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Dynamic Modeling and Process Simulation of Steam Bottoming Cycle

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Natural Gas Technology
Submission date: June 2017
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NTNU
Norwegian University
of Science and Technology

Department of Energy
and Process Engineering

EPT-M-2017-69

MASTER THESIS

for

Student Jairo Rúa Pazos

Spring 2017

Dynamic process simulation of steam bottoming cycle

Background and objective

On offshore oil and gas installations the power demand is high. Process heat is also needed. In addition, the power plant should be flexible to be able to adjust to the needs of the oil and gas processes on the platform or FPSO both short and long term. The current dominating technology is based on simple cycle gas turbines. Other options currently installed on the Norwegian continental shelf includes gas turbines with waste heat recovery units, electricity from onshore, and combined cycle gas turbine plants with steam bottoming cycle.

The main objective for the student is to design and build a dynamic process model of a steam bottoming cycle. The work will build on the specialization project where the student developed steady-state design and off-design process models including steam cycle control strategies. The focus should primarily be on load changes.

The following tasks are to be considered:

1. Literature study on transient performance of steam cycles and combined cycles.
2. Learning Modelica/Dymola for dynamic process modeling and simulation.
3. Development of a physical model of a combined cycle in Modelica/Dymola based on components from a selected Modelica library, preferably the ThermoPower library.

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

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

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

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

Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

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

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

☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, 18. January 2017

Lars O. Nord
Academic Supervisor

Research Advisor: Rubén Mocholí Montañés (PhD Candidate, NTNU)

Preface

This Master thesis is submitted as a partial fulfillment of the requirements for a M.Sc. in Natural Gas Technology at the Norwegian University of Science and Technology (NTNU). The work was carried out at the Department of Energy and Process Engineering at the Faculty of Engineering Science and Technology, with Associate Professor Lars Olof Nord as supervisor. Ph.D. Canditate Rubén Mocholí Montañés was appointed co-supervisor.

Abstract

Steam Rankine cycles are an efficient and mature technology for heat recovery of the energy contained in the exhaust gases of gas turbines. Offshore oil and gas facilities currently employ simple gas turbine cycles where the heat in the exhaust is dropped into the atmosphere. Therefore, an increase in the efficiency and a reduction in the fuel consumption and its associated greenhouse emissions could be achieved if steam bottoming cycles were installed together with the gas turbines employed in the offshore platforms. Low weight and high operation flexibility are constraints to the power generation systems on these facilities, and a trade-off among compactness, flexibility and high efficiency must be achieved.

In this work, a flexible combined heat and power plant with a steam Rankine bottoming cycle is proposed as a suitable alternative for the power generation system of an oil and gas offshore facility. The power system should be able to produce simultaneously 58 MW of power and 52 MW of heat at 150 °C, and to respond rapidly to the sudden changes of load that may be expected. The nominal operating conditions and the sizes of the components integrating the thermal power plant were obtained from a multi-objective optimization where maximum efficiency and minimum weight were the objective variables. A dynamic model of the combined heat and power plant was developed in the Modelica language by means of the software Dymola and the specialized library ThermoPower. Specific components were expressly programmed for this purpose due to the special requirements of the power system in terms of heat and power demand. This model was validated with both design and off-design steady state data generated by the software Thermoflex. Dynamic simulations were carried out in order to test the correct performance of the developed models and their ability to predict the dynamic behavior of a thermal power plant. Preliminary results of the transient performance of the proposed combined heat and power plant for a sudden gas turbine change of load were obtained, both in open-loop and with a control structure implemented. It was found that the power plant model was able to predict the dynamic behavior that could be expected for a reduction in the gas turbine load, proving the feasibility of the developed models to be utilized in a deeper assessment of the transient performance of thermal power generation systems.

Acknowledgments

First I would like to express gratitude to Associate Professor Lars Olof Nord, Ph.D. Candidate Rubén Mocholí Montañés and Postdoctoral Fellow Luca Riboldi for the help provided throughout these five months. Their guidance during this period has been precious.

Many thanks to Roberto for all the endless, interesting and sometimes productive discussions. We have taken our patience limits to new levels but we cannot say it has not been worth it.

Mother, father and sister, Ángela, for their unconditional support and love throughout these two years far from home, and for always showing interest in my life and work. *Morriña* has always been present in me but the long and close calls have made me feel at home.

Special thanks to Marta. You are always there when I need you.

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Chapter 1

Introduction

1.1 Motivation

Fossil fuels are the main contributors to the energy sector in Norway, albeit the clean electricity that is produced due to the hydropower plants installed over the country. The oil and gas sector is responsible of an important share of this fuel consumption. Simple power generation systems with poor performances and large energy wastes are currently installed in the oil and gas facilities, leading to excessively high greenhouse emissions.

Waste heat recovery applications are regarded as feasible applications to improve the energy utilization in these facilities and to reduce the associated emissions. Gas turbine combined cycles are considered the most promising alternative, but the continuous fluctuations in the operation of the offshore platforms suppose a significant challenge for these power generation systems. Thus, the study and analysis of the combined cycle's dynamic performance becomes fundamental for the installation assessment of these thermal power plants.

1.2 Objectives

For this master thesis work, the objectives were:

- Literature study on transient operation of combined cycles, focusing on the utilization of water/steam as working fluid and waste heat recovery applications.
- Learning the modeling and programming language Modelica, its main advantages for dynamic modeling and simulation, how it is implemented in the software Dymola, and the specialized ThermoPower library based on it.
- Develop a dynamic model in Modelica and Dymola of a proposed combined heat and power plant for offshore applications, and validate it under steady-state conditions.
- Perform dynamic simulations based on load changes with the developed model and analyze the transient operation of the proposed thermal power plant.

1.3 Organization

This master thesis includes six chapters and one appendix. Technical background is given in Chapter 2, where the waste heat recovery concept is introduced, the importance of dynamic modeling and simulation is discussed, and the motivation for the analysis of transient performance of thermal power plants is presented. In addition, a detailed analysis of the main features of Modelica and the ThermPower library for the dynamic modeling of power plants is presented on this chapter. The case study and the combined heat and power plant design are introduced in Chapter 3. Furthermore, the procedure employed to achieve this design and the methodology followed to develop the dynamic model and perform the transient simulations are also described. Chapter 4 covers the dynamic modeling of the components integrating the power system. The obtained results are presented and discussed in Chapter 5. Final conclusions and suggestions for further work are given in Chapter 6. Detailed validation data results are included in Appendix A.

Chapter 2

Technical Background

This chapter covers the fundamentals and state of the art required to understand and motivate the study and analysis of transient operation, and the development of dynamic models for power generation systems. The energy situation in the world and in Norway is presented in Section 2.1, pointing out the necessity of improvement of the current energy sector. The concept of heat recovery and its many application in the energy market are introduced in Section 2.2. The motivation to utilize dynamic models and simulation is given in Section 2.3, where the increasing energy market variability is presented along with the challenges that it supposes for the current power cycles. Section 2.4 describes the main characteristics of Rankine power cycle's dynamics, and the relevance of each component during the modeling stage. Modelica, a modeling and programming language, is presented and briefly described in Section 2.5, where its ideal features for dynamic modeling are highlighted. Lastly, the ThermoPower library employed in this work is introduced in Section 2.6.

2.1 Energy Scenario

The climate has experienced in the last decades its most tumultuous period. Natural systems have undergone drastic changes since 1950, leading to a situation where human systems and the environment are in danger. The globally averaged surface temperature of the land and the ocean has increased substantially in this period, following a warming process never seen before. Ocean's surface salinity has also suffered some changes, proving the effect of the climate changes in the water cycle. In addition, regions like Greenland and Antarctica are losing considerable amounts of ice due to the temperature increase, which leads to an increase of the sea-level [1].

Human activity is considered as the main responsible of this behavior, as the climate changes can not be explained by themselves. Anthropogenic greenhouse gas (GHG) emissions has increased since the industrial revolution and they reached its peak in the last years. Thus, these prolonged and uncontrolled emissions have resulted in atmosphere concentrations of carbon dioxide (CO_2), methane (CH_4) and nitrous oxides (N_2O) never registered before. Moreover, some of these emissions are absorbed by the ocean, which generates its acidification and the change in some oceanic natural systems.

The main source of these anthropogenic GHG emissions is the combustion of fossil fuels, which accounts for 78% of the total emission increase in the period from 1970 to 2010. Therefore, the connection between the emissions attached to fossil fuels and the climate change that the world has experienced in the last decades can be stated [1].

In view of this situation, Governments from countries belonging to the *United Nations Framework Convention on Climate Change* (UNFCCC) created and signed the Kyoto Protocol (1998). In this treaty, the State Parties admitted the existence of the global warming and the fact that the human CO_2 emissions had generated it, and committed themselves to reduce the GHG emissions in assigned amounts specified in this treaty [2].

As the second period of commitment of the Kyoto Protocol ends in 2020, a new treaty was signed by the State Parties of the UNFCCC, the Paris Agreement. The main objective established in this agreement is to keep the global average temperature below 2 °C above the pre-industrial temperature, having 1.5 °C as a reasonable goal. A significant reduction of the environmental risks and global warming effects can be obtained if this objective is accomplished.

In addition, adaptation and mitigation goals are established in order to achieve a level of development in a low GHG emission society without compromising the food production and supply. Economy policies that can ease the accomplishment of this adaptation and mitigation goals are also covered in the Paris Agreement [3].

From the stated above it seems clear that the development of new technologies that can provide the increasing demand of energy in a growing population with low GHG emissions is needed. Some of the alternatives are based on carbon-free energy sources, while others rely on the improvement of the existing technology to reduce the emissions.

In this context is where the motivation of this project arises. The petroleum sector is one of the main generators of GHG emissions in Norway, see Section 2.1.3, and thus, new alternatives to the current energy and heat generation systems are discussed. Combined cycles are considered as a feasible alternative to reduce the CO_2 emissions in the off-shore oil and gas facilities, albeit their application on the Norwegian Continental Shelf is limited by the frequent fluctuations on the demand. Therefore, this Master's thesis is focused on the study of the dynamic behavior of the bottoming cycle of a combine cycle and the development of a suitable control strategy.

2.1.1 World's energy scenario

The energy situation in the world is clearly dominated by fossil fuels. Its use in the transport sector, the industry and the heating and cooling of buildings yields to a universal energy scenario where the presence and relevance of renewable energy sources are symbolic.

According to the *Energy International Agency* (EIA) [4] and the oil and gas company *British Petroleum* [5], more than the 85% of the energy used in the world comes directly from fossil fuels, namely, oil, coal and natural gas. This energy distribution is shown in Figure 2.1.

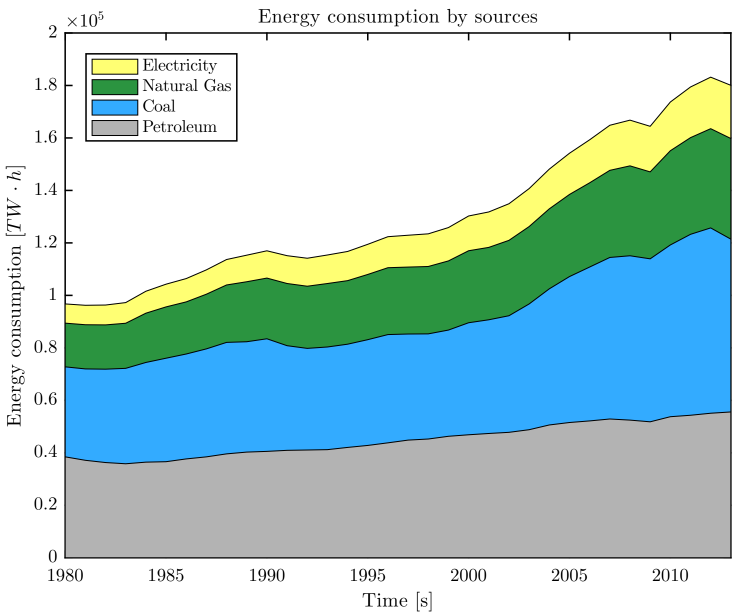


Figure 2.1: World's energy consumption by sources.

The type of sources involved in the electricity generation is another important topic in the understanding of the measures that may be taken to reduce the global GHG emissions. Although the electricity share is not remarkable in the world's framework, it is an important sector since it is where the renewable energies can easily enter the energy market. Moreover, the growth of its contribution to the electricity generation could also lead to a reduction of the direct fossil fuel's consumption, as the transport sector could switch to a electric vehicle fleet and the heating and cooling of buildings could be done by heat pumps instead of traditional gas boilers. The world's electricity generation by sources is shown in Figure 2.2.

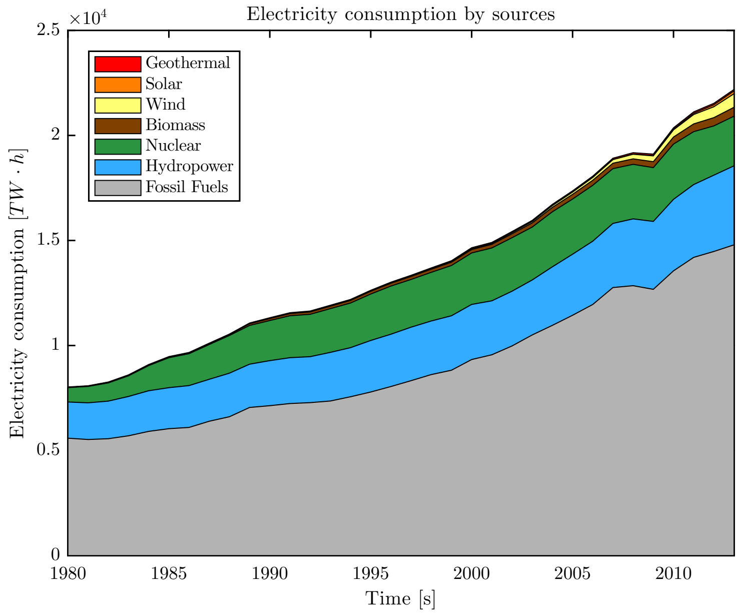


Figure 2.2: World's electricity generation by sources.

As it occurred for the total energy consumption, the electricity generation is dominated by the combustion of fossil fuels in thermal power plants. Moreover, the nuclear energy has an important share of the remaining electricity generated in the world. Therefore, the contribution of renewable sources to the electricity production is small, becoming less than 6% when the entire energy demand is considered.

From Figure 2.1 and Figure 2.2 can also be observed that the total energy and electricity demand increases constantly throughout the years, and it is expected that it continues with this growth in the next decades [4]. This demand's increase is mainly matched by the use of more fossil fuels in both electricity generation and direct fuel consumption. Thus, in order to meet the goals established in the Kyoto Protocol and the Paris Agreement, different strategies regarding the energy sector have to be applied so the growth of the energy demand is not linked to an increase in the GHG emissions as it has happened in the last decades. The emissions since 1980, in CO_2 tonnes equivalent, are shown in Figure 2.3 [4], proving the tendency mentioned above.

The data shown above exposes that, despite the efforts put into the creation of new legislation that boosts the development and integration of renewable energy sources in the energy sector, more research and more policies are needed to achieve the goal of having a sustainable society.

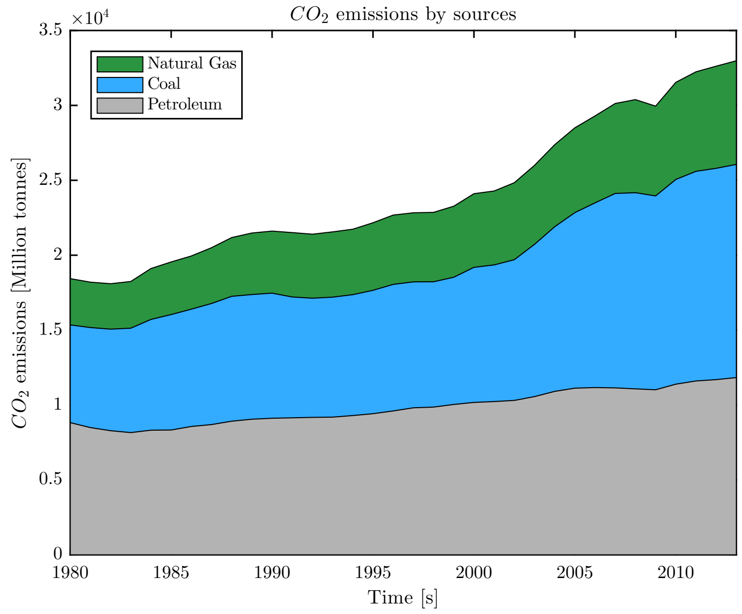


Figure 2.3: World's CO_2 emissions associated to the fossil fuel's use.

2.1.2 Norway's energy scenario

Norwegian energy scenario is a unique case in the world since its generated electricity is almost entirely produced by renewable energy. The geographical location together with its advantageous topography make Norway the perfect place to install hydraulic power plants. Thus, the consumed electricity has been traditionally produced by this power stations, reaching peaks of 98% in the last years.

The Norwegian electricity generation by sources is shown in Figure 2.4. Here, it can be seen that wind is also being introduced in the electricity share, but it is in an early stage. With less than a 2% of share, fossil fuels also enter the electricity generation scenario as they act as buffers of the possible fluctuations that the renewable sources may experience.

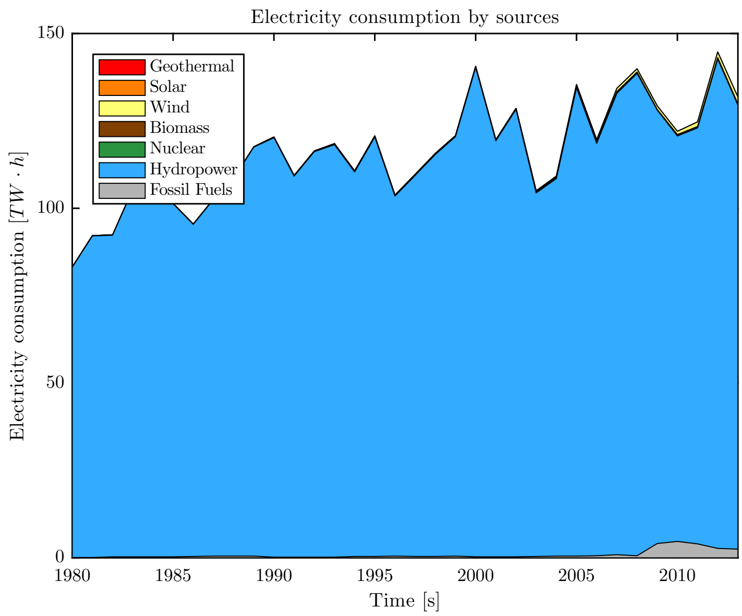


Figure 2.4: Norway's electricity generation by sources.

The world's electricity production is away from the clean electric system that Norway poses. However, when the distribution of the energy consumption by sources is compared, a more similar pattern can be observed, as fossil fuels are also the main contributors to the energy sector in Norway. This Norwegian energy consumption by sources can be observed in Figure 2.5.

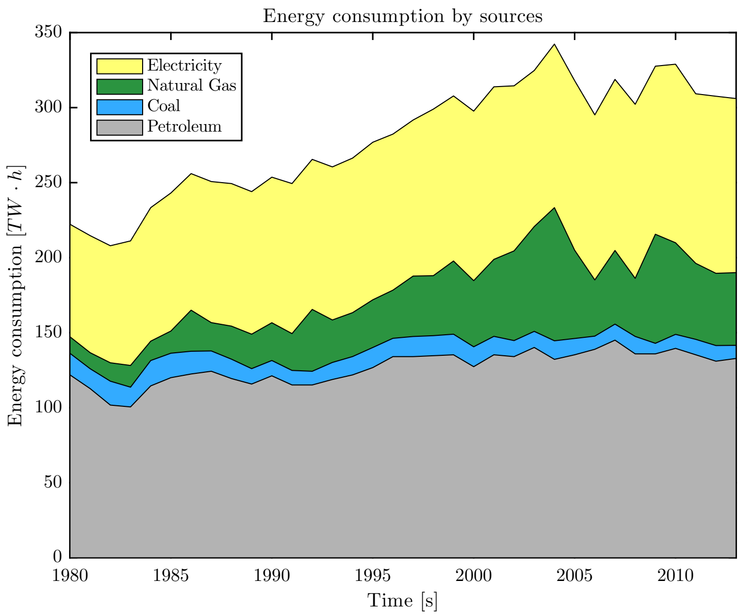


Figure 2.5: Norway's energy consumption by sources.

Norway's energy system is in a better scenario than the average world energy system shown in Figure 2.1, as the clean electricity that is produced in the Scandinavian country makes it to depend less on the fossil fuels. Nevertheless, the 70% of the energy consumed in Norway still comes from the combustion of fossil fuels, what means that the Norwegian energy sector is still far from being fossil fuel independent.

This percentage accounts for the energy consumption of many sectors, being the transport and the petroleum industry the main consumers. Within the transport sector the kerosene burnt in the gas turbines of the planes is included, as well as the diesel and gasoline burnt in the Norwegian vehicle fleet and the fuel burnt in the ships sailing in international waters. The petroleum sector is one of the main consumers as the off-shore platforms are run by gas turbines that burnt natural gas.

As a result of this energy consumption, GHGs are dropped into the atmosphere. The total amount of GHG emissions by source are given in Figure 2.6.

In contrast to the world's trend, where the energy consumption and the associated GHG emissions seem to have a constantly increasing inertia, the Norwegian energy consumption and GHG emissions have stabilized in the last years. This proves that the measures taken by the Norwegian Government are having positive results, and that many countries have to start to make more efforts on the fight against global warming.

2.1.3 Norwegian petroleum sector

As the core business of the Norwegian economy, the petroleum sector is the most important activity of the industry. Thus, the amount of GHG emissions linked to its functioning is a considerable fraction of the Norway's total emissions. Carbon dioxide (CO_2), nitrogen oxides (NO_x), methane (CH_4), non-methane volatile organic compounds (NMVOCs) and sulphur oxide (SO_2) are the main gases released to the atmosphere during its activity.

The sources of these emissions are the combustion of natural gas and diesel in the gas turbines, engines and boilers that run the platform, the flaring of natural gas for safety reasons, the gas venting, and the storage and loading of crude oil.

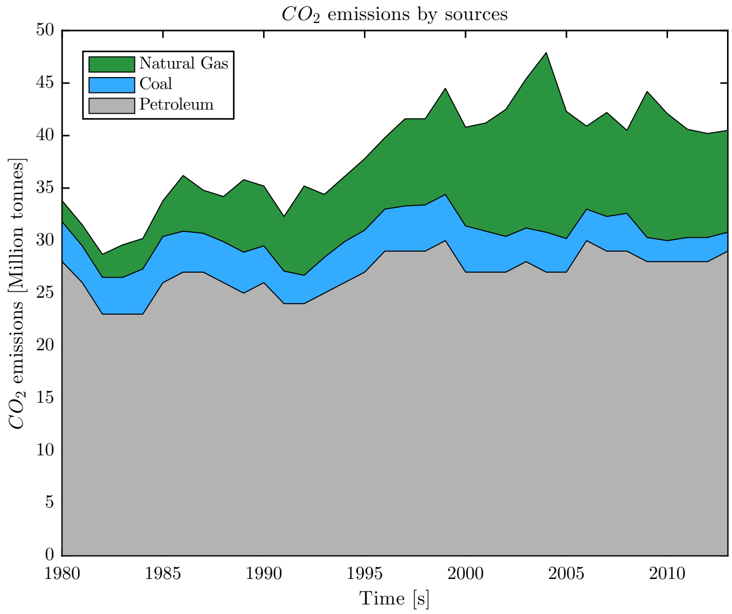


Figure 2.6: Norway's CO_2 emissions associated to the fossil fuel's use.

The emissions originated from the oil sector are regulated through several acts, which are policy instruments created by the Norwegian Government whose purpose is to ensure that climate and environmental considerations are followed during any activity of the sector. Within this measure package, the carbon tax and the Greenhouse Gas Emission Trading Act stand out. The former charges 436 NOK per tonne of CO_2 released to the atmosphere. The later allowed Norway to join the EU Emissions Trading System (EU ETS), which is a system that pretends to set a limit, or "cap", to the GHG emissions of the European Union (EU) as a whole. Thus, companies with low emissions within the EU can sell their surplus of allowances to companies that exceeded its free allowance. In this way, motivation for the application of low-emission technology is generated.

Another of the measures applied in Norway is the data collection of the emissions released into both sea and atmosphere by the *Norwegian Oil and Gas Association*. Therefore, all the emissions originated by the oil sector are registered in a national database called *EPIM Environment Hub* (EEH).

This emission data is shown in Figure 2.7, where all the emissions are presented in million tonnes CO_2 equivalent. In 2015, from the 14.2 million tonnes CO_2 equivalent, 13.5 corresponded to CO_2, while the rest was mainly CH_4 with some traces of the components mentioned above [6].

From the previous figure can be observed that the amount of GHG emissions coming from the oil sector are expected to be stable in the next years. If the fact that the energy consumption increases as time passes is taken into account, the stability of the GHG emissions means that the measures taken by the Government and the companies operating in the Norwegian Continental Shelf (NCS) are effective. Nevertheless, if Figure 2.6 and Figure 2.7 are compared, it can be seen that the oil sector accounts for about the 30% of the Norway's GHG emissions. Hence, if the national emissions want to be reduced, more technological solution have to be applied (see Section 2.2).

The share of the emissions in 2016 is summarized in Table 2.1. Although the data is given for a single year, the general distribution of GHG emission by source over the years is similar to this.

As gas turbines are the main devices utilized for producing the power needed in the off-shore platform, they are the main contributors to the GHG emissions generated by the oil industry. Engines and boilers are used as buffers and as backup power units when gas turbines are under maintenance, so their contribution to the emissions is low.

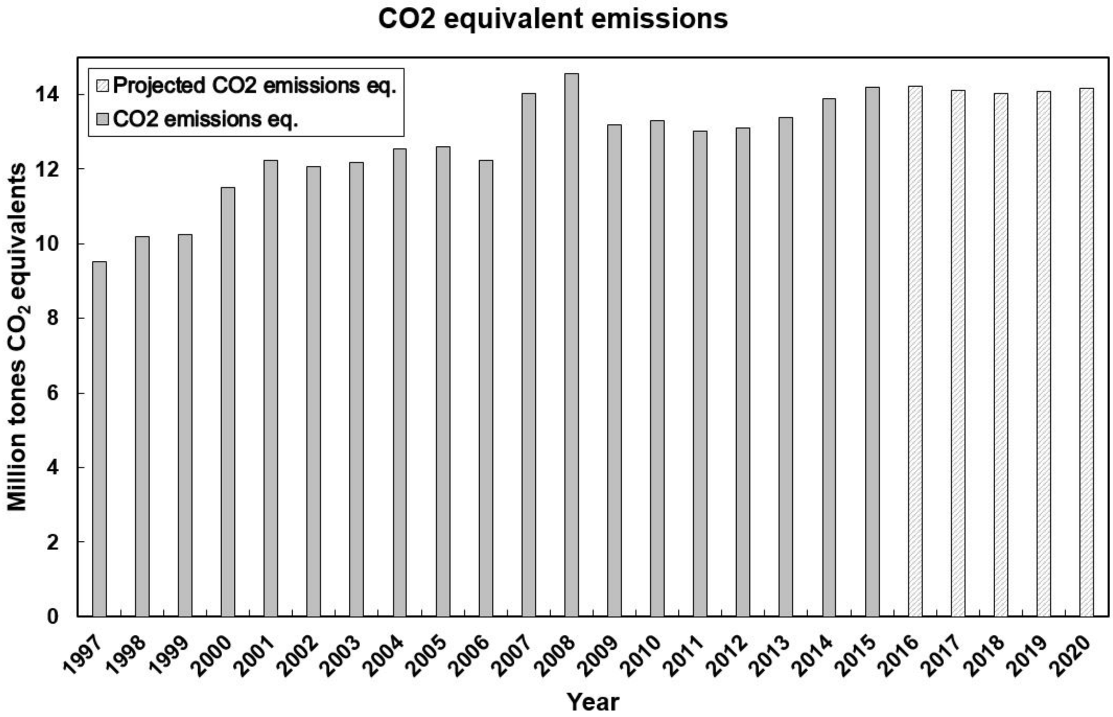


Figure 2.7: Petroleum sector GHG emissions associated to the fossil fuel's use.

Table 2.1: Oil sector's GHG emissions by source.

Source	CO_2 **[1000 tonnes]**	**Share [%]**
Boilers	230	1.7
Engines	955	7.1
Flaring	1307	9.7
Turbines	10930	81.1
Well testing	20	0.2
Other sources	40	0.3

This uneven share implies that the gas turbines play a fundamental role in the reduction of the GHG emissions of the oil sector. Several options will be discussed in the next section, but the improvement of the efficiency of the gas turbines by recovering the waste heat that is currently released to the atmosphere seems to be the most feasible and promising alternative in the off-shore power generation.

2.2 Heat Recovery

The impossibility of many industries to integrate excess heat in any other process or to generate district heating water yields to the release of large amounts of energy to the atmosphere. Heat recovery is based on the utilization of this thermal energy, that otherwise would be wasted, for alternative applications. According to Sternlicht et al. [7], the recovery of thermal wastes is an attractive possibility because of the convenient combination of available temperature and heat volume.

Power generation and process heat are the main uses found for the surplus of heat. However, in this work, power generation is the only utilization considered for the waste heat. There are several factors that may motivate the adoption of waste heat as a source for power generation systems:

- **Economy.** Industries where the waste thermal energy is large can save substantial amounts of money if they convert that heat into power, which could be used to reduce the electricity consumption as in the case of the steel industry or to reduce the fuel consumption as in the power generation industry.
- **Emissions.** The fact of reducing the electricity or fuel consumption not only has economic advantages but also has environmental benefits as the greenhouse gas emissions are reduced.
- **Thermal pollution.** The thermal pollution linked to the direct release of hot gas or liquid streams to the atmosphere is reduced due to the fact that the temperature of these streams is lowered by the recycling of the thermal energy that they contain.

Waste heat can be characterized by its quality, which represents the temperature interval where it can be categorized. There are three main levels: low temperature (<230 °C), medium-temperature (230-650 °C) and high-temperature (>650 °C). The quality of the waste heat influences the area of application, being the low-temperature heat normally used with heating purposes whereas the high-quality heat, i.e. medium and high-temperatures, is employed for power generation [8].

Within the power generation temperature interval of the waste heat, numerous cycle configurations (subcritical, transcritical and supercritical) and working fluids (organic and inorganic compounds) may be consider. The study carried out by Pasetti et al. [9] shows that organic fluids may suffer chemical decomposition when they are subjected to higher temperatures than 350 °C, therefore their application is restricted to heat sources with lower temperatures in organic Rankine cycles. Nevertheless, the application of organic compounds for power generation with low waste heat temperature is not exclusively determined by their chemical composition. As discussed by Larjola [10], the lower specific vaporization heat of the organic fluids allows to extract more energy from the low temperature waste heat sources than the traditional steam cycles. Moreover, the low specific enthalpy change of the organic fluids in the expander permits the utilization of simple single-stage turbines, which reduces the maintenance and initial costs of the unit.

On the contrary, steam Rankine cycles are not capable of recovering energy efficiently when the temperature of the waste heat is low. The large enthalpy drop of the steam during the expansion in the turbine is another drawback of the steam Rankine cycles as it forces the installation of bulky and expensive multi-stage turbines even for small-scale plants [10, 11]. Thus, steam Rankine cycles are suitable options for large-scale power plants and medium- to high-temperature applications where large grades of superheating may be obtained and the formation of droplets during the expansion might be avoided. Despite of this statement, steam cycles can also be used for medium-temperature applications as shown by Kurana et al. [12], who proved the feasible installation of a steam cycle for the heat recovery of a cement plant with exhaust temperatures of 400 °C. Good perfomance of two different steam cycles configurations for even lower waste heat temperatures is observed by Wang et al. [13]. In this study, the higher exergy efficiency of two steam cycles than an organic Rankine cycle was shown, whereas the importance of cycle configuration was proven, as the exergy efficiency difference between the single flash and dual-pressure steam cycles is 1.4%. Inorganic fluids as CO_2 may also be employed for both low- [14] and high-temperature [15] heat sources, showing better performances than organic fluids and steam for certain cycle configurations.

In addition to the temperature level of the waste-heat source, the selection of the working fluid also depends on whether this temperature is constant throughout all the heat exchange process, as in a geothermal applications where the heat source is assumed as a hot reservoir, or it decreases as heat is transferred to the working fluid, as it occurs with the exhaust gas of a gas turbine in a combined cycle. This fact was covered by Liu et al. [16] in their study of the selection of organic working fluid for waste heat recovery applications, pointing out the large deviation that may occur if a constant waste heat temperature is assumed. Larjola [10] also identified this characteristic, showing the great potential of the organic compounds when variable low temperature heat sources are available, as these working fluids follow better the temperature decrease of the waste heat leading to higher efficiency than if steam was utilized.

In this work the traditional steam cycle is utilized due to the high temperatures and the large mass flow that can be reached in the exhaust stream of the gas turbine employed in the off-shore oil and gas facilities. Moreover, this technology is more mature than the available for other inorganic fluids as CO_2 and hence its installation is more likely to occur in the short-term.

2.3 Dynamic Modeling and Simulation

As engineering systems are evolving towards more elaborated and complicated configurations, more detailed studies and analysis are required in order to achieve a deeper understanding. As a consequence, dynamic modeling and simulation are gaining continuous relevance in the research and industry sectors in the recent years. Unsteady operation importance lies in the necessity of describing and predicting the behavior of systems formed by elements that belong to different domains when they undergo abnormal and fast alterations.

Traditional studies are based on systems working at nominal operation conditions subjected to quasi-steady deviations where the system reaches another stable operation point. This approach allows to evaluate whether the new operation conditions are adequate or unfavorable and how the final state of each element of the system is influenced by a specific change. However, when time is taken into account, additional information can be extracted since not only the initial and final states of the system are known but also its behavior throughout the process. In this way, the effect of the control strategies can be extensively evaluated whereas the tuning of the control variables is eased. These features enhance the utilization of dynamic modeling and simulation as a testing tool for hazard and operability analysis as well as emergency control procedures.

In addition, the training possibilities of plant operators are widen due to the capability to replicate unstable operation and emergency scenarios. Several plant working conditions may be generated where the operators have to follow certain procedures, yielding to a more relevant and complete preparation of the plant staff responsible of the control area. This training may be employed with several goals: firstly, to ensure that emergency situations can be solved without any risk, secondly, to operate the plant with the highest benefits, and lastly, to avoid the infringement of environmental regulations.

Moreover, the study of the dynamic performance can also be integrated in the design stage of a system, e.g. process plants and automotive or aeronautic applications. Traditional steady design procedures only aim to achieve the highest possible efficiency, both first and second law, under certain restrictions. Unsteady performance is not considered at this stage but it is studied when the control strategy is being designed, which leads to excessively aggressive control configurations where the off-design operation, a regime where many systems normally operate, is worsen as a result of the limitations established by the dynamic operation. Furthermore, the dimension of the different components of the system may be unsuitable if only a steady-state approach is followed as it is during unstable conditions when the size of the equipment has the largest effects on the operation of the entire system. Thus, inadequate configurations or designs may be avoided in an early stage of the design process if the performance under dynamic conditions is considered.

Many approaches may be utilized for the dynamic modeling of different plants and systems. Several classifications of these approaches can be done according to whether the models are lumped or distributed parameters, the followed modeling strategy is based on modular or simultaneous formulation, or if the causality principle is utilized. The selection of the modeling language determines the approach that is followed and dictates the restrictions of the user when modeling any system, e.g. outputs are on the left side of the equation and inputs are on the right when a causal language is employed. Therefore, the selection of the modeling language is a key step when modeling and simulating dynamic systems. Modelica is selected as the modeling language in this project. A thorough description of Modelica is covered in Section 2.5, where its main characteristics are detailed and the advantages of its utilization for dynamic modeling and simulation are pointed out.

2.3.1 Energy Market Variability

The climate changes and environmental risks addressed in Section 2.1 motivate the modification of the energy market towards more sustainable energy horizons. Renewable sources play a significant role in this transition to a sustainable power generation market since they are carbon-free, or CO_2 neutral in case of the biomass (if utilized correctly), and therefore their integration in the system seems mandatory.

The evolution of the electric market has already started as renewable energy sources have been gaining more relevance in the generation share, albeit their contribution is still fairly low and more effort is needed in order to substitute the traditional hydrocarbon sources. This tendency can be observed in Figure 2.2, where it is clear that since 2006 the share of wind and geothermal energy along with biomass has notably increased with respect to the previous decades, which proves the beginning of this transition.

Higher levels of renewable source penetration in the electric market may be expected in the short-term. The integration of this type of energy sources, despite of their environmental and economic benefits, presents technical and structural challenges for the electric grid management and stability, and for the operation of other traditional power plants, e.g. nuclear and coal plants. The issues related with the integration of renewable energies in the electric market are due to the non-dispatchability, the frequent fluctuations and high uncertainty in the availability of sources as the wind or the sun. Thus, the power generated coming from wind farms, photovoltaic panels and thermo-solar power plants experiences the same issues. In addition, other renewable sources as hydroelectric and geothermal are strongly affected by physical, economic and environmental constrains, and their dispatchability varies according to such restrictions increasing their variability and unpredictability [17].

As a consequence of the uncertainty in the availability and dispatchability of power generated by renewable sources, the traditional thermal power plants, coal plants and combined cycles mainly, are forced to operate in a more flexible manner. These power plants have to be able to increase or decrease their production according to the renewable power production in order to maximize the share of electricity generated by zero-emission sources. According to Garcia et al. [17], this solution is less complex and more cost-effective than introducing modifications in the electric grid so it is able to handle supply variability.

The necessity of developing a reliable and efficient power system where the intermittent supply coming from renewable sources increases is discussed by Lise et al. [18]. In this study, the role of different sources of flexibility for the energy marker are treated, pointing out the need of developing other flexible resources that complement the flexible power generation coming from hydropower and combined cycles. Garcia et al. [17] studied the dynamic behavior of hybrid energy systems where the obtaining of energy products is related with the power generation system in an energy flow network. It was found that renewable penetration supposes a challenge but different strategies may be applied to meet the established requirements, as the facilitation of flexible operation or the design of more flexible primary heat generation for chemical processes. Hittinger et al. [19] analyzed the behavior of a hybrid system formed by a wind turbine and a natural gas turbine, and the effect that a small storage unit produces in the power generation. The hybrid system was shown to be economically viable whereas the fluctuations of the system were smoothed allowing the gas turbine to compensate the power generation without abrupt changes that could generate short periods of large emissions. This increase in the emissions generated by natural gas generators due to sudden ramp-ups forced by fast variations in wind and photovoltaic power production was studied by Katzenstein et al. [20]. The need of taken into account the poorer performance of the equipment utilized for compensating the fluctuations associated to large penetration of renewable systems in the energy market is addressed in this study, motivating the development of more flexible power generation systems that compensate such variations with better performance.

2.3.2 Power Cycle Dynamics

The growing variability induced by the renewable source penetration in the energy market leads to the flexible operation of traditional power generation plants in order to match such fluctuations and do not alter the normal operation of the energy system. As a result, coal and gas-fired power plants are subjected to frequent changes in their operation conditions that may result in poor performances, high emissions and low benefits (see Section 2.3.1). Hence, a deeper understanding of this type of processes and the development of new operation strategies are needed in order to adapt the energy system to the integration of the intermittent supply generated by renewable sources. The dynamic analysis of these power plants is a powerful technique that allows the evaluation of their behavior under fast operation changes, and the development of models that may predict responses to certain inputs and ease the design of new control strategies that enhance the flexible operation of these plants.

Dynamic modeling and simulation may be employed to the analysis of the unstable behavior of power plants and to improve the development of control strategies and tuning of control parameters, as discussed in the beginning of Section 2.3. Nevertheless, power systems present a number of peculiarities that hinder the modeling of these plants and make more challenging the evaluation of their performance. More specific models and modeling techniques are necessary to characterize the behavior of different power systems since certain elements as turbines or furnaces are not commonly utilized in other sectors.

Efforts of many researchers and institutions have lead to the creation of specialized software for the dynamic modeling and simulation of energy conversion systems in the last years. For such purpose, several approaches may be utilized. Colonna et al. [21] presented a software based on a modular paradigm where elementary systems are represented by blocks, or modules, whose behavior is determined by equations following causal relations. The modeling approach employed during the development of the software is discussed, whereas the creation of elementary modules is demonstrated by means of some examples. Lastly, the validation is done by comparing the results obtained by the software with the measurements of a experimental setup, proving in this way the possibility of predicting the dynamic behavior of power cycle elements by means of specialized software. A modular approach was also followed by Casella et al. [22] during the development of the ThermoPower library. Within this library, many components and physical behaviors representative of power plants are represented by blocks, but in this case the performance is specified by acausal relations. A more extensive description of this Modelica library and its development may be found in Section 2.6.3. In addition to the tools created by reserachers and academic institutions for the study of dynamics in power plants, there are also other softwares developed by private companies that may be utilized. Dymola and Simulink, two different modular-based software, were employed by Benato et al. [23] to develop two different models of a steam bottoming cycle with the objectives of studying the transient behavior of the unit and the reduction of the operation-life due to the stresses suffered by the heat recovery unit. The results obtained by the two models are in good agreement since the trends observed during the dynamic response are similar and the steady states reached after such variations are equal.

The fact that modular-based software have been exclusively mentioned could create the erroneous impression that this approach is the unique option when modeling and simulating the dynamic behavior of power plants. There exist other alternatives as the utilization of a simultaneous approach where a single routine of many equations is used to characterize the performance and behavior of the power plant. However, due to the complexity and heterogeneity of these systems, modular approaches as the mentioned previously are normally employed for the study of power cycle dynamics, albeit the higher demand that they require in terms of simulation time.

The horizon of utilization possibilities of these dynamic modeling and simulation software is broaden when they are combined with other computational tools. This merging capability enhances and eases the integration of dynamic modeling in the design phase of power cycles, a fundamental step where several plant configurations can be discarded due to the unfulfillment of any restriction. A power system design methodology accounting for the dynamic performance was proposed by Pierobon et al. [24]. The procedure consists in two clearly differentiated steps. In the first step, using Matlab, the boundary conditions are fixed and all the thermodynamic states are calculated following basic mass and energy conservation laws. From these results, and using the adequate equations, the dimensions of every component integrating the power cycle are calculated. A multi-objective optimization is employed in order to find the best thermodynamic states and equipment dimensions. As a result, a Pareto front of optimal solution is obtained. In the second step, each of the solutions that form the Pareto front are evaluated under certain specified dynamic conditions, normally critical scenarios, employing a dynamic model of the power plant built in Dymola. Once that the solutions that do not meet the dynamic requirements are discarded, the assessment of the remaining can be done. The optimal solution can then be selected attending to the desired criteria.

The development of two models, one for design operation and other for dynamic performance, is needed in order to encompass the whole operation range of a power cycle. As discussed above, the first model is built in order to define the thermodynamic states at the inlet and outlet of each element. These states are limited by the boundary conditions imposed to the model and determined by steady conservation laws. Once the nominal states of the entire cycle are known, it is possible to dimension every component through specific equations and correlations that may be found in the literature. The dynamic model is developed in a different way as the sizes and geometry of the components are already defined. The equations that govern the performance of each component are not steady in these models, as the time derivative must be included in order to account for the fast fluctuations that occur during dynamic operation. Thus, mass and energy storage are taken into account in the dynamic model. Auxiliary equations are also required to consider the deviations of certain parameters from their values during nominal operation, e.g. heat transfer coefficients or expander inlet pressure, and therefore they also have to be included in the dynamic model. This process is followed by Mazzi et al. [25], who developed a steady and a dynamic model of an organic Rankine cycle with regenerator for waste heat recovery applications with the objective of analyzing the transient response of the system, the effectiveness of the selected control strategy and the influence of the evaporator volume during transient performance. The latter was done by means of a sensitivity analysis and proved that the evaporator pressure and the power output of the expander are not strongly influenced by the volume of the evaporator. This highlights the benefits that can be extracted from including dynamic models and analyses in the design phase of a power cycle since the heat exchanging equipment is normally a sensitive element during transient operation and discarding any issue related with its size may ease the design of the cycle.

In addition to improvements associated to the consideration of the dynamic operation during the design of power cycles, dynamic modeling and simulation are also powerful tools to evaluate the performance and behavior of a plant when it is operating close to the limits. These bounds may be imposed because of environmental regulations, material specifications or economic restrictions. A remarkable procedure where these limits might be reached is the start-up of thermal power plants with waste heat recovery since the time scale of the dynamics differs notably among all the components integrating the plant, ranging from seconds or a few minutes to tenths of minutes or even hours. Therefore, the study of the dynamic behavior of every component during this process is fundamental in order to develop procedures that reduce the required starting time while keeping safe conditions in each element that do not reduce the life-time of the power plant. Both facts were studied by Casella et al. [26], who developed a dynamic model of a combined cycle with a three pressure level heat recovery steam generator. After an initial analysis of the plant and the current start-up procedure applied in the power plant, it was noted by the authors that the followed strategy was excessively conservative due to the focus on safety and availability rather than on flexibility and efficiency. Therefore, two alternative routines were proposed: the first one focused on minimizing the start-up time without exceeding the stress limits, and the second one focused on reducing the maximum stress peak in order to extend the life-time of the plant without increasing the start-up time. Positive results were obtained for both alternatives, halving the start-up time in the first case and reducing the start-up time and the peak of stress in the second one. Alobaid et al. [27] developed a very detailed static and dynamic model of a combined cycle in order to study and improve its start-up process. Higher thermal gradients achieved by the modification of the exhaust gas conditions were proposed as new start-up conditions. As a result, it was found that the time to reach the maximum power generation was reduced to the half and that savings in fuel consumption and electricity production accounting for 3500 euros per start-up could be achieved. Temperatures in sensitive components were monitored and it was observed that the maximum temperature was not increased, however, it was found that the pressure increased substantially in certain components. Therefore, substantial gains in economic and operational terms could be achieved if a material that could handle such pressures was employed. A control strategy for improving the start-up time of a heat recovery steam generator was studied by Kim et al. [28]. Only the effect of the control variables on the behavior of the steam generator were analyzed, showing the relation among them that leads to the best performance.

Operational limits may also be reached when power plants operate on an island, i.e. as stand-alone systems. The fact of being disconnected from the grid can lead to frequency instabilities that may generate the failure of the power system. Since the power consumed in certain isolated locations exclusively depends on standing alone power cycles, the study and avoidance of this scenario is fundamental in order to ensure the power supply. Examples are the off-shore oil and gas facilities, recuperation systems equipped in means of transport or power plants linked to process plants. Frequency fluctuations occur during transient performance due to sudden changes in the operation conditions of the power plant. Therefore, studying the dynamic behavior of such plants eases the development of control strategies and component geometries that enhance a more stable operation where frequency is kept within the limits. The rise time and the frequency undershooting or overshooting of a combined cycle with two gas turbines connected to an air bottoming cycle for off-shore applications under dynamic operation were studied by Casella et al. [29]. The critical scenario where the trip of one of the gas turbines occurs was selected. Results addressed the need to modify the load distribution among the gas turbines and the air bottoming cycle since an overshooting exceeding the limits set by the plant owner occurred for certain load set-point variation. Moreover, by means of a sensitivity analysis, it was found that this overshooting could be avoided if a lighter recuperator was installed. Kazuyohsi et al. [30] studied the behavior of a combined cycle that undergoes a low-frequency excursion. It was observed that, as a consequence of the compensation action followed by the gas turbine, the temperature of the exhaust increases until the temperature control acts on the inlet guide vanes. Therefore, the relevance of considering dynamic frequency fluctuations in stand-alone system is pointed out, as undesirable control strategies may lead to temperatures excessively high that can damage the equipment.

2.4 Dynamic Modeling of Rankine Power Cycles

Closed and open thermodynamic cycles may be employed for waste heat recovery applications. Both present advantages and disadvantages, and the selection criteria strongly depends on the scenario. Nevertheless, close Rankine cycles are the thermodynamic cycles most employed by the industry due to the wide range of possibilities they present in terms of working fluids and cycle configurations.

These cycles are formed by different types of heat exchangers, turbines, condensers, pumps and auxiliary equipment. However, from a modeling perspective, the auxiliary equipment is not normally considered as it does not affect the performance of the cycle directly. In this section, the main components of the Rankine cycle are described, whereas the main features and challenges of their dynamic modeling are presented.

2.4.1 Primary Heat Exchanger

The primary heat exchanger is utilized for the recycling of the waste heat. The hot fluid acts as a heat source releasing its energy through a metal wall. The cool fluid circulating in the other side, which is the working fluid of the Rankine cycle, is heated up, increasing its energy before being expanded in the turbine. The configuration of this component may differ from an application to another, as temperature gradients, heat source's mass flow and temperature, working fluid, presence of fins, weight, and space affect the selection.

The modeling of this component is a crucial step in the development of a dynamic model since fluid phase changes may occur and large gradients, not only in temperature but also in fluid properties, are present throughout the heat exchanger. Hence, detailed and robust models are needed in order to satisfy such requirements. Two different modeling approaches can be utilized for this component: discretized models and moving-boundary models.

- **Discretized models.** The flow path is divided into cells, or volumes, where the fluid is characterized by the properties within the cell or at its faces. The calculation of these properties can be done in several ways depending on the selected approach and mathematical method, e.g. finite difference method (FDM), finite volume method (FVM), or finite element method (FEM). In addition, the materials like the metal tubes are also discretized in several volumes and their properties are calculated in the same manner, as it is in these region where the thermal capacity and inertia affect the most the dynamic operation of the heat exchanger. Thus, the calculations of the heat transfer and pressure drop occurring among both fluids and the metal wall are subdivided in as many sections as discretizations required, obtaining individual results in each of them. The accuracy of the calculations increases with the number of cells since the mathematical process is smoothed due to the decrease in the property change from one cell to the next one, i.e. infinite number of cells would produce exact results.

- **Moving-boundary models.** The flow path is divided in the same number of zones as fluid phases within the heat exchanger. Hence, a maximum of three zones may co-exist. With this modeling approach the phase boundaries are tracked during the operation of the heat exchanger and adapted to the changes that occur during different off-design and dynamic conditions. Fluid, flow and material properties are calculated for each zone and the overall results for heat transfer and pressure drop are obtained by the combination of the three zone's values.

The higher complexity of the discretized models yields to more accurate results. Moving-boundary models are faster and more robust, specially in applications with high property gradients, e.g. changes of phase, where the equations of state may cause computational instabilities. In addition to its higher flexibility, moving-boundary models are also characterized by lower orders, which makes them more suitable for control applications where high computational speed and robustness are required. This result was found by Wei et al.[31] by means of a comparison between these two approaches. The condenser and evaporator of an organic Rankine cycle were modeled following both techniques. Similar results and accuracy were obtained by the two models, however, the moving-boundary model was proven to be faster than the discretized version. Horst et al. [32] also employed a moving-boundary approach for the development of a heat exchanger model for heat recovery of the exhaust gases of a car engine. Results were in good agreement with the measured data and fast response of the model was obtained even during fast dynamics of the engine. On the contrary, a discretized model of a heat recovery steam generator is utilized by Alobaid et al.[27] in order to achieve high accuracy in the temperature profiles and temperature gradients calculations due to the high sensitivity of these parameters during the start-up of this component.

The configuration of the primary heat exchanger also affects the modeling. Large steam power plants are typically characterized by several differentiated pressure levels with steam drums and daerators, while organic Rankine cycles and steam plants with space and weight restrictions normally utilize compact heat exchangers where the phase change of the working fluid occurs in the same component, and the installation of drums and daerators is discarded. The former configuration is usually modeled following the distribution that it has in practice, where three separate sections representing the economizer, evaporator and superheater are utilized [22, 23, 26, 33, 34, 35]. Compact heat exchangers may also be modeled in this way even if not such divisions are present in practice. Higher accuracy and detail are obtained since the different fluid phase regions are modeled individually and the heat transfer process can be described with more exactitude in each of the sections [25, 36]. Conversely, this compact heat exchangers may be modeled as unit where the overall performance of the heat transfer process is obtained. This approach does not provide as much information as the previous modeling technique but it is computationally faster since less detail is required. The modeling of a once-through boiler using this approach was carried out by Pierobon et al.[24], obtaining good results albeit the simplifications assumed in the modeling stage.

2.4.2 Turbine

The turbine is the element where power is extracted by means of the expansion of the working fluid through one or many stages of blades that drive a shaft connected to a load or generator. The dynamic behavior of the fluid within the component is normally too complex to be modeled, and therefore it is not taken into account. In addition, the dynamic analysis of a power cycle is usually focus on the thermodynamics of the cycle and how variables are affected by certain changes. Fluid behavior through the turbine is not relevant for such analysis, and its effects on the performance of the entire cycle are accounted by parameters as the isentropic or politropic efficiency of the expander by means of equations and correlations. The mass, momentum and energy inertia and capacitance of the turbine are negligible compared to those of the heat exchangers and they are not normally included in the models. Thus, quasi-static models are utilized to describe the performance of this element. Examples of this practice may be found in Refs.[25, 37, 38].

2.4.3 Condenser

The condenser is the heat exchanger where the fluid leaving the turbine after being expanded is condensed to be able to enter the pump without damaging it. Shell and tube is the most common configuration in Rankine cycles, although other technologies are available. The cooling fluids commonly utilized are water, seawater or air. Its choice depends on the application and the location of the power plant, as air is a suitable cooling fluid if large condensers can be installed while seawater is an ideal cooling fluid in offshore applications due to its availability and high heat transfer coefficients.

The dynamic modeling of the condenser is similar to the primary heat exchanger. Large property changes and temperature gradients are encountered throughout this component. Therefore, the discretization of the shell, the flow along the tubes and the metal walls is required if details of the condensing process are needed. This approach is specially convenient when superheated steam is present at the inlet of the condenser since varying conditions may be encountered during the transient performance along the heat exchanger. Moving-boundary models may also be applied in this scenario (see Section 2.4.1), and can become specially useful when subcooled conditions can be expected, as abrupt property changes occur when some working fluids get closer to the saturation state, e.g. water. A discretized model can also be utilized, but excessively dense discretizations could be needed, leading to large computing times.

In addition, if only small fluctuations are expected, even during dynamic operation, and the working fluid at the outlet of the turbine is saturated or slightly superheated, a simpler modeling where only the inlet and the outlet states of the condenser are considered may be utilized. This approach considers that the state of the working fluid at the outlet of the condenser is saturated liquid, which eases the modeling work and reduces the computational time.

2.4.4 Pump

The pump is the active element of the Rankine cycle. Its function is to control the mass flow rate of the working fluid circulating in the closed loop and the high pressure of the cycle, which corresponds to the entrance of the heat exchanger. There are many types of pumps, but the ones employed in Rankine cycles are normally centrifugal pumps with variable speed.

As it occurred with the turbine, the flow pattern inside the pump is too complex to be modeled and its knowledge is not relevant for dynamic analysis purposes. Therefore, the variations in the fluid conditions within the pump are represented on its efficiency. Quasi-static models are employed and the performance of this component is represented by sets of characteristics curves, where the head generated by the pump is a function of the mass flow rate circulating and the number of revolutions.

2.5 Modelica Language

The complexity in the study of human and natural systems is continuously increasing. Nature becomes more difficult to analyze the deeper understanding we want to achieve while human-made systems are consistently evolving towards better, but more involved, designs. Experimentation is normally required in order to collect more information about a system or to verify whether the hypothesis developed theoretically are correct or not. However, some experiments may be unfeasible because of the price, the associated risk, the impossibility of measuring certain inputs or outputs, or simply because the system does not exist yet. Modeling and simulation can be utilized instead of experimentation to study these systems.

In engineering, the importance of modeling and simulation is growing as the study of current systems becomes more challenging due to the increase in their complexity and heterogeneity. Several languages and tools have been developed according to the needs of specific domains but their weakness handling components of other domains makes them too exclusive. General tools have also been created in order to be able to treat multi-domain models, but the enormous work load that they require in the case of the block-oriented tools, and the lack of re-usability in the object-oriented attempts, points out that these computer programs are not the best suited for the simulation of large complex heterogeneous systems. A standardized modeling format is required in order to be able to generate multi-domain models while keeping the re-usability approach. From this idea and the shortage of tools that can handle complex heterogeneous systems in a easy manner Modelica emerges [39, 40].

Modelica is a general object-oriented equation-based programming language whose main objective is the mathematical modeling and simulation of complex systems in a multi-domain framework. What it is intended with Modelica is to create a language that models complex physical systems at the same time it allows the exchange of models between different tools. Re-usability is a key feature when modeling large heterogeneous systems. Thus the Modelica approach is selected in such a way that this characteristic is achieved.

2.5.1 Object-Oriented Mathematical Modeling and Programming

As its name suggests, object-oriented programming is a programming approach based on the utilization of objects. How an object is described, and therefore behaves, depends on the employed programming language. In the traditional object-oriented languages, an object is a combination of stored data and code, which is normally a set of operations that define a specific procedure that determines the behavior of the object. In this way, more elaborated systems can be created by means of groups of simpler objects where the interaction among them is specified in the procedure written in each element. An object may modify the stored data of another related object creating in this manner a dynamic flow of information.

In Modelica, the object orientation is viewed with a different perspective than in the traditional languages since it is considered as a structural approach for mathematical modeling instead of for procedural description. This is done by the specification of equations in the object instead of procedures. As it occurs in the traditional programming languages, a Modelica object also includes a set of data and instance variables, however, the fact of introducing equations leads to a declarative description of the object properties and behavior that the objects defined by the traditional approach do not posses. In other words, the utilization of equations describes what truly occurs physically, what an object holds, its behavior, while a procedural description is just an algorithm to achieve a certain goal following a specific order.

The equation-based programming allows an easier programming from the user point of view since a direct physical modeling is possible. In this context direct means that the modeling in the programming environment is done as in the paper, writing the equations that model the physical behavior of the object without any re-arrangement, which avoids human mistakes during the adaptation of the physical equations to a form that a procedural language can follow. Thus mathematical modeling and programming become closer to the human way of thinking at the same time that generality is kept since no algorithms are specified.

On the contrary to what could be thought, the declarative object-oriented way of describing systems supported by Modelica leads to a higher level of abstraction as some steps are skipped during the programming. In the object-oriented procedural-based languages all the steps required for the modeling and simulation of any system have to be specified, including the object interaction relations that regulate the data flow among them. This code can be omitted when programming with Modelica since it is automatically generated by the Modelica compiler according to the equations that are specified in each object [41, 42].

2.5.2 Acausal Modeling and Reusability

The Modelica approach of object-oriented programming and modeling is based on the description of physical behavior through equations instead of statements. This feature is possible due to the acausal, or non-causal, modeling. This modeling strategy means that data flow direction is not dependent on how the code within each element is written since inputs and outputs are not defined, information can flow in both ways. From a programming perspective, the acausal modeling can be explained as that the equal sign, =, means equality, not an assignment as in other programming language. Thus, as the information direction is not pre-established, objects can include equations to define a specific physical behavior.

The combination of an object-oriented programming approach with an acausal modeling strategy results in a programming and modeling language that enhances the object reusability. Objects describe physical behavior through sets of data, instance variables and equations. As these equations do not have a fixed information flow direction due to the acausal modeling that is implemented in Modelica, an object may be utilized in any system where the physical behavior described by the object appears. This is possible since the data that is known is not relevant, Modelica's capability to handle both data flow directions allows it to solve the system if the number of variables is equal to the number of equations.

Reusability is a key feature for any kind of language that aims for modeling complex systems. Modelica objects that model many different physical behaviors can be used in several scenarios since they are not limited by the need of pointing out which variables are inputs and which variables are outputs. Thus, objects that represent behaviors from different domains, e.g. mechanic, electric, fluid and control domains, can be linked and utilized together for describing a more complex system as the object-oriented acausal modeling enhances these combinations. Modelica does not distinguish among domains, it just characterizes behaviors through objects that contain data and equations. If these components are mixed in a multi-domain system is totally irrelevant for Modelica as long as the connections among components make sense, e.g. mechanical with mechanical or electrical with electrical, and the number of variables is equal to the number of equations. Therefore Modelica eases the modeling of large complex systems where components from different fields have to be included.

In addition to reusability, Modelica also enhances the ease of understanding the physical modeling of complex system. The fact of being based on an acausal object-oriented modeling approach allows to generate models of systems that maintain the same topology as the real system, that is, the created model and the real system have the same appearance. This characteristic eases the modeling of the real case since the connections between the components of the model correspond to the same connections in the real system. Causal block-oriented modeling languages lack this feature due to the fact that information flow directions have to be specified, and therefore the order of components may be altered during the modeling procedure. In this way, the physical topology of the real system is lost throughout the modeling and the expression physical modeling is not the best suited.

2.5.3 Modelica Structure

The modeling and programming by means of equations is supported by Modelica as explained above. However, modeling and programming by composition can also be utilized if a graphical model editor based on Modelica, like DYMOLA, is employed. This kind of tools allow graphical programming, that is, more complex models can be created from the union of icons that represent simpler models, forming what is called a composition diagram, the graphical representation of a composite model. These icons may be basic objects where the description of physical behavior is included with equations, storage data and instance variables, or may also be models formed by more elementary objects or by the extension of a more generic class. A diagram that symbolizes the process of creation of a composite model through the use of simpler models and objects is shown in Figure 2.8.

As it can be observed, some specific words as class, model, object and instance have been utilized. They are used in Modelica as keywords to classify different physical phe-

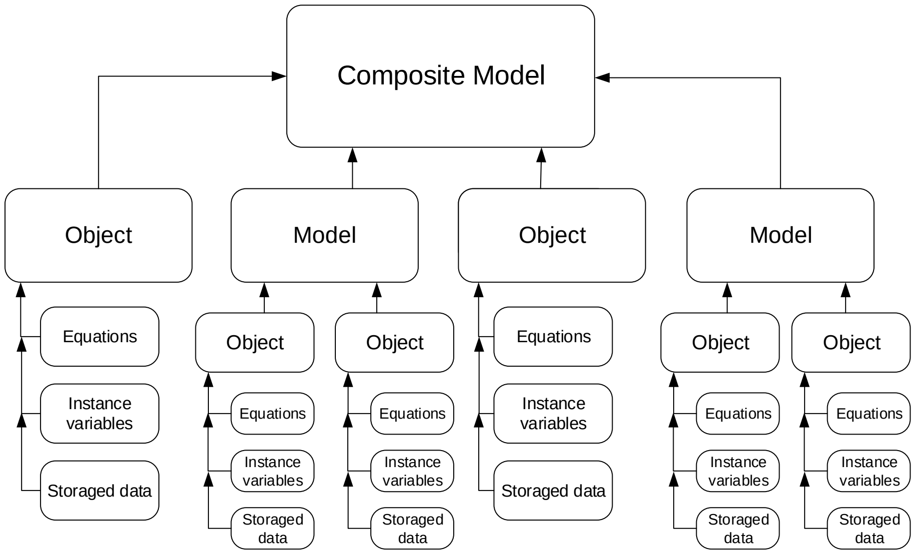


Figure 2.8: Diagram of a composite model formation.

nomena and devices. Thus, in order to understand how Modelica is internally structured, it is fundamental to know the meaning of these keywords as well as the hierarchy existing among them.

Classes, also called models, are the main structuring element of the Modelica language. They can be regarded as blueprints or molds from where is possible to create objects, known as instances of the class. A class may be formed by the following members:

- **Fields**. Variable declarations associated with a class and therefore its instances. They are constants, several type of parameters and the results obtained from the computation, acting as a information storage element during the calculation process.

- **Equations**. They specify the behavior of the class. They are the representation of the physical phenomena that is modeled.

- **Classes**. Classes may also be part of other classes.

As objects are instances of a class, it is clear that an object is a collection of variables and equations that share a set of data. From a Modelica class as many objects as needed can be generated.

Reusability is also enhanced from the class concept since any class can be modified to include a special behavior. Using modifiers, which are attribute equations that modify some parameters of the original class, the desired performance can be obtained in the new generated class without re-writing the entire code. Inheritance is another technique to reuse any kind of class. It consists in the extension of the properties or the behavior of an existing class, called superclass or base class, by adding new parameters or extra equations to the created class, known as subclass or derived class. This is done using the keyword *extend*. It is important to point out that when inheritance is utilized, all the parameters and equations of the base class are inherited by the derived class, which is extended by extra parameters or equations; while the same type and number of parameters are employed when a class is modified. As an special type of class that tries to make use of the inheritance concept is the partial class, which is a class that does not have enough equations to completely define a physical phenomena but that is so general that can be extended by several classes within the same domain, e.g. electrical ports or mechanical flanges.

As the last structuring element, packages, also called libraries, are introduced. It is originated from the need of avoiding name collisions between different classes, functions and other definitions. The possibilities that Modelica offers to create new classes and to reuse the existing ones can yield to situations where the same name is used for different concepts. Hence, in order to avoid the overlapping of names and the consequent computation error, packages are utilized. In this way, the different elements contained in the package have as a prefix the name of the package and therefore no name collision can occur.

More types of structuring elements could be presented in this section, as the types of class definitions. However, these kind of classes may be considered as special cases of the generic class concept and they are not fundamental to understand the main organization of the Modelica language. For more information the reader is referred to Ref.[42].

2.5.4 Continuous, Discrete-Event and Hybrid System Modeling

Physical systems normally evolve as a function of time in a continuous manner as the physical laws establish. The study of the dynamics of this kind of systems can be done by using directly the laws that govern their behavior. On the contrary, human-made systems and system with extremely fast dynamics can be studied as if the changes occurred instantaneously and discontinuously, that is, their dynamics can be analyzed as discrete events. The traditional approach to study this type of events is to utilize *if* or *when* clauses followed by certain actions. However, in Modelica, it is possible to implement equations instead of causal actions in order to maintain the acausal modeling approach [41, 42]. Algorithms may also be implemented with Modelica, but they should be avoided whenever is possible since the use of equations describes physical phenomena in a more detailed way.

In a real application may be possible to find a system that mixes continuous and discrete time events. Such systems are called hybrid systems and they suppose a challenge for the modeling approach. The synchronous principle is utilized in Modelica to handle this type of systems. It states that at any moment the active equations describing a physical phenomena by means of a set of variables have to be fulfilled concurrently. This means that during the continuous time simulation the number of continuous equations has to be equal to the number of variables involved, while during the discrete time events the number equations involved, both continuous and discrete, has to be match the number the number of variables. In addition, as Modelica cannot predict how many discrete events occur at a certain time, it is always assumed that all the discrete events may take place at the any time, so the total number of equations and variables have also to be the identical. If this is accomplish, Modelica is able to get an automatic synchronization among the continuous and discrete equations by means of data flow analysis. As a consequence, no simulation time is spent in the calculation and evaluation of the discrete equations [43]. If more information about hybrid modeling is needed the author of this work refers the reader to Ref.[44], where the synchronous data flow principle was introduced for the analysis of hybrid systems.

2.5.5 Software Component

The software component is one of the strong features of Modelica. It contains constructs that allow the development of interfaces for creating and connecting different components. Thus, the previously mentioned high level of abstraction during the modeling process is partially possible due to the software since the equations associated to the connections among components are automatically generated. Its strong software makes that Modelica can be regarded as an ideal architectural modeling and programming language for complex physical systems.

Three steps can be clearly differentiated during the simulation of a model. Firstly, the Modelica environment compiles the written code of the corresponding system to a some intermidiate code, usually C code. Secondly, this generated code is further compiled to machine code. Lastly, in order to get a solution, this code is executed with the proper solver, which may be an ordinary differential equation solver or a differential algebraic equation solver depending on the system that is simulated [42].

As it can be observed, just two types of equations systems have been mentioned. Partial differential equations have been omitted since they are still not contained in the Modelica languages. Efforts are being put in the development of this extension but further work is needed since the inclusion of concepts as geometric domains, domain boundaries, multi-variable functions and connections regions need to be done before releasing such extension [41].

2.6 Thermo-Hydraulic Modeling

The incorporation of renewable energy sources to the power market, the increasing environmental limitations and the rapid evolution of the power generation technology motivate the development of thermal power plants with high efficiency and flexible operation. Consequently, modern power plants are characterized by a high level of complexity and heterogeneity. In such scenario, modeling and simulation tools for these systems are required in order to ease the engineering work during both design and operation stages.

2.6.1 Thermo-Hydraulic Systems

The study of thermal power plants, specifically, and that thermo-hydraulic systems, generally, has an intrinsic difficulty due to the hetereogenity of the involved energy domains and the complexity of the equations that govern their behavior. Hence, the modeling and simulation of this type of systems may become extremely lengthy and tedious if a correct modeling language is not utilized. Many desirable features for the modeling of thermo-hydraulic processes, including power generation systems, are discussed in [45]. Among all of the characteristics mentioned in this paper, only the most important ones according to the author's opinion are here covered.

- **Modularity**. As discussed in Section 2.5, the modeling of large complex system may become easier if the assembling of simpler models that represent physical behaviors is allowed. Thus, the capability to utilize and generate these "blocks" and to put them together to create a more elaborated model should be available in a modeling environment for thermo-hydraulic systems.

- **Model availability**. If modularity is a characteristic of the modeling environment a large number of models are needed to develop more complex systems. As the number of the included models increases the modeling effort is reduced since a broader range of modeling and simulating possibilities is available for the user.

- **Transparency**. This characteristic refers to the closeness between the equations that describe a certain physical phenomena and how it is modeled. A modeling language where the resemblance between the model and the equations is achieved yields to a lost of the abstraction during the modeling procedure that is convenient for the description of thermo-hydraulic applications.

- **Openness**. Dealing with complex systems may lead to situations where specific models for certain applications are not available, even if the modeling environment contains rich libraries. Thus, the user have to be able to create their own models by either modifying or extending existing models or writing a new one from scratch.

- **Substance properties**. Physical properties of different substances are needed when computing and simulating any thermo-hydraulic system. Therefore, libraries with the required formulation and funtions to calculate these properties are essential.

2.6.2 Modelica as a Thermo-Hydraulic Modeling Tool

The adoption of Modelica as a thermo-hydraulic modeling tool is supported by the principles that conform this modeling language. From the object-orientation modeling approach modularity is enhanced as each object represents a physical phenomena and complex systems can be built from the assembly of elementary objects. This characteristic is boosted by the fact that Modelica has an extensive set of libraries where many models for different fields of application are available. Substance libraries are also included. They contain a generic interface to media property calculations.

Transparency is also enhanced in the Modelica language as the acausal modeling approach stimulates the similarity between the equations describing the physical behavior and the code included within the model. The acausal modeling principle included in Modelica also eases the reusability of the existing models since the bidirectional flow of information allows the utilization of an object independently of the unknown variables.

In addition, Modelica's open nature allows the development of new models through the modification of the existing ones and the creation of special models for specific applications. This characteristic favors the growth of the number of libraries available in Modelica.

Tummescheit et al. [46] present the overall concepts behind the design of a thermo-hydraulic model library in Modelica for homogeneous one- and two-phase flows. As Modelica does not support partial differential equations, a staggered grid method is utilized for the spatial discretization in this library. A modularity approach is constantly kept during the development of this library by means of using fluid control volume models where the equations of state, the medium model and all the necessary equations to describe the physical behavior are covered. This structure together with Modelica's openness permit the extension of the created models by the inclusion of the equations that describe other phenomena, as special heat transfer or pressure drop, which enhances the reusability of this library as well as inheritance.

The design of the Fluid and Media libraries contained in the Modelica standard library is described by Elmqvist et al. [47]. These libraries are created as a basis that may be utilized in the development of specialized thermo-hydraulic libraries. Thus, features as modularity, acausality, reusability and inheritance are deeply introduced during the design procedure. In the Fluid library, control volumes are also utilized for the study of each model, which contains state, media, mass, momentum and energy equations. Special attention is put in the replaceability of the media models and the bidirectional flow of information so general models capable to work with any fluid are created. Mass, momentum and energy equations are discretized by a finite volume method and integrated in the models in a generic way, which permits the expansion of the models for specific applications as well as the increase in accuracy by the modification of the discretization method. In the Media library, five thermodynamic variables (pressure, temperature, density, specific internal energy and specific enthalpy) and three algebraic equations are utilized for the characterization and calculation of a thermodynamic state. The relation among these variables is obtained in the medium model by the description of three of these variables as a function of the other two. Mass fractions are utilized when the media used is a mixture.

2.6.3 ThermoPower Library

The ThermoPower library [48] is an open Modelica library developed at Politecnico di Milano to provide the basic components for the modeling and simulation of thermal power plants. The approach followed during the development of this library is based on the same principles mentioned in Section 2.6.1 and widely discussed by Casella et al. [45].

The difference with other thermo-hydraulic libraries, as those covered in Section 2.6.2, is the narrower scope that the ThermoPower library possess. Generality is partially lost since some fluid behaviors never occur in thermal plants and they do not have to be covered by the models. Therefore, simplifying assumptions are introduced in order to ease the modeling and understanding of the processes involved. On the contrary, the lack of a general approach also allows to increase the level of detail in the models included in the library as they may be more relevant in power plants than in any other scenario. A remarkable example of this specialization characteristic is the fact that only specialized models for water and gas are available in the library. Attempting to describe the behavior of this fluids in different devices in a general way may lead to unnecessary complex models and since these are essentially the two fluids currently used in power plants the creation of different and more detailed models for each fluid was considered the best alternative.

The library is structured into five packages where simpler models are described [49] and an extra package where more complex and elaborated models are presented. These packages are:

- **Water**. Basic models of many components where water and steam are the working fluids are contained in this package.

- **Gas**. Elementary models of similar components as in the water package are contained in this package with the difference that the working fluid is here an ideal gas mixture.

- **Media**. The water and gas models, i.e. the models of the fluids, are included in this package. They are provided by the Modelica Media library described by Elmqvist et al. [47]. The water model is based on the EAPWS-IF97 formulation and the default gas model is based on a NASA property database.

- **Thermal**. Basic models to describe heat transfer phenomena are included in this package.

- **Electrical**. The content of this package is based on models that act as boundary conditions of the power plant, mainly electric generators and power grid connections. It is not a specialized library for modeling of complex electrical systems.

- **Power Plants**. Complex models created from the elementary models contained in the previously described packages are contained in this library. Gas turbines, heat recovery steam generation units, steam turbines and other specialized systems are included in order to ease the modeling of more elaborated structures.

Within these elementary packages, i.e. Water, Gas, Media, Thermal and Electrical, several types of models may be included. The qualitatively description of these models, the validation procedure and the overview of the equipment employed for such purpose is done by Casella et al. [22], proving the high-fidelity results that can be obtained with the ThermoPower library. If the reader is interested in a more mathematical description of the development of the models included in the library it is referred to the work done by Casella et al. [50], where the equations that govern the behavior of a 1-D heat exchanger are presented, the followed discretization is detailed and its implementation in Modelica is described. In this work, only a brief summary of the types of models contained the packages of the ThermoPower library is included.

- **Boundary Conditions**. Ideal pressure and mass flow sources and sinks are included in this classification. It is worth mentioning that a source can act as a sink and vice versa due to the bidirectional character allowed in the fluid by Modelica. Nevertheless, both elements are included for pure convention.

- **Branching components**. Flange terminals only allow two connections, so splits and joints are required when more than two streams are required.
- **Elementary components**. Within this category elements as valves, pumps, drums, mixers, collectors and more physical components are included. These basic models are created with a general approach in order to be able to use them when building more complex systems.
- **Building Blocks**. Pressure drops, 1-D fluid flow, metal walls and other physical phenomena and modules, as basic heat exchangers, are included in this type of model. They are meant as blocks for complex models where the work of modeling certain physical behavior is avoided with the utilization of these elements.

Chapter 3

Case Study and Methodology

A case study is utilized to asses the dynamic performance of a flexible combined heat and power plant for offshore applications in the Norwegian Continental Shelf. This case study and its main features regarding the expected heat and power demands are presented in Section 3.1 of this chapter. The proposed combined heat and power plant configuration and the description of its principal components are covered in Section 3.2. The methodology followed to study and analyze the dynamic performance of the proposed power generation system is described in Section 3.3. Firstly, some guidelines for the development of the steady-state modeling and design are given. Secondly, the assumptions selected for this stage are presented. Then, the optimization procedure and the selected steady-state design of the CHP plant are presented. Lastly, the methodology that was followed for the development of the dynamic model of the chosen power generation system is described. Its is important to note that a qualitative description of the dynamic modeling is here exposed. Details of this process are covered in Chapter 4.

3.1 Case Study

The case study utilized in this work to analyze the dynamic performance of a combined heat and power plant is the Johan Castberg field in the Norwegian Continental Shelf. It is situated in the Barents sea, 100 km north of the Snøhvit-field, 150 km from Goliat and nearly 240 km from Melkøya (see Figure 3.1). Johan Castberg field is a project formed by three different oil discoveries located in PL 532: Skrugard (2011), Havis (2012) and Drivis (2014) [51].

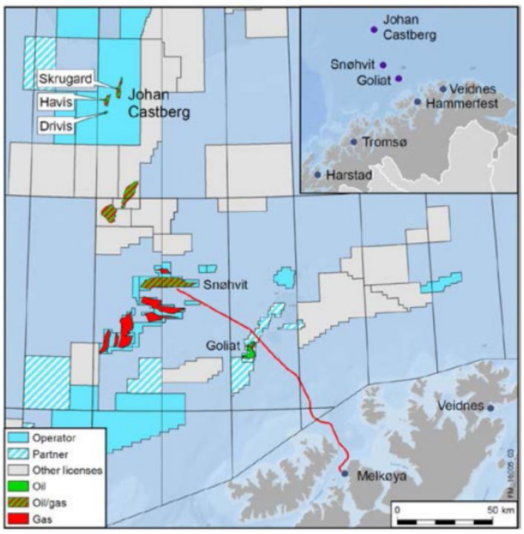


Figure 3.1: Location of Johan Castberg field.

The power generation system has to provide both power and heat to cover the demand of the offshore platform. The power demand can be separated into two types: shaft power, which is the mechanical work directly provided from the gas and steam turbines to the compressors' shaft, and electricity, which is used for running different components of the platform. The heat demand sources are the different processes that take place in the platform for conditioning the extracted oil and gas, so they meet the requirements demanded for its transport to shore.

The power and heat demand of Johan Castberg field has been assessed for its life-time [51]. The estimated distribution of the total power and heat demand throughout its production period is shown in Table 3.1. The represented values are the year average demands, but it is important to keep in mind that many fluctuations from these reference values may be expected during the regular operation of the combined heat and power plant.

Table 3.1: Power and heat demand throughout Johan Castberg's life-time operation.

Year	Electricity [MW]	Shaft Power [MW]	Elec+Shaft [MW]	Heat [MW]	Total Demand [MW]
2023	18	24	42	32	74
2024	22	22	44	36	80
2025	22	24	46	42	88
2026	22	32	54	46	100
2027	22	36	58	52	110
2028	22	36	58	50	108
2029	22	36	58	48	106
2030	22	36	58	46	104
2031	22	36	58	44	102
2032	22	36	58	42	100
2033	22	36	58	42	100
2034	22	36	58	42	100
2035	22	36	58	42	100
2036	22	36	58	42	100
2037	22	36	58	42	100
2038	22	32	54	40	94
2039	22	32	54	40	94
2040	22	32	54	42	96
2041	22	32	54	42	96
2042	22	32	54	44	98
2043	22	28	50	38	88
2044	22	28	50	38	88
2045	22	28	50	38	88
2046	22	28	50	38	88
2047	22	28	50	40	90
2048	22	28	50	40	90
2049	22	28	50	40	90
2050	22	28	50	42	92
2051	22	26	48	40	88
2052	22	26	48	42	90

The electric and shaft power are added in a column since this value will be more convenient when performing the simulations of the proposed alternatives for the power generation. The addition of both values represents the total power that has to be produced by the gas and steam turbines, independently of the use that will be given to this power, e.g. compressors' shaft power or electricity consumption. The demanded heat is not considered in the total amount of power (turbines' shaft power) that the gas turbines have to generate as it can be extracted from the high temperature exhaust gases. In this manner, heat that otherwise would be wasted is used.

A summary of Table 3.1 is represented in Figure 3.2, which shows a more visual image of the Johan Castberg's power demand by source.

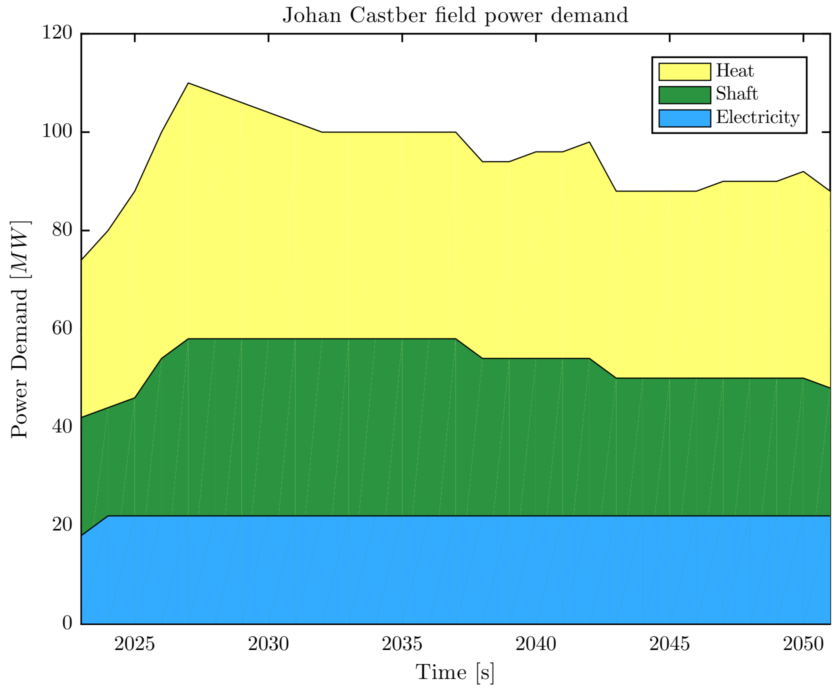


Figure 3.2: Johan Castberg's power and heat demand.

As it can be observed, the heat fraction of the total power demand is the largest demand of the field in every year of its life-time. However, if the parameter previously introduced turbine's total power is considered, the heat demand is always smaller. Nevertheless, the distribution between the power that has to be generated by the gas turbines and the heat needed for the process activities is unusual, as the turbine's total power is normally much larger than the heat demand. Thus, the energy requirements of Johan Castberg field suppose a challenge when designing a flexible power generation system able to satisfy both demands during dynamic operation.

The challenge becomes bigger when one realizes that the peak heat and power demands are expected to occur the same year, in 2027. Therefore, the designed power system has to be able to generate these peak quantities at the same time, which is the case where the heat demand-turbines' power output ratio is the highest. Then, if an offshore power system is able to provide the total power demand in 2027 it will be able to generate the needed heat and power in any other year of its operation-time.

3.2 Power System Layout Description

The selection of a power plant configuration is a fundamental step in the design of a power generation system. It not only affects its nominal operation but also the performance of the plant under changing demand scenarios, both quasy-steady and dynamic. Therefore, the functioning of a power plant during different operating conditions has to be assessed.

Several combined heat and power plant configurations were analyzed in Ref.[52]. A screening procedure where the performance of the proposed power plant designs was analyzed throughout the expected life-time operation of the offshore facility was followed. It was found that the best design was formed by a steam Rankine cycle where the hot source was formed by the exhaust gases of two gas turbines. The layout of the combined heat and power plant is shown in Figure 3.3.

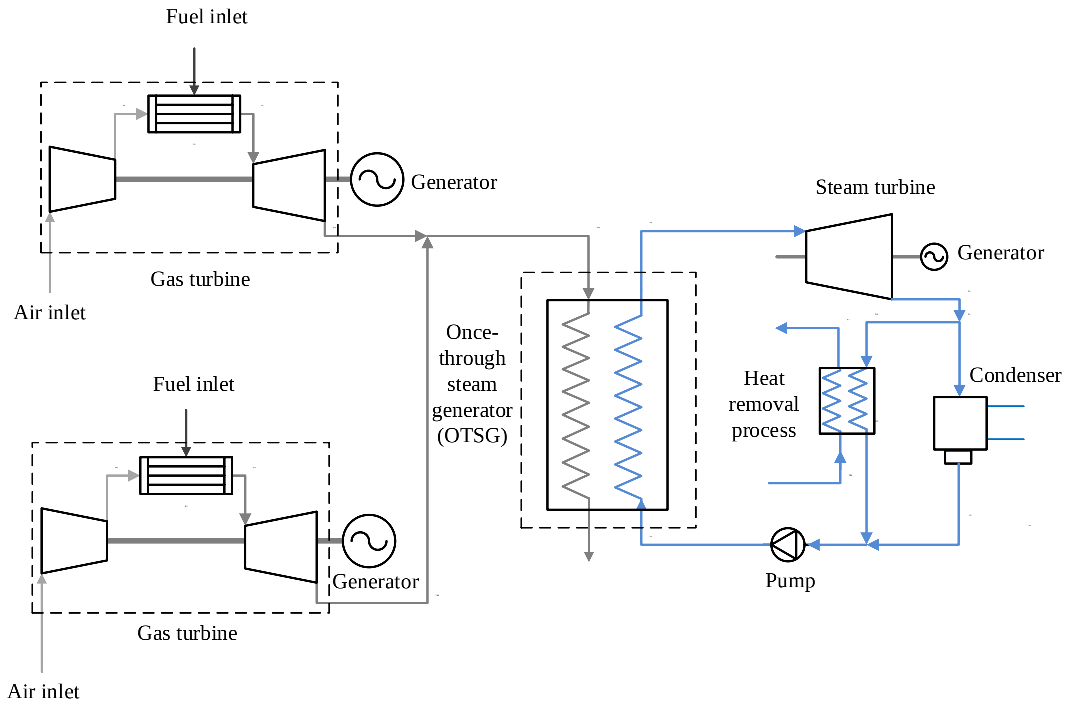


Figure 3.3: Layout of the proposed combined heat and power plant.

This cycle configuration recovers the energy contained in the exhaust gases of the gas turbines to produce the steam that is required in the Rankine bottoming cycle. The heat recovery is done in a once-through steam generator (OTSG) (see Section 3.2.1), where the temperature of the gases is considerably reduced and the entering subcooled water is heated up until it leaves as superheated steam. This steam is expanded in a back-pressure steam turbine, where some energy is left to produce the required process heat (see Section 3.2.2). A fraction of the turbine outlet steam is sent to the heat removal process where pressurized water is heated up in a condenser until the heat demand of the offshore facility is met. The remaining heat is condensed with seawater in the other condenser. The two flows of water leaving the condensers in the parallel configuration are joint and pumped to the OTSG.

3.2.1 Heat Recovery

Conventional combined cycles are normally composed by heat recovery systems with several pressure leves, drums and deaerators. These systems allow the working fluid to exchange energy with the waste heat stream more efficiently since the temperature profile along the heat recovery steam generator (HRSG) follows more closely the temperature curve of the waste gases, reducing the temperature difference in the heat exchange process and, hence, the exergy destruction (see Figure 3.4). Consequently, this type of configurations are characterized by high weights and volumes due to the amount of equipment and the large sizes of the heat exchangers, which need large heat transfer areas in order to reduce the temperature difference.

Offshore facilities are limited by the lack of space and the restrictions in weight. Costs increase rapidly with these two parameters and therefore they have to be kept as low as possible. Thus, the combined cycle configurations currently employed in onshore power plants are not suitable for offshore applications due to these limitations. The utilization of a once-through steam generator (OTSG), where a single pressure level is employed, is considered as a feasible alternative for power plants installed in offshore facilities. Nord and Bolland [53, 54] proved that a trade-off among compactness, flexibility and high efficiency may be achieved, even during off-design conditions, if the traditional heat recovery steam generator with several pressures levels and steam drums installed in the land-base power plants is replaced by a once-through steam generator for offshore applications. Therefore, this was the configuration selected for the combined heat and power plant proposed as the power generation system in Johan Castberg field.

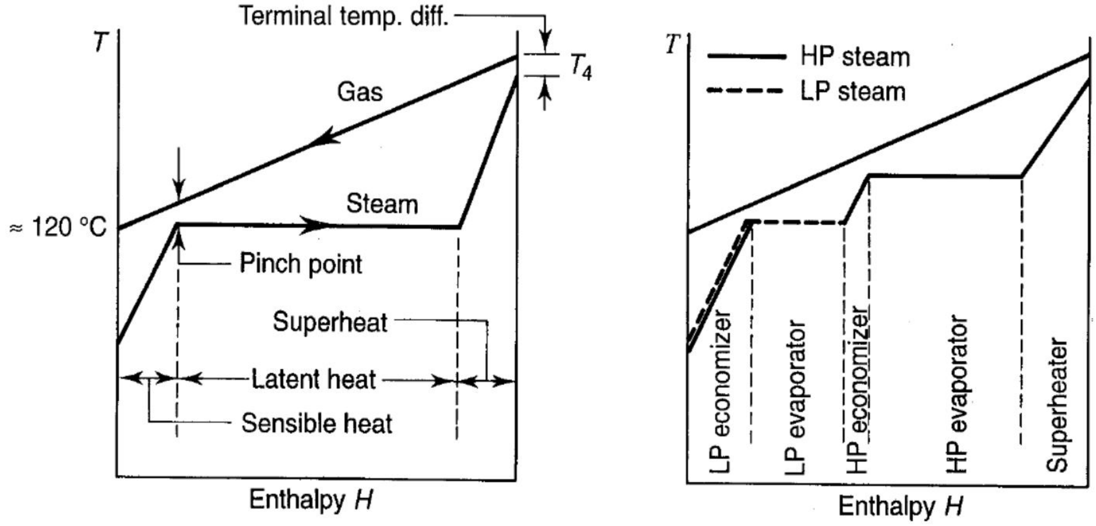


Figure 3.4: Comparison of T-h diagrams for single and dual-pressure heat recovery steam generators.

3.2.2 Steam Turbine

Extraction steam turbines are the most common expanders utilized in land-base power plants. Large amounts of power may be produced by these components as the expansion might occur until sub-atmospheric levels close to zero, e.g. 0.07 bar, which leads to considerable reductions in the power that has to be generated by the gas turbines and the fuel burnt by these units. In addition, steam can be extracted along the turbine at different pressures and in different quantities. This feature allows to produce process heat in a flexible manner since steam is exclusively extracted in the required amount to meet the heat demand, which allows to maximize the power generation of the bottoming cycle as the steam expanded in the turbine is always maximum.

The main drawback of this technology is its limitation in the heat production, as only small quantities of heat may be generated in comparison to the power produced. Therefore, when large amounts of heat are needed back-pressure steam turbines are required to be able to meet the heat demand while producing considerable electricity or shaft power. This turbine technology is based on an incomplete expansion of the working fluid to leave some energy at the outlet of the turbine that may be utilized to produce process heat. The main disadvantage of this type of turbines is its limited flexibility, as the energy remaining in flow at the outlet may be excessive in different operation points and, hence, it is wasted in the condenser. Although this limitation, a back-pressure steam turbine is chosen for the proposed CHP plant due to the large ratios between heat and power demands.

3.2.3 Process Heat Production

Changing conditions may be encountered during the life-time operation of the power plant. Thus, a process heat generation system that ensures the flexible operation of the entire power plant is fundamental. A parallel configuration where two shell and tube condensers are utilized for process heat production and steam condensation is selected. The main branch is where the process heat generation occurs. The flow sent to this condenser is controlled to ensure that the corresponding mass flow of pressurized water circulating in the condenser leaves the unit at the required temperature. The remaining steam at the outlet of the turbine is sent to the condenser in the secondary branch, where seawater is utilized as cooling fluid due to its availability and high heat transfer coefficient. In this manner, it is ensured that the working fluid enters the pumping system in liquid phase.

3.3 Methodology

3.3.1 Steady-State Design Methodology

The life-time assessment [51] of the energy requirements expected for the offshore oil and gas field shows that maximum demands of heat and power may occur simultaneously, as previously mentioned in Section 3.1. Hence, the nominal operation point is selected to correspond to peak demands of heat and power, 52 MW and 58 MW respectively, in order to ensure that these conditions are met by the power generation system.

Once the nominal operating conditions of the CHP plant are defined, the steady-state modeling of the components and the design of the thermodynamic cycle is performed. The detailed modeling and steady-state design of this thermal power plant is out of the scope of this work, and hence, only some guidelines are given in order to show the reader the methodology that should be followed to reach the end result of this process (see Ref.[55]). Firstly, the thermodynamic states at the inlet and outlet of each component are calculated based on mass and energy conservation laws, heat transfer correlations, pressure loss equations, and the boundary conditions imposed to the model. Secondly, from the thermodynamic data previously obtained, the sizing of the components integrating the power cycle is performed. From these two steps infinite possible combinations are available. Therefore, in order to find an optimal solution for the requirements set by the expected functioning in the offshore facility, a multi-objective optimization based on a genetic algorithm is carried out. As a result, a Pareto front of solutions of the preliminary design of the CHP plant is obtained, from where the most suitable alternative is selected. Once the nominal operation point of the thermal power cycle and the size of its components are known, the size of the secondary condenser is recalculated in off-design conditions for maximum power demand and minimum heat demand, since this is the operation point where more mass flow circulates through this unit.

The specialized software Thermoflex [56] was chosen to perform the steady-state modeling and design of the CHP plant. This selection was based on two fundamental reasons:

- **Reliability.** Thermoflex is a modeling and simulation tool that has been extensively tested and validated with experimental data. Therefore, the results produced by this software are expected to be both accurate and reliable.

- **Sizing feature.** Thermoflex includes all the equations and correlations needed to calculate the dimensions of the cycle components, indispensable information to carry out the dynamic operation assessment of any power plant. Sizing of the elements integrating the thermal power plant was included due to the importance of the components' sizes in the dynamic model that had to be developed in order to asses the unsteady operation of the proposed CHP plant. The dimensions of each element affects the transient operation of the thermal power plant, as their inertia and capacitance are strongly influenced by their dimensions.

3.3.2 Steady-State Design Assumptions

The gas turbine model GE LM2500+G4 was selected for both gas turbines of the proposed CHP plant. Its operation is modeled by performance maps provided by the manufactured and included in Thermoflex. The main design point specifications of this gas turbine model are covered in Table 3.2.

Table 3.2: Design point specifications of the GE LM2500+G4 gas turbine.

Variable	Design point value
Net power output [MW]	32.50
Net efficiency [%]	36.50
Exhaust gas flow rate [kg/s]	89.90
Exhaust temperature [°C]	552
Net heat rate [kJ/kWh]	9867
Frequency [Hz]	60
Inlet pressure drop [mbar]	10
Outlet pressure drop [mbar]	10
Fuel	Natural gas

The assumptions selected for the components of the steam Rankine cycle are summarized in Table 3.3.

Table 3.3: Overall assumptions of the steam Rankine cycle components.

Component	Parameter	Assumption
OTSG	Tube material	Incoloy
	Fin material	T409
	Tube arrangement	Staggered
Condenser (Process heat)	Inlet temperature [°C]	70
	Outlet temperature [°C]	150
	Inlet pressure [bar]	20
	Tube material	Titanium
Condenser	Inlet temperature [°C]	25
	Inlet pressure [bar]	1.021
	Tube material	Titanium
Pump	Efficiency [%]	75

3.3.3 Optimization

Combined heat and power cycles are regarded as a promising and feasible alternative to the traditional gas turbines due to their higher efficiency and to their technology maturity. However, when offshore applications are considered, weight and space are fundamental criteria in the design stage. Therefore, a constrained multi-objective optimization was carried out in order to account for both features. The objective functions selected for the optimization were the minimization of the weight and the heat rate HR, which is defined in Equation 3.1 as:

$$HR = \frac{3600}{\eta_{net}} = \frac{3600 \cdot \dot{m}_f \cdot LHV_f}{\dot{W}} \tag{3.1}$$

where η_{net} is the net efficiency of the power plant, $\dot{m}_f$ and LHV_f are the mass flow of fuel and its low heating value, and $\dot{W}$ is the power generated.

The decision variables were chosen according their influence on the overall performance they have on both objective functions. These variables were the steam turbine inlet pressure (p_{steam}), the steam turbine inlet temperature (T_{steam}), the minimum temperature difference in the evaporating section of the OTSG (ΔT_{pinch}), and the load of the second gas turbine ($GT_{2,load}$). The load of the first gas turbine is modified in order to match the power demand. The lower and upper bounds of these decision variables are covered in Table 3.4.

Table 3.4: Lower and upper bounds of the selected decision variables.

Decision variables	Lower bound	Upper bound
p_{steam} [bar]	10	40
T_{steam} [°C]	400	515
ΔT_{pinch} [°C]	10	30
$GT_{2,load}$ [%]	75	94

The multi-objective optimization process produces a two-dimension Pareto front of solutions of preliminary designs for the combined heat and power plant. These solutions are represented in Figure 3.5. The efficiency of the thermal power plant is plotted in the x-axis instead of the heat rate since it eases the understanding of the Pareto front and no error is introduced as they are inversely proportional (see Equation 3.1). The weight of the power generation system is plotted in the y-axis. The final values of the decision variables utilized during the multi-objective optimization for each of the found optimal solutions are covered in Table 3.5.

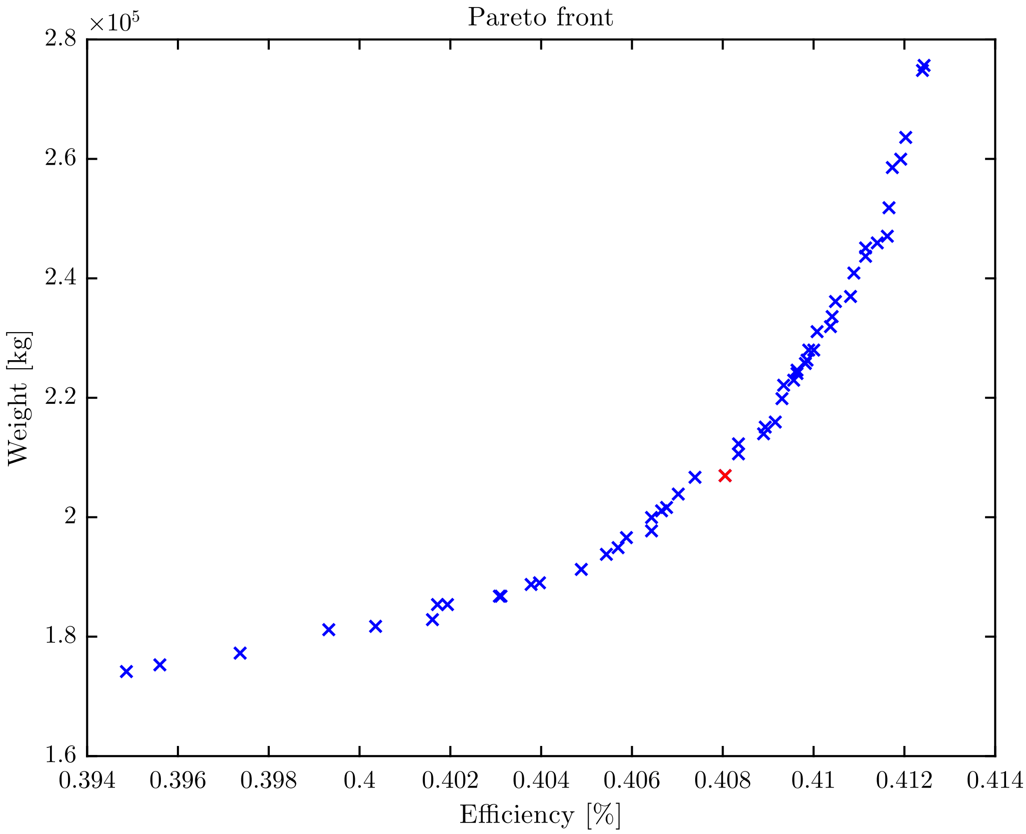


Figure 3.5: Pareto front of optimum solutions for the preliminary design of the CHP plant. Selected design is highlighted with the red marker.

Table 3.5: Efficiency, weight and decision variable values of the Pareto front of optimum solutions.

Case	Efficiency, η_{net} [%]	Weight [tonne]	p_{steam} [bar]	T_{steam}[°C]	ΔT_{pinch} [°C]	$GT_{2,load}$ [%]
1	41.24	274.74	36.44	504.84	10.55	85.88
2	40.93	219.77	34.10	472.14	17.40	87.01
3	40.40	188.94	28.02	404.06	28.29	76.99
4	40.97	224.53	35.84	453.95	16.06	85.68
5	40.64	199.89	29.44	458.22	25.21	83.78
6	40.67	200.96	30.86	445.23	24.22	84.48
7	40.49	191.35	28.31	437.90	28.60	80.74
8	40.64	197.63	30.89	440.63	24.81	83.66
9	40.59	196.72	29.59	441.61	25.15	78.30
10	40.80	207.07	33.79	444.02	20.30	85.19
11	41.04	232.02	35.61	470.86	14.59	85.69
12	40.31	186.85	24.42	437.44	29.18	79.40
13	41.16	246.97	36.57	491.71	11.44	85.86
14	40.91	215.98	35.28	457.32	17.66	87.16
15	40.04	181.68	20.59	403.53	28.91	78.16
16	41.20	263.59	36.45	497.34	10.91	85.87
17	40.83	210.50	34.09	446.52	19.78	85.71
18	40.83	212.19	33.72	456.01	19.23	78.96
19	41.17	258.64	35.97	494.53	11.27	85.89
20	41.09	240.80	35.70	483.80	12.94	85.74
21	40.70	203.76	31.36	446.42	21.86	82.14
22	40.38	188.75	27.52	404.56	28.29	76.99
23	41.08	236.86	36.07	478.87	13.63	85.08
24	40.74	206.72	31.66	443.93	20.07	86.05
25	41.19	259.98	36.49	495.03	11.02	85.87
26	40.57	194.98	32.21	409.83	26.78	84.37
27	40.54	193.81	28.61	447.66	27.23	80.79
28	39.56	175.27	15.16	412.22	28.73	79.52
29	41.01	231.16	35.67	461.86	14.61	85.60
30	40.98	225.83	35.96	456.56	15.63	85.73
31	41.11	243.62	35.81	482.24	12.21	85.85
32	40.17	185.48	22.54	414.36	29.28	77.20
33	40.96	222.84	35.52	459.34	16.90	85.06
34	39.93	181.09	18.88	405.82	28.19	79.86
35	41.05	236.00	35.40	475.61	14.16	85.96
36	41.24	275.75	36.44	504.84	10.30	85.88
37	40.89	214.11	34.29	456.19	18.32	85.22
38	40.99	227.92	35.58	464.03	15.88	85.01
39	41.00	228.05	35.20	470.68	16.04	85.78
40	40.93	222.14	35.41	452.44	16.96	85.66
41	41.14	245.87	35.82	492.69	11.71	85.87
42	40.99	226.27	35.96	457.13	15.48	85.71
43	40.89	215.15	35.02	449.02	17.75	85.02
44	41.16	251.72	36.50	486.06	11.34	85.87
45	41.04	233.71	36.25	463.65	13.86	85.73
46	40.68	201.76	31.80	440.07	23.02	79.75
47	39.74	177.34	16.22	412.03	28.22	84.51
48	40.96	224.02	35.52	459.34	16.40	85.06
49	41.11	244.99	36.07	483.44	11.70	85.23
50	40.16	182.90	22.84	406.58	29.49	78.59
51	40.19	185.50	23.04	413.86	29.53	77.20
52	40.31	186.89	24.42	437.94	29.18	79.40
53	39.48	174.25	15.11	402.80	28.82	78.79

As it may be observed in Table 3.5 and in Figure 3.5, the solution that is belief that provides the best balance between weight and efficiency is selected. The T-s diagram of the heat exchange occuring in the OTSG between the exhaust gas and the working fluid of the selected design conditions of the combined heat and power plant is shown in Figure 3.6. The large pinch point temperature, i.e. ΔT_{pinch}, and the utilization of a single pressure level produce big temperature differences during the heat exchange process and, hence, large exergy destruction. However, even if this configuration is not ideal from a thermodynamic perspective, the weight savings associated to this configuration make it the most suitable for the offshore facility considered in this work.

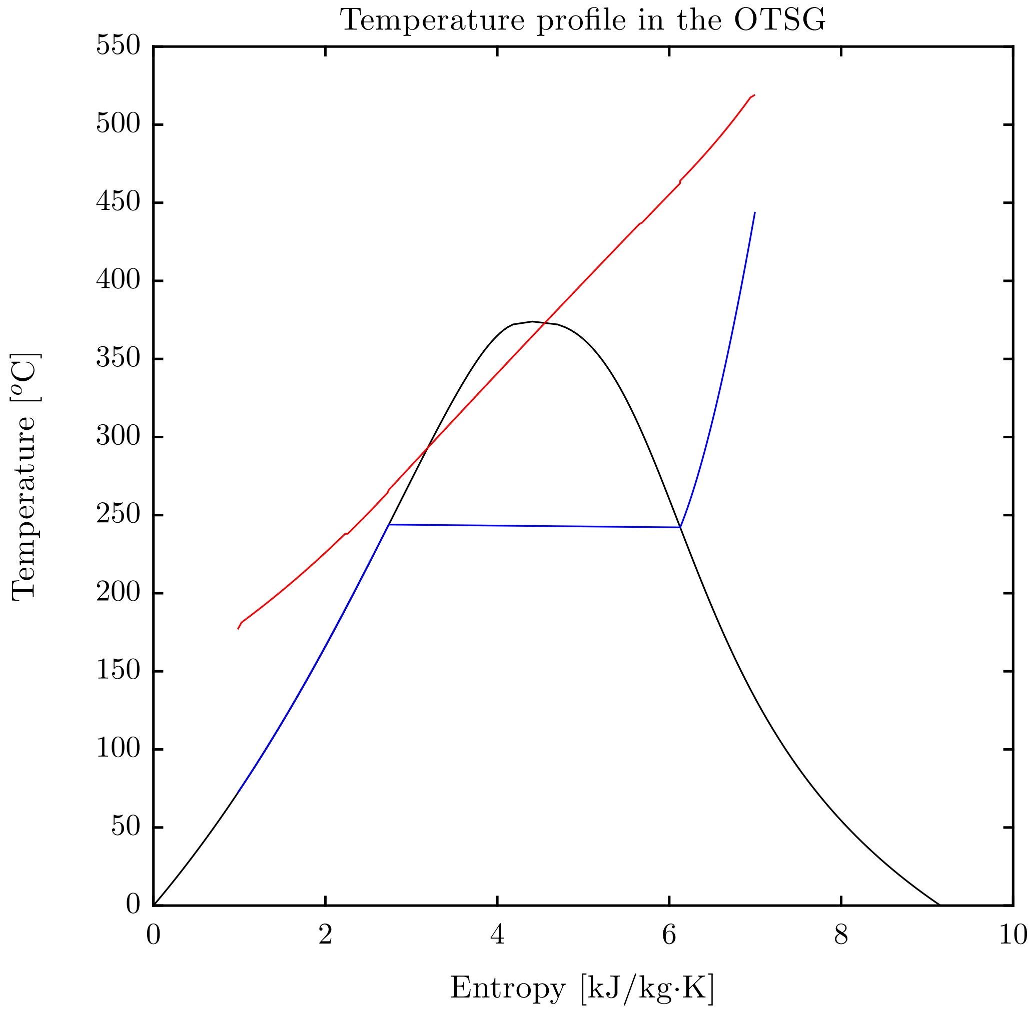


Figure 3.6: T-s diagram of the temperature profile of the exhaust and working fluid along the once-through steam generator (OTSG).

3.3.4 Dynamic Modeling Methodology

The assessment of the dynamic operation of thermal power plants is becoming a fundamental step in the design stage since it may predict possible imbalances between the demand and generation in certain scenarios. Offshore applications are examples of where transient operation analysis is specially useful, as they operate as stand-alone systems that have to provide the entire heat and power demand of the facility. Therefore, in order to guarantee that a specific power cycle design is able to perform correctly under varying scenarios, e.g. load changes or unit trips, a dynamic model of such system must be built.

The dynamic modeling language Modelica was utilized for the development of the proposed CHP plant dynamic model. Its object-oriented nature and the existence of specialized libraries eases the implementation of unsteady conservation laws and correlations (see Section 2.5 and Section 2.6 for a more detailed description of Modelica and the library utilized). In addition, the specialized software Dymola was employed, as it enhances the graphical programming feature that object-oriented modeling languages provide. Therefore, it allows to develop the dynamic model by linking the different objects that represent the elements forming the CHP plant. Each of the components integrating the combined heat and power plant were modeled individually by means of the components included in the ThermoPower library and objects specially programmed for this dynamic model. This task is extensively covered in Chapter 4. The sizes of the components that were calculated during the steady-design stage were implemented in the dynamic model of each component in order to account for their effect during transient operation.

Once the dynamic model of the proposed CHP plant was built and the characteristics of each component were included in their models, a extensive validation was carried out. Transient operation data was not available and only steady-state validation could be done. Nevertheless, in order to ensure that the system was able to provide reliable yet accurate results, both design and off-design steady-state validation was performed. This data was obtained by the steady-state model developed in Thermoflex, since it was assumed that the results provided by this software were reliable enough for this purpose due to the extensive experimental validations that it has followed throughout its development.

Subsequently, open-loop simulations, i.e. without any control strategy implemented, were carried out in order to achieve a deeper understanding of the intrinsic unsteady behavior of the proposed combined heat and power plant. This provided more detailed information about the time orders that may be expected in each component and where the transient performance had larger effects. The transient performance of the power plant was triggered by step changes in the mass flow and temperature of the exhaust gases, and in the mass flow of the pressurized water circulating in the heat production process.

Dynamic simulations of the power plant with a control strategy were also performed, and compared with the open-loop results previously obtained in order to analyze the effects that the proposed control strategy had in the CHP plant. Three main control structures were implemented:

- **Pressurize water outlet temperature control.** The mass flow of pressurized water is modified according to the heat demand required in the offshore facility. Therefore, as a specific outlet temperature of 150 $°C$ is needed, the mass flow of steam circulating in the shell of the primary condenser has to be controlled in order to ensure that the temperature of the pressurized water at the outlet of this component is adequate.

- **Steam turbine inlet temperature control.** The temperature at the inlet of the steam turbine has to be controlled to ensure that no damages occur in the first stage of this component due to high temperatures. Thus, the mass flow of water circulating in the Rankine cycle is modified by the pump in order to fix this temperature. A cascade control strategy was followed for this purpose.

- **Hotwell's level control.** The water level of one of the condenser hotwells needs to be controlled so a stable operation of the power plant is achieved. The other hotwell acts as a buffer, fluctuating with the operation changes that the CHP plant may experience.

Chapter 4

Dynamic Modeling Approach

The description of the models developed for the simulation of the transient operation of a combined heat and power plant is done throughout this chapter. The modeling of the system is presented component by component, as the object-oriented feature of Modelica allows to model individually the behavior of each element that integrates the power system. Therefore, the mathematical equations that govern the dynamic performance of the power plant components are shown, pointing out the assumptions utilized and the methods employed to make possible the implementation of these models, i.e. discretization methods and development of performance maps. This chapter is divided into sections, corresponding each of them to the model of a different component of the combined heat and power plant. Thus, Section 4.1 describes how the gas turbine is modeled and the assumptions utilized regarding its transient performance. A thorough description of the once-through steam generator is presented in Section 4.2. The approach followed to model the steam turbine off-design performance is detailed in Section 4.3. The importance of the condenser model in the correct modeling of the entire plant dynamic performance are discussed in Section 4.4. Lastly, the pump model is described in Section 4.5.

4.1 Gas turbine

Gas turbines are devices characterized by their operation flexibility. Changes of load occur in the order of seconds when the turbine is operating regularly, i.e. longer times may be required in more critical scenarios like the start-up of the turbine. Therefore, gas turbines' dynamics were not considered as their transient operation is much faster than that of the steam Rankine cycle, which is characterized by tens of minutes due to the large thermal capacitance associated to the steam generator.

Gas turbines were modeled with validated quasi-static models, and the exhaust gas stream leaving the turbines was utilized as boundary conditions to the dynamic process model. Off-design performance conditions were obtained from validated operation maps of the gas turbine model provided by the software Thermoflex. This set of different states was implemented in the dynamic model by a variable characteristic that models the varying exhaust conditions. The composition of the exhaust gas is covered in Table 4.1:

Table 4.1: Mole composition of the gas turbines' exhaust gas.

Component	Mole Composition [%]
N_2	74.904
O_2	13.918
CO_2	3.119
H_2O_v	7.157
Ar	0.902

4.2 OTSG model

Once-through steam generators are heat exchangers where heat transfer takes place at single pressure and in a continuous form, i.e. no drums or dearators are installed between sections of the OTSG. Phase changes occur along the heat exchanger without delimited regions for each fluid state.

Three different sections were defined for the dynamic modeling of the OTSG, albeit its continuous configuration. This approach was selected due to the large differences between the physics that govern the heat transfer process for the different states, i.e. subcooled water, two-phase flow, and superheated steam; that are present along the heat exchanger. Convection heat transfer coefficients are strongly influenced by the working fluid phase and, hence, a more detailed modeling of the heat exchange is achieved if these parameters are calculated separately for each fluid phase. A general model of the OTSG could also be done assuming an overall heat transfer coefficient. This model would reduce the amount of equations, the complexity of the power plant model and the computational time. However, it was belief by the author that this approach may be implemented once that it was proven that the detailed OTSG model performed correctly, as simplifications should be done from detailed and complex to simple and general.

The three differentiated sections represent an economizer, an once-through boiler (OTB), and a superheater. Preheating of the working fluid occurs in the economizer, where pressurized water enters and leaves in subcooled conditions. Sufficient difference between the boiling point temperature and the economizer outlet temperature was left in order to ensure that no evaporation took place in this component during off-design conditions. This temperature difference is called approach temperature and it was defined by Thermoflex, as it is a fundamental parameter during the design stage of the OTSG. The intermediate section, the OTB, mainly modeled the vaporization of the working fluid, but it also included the preheating of the subcooled water leaving the economizer until saturated conditions, and some superheating of the steam produced. The superheating section of the OTB only accounts for less than 2% of the total surface are of the OTB and, hence, it was possible that superheated steam may not be produced in the OTB during off-design operation. The last modeled section of the OTSG was the superheater, where heat is exchanged to achieve the final degree of superheating of the steam produced in the OTB.

The overall data of the dimensions utilized for the dynamic modeling of the OTSG is covered in Table 4.2:

Table 4.2: Geometry data employed during the development of the OTSG dynamic model.

Parameter	Section		
	Economizer	OTB	Superheater
Number of tubes	320	960	240
Tube length [m]	9.74	9.74	9.74
Tube outer diameter [mm]	25.40	31.75	31.75
Tube inner diameter [mm]	22.10	27.94	27.94
Fin height [mm]	12.70	9.53	9.53
Fin spacing [mm]	2.63	2.34	6.41
Gas path frontal area [m2]	27.10	27.10	27.10
Tube surface/Heat transfer area	0.0658	0.0809	0.1946
Heat exchanger total outside area [m2]	2737.10	8070.00	1036.30
Preheating section (design cond.) [%]	-	28.70	-
Evaporating section (design cond.) [%]	-	70.20	-
Superheating section (design cond.)[%]	-	1.10	-

Modelica language can handle time derivatives, which are calculated by the solvers integrated in Dymola. However, space and partial derivatives are still not included as a feature of this modeling and programming language (see Section 2.5). Thus, each of the heat exchangers integrating the dynamic OTSG model was spatially discretized in (N-1) cells, or N nodes, where dynamic mass and energy conservation laws, and heat transfer correlations were applied (see Figure 4.1). Dynamic momentum conservation law was not included in the modeling of these sections as the inertia of the fluid was neglected and the pressure drops occurring along the heat exchangers were modeled as lumped parameters at the outlet of each component.

The equations that govern the gas and water/steam one-dimensional flow in the heat exchangers, assuming that the cross-section of the tubes in the tube bundle, A, is constant; the velocity profile, u, is uniformly distributed; and that the kinetic and thermal diffusion terms are negligible in the energy equation, read:

$$\frac{\partial \rho}{\partial t} = \frac{\partial (\rho u)}{\partial x} \quad \text{or} \quad A\frac{\partial \rho}{\partial t} = A\frac{\partial (\rho u)}{\partial x} \tag{4.1}$$

$$A\rho\frac{\partial h}{\partial t} + A\rho u\frac{\partial h}{\partial x} = \omega \dot{q} \tag{4.2}$$

where ρ is the medium density, the product $A\rho u$ is the mass flow rate $\dot{m}$, h is the medium specific enthalpy, ω is the wet perimeter, and $\dot{q}$ is the heat flux.

The density variation respect to time, $\frac{\partial \rho}{\partial t}$, in Equation 4.1 may be written as:

$$\frac{\partial \rho}{\partial t} = \frac{\partial \rho}{\partial h}\bigg|_p \frac{\partial h}{\partial t} + \frac{\partial \rho}{\partial p}\bigg|_h \frac{\partial p}{\partial t} \tag{4.3}$$

where $\frac{\partial \rho}{\partial h}\Big|_p$ is the medium density variation respect to enthalpy at constant pressure, $\frac{\partial \rho}{\partial p}\Big|_h$ is the medium density variation respect to pressure at constant enthalpy, and $\frac{\partial h}{\partial t}$ and $\frac{\partial p}{\partial t}$ are the specific enthalpy and pressure variations respect to time.

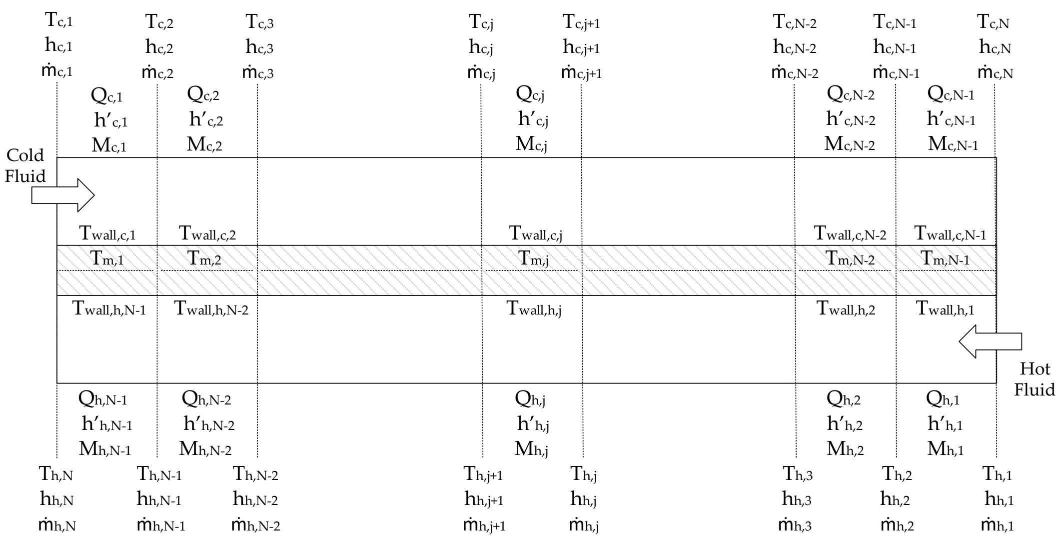


Figure 4.1: Modeling paradigm of the once-through steam generator.

If Equation 4.1 and 4.2 are integrated along the heat exchanger tube length L:

$$\int_0^L A\frac{\partial\rho}{\partial t}dx = \int_0^L A\frac{\partial(\rho u)}{\partial x}dx \tag{4.4}$$

$$\int_0^L A\rho\frac{\partial h}{\partial t}dx + \int_0^L A\rho u\frac{\partial h}{\partial x}dx = \int_0^L \omega\dot{q}\,dx \tag{4.5}$$

the following expressions are obtained:

$$V\frac{d\rho}{dt} = \dot{m}_{in} + \dot{m}_{out} \quad \text{or} \quad \frac{dM}{dt} = \dot{m}_{in} + \dot{m}_{out} \tag{4.6}$$

$$V\rho\frac{dh}{dt} + \dot{m}_{out}h_{out} - \dot{m}_{in}h_{in} = \omega L\dot{q} \tag{4.7}$$

where V and M are the fluid volume and mass respectively, $\dot{m}_{in}$ and $\dot{m}_{out}$ are the mass flow rates entering and leaving the heat exchanger, and h_{in} and h_{out} are the fluid specific enthalpies at the inlet and outlet. The product $\omega L\dot{q}$ represents the heat flow and may be written as $\dot{Q}$.

Once that the equations that describe the gas and water/steam flow circulating in the heat exchangers are presented, their discretization is carried out in order they can be implemented in Modelica. The spatially discretized dynamic mass conservation law for a single cell reads:

$$\frac{dM_j}{dt} = \dot{m}_j + \dot{m}_{j+1} \tag{4.8}$$

where M_j is the fluid mass in cell j, $\dot{m}_j$ is the mass flow rate in node j, i.e. the left node of cell j, and $\dot{m}_{j+1}$ is the mass flow rate in node $j+1$, i.e. the right node of cell j. Similarly to Equation 4.3, the mass change respect to time in a single cell $\frac{dM_j}{dt}$ may be expressed as:

$$\frac{dM_j}{dt} = V_j\left(\overline{\frac{\partial\rho}{\partial h}}\bigg|_j\frac{dh'_j}{dt} + \overline{\frac{\partial\rho}{\partial p}}\bigg|_j\frac{dp}{dt}\right) \tag{4.9}$$

where V_j is the fluid volume in cell j, $\frac{dh'_j}{dt}$ is the time derivative of the specific enthalpy evaluated in the center of the cell, and $\frac{dp}{dt}$ is the pressure variation respect to time. The overall density derivative respect to the enthalpy, $\overline{\frac{\partial\rho}{\partial h}}\Big|_j$, and pressure, $\overline{\frac{\partial\rho}{\partial p}}\Big|_j$, in the center of cell j are defined as the average values of the these partial derivatives evaluated at the nodes j and $j+1$ of cell j:

$$\overline{\frac{\partial\rho}{\partial h}}\bigg|_j = \frac{1}{2}\left(\frac{\partial\rho}{\partial h}\bigg|_{j+1} + \frac{\partial\rho}{\partial h}\bigg|_j\right) \tag{4.10}$$

$$\overline{\frac{\partial\rho}{\partial p}}\bigg|_j = \frac{1}{2}\left(\frac{\partial\rho}{\partial p}\bigg|_{j+1} + \frac{\partial\rho}{\partial p}\bigg|_j\right) \tag{4.11}$$

It is important to note that the partial derivatives of the density ρ respect to other thermodynamic variables, namely enthalpy and pressure, are calculated from the fluid properties incorporated in the Modelica and ThermoPower libraries entering the values of pressure p and enthalpy h (note that it is not h') at the nodes of the cell.

The semi-discretization of the dynamic energy conservation law for a single cell reads:

$$V_j\bar{\rho}_j\frac{dh'_j}{dt} - \bar{\dot{m}}_j(h_{j+1} - h_j) = \dot{Q}_j \tag{4.12}$$

where $\bar{\rho}_j$ is the average density in cell j, h'_j is the enthalpy state variable at the center of the cell, $\dot{Q}_j$ is the heat flow in the center of cell j, h_j and h_{j+1} are the specific enthalpies at the left and right nodes of cell j, respectively; and $\bar{\dot{m}}_j$ is the average mass flow rate in each cell, defined as:

$$\bar{\dot{m}}_j = \dot{m}_{in} - \sum_{j=1}^{j-1} \frac{dM_j}{dt} - \frac{1}{2}\frac{dM_j}{dt} \tag{4.13}$$

The heat flow appearing in Equation 4.12 between the hot and cold fluids, denoted by the sub-index h and C respectively, and the metal tube were calculated in the center of each cell as follows:

$$\dot{Q}_{h,j} = \gamma_{h,j} \cdot A_{h,j} \cdot \left(\frac{T_{h,j+1} + T_{h,j}}{2} - T_{wall,h,j}\right) \tag{4.14}$$

$$\dot{Q}_{c,j} = \gamma_{c,j} \cdot A_{c,j} \cdot \left(\frac{T_{c,j+1} + T_{c,j}}{2} - T_{wall,c,j}\right) \tag{4.15}$$

being T_j and T_{j+1} the temperature of the fluid in the left and right nodes of cell j, A_j the heat surface area of each discretization, $Twall, j$ is the temperature of the metal wall in the contact surface with the hot or cold fluid evaluated in the center of each cell, and γ_j is the convective heat transfer coefficient for the hot and cold fluids in the the center of each cell.

The heat transfer process is accounted in Modelica and Dymola by specific blocks connected with the objects that model the physics of the fluid flow and the temperature gradient in the metal tube. Therefore, the sign of each discretized heat flow, $\dot{Q}_j$ is included in the heat exchanger model as boundary conditions between the models that characterize the behavior of the flow and the heat transfer. In other words, the heat flow from the hot fluid to the external metal tube surface should be negative according to the sign criteria employed, however, it is calculated as positive in the object modeling the hot fluid flow (see Equation 4.14), and in the connection between this object and the object modeling the heat transfer process between the hot fluid and the metal tube it is defined as negative.

The main difference between the modeling of the gas and water/steam flow lies in the calculation of the convective heat transfer coefficients. In the gas side, this parameter, $\gamma_{gas,j}$, was calculated with the relation [57]:

$$\gamma_{gas,j} = \gamma_{gas,nom} \left(\frac{\bar{\dot{m}}_j}{\dot{m}_{nom}}\right)^{0.6} \tag{4.16}$$

where $\dot{m}_{nom}$ is the gas mass flow rate in nominal conditions, $\bar{\dot{m}}_j$ is the mass flow rate of gas at any instant in the center of the cell j, and $\gamma_{gas,nom}$ is the convective heat transfer coefficient in the gas side calculated by Thermoflex during design conditions.

The convective heat transfer coefficients in the water side were calculated following different approaches depending on the fluid phase. A constant heat transfer value calculated during the design phase of the CHP plant was utilized in the economizer and the superheater since the main thermal resistance for the heat exchange process was found in the gas side of the heat exchanger. This assumption was also possible since no evaporation was expected to occur in the economizer model due to the approach temperature chosen during the design stage of the thermal power plant, and almost all the evaporation takes place in the OTB even during off-design conditions. However, the convective heat transfer coefficient could not be assumed constant in the OTB due to the coexistence of three different fluid states (subcooled water, two-phase flow, and superheated steam) that may occur in this model. Therefore, the Dittus-Boelter correlation with constant value consisting on the saturated boiling region was utilized. This implies that the Dittus-Boelter correlations shown in Equation 4.17 was employed when the fluid was a single-phase flow, i.e. subcooled water or superheated steam, and a constant coefficient accounting for the nucleate and convective phases of the evaporation was utilized for the two-phase flow. The Dittus-Boelter correlation reads:

$$\gamma_{w,j} = 0.023 \frac{k_j}{D_{hyd,j}} Re_j^{0.8} Pr_j^{0.4} \tag{4.17}$$

where k_j is the thermal conductivity of the fluid evaluated at the nodes, $D_{hyd,j}$ is the hydraulic diameter in cell j, Re_j is the Reynolds number, and Pr_j is the Prandtl number. The latter three parameter are defined as:

$$D_{hyd,j} = \frac{4 * A_j}{Pe_j} \tag{4.18}$$

$$Re_j = \frac{\bar{\dot{m}}_j \, D_{hyd,j}}{A_j \mu_j} \tag{4.19}$$

$$Pr_j = \frac{Cp_j \, \mu_j}{k_j} \tag{4.20}$$

where Pe_j is the perimeter of the metal tubes, μ_j is the dynamic viscosity evaluated at the nodes, and Cp_j is the specific heat calculated at the nodes.

The dynamic modeling of the tubes' thermal performance is a fundamental step in the development of heat exchanger models for predicting the unsteady operation of a thermal power plant. The material they are composed of possesses a heat capacity that affects substantially the transient heat transfer. Therefore, Equation 4.21, 4.22 and 4.23 are included in the dynamic heat exchanger models in order to account for this phenomena. The former represents the energy balance through the metal tube wall, whereas the two latter describe the heat conduction through the wall. These equations are given for a single tube, and read:

$$A_m \, L \, \rho_m \, Cp_m \frac{dT_{m,j}}{dt} = \dot{Q}_{h,j} + \dot{Q}_{c,j} \tag{4.21}$$

$$\dot{Q}_h = \frac{\lambda \frac{2\pi L}{N-1} \cdot (T_{wall,h,j} - T_{m,j})}{\log\left(\frac{2 \cdot r_{ext}}{r_{ext} + r_{int}}\right)} \tag{4.22}$$

$$\dot{Q}_c = \frac{\lambda \frac{2\pi L}{N-1} \cdot (T_{wall,c,j} - T_{m,j})}{\log\left(\frac{r_{ext} + r_{int}}{2 \cdot r_{int}}\right)} \tag{4.23}$$

where A_m is the cross-section area of the metal tube, ρ_m is the density of the material, Cp_m is the specific heat capacity of the material, $T_{m,j}$ is the metal temperature in the middle of the inner a outer surfaces of the tube in each cell, λ is a constant thermal conductivity associated to the material, and r_{int} and r_{ext} are the internal and external radius of the tube.

4.3 Steam Turbine model

A quasy-static model of the steam turbine was utilized in this work. The dynamics of this unit are faster than those of the OTSG, and since frequency fluctuations were not analyzed, the dynamic behavior of the steam turbine was neglected. This approach is extensively followed in the literature, e.g. see [24, 25, 37, 38], as the thermal capacitance of the rotor and stator blades, and the shaft are small, easing in this way the modeling of the expander.

Therefore, the differential conservation equations that govern the flow behavior along the expansion in the turbine were reduced to algebraic equations, where no accumulation of mass or energy is considered. Mass conservation law reduces to:

$$\dot{m}_{in} = \dot{m}_{out} \tag{4.24}$$

However, the mass flow rate and pressure may vary when the bottoming cycle operates in off-design conditions. Thus, the Stodola's cone law [58] was utilized to predict the steam turbine performance during off-design operation. This equations is described as:

$$\dot{m}_{in} = K_t \sqrt{\rho_{in} p_{in} \sqrt{1 - \left(\frac{1}{PR}\right)^2}} \tag{4.25}$$

with the Stodola's coefficient, K_t, given by:

$$K_t = \frac{\dot{m}_{in,nom}}{\sqrt{2\ \rho_{in,nom}\ (p_{in,nom} - p_{out,nom})}} \tag{4.26}$$

and the pressure ratio, PR, being defined as:

$$PR = \frac{p_{in}}{p_{out}} \tag{4.27}$$

where ρ_{in} is the density at the inlet, p_{in} and p_{out} are the pressures at the inlet and the outlet, and the subscript *nom* stands for nominal conditions.

The shaft power, P_m, produced by the expansion of the working fluid along the turbine is obtained by:

$$P_m = \eta_{mech}\ \dot{w}\ (h_{in} - h_{out}) \tag{4.28}$$

where η_{mech} is the mechanical efficiency, h_{in} is the specific enthalpy at the inlet of the turbine, and h_{out} is obtained assuming constant isentropic efficiency η_{iso} from:

$$\eta_{iso} = \frac{h_{in} - h_{out}}{h_{in} - h_{iso}} \tag{4.29}$$

4.4 Condenser model

The selected configuration for the modeled condenser was a shell and tube heat exchanger where the cooling fluid circulates within the tubes while the working fluid flows through the tube bundle in the shell side. Both condensers of the CHP plant were considered to have the same configuration, being only differentiated by the cooling fluid employed. Thus, the dynamic modeling of the flow and heat transfer process of both components was done in the same manner, and only one object was programmed for such purpose.

In many applications, two-phase flows with high quality, i.e. vapor with some droplets, are found at the outlet of the turbine and at the inlet of the condenser. Pure condensation is modeled in these cases as the fluid already enters as a two-phase mixture. However, the utilization of a back-pressure steam turbine implies that the steam entering the condensers is superheated, as the pressure at the outlet of the turbine is relatively high in order to leave some energy to produce the process heat. The energy that the steam contains because of its superheated state is a considerable percentage, between 10% and 25%, of the energy employed for the production of the demanded heat. Thus, this feature must be taken into account in the model of the component.

The modeling of the cooling flow in the tube side was carried out similarly as in the OTSG model in Section 4.2. For the sake of shortness the equations are not written again, but some guidelines are given instead in order to ease the understanding of the developed condenser model. Mass conservation was given by Equation 4.8, whereas Equation 4.12 was utilized to describe the energy balance. Heat transfer from the inner surface of the metal tube to the cooling fluid was modeled by Equation 4.15, where the heat transfer coefficient was calculated by means of the Dittus-Boelter correlation (see Equation 4.17). The thermal dynamics of the metal tubes of the condenser were accounted by the utilization of Equation 4.23 and 4.23. A scheme of the discretization of the condenser tube bundle is shown in Figure 4.2. This tube bundle is formed by many tubes of the same length disposed parallel to each other, however, this configuration was modeled as a continuous single tube of length equivalent to the total length of all the tubes integrating the tube bundle.

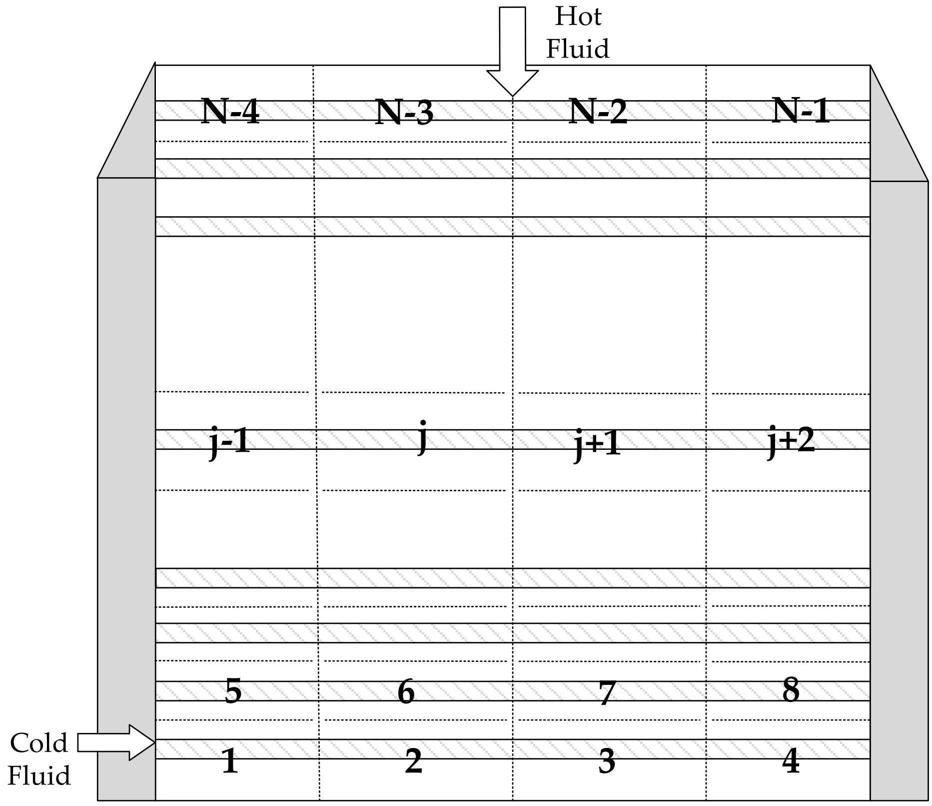


Figure 4.2: Discretization scheme of the condenser tube bundle.

As the working fluid enters the condenser in superheated conditions, the shell was discretized in order to account for the change in temperature and fluid properties more accurately. A moving-boundary model could have also been developed, but a discretized one was chosen in order to have more information about the fluid state within the shell.

Saturated conditions at the outlet of the shell side of both condensers and constant volume of the shell, V_{shell}, were set as constraints of the model. The division of this volume into sub-units was done gradually in order to obtain more detail and accuracy at the outlet of the shell. The reason to such distribution lies in the abrupt density changes that occur when the two-phase is close to become saturated liquid. Thus, in order to have a more robust model with a lot of detail, a larger density of cells was needed at the outlet than at the inlet, where the change of density is gradual. Hence, the volume of each discretization, from top to bottom of the condenser, is defined as:

$$V_{shell,j} = \frac{V_{shell}}{2^j} \tag{4.30}$$

The rapid cell density increase with the number of discretizations at the outlet of the condenser shell may be observed in Figure 4.3:

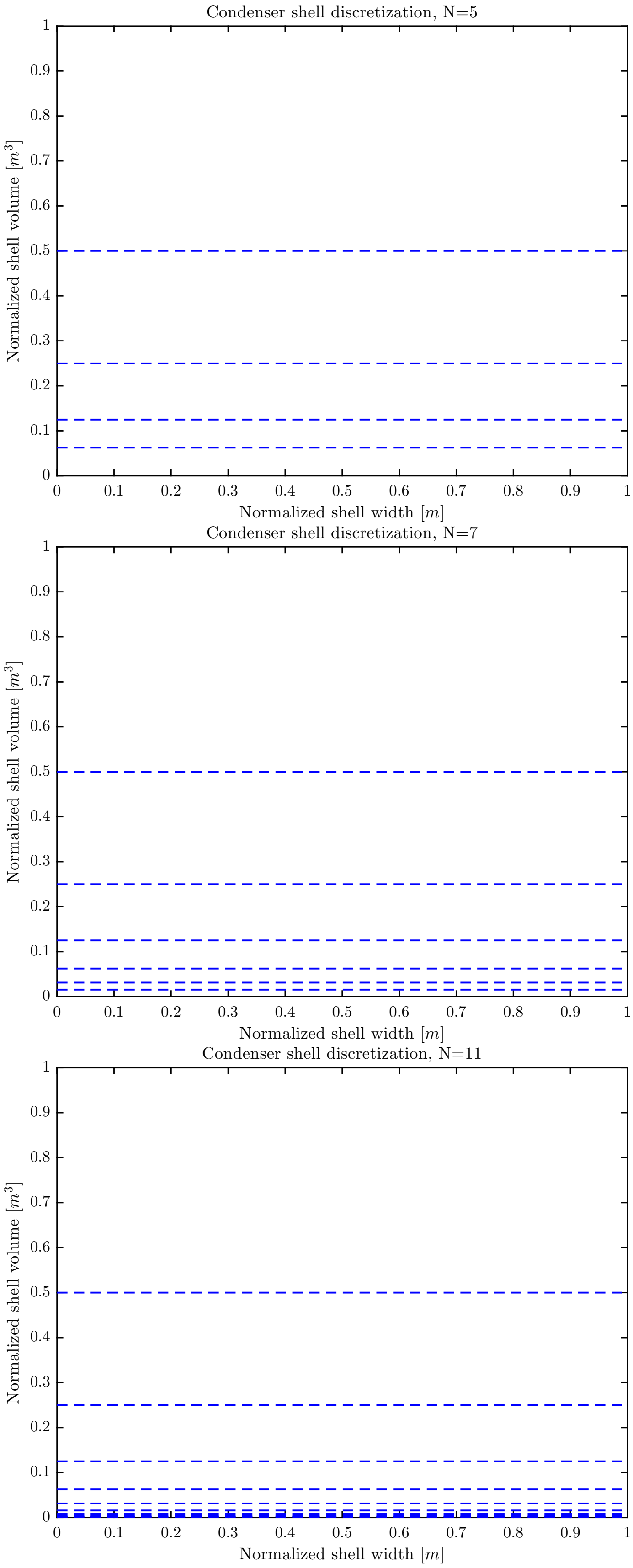


Figure 4.3: Shell gradual discretization for different number of nodes N.

The mass of working fluid in each cell, $M_{shell,j}$ and energy it contains, $E_{shell,j}$, were calculated using the average of the density and the specific enthalpy evaluated at both nodes of cell (see Figure 4.4):

$$M_{shell,j} = V_{shell,j} \frac{\rho_{j+1} + \rho_j}{2} \tag{4.31}$$

$$E_{shell,j} = M_{shell,j} \frac{h_{j+1} + h_j}{2} - p \cdot V_{shell,j} \tag{4.32}$$

where p is the pressure in the shell, which was assumed constant along the entire shell.

Dynamic mass and energy conservation laws were employed to model the flow in the shell side (see Figure 4.4):

$$\frac{dM_{shell,j}}{dt} = \dot{m}_j - \dot{m}_{j+1} \tag{4.33}$$

$$\frac{dE_{shell,j}}{dt} = \dot{m}_j\, h_j - \dot{m}_{j+1}\, h_{j+1} - \dot{Q}_j \tag{4.34}$$

where $\dot{m}_j$ and $\dot{m}_{j+1}$ are the mass flow rates at both nodes of cell j, h_j and h_{j+1} are the specific enthalpies in both nodes of the same cell, and $\dot{Q}_j$ is the heat flow rate leaving the cell evaluated at its center, which is calculated as in Equation 4.14.

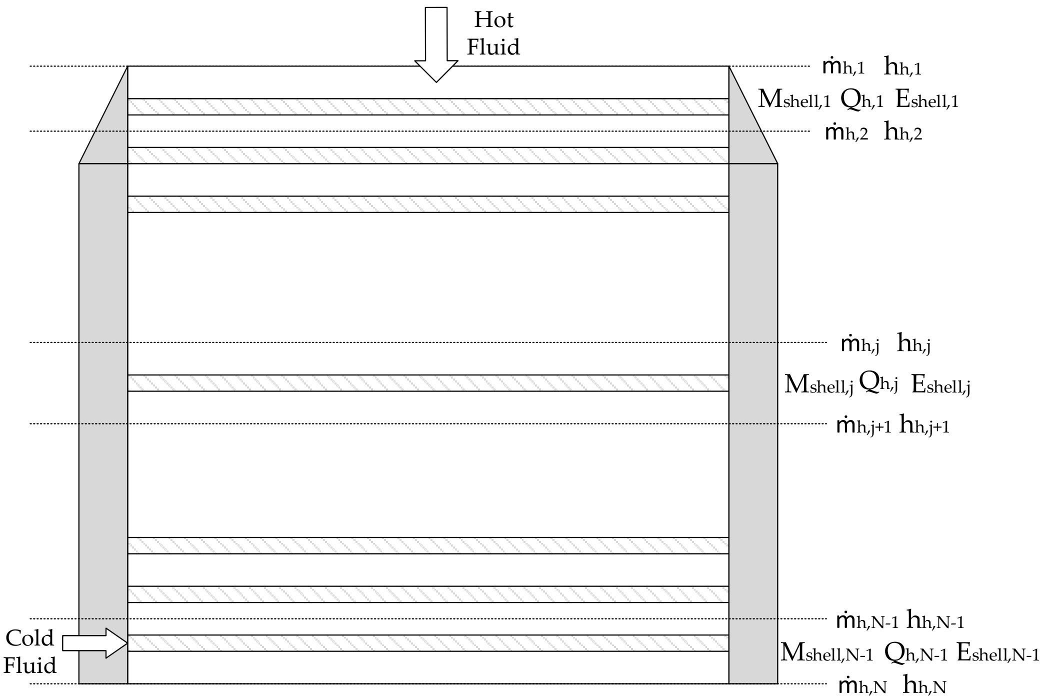


Figure 4.4: Modeling paradigm followed for the condenser shell.

In addition to the shell, the condenser also incorporates a hotwell where the condensed steam is accumulated before it is pumped to the OTSG to start the cycle again. The modeling of this component is fundamental for dynamic simulations as it is in this unit where the accumulation or emptying of water occurs during transient operation of the power plant. Therefore, its effect on the performance of the power generation system is notable. Moreover, the hotwell was also modeled to account for the increase in the outlet pressure due to the static pressure of the water column. Hence, the pressure at the outlet of the hotwell, p_{out}, and the time rate of mass change, $\frac{dM_{hotwell}}{dt}$, are described by:

$$p_{out} = p_{in} - h_{hotwell}\ g\ \bar{\rho}_{hotwell} \tag{4.35}$$

$$\frac{dM_{hotwell}}{dt} = \dot{m}_{in} - \dot{m}_{out} \tag{4.36}$$

where g stands for gravity, $\dot{m}_{in}$ and $\dot{m}_{out}$ are the mass flow rates entering and leaving the hotwell, respectively, $\bar{\rho}_{hotwell}$ is the average density of the water contained in the hotwell, and $h_{hotwell}$ is the water level, which is calculated assuming a constant cross-section of the hotwell, $A_{hotwell}$, by:

$$h_{hotwell} = \frac{V_{hotwell}}{A_{hotwell}} \tag{4.37}$$

The change in the outlet mass flow rate due to the accumulation or emptying of water in the hotwell and the increase or reduction of the static pressure was modeled based on:

$$\dot{m}_{out} = \sqrt{2gh_{hotwell}} \cdot A_{duct} \cdot \bar{\rho}_{hotwell} \tag{4.38}$$

with A_{duct} being the area of the duct leaving the hotwell, and calculated with the same equation and the nominal values of each variable.

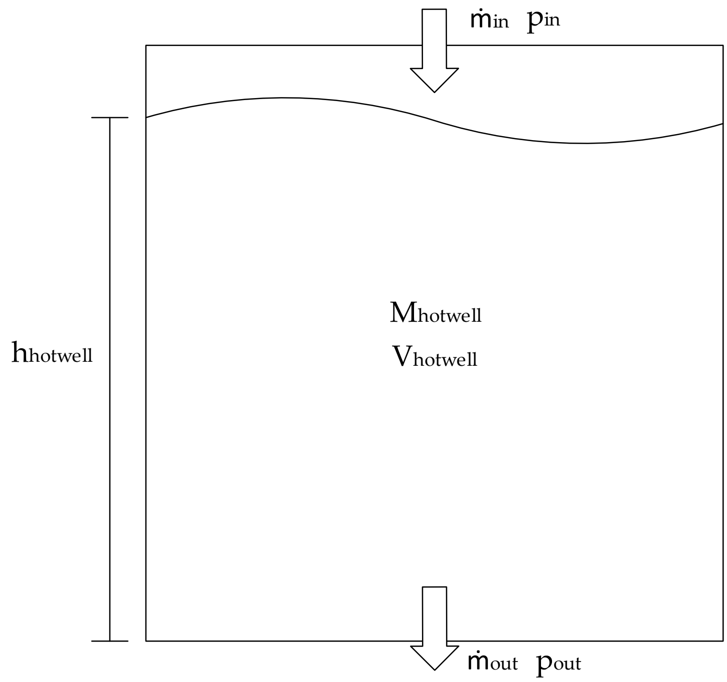


Figure 4.5: Condenser hotwell model.

4.5 Pump model

A variable speed pump model was employed in the dynamic model of the combined heat and power plant. As it occurred with the gas turbines, pumps are able to change their operation point in seconds and, hence, its transient performance time is negligible compared to the dynamics of the heat exchangers. A quasy-static model is employed to simulate the operation of the pump in the dynamic model.

Performance maps of this device were not available and therefore a characteristic equation relating mass flow versus pressure gain was implemented. The data needed to develop this equation, i.e. mass flows and their associated pressure increases along the pump, was obtained from the steady design and off-design simulations carried out with Thermoflex during the design stage of the thermal power plant. However, since this set of information was generated by a pump model that incorporates performance maps relating mass flow, pressure gain and number of revolutions, the characteristic curve created was excessively abrupt for some mass flow intervals. Thus, instead of using the this curve, a regression curve from the set of data was generated in order to smooth the transient behavior of the pump model. The pump characteristic is represented by Equation 4.39:

$$dp = -0.0097\,\dot{m}^3 + 0.3911\,\dot{m}^2 - 3.3895\,\dot{m} + 22.5770 \tag{4.39}$$

where dp is the pressure gain of the working fluid along the pump, and $\dot{m}$ is the circulating mass flow.

This equation is shown in Figure 4.6 together with the steady operation points employed for its development.

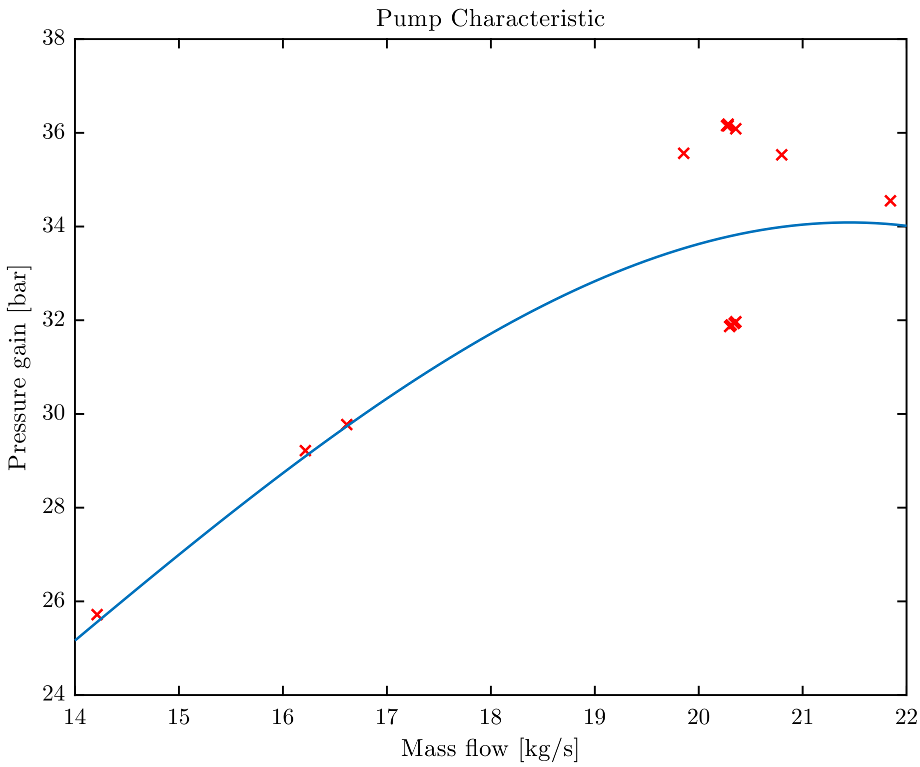


Figure 4.6: Pump characteristic generated as a regression curve.

Chapter 5

Results and Discussion

The results obtained throughout the development of this Master thesis are presented in this chapter. The validation procedure followed to ensure that the developed dynamic models are able to produce reliable results is detailed in Section 5.1. Open-loop simulations are presented in Section 5.2, where the inherent dynamics of the system are shown and discussed in order to observe whether the dynamic model system predicts the transient performance in a reasonable manner. Section 5.3 includes the dynamic simulation results obtained when a preliminary control structure is implemented in the combined heat and power plant model. The main goal of this section is to verify if the system is able to fulfill the pre-established requirements imposed to the power plant rather than assessing if the control strategy applied is the most suitable for the system.

5.1 Validation

Steady-state data was utilized to validate the models integrating the combined heat and power plant proposed in this work. Both design and off-design operating conditions were evaluated in order to have a wider set of validation results and show the model's capability to produce accurate and reliable results. For brevity, the detailed validation results obtained and their error respect to the available data are not shown in this section, but they are presented in Appendix A. A summary of the cases employed during the validation procedure is included in Table A.1.

As discussed in Appendix A, the presented validation results were obtained with a pump characteristic generated by the utilization of the operation points provided by the specialized software Thermoflex with the steady-state design and off-design data, i.e. the red points in Figure 4.6. These pump operation points were included in order to ensure that similar conditions and thermodynamic states were obtained by the dynamic model during its validation. However, this equation representing the pump characteristic is a ninth order equation and, hence, it yields to sharp fluctuations among different operation points. Therefore, it was only utilized during the steady-state validation, where the performing of the pump does not change.

Conditions representing the operation of the pump with the performance characteristic given by Equation 4.39 were implemented in the dynamic simulation in order to ensure a smoother behavior of the entire system. Steady-state off-design validation was not carried out with this performance characteristic as large variations respect to the values covered in Appendix A were not expected. Nevertheless, relevant parameters and variables produced with the smoothed pump being implemented in the model are shown in Table 5.1 in order to prove that the employment of the conditions produced by the smoothed pump characteristic do not introduce large errors to the simulations.

Table 5.1: Validation results during nominal operation.

	Dymola	Thermoflex	Error
$\dot{Q}_{economizer}$ [MW]	10.403	10.343	0.580%
$\dot{Q}_{evaporator}$ [MW]	40.497	40.582	0.209%
$\dot{Q}_{superheater}$ [MW]	9.729	9.652	0.798%
P_m [MW]	8.348	8.601	2.942%
$\dot{Q}_{condenser,1}$ [MW]	52.015	52.050	0.067%
$T_{turb,in}$ [°C]	442.220	442.600	0.086%
$p_{turb,in}$ [bar]	33.809	33.730	0.234%
$p_{turb,out}$ [bar]	5.120	4.896	4.575%

Good agreement between the values generated by the model in Dymola and the steady-state data produced by Thermoflex was obtained, albeit the change in the pump characteristic curve. A larger difference may be observed in the shaft power produced by the steam turbine, P_m, as a consequence of the steam turbine's outlet pressure increase. However, it was considered that these changes respect to the original steady-state data were in an acceptable error range and, hence, the smoothed pump performance equation (Equation 4.39) was employed in the dynamic simulations.

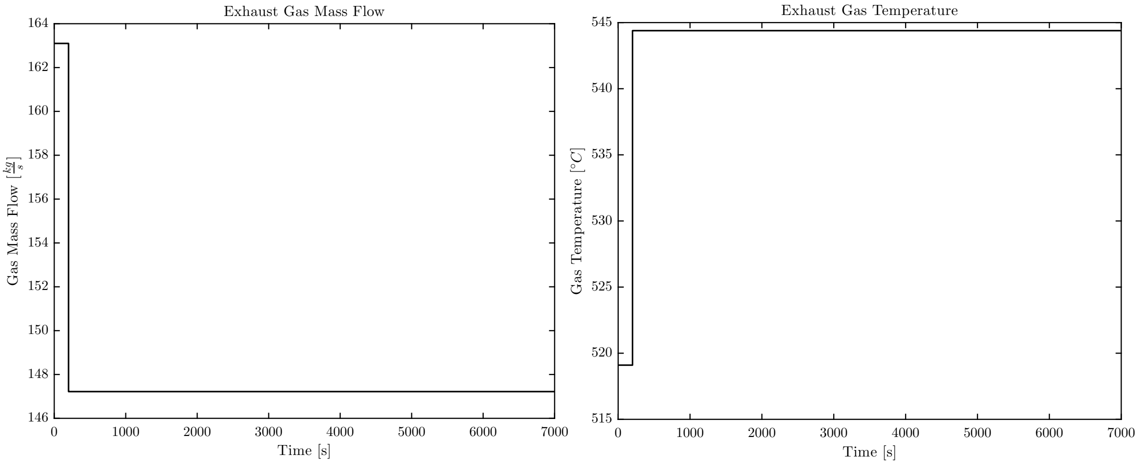
5.2 Open-Loop Dynamic Simulations

The dynamic model of the combined heat and power plant was developed by putting together the component models included in the specialized ThermoPower library and the dynamic models specially programmed in the Modelica language. An open-cycle configuration was selected since the main goal of the simulations included in this work was to obtain preliminary results that verify the correct and reliable performance under transient operation of the individual models developed, so they can be used in the future for the assessment of the dynamic performance of the proposed CHP plant.

An open-loop dynamic simulation, i.e. without any control strategy implemented, was firstly carried out in order to observe the intrinsic transient behavior of the system. Thus, a mass source was utilized to simulate the working fluid mass flow entering the once-through steam generator. The state of the fluid was set equal to the conditions established by the pump characteristic, whereas a constant mass flow rate equal to the nominal value was established. A pressure sink was included at the end of the cycle, i.e. the position corresponding to the inlet of the pump, to impose a boundary condition where the corresponding pressure set by the pump characteristic was utilized.

The ability of the developed combined heat and power plant model to simulate operation under transient conditions was assessed by a dynamic simulation where the load of both gas turbines were decreased from nominal operation point, which corresponds to gas turbine loads of 72.38% and 85.19%, to 60% load. This decrease in the gas turbine power production represents a 20% overall reduction in the power generated by the thermal power plant. Therefore, the capacity of the steam bottoming cycle to reject a disturbance in the gas turbine load is utilized as case scenario.

The load reduction in the gas turbines is simulated as a step reduction in the exhaust gas mass flow stream, which acts as a boundary condition of the heat source in the dynamic model of the power plant. In addition, the variation of the gas turbine load has a strong influence on the exhaust gas temperature that also has to be introduced in the dynamic model. Hence, a step change in the exhaust gases is implemented simultaneously with the mass flow rate variation. The variation of the mass flow rate and temperature of exhaust gas was calculated by means of the quasi-static model explained in Section 4.1. The step-changes introduced in the exhaust gas stream are represented in Figure 5.1. As it may be observed in this figure, an increase in the exhaust gas temperature is associated to a decrease in the mass flow as a result of the changing operating conditions of both gas turbines.



(a) Exhaust gas mass flow rate. (b) Exhaust gas temperature.

Figure 5.1: Dynamic model boundary conditions.

The introduced variations in the exhaust gas stream have encountered effects, as an increase of the temperature leads to a rise in the temperature of the steam produced whereas the reduction of the mass flow decreases the heat transferred to the working fluid in the once-through steam generator. The temperature at the inlet of the steam turbine, i.e. the steam live temperature, is shown in Figure 5.2. The increase in the temperature difference has an initial larger effect when the exhaust gas changes occur, since a rapid increase in the steam live temperature is produced. However, some time after the gas turbine load change, the mass flow rate reduction outbalances the exhaust gas temperature rise and a new steady-state value lower than the initial one is reached by the steam live temperature.

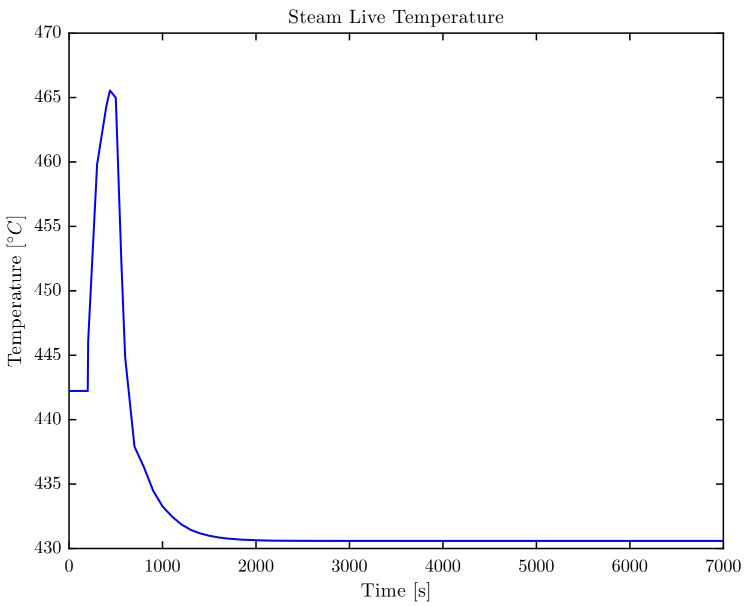


Figure 5.2: Live steam temperature.

In addition to the temperature variation, the pressure at the inlet of the steam turbine also experiences some changes as a consequence of the pressure fluctuations that occur due to the water level variations in the condensers. The pressure at the inlet of the steam turbine is represented in Figure 5.3:

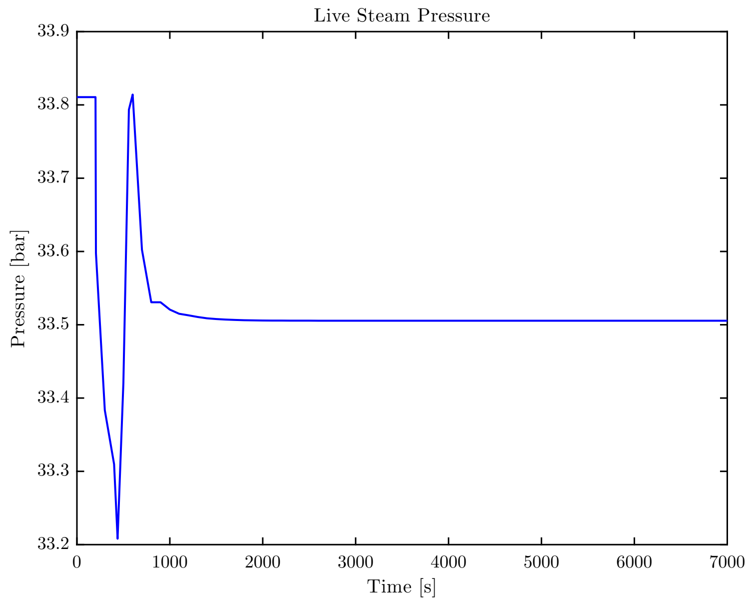


Figure 5.3: Live steam pressure.

The shaft power produced by the steam turbine has a trajectory that result from the combination of the behavior of both live steam temperature and pressure. As it may be observed in Figure 5.4, the shaft power is slightly decreased due the faster reduction in the inlet pressure than the increase in the steam temperature. However, when the inlet pressure increases again and the inlet temperature reaches its peak value, a maximum shaft power generation is achieved. Once that the effects of the decrease in the mass flow rate are felt by the system the shaft power is reduced abruptly until it reaches a new steady-state value.

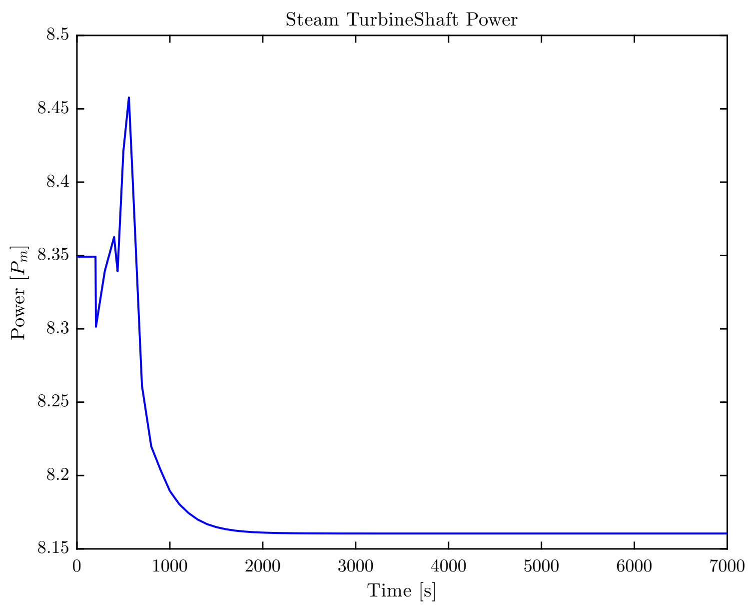


Figure 5.4: Steam turbine shaft power.

The effect of the gas turbine load reduction on the process heat temperature, i.e. the temperature of the pressurized water utilized in the process plant of the offshore facility, is illustrated in Figure 5.5. The requirements established by the plant operator were not met, as a stationary temperature below 150 °C is reached when the power plant is operated in open-loop and a reduction in the gas turbine load is experienced.

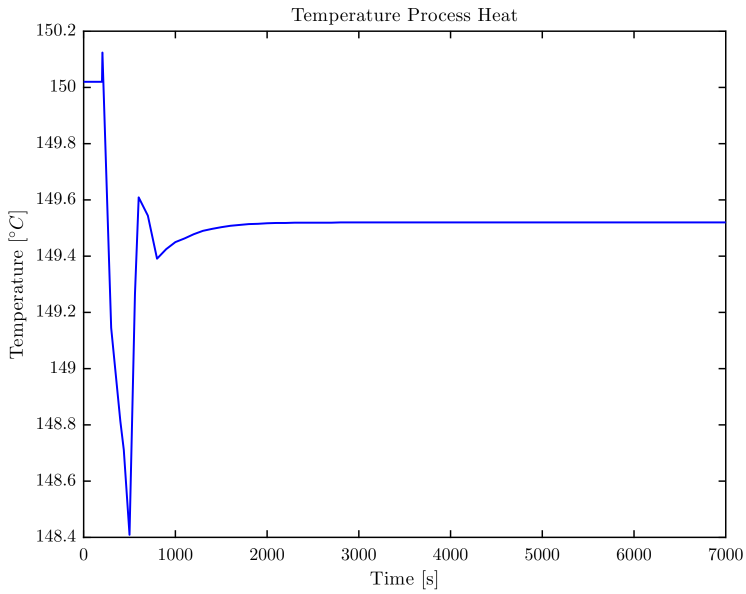


Figure 5.5: Process heat temperature.

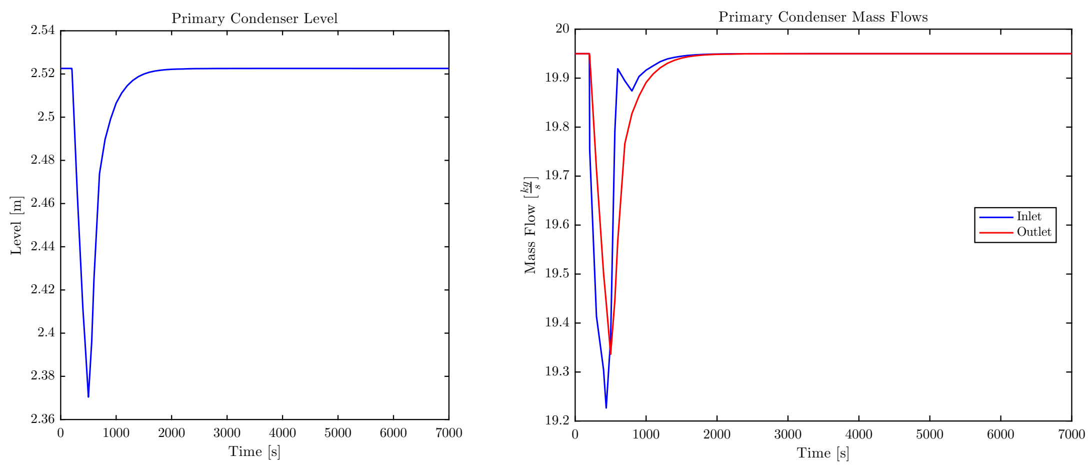
The behavior of the process heat temperature is explained by the variation of the mass flow of working fluid entering and leaving the primary condenser (see Figure 5.6b). As a result of the gas turbine load change and the pressure variations introduced in the system, the mass flow rate at the inlet and outlet of the condenser is reduced and, hence, the temperature of the pressurized water circulating in tubes is also decreased. The mass flow rate returns to its steady nominal value when the fluctuations have been absorbed by the system. However, as a consequence of the reduction of the steam temperature, the same amount of mass flow rate circulating in the primary condenser is not capable of heating the pressurized water at the required temperature. The effect of the modification in the inlet and outlet condenser mass flow rates can also be seen in the water level of its hotwell (see Figure 5.6a). The hotwell acts as buffer, smearing out the fluctuations in the outlet mass flow rate and maintaining it higher than the entering superheated steam. Thus, the difference between the dynamics of both mass flow rates generates a momentary decrease in the water level of the primary condenser hotwell.

5.3 Closed-Loop Dynamic Simulations

A control structure was implemented in the open-cycle model in order to test if the component models were able to operate under the constraints imposed by the control strategy. In addition, these preliminary results provided a first idea of the steam bottoming cycle capability to reject changes in the gas turbine loads.

The temperature of the pressurized water utilized to deliver the process heat in offshore facility is a fundamental parameter in the correct operation of the combined heat and power plant. Thus, it was controlled by regulating the flow of working fluid that circulates in the shell side of the condenser. The manipulated variable was the opening of a valve that regulates the hydraulic resistance of each of the parallel branches and fixes the mass flow rate circulating along them. A PI controller was utilized to carry out such control.

The control of the steam live temperature was also implemented in the open-cycle model as it is a common approach utilized in the operation of thermal power plants. This



(a) Water level in the primary condenser.

(b) Mass flow rates in the primary condenser.

Figure 5.6: Water level and mass flow rate in the primary condenser of the thermal power plant.

methodology provides a more stable operation of the steam turbine as well as it ensures that temperatures exceeding the material limits are not reached. Nevertheless, the latter is not a big issue in this case since the temperatures that were expected were low compared to the temperatures that may be found in other applications of the same technology, e.g. power plants with reboilers. This control of the live steam temperature was implemented in the model by means of a PI controller that manipulates the mass flow generated by the mass source. If a closed-cycle was utilized, the rotation speed of the pump would be the manipulated variable that controls the turbine inlet temperature.

A control system was also included in the dynamic model to maintain constant the water level in the primary condenser and ensure a stable operation of the process heat generation section. The opening of a control valve was utilized as the manipulated variable of a PI controller in the open-cycle model. However, a pump at the outlet of the hotwell should be employed if a closed cycle was considered as this is the normal procedure followed in real power plants.

It is worth mentioning that the tuning parameters that were set in the PI controllers were not optimized, as this task is out of the scope of this work. Thus, values that allowed to reach a stable operation point after the transient performance were implemented without considering if some improvements could be achieved with different tuning parameter values. This is also the motivation to utilize PI controllers instead of PID, which could provide a faster control but need to be carefully calculated to avoid instabilities during the dynamic operation.

The same step changes as in Section 5.2 were implemented in these dynamic simulations in order to have results to compare and check the effect that a control structure may have in the behavior of the combined heat and power plant during transient operation conditions. The step changes implemented may be observed in Figure 5.1.

The live steam temperature during both open-loop performance and with the implemented control structure is represented in Figure 5.7. It can be observed that, although larger fluctuations are produced due to the actuation of the control, the set point is reached after the transient operation of the power plant. The manipulated variable, i.e. the working fluid mass flow, is presented in Figure 5.8. Same patterns are observed in both variables, being the mass flow slightly shifted in time due to the delay existing between the measurement of the control variable and the response of the manipulated variable. Nevertheless, as it could be expected, a reduction in the mass flow of working fluid circulating in the bottoming cycle was produced in order to maintain a constant temperature at the inlet of the steam turbine.

Changes in the mass flow passing through the steam turbine have also effect on its inlet pressure. The sliding operation mode during off-design conditions leads to larger changes in the inlet pressure than those occurring without a control structure, as no modifications in the steam mass flow happen in the latter. This behavior is represented in Figure 5.9.

Shaft power also experiences larger changes when the control strategy is applied than when the thermal power plant is operated in open-loop. As it may be observed in Figure 5.10, the dynamic response of the power produced in the steam turbine has a similar behavior to the mass flow of working fluid, with a rapid initial increase followed by an equal reduction to then reach a steady value smaller than the nominal point. It is worth mentioning that the shaft power produced when a control structure is implemented is larger than in the open-loop case, which proves that the increase in the live steam temperature weights more than the reduction of the inlet pressure and the mass flow rate.

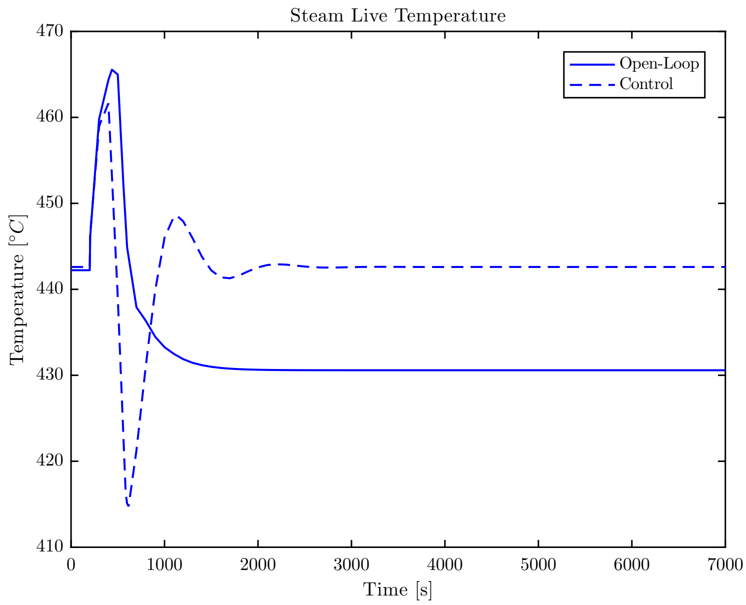


Figure 5.7: Comparison between live steam temperatures in open-loop configuration and with a control structure implemented.

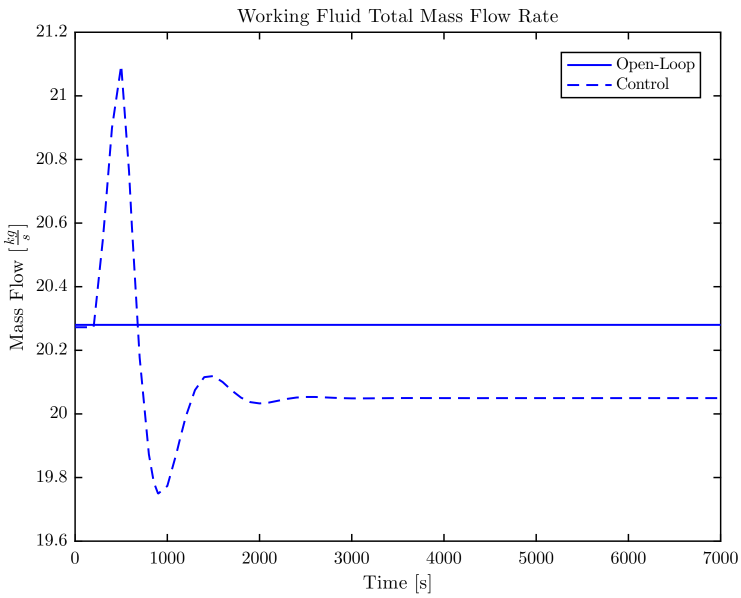


Figure 5.8: Comparison between the working fluid mass flow in open-loop configuration and with a control structure implemented.

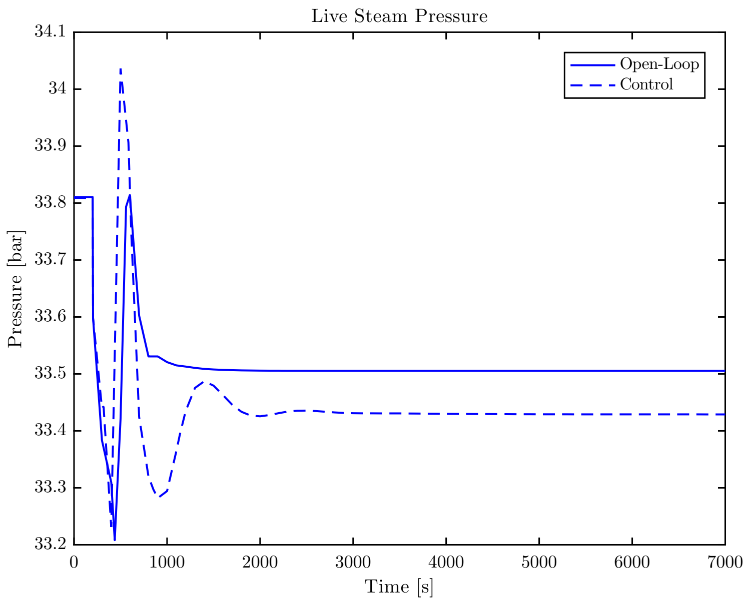


Figure 5.9: Comparison between steam turbine inlet pressures in open-loop configuration and with a control structure implemented.

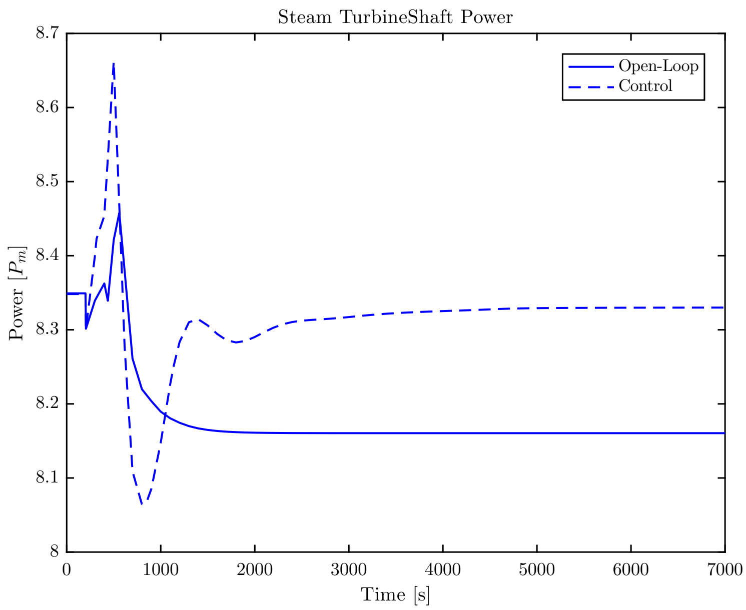


Figure 5.10: Comparison between shaft power generation in open-loop configuration and with a control structure implemented.

The control structure implemented was also able to maintain the temperature of the pressurized water used to produce process heat at the required temperature (see Figure 5.11). Nevertheless, as a consequence of the control strategy, larger times were needed to reach this steady operation point whereas a more fluctuating dynamic behavior was obtained. This slow response is represented by the opening of the control valve utilized to regulate the flow distribution between both branches in the process heat generation section of the combined heat and power plant (see Figure 5.12). However, the optimization of the tuning parameters and the integration of a derivative control structure in the current control structure could notably reduce the time required by the controlled variable to return to its set point.

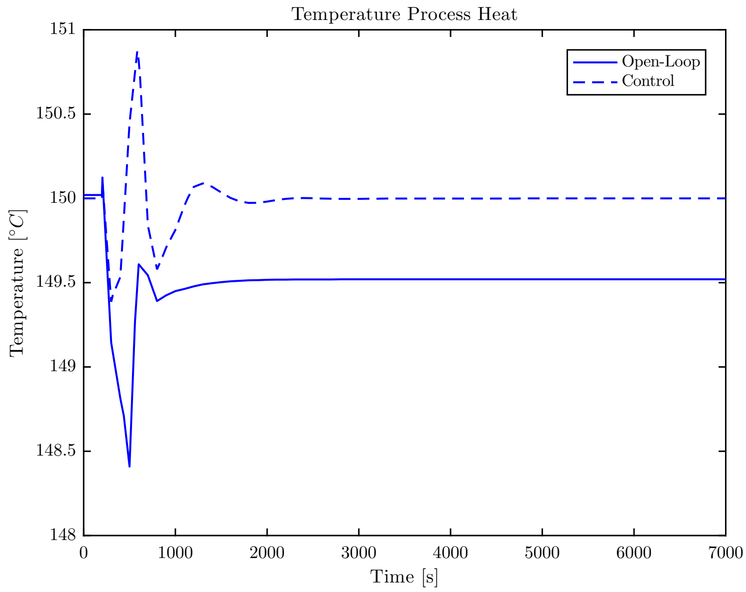


Figure 5.11: Comparison between process heat temperature in open-loop configuration and with a control structure implemented.

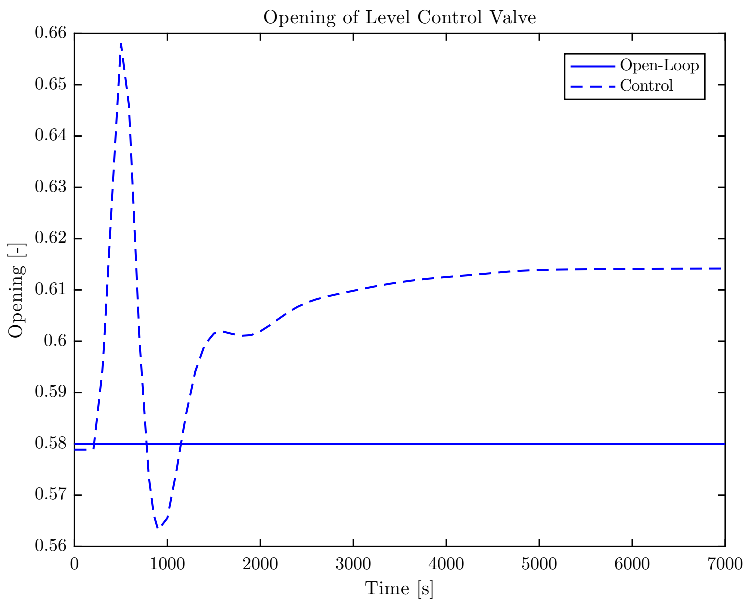
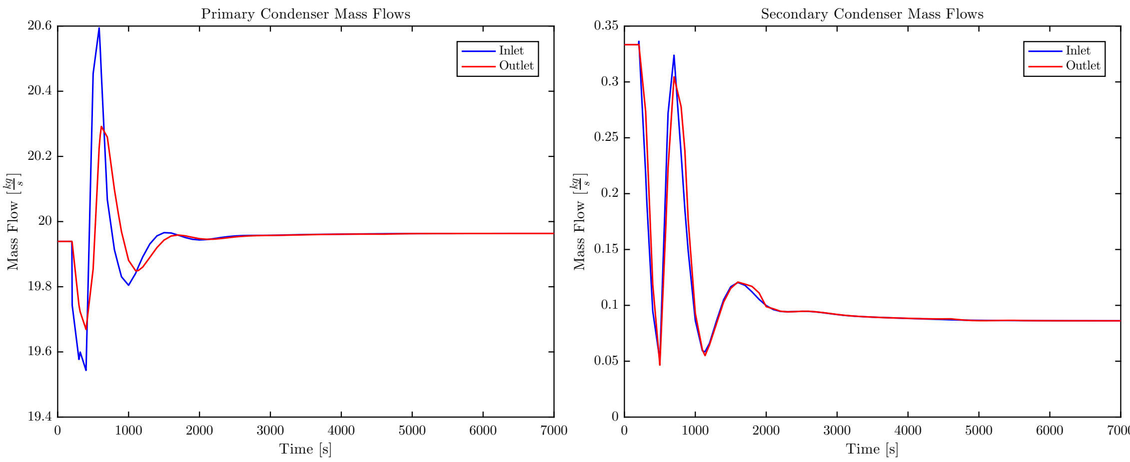


Figure 5.12: Comparison between level control valve opening in open-loop configuration and with a control structure implemented.

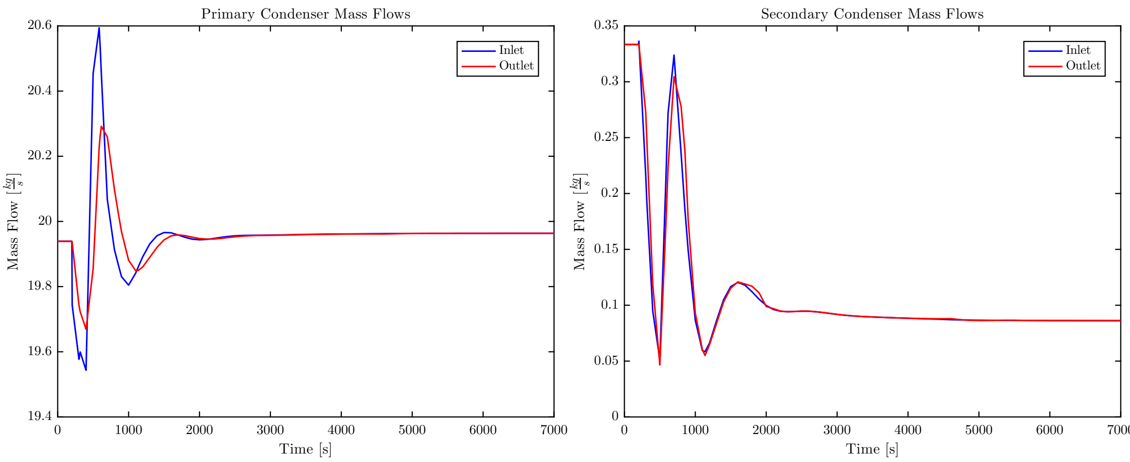
The mass flow rate of working fluid circulating in the primary condenser was the manipulated variable employed to control the temperature of the process heat and, thus, an increase in its value was expected in order to meet this temperature requirement. This behavior is shown in Figure 5.13, where the mass flow in the primary condenser reaches a set point higher than the original while a reduction in the steam flowing through the secondary condenser is considerably reduced. A similar behavior is observed in the water levels of the hotwells (see Figure 5.14). The mass flow increase in primary condenser yields to a rise in the water level as a result of the increase of static pressure that has to occur in order to balance the water entering and leaving the hotwell. On the contrary, the reduction in the mass flow rate flowing trough the secondary condenser yields to a decrease in its water level. If Figure 5.6 and Figure 5.13a and 5.14a are compared, it can be observed that fluctuations in the mass flow and the water level in the condenser are enhanced due to the control strategy implemented to ensure the correct temperature of the pressurized water flowing in the tube side.

The buffering effect of the hotwell can also be identified when a control strategy is implemented. Figure 5.13a shows how the fluctuations in the mass flow leaving the condenser are smoothed respect to those suffered by the flow at the inlet of the component. In addition, it may be observed in Figure 5.13b and 5.14b the larger variations that both mass flow rate and water level experience due to the low water accumulation.



(a) Mass flow rates in the primary condenser. (b) Mass flow rates in the secondary condenser.

Figure 5.13: Mass flow rates at the inlet and outlet of the thermal power plant condensers with a control structure implemented.



(a) Water level in the primary condenser. (b) Water level in the secondary condenser.

Figure 5.14: Water levels in the condensers with a control structure implemented.

Chapter 6

Conclusions and Further Work

6.1 Conclusions

A throughout study of the importance of waste heat recovery applications and the analysis of the dynamic operation of power generation systems was done throughout this work. Modelica was shown to be an ideal modeling and programming language to perform these analysis due to its object-oriented nature, its acausal modeling feature, the availability of specialized libraries in different fields, e.g. ThermoPower library, and the possibility of merging it with software like Dymola, which enhances the main characteristics of Modelica allowing the development of heterogeneous and complex models by means of graphical programming.

The main guidelines for the development of an efficient yet compact combined heat and power plant were detailed, discussing the assumptions that had to be taken in order to achieve a trade-off among weight, volume and performance. The heat recovery steam generator configuration was extensively discussed through a literature study, yielding to the selection of a continuous single pressure once-through steam generator.

The different components integrating the selected design of the power generation system proposed for the case study were dynamically model. Elements included in the ThermoPower library were utilized to this purpose, albeit some components were specially programmed due to the demanding requirements established by the case study and the unavailability of components in the library that fulfilled them. Details of the individual models were presented, pointing out the assumptions followed during its development.

Steady-state validation procedure of the open-cycle power plant model showed the capability of the individual component models to produce reliable and accurate results during both design and off-design performance. Detailed results of this procedure were presented, pointing out the correct functioning of complex components, e.g. the once-through steam generator or the condenser.

Dynamic simulations in open-loop and with a control structure implemented were carried out in order to obtain preliminary results of the performance of the developed models under transient operation. Results proved that the dynamic models were able to predict the dynamics of the power plant system under a change of load in the gas turbines. In addition, it was also shown the ability of the proposed combined heat and power plant to meet the requirements imposed to the thermal power system when a control strategy was implemented, proving the capability of the developed models to be used for a detailed assessment of the transient performance of a power generation system.

6.2 Further Work

Further lines of work to complement what has been done in this Master thesis include:

- Develop a closed-cycle model of the combined heat and power plant that allows a detailed and accurate assessment of its transient performance under many different changes of gas turbine loads and variations in the heat demand, occurring both separately and simultaneously. Start-up and shut down are also operation regimes that could be included in the dynamic study. The utilization of pump characteristic maps would be recommendable in this stage due to the different operation points that may be found during the dynamic operation of this power system.
- Study and implement a suitable control strategy that allows to meet the requirements of the power plant in the most efficient manner while ensuring a reliable transient performance.
- Develop component models with reduced orders, e.g. moving boundary models for the primary heat exchanger and the condensers, in order to compare the accuracy of the results with the discretized models and the simulation time required for each case. This line of work is specially interesting if model predictive control strategies were considered to be implemented.
- Different organic and inorganic working fluids could be considered in the design procedure and the dynamic operation assessment in order to reduce the size of the equipment while maintaining high performances both under steady and transient operation.

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Appendix A

Validation Results

The detailed validation results for the dynamic model developed in this work are included in this appendix. Steady-state design and off-design conditions were simulated in order to validate the model for a wide range of operating points and to ensure that it was able to provide accurate and reliable in different scenarios.

A pump characteristic was developed by means of a regresion equation of the operation point generated by Thermoflex in order to achieve a smooth performance during the dynamic simulations (see Section 4.5). However, a different pump performance characteristic including these operation points was utilized during the validation simulations. This approach was selected in order to ensure that the pressure in critical components, e.g. steam turbine, condensers and pump, were as close as possible to the data provided by the steady simulations carried out with Thermoflex. If the smoothed characteristic pump had been utilized, differences larger than 2 bar would have been introduced in the pressure gain of the pump and, although the overall performance of the cycle had not changed substantially, the validation procedure had not been so rigorous.

The lack of pump characteristics, not only for the pump's head but also for its efficiency and power consumption, affects the fluid conditions, i.e. pressure and temperature, at the outlet of the component as can be seen in the tables below. Nevertheless, the performance and accuracy of the other components of the cycle were considered to be in satisfactory agreement with the validation data.

The power and heat demand conditions, and the gas turbine loads that were selected for the validation procedure are summarized in Table A.1:

Table A.1: Summary of the cases utilized in the validation procedure.

Case	Power Demand	Heat Demand	GT1 Load	GT2 Load
Design	58.00	52.00	72.40%	85.20%
Off-design 1	58.00	42.00	72.40%	85.20%
Off-design 2	69.63	52.00	95.00%	95.00%
Off-design 3	58.00	46.00	72.40%	85.20%

Table A.2: Design Conditions. Power = 58 MW and Heat = 52 MW. GT load: 72.4% and 85.2%.

			Dymola	Thermoflex	Difference	Difference [%]
Economizer	Water	Pressure [bar]	36.404	36.330	0.074	0.20%
		Temperature [°C]	70.584	72.320	-1.736	2.40%
		Mass Flow [kg/s]	20.280	20.280	0.000	0.00%
	Gas	Pressure [bar]	1.015	1.015	0.000	0.00%
		Temperature [°C]	176.805	176.900	-0.095	0.05%
		Mass Flow [kg/s]	163.100	163.100	0.000	0.00%
OTB	Water/Steam	Pressure [bar]	35.685	35.610	0.075	0.21%
		Temperature [°C]	197.462	196.900	0.562	0.29%
		Mass Flow [kg/s]	20.280	20.280	0.000	0.00%
	Gas	Pressure [bar]	1.020	1.020	0.000	0.00%
		Temperature [°C]	240.164	239.300	0.864	0.36%
		Mass Flow [kg/s]	163.100	163.100	0.000	0.00%
Superheater	Steam	Inlet Pressure [bar]	34.905	34.830	0.075	0.22%
		Inlet Temperature [°C]	254.439	256.500	-2.061	0.80%
		Oulet Pressure [bar]	34.495	34.420	0.075	0.22%
		OutletTemperature [°C]	443.572	444.200	-0.628	0.14%
		Mass Flow [kg/s]	20.280	20.280	0.000	0.00%
	Gas	Oulet Pressure [bar]	1.047	1.046	0.001	0.10%
		OutletTemperature [°C]	466.738	467.100	-0.362	0.08%
		Inlet Pressure [bar]	1.051	1.051	0.000	0.00%
		Inlet Temperature [°C]	519.100	519.100	0.000	0.00%
		Mass Flow [kg/s]	163.100	163.100	0.000	0.00%
Steam Turbine		Inlet Pressure [bar]	33.816	33.740	0.076	0.23%
		Inlet Temperature [°C]	442.900	442.600	0.300	0.07%
		Outlet Pressure [bar]	5.103	4.945	0.158	3.20%
		Outlet Temperature [°C]	220.910	220.500	0.410	0.19%
		Mass Flow [kg/s]	20.280	20.280	0.000	0.00%
Heat Exchanger	Steam/Water	Pressure [bar]	5.054	4.896	0.158	3.23%
		Mass Flow [kg/s]	20.059	20.000	0.059	0.30%
		Inlet Temperature [°C]	220.765	219.200	1.565	0.71%
		Outlet Temperature [°C]	70.074	71.680	-1.606	2.24%
	Cooling Water	Pressure [bar]	20.000	20.000	0.000	0.00%
		Mass Flow [kg/s]	153.700	153.700	0.000	0.00%
		Inlet Temperature [°C]	70.011	70.000	0.011	0.02%
		Outlet Temperature [°C]	150.393	149.800	0.593	0.40%
Condenser	Steam/Water	Pressure [bar]	0.314	0.271	0.043	15.87%
		Mass Flow [kg/s]	0.221	0.220	0.001	0.45%
		Inlet Temperature [°C]	210.827	209.200	1.627	0.78%
		Outlet Temperature [°C]	70.164	66.830	3.334	4.99%
	Cooling Water	Pressure [bar]	1.021	1.021	0.000	0.00%
		Mass Flow [kg/s]	23.770	23.770	0.000	0.00%
		Inlet Temperature [°C]	25.021	25.040	-0.019	0.08%
		Outlet Temperature [°C]	30.805	33.150	-2.345	7.07%
Pump		Inlet Pressure [bar]	0.314	0.271	0.043	15.87%
		Inlet Temperature [°C]	70.164	66.830	3.334	4.99%
		Outlet Pressure [bar]	36.404	36.330	0.074	0.20%
		Outlet Temperature [°C]	70.576	72.320	-1.744	2.41%

Table A.3: Off-design Condition 1. Power = 58 MW and Heat = 42 MW. GT load: 72.4% and 85.2%.

			Dymola	Thermoflex	Difference	Difference [%]
Economizer	Water	Pressure [bar]	36.387	36.45	-0.063	0.17%
		Temperature [°C]	70.373	87.35	-16.977	19.44%
		Mass Flow [kg/s]	20.350	20.35	0.000	0.00%
	Gas	Pressure [bar]	1.015	1.015	0.000	0.00%
		Temperature [°C]	176.704	183.3	-6.596	3.60%
		Mass Flow [kg/s]	163.100	163.1	0.000	0.00%
OTB	Water/Steam	Pressure [bar]	35.668	35.73	-0.062	0.17%
		Temperature [°C]	197.385	201.9	-4.515	2.24%
		Mass Flow [kg/s]	20.350	20.35	0.000	0.00%
	Gas	Pressure [bar]	1.020	1.02	0.000	0.00%
		Temperature [°C]	240.127	241.1	-0.973	0.40%
		Mass Flow [kg/s]	163.100	163.1	0.000	0.00%
Superheater	Steam	Inlet Pressure [bar]	34.888	34.93	-0.042	0.12%
		Inlet Temperature [°C]	254.362	256.5	-2.138	0.83%
		Oulet Pressure [bar]	34.478	34.51	-0.032	0.09%
		OutletTemperature [°C]	443.552	443.8	-0.248	0.06%
		Mass Flow [kg/s]	20.350	20.35	0.000	0.00%
	Gas	Oulet Pressure [bar]	1.047	1.047	0.000	0.00%
		OutletTemperature [°C]	466.722	467	-0.278	0.06%
		Inlet Pressure [bar]	1.051	1.051	0.000	0.00%
		Inlet Temperature [°C]	519.100	519.1	0.000	0.00%
		Mass Flow [kg/s]	163.100	163.1	0.000	0.00%
Steam Turbine		Inlet Pressure [bar]	33.799	33.84	-0.041	0.12%
		Inlet Temperature [°C]	442.88	442.3	0.580	0.13%
		Outlet Pressure [bar]	4.995	4.945	0.050	1.01%
		Outlet Temperature [°C]	218.84	219.9	-1.060	0.48%
		Mass Flow [kg/s]	20.350	20.35	0.000	0.00%
Heat Exchanger	Steam/Water	Pressure [bar]	4.942	4.896	0.046	0.94%
		Mass Flow [kg/s]	16.065	16.12	-0.055	0.34%
		Inlet Temperature [°C]	218.683	218.7	-0.017	0.01%
		Outlet Temperature [°C]	70.018	70.35	-0.332	0.47%
	Cooling Water	Pressure [bar]	20.000	20	0.000	0.00%
		Mass Flow [kg/s]	153.700	124.1	29.600	23.85%
		Inlet Temperature [°C]	70.011	70	0.011	0.02%
		Outlet Temperature [°C]	134.499	149.9	-15.401	10.27%
Condenser	Steam/Water	Pressure [bar]	4.547	4.5	0.047	1.04%
		Mass Flow [kg/s]	4.215	4.233	-0.018	0.43%
		Inlet Temperature [°C]	217.862	217.8	0.062	0.03%
		Outlet Temperature [°C]	148.315	147.9	0.415	0.28%
	Cooling Water	Pressure [bar]	1.021	1.819	-0.798	43.87%
		Mass Flow [kg/s]	23.770	255.9	-232.130	90.71%
		Inlet Temperature [°C]	25.021	25	0.021	0.08%
		Outlet Temperature [°C]	100.190	34.41	65.780	191.17%
Pump		Inlet Pressure [bar]	4.547	4.5	0.047	1.04%
		Inlet Temperature [°C]	70.025	86.35	-16.325	18.91%
		Outlet Pressure [bar]	36.387	36.45	-0.063	0.17%
		Outlet Temperature [°C]	70.985	87.35	-16.365	18.73%

Table A.4: Off-Design Condition 2. Power = 69.63 MW and Heat = 52 MW. GT load: 95% and 95%.

			Dymola	Thermoflex	Difference	Difference [%]
Economizer	Water	Pressure [bar]	39.136	39.044	0.092	0.23%
		Temperature [°C]	70.463	80.216	-9.753	12.16%
		Mass Flow [kg/s]	21.840	21.837	0.003	0.01%
	Gas	Pressure [bar]	1.015	1.016	-0.001	0.05%
		Temperature [°C]	182.216	184.967	-2.751	1.49%
		Mass Flow [kg/s]	176.400	-	-	-
OTB	Water/Steam	Pressure [bar]	38.361	38.224	0.137	0.36%
		Temperature [°C]	197.722	201.712	-3.990	1.98%
		Mass Flow [kg/s]	21.840	21.837	0.003	0.01%
	Gas	Pressure [bar]	1.021	-	-	-
		Temperature [°C]	245.424	-	-	-
		Mass Flow [kg/s]	176.400	-	-	-
Superheater	Steam	Inlet Pressure [bar]	37.521	37.374	0.147	0.39%
		Inlet Temperature [°C]	269.247	263.506	5.741	2.18%
		Oulet Pressure [bar]	37.080	36.931	0.149	0.40%
		OutletTemperature [°C]	443.772	444.047	-0.275	0.06%
		Mass Flow [kg/s]	21.840	21.837	0.003	0.01%
	Gas	Oulet Pressure [bar]	1.049	-	-	-
		OutletTemperature [°C]	474.484	-	-	-
		Inlet Pressure [bar]	1.054	1.063	-0.009	0.85%
		Inlet Temperature [°C]	522.400	522.438	-0.038	0.01%
		Mass Flow [kg/s]	176.400	176.362	0.038	0.02%
Steam Turbine		Inlet Pressure [bar]	36.348	36.204	0.144	0.40%
		Inlet Temperature [°C]	443.051	442.503	0.548	0.12%
		Outlet Pressure [bar]	5.344	4.952	0.392	7.92%
		Outlet Temperature [°C]	218.190	213.433	4.757	2.23%
		Mass Flow [kg/s]	21.840	21.837	0.003	0.01%
Heat Exchanger	Steam/Water	Pressure [bar]	5.291	4.499	0.792	17.61%
		Mass Flow [kg/s]	20.280	20.280	0.000	0.00%
		Inlet Temperature [°C]	218.041	212.138	5.903	2.78%
		Outlet Temperature [°C]	70.081	74.303	-4.222	5.68%
	Cooling Water	Pressure [bar]	20.000	-	-	-
		Mass Flow [kg/s]	153.700	153.700	0.000	0.00%
		Inlet Temperature [°C]	70.011	70.004	0.007	0.01%
		Outlet Temperature [°C]	151.052	150.153	0.899	0.60%
Condenser	Steam/Water	Pressure [bar]	4.895	4.499	0.396	8.80%
		Mass Flow [kg/s]	1.560	1.557	0.003	0.19%
		Inlet Temperature [°C]	217.211	-	-	-
		Outlet Temperature [°C]	151.065	-	-	-
	Cooling Water	Pressure [bar]	1.021	-	-	-
		Mass Flow [kg/s]	23.770	41.012	-17.242	42.04%
		Inlet Temperature [°C]	25.021	-	-	-
		Outlet Temperature [°C]	60.427	-	-	-
Pump		Inlet Pressure [bar]	4.895	-	-	-
		Inlet Temperature [°C]	70.089	-	-	-
		Outlet Pressure [bar]	39.136	-	-	-
		Outlet Temperature [°C]	71.122	-	-	-

Table A.5: Off-Design Condition 3. Power = 58 MW and Heat = 46 MW. GT load: 72.4% and 85.2%.

			Dymola	Thermoflex	Difference	Difference [%]
Economizer	Water	Pressure [bar]	36.602	36.40	0.205	0.56%
		Temperature [$°C$]	70.387	81.56	-11.169	13.70%
		Mass Flow [kg/s]	20.320	20.32	-0.000	0.01%
	Gas	Pressure [bar]	1.015	1.02	0.000	0.00%
		Temperature [$°C$]	176.920	180.870	-3.953	2.19%
		Mass Flow [kg/s]	163.100	163.1	0.000	0.00%
OTB	Water/Steam	Pressure [bar]	35.881	35.68	0.202	0.57%
		Temperature [$°C$]	197.578	200.07	-2.491	1.25%
		Mass Flow [kg/s]	20.320	20.32	-0.000	0.01%
	Gas	Pressure [bar]	1.020	-	-	-
		Temperature [$°C$]	240.557	-	-	-
		Mass Flow [kg/s]	163.100	163.10	0.000	0.00%
Superheater	Steam	Inlet Pressure [bar]	35.100	34.89	0.214	0.61%
		Inlet Temperature [$°C$]	263.065	257.37	5.690	2.21%
		Oulet Pressure [bar]	34.689	34.47	0.218	0.63%
		OutletTemperature [$°C$]	448.227	-	-	-
		Mass Flow [kg/s]	20.320	20.32	-0.000	0.01%
	Gas	Oulet Pressure [bar]	1.047	-	-	-
		OutletTemperature [$°C$]	470.446	-	-	-
		Inlet Pressure [bar]	1.051	1.06	-0.005	0.48%
		Inlet Temperature [$°C$]	521.073	521.09	-0.018	0.00%
		Mass Flow [kg/s]	163.100	163.10	0.000	0.00%
Steam Turbine		Inlet Pressure [bar]	34.008	33.80	0.213	0.63%
		Inlet Temperature [$°C$]	447.567	442.89	4.679	1.06%
		Outlet Pressure [bar]	5.150	4.94	0.205	4.15%
		Outlet Temperature [$°C$]	224.860	220.49	4.370	1.98%
		Mass Flow [kg/s]	20.320	20.32	-0.000	0.01%
Heat Exchanger	Steam/Water	Pressure [bar]	5.105	-	-	-
		Mass Flow [kg/s]	17.750	17.75	0.000	0.00%
		Inlet Temperature [$°C$]	224.574	219.21	5.365	2.45%
		Outlet Temperature [$°C$]	70.031	71.00	-0.969	1.37%
	Cooling Water	Pressure [bar]	20.000	-	-	-
		Mass Flow [kg/s]	153.700	135.97	17.735	13.04%
		Inlet Temperature [$°C$]	70.011	70.00	0.007	0.01%
		Outlet Temperature [$°C$]	141.489	-	-	-
Condenser	Steam/Water	Pressure [bar]	4.705	4.50	0.205	4.56%
		Mass Flow [kg/s]	2.570	2.57	-0.000	0.00%
		Inlet Temperature [$°C$]	223.785	-	-	-
		Outlet Temperature [$°C$]	149.582	-	-	-
	Cooling Water	Pressure [bar]	1.021	-	-	-
		Mass Flow [kg/s]	23.770	92.77	-68.999	74.38%
		Inlet Temperature [$°C$]	25.021	-	-	-
		Outlet Temperature [$°C$]	83.845	-	-	-
Pump		Inlet Pressure [bar]	7.705	-	-	-
		Inlet Temperature [$°C$]	70.039	-	-	-
		Outlet Pressure [bar]	36.602	-	-	-
		Outlet Temperature [$°C$]	70.390	-	-	-