The difference between the "best"- and "worst" case when one process train is running is 3291 kW (18954 kW – 15663 kW). In this case an extra generator has to be fired up to deliver enough power to the vessel.

The writer has not found any data from Höegh LNG or CB&I documents displaying what power demand to expect when only one process train is running. In this context it is difficult to give precise data of what supplementary amount of power will be needed from external generators. Yet, running with one process train is a situation not often encountered and the power plant should not be optimized for this situation.

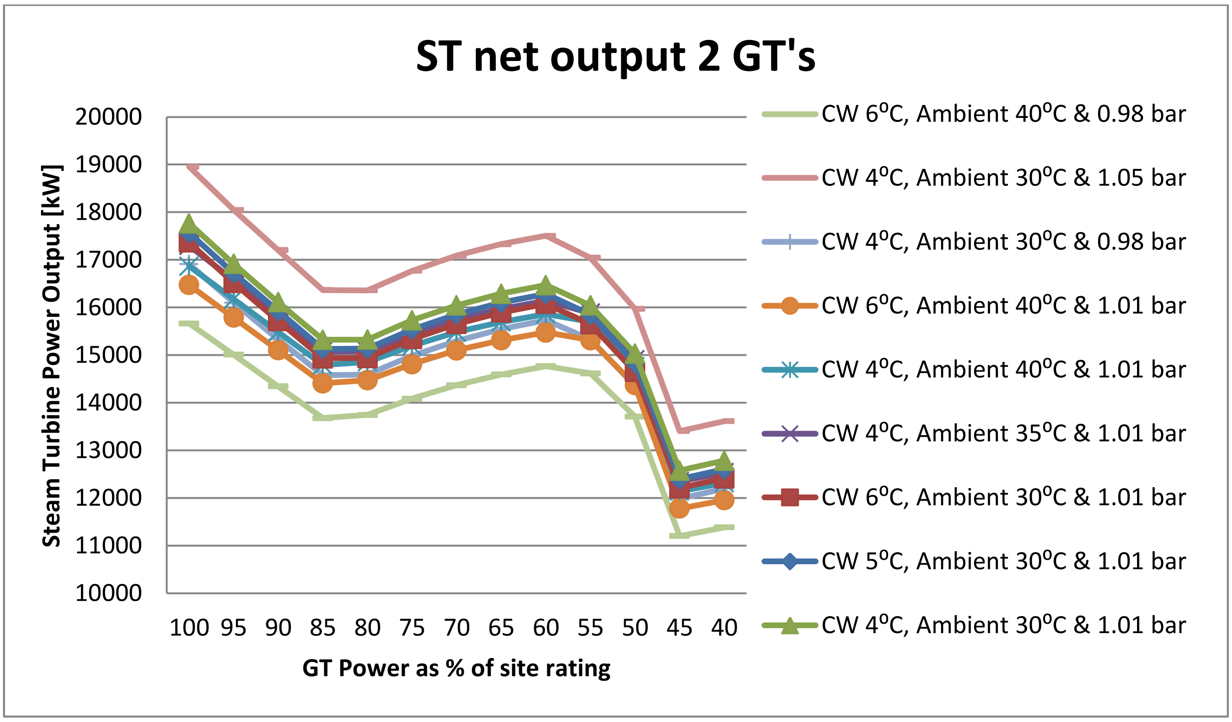

Figure 4-4 Steam turbine net power output without reheat and one process train running

4.1.4 Electric efficiency for power plant without reheat

As GT PRO is originally a program developed for calculations on land based power plants it calculates electrical-, CHP- and PURPA efficiencies. The CHP- and PURPA efficiencies is not so relevant in this context owing the fact that they are designed to inform on the total (heat and power) system efficiency and not the effective electric/mechanical efficiency. These efficiency measurements are often used for power plants with surrounding urban areas where all the low worthy energy (low temperature waste heat) is used as heating for buildings.

Unfortunately GT PRO does not calculate the mechanical efficiency as would be interesting in this context since the gas turbines in real life is connected directly to compressors. However the electrical efficiency gives us a good indication of what mechanical efficiency to expect. GT PRO includes electrical generator efficiency on 97.22% (Please see Figure C-1 or Figure D-1), so a somewhat higher mechanical efficiency should be expected. To sum up, since we also have a steam turbine producing electrical efficiency, the interesting information to see is what overall mechanical efficiency we get by utilizing COGES (total available shaft power compared to using standalone gas turbines).

The plant net electric efficiency in Figure 4-5 shows a descending trend as we would expect at part load operations when both process trains are in operation. It is also clear that the ambient conditions make the net electric efficiency swing from over 50% to fewer than 49%. In comparison a standalone GE LM2500+G4 gas turbine has an efficiency of approximately 37% under ambient operation. Clearly COGES configuration considerably raises the overall efficiency.

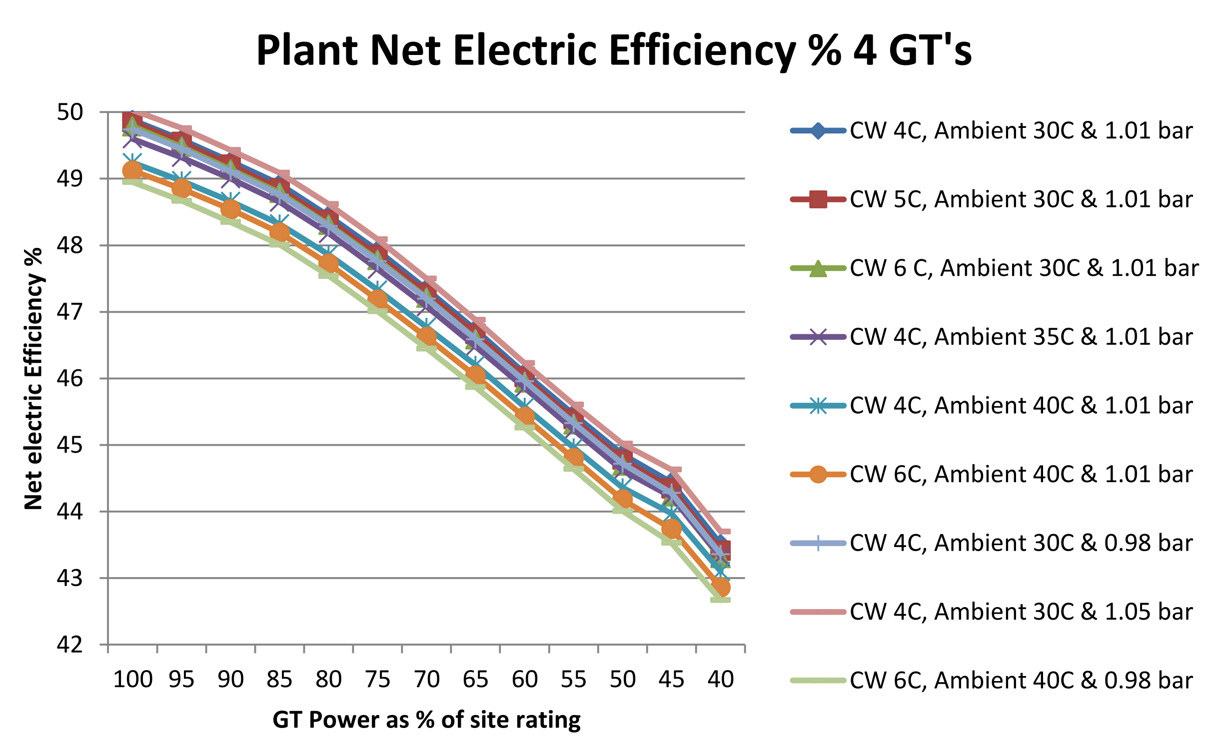

Figure 4-5 Plant Net Electric Efficiency % 4 GT's without reheat

When one process train is stopped (2 GT's in operation) we can spot in Figure 4-6 a somewhat lower efficiency over the whole specter. This has to do with part load operation (reduced efficiency) of the steam- and gas turbine, which has been mentioned earlier in this chapter. In this operational mode the different non-ambient conditions force the net electric efficiency to swing with approx. 1%.

Even though with one process train running (50% production) very high efficiency is maintained and part load production is no "disaster" in an energy efficiency perspective. Operation of one process train will/should seldom occur and only be the case under repairs, and unforeseen shutdowns.

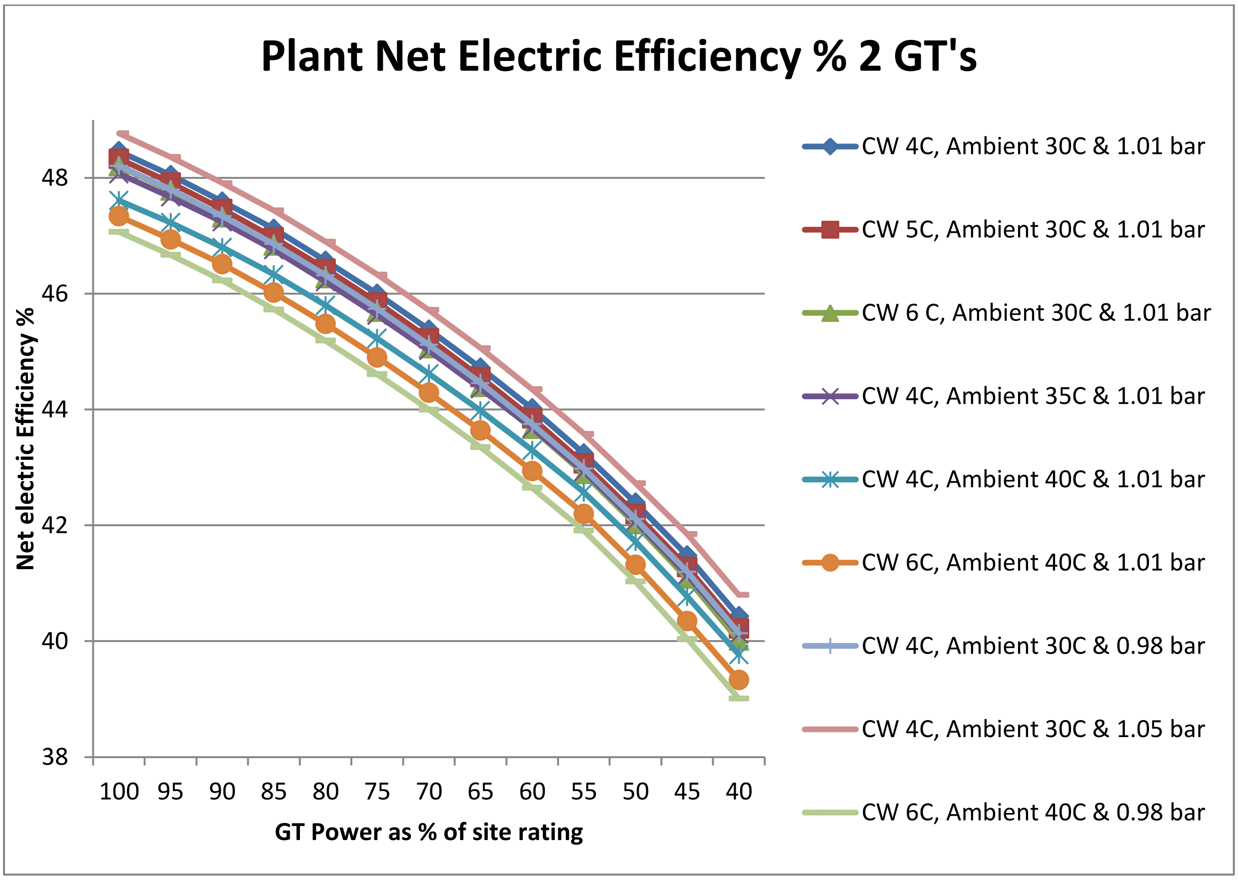

Figure 4-6 Plant Net Electric Efficiency % 2 GT's without reheat

When the efficiency of both gas- and steam turbine is calculated as one figure we can display nice descending lines. The worsening efficiency of the gas turbine is as mentioned before is “recaptured” by the steam cycle, increasing the overall efficiency at part load.

4.2 Steam cycle with reheat

Likewise as for the former chapter the simulation for power plant with reheat are outlined below, divided in subchapters between with one and both process trains running (2 or 4 gas turbines in operation).

4.2.1 Both process trains running at ambient condition

This solution has like the power plant without reheat a steady power production over the part load specter. Reheat in the steam cycle makes the net power output higher at 100% and at part load, and as mentioned before excess power is also produced.

With one process train running the net power output is slightly higher than for the solution without reheat 689 kW (18454 kW - 17765 kW @ 100% site rating) and most likely an external generator is also needed in this case (details of power consumption with one process train running is as previously stated not known).

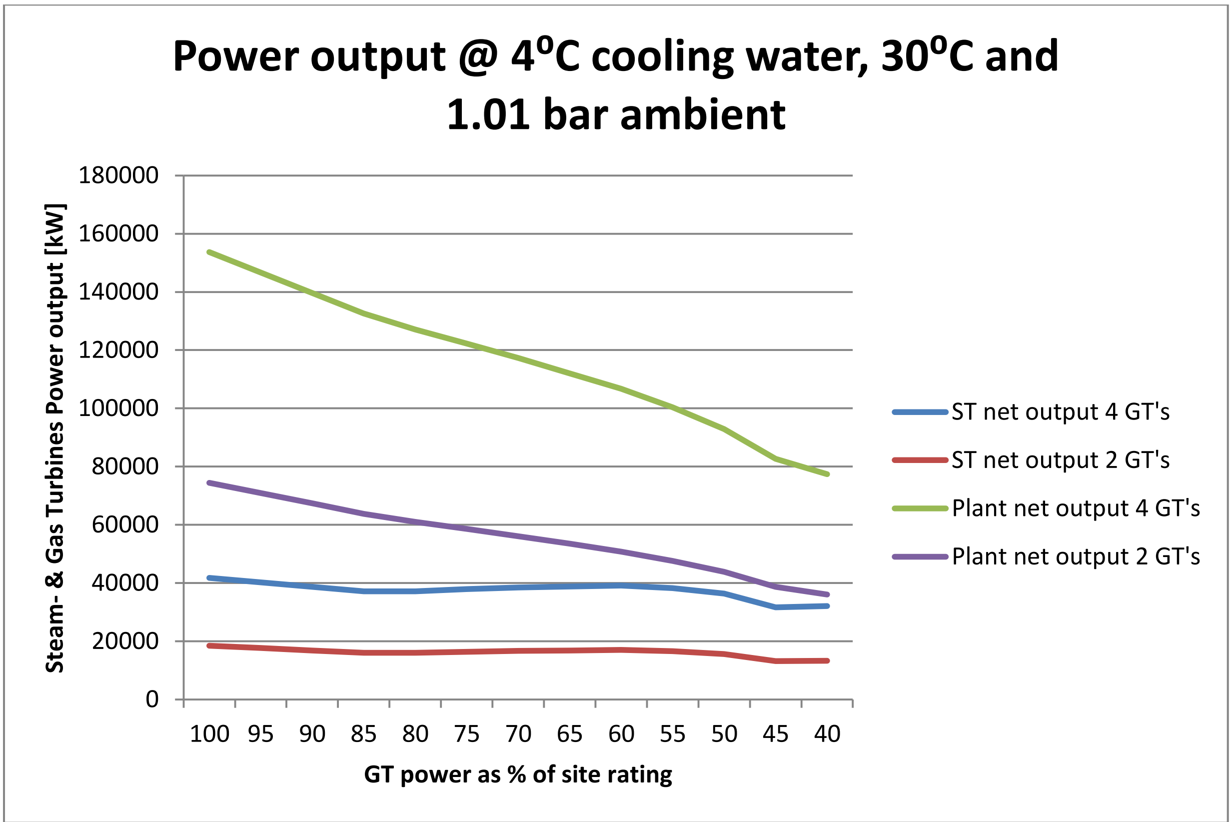

Figure 4-7 Power output with reheat @4°C cooling water, 30°C and 1.01 bar ambient

4.2.2 Both process trains running at non-ambient condition

With reheat the “best” case gives 44058 kW at 100% site rating, while the “worst” case have a power output 37982 kW. This result in a difference in power output of 6076 kW (see Figure 4-8). Further this means that even when the ambient conditions is unfavorable there is enough to maintain 100% production and full power on thrusters (power demand: 37182 kW) which is favorable in terms of easy operation and power safety (no need for standby generators, and enough power available in almost all situations). Only at offloading + 100% production and thrusters working there is not enough power available if the ambient conditions are unfavorable. As for the solution without reheat, an extra generator has to be fired up for this situation.

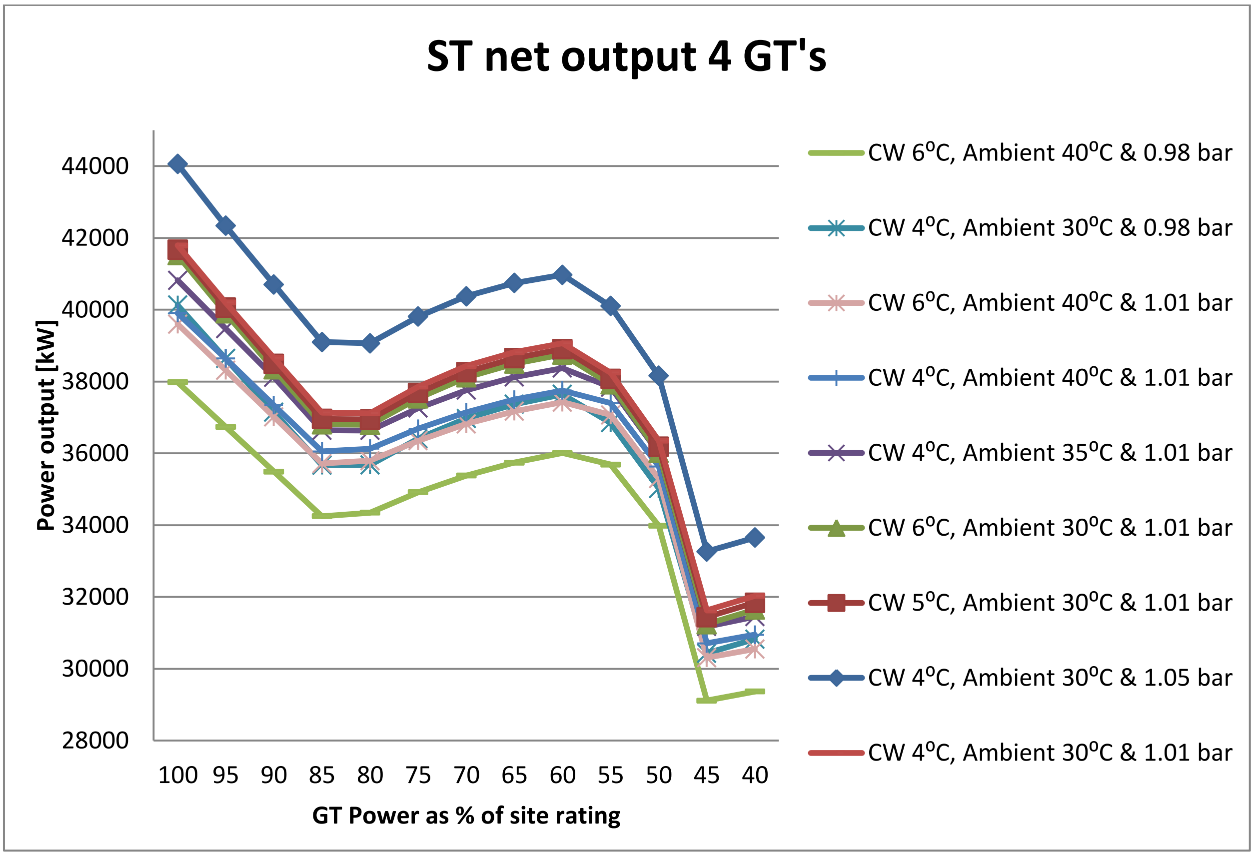

Figure 4-8 Steam turbine net power output with reheat and two process trains running

The steam cycle with reheat displays the same "dip" in power output from as the one without reheat. This is related to the gas turbines LHV efficiency and is explained earlier under the subchapter: Both process trains running at non-ambient condition under Steam cycle without reheat.

4.2.3 One process train running at non-ambient condition

The steam cycle with reheat is more energy efficient, and also better at part load. Figure 4-9 displays this statement.

Equal to the cycle without reheat, the writer has not found any data from Höegh LNG or CB&I documents displaying what power demand to expect when only one process train is running. It is also here difficult to give data precise of what supplementary amount of power needed from external generators.

The difference between "best"- and "worst" case is 3491 kW (19647 kW – 16156 kW) which is slightly larger than for the steam cycle without reheat.

