Analyses and dynamic modelling of the compressor section in a PCC-process for coal-fired power plants with offshore storage

M.H.L. Ogink
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MASTER OF SCIENCE THESIS

For the degree of Master of Science in Mechanical Engineering at Delft University of Technology

M.H.L. Ogink

June 11, 2015

Faculty of Mechanical, Maritime and Materials Engineering (3mE) · Delft University of Technology
Delft University of Technology
Department of
Process & Energy (P&E)

The undersigned hereby certify that they have read and recommend to the Faculty of Mechanical, Maritime and Materials Engineering (3mE) for acceptance a thesis entitled

ANALYSES AND DYNAMIC MODELLING OF THE COMPRESSOR SECTION IN A PCC-PROCESS FOR COAL-FIRED POWER PLANTS WITH OFFSHORE STORAGE

by

M.H.L. Ogink

in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE MECHANICAL ENGINEERING

Dated: June 11, 2015

Exam committee:

ir. C.M. De Servi (supervisor)    ir. R.C.F. de Kler (supervisor)
Dr.ir. W. de Jong
Dr.ir. C.A. Infante Ferreira

Prof.Dr.ir. B.J. Boersma
Abstract

Carbon dioxide (CO\textsubscript{2}) is one of the main greenhouse gases that contributes to the current climate change. Its increasing share in the Earth atmosphere leads to an increase in global temperature. To prevent the sea level from rising and increased severity of extreme weather, many governments are now trying to reduce their carbon footprint. One third of the global CO\textsubscript{2} emissions are produced by fossil-fuelled power plants. Carbon Capture and Storage (CCS) is considered one of the leading technologies for reducing these emissions.

This thesis is part of investigating the feasibility of the ROAD 2020 project. This project will be one of the world’s first large-scale Monoethanolamine (MEA)-based Post Combustion Capture (PCC) demonstration project with offshore storage. The power plant in this project is designed so that it can rapidly change between full and part load operation. The combination of the resulting fluctuating flow and offshore storage makes compressing the CO\textsubscript{2} a challenging problem. An integrally geared centrifugal compressor train seems the most suited compressor system for a PCC process. However, their operational limitation is their short operating range for compressing CO\textsubscript{2} at high pressures. If not properly controlled, this compressor train can significantly reduce the capture unit efficiency. Moreover, possible flow instabilities like surge, two phase flow and hydrate formation could inflict unrepairable damage to the systems in the PCC process.

This thesis investigates the dynamic performance of the integrally geared centrifugal compressor train for a PCC process retrofitted to a coal-fired power plant. The dynamic performance of the compressor train is evaluated for different process configurations which include different compressor control strategies, addition of a well control valve and a single and double compressor train configuration. The figures of merit used to assess the dynamic performance are: compressor work, impact on stripper pressure, occurrence of surge and choke, occurrence of two phase flow. To evaluate these figures of merit a dynamic model of the integrally geared centrifugal compressor train has been successfully developed. The model also includes the transportation pipeline and offshore storage well. The open source Modelica language is used for the development of the dynamic model.

The simulations performed with this model give more insight in the performance of the compressor train and its impact on the CO\textsubscript{2} capture process, transportation pipeline and
storage well. Recycle valves seem the most suited solution for the compressor train to extend its operating range during minimum load of the power plant. Simulations show that all evaluated process configurations meet the required performance and guarantee a stable filling of the offshore storage well. However, each configuration has a different power consumption, possible process complications and impact on the process capital and operational costs. To propose an optimal process configuration for the ROAD project a further techno-economic investigation is required. Two future research directions can be distinguished to obtain this goal. The first direction should focus on extending the developed dynamic model with a capture unit model. The second direction should focus on the technology gaps that make the integration of MEA-based PCC processes with offshore storage challenging.
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Chapter 1

Introduction

The 5th assessment report of the Intergovernmental Panel on Climate Change (IPCC), shows that CO\textsubscript{2} is one of the main contributors to the current climate change [9]. Its strong positive radiative forcing\textsuperscript{1} leads to an increased global energy accumulation. This energy accumulation will cause global warming, a sea level rise and will increase the frequency and severity of extreme weather [10]. These projections have triggered an international course of action and many governments are now trying to reduce their CO\textsubscript{2} emissions. The main contributor to these CO\textsubscript{2} emissions is the fossil energy sector, which emits roughly two thirds of the global greenhouse-gas emissions [11]. Of these fossil fuels, coal is the most common used for electricity production and remains so at least until 2035, as predicted by the International Energy Agency (IEA) [11].

Carbon Capture and Storage (CCS) is one of the main technologies that could significantly help to reduce the CO\textsubscript{2} emissions from coal-fired power plants [12] [13]. CCS comprehends all the systems necessary for capturing, transportation and storage of CO\textsubscript{2}. This is acknowledged by the IEA and they have concluded that CCS can contribute to 14\% (see Figure 1.1) of the required CO\textsubscript{2} reduction to prevent the average world temperature from reaching critical levels. The IEA has predicted that without CCS the overall cost of reducing CO\textsubscript{2} by 50\% would increase by 70 \% in 2050 [1].

There are three main CCS technologies that could be used for coal-fired power plants:

- Post-Combustion Capture
- Pre-Combustion Capture
- Oxyfuel combustion

A description of each of these technologies can be found in many literature sources i.e. Gibbins et al. [14]. Post Combustion Capture (PCC) has the significant advantage that it can be retrofitted to existing coal-fired power plants without large investments or substantial modifications. This makes it very suitable for retrofitting the large number of conventional power plants existing today.

\textsuperscript{1}In climate science, radiative forcing or climate forcing, is defined as the difference of sunlight absorbed by the Earth and energy radiated back to space
2 Introduction

Figure 1.1: Energy-related CO\textsubscript{2} emissions by technology\textsuperscript{2} [1]

Project ROAD 2020

The work of this thesis is part of a larger research project currently investigated by TNO, namely the ROAD 2020 project [15]. This project studies the design and construction of a PCC demonstration plant at a newly built 1100 MW coal-fired power plant in the port of Rotterdam. When in full operation the PCC plant will capture 1.1 million tonnes of CO\textsubscript{2} each year which will be stored in a depleted gas field 25 km offshore. The objective of this demonstration plant is to show the technical and economic feasibility of a large scale PCC process. When constructed, this project will be one of the first large-scale PCC demonstration plants in the world with offshore storage [16]. With the knowledge created in this project, the road is paved for other energy intensive industries to embrace CCS as suitable transition technology. This will be a major step forward in reducing our carbon footprint until the world makes the full transition to renewable energy technologies.

1.1 Post Combustion Capture

PCC technology captures CO\textsubscript{2} from flue gas generated by the combustion of fossil fuels. A PCC process retrofitted to coal-fired power plants can be divided into four separate systems as shown in Figure 1.2. CO\textsubscript{2} rich flue gas is produced in the power plant and fed to the capture unit. Here the CO\textsubscript{2} is absorbed from the flue gas and later stripped in the stripper. Almost pure CO\textsubscript{2} leaves the capture unit and is compressed in the compression section. The compressed CO\textsubscript{2} is transported through a pipeline and eventually stored in a storage well. Every system has its own dynamic behaviour which become interlinked when all systems are connected. During transient operation of the power plant, this combined dynamic behaviour,\textsuperscript{2}Percentages represent the share of cumulative emissions reduced until 2050 to prevent critical temperature levels. Percentages in brackets represent the share of emissions reduced in the year 2050.

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will have dramatic consequences for the different systems if not taken into account in the early design phase.

![Schematic of a post combustion capture process](image)

**Figure 1.2:** Schematic of a post combustion capture process

**Transient operation**

As a result of upcoming political conflicts, various renewable energy sources and exploitation of shale gas, the conventional energy market is rapidly changing. This future market requires flexible systems that can react to rapid variations in energy demand and supply. Moreover, because renewable sources like solar energy have their peak power production during the day, it is expected that coal-fired power plants will be in full operation during the night and part load operation during the day. Until an economical efficient way of storing huge amounts of energy is developed, coal-fired power plants will keep playing a central roll during the transition to renewable energy [17].

The coal-fired power plant in the ROAD project is designed for rapid changes in power supply. This ramping up and down of the power plant has a direct effect on the amount of CO\(_2\) captured in the capture unit, which could range between 40-100 % of the nominal mass-flow. If all systems within the PCC process are not optimised and equipped to cope with these load changes, it will lead to difficulties for the operators as well as a decrease in energy efficiency and reliability of the whole PCC process.

**1.2 Thesis scope**

There is little knowledge about the dynamic behaviour of transient operating PCC processes. A recent review is given by Bui et al. [18]. A number of studies investigate the dynamic behaviour of a power plant integrated with a capture unit [19] [20], but, at the time of this research no dynamic simulation was performed of a whole PCC process including a capture unit, compression train, transportation pipeline and offshore storage well.