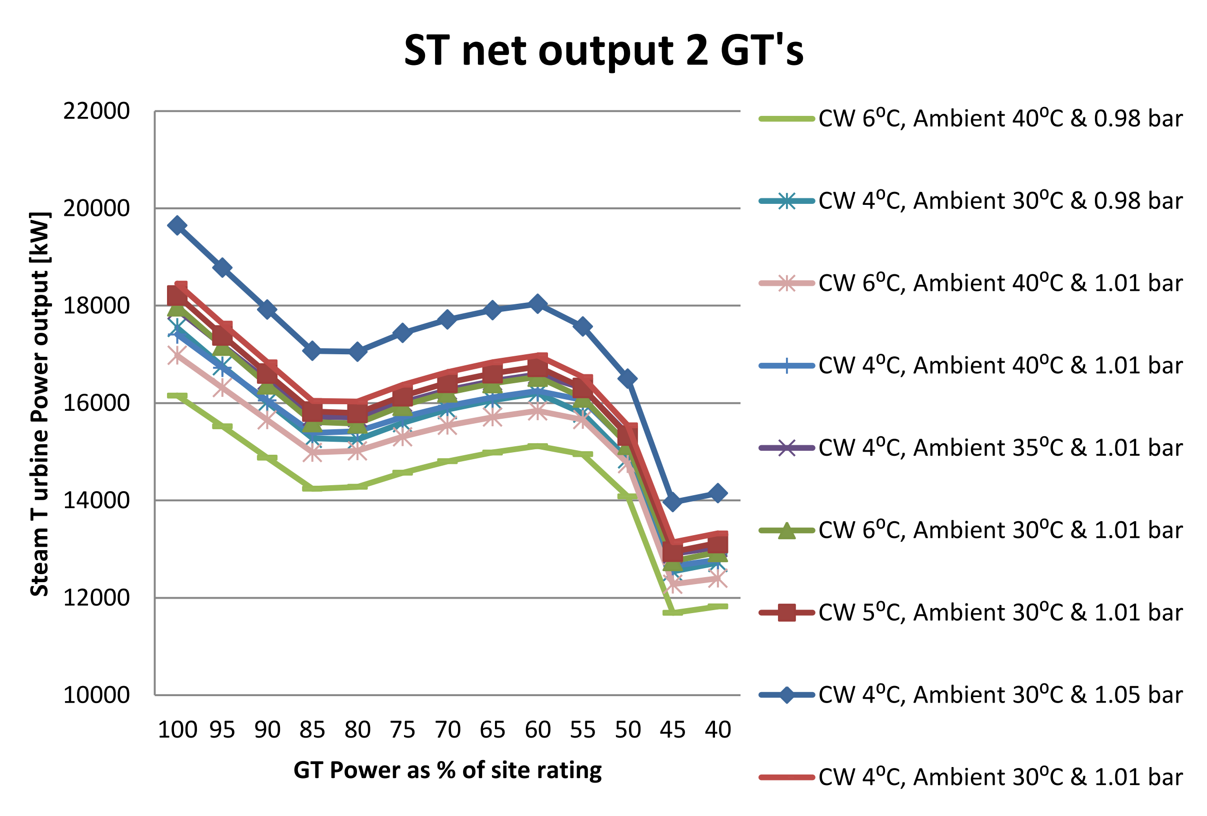

Figure 4-9 Steam turbine net power output with reheat and one process train running

Effort has been made to spread the number of non-ambient conditions widely over the different temperature and pressure variables. Even though there are still several conditions displayed here, even more should be taken into consideration at a later stage (e.g. higher ambient- and cooling water temperature). The number of conditions has been restricted due time limit and time consuming simulations. However the trend in the diagrams is clearly visible.

4.2.4 Electric efficiency for power plant with reheat

When reheat is implemented in the steam cycle we can from Figure 4-10 see that the electric efficiency lies noticeably higher. This is as expected as reheat systems are more efficient. But if approximately half a percentage point in increased electric efficiency is worth the building and operational costs are another discussion, and not addressed in this thesis.

An interesting point to observe is difference in electric efficiency between the steam cycle without and with reheat is reducing at part load, meaning they become more and more alike at reduced site ratings.

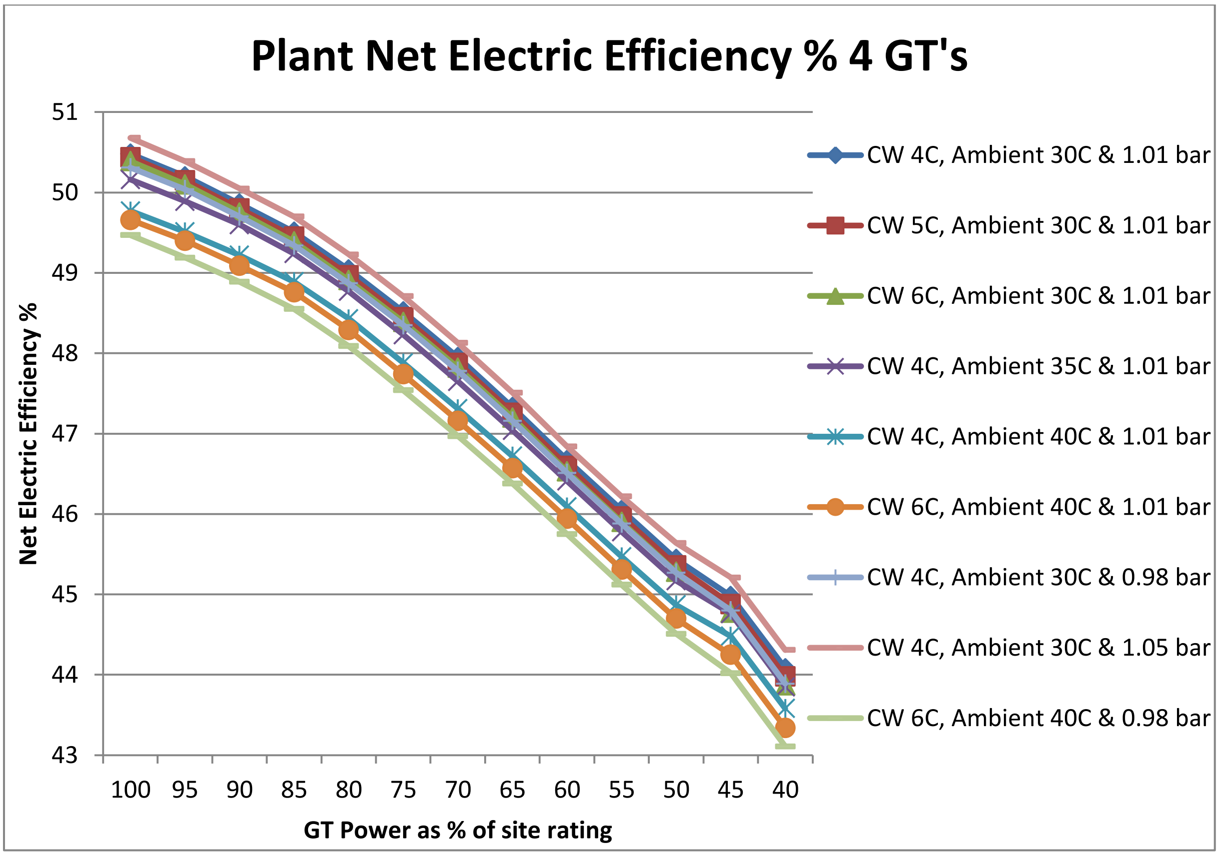

Figure 4-10 Plant Net Electric Efficiency % 4 GT's with reheat

When one process train is running (see Figure 4-11), the same trend as the steam cycle without reheat is vissible; reduced efficieny at 100% load due to part load on the steam turbine. Beyond that the electric efficiency lays just above the steam cycle without reheat.

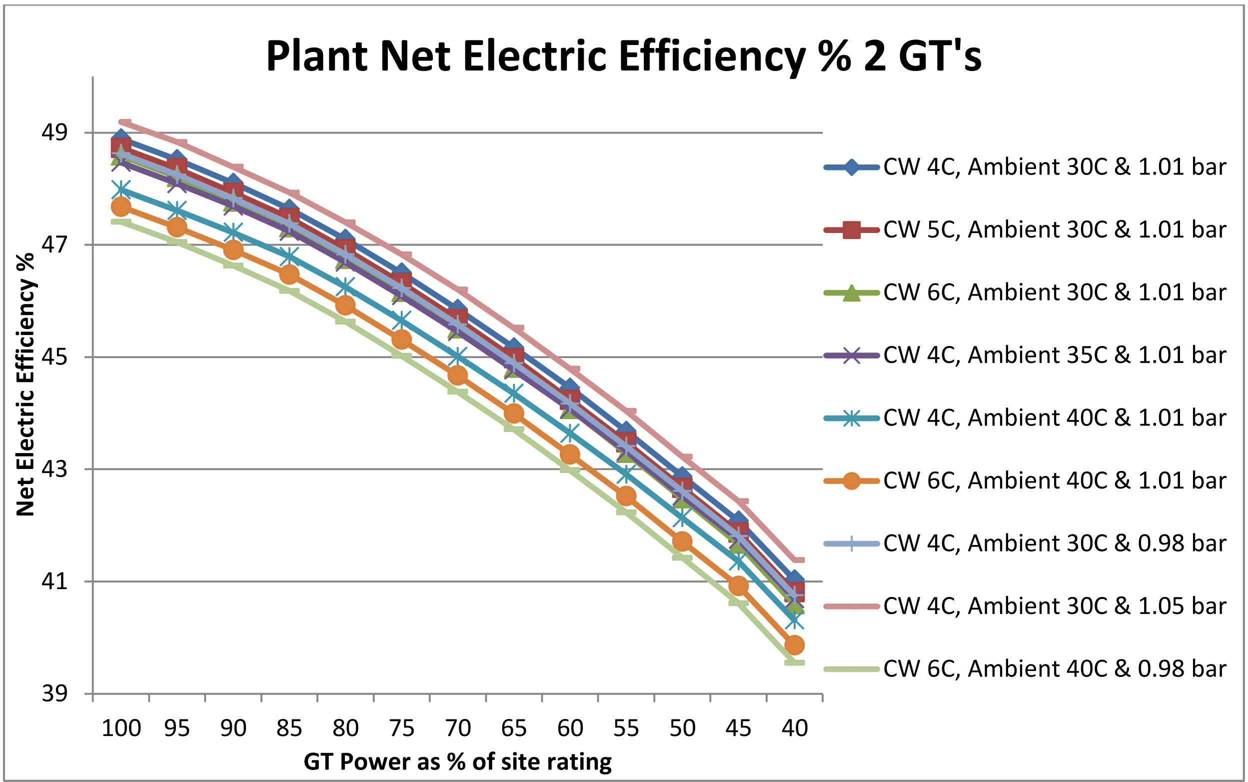

Figure 4-11 Plant Net Electric Efficiency % 2 GT's with reheat

The diagrams shown in this chapter is a summary of all data produced by GT PRO and GT MASTER. The respective Excel sheets: Off_Design_Calculation.xlsx and Off_Design_Calculation_REHEAT.xlsx is added on the enclosed CD.

5 Power production at deviating situations

The power systems above with and without reheat are designed with the intention to make the stationary LNG production as energy efficient as possible. As this power system supplies electrical energy only when the gas turbines connected to the compressors are running, electricity has to be provided otherwise when LNG production is putt to a halt/ reduced operation.

Appendix A made during the project assignment displays three (2+1) gas turbine generators independent from the main system which should provide the necessary power in off-production/ part-production situations. In this chapter we will discuss if other layouts is possible and/or favorable in terms of energy efficiency.

The initial thought during the project assignment was that one or two of the generators would step in and supply the electric power when needed depending on the deviating situation. This way of thinking is okay at transit and start-up where large amounts of electricity and heat (only during start up) is needed and can't be supplied by the gas- and steam turbine(s) in the process trains. But as displayed earlier, some production situations is almost self-sustained (the steam turbine delivers almost enough electric power), it would therefore be very energy inefficient to start up a ~30 MW gas generator to supply the needed 2- 4 MW of electric power ensuring enough electricity to the vessel.

With this in mind; the writer thinks it is important to do a brief investigation if other power solutions could prove worthy. Initial thoughts are to implement a high efficiency diesel generator running on fuel gas which can step in when needed, and remove one of the gas turbine generators. Please see Appendix B for updated layout for the overall power production.

5.1 Unfavorable ambient conditions

At unfavorable ambient conditions the steam cycle without reheat does not deliver the electricity needed during 100% production with thrusters operating and also at offloading. On the most there is lacking 4349 kW (40884 kW – 36535 kW).

Figure 5-1 2.1 MW Diesel Generator. Source: catepillar.com

For the solution with reheat there is not enough power at offloading if the thrusters are working under unfavorable ambient conditions. The extra power needed if thrusters are running at 100% under unfavorable ambient conditions is 2902 kW (40884 kW – 37982 kW).

The results exhibited are the "worst" case scenarios. Often the power demand would be less as the thrusters often are not operating at full power, 100% of the time and the ambient conditions is a little more favorable. This further emphasize why it's reasonable to have a diesel generator instead of a large gas turbine running on part load at very low efficiency. Please refer to Figure 4-3 for the LM2500+G4 LHV efficiency diagram. Keeping the diesel generator at operating temperature while at standby (tempering of cooling water) makes it possible to power up at very short notice when the power grid demands it.

The generator can run on fuel gas only (SI engines) or by utilizing a small diesel pilot flame for ignition. For more information please see the project assignment, under Energy production for further details.

5.2 Transit

The LNG FPSO is a self-propelled vessel with three thrusters á 5500 kW each. At full power the LNG FPSO is capable to do approximately 8 knots. As the LNG FPSO is thought to be stationed at one specific field for most it's lifetime, this operational mode will not be often used. Exceptions are at very foul weather when the LNG FPSO has to interrupt the production, and take refuge to avoid the worst weather. During this case, the steam turbine can no longer provide the electricity needed since the process trains have to shut down. One of the gas turbine generators in "generator package" has to step in to fulfill the energy demand in this case.

It should be mentioned that the LNG FPSO is a very huge vessel and the weather has to be extremely poor before a disconnection of the turret has to be done.

Figure 5-2 View of the dis-connectable turret and of a LNG membrane tank. Source: hoeghlng.com

In total the thrusters has a power demand of 16500 kW in transit mode at full load, and the electric top side consumption is assumed to be at a minimum (exact figures is not known by the writer, not given in available documents). An estimate for the transit consumption is set to be 20 MW. Each of the two LM2500+G4 Turbine Power Generators can deliver approximately 27.25 MW at 30^{o}C air temperature[7] and therefore one gas turbine in operation should be enough, whereas the other is in standby (redundancy maintained).

As said previously, transit mode for the LNG FPSO is a rare happening, and therefore it is not economical to adjust the “generator package” to be fuel efficient especially for this operational mode.

5.3 Start-up

LNG process trains are complex systems demanding special start-up procedures with changing heat and power demands over time. With the previously “all electric” solution chosen by Höegh LNG this start-up procedure was easily encountered, as the heat and power supply to the process is not directly coupled to the process trains. However in this case, when the gas turbines are directly coupled to the compressors which are also producing the majority of the steam, this is not as easy.

In this operational mode the two gas turbine generators which also are equipped with HRSGs will provide the necessary power and heat, before the gas turbines driving the compressors in the process trains can step in and after a while stabilize the power and heat requirements.

Both gas turbines in the “generator package” are needed in the start-up procedure to fulfill the requirements. This in turn means that there is not any redundancy before the gas turbines in the process trains are online, and further produce the required steam needed themselves to generate electricity and process requirements.

The start-up procedure should be a rare event, and the time span where there is not any redundancy present is limited. As soon as the gas turbines in the process trains have been running a certain time, the HRSGs will produce steam and further redundancy in the electricity and heat deliverance is ensured.

The cost of having a dedicated extra gas turbine to eliminate the redundancy-gap at start-up is very high and is therefore disregarded in evaluation.

A contingency is to fire up the gas turbines in the process trains and run them at part load while the compressors are in surge (no actual compression occurring), or let the cooling cycle do precooling (no natural gas running thru the process cooling cycle). By doing so the HRSGs will after some time produce steam and electricity and the start-up procedure can continue.

Another solution to eliminate the redundancy gap is simply to install extra burners in the HRSGs allowing them to produce the necessary hot water and steam needed before start up. This solution is not particularly energy efficient but it is effective, cheap and also very reliable (no moving parts). This solution will further be recommended.

[7] From ref [16] Höegh LNG “Fuel Gas Heating Medium Review for WO3-Driver Study_Rev 2”, page 2.

5.4 Unforeseen situations

Every production facility, unconditionally if it is land based or offshore, has unplanned shutdowns from time to time due to equipment failure, gas leakage, fire etc.

Emergency shutdowns can pose a problem for COGES systems (especially in this configuration) since the steam cycle is providing all the electricity needed at normal operation. If the operator has to shut down the process trains, the steam cycle will shut down as well (after some time). To have the two gas turbines in the “generator package” in standby at all times is therefore very important. In the next chapter it is questioned whet ever a stream drum could work as a “pressure accumulator” for storing steam. This component could keep the steam turbine running for an extended period of time after the gas turbines has been shut down, which could be useful in an emergency shutdown.

Another dilemma by utilizing steam to produce electric power is when an unwanted shortcut in the electrical grid occurs. To be able to stop the power production very fast, steam has to be dumped or directed directly to the condenser in the steam system. This capacity has to be implemented in the design.

In this case it is also very important to have independent safety generators in standby, to prevent a dead ship situation.

5.5 Transient behavior

The calculations done in GT PRO and GT MASTER are steady-state calculations under given ambient conditions and site rating. A steady-state is a condition where all state variables are constant, unconditionally of ongoing processes that strive to change them. This implies that the steady-state calculations don’t say anything how the system behavior is before the steady-state is reached. In other words, how or how long it takes before the system reaches steady-state is difficult to predict.

In a practical sense, this implies that it is difficult to tell how long it takes for the system to go from e.g. 100% to 80%.

Since GT PRO and GT MASTER is steady-state software, it has not been possible to go further into the dynamic set of problems related to the transient situations. This would in that case demand a different software package and analysis.

6 Layout

The proposed systems made in GT PRO have been carefully outlined on schematic drawings produced by the Thermoflow software. However these drawings do not show the physical location of where the different components should be situated on the vessel. This chapter has therefore been added to clarify this. The direct drive configuration of the gas turbines forces them to be situated with the compressors in the liquefaction train number 1 and 2 in the vessels aft section. Naturally this is also the location were the HRSGs are situated, and therefore also all the steam and hot water production at normal operation.

The steam turbine has been positioned behind the compartment section with the other electricity production. At first eyesight it may seem unreasonable to have the steam turbine far away from the steam production; however there are several reasons for this. The first is because there is likely to be restrictions in available space nearby the HRSGs in the process trains. Secondly a large part of the steam and hot water produced are already shipped to the gas cleaning section behind the turret, so the stream are already going that direction. Thirdly it is reasonable to gather all the electricity in one section.

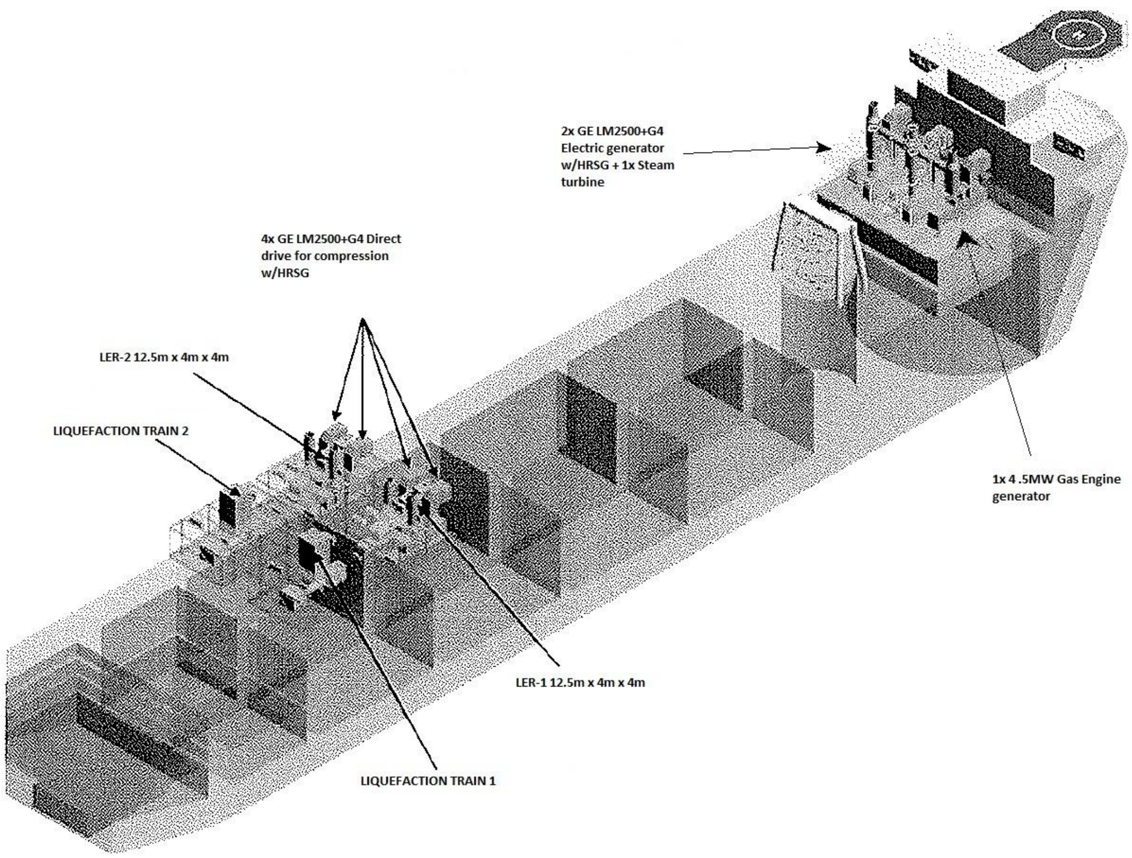

Figure 6-1 LNG FPSO sketch of the power plants components location. From ref [10] page 39.

As mentioned earlier, the distance between the steam production and consumption is fairly long and it is believed that this will increase pressure- and heat loss. GT PRO has input where increased pipe losses could be adjusted. How much adjustment needed is uncertain and therefore not changed. This should however be revised at a later stage.

7 Possible changes or improvements

During the design process several ideas and suggestions to how the system can be modified has emerged. Some of the possible changes would pose rather large changes to the already designed process trains on the top sides, and in this context not been further evaluated. Should however the topside be designed from scratch again with the intention of implementing COGES, the ideas presented here should be considered once more.

7.1 One gas turbine instead of two

As aforementioned in the project assignment it could be advantageous to have one large gas turbine driving both compressors in each train, instead of two smaller ones. Since both gas turbines has to run to keep the process train operational, there is no loss in reliability by using one gas turbine instead. Actually if the larger one (for example a GE LM6000 gas turbine) has the same reliability as the smaller one, the solution with one gas turbine will have the overall higher reliability. Because the smaller ones operates in series they are dependent on each other, and this will reduce the overall reliability.

Again if this design proposal could be realized, heavy modification of the design in the process trains has to be done which is both cost- and time consuming.

A single turbine configuration could be a nice solution in combination with single shaft type combined cycle which the next segment will describe more in detail.

7.2 Single shaft type combined cycle

Producing electricity with the help of the excess steam produced gives very high energy utilization. However there are some operational challenges when the operational mode is deviating from the normal.

If there is enough available space in the process trains a solution could be to boost the compressors with steam turbines directly on a single shaft (see Figure 7-1). In this case the electricity production has to come from standalone gas turbine generators located astern the compartments.

By doing so, the electricity production does no longer rely upon that the process trains are running to be able to produce steam (and further electricity). At start up the steam turbines would gradually boost the compressors as more and more steam is produced, and after some while the situation would stabilize. Higher reliability and easier start-up would be the result of this configuration.

The overall efficiency will be somewhat reduced compared to the solutions in this master thesis unless there are HRSGs mounted on the standalone gas turbine generators astern the compartments. These HRSGs could then produce steam to the process requirements (and also boost the steam turbines if there is enough steam available).

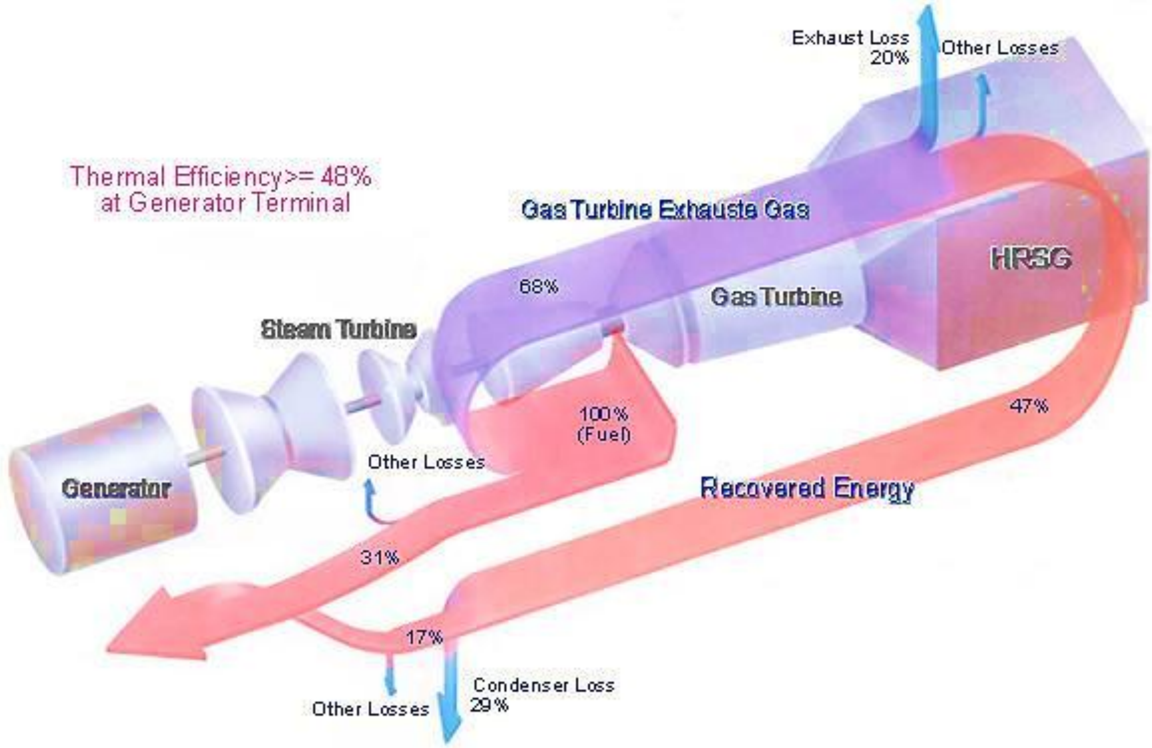

Figure 7-1 Illustration of Combined cycle with single shaft. Source: tic.toshiba.com

Figure 7-1 displays a generator at the end, but in this context the shaft would be connected directly to the compressors for the cooling cycle.

A possible challenge with this design is the long shaft. The LNG FPSO is in constant movement due to restless sea, and this result in torsion and small movements in the hull which could bend the shaft. A strong frame for the power module or flexi shaft at strategic points could solve this possible issue.

As a final note, it should be mentioned that GT PRO also has the possibility to simulate single shaft combined cycles.

7.3 Steam drum

To some degree it is thought that peak electric power demands could be covered by collecting excess steam at reduced load to boost when necessary during high loads on the thrusters. This solution is only applicable or relevant to the steam cycle without reheat, as this does not deliver enough power at 100% production + offloading and thrusters.

It should be noted that the LPB, IPB and HPB (Low-, Intermediate- and High Pressure Boilers) inside the HRSGs already works as a small scale steam drums, and also acting as a phase-separator for the steam/ water mixture. However, if they should work as a buffer they should be enlarged or a separate steam drum has to be mounted after the HRSGs.

8 HAZOP

In the following subchapters a brief HAZOP analysis is made for the potential hazards and difficulties appurtenant to implementing a COGES system to the LNG FPSO. The Norwegian company Safetec (Risk and asset management services) has the following definition on their internet page:

"HAZOP (HAZard & OPerability analysis) is a well-established method for identifying potential safety and operational problems associated with the design, maintenance or operation of a system. A HAZOP is a formal and objective process, where a group assesses the different parts of a given system with the aid of "guidewords". This ensures a systematic and well documented evaluation of potential problems/hazards."[8]

The HAZOP study is also called a "What if?"- Study. It can be summarized to be a structured and systematic examination of a planned or existing process/operation in order to identify and evaluate problems that may represent risks to personnel or equipment, or prevent efficient operation. As a reference, this kind of study has been unquestionably successful in reducing the incidence and mitigating the consequences of major accidents in all industries dealing with toxic, reactive, flammable and explosive substances.

The study is further based on a qualitative technique based on guide words. HAZOP study in short is done by following the list below:

1. Divide the system into study nodes
2. Choose a study node
3. Describe the design intent
4. Select a process parameter
5. Apply a guide word
6. Determine causes
7. Evaluate consequences/problems
8. Recommend action: What? When? Who?
9. Record information
10. Repeat procedure from step 2

A visual representation of the HAZOP procedure is displayed in Figure 8-1.

It is important to be aware of that the designed power systems in this master thesis is a component in a much larger process system on the top side of the LNG FPSO. Both systems rely on each other. The process facilities need power and heat from the power plant and the power plant need fuel gas from the process trains (only dependent if sulfur has to be removed first). Failure to deliver power and heat does directly influence the process system. In addition if we look on the LNG FPSO in a totality with the LNG tanks as a node in the HAZOP analysis, an explosion or fire could have large consequences.

The writer has not the knowledge nor the time to do a full HAZOP analysis on the whole process system as this is a huge and complex task. However the COGES system will be investigated, further it will be emphasized where the system could influence the overall process system on the LNG FPSO.

[8] From Safetec: http://www.safetec.no/index.php?c=78&kat=HAZOP+-+HAZID+-+CRIOP

In reality a HAZOP study should be executed in teams where two types of persons is needed; those with detailed technical knowledge of the process, and those with knowledge and experience of applying highly structured, systematic HAZOP approach. Further the team should not exceed more than 10 participants and no less than 3. Note! The last principle has not been followed in this study as this is an individual master thesis. However discussion and addition from external sources has been taken into account.

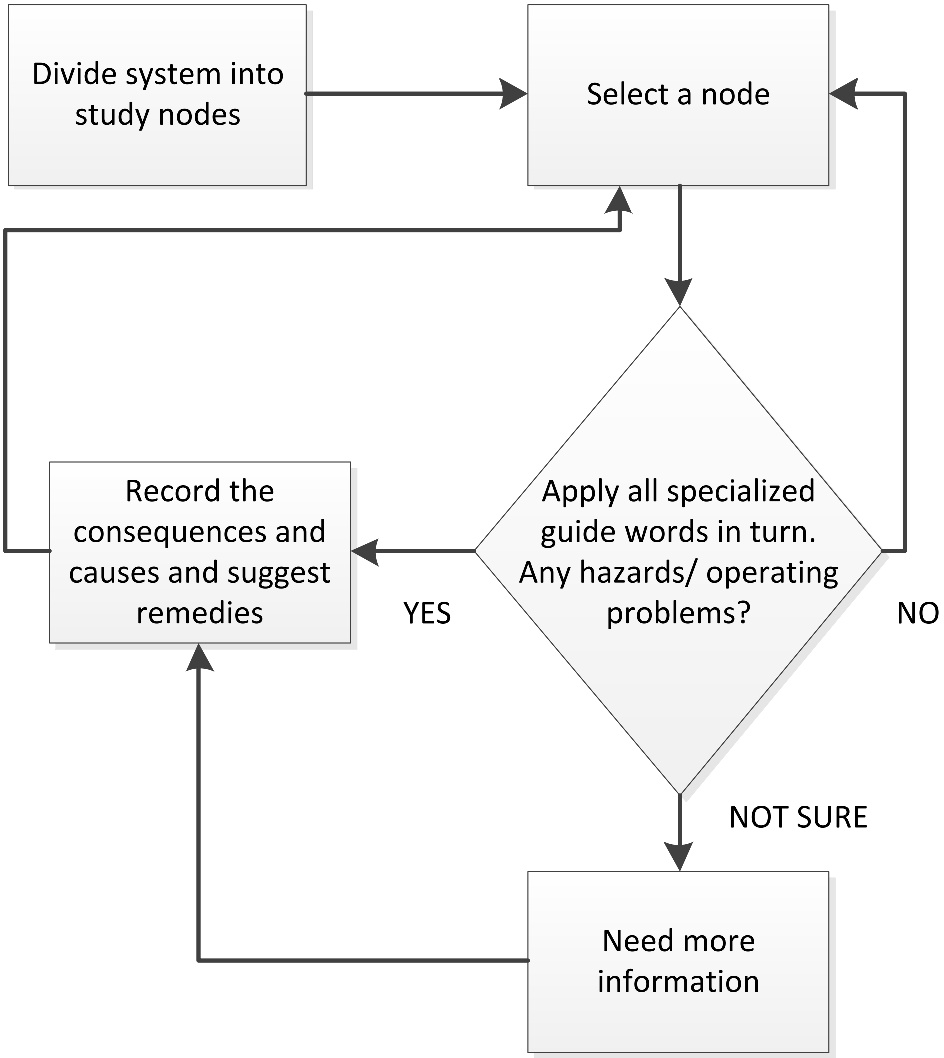

Figure 8-1 HAZOP flow diagram

In the following table the HAZOP guide words[9] are listed:

[9] From Reference [39], page 15.

Table 8-1 HAZOP guide words

Guide Word	***Meaning***	***Example***
No, Not, None	None of the design intent is achieved	No flow when production is expected
More, More of, Higher	Quantitative increase in a parameter	Higher temperature than designed
Less, Less of, Lower	Quantitative decrease in a parameter	Lower pressure than normal
As well as, More than	An additional activity occurs	Other valves closed at the same time
Part of	Only some of the design intention is achieved	Only part of the system is shut down
Reverse	Logical opposite of the design intention occurs	Back-flow when the system shuts down
Other than, Other	Complete substitution – another activity takes place	

The power plant will be divided in the following parts: Inlet filter, gas turbines, HRSGs, piping, steam turbine, condenser, steam cycle pumps, cold water pump and feed water tank. This is a superficial partition, but involves the most important components.

In the following HAZOP will be applied on the previously mentioned parts using standardized tables. The risk sat is subjective and should only act as a guideline.

8.1 Inlet filter

The inlet filter is a component with no moving parts and not subject to sudden failures in the same way as rotating machinery is. However it is important that this component work flawlessly to ensure safe and efficient operation.

Table 8-2 HAZOP table for inlet filter

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Inlet filter						
FLOW	Less	Filter is clogged	Decreased efficiency on gas turbines	Cleaning of air filter	Reduced power output/efficiency	1
	More	Leakage in the filter	Impurities & salt aerosols in GT	Seal leakage	Increased power and wear	1
	NO	Blockage	Shut down	Remove blockage	No power	1

8.2 Gas turbines

The four gas turbines are the “heart” of the system, as this is where the power and heat is produced first. The electricity production at normal operation is dependent on the gas turbines to ensure enough steam to the steam turbine.

Table 8-3 HAZOP FLOW table for Gas turbine

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Gas turbine ***FLOW***	Less	Filter is partly clogged	Decreased efficiency on gas turbines	Cleaning of air filter	Reduced power output/efficie ncy	1
	More	Leakage in filter/low ambient temperature	Impurities & salt aerosols in GT	Seal leakage	Increased power/efficien cy	1
	NO	Blockage, critical failure	Shut down	Remove blockage, repair	No power	2

Table 8-4 HAZOP PRESSURE table for Gas turbine

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Gas turbine ***Pressure***	Less	Deteriorated compressor blades	Reduced power	Change blades	Poor efficiency, reduced power output	1
	NO	Blockage, critical failure	Shut down	Remove blockage	No power	2

Deteriorated compressor blades can be an outcome of defect inlet filters. As mentioned under inlet filter, increased wear will happen when salt aerosols and impurities get in the gas turbines.

Table 8-5 HAZOP TEMPERATURE table for Gas turbine

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Gas turbine						
TEMPERATURE	Less	Not enough fuel	Reduced exhaust temperature	Check fuel system	Reduced power output & steam production	2
	More	Too much fuel	Increased wear on HP turbine blades and HRSGs	Same	Higher exhaust temperature & power output	1

The next table looks on the potential hazards connected by having gas turbines inside the process trains. This is where some of the major safety issues are assigned. Due to combustion machinery inside a process facility, an undetected gas leakage could have large consequences if not detected early. By default the gas turbines should trip when gas is detected in the inlet, but this does not always happen in time. If gas gets into a gas turbine when running, an explosion is often a fact.

Höegh LNG has also expressed that a gas leak could be dangerous because of high exhaust stack temperature. Exhaust stack is approx. 500°C and LPG self-ignite at 400°C[10]. The exhaust stack should be insulated, but this could deteriorate over the LNG FPSO's life time.

Table 8-6 HAZOP FUEL SYSTEM table for Gas turbine

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Gas turbine						
PROCESS TRAINS	Small leakage	Seal leakage	Gas turbine trip shutdown	Seal leakage	Gas detection alarm	3
	Large leakage	Pipe rupture	Gas turbine trip (chance of explosion), self-ignition exhaust stack	Shut down, fire extinguishing	Automatic shutdown	4

[10] Please see: Safety considerations in the project assignment for further details.

8.3 HRSG

In the two systems proposed in this master thesis the HRSGs play an important role since both the steam turbine and process trains are dependent on these components. Luckily, there are 4+2 HRSGs in the system, ensuring that if one or several fails the situation is not critical.

In a safety perspective to personnel and equipment the HRSGs could be very dangerous (this also concern the rest of the steam system). The danger of a catastrophic pressure vessel failure (boiler explosion) is always present when HP systems are utilized.

The reason why HP systems could be very dangerous is related to the nature of the liquid used. Water under pressure will not boil at 100°C (e.g. as boiling water in the kitchen) because of limited volume. Since the pressure rises, so does the temperature needed to get the water to boil/evaporate. As a result, massive amounts of energy get stored in HP boilers. When/if a pressure vessel or boiler fails, an instant drop in pressure will happen. The water will still have the same temperature (in this HP cycle ~500°C), but pressure is instantly reduced to ambient where the evaporation point is lower. Then without pressure the water will instantly flash boil into steam and produce a very hot pressure wave. Historically this is known to be very dangerous and deadly. Examples are steam locomotive boiler explosions from the start of the 19th- and through the middle of the 20th century. The picture below clearly displays which forces are involved even when dealing with a relatively small pressure vessel.

Figure 8-2 Example boiler explosion. Source: http://www.flickr.com/photos/beamishmuseum/

Table 8-7 HAZOP FLOW table for HRSG

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
HRSG						
FLOW	Less	Defect pressure pump	Overheating in HRSG, shutdown of GT	Replace pressure pump	Reduced delivery of steam & hot water	2
	More	Leakage in steam cycle	Feed water tank runs out	Fix leakage	Drop in feed water level	2
	NO	Blockage	Shut down, excessive pressure	Remove blockage	No power, steam, hot water	2

Table 8-8 HAZOP PRESSURE table for HRSG

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
HRSG						
PRESSURE	Less	Leakage/ defect pressure pump	Power- and fresh water loss	Block leakage/ change pressure pump	Reduced power, steam & hot water	2
	More	Too much exhaust heat, blockage	Pressure leakage, heat damage	Reduce GT power rating	Excess steam, water & steam temperature & pressure	1
	NO	Blockage	Shut down	Remove blockage	No power, pressure leakage	2

Table 8-9 HAZOP TEMPERATURE table for HRSG

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
HRSG						
TEMPERATURE	Less	Not enough exhaust	Reduced steam production	More heat required	Reduced electric power output	2
	More	Error in GT's	Increased steam production	Steam dump	Higher steam temperature and pressure	1
	NO	GT's shut down	No electricity, steam or hot water	Start GT's	No power	2

8.4 Piping

When HP steam is distributed to different parts of the vessel it is important to remember that it could be very dangerous in case of a leakage. If a sudden rupture occurs the same safety hazards to personnel and equipment would apply as mentioned for the HRSGs above. Piping around the vessel with superheated steam is also a safety concern.

Since superheated steam is not visible when released to air (not before it condenses) this could be a safety hazard for personnel because the leakage is difficult to spot. To notice a steam leakage with the help of hearing is also difficult due to the high sound levels already present in the process facilities.

The condition of the piping relies on a large degree on the quality/purity of the steam. The combination of air and other non-condensable gases (carbon dioxide and oxygen) in steam systems could accelerate corrosion and further block flow. Over time these set of problems could result in leaks, steam/water hammer, reduced heat transfer which accumulate in expensive repairs and costly down time.

If the deaerator does not work as it should the consequences mentioned above could prove real if not gases in the feed water are removed properly. Poor cleaning of the deaerator directly affects piping and all other components as HRSGs, steam turbine, heat exchangers and so forth.

It should be emphasized that carbonic acid only forms below steam temperature when carbon dioxide and condensate combines. This means that return- and drain lines, heat exchanger walls, and tubes are especially exposed to corrosion.

"Corrosion is often so severe that condensate discharged from a steam trap may be bright red or dark brown from iron content. Under such conditions, components within the heat transfer equipment may not withstand the pressure of the system"[11]

For further information please see chapter: Steam- and process water, and also reference [40].

[11] From ref [40] The Dangers of Uncontrolled Gases in Steam Systems

Table 8-10 HAZOP FLOW table for Piping

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Piping **FLOW**	Less	Leakage, Water pump defect	Increased temp in HRSG, reduced power, steam & heat	Seal, Replace water pump	Reduced power output & steam production. Increased steam temp	2
	More	Failure to reduce pump level	Steam at to low temperature	Replace control unit	Reduced steam turbine output	1
	NO	Blockage	Shut down	Remove blockage	No power	2

Table 8-11 HAZOP PRESSURE table for Piping

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Piping **PRESSURE**	Less	IP or HP pump failure, leakage	Increased temp in HRSG, reduced power, steam & heat	Replace water pump, seal leakage	Reduced power output & steam production	2
	More	Failure to reduce pump level, blockage	Pressure leakage	Replace pump, remove blockage	Leakage in seal, connections	1
	NO	Blockage	Shut down	Remove blockage	No power	3

8.5 Steam turbine

The reliability to the steam turbine directly influences the process trains as the electricity production comes exclusively from this source in normal operation. In that context it is important to highlight the weaknesses of this component.

Table 8-12 HAZOP PRESSURE table for Steam turbine

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Steam turbine ***PRESSURE***	Less	Problems in piping or HRSG or IP/HP pump	Reduced electric power production, out of sync (not 60Hz)	Check piping, HRSG or IP/HP pump	Reduced power output & steam production	2
	More	Same and defect throttle, safety valve	Excessive rpm, bearing damage, out of sync (not 60Hz)	Shut down, replace throttle, safety valve	Excessive rpm	2
	NO	Blockage	Shut down	Remove blockage	No power	3

In addition the steam turbine is especially sensitive to impurities in the steam, where increased wear on the turbine blades could be the result. In this context it is very important that the deaerator operates correctly ensuring that the feed water is clean and without any impurities.

8.6 Condenser

The condenser is a large heat exchanger which could be subject to corrosion in the same way as the equipment mentioned above. The risk of break down is low, if scheduled maintenance is followed. In this way excessive wear can be revealed, and further unforeseen breakdowns reduced.

Table 8-13 HAZOP FLOW table for Condenser

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Condenser ***FLOW***	Less	Blockage in inlet, CW pump damaged	Increased condenser pressure =>reduced power output	Remove blockage, replace CW pump	Reduced power output from steam turbine, increased temp in feed water tank	2
	NO	Blockage	Shut down	Remove blockage	No power	3

Table 8-14 HAZOP PRESSURE table for Condenser

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Condenser **PRESSURE**	Less	Partial blockage from steam turbine	Decreased condenser pressure	Remove blockage	Reduced need for cooling water flow	1
	NO	Blockage	Shut down	Remove blockage	No power	2

8.7 Steam cycle pumps

These pumps are small but vital components to ensure the operation of the steam cycle and power production. To ensure high up time two pumps in parallel should be considered, with one pump in standby.

Table 8-15 HAZOP PRESSURE table for Steam cycle pumps

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Steam cycle pumps **PRESSURE**	Less	IP or HP pump failure, leakage	Increased temp in HRSG, reduced power, steam & heat	Replace water pump	Reduced power output & steam production	2
	More	Blockage after pump	Increased system pressure/ less flow	Remove blockage	Increased pressure	1
	NO	Blockage, no power delivered to pumps	Shut down	Remove blockage, reconnect power	No power from steam turbine or process heat /steam	2

8.8 Cold water pump

To ensure the lowest possible condenser pressure huge cold water pumps has been installed. Reduced flow will result directly in lowered power output in the steam turbine. This is not critical, but undesirable since an extra gas turbine generator may have to be started. As for the steam cycle pumps, a parallel configuration should be installed to ensure high up-time.

Table 8-16 HAZOP PRESSURE table for Cold water pump

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Cold water pump ***PRESSURE***	Less	CW pump failure, leakage	Increased temp in condenser and pressure. Less power output	Replace water pump, seal leakage	Reduced power output	2
	More	Failure to reduce pump level, blockage	Pressure leakage. Increased temp & pressure in condenser	Replace pump, remove blockage	Leakage in seal, connections. Increased condenser pressure	1
	NO	Blockage	Shut down	Remove blockage	Decreased turbine power	3

8.9 Feed water tank

The feed water tank is the water "buffer" in the steam cycle acting as an accumulation tank for the different return lines from process and steam turbine. This tank gives a great insight of the current water consumption in the system i.e. if there is a leakage etc.

Table 8-17 HAZOP LEVEL table for Feed water tank

Equipment Reference	Deviations from operating conditions	Possible causes of deviation	Consequences of the deviation	Action required	Process Indications	Risk (Min=1, Max=5)
Feed water tank ***LEVEL***	Less	Reduced makeup or process return, leakage	Feed water tank drains out, HRSG runs dry	Troubleshoot process return, block leakage	HRSG runs dry,	2
	More	Excessive return from process	Feed water tank overfills	Release water from system	Excessive pump resistance	1
	NO	Leakage, blockage in process return water	Shut down, HRSG runs dry, heat damage	Remove blockage	No power	3

8.10 HAZOP round-up

The analysis points out some interesting clues of where there are safety issues and possible operational difficulties.

From this analysis it has been revealed two noteworthy concerns about implementation of COGES in LNG process trains. Implementation of larger and more complex steam system in the process trains introduces an increased safety hazard for personnel working in these areas. Larger streams under higher pressure increases wear and how fast a small malfunction can evolve into a larger problem. Increased risk of disastrous boiler explosions increases as higher steam pressures are introduced.

Passage ways, stairs etc. should be positioned in a safe distance away from HP equipment and piping to avoid exposing workers too much to these possible hazardous components. This is necessary since Höegh LNG (and most other companies) has a zero injury policy.

The other major concern is by utilizing direct drive on the compressors, the designer is forced to position the gas turbines in the middle of the process trains. This is unfortunate, but necessary. However by designing the air intakes correct (to minimize the risk of a potential gas leak reaching the inlets), and further strengthen the floor under the gas turbines these safety issues should be able to overcome.

Questions of what happens if the power system fails to deliver the requested amount of hot water and steam has not been answered, since the writer does not obtain enough information on the overall process facility. At reduced delivery it's logical to think that the LNG production has to slow down, and further shut down if the hot water and steam flow becomes too low.

9 Conclusion

This master thesis has studied in detail the performance of two differently designed COGES systems. The first has three pressure cycles with a condensing steam turbine. The second is a more powerful and complex steam system with 3 pressures at higher levels and a reheat cycle.

The power plants have been simulated at off-design under different non-ambient conditions. Changes in ambient pressure & temperature, and cooling water temperature have been tested under different combinations.

The off-design calculations show a steady decline in both gas- and steam turbine output from 100% site rating until 85%. From 80% until 60% site rating on the gas turbines, the steam turbine power output steadily increases, before it falls steeply again down to 45%. From 45 % there is an increase in the steam power output again until 40%.

Increased power output for the steam turbine is due to reduced thermal efficiency for the gas turbines at part load which increases the exhaust developed per kWh produced. This energy is further utilized in the steam turbine, making possible to maintain the power output of the steam turbine. At 60% the steam turbine experience worsening efficiency itself combined with decreased part load operation on the gas turbine, and therefore shows a declining power output.

Electric power produced by the steam turbine, has consequently minor variations in power delivered from 100- 60% site rating on the gas turbines. This characteristic has proven to be advantageous and is making it unnecessary to fire up a spare generator under part load operation.

At 100% site rating the net steam turbine power output has a difference between the “best”- and “worst” case due to changing ambient conditions of 5425 kW (41960 kW - 36535 kW) for the cycle without reheat. For the steam cycle with reheat, the “best” case gives 44058 kW at 100% site rating, while the “worst” case have a power output 37982 kW. This gives a net power output difference of 6076 kW. The steam cycle without reheat delivers enough power to maintain 100% production and thrusters working at part load, but there is power lacking when offloading under unfavorable conditions. The cycle with reheat has enough power to all situations, and also to deliver enough under most unfavorable conditions.

A simplified equipment cost estimate calculated by GT PRO shows that the steam cycle without reheat will cost: 556.1 USD/kW and with reheat: 559.1 USD/kW. The extra costs connected to the complex steam cycle are not considerable, and this system also offers extra flexibility at off-design operation under non-favorable conditions. The steam cycle with reheat will further be recommended with the reservations that there is enough space onboard due the larger space requirements of the HRSGs.

The HAZOP (Hazard and Operability) study points of some important safety aspects by implementing a direct-drive COGES system in the process trains of the LNG FPSO. Higher steam pressure and larger steam flows increases the danger in case of a boiler explosion or leakage. Further there are some safety issues around having fired machinery in the process trains.

However by careful design and awareness of these safety aspects, the writer believes that a COGES system is justifiable.

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Conversations

Harald Valland	– NTNU, Department of Marine technology
Lars Petter Revheim	– Höegh LNG
Roy Scott Heiersted	– Höegh LNG
Paul Andreas Marchioro Ystad	– NTNU, Department of Energy and Process Engineering
Patrick R. Griffin	– Vice President Thermoflow
Maher Elmasri	– President Thermoflow

Appendix A Process diagram option 3

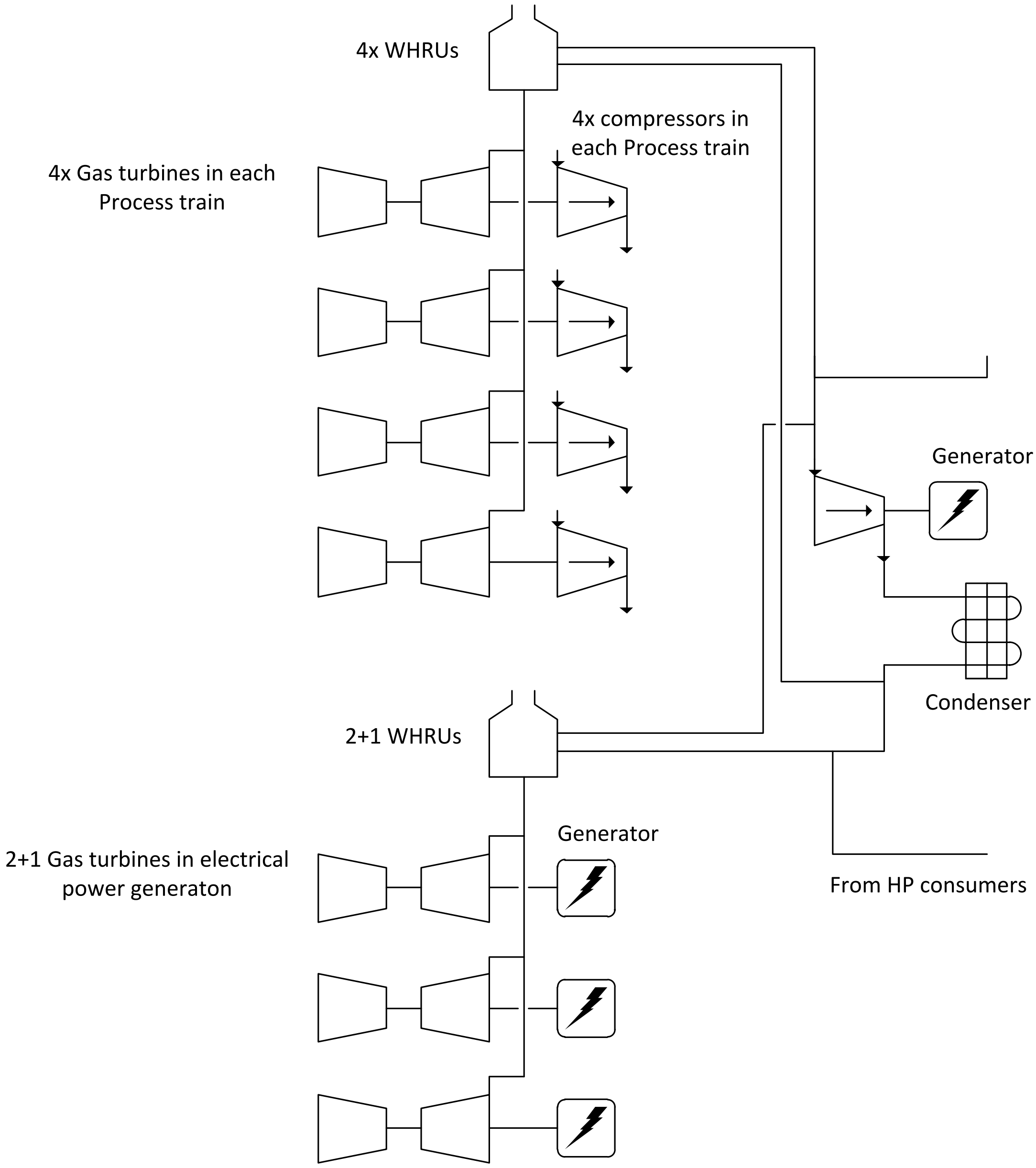

Figure A-1 Layout of option 3 from project assignment

Appendix B Revised process diagram option 3

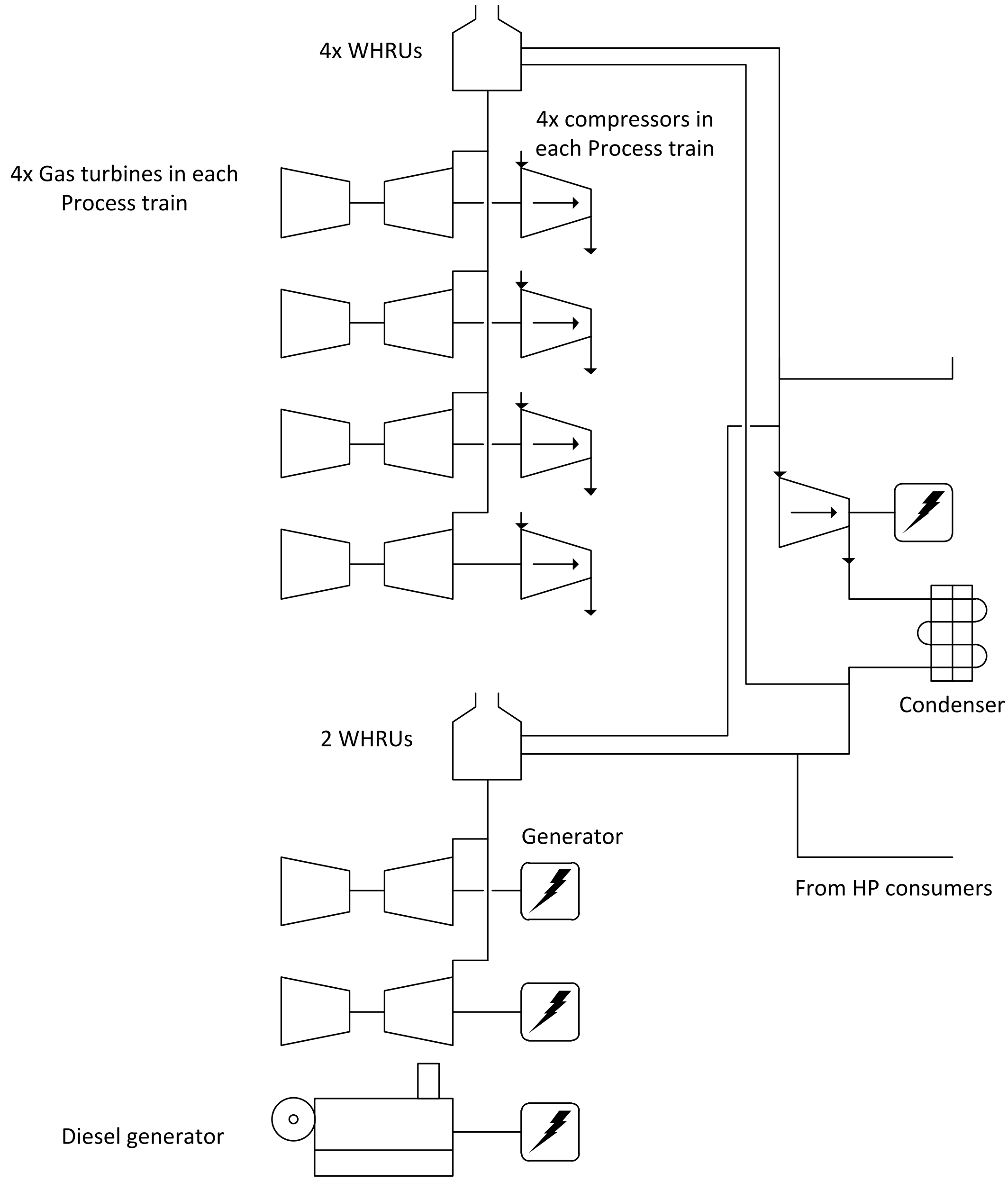

Figure B-1 Revised layout of option 3 from project assignment

Appendix C Data for steam cycle without reheat

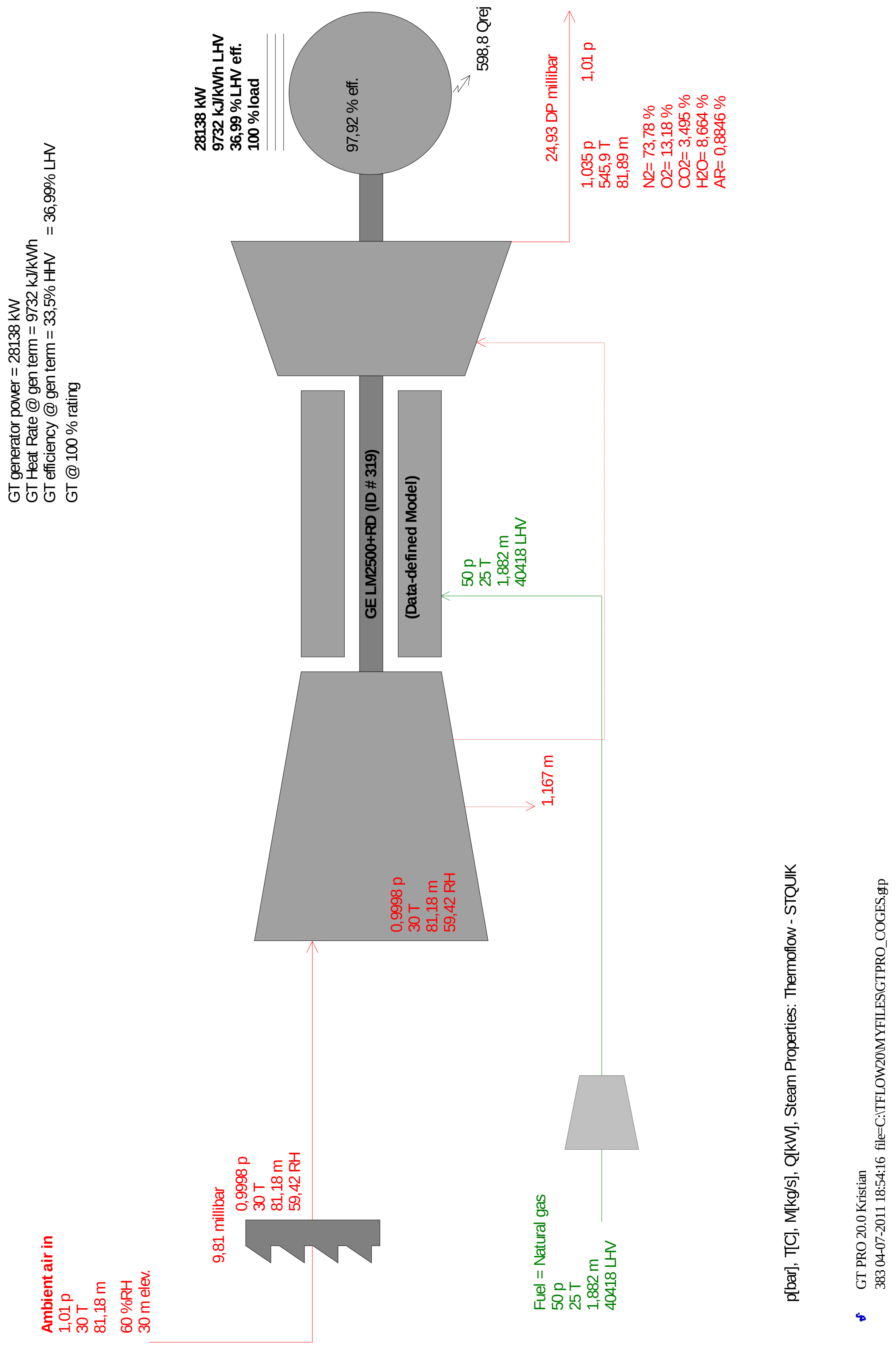

Figure C-1 Schematic drawing for GE LM2500+RD for power plant without reheat

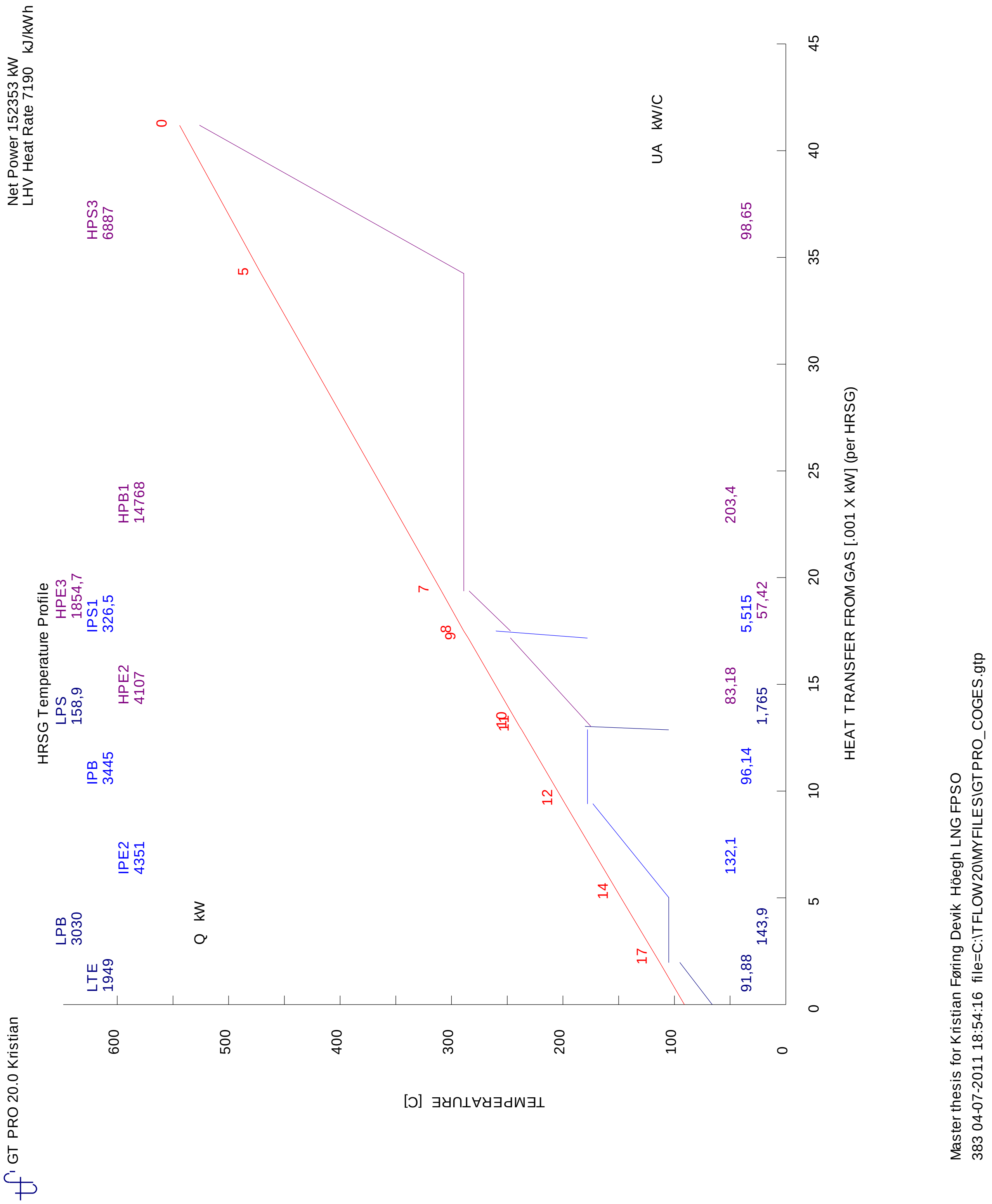

Figure C-2 HRSG temperature profile for power plant without reheat

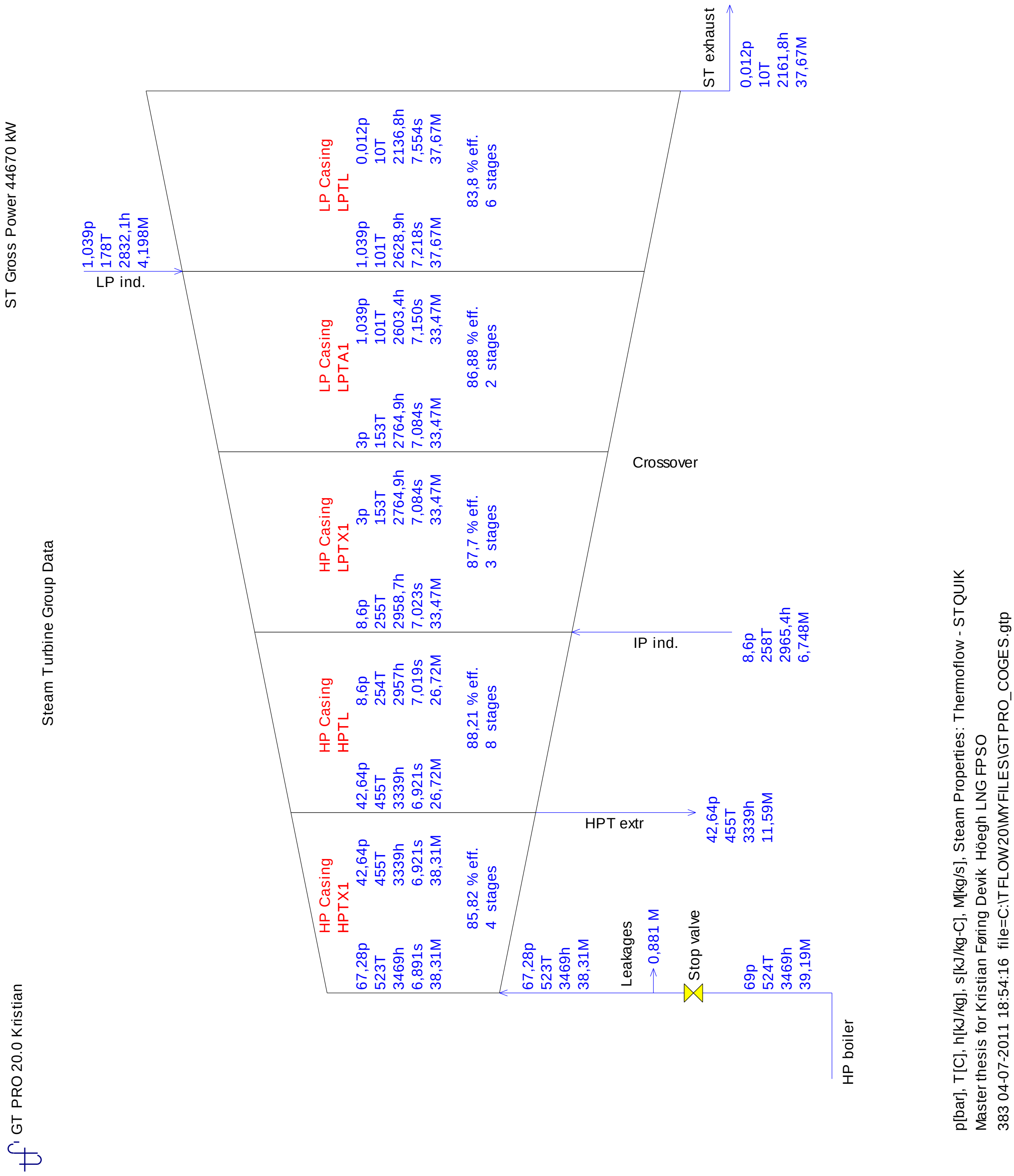

Figure C-3 Steam turbine group data for power plant without reheat

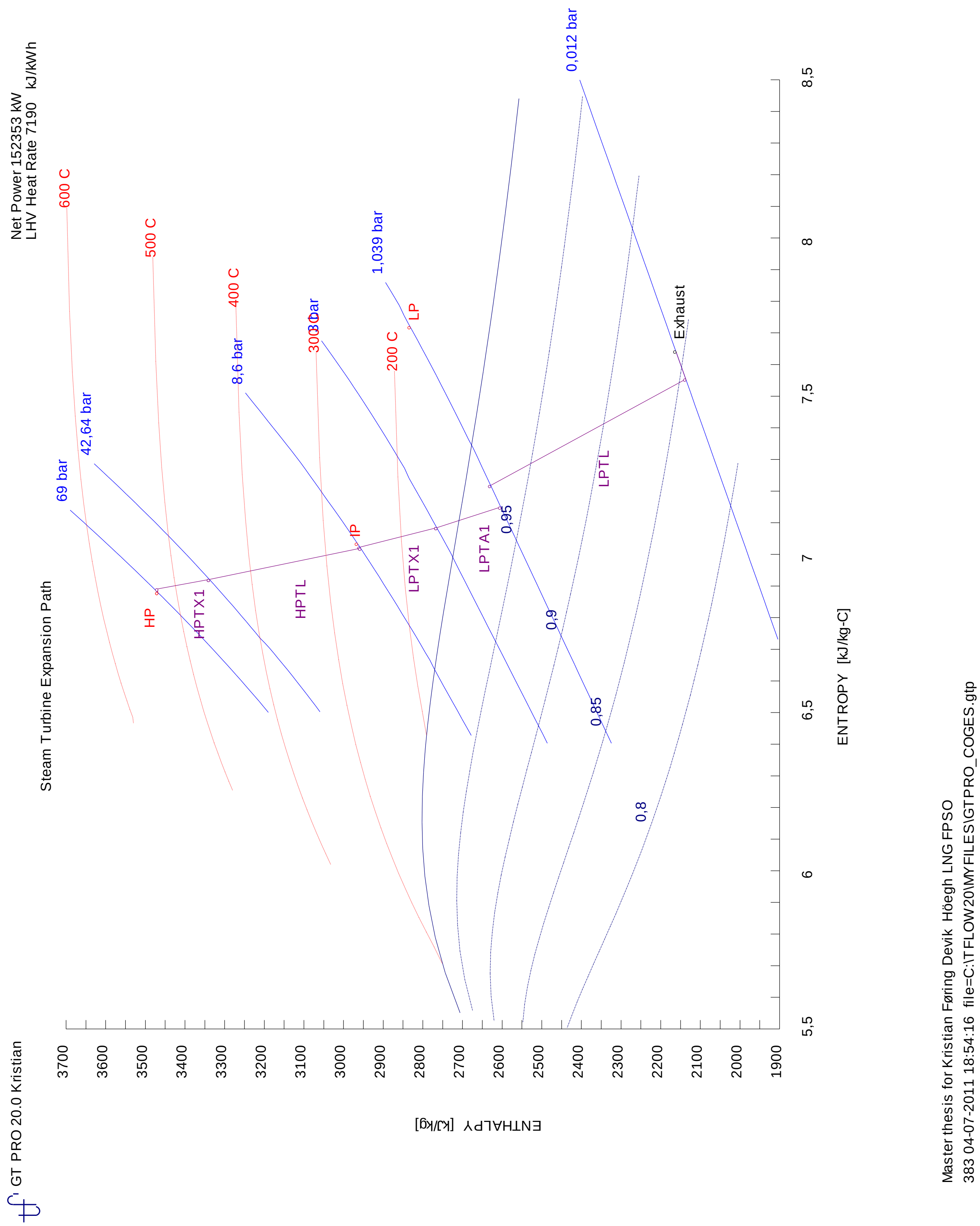

Figure C-4 Steam turbine expansion path for power plant without reheat

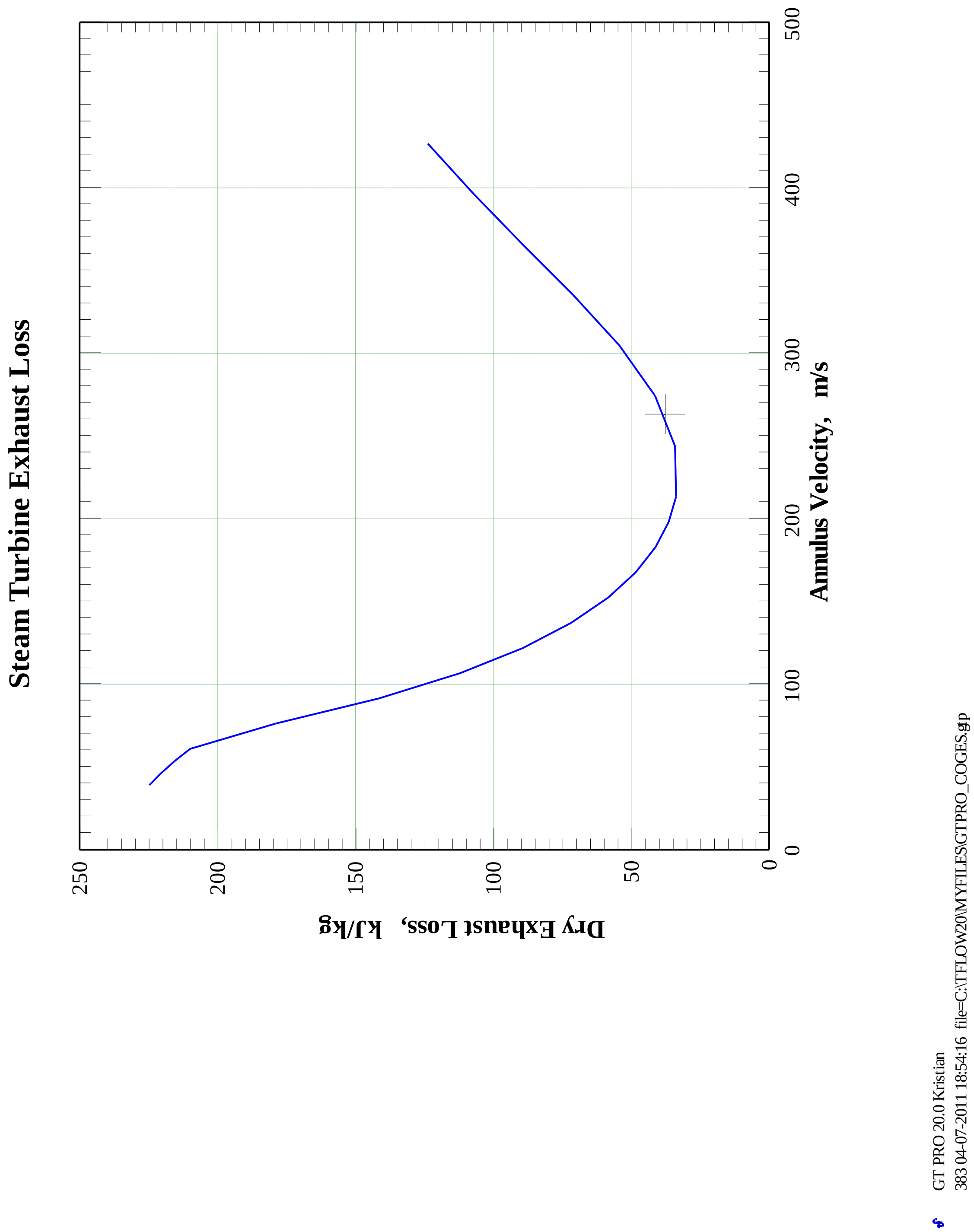

Figure C-5 Steam turbine exhaust loss for power plant without reheat

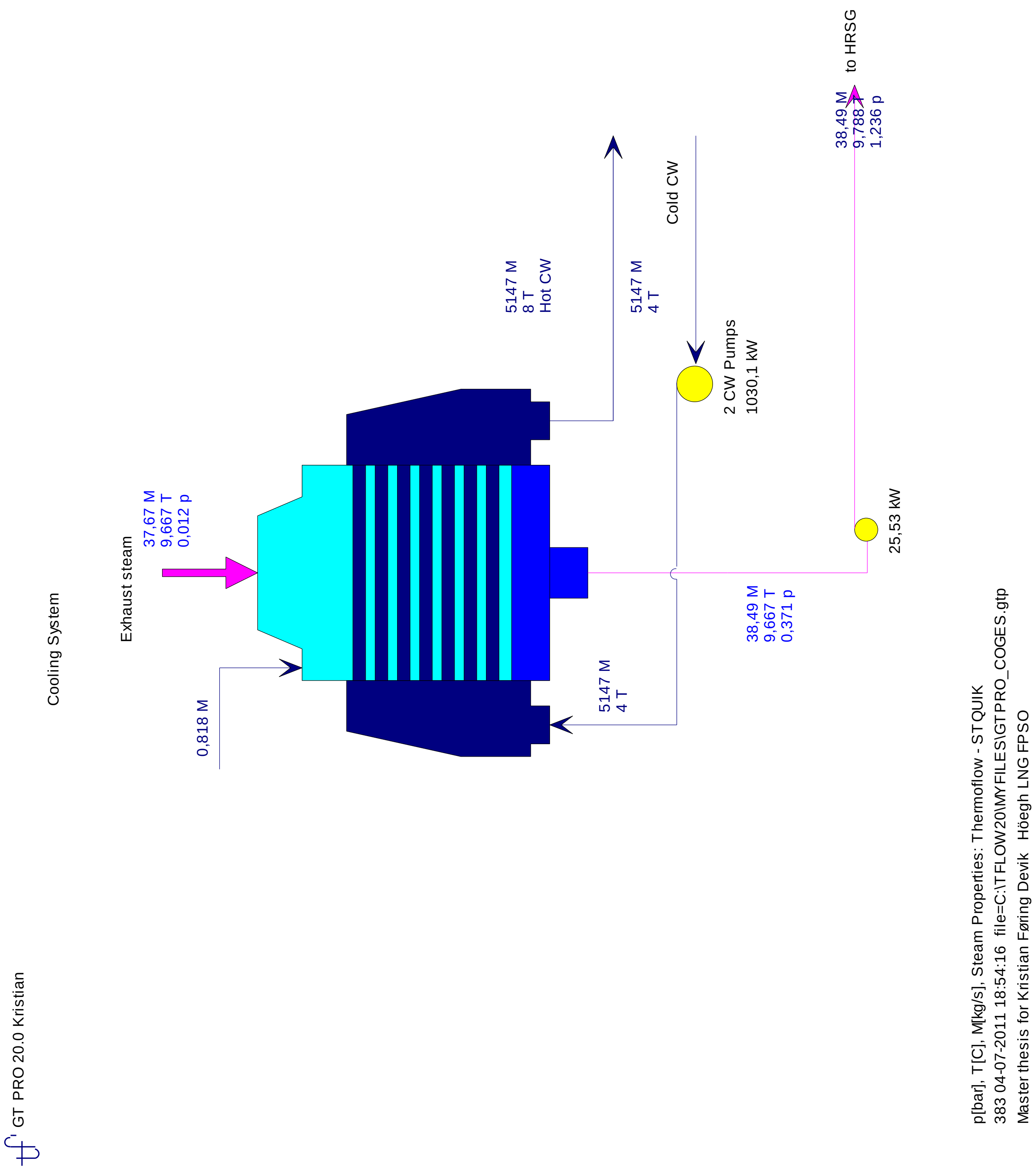

Figure C-6 Cooling system for power plant without reheat

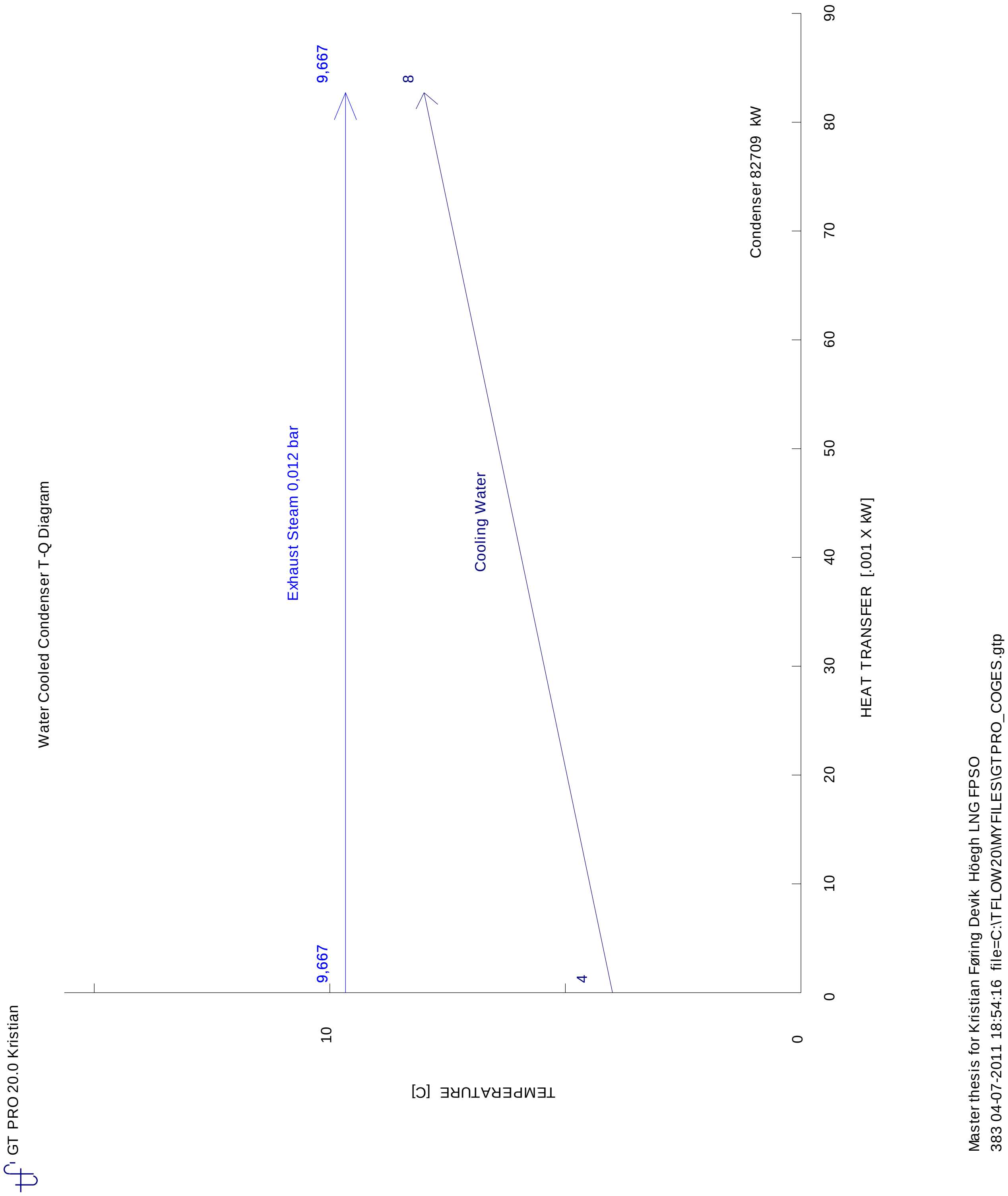

Figure C-7 Water cooled condenser T-Q diagram for power plant without reheat

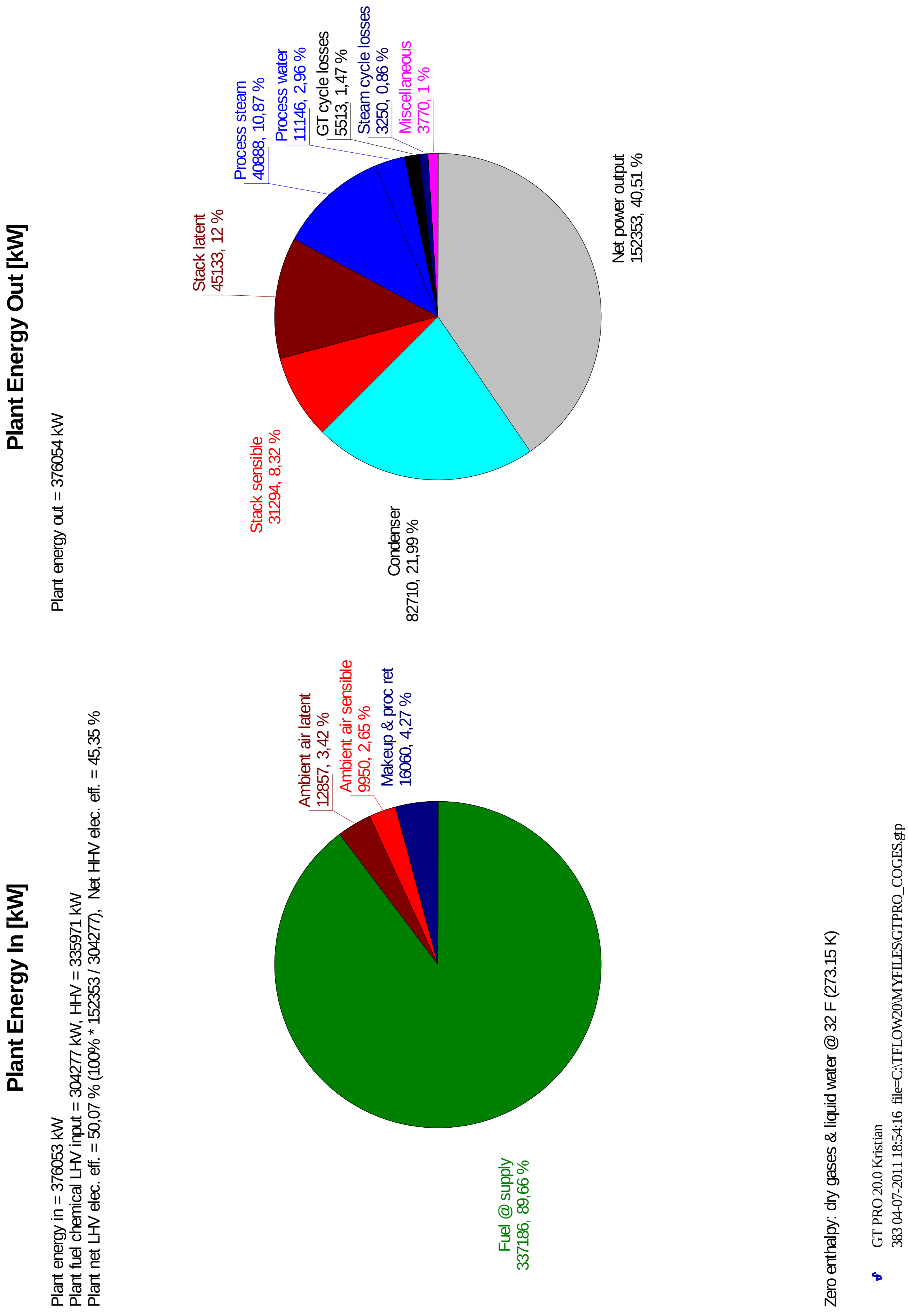

Figure C-8 Plant energy In- Out for power plant without reheat

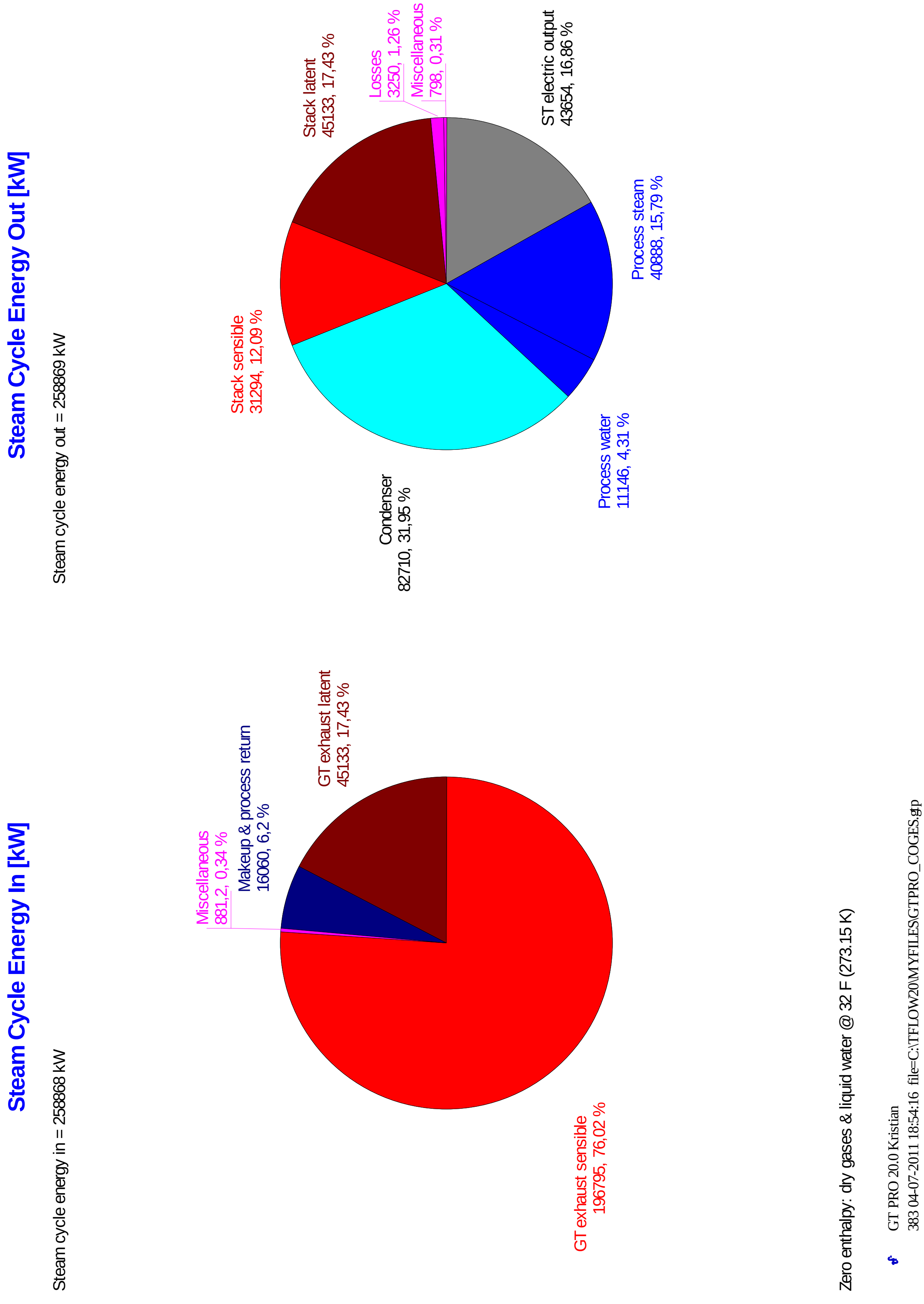

Figure C-9 Steam cycle energy In- Out for power plant without reheat

Appendix D Data for steam cycle with reheat

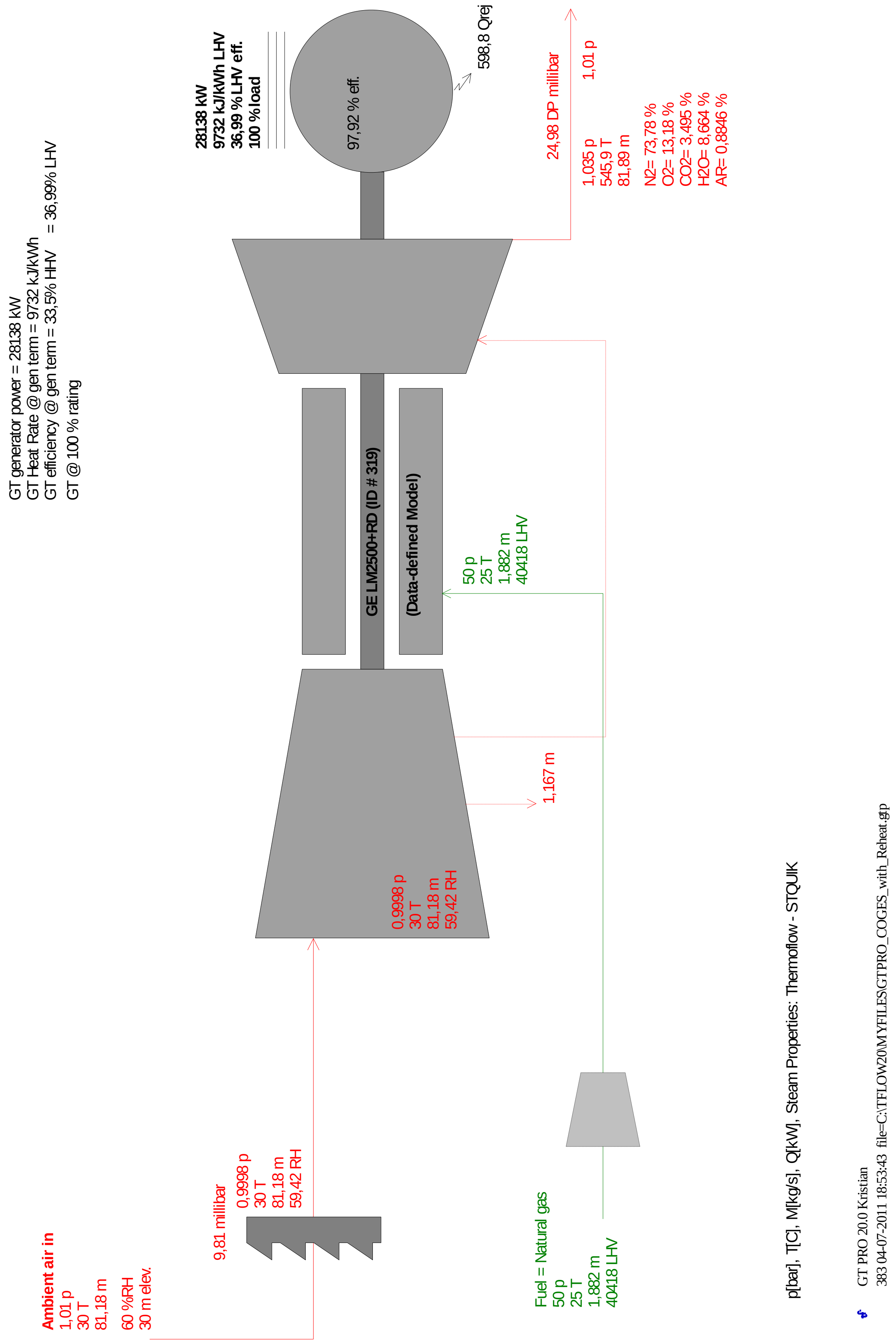

Figure D-1 Schematic drawing for GE LM2599+RD for power plant with reheat

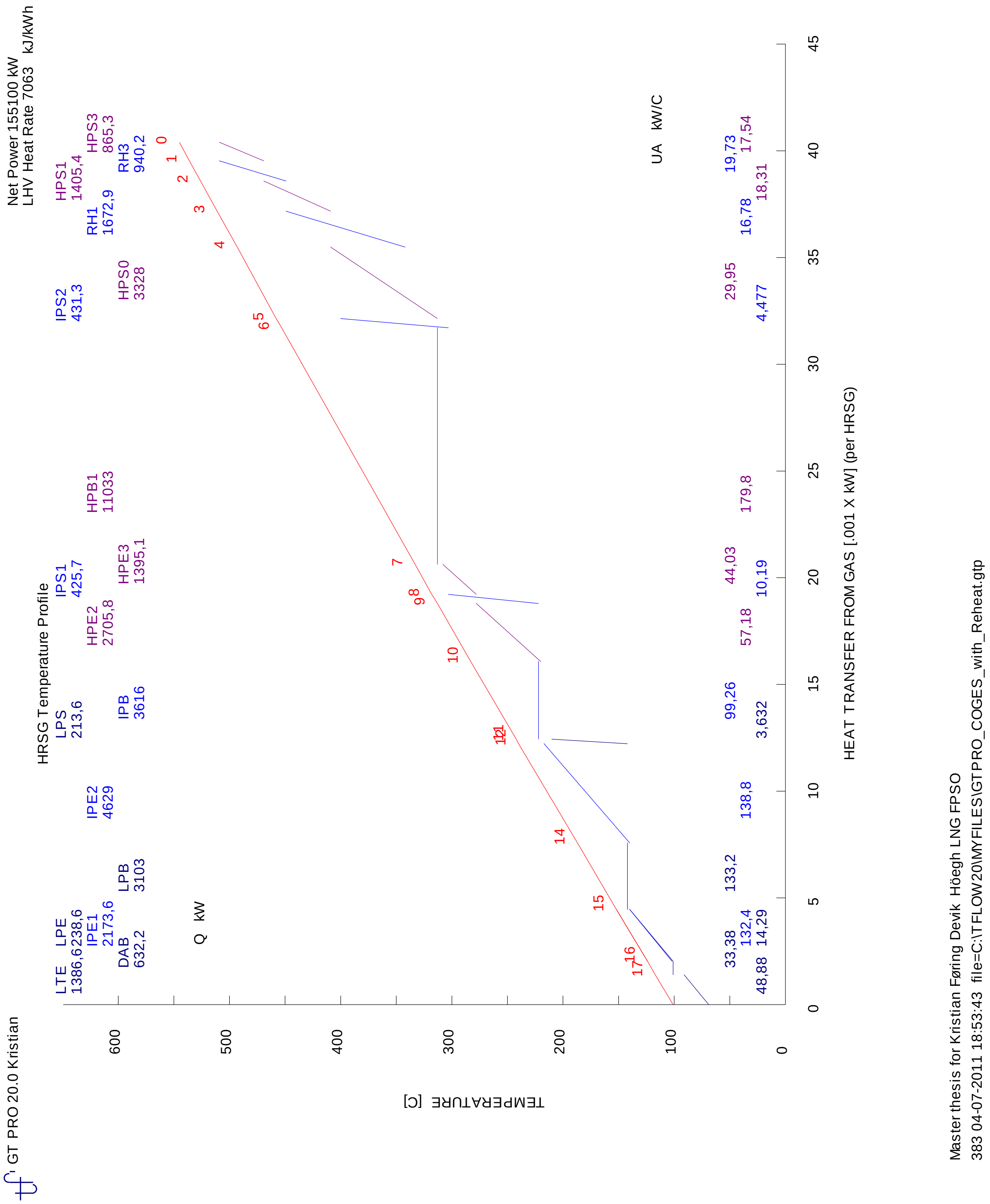

Figure D-2 HRSG temperature profile for power plant with reheat

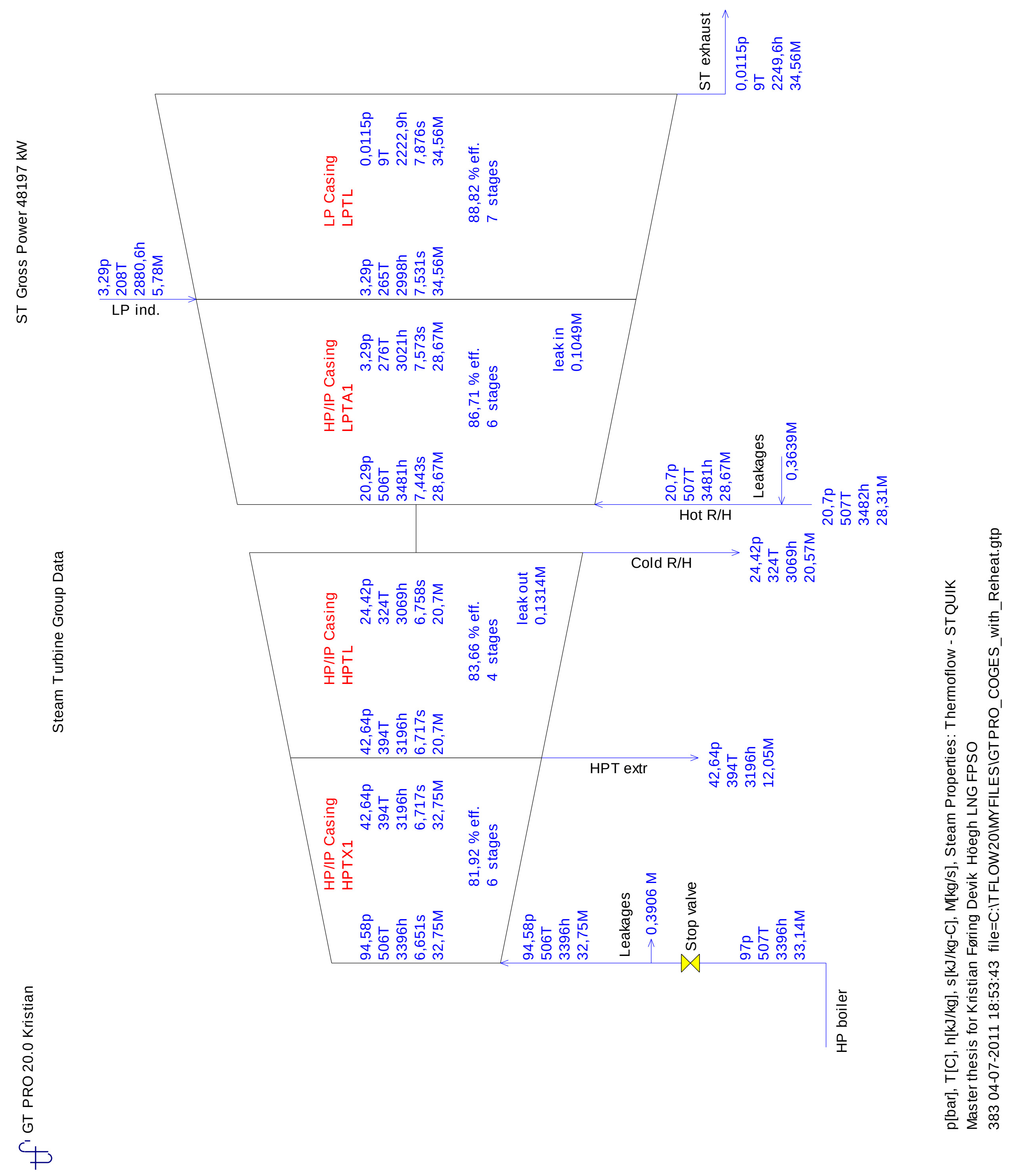

Figure D-3 Steam turbine group data for power plant with reheat

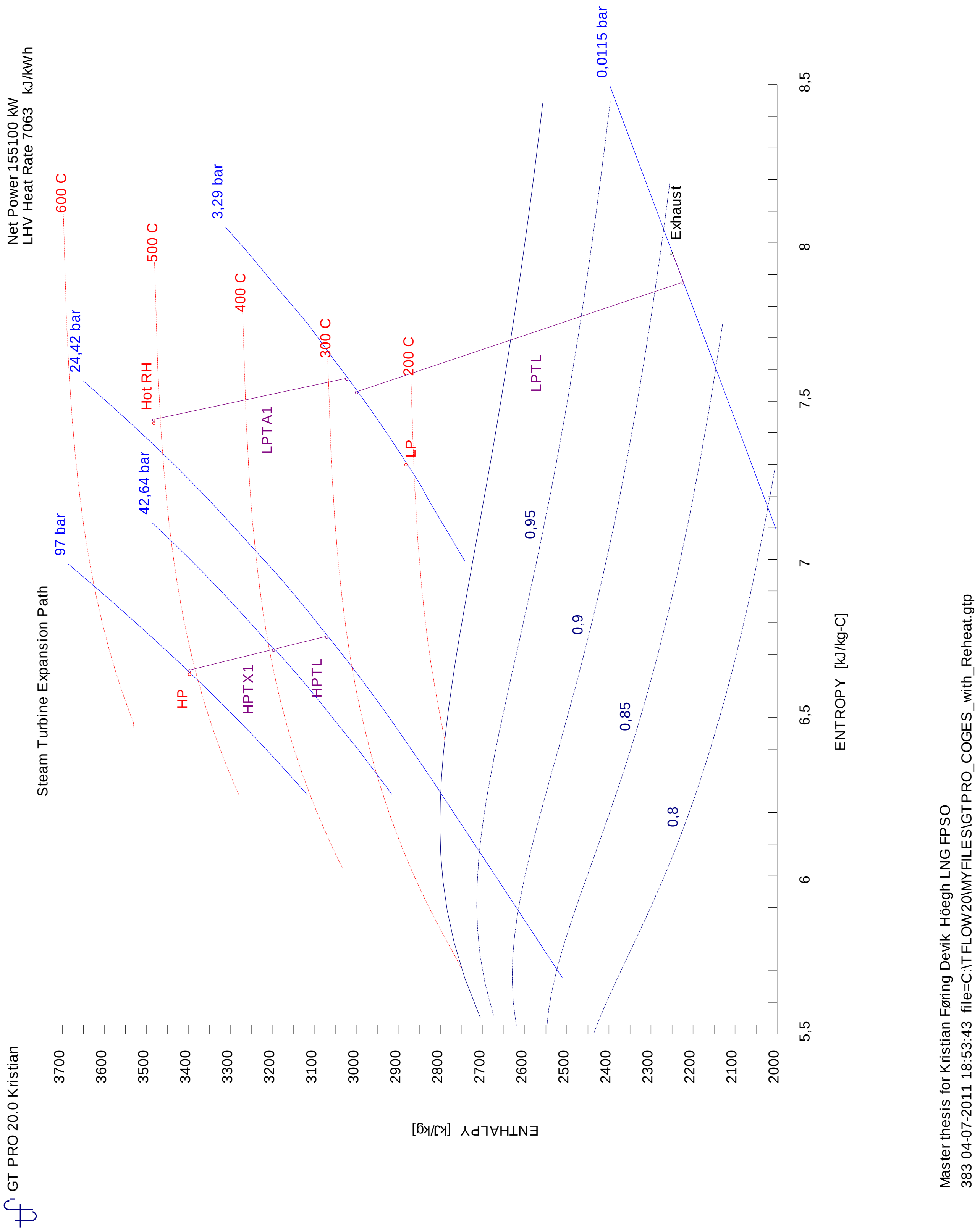

Figure D-4 Steam turbine expansion path for power plant with reheat

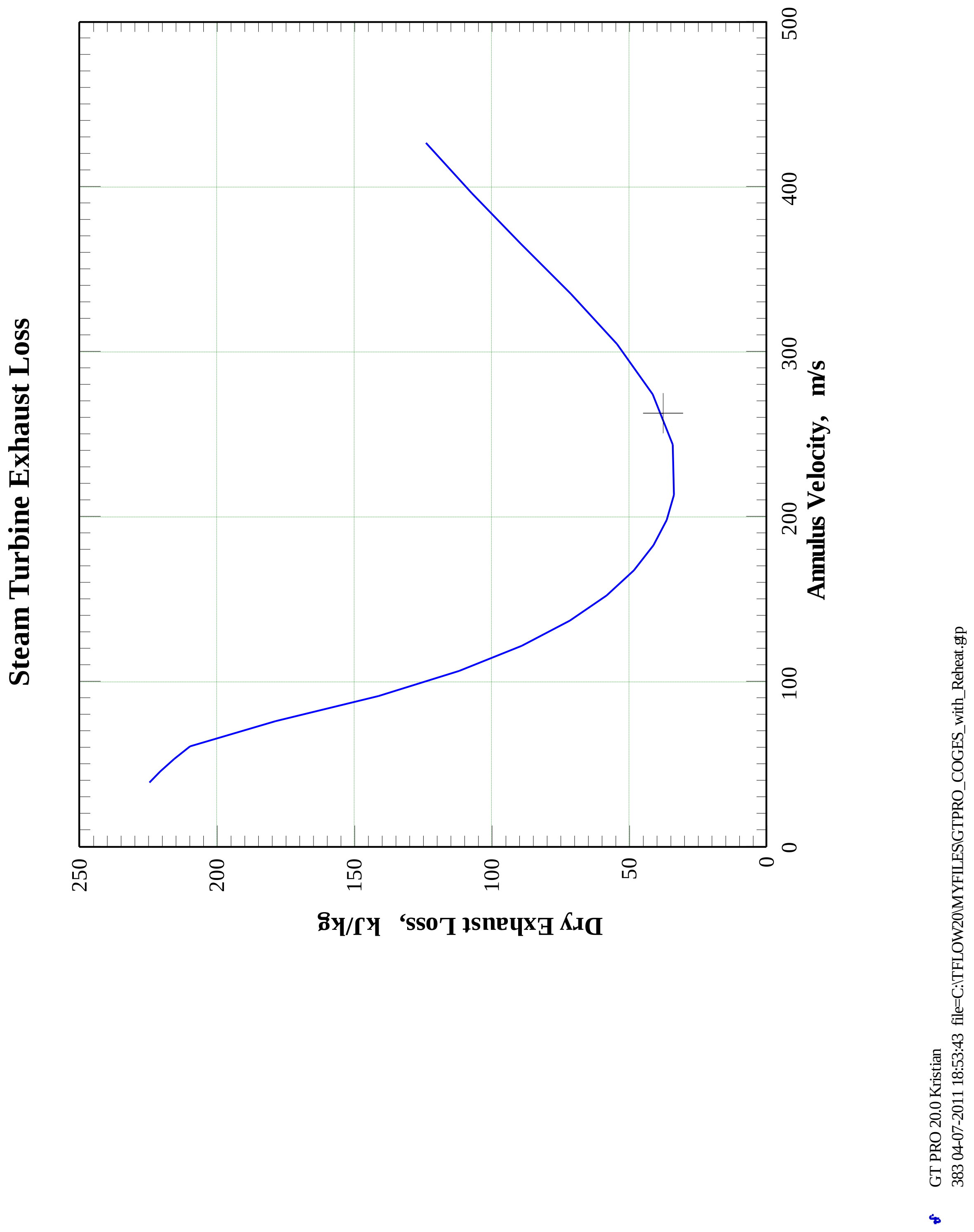

Figure D-5 Steam turbine exhaust loss for power plant with reheat

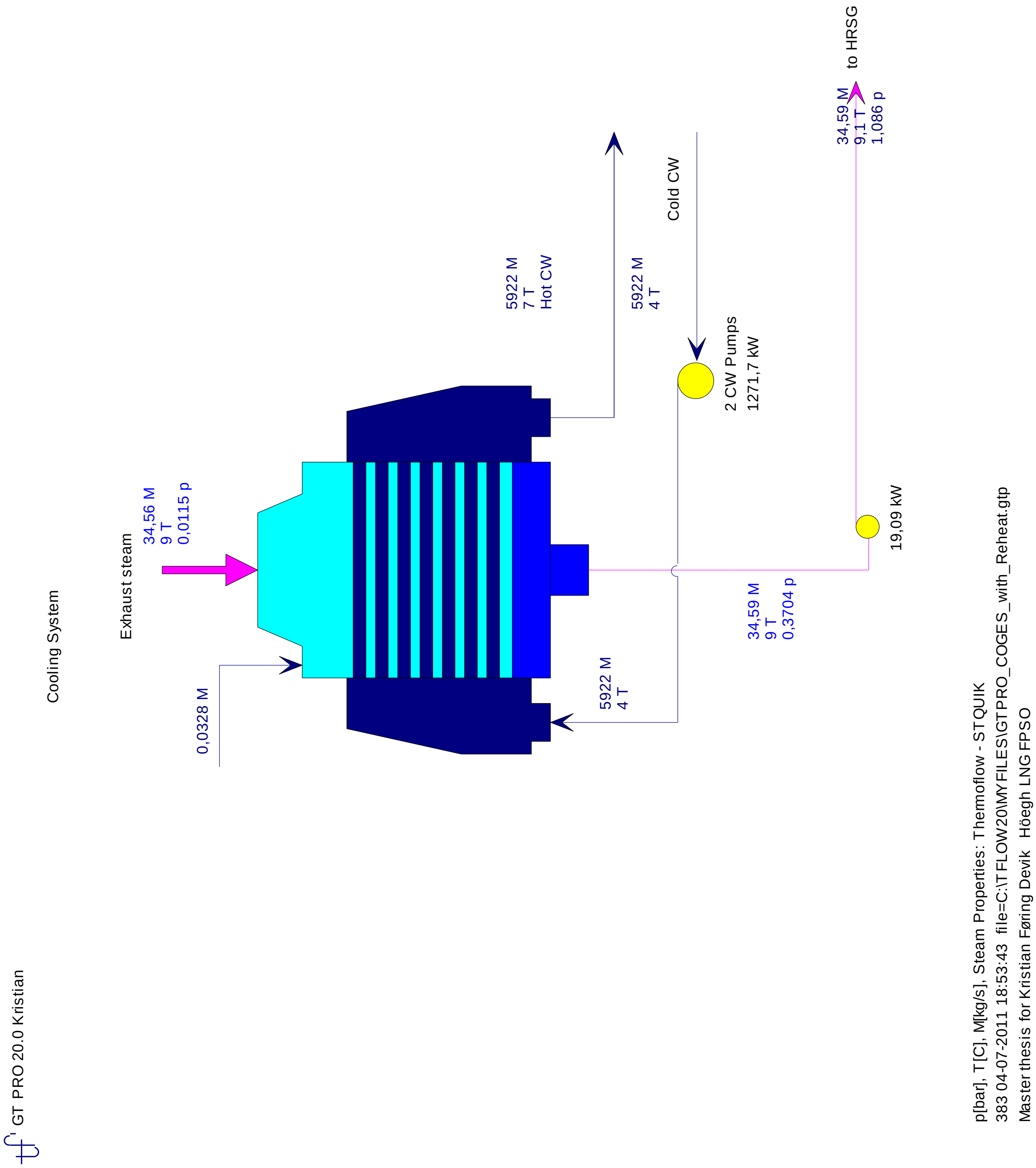

Figure D-6 Cooling system for power plant with reheat

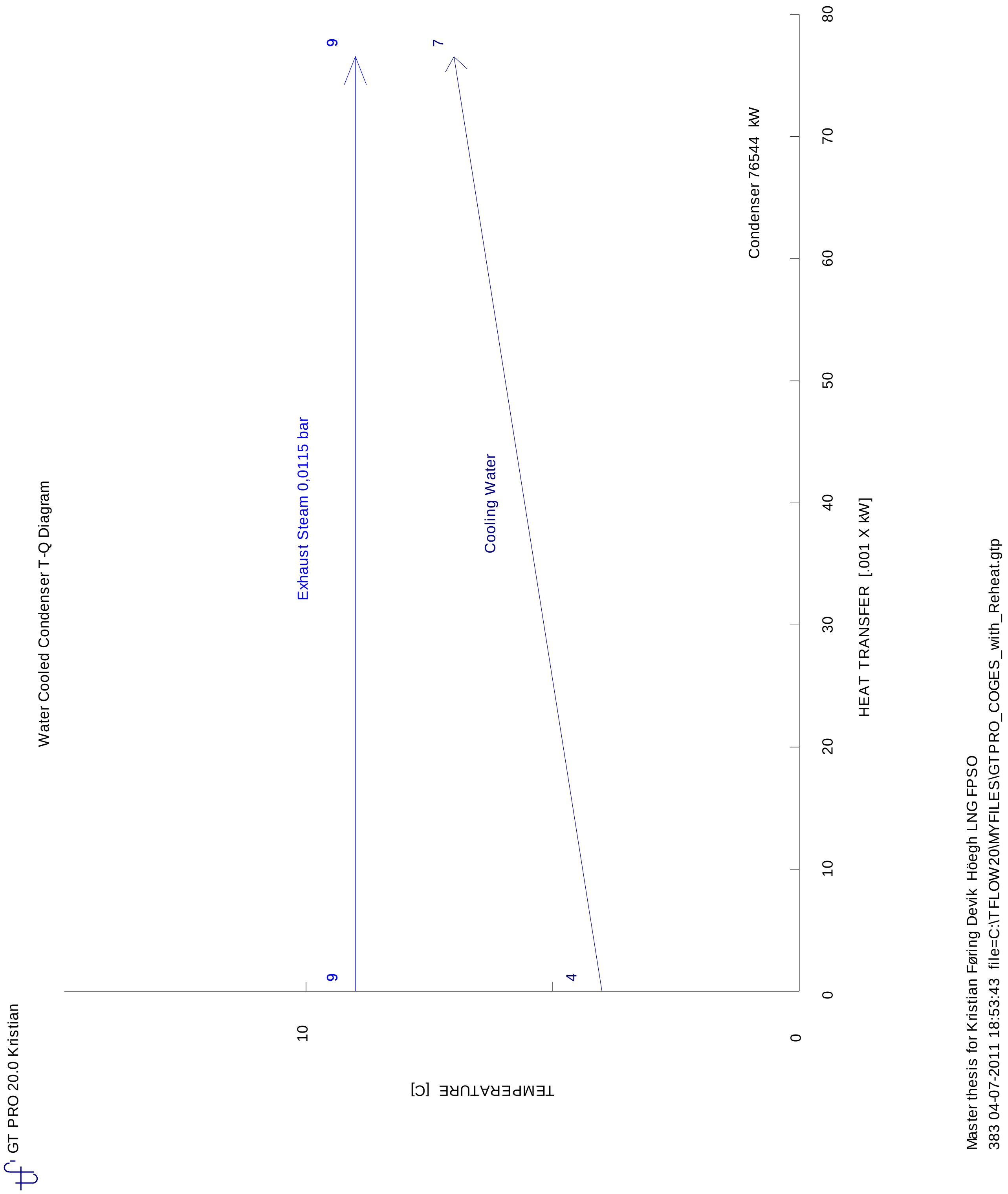

Figure D-7 Water cooled condenser T-Q diagram for power plant with reheat

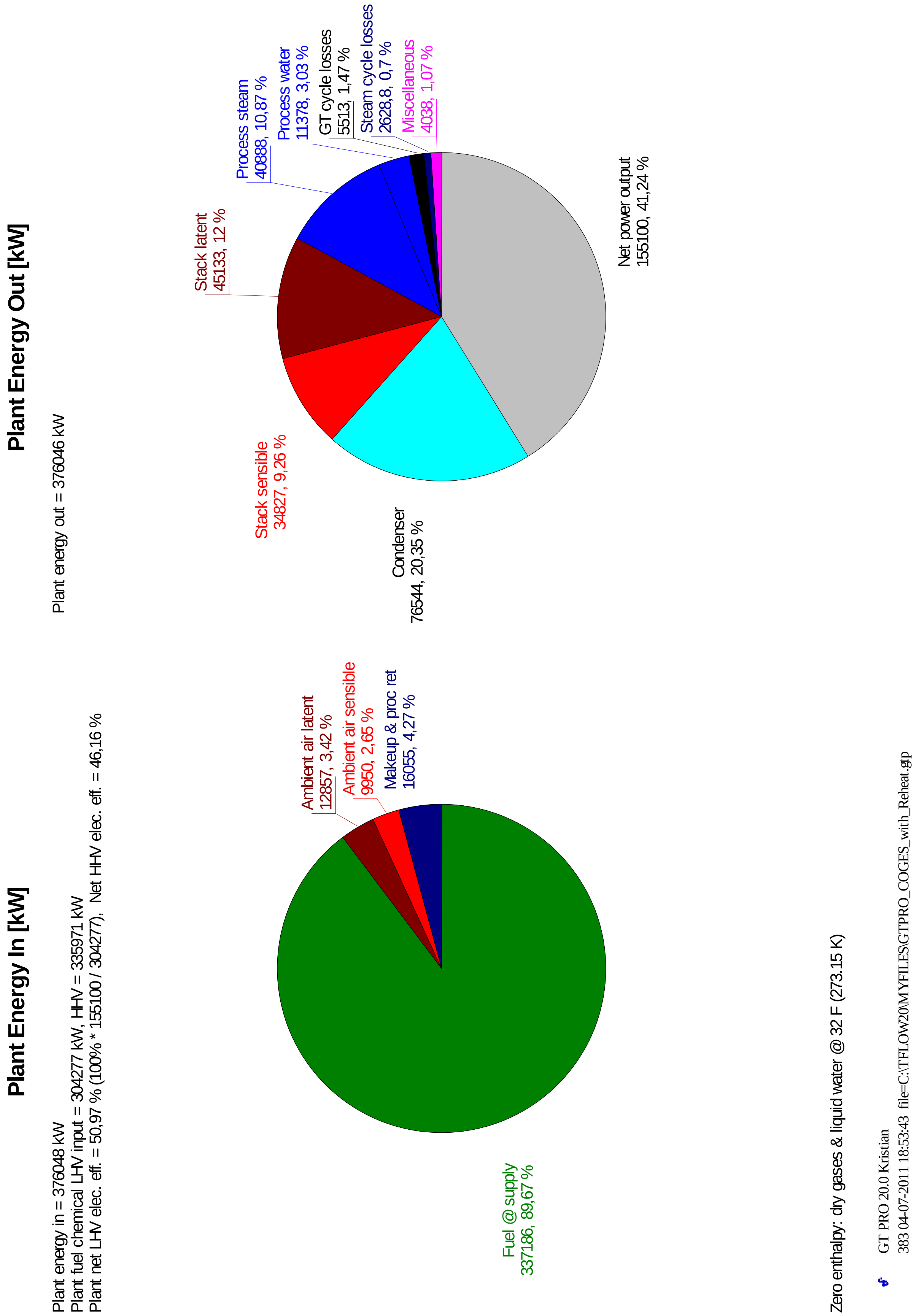

Figure D-8 Plant energy In -Out for power plant with reheat

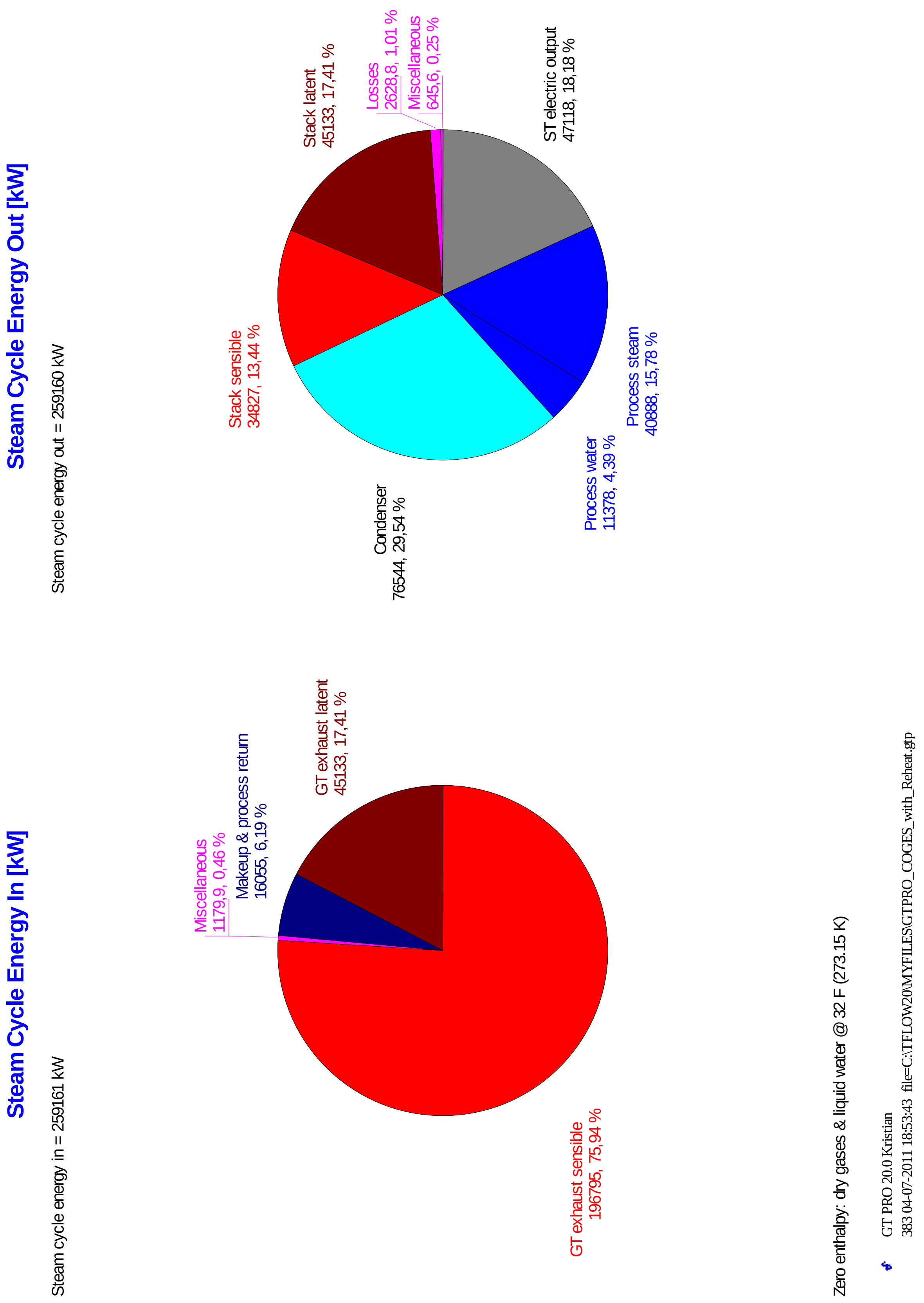

Figure D-9 Steam cycle energy In -Out for power plant with reheat

Appendix E A3 outline for steam cycle without reheat

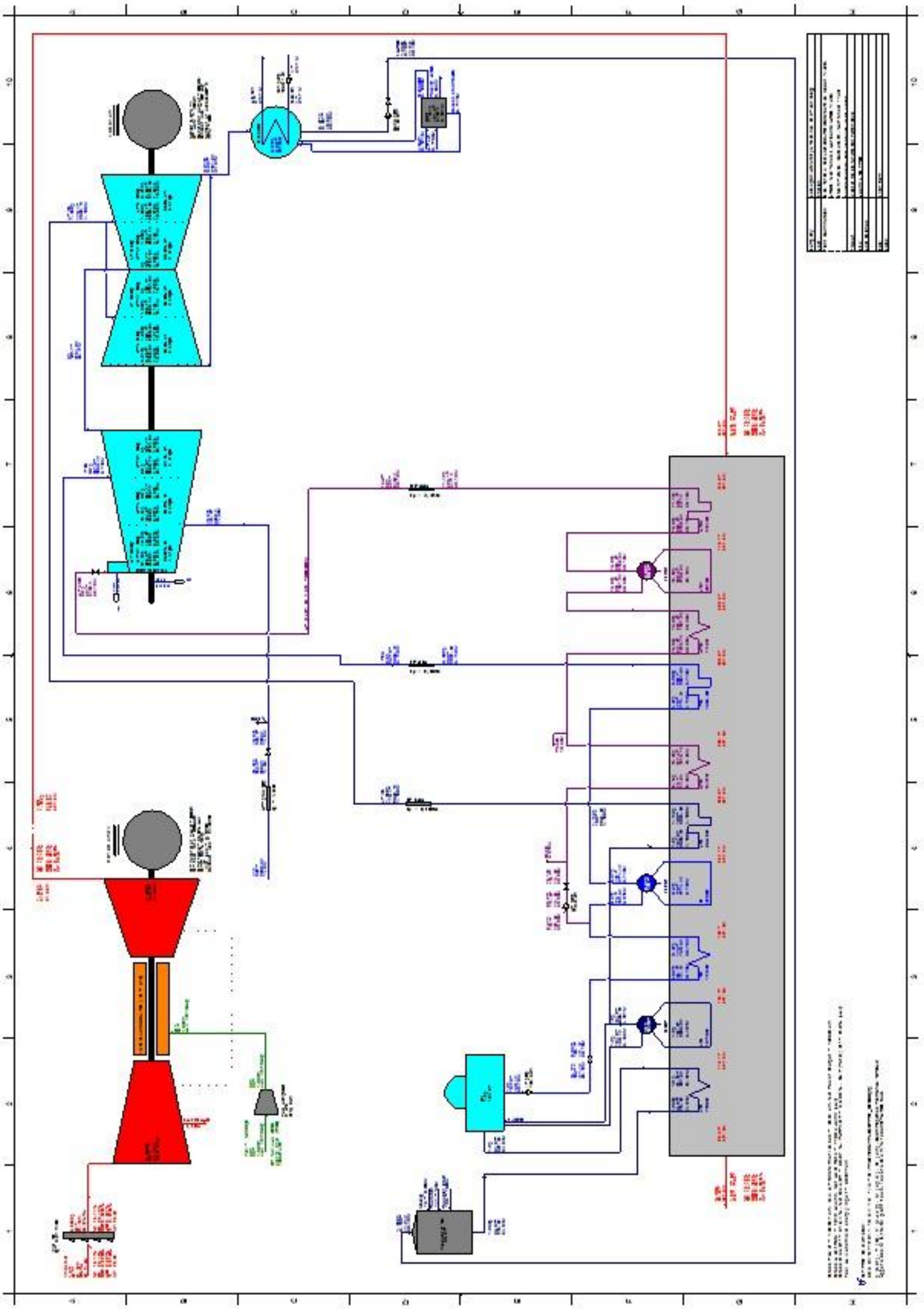

Figure E-1 Small view of outline for steam cycle without reheat

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Appendix F A3 outline for steam cycle with reheat

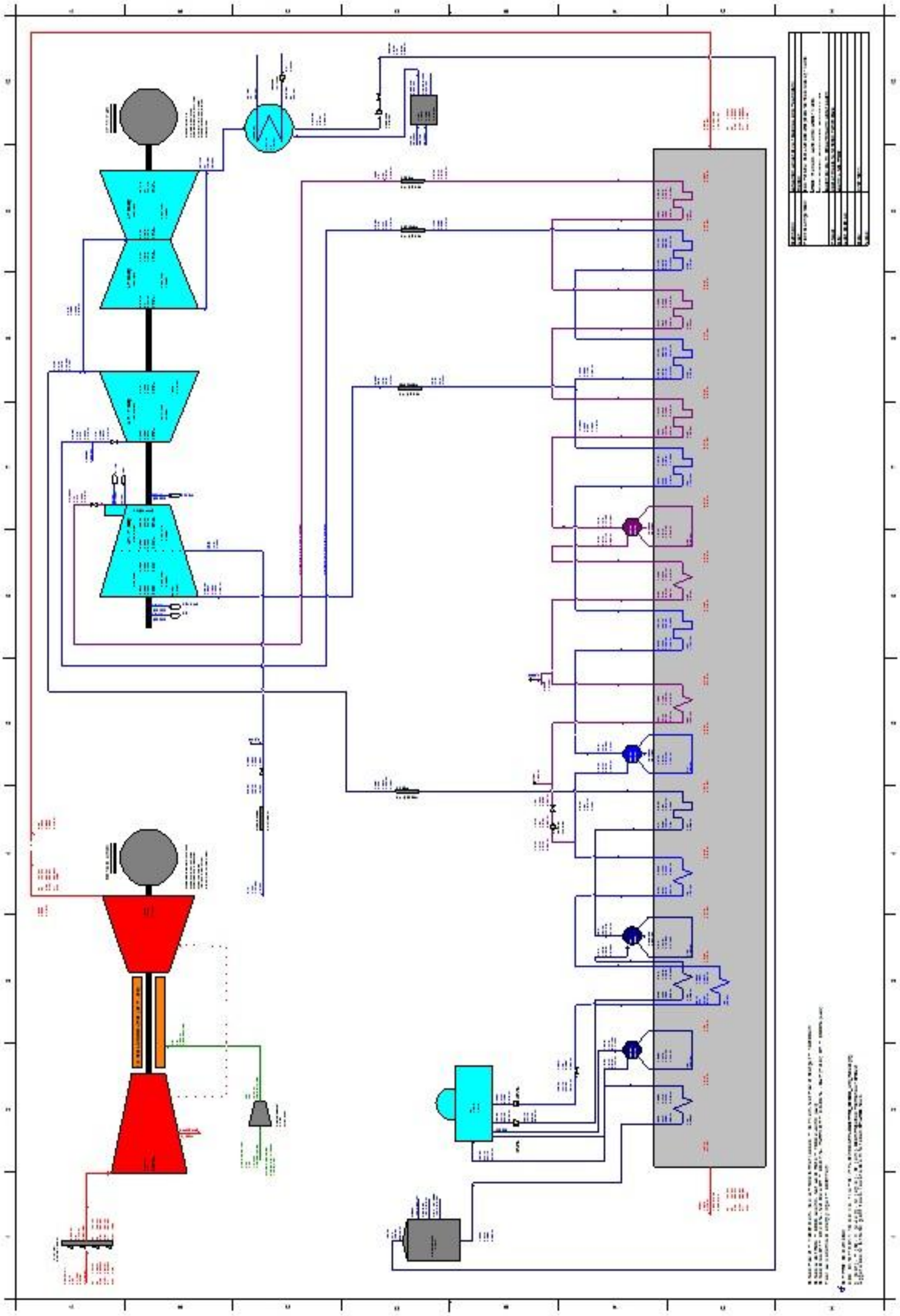

Figure F-1 Small view of outline for steam cycle with reheat

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Appendix G Preliminary HRSG Design (Hot end) without reheat

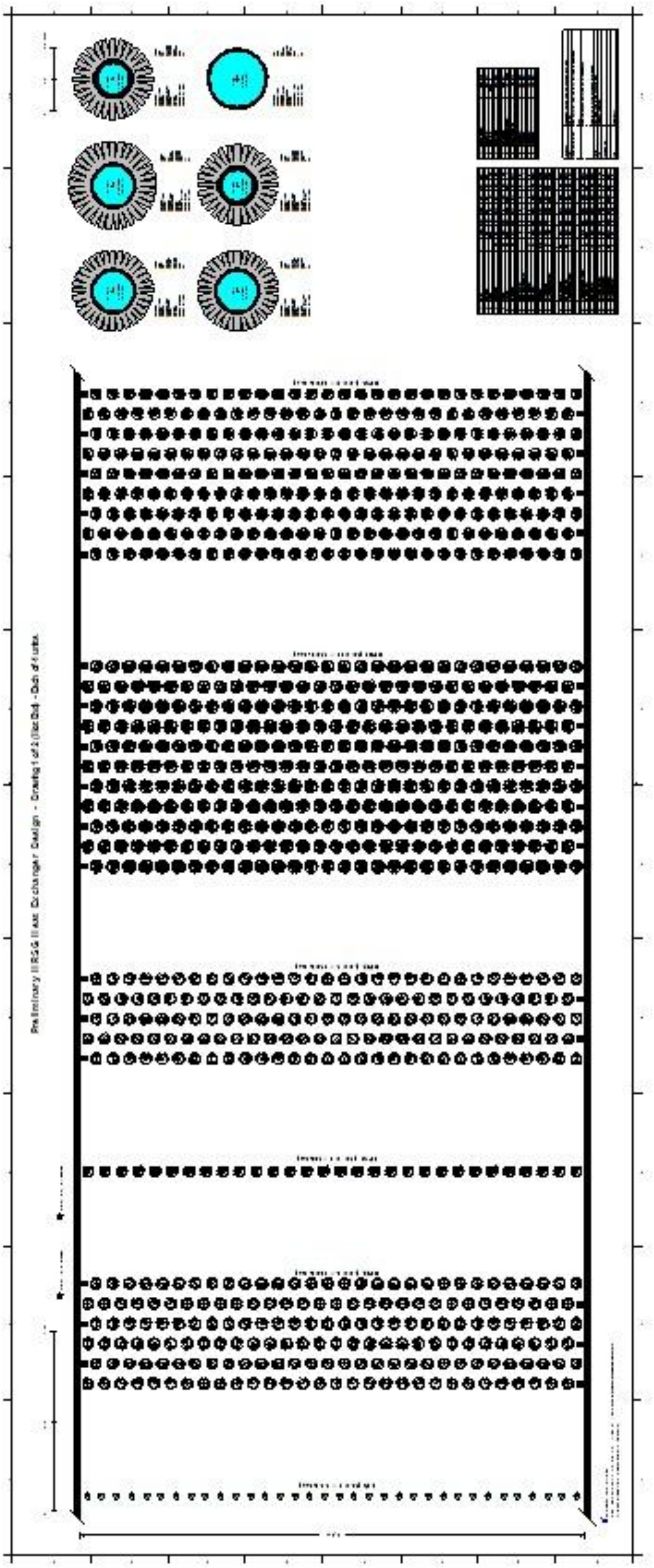

Figure G-1 Preliminary HRSG design (hot end) for steam cycle without reheat

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Appendix H Preliminary HRSG Design (Hot end) with reheat

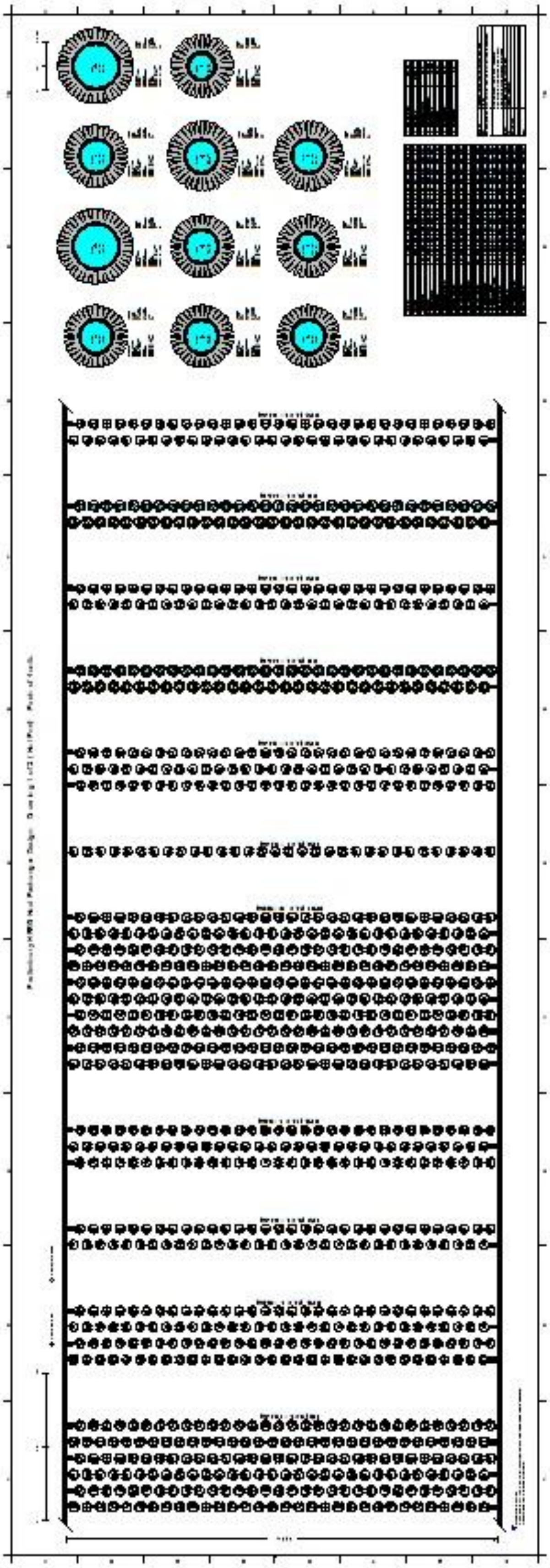

Figure H-1 Preliminary design (hot end) for steam cycle with reheat

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Appendix I Preliminary HRSG Design (Cold end) without reheat

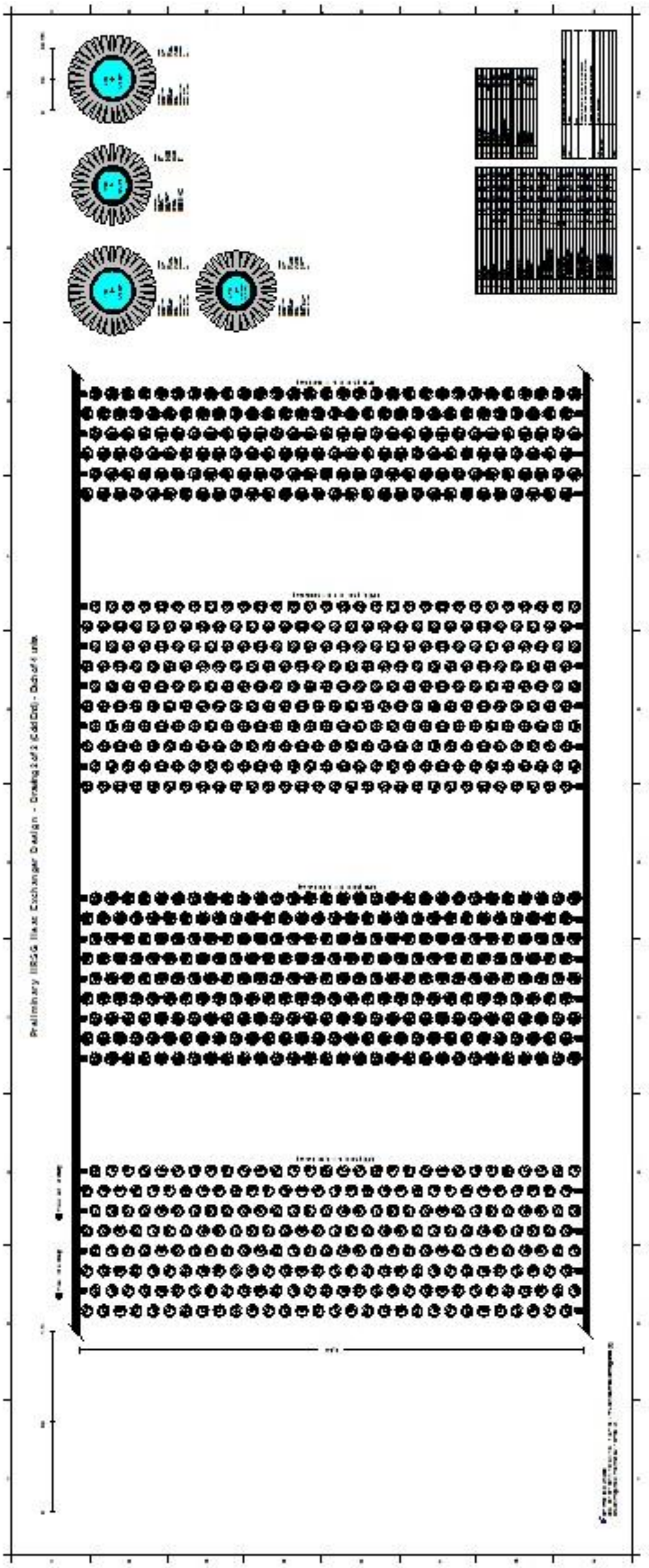

Figure I-1 Preliminary HRSG Design (cold end) for steam cycle without reheat

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Appendix J Preliminary HRSG design (Cold end) with reheat

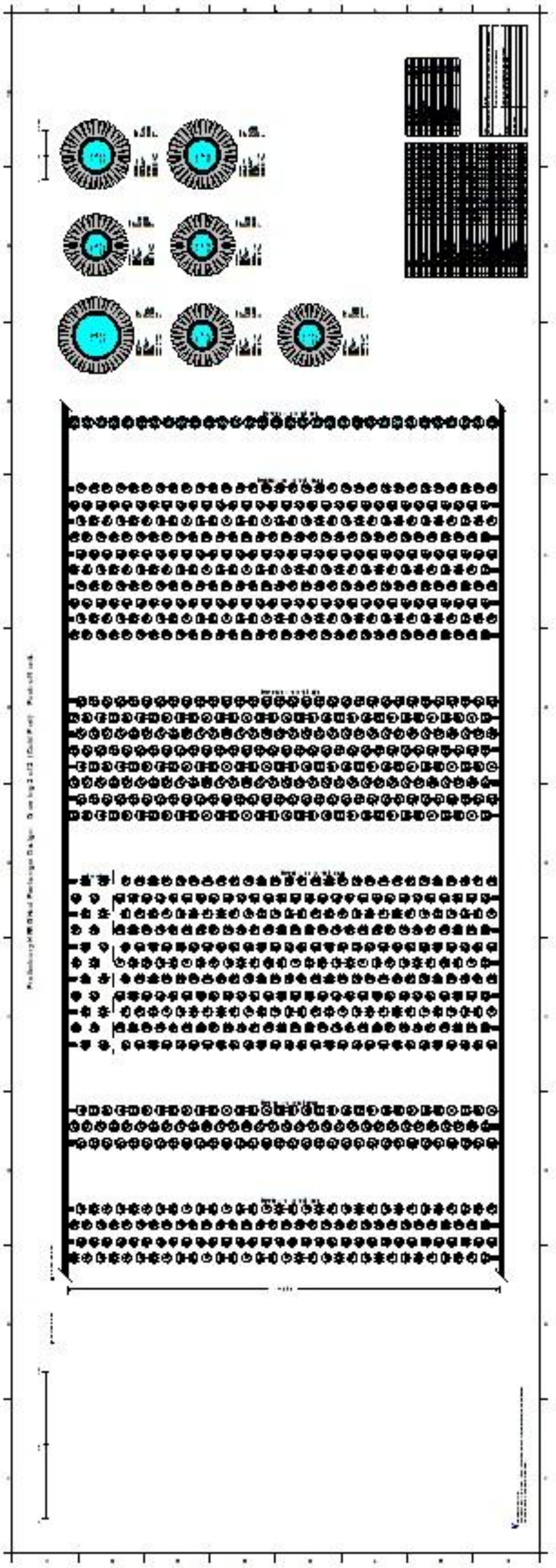

Figure J-1 Preliminary HRSG design (cold end) for steam cycle with reheat

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