To guarantee safe and efficient operation of the PCC process during transient operation in the ROAD project, more research is needed before construction of can be started. For this
reason, in the framework of the ROAD project, a validated dynamic model of the capture unit was made by Van De Haar et al. [21]. In addition, a steady state analysis of the pipeline and storage well was performed by TNO [22]. Despite these studies, still little is known about the impact of the compression section on the PCC process during transient operation. The focus of this work is therefore on analysing the dynamics of the compressor section and its impact on the capture unit, transportation pipeline and storage well.

### 1.2.1 Compressing CO₂

In the compression section the CO₂ is compressed to a desired transportation pressure. This pressure is case specific and determined by the method of transportation and the varying well pressure. In the ROAD project existing pipelines are used, formerly utilized for the transportation of natural gas. Cost efficient transportation with pipelines is achieved when the CO₂ pressure exceeds its critical point [23]. At this pressure, the CO₂ enters a dense phase where it has a density similar to that of a fluid while maintaining a viscosity of that of a gas. In this phase high mass-flows can be achieved, minimizing frictional forces and at the same time reducing the required compressor energy.

![Integrally geared centrifugal compressor train](www.siemens.com)

**Figure 1.3:** Integrally geared centrifugal compressor train, source: www.siemens.com

![Characteristic map of a high pressure centrifugal compressor operating with CO₂](TNO)

**Figure 1.4:** Characteristic map of a high pressure centrifugal compressor operating with CO₂ obtained from a compressor manufacturer. On the y-axis the normalized pressure ratio against the normalized mass-flow on the x-axis. [TNO]

High pressures can be achieved with different types of compressors namely reciprocating, centrifugal or axial compressors. Without exploiting possible heat integration opportunities, the compressor process will consume 6-12 % of the nominal power generated by power plant [24]. This makes it a large energy consuming system within the PCC process. The choice for a compressor type is therefore based on the three most influential variables that influence the total efficiency of the compression system: pressure ratio, compressor efficiency and inlet temperature. A number of studies deal with this subject and give more than 15 possible compressor strategies for compressing CO₂ to high pressures [25] [26]. The most efficient configuration seems to be an integrally geared centrifugal compressor train (see Figure 1.3). With respect to the other compressor types, this solution results in less capital and operational cost as well as more flexibility for optimizing the compressor design during operation [27].

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Despite the favourable aspects of the integrally geared centrifugal compressor train, its main operational drawback is the inability to cope with large and sudden fluctuations in mass-flow. This is especially a challenging problem for the high pressure compressor stages of the compression process. Due to the high molecular weight, low critical point and high compressibility of CO\textsubscript{2}, the high pressure centrifugal compressors operate with steep speed-lines (see Figure 1.4). These speed-lines represent the working range of a centrifugal compressor operating at constant speed (see Section 2.3). The combination of steep speed lines and the fast variation in mass-flow during transient operation, create the conditions for flow instabilities like surge, rotating stall and choke. While rotating stall can occur at centrifugal compressors, it is argued to have little effect on the compressor performance [28] [29]. In contrary to rotating stall, surge will result in large mechanical and thermal stresses. Moreover, a compressor failure could cause a sudden pressure rise at the inlet of the compressor, causing unreparable damage to the capture unit. Without adequate control of the compressor train, rapid flow variations greatly reduce the efficiency, safety and operating range of the PCC process.

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<td>Low</td>
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Different control strategies are possible to prevent surge and choke. The five main strategies for integrally geared centrifugal compressor trains are listed in Table 1.1. Each of these control strategies has a different impact on the PCC process, level of system complexity as well as a different operating range in which they can successfully manage mass-flow fluctuations. Which of these options is best suited for controlling a compressor train is strongly dependent on the overall compression process and its required inlet and outlet conditions.

**Compressor inlet conditions**

The boundary conditions at the inlet of the compression process are set by the stripper of the capture unit. In the ROAD project this stripper is controlled in order to maintain a constant temperature and pressure to ensure favourable thermodynamic conditions for the stripping of CO\textsubscript{2}. A small increase in pressure will have a large negative effect on the efficiency of the capture unit. To maintain an optimal efficiency, the compressor train and its control strategy have to be designed such that the compression process has minimal impact on the stripper pressure.
Compressor outlet conditions

Since the pressure in the stripper cannot be varied, the pressure ratio across the compression process can only be altered by changing the outlet pressure. TNO performed a steady state analysis of the pipeline and storage well to determine the operating conditions required for safe and efficient transportation of the CO$_2$. To cope with process complications like the formation of hydrates and two phase flow, two design solutions were proposed. In Case A, the pressure at outlet of the compression process is a function of the reservoir pressure and the pressure drop caused by the frictional forces in the pipeline and well. In Case B, a well control valve is added which keeps the pipeline at constant pressure (see Figure 1.5). Both cases allow different variations of the pressure ratio and therefore may require a different control strategy for the compressor train.

![Figure 1.5: Schematic of the well control valve (Case B)](image)

1.2.2 Previous research efforts

A number of studies include a compressor section in their study for analysing a PCC process during transient operation. Ziaii et al. [30] made a dynamic model of the capture unit including a compression section, but this model did not take into account the limited operating range of the high pressure CO$_2$ compressors. An investigation for an adequate control strategy is required to increase the operating range of the high pressure compressors and to prevent flow instabilities like surge and choke. Roeder et al. [26] used a steady state model and concluded that by using multiple operating compressor trains in parallel, the efficiency of the PCC process could be increased. Altogether, these studies did not include the dynamic behaviour of the transportation pipeline and storage well. Moreover, they analysed different compressor outlet conditions than those required in the ROAD project to prevent two phase flow and hydrate formation in the pipeline and well.

In general it can be concluded that there is a lack of knowledge about the dynamic performance of the compression section for PCC processes with offshore storage, especially about the impact the latter will have on the other systems during transient operation of the power plant. For the ROAD project it is interesting to know if the dynamic performance of such a compression train is affected by the different well control solutions. Furthermore, it would
be interesting to investigate if a double compressor train configuration would improve the efficiency over a single compressor configuration while complying with the process constraints.

1.3 Research objectives

The objective of this thesis is to investigate the dynamic performance of an integrally geared centrifugal compressor train for a PCC process retrofitted to a coal-fired power plant. The dynamic performance of the compressor train is evaluated for different test cases in order to analyse different control strategies, and to verify the interactions of the system in a single and double compressor train configuration with the pipeline and the stripper. Based on these evaluations a configuration for the compressor section of the ROAD project power plant is proposed. The figures of merit used to assess the dynamic performance are: compressor work, impact on stripper pressure, occurrence of surge and choke, occurrence of two phase flow. The transient operation of the power plant, taken as a reference load profile, consists of a gradual reduction of the load from nominal to 40% minimum load conditions and vice versa. The following objectives are formulated:

1. Identify and implement a control strategy for the compressor train
2. Analyse the dynamic performance of the system with and without a well control valve
3. Analyse the dynamic performance of a single and double compressor train configuration
4. Analyse the impact of the compression process on the stripper pressure
5. Select the best suited process configuration for the PCC system of the ROAD project

This thesis work could form the basis for further research aiming at the optimal design of the compression process for the ROAD project. The developed dynamic models can also be used in similar PCC projects to support the definition of the compressor train configurations and their control strategy. Furthermore, the dynamic models can be a valuable tool for engineering companies and compressor manufacturers, to evaluate the dynamic performance of the system and include eventual extra requirements into the initial design phase. This will avoid costly rectifications in a later stage of the construction phase.

1.4 Approach

To achieve the objectives previously mentioned, a dynamic model of the integrally geared compressor train has been developed. The model includes the transportation pipeline and a storage well. The open source Modelica language [31] is used to develop the dynamic models. Modelica is a fully a-causal and equation-based modelling language which supports an object-oriented approach. This facilitates the re-use and further development of the dynamic models in future research. Moreover, this approach enables the use of component models from already existing Modelica libraries notably the ThermalPower package. This package was specially developed for modelling energy conversion systems and thermal power plants. It has been
developed by the Politecnico di Milano and Delft University of Technology. The commercial software Dymola [32] is used as simulation environment to solve the set of equations defined in Modelica.

To developed the models in this thesis, components from the mentioned library were used or partly modified. When there were no suitable component available, new components have been developed. To keep the index of the dynamic model equal to one, a modelling approach based on bilaterally coupled variables is used [33]. To verify the correct implementation of the dynamic model, a steady state model is developed with ASPEN [34]. This commercial software is widely used in the industry for modelling of chemical processes.

1.5 Thesis outline

Chapter 2 gives a theoretical background on the boundary conditions of the compression process. In addition the characteristics, control strategy and the EoS for describing the process conditions of the integrally geared centrifugal compressor train are discussed.

Chapter 3 describes the development of the dynamic and steady state models.

Chapter 4 presents the verification of the dynamic model.

Chapter 5 presents the simulation results obtained with the dynamic model.

Chapter 6 discusses the simulation results with respect to the figures of merit.

Chapter 7 summarizes the conclusions of this work and the recommendations for future research.
Chapter 2

Theoretical background

In order to define the design and control strategy of a compressor train, first of all it is fundamental to determine the boundary conditions of the system. This chapter discusses such boundary conditions in Section 2.1, while the compressor design is treated in Section 2.2. The used compressor train control strategies are presented in Section 2.3. Finally, the chapter concludes with describing the EoS used to accurately predict the properties of the CO₂ stream (see Section 2.4. The theory presented here forms the basis for the development of the dynamic model in Chapter 3.

2.1 Compressor boundary conditions

In this case study the boundary conditions at inlet of the compressor train are those imposed by the capture unit (see Section 2.1.1). While on the outlet, the conditions has to be such that a safe operation and stable filling of the storage well is guaranteed. These specific conditions can be met with two different control strategies, which differ in the way they control the CO₂ pressure in the transpiration pipeline and storage well. This is discussed in Section 2.1.2.

2.1.1 Capture unit

There are three main technologies available for extracting CO₂ form the flue gas in Post Combustion Capture (PCC) processes: adsorption, absorption and membrane separation. Currently, absorption is the most promising method for PCC retrofitted to high capacity power plants. In a chemical absorption process the CO₂ reacts and is captured from the flue gas with a solvent. In the ROAD project the amine-based Monoethanolamine (MEA) is used as solvent. MEA has a high reaction rate with CO₂ and can be produced with low production costs. This combination makes it an effective solvent for a PCC process.

The layout of the capture unit is shown in Figure 2.1. The CO₂ is captured in the absorber, and separated in the stripper where the solvent is regenerated. The regeneration process
Theoretical background

2.1 Theoretical background

2.1.1 Standard process configuration for CO₂ absorption and desorption from Flue Gas Desulfurization unit to pipeline [2]

Figure 2.1: Standard process configuration for CO₂ absorption and desorption from Flue Gas Desulfurization unit to pipeline [2]

requires thermal energy that is added to the system with a reboiler. To ensure the maximum performance of the capture unit, stable operation conditions are crucial. The stripper is controlled in order to maintain a constant temperature of 120°C. At temperature levels exceeding 120°C, the MEA solvent will start to thermally decompose in products that will react with other flue gas components. The resulting products are highly corrosive and may cause a further decomposition of the solvent. This will lead to a drastic decrease in efficiency of the capture unit. At 120°C the optimal stripper pressure is 1.8 bar. When developing the compressor train, it should therefore be designed to minimize pressure fluctuations in the stripper.

2.1.2 Pipeline and Storage well

TNO performed a steady state analysis of the pipeline and storage well of the ROAD project in order to determine the operating conditions at the outlet of the compressor section. In this study two design solutions for controlling the pressure in the pipeline and well head where proposed to prevent process complications. To understand these complications an explanation of the governing forces in the storage well is given in the next paragraph.

Governing forces in the well

When the CO₂ arrives at the well platform (or well head), 25 km offshore, it is transported by means of a pipeline 3 km into the ground and pushed into the well reservoir (see Figure 2.2). From this point two dominant forces are at play: frictional forces and gravitational forces. The frictional forces depend on the velocity and viscosity of the fluid, the roughness of the pipelines, the tubing geometry and the porosity of the rock. All these forces determine the minimal pressure required at the inlet of the pipeline to ensure a constant flow into the well. While the reservoir at the bottom of the well is filled, the pressure starts to increase. The
minimal required pressure at the inlet of the pipeline for maintaining stable filling is shown in Figure 2.3 for different reservoir pressures. It is expected that the maximum reservoir pressure of 300 bar will be reached after 8 to 16 years, depending on the utilization of the power plant. A decrease in fluid velocity reduces the frictional forces in the well and therefore requires a lower inlet pressure. This is also shown in Figure 2.3 where the minimum required pressure at different mass-flows is given in percentages of the nominal flow (47 kg/s) corresponding to the full and minimum load of the power plant.

![Figure 2.2: Schematic of the well layout](image)

When the pressure in the reservoir exceeds 100 bar gravity becomes the dominant force. At this pressure the CO$_2$ is above its critical point were its density increases considerably. This higher density increases the downward gravitational force, creating a large static head from top to bottom of the well. Moreover, a large increase in density also decreases the fluid velocities hence the frictional forces. With low frictional forces and a large static head, high reservoir pressures can be achieved with relative low pressures at the well head.

**Process complications**

Two process complications can affect the operational safety of the pipeline and storage well. The first complication occurs in the pipeline. The pipeline runs for 24 km on the seabed where the average temperature of the sea is 8°C. Despite good insulation, the heat loss through the pipeline wall will cause a significant temperature drop. This temperature drop could lead to the condensation of CO$_2$ when the pressures are below its critical point. The resulting two phase-flow could form a plug flow in the pipeline which will severely damage process equipment. A multi phase design is possible but will require extra investments in process
equipment, which will significantly increase the project costs. The second complication only occurs in the beginning of the well lifetime at the point where the CO$_2$ expands through a small tubing, down at the bottom of the well, into the reservoir. When the reservoir pressure is low, this expansion results in a significant decrease in temperature. When temperatures drop below 15$^\circ$C degrees, formation of hydrates impose a risk of clogging the reservoir.

Well control method

To avoid process complications, TNO investigated two different design solutions which differ in the way they control the pipeline pressure. For both designs, TNO determined the minimum required temperatures and pressures at the inlet of the pipeline to prevent process complications. The design of the compressor train should be such that it is always able to meet these requirements during full and part load conditions of the power plant. In the first design case (Case A), the required pipeline inlet pressure is a direct function of the forces acting in the well. The minimum required pressure and temperature at the pipeline inlet for this case are shown in Figure 2.4a and 2.4c. It can be observed that the compressor train should be able to manage large pressure variations between full and minimum load operation.
Moreover, during the well lifetime the pressure significantly increases. To prevent these large variations and reduce the pressure difference between beginning and end of well exploitation, a second design case is analysed (Case B).

In Case B, a control valve is placed between the pipeline and well head (see Figure 1.5). This valve is set to regulate the pipeline pressure at 85 bar. This pressure is higher than the critical point of CO\textsubscript{2} at which it no longer forms a two phase flow. This allows for a significant decrease of the temperature in the pipeline, hence increasing the density and the static head in the well. In this case, less compression energy is required to achieve high pressures in the reservoir. The minimal required inlet pressures and temperatures for this case are shown in Figure 2.4b and 2.4d. At the beginning of the well lifetime a slightly higher pressure and temperature is required with respect to the well regular conditions in order to prevent hydrate formation. Whereas, approaching the end of the well lifetime, the control valve is fully opened and the pressure becomes a function of the reservoir pressure.

Both well control strategies require a different layout and control strategy for the compressor train. Furthermore, it is unknown how the control valve and the compressor train interact during transient operation of the power plant. To investigate this interaction and decide which design would require less compression energy, both well control strategies have been considered in this thesis.

## 2.2 Compressor design

The design of the compressor train is determined in this section. Notably, Section 2.2.1 deals with the selection of the compressor system while Section 2.2.2 deals with the layout of the designed compressor train.

### 2.2.1 Compressor system

To minimize the efficiency penalty of a PCC process, Witkowski et al. [25] investigated the power requirements of different compression strategies. They concluded that a shock wave compressor[35] has the most potential of minimizing the energy penalty of the compression section in a PCC process. Because shock wave compressors are not a fully developed technology yet, this option is not considered in this thesis. The second best solution proposed is an integrally geared centrifugal compressor where the CO\textsubscript{2} is liquefied at medium pressures to -25\degree C. However when pipelines are used for transportation like in the ROAD project, economical cooling can only be achieved at Arctic latitudes. An integrally geared centrifugal compressor which compresses the CO\textsubscript{2} stream to high pressures in 6-8 stages would be the third best strategy. This was also concluded by a compressor manufacturer in Ref. [3]. This strategy is used as basis for the compressor train layout in the next section.

### 2.2.2 Compressor train layout

An integrally geared centrifugal compressor train consist of several centrifugal impellers (stages), which are connected with gears to one main shaft. Between two consecutive compression stages, the gas stream can be treated to meet the desired process conditions required
for the next stage. The shaft can be driven by an electric motor or steam expander. The efficiency of the compressor train depends on the efficiency of each compression stage, which in turn is a function of the pressure ratio, inlet temperature, gas properties, working point and impeller efficiency. Finding an optimal layout and process conditions is therefore a complex optimisation problem and is beyond the scope of this research.

In Ref [3], a similar compression process was investigated and illustrated three optimal compression paths (see Figure 2.5). The path labelled with C is considered in this work for the design of the compressor train configuration. A process schematic is shown in Figure 2.6. It shows the main components of the compressor train. The recycle valves are part of the anti-surge system and are discussed in Section 2.3.1. Inter-coolers are used to cool the gas at the outlet of a compression stage, while the knock-out drums and dehydrator are used for removing stream impurities. This equipment is discussed in the next paragraph.

**Removal of impurities from the CO\textsubscript{2} stream**

The CO\textsubscript{2} stream from the stripper contains a considerable amount of water as well as other flue gas components like N\textsubscript{2}, O\textsubscript{2} and Ar in trace quantities (see Table 2.1). These impurities will damage the impeller blades or create dangerous operating conditions downstream of the compressor train as reported by Visser et al. [36].
Based on the work of Visser et al. it is concluded that the concentrations of all impurities, except that of H$_2$O, are low enough that their effect on the compressor performance is negligible. H$_2$O however, is well above the 500 ppm specified for safe operation. This amount of water in the stream could condense in the pipeline and form hydrates in the well. When CO$_2$ dissolves in the condensed water, carbonic-acid is formed which corrodes the metal parts of the compressor and the pipeline.

A cheap way to remove the H$_2$O from a gas flow is to use a knock-out drum (see Section 3.1.5). However, to obtain the high purity required, knock-out drums will not be sufficient. For the last purification step, dehydrators are used (see Section 3.1.5). To protect the impeller blades from corrosion and the impact of large water droplets, a knock-out drum is placed before each compressor stage upstream of the dehydrator. The optimal place in the compressor train for the dehydration unit is that where the stream pressure is around 30 bar [37].
2.3 Compressor control

Compressor control aims at maintaining a compressor in its stable operating range. In a compressor performance map this operating range is bounded by two lines, the surge line and the capacity limit (see figure 2.7).

![Figure 2.7: Typical centrifugal compressor map. On the y-axes the pressure rise is plotted against the mass-flow on the x-axes. [4]](image)

The latter represents the maximal operating region of the compressor from where further increase in mass-flow will choke the compressor (stonewall area). In this region the gas velocity in the compressor reaches a velocity in the diffuser equal to the speed of sound. At these conditions shock waves cause a excessive decrease in head and extra mass-flow cannot flow through the compressor. When choke occurs at a single compressor stage, it will limit the mass-flow through the whole compressor train.

Surge determines the minimum flow through a compressor and is initiated by two separate phenomena [38]. When the flow through a compressor is reduced, the angle of the flow leaving the impeller decreases. This results in a gas particle travelling a longer flow path from impeller tip to the diffuser outside diameter. During this longer path, generated flow momentum by the impeller dissipates by friction at the diffusers walls. Next to the outlet angle, the incidence angle at the impeller inlet increases during a flow reduction causing flow separation that tends to shift round the impeller blades. Because of this flow separation and when the friction in the diffuser becomes large enough, the compressor is unable to generate enough head to overcome the back pressure at the outlet. This results in a reversed mass-flow (surge) until the back pressure is returned to a level at which the compressor can generate sufficient head again. Without intervention this process will keep repeating causing high oscillations in mass-flow and pressure (surging). The occurring vibrations result in high stresses on the compressor components. If the system is not adequately protected, the compressor could break down and a sudden increase of pressure at the inlet of the compressor train can occur causing unreparable damage to the stripper.
During transient operation of the power plant, the CO$_2$ flow will vary between 40-100% of its nominal mass-flow. Given the operational constraints of the stripper and the pipeline, a suitable control strategy is needed for the compressor train. This strategy should be able to keep all compressors of the compression train in their stable operating range during power plant transients. A review of possible control strategies for centrifugal compressors is presented in the next section.

### 2.3.1 General compressor control strategies

Two different control methods for stable operation of a compressor train can be distinguished, namely avoidance control and active control. Avoidance control aims at keeping the compressor operating point in a region between the surge-line and the capacity limit. Active control aims at stabilizing the system beyond the surge-line during low mass-flows. This control method uses actuators to increase the stable operating region by minimizing and stabilizing the flow instabilities which lead to surge. An overview of these methods can be found in Ref. [29]. While active control can increase the operating range and efficiency of a centrifugal compressor, it is not considered in this thesis because of its limited increase in operating range, controlling complexity and large increase in capital cost. Avoidance control comprehends the general compressor methods: gas circulation valves, suction/discharge throttling, guide vane positioning and variable speed control. Each of these strategies is discussed in the following paragraphs with reference to the ROAD process conditions.

**Recycle valve control**

When the mass-flow through the compressor suddenly decreases, part of the compressed gas from the discharge of the compressor can be fed back through a recycle valve to the suction side of the compressor. As a consequence, the compressor operating point moves away from the surge line because the pressure ratio across the compressor decreases and the mass-flow through the compressor is increased. Recycle valves can either be used as backup system to prevent surge during high flow fluctuations when other avoidance control strategies fail (anti-surge control), or themselves be used to keep the compressor in its stable operating range when the mass-flow decreases (avoidance control). If properly designed, recycle valves enable the compressors to operate at 0-100% of the nominal mass-flow. However, when compressed gas is recycled, the compression process efficiency decreases considerably. Thus the use of this strategy should be minimized as much as possible.

There are many ways of installing recycle valves, notably the number of valves, its control algorithm or valve placement for making it a hot or cold bypass can be changed. An overview is given in Ref. [39]. They distinguish three situations in which recycle valves are expected to prevent surge:

– Start-up situation
– Normal process situation
– Emergency shut-down situation

For each situation different strategies and layouts are used of controlling the recycled flow to prevent surge. In this research only the normal process situation is considered in which the recycle valves are used as avoidance control.


Suction and discharge throttling

These control strategies modulate the inlet or outlet pressure by means of a throttling valve in order to modify the pressure ratio across the compressor train. By altering the pressure ratio, the operating point of the compressor moves to the desired position. As explained in Section 2.1.1, the compressor control system should maintain an optimal operating pressure in the stripper. Since this pressure is close to the atmospheric pressure (1.8 bar), suction throttling at the inlet of the compressor train has therefore only a limited range of modifying the pressure ratio. Moreover, both suction and discharge throttling are an inefficient way of reducing the pressure ratio. This inefficiency is caused, because during throttling control energy is dissipated by increasing the dynamic loss, thus the energy to drive the compressor is not reduced.

Guide vane positioning and variable speed control

Guide vane positioning is based on the possibility to change the guide vanes orientation in order to control the flow angle at which the flow enters the compressor. The guide vanes are progressively closed when the mass-flow through the compressor is reduced. This optimises the angle at which the mass-flow enters and leaves the impeller into the diffuser reducing frictional losses and moves the operating point of the compressor away from the surge line. Even though this method is more efficient than using recycle and throttling valves, closing the guide vanes gives an increased efficiency penalty. When the guide vanes are closed, the increasing friction generated by the vanes makes them act more and more like throttling valves.

A variable speed controller reduces the shaft speed of the compressor when the flow is reduced. In this way the compressor operates at a new speed line corresponding to the reduction in speed. At this new speed line the operating point of the compressor is farther from the surge line. Because the compressor power is a function of the speed cubed, a speed reduction can
2.3 Compressor control

significantly reduce the required compression energy during part load operation. This makes variable speed control the most efficient method for controlling flow variations.

Both control strategies give a wide operating range for the low pressure compressors. Whereas this is not the case for high pressure CO\textsubscript{2} compressors. Figure 2.8 represents a normalized performance map for a high pressure CO\textsubscript{2} guide vane controlled compressor and represents the same map for a variable speed controlled compressor. The triangle represents the optimal design point at which the compressor operates at maximum efficiency. The dashed lines represent the maximum operating range when the compression ratio remains constant during the flow reduction at its optimal design point. It shows these control strategies have a limited operating range when the pressure ratio is fixed like in Case B were the well valve is used to fix the pressure in the pipeline. Without this control valve, the pressure ratio across the compressor train varies allowing a wider operating range. Nevertheless, both strategies will not achieve the full operating range of the flow from the capture unit corresponding to 40-100\% of the nominal CO\textsubscript{2} flow.

2.3.2 Control strategies applied in this research

Because of their limited operating range, both guide vane control and variable speed control will require recycle valves to cover the full part load operation range. This will result in a large efficiency penalty. Roeder et al [26] concluded that by using parallel operating compressor trains (see Figure 2.9), the efficiency of the PCC process could be increased. This configuration offers the possibility of shutting down one of the compressor trains at part load operation while the other train will operate close to its design point. In this way, no recycled flow is needed to extend the operating range of the compressor train. However, adding an extra compressor train will increase the capital costs and maintenance cost during plant operation. To evaluate advantages of such a solution, both a single and double compressor train have been considered in this research.

![Figure 2.9: Process schematic of a double compressor train](image)

For a single compressor train, in order to minimize the use of recycle valves, the operating
range can either be extended with variable speed control or guide vane control systems. Guide vanes are generally less effective in multi-stage compressor systems. Placing guide vanes at the inlet of the compressor train will only effect the first compressor stage. For adequate control of the whole compressor train, guide vanes need to be installed at the inlet of each centrifugal compressor. This greatly increases the complexity of controlling the compressor train. Moreover, the friction created when closing all these guide vanes will significantly decrease the efficiency of the compressor train during minimum load operation. For this reason only variable speed control is considered a valid option.

Ziaii et al. [30] analysed the overall efficiency of a capture unit including a compressor train during transient load operation. They concluded that the power saved by decreasing the compressor train speed is less than the caused efficiency penalty at the capture unit, inflicted by the pressure increase in the stripper. Based on this analyses, when a well control valve is used, the variable speed controller is only effective in increasing the operating by a small range as indicated in Figure 2.8. However, when this valve is not controlled, the pressure in pipeline will drop when the CO₂ stream is reduced at minimum load conditions of the power plant. At these conditions the pressure ratio across the compressor decreases, which should allow the variable speed controller to reduce the compressor speed while maintaining a constant pressure in the stripper. How the different control strategies have been implemented in the dynamic model of the compressor train is discussed in the next paragraphs.

Recycle valve control

To maintain a stable operation region for a single compressor, a recycle valve should start opening just before the operating point of the compressor reaches the surge line. This is needed to take into account variable measurement/response delays and inaccuracies in the valve control chain. For a variable speed compressor, this safe distance is represented by the control line shown in Figure 2.10. That line is typically placed to the right of the surge line at a distance equal to 5-10% of the surge mass-flow and pressure values [40]. The control line is generally described as a square function of the pressure ratio across the compressor:

\[ \phi_c = k_1 \cdot (PR)^2 + k_2 \]  

(2.1)

In this formula \( \phi_c \) is the corrected mass-flow (see Section 3.1.5), \( PR \) the pressure ratio and \( k_1, k_2 \) are constants that determine the control line at 5-10% from the surge line of the compressor. The resulting control scheme to control the recycle valve is schematically shown in Figure 2.11. A PID controller is used to control the recycle valve. Its Set-Point (SP) is set by the corrected mass-flow that represents the minimum allowable flow for the compressor at its current operating Pressure Ratio (PR). The actual corrected mass-flow is received by the PID controller as Process Variable (PV). When the actual flow drops below the minimum flow at which the compressor is from the surge line the controller will attempt, by opening the recycle valve, to eliminate the difference between process variable and set-point. In this schematic the pressure ratio across the compressor is measured in the Pressure Ratio Transmitter (PRT). A Ratio Relay (RR) multiplies the pressure ratio from the PRT with the slope \( k_1 \) and adds the constant \( k_2 \) of the control line determined for the compressor.
actual corrected mass-flow is determined by the Flow Relay (FR) based on the mass-flow, pressure and temperature measurement instruments (respectively FI, PI, TI).

**Variable speed control**

When the CO$_2$ stream reduces by a load reduction of the power plant, the stripper control valve will slowly close to maintain a pressure of 1.8 bar in the stripper. When the compressor train speed is maintained constant at these conditions, the pressure at the inlet of the compressor train will start to drop. The decrease in inlet pressure increases the pressure ratio across the compressor train. This allows the variable speed controller to reduce the compressor speed which will decrease the energy consumption of the compressor train. A control schematic of the variable speed controller is shown in Figure 2.12. Because the pressure in the stripper should always be maintained at 1.8 bar, the inlet pressure of the compressor train is used as PV for the variable speed controller.

**Control of two compressor trains**

Though identical, every compressor train placed in parallel will operate at a slightly different flow, pressure ratio and shaft speed. The compressor train with the least frictional resistance will eventually compress a larger part of the mass-flow. This can cause one train to be fully loaded, pushing the other train into surge. To prevent this from occurring a ratio controller is added. This controller is used to keep the flow evenly divided between the two compressor trains. Next to this controller an extra control logic is needed to diverge the flow to a single compressor train during minimum load operation.

The used control schematic is shown in Figure 2.13. A PID controller is used to control the valve at the inlet of train B. The flow at the inlet of compressor train B is used as PV. The
amount of gas that should flow to compressor train B is set as SP for the controller and determined in the Set-point Relay (SR). This relay calculates this flow based on the desired ratio from the Ratio Relay (RR) and the actual flow at the inlet of compressor train A and B. During full load conditions of the power plant the RR transmits a ratio of 0.5 to the SR. When the total flow from the stripper drops below a capacity of a single compressor train, the ratio is switched to zero, and the valve at inlet of the compressor train B is gradually closed.

2.4 Equation of State

To accurately model the thermodynamic properties of CO\(_2\) at high pressures the ideal Equation of State (EoS) is not adequate. It particularly fails for thermodynamic states close to the critical point. At these conditions, the molecules can no longer be modelled as single entities with no mutual interaction. Mazzoccoli et al. [41] have done an elaborate research to investigate the most accurate EoS for modelling pure CO\(_2\) and CO\(_2\) based mixtures. In their research they concluded that the Span Wagner (S&W) EoS shows the smallest error in predicting the CO\(_2\) properties at high pressures.

The S&W equation of state was developed by Span et al [42], and is an empirical equation explicit in the Helmholtz free energy with two independent variables density \(\rho\) and temperature \(T\). The Helmholtz free energy is split in a solution depending on the ideal-gas behaviour and one that describes the residual part. The latter part is determined in an empirical way by fitting its coefficients to experimental results. This makes the S&W EoS not reliable when it is used for describing the gas beyond the conditions used to determine the experimental data, for instance to describe mixtures of CO\(_2\) and H\(_2\)O. To adequately predict the fluid properties of the CO\(_2\) mixture until the dehydrator separates the H\(_2\)O, the ideal EoS is used. This ideal EoS can be modelled with the 'FlueGas'-media model from the ThermalPower library in Modelica. For the components downstream the dehydrator the 'CO2CoolPropTABLES'-media model according to the S&W EoS is used from the ExternalMedia library [43]. This
media model consist of tables of the CO$_2$ properties. This allows for an accurate generation of the CO$_2$-rich gas streams thermodynamic properties without penalising the computational efficiency of the model.
Chapter 3

Model development

This chapter describes the development of the dynamic model used to simulate the compressor
train, pipeline and storage well (see Section 3.1). The chapter concludes with describing the
steady-state model developed to verify the dynamic model (see Section 3.2).

3.1 Dynamic model

For the development of the dynamic model a procedure consisting of 9 fundamental steps
is followed. This methodology is specifically developed for designing dynamic models and is
presented in Ref. [33]. The steps procedure consists in defining:

1. Model purpose
2. System border and variables
3. Relevant phenomena
4. Hypotheses and assumptions
5. Sub models
6. Conservation laws and constitutive equations
7. Simplifications
8. Implementation and simulation
9. Verification, documentation and application

This method is an iterative process, hence after modelling and verification it is important to
check if the model purpose is met. This may lead to repetition of previous steps, like recon-
considering the relevant phenomena, hypotheses and assumptions. The first five steps represent
the next five sections. Steps six and seven are described in the Appendix B. The verification
of the model is described in Chapter 4 and the results are presented in Chapter 5.
3.1.1 Purpose

The purpose is to develop a simulation tool in order to predict the dynamic performance of the compression section of the Post Combustion Capture (PCC) system of the ROAD project power plant. The model should describe the dynamic behaviour of the compressors, their utility systems (e.g. knock-out drums), transportation pipeline and storage well. The model should also be compatible with a previous developed dynamic model of the capture unit.

![System border of the dynamic model](image)

**Figure 3.1:** System border of the dynamic model

3.1.2 System border and variables

Before starting the development of a model it is important to define the system boundaries, namely the model input and output variables. The boundaries are placed where there are environmental disturbances, controlled inputs and wanted outputs consisted with the purpose. The model boundaries are shown in Figure 3.1. In addition, in this figure the system is divided into two parts depending on the Equation of State (EoS) used to predict the thermodynamic properties of the process streams. The input and output variables of the system are:

- Gas stream from stripper \((\dot{m}, h, P, X)\) / input
- Liquid stream at knock-out drums \((\dot{m}, h, P, X)\) / output
- Heat exchanged at inter-cooler \((Q)\) / output
- Mechanical power to compressor \((\omega, T_q)\) / input
3.1 Dynamic model

- Heat loss in the pipeline and storage well ($\dot{Q}$) / output
- CO$_2$ stream stored in well reservoir ($\dot{m}, h, P, X$) / output

3.1.3 Relevant phenomena

The principle of Parsimony states that among different models that predict system characteristics evenly well, the simplest one should be selected. So the development of a dynamic model should start investigating the relevant physical phenomena which are essential to understand the system behaviour. The relevant phenomena for the developed system model are listed below. Those of each sub-model are listed in the appendix.

- Conversion of mechanical energy into pressure in the compressor stages
- Frictional losses in all components
- Convective heat transfer in the inter-coolers, pipeline and well
- Accumulation of mass and energy in the knock-out drums, dehydrator, pipeline and storage well
- Heat and mass transfer between the liquid and gas phase in the knock-out drums
- Condensation of water in the knock-out drums and heat exchangers
- Adsorption of water in the dehydrator
- Partial condensation of the CO$_2$ rich stream in the pipeline
- Hydrate formation in the well
- Flow instabilities in the compressors (e.g. surge, stall)

3.1.4 Hypotheses and assumptions

To simplify the modelling of real physical systems, adequate hypotheses and assumptions should be made. For each sub-model the hypotheses and assumptions are listed in the appendix. The following are made for the general development of the dynamic model:

- Because the gas stream components O$_2$, Ar, N$_2$, have negligible influence on the compressor train performance (see Section 2.2.2), they are omitted
- The variations in temperature and composition of the feed stream from the stripper are relative small and therefore the flue gas composition is assumed constant during power plant load variations
- The volume of the pipes and ducts between subsystems are small relative to the modelled components and therefore omitted
- The dehydrator is assumed capable of adsorbing all water in the gas stream (see Section 3.1.5)
- Due to proper insulation, heat loss to the environment is assumed negligible for all components except for the pipeline and storage well
- The expected flow fluctuations are assumed to be within the control capability of the anti-surge control system. It is therefore not required to model compressor behaviour in surge or stall conditions
- Frictional forces in all pipes and ducts connecting the different model subsystems are assumed relatively small and can therefore be neglected
Because of the large energy capacity of the flow and relative small temperature fluctuations, the energy accumulation in the metal parts of all components can be neglected. The mechanical motion of the compressor train are assumed much slower than the dynamics related to the energy and mass fluctuations in the system, therefore this mechanical motion is neglected. The control strategies applied, constraints the operating conditions such that hydrate formation in the well does not occur (see Section 2.1.2).

Figure 3.2: Example of thermo-hydraulic systems described by resistive and storage modules connected in series [6]

3.1.5 Subsystems

To develop the dynamic model of the system a modular approach is used. This approach establishes that all calculations performed in a module or sub-module are only dependent on its inputs, internal variables and parameters. By using this approach, large systems can be decomposed into smaller sub-systems or components. Generally these sub-systems represent a physical component of the modelled system. This also makes it easier to re-use developed modules and to adopt them for modelling different system configurations.

Table 3.1: Bilateral coupled variables

<table>
<thead>
<tr>
<th>Potential variable</th>
<th>Flow variable</th>
<th>Type of power transfer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure</td>
<td>Mass flow</td>
<td>Mechanical</td>
</tr>
<tr>
<td>Temperature</td>
<td>Heat flux</td>
<td>Thermal</td>
</tr>
<tr>
<td>Rotational speed</td>
<td>Torque</td>
<td>Mechanical</td>
</tr>
<tr>
<td>Velocity</td>
<td>Force</td>
<td>Mechanical</td>
</tr>
<tr>
<td>Voltage</td>
<td>Current</td>
<td>Electrical</td>
</tr>
</tbody>
</table>

In order to prevent numerical issues during the simulation it is fundamental to avoid that two bilaterally coupled variables are specified as inputs at the same model boundary. Examples
of bilateral coupled variables are given in Table 3.1. To ensure that the developed models comply with this requirement, the system is decomposed into a series of 'storage' and 'resistive' modules. When modelling thermodynamic modules the former can store mass and energy but not contemplate momentum variations, whereas the latter vice versa. An example of system components that follow this modelling approach is given in Figure 3.2, while the resulting structure of the complete dynamic module developed in this thesis is shown in Figure 3.3. The remainder of this section provides an overview of the main characteristics of the different component models of the analysed systems.

**Figure 3.3:** Composition of the system in 'Storage' and 'Resistive' modules

**Compressor**

In literature several lumped parameter models for centrifugal compressors exist. An overview of these is given in Ref. [29]. Most of these models are obtained by extending a model developed by Greitzer [44]. This model uses the NASA shock-loss theory to describe the frictional forces in the compressor that act on the flow [45]. To apply this theory, reliable geometric data of the internal compressor components are required. Moreover, most of these models assume the flow incompressible. This assumption is not completely valid for fluids at high pressures, especially for operating conditions close to the critical point.
Another method is to use compressor maps. These maps are generally used to predict the performance of the compressor in on- and off-design conditions and are generated with advanced computer models or based on extensive experimental data. Since the main objective is to understand the impact of the compressor train operation on the dynamic performance of the other systems within the PCC process, the fast dynamics that can occur at single compressors are neglected. Compressor maps are adequate to investigate the main dynamics of the system and are therefore used in this research. Even though they cannot predict surge and stall, as long as the anti-surge system can adequately react to the predicted flow fluctuations these flow instabilities in the compressor will not occur.

Notably, the compressor maps used to model the compressor train performance were provided by TNO and are generated with the program CompAero, a program to analyse centrifugal and axial-flow compressors. These maps where originally generated for compressors with a higher power capacity. It was therefore necessary to make them dimensionless and adopt them for the ROAD project conditions. The obtained maps are presented in Appendix B. A generalized efficiency map, provided by a compressor manufacturer, is used to describe the efficiency of each compressor in the dynamic model. These maps are made dimensionless by using the Buckingham II theorem, namely by correcting the maps variables in the following dimensionless quantities:

- **Pressure ratio** $\Psi$; which is given as $\frac{P_{out}}{P_{in}}$, where $P_{out}$ is the discharge pressure and $P_{in}$ the inlet pressure.

- **Compressor efficiency** $\eta$; which is given by $\frac{h_{s, out} - h_{in}}{h_{s, out} - h_{in}}$. The numerator represents the difference between the inlet enthalpy $h_{in}$ and $h_{s, out}$, the isotropic outlet enthalpy.

**Figure 3.4:** Auxiliary beta lines parallel to the surge line [7]

**Figure 3.5:** Tabulated flow coefficient, pressure ratio and efficiency [7]
The denominator represents the difference between the inlet enthalpy and the outlet enthalpy $h_{\text{out}}$.

- **Corrected mass-flow** $\phi_c$: which is defined as $\dot{m} \sqrt{\frac{T_{\text{in}}}{P_{\text{in}} D^2}}$, where $\dot{m}$ is the mass-flow, $T_{\text{in}}$ the inlet temperature, $D$ the diameter of the impeller and $P_{\text{in}}$ the inlet pressure.

- **Corrected speed** $N_c$: which is given by $\frac{N D}{\sqrt{T_{\text{in}}}}$, where $N$ is the rotational speed of the impeller.

To efficiently implement the performance maps in the dynamic model, auxiliary lines called beta lines are introduced (see Figure 3.4). These lines are parallel to the surge line of the compressor and allow preventing of singularities that would arise when using speed lines. These beta lines are then used to construct three tables with the beta line number on the y-axes, and the corrected speed on the x-axes (see Figure 3.5), which tabulates the normalized pressure ratio, flow coefficient and efficiency of the compressor. After their adaptation to match the process conditions of the ROAD project power plant, the used compressor parameters are tabulated in Appendix B.

**Recycle valve**

There are multiple strategies of installing recycle valves for a compressor train. A comprehensive list of these methods is given in Ref. [39]. To prevent a constant increase of the temperature at the inlet of the compressor when the gas flow is recycled, the recycle valve is placed downstream an inter-cooler. When multiple compressor stages are used, a single recycle line is not sufficient to prevent surge in all the compressors. If a compressor not directly connected to the outlet pipeline of the recycle valve operates near the surge line, the recycled gas flow would not reach this compressor in time to prevent surge. This delay is further increased when there are large volumes between the compressor stages. This could be extremely dangerous for the high pressure compressors. For the reasons mentioned above, it has been decided to place a recycle valve between each compression stage. The valve component of the ThermoPower library with a linear flow characteristic is used to model the recycle valves. The valves are sized such that they can recycle the maximum process flow.

**Inter-cooler**

For inter-cooling between the compressor stages normal heat transfer equipment is used. The most commonly type of heat exchanger used in the process industry is the shell and tube heat-exchanger. In these type of heat-exchangers the hot CO$_2$ gas will flow trough the tubes, transferring heat to the cooling fluid flowing through the shell. Inter-cooling the gas stream will cause a pressure drop and delay in mass flow variations which will influence the performance of the compressor train. To calculate this pressure drop and the volume of the heat exchanger, Kern’s design method is used [8]. The detailed results and the assumptions made for these calculations can be found in Appendix A. The calculated volumes and pressure drop in the inter-coolers for a configuration with a single compressor train are listed in Table 3.2.
Table 3.2: Inter-cooler pressure drop and volume for a single compressor train

<table>
<thead>
<tr>
<th>Inter-cooler</th>
<th>( \Delta P ) [bar]</th>
<th>( V ) [m³]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0,03</td>
<td>10,1</td>
</tr>
<tr>
<td>2</td>
<td>0,05</td>
<td>5,7</td>
</tr>
<tr>
<td>3</td>
<td>0,10</td>
<td>4,8</td>
</tr>
<tr>
<td>4</td>
<td>0,32</td>
<td>2,1</td>
</tr>
<tr>
<td>5</td>
<td>0,23</td>
<td>2,3</td>
</tr>
<tr>
<td>6</td>
<td>0,01</td>
<td>6,7</td>
</tr>
</tbody>
</table>

It is assumed that the heat exchangers used for the compressor train of the ROAD project power plant are of such capacity that they are able to cool the stream to the desired temperature in all operating conditions. Moreover, no large temperature fluctuations in the gas stream are expected during normal and transient operation of the compressor train. Therefore, to reduce the complexity of the dynamic model and shorten the simulation time, only the tube side of the heat exchangers is modelled. This is represented in the model by a storage module with a thermal port to account for the heat transfer through the heat exchanger walls. The amount of heat transferred is calculated through a PID controller that simulates the head duty of the heat-exchanger.

**Knock-Out Drum**

Knock-out drums are large vertical vessels where the liquid phase in the inlet stream is separated by gravity (see Figure 3.6). A pump drains the condensed water from the bottom of the vessel. As the pressure increases of the CO₂ stream after each compression stage, the water content in the stream diminishes in each knock-out drum.

![Vertical Knock-out drum](image)
Before the knock-out drums, water will already condense in the inter-coolers releasing significant amounts of thermal energy. The flue gas model used to model the mixture of CO$_2$ and H$_2$O in Modelica is unable to calculate this released condensation heat. If this heat is neglected, the amount of heat transferred in the inter-cooler is underestimated up to 30% (see Chapter 4). To reduce this error, this condensation heat is calculated in the knock-out drum. An new component model was developed for this purpose. The component consists of a single storage module. The volume, height and diameter of the knock-out drums are calculated based on the guidelines given in Ref. [8]. The calculated values are presented in Table 3.3

### Table 3.3: Knock-out drum volume, diameter and height for a single compressor train

<table>
<thead>
<tr>
<th></th>
<th>V [m$^3$]</th>
<th>D [m]</th>
<th>H [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Knock-out drum 1</td>
<td>55.9</td>
<td>4.1</td>
<td>4.1</td>
</tr>
<tr>
<td>Knock-out drum 2</td>
<td>32.5</td>
<td>3.4</td>
<td>3.4</td>
</tr>
<tr>
<td>Knock-out drum 3</td>
<td>19.0</td>
<td>2.8</td>
<td>2.8</td>
</tr>
<tr>
<td>Knock-out drum 4</td>
<td>11.0</td>
<td>2.2</td>
<td>2.2</td>
</tr>
<tr>
<td>Knock-out drum 5</td>
<td>6.3</td>
<td>1.8</td>
<td>1.8</td>
</tr>
</tbody>
</table>

**Dehydrator**

A dehydrator consists of a column filled with an adsorbent material. During operation the water molecules will attach to the surface of the adsorbent. Commonly used adsorbent’s for removing water from industrial gases are silica gel and activated alumina. When the adsorbent is saturated, it is regenerated by blowing hot air through the column. Therefore, multiple columns are placed in parallel to ensure continuous drying of the gas stream. Each of these is used in cycles, 3-24 hours, depending on its size and design. In this work it is assumed that switching between these columns has negligence influence on the compressor performance. For this reason only one column is modelled.

Because the dehydrator is placed at a high pressure, most of the H$_2$O is already condensed in the knock-out drums. The amount of water that is adsorbed in the dehydrator will therefore only release a small amount of energy compared to the energy of the total flow. The dehydrator will therefore only cause a pressure drop and delay in mass variations in the compressor train. The dehydrator column is therefore modelled as a resistive and storage module. The dimensions of the dehydrator are determined by the purity required in the outlet stream and type of adsorbent used. The volume and pressure drop are calculated based on the guidelines in Ref. [46]. The properties of the adsorbent and the column dimensions adopted in the model are listed in Table 3.4. This data is retrieved from the information charts of a common adsorbent manufacture.

Master of Science Thesis

M.H.L. Ogink
Table 3.4: Dehydrator dimensions and adsorbent data

<table>
<thead>
<tr>
<th>Adsorbent</th>
<th>Activated Alumina</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surface area [m$^2$/g]</td>
<td>340</td>
</tr>
<tr>
<td>Packed bulk density [kg/m$^3$]</td>
<td>769</td>
</tr>
<tr>
<td>Equivalent spherical diameter [mm]</td>
<td>4.7</td>
</tr>
<tr>
<td>Porosity</td>
<td>0.38</td>
</tr>
<tr>
<td>Pore volume [cc/g]</td>
<td>0.5</td>
</tr>
<tr>
<td>Cycle time [hour]</td>
<td>24</td>
</tr>
<tr>
<td>Total volume [m$^3$]</td>
<td>17.5</td>
</tr>
<tr>
<td>Total pore volume [m$^3$]</td>
<td>6.7</td>
</tr>
<tr>
<td>Pressure drop [bar]</td>
<td>0.3</td>
</tr>
</tbody>
</table>

Pipeline & Storage well

After compression the CO$_2$ is transported through a pipeline of 23 km on the seabed. Despite insulation of the pipeline, the CO$_2$ flow loses considerable heat. Furthermore, friction in the pipeline causes a pressure drop in the CO$_2$ stream and delay in mass flow variations throughout the system due to its considerable size. Based on the steady state analyses performed by TNO, it has been concluded that an adequate prediction of the pipeline performance is achieved by disconnecting the system into five storage modules with thermal ports to account for the heat transfer losses. The storage modules are connected with resistive modules in order to describe the pressure drop in the CO$_2$ stream.

The storage well is described into three different control volumes (see Figure 2.2). One storage module to account for the tubing consists of a large diameter, one for the tubing with a smaller diameter and one for the mean reservoir. In between a resistive component is added to calculate the pressure drop. The full equation derivation is given in Appendix B. The frictional forces are modelled on the basis of the results from the steady state analyses carried out by TNO for the storage well. The parameters of the pipeline and storage well are listed in Table A.4.

Table 3.5: Pipeline and storage well parameters

<table>
<thead>
<tr>
<th></th>
<th>Pipeline</th>
<th>Large tubing well</th>
<th>Small tubing well</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pipeline length [m]</td>
<td>23.375</td>
<td>3700</td>
<td>383</td>
</tr>
<tr>
<td>Pipeline diameter [m]</td>
<td>0.375</td>
<td>0.155</td>
<td>0.1</td>
</tr>
<tr>
<td>Depth into well [m]</td>
<td>0</td>
<td>-3020</td>
<td>-3182</td>
</tr>
<tr>
<td>Overall heat transfer coefficient [W/m$^2$K]</td>
<td>1.5</td>
<td>9.5</td>
<td>9.5</td>
</tr>
<tr>
<td>Average friction coefficient [-]</td>
<td>0.0068</td>
<td>0.0106</td>
<td>0.1485</td>
</tr>
<tr>
<td>Average surrounding temperature [$^\circ$C]</td>
<td>8</td>
<td>54</td>
<td>120</td>
</tr>
</tbody>
</table>
Stripper

At the beginning of this research there was the intention to connect the developed dynamic model to an existing dynamic model of the capture unit. Due to the complexity of the models and the sensitivity of the stripper model, the process variables could not converge during the initialisation of the model. These convergence problems were caused by the used approach adopted to predict the thermodynamic equilibrium in the stripper that are based on experimental correlations. During the solution initialization of the problem, the solver was unable find a solution because during the initialization the correlations are outside their validated range and provide inconsistent predictions. Further work is required to solve these numerical problems.

Figure 3.7: Stripper flow response to the powerplant ramping: (a) from full to minimum load operation (b) from minimum to full load operation.

To still evaluate the influence of the compressor train on the stripper, a simplified stripper model was developed. A response from nominal operating conditions to 40% of the rated power, and vice versa, was simulated with the already developed CO\textsubscript{2} capture unit model. The mass-flow leaving the stripper is shown in Figure 3.7. A flow source from the ThermalPower library is used to mimic this response. A volume resembling that of the stripper is placed between the flow source and a control valve, used to keep the pressure at 1.8 bar.

3.2 Steady state model

To verify the correctness of the developed dynamic model, a steady-state model has been developed with the commercial software tool ASPEN. The components of the compressor train are modelled with resembling components from the component library in ASPEN. Because for the dehydrator component no identical component was available it is modelled with a separator component. For the knock-out drum components an adiabatic flash vessel component is used. The model process flow diagram is shown in Figure 3.8. The recycle valves needed to verify the dynamic model at minimum load conditions are not shown in this figure. The model settings are shown in Table 3.6.
To verify if the ideal FleuGas-media model used in the dynamic model is adequate in describing the mixture of CO$_2$ and H$_2$O, a more accurate EoS is used in the ASPEN model. Austegard et al. [47] researched the solubility of H$_2$O in CO$_2$ for a wide range of process conditions. They collected experimental data from literature and investigated which EoS and mixing rules were best to describe the CO$_2$ mixtures. The second order Soave Redlich Kwong EoS with modified Huron Vidal mixing rules (SRKHV) produced the most accurate results for the pressure and temperature ranges of interest for this research. Although the SRKHV is accurate at describing a mixture of CO$_2$ and H$_2$O, it slightly miscalculates the CO$_2$ density at high pressures. To verify the steady state results of the components operating at high pressures the Lee Kesler Plocker (LKP) EoS was used in ASPEN. According to Ref. [41], this EoS is capable of predicting accurate CO$_2$ properties at high pressures.
Table 3.6: ASPEN input selections, variables and parameters for the steady state model of the single compressor train in full load operation

<table>
<thead>
<tr>
<th>Description</th>
<th>Selection/value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor 1</td>
<td></td>
</tr>
<tr>
<td>Model type</td>
<td>Isentropic</td>
</tr>
<tr>
<td>Discharge pressure</td>
<td>3.19 bar</td>
</tr>
<tr>
<td>Isentropic</td>
<td>0.848</td>
</tr>
<tr>
<td>Mechanical</td>
<td>0.98</td>
</tr>
<tr>
<td>Heater 1 (Inter-cooler)</td>
<td></td>
</tr>
<tr>
<td>Flash type</td>
<td>Temperature/Pressure</td>
</tr>
<tr>
<td>Temperature</td>
<td>38 °C</td>
</tr>
<tr>
<td>Pressure</td>
<td>0 bar</td>
</tr>
<tr>
<td>Flash 1 (Knock-out drum)</td>
<td></td>
</tr>
<tr>
<td>Flash type</td>
<td>Temperature/Pressure</td>
</tr>
<tr>
<td>Temperature</td>
<td>40 °C</td>
</tr>
<tr>
<td>Pressure</td>
<td>0 bar</td>
</tr>
<tr>
<td>Separator (Dehydrator)</td>
<td></td>
</tr>
<tr>
<td>Substream</td>
<td>MIXED</td>
</tr>
<tr>
<td>CO₂ split fraction</td>
<td>1</td>
</tr>
<tr>
<td>Input</td>
<td></td>
</tr>
<tr>
<td>Inlet flow</td>
<td>49 kg/s</td>
</tr>
<tr>
<td>Inlet flow composition (CO₂, H₂O)</td>
<td>0.965, 0.035</td>
</tr>
<tr>
<td>Inlet flow temperature</td>
<td>40 °C</td>
</tr>
<tr>
<td>General</td>
<td></td>
</tr>
<tr>
<td>Input mode</td>
<td>Steady-State</td>
</tr>
<tr>
<td>Solution method</td>
<td>Sequential Modular</td>
</tr>
<tr>
<td>Property method</td>
<td>RKSMHV2</td>
</tr>
</tbody>
</table>
Chapter 4

Model Verification

This chapter reports the results of a preliminary verification performed to investigate if the dynamic model was correctly implemented and if the assumptions and hypotheses made in the previous chapter are justified. First, the dynamic model is verified by comparing its steady state results with those of steady state models of the system (see Section 4.1). Second, a qualitative verification of the dynamic trends predicted by the dynamic model is evaluated in Section 4.2.

4.1 Steady State Verification

Two steady-state models where available to verify the dynamic model. One model was created in ASPEN [34], as described in Chapter 3, and deals with the compressor train. The second model describes the performance of the pipeline and storage well and was developed by TNO with the program OLGA [48]. This modelling program is specifically designed to simulate multiphase flows in pipelines and wells for the oil and gas industry. The verification of the compressor train model is presented in Section 4.1.1, while the verification of the pipeline and storage well models is presented in Section 4.1.2.

4.1.1 Compressor train verification

To verify the results of the compressor train different test cases were considered. In the first simulation the inlet mass-flow of the compressor train was set equal to the nominal mass-flow (49 kg/s), while in the second simulation the mass-flow was set equal to 40% of the nominal flow.
### Component Verification

<table>
<thead>
<tr>
<th>Component</th>
<th>Units</th>
<th>ASPEN</th>
<th>Modelica</th>
<th>Difference(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor 1</td>
<td>P_{in} [bar]</td>
<td>1.62</td>
<td>1.61</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>P_{out} [bar]</td>
<td>3.19</td>
<td>3.19</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>m [kg/s]</td>
<td>48.22</td>
<td>48.21</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>T_{in} [°C]</td>
<td>40.00</td>
<td>40.00</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>T_{out} [°C]</td>
<td>98.02</td>
<td>98.15</td>
<td>-0.1</td>
</tr>
<tr>
<td></td>
<td>W [MW]</td>
<td>2.52</td>
<td>2.49</td>
<td>1.2</td>
</tr>
<tr>
<td>Compressor 2</td>
<td>P_{out} [bar]</td>
<td>3.19</td>
<td>3.19</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>m [kg/s]</td>
<td>47.69</td>
<td>47.70</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>T_{in} [°C]</td>
<td>39.00</td>
<td>38.00</td>
<td>0.0</td>
</tr>
<tr>
<td>Intercooler 1</td>
<td>P_{in} [bar]</td>
<td>6.37</td>
<td>6.37</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>P_{out} [bar]</td>
<td>25.63</td>
<td>25.63</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>m [kg/s]</td>
<td>47.48</td>
<td>47.49</td>
<td>-0.1</td>
</tr>
<tr>
<td></td>
<td>T_{in} [°C]</td>
<td>39.00</td>
<td>38.00</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>T_{out} [°C]</td>
<td>97.75</td>
<td>98.83</td>
<td>1.0</td>
</tr>
<tr>
<td></td>
<td>W [MW]</td>
<td>2.41</td>
<td>2.44</td>
<td>-0.9</td>
</tr>
<tr>
<td>Intercooler 2</td>
<td>P_{in} [bar]</td>
<td>51.21</td>
<td>51.21</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>P_{out} [bar]</td>
<td>88.66</td>
<td>88.66</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>m [kg/s]</td>
<td>47.25</td>
<td>47.29</td>
<td>-0.1</td>
</tr>
<tr>
<td></td>
<td>T_{in} [°C]</td>
<td>39.00</td>
<td>38.00</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>T_{out} [°C]</td>
<td>99.29</td>
<td>97.02</td>
<td>2.3</td>
</tr>
<tr>
<td>Knockout drum 1</td>
<td>P_{in} [bar]</td>
<td>12.76</td>
<td>12.76</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>P_{out} [bar]</td>
<td>25.63</td>
<td>25.63</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>m [kg/s]</td>
<td>47.25</td>
<td>47.29</td>
<td>-0.1</td>
</tr>
<tr>
<td></td>
<td>T_{in} [°C]</td>
<td>39.00</td>
<td>38.00</td>
<td>0.0</td>
</tr>
<tr>
<td>Knockout drum 2</td>
<td>P_{out} [bar]</td>
<td>12.76</td>
<td>12.76</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>m [kg/s]</td>
<td>47.25</td>
<td>47.29</td>
<td>-0.1</td>
</tr>
<tr>
<td></td>
<td>T_{in} [°C]</td>
<td>39.00</td>
<td>38.00</td>
<td>0.0</td>
</tr>
</tbody>
</table>

**Model Verification**

[M.H.L. Ogink](#) Master of Science Thesis
4.1 Steady State Verification

The results for the first test case are presented in Table 4.1. The last table column shows the discrepancy between the results of the ASPEN model and those of the Modelica model. Between most of the process variables exists only a small deviation. Large difference are however observed for the cooling duty of the inter-coolers. As explained in Section 3.1.5, the Modelica model of the inter-cooler does not take into account the heat released due to condensation. For this reason the cooling duty of this component is structurally lower compared to the ASPEN results. By adding the heat of condensation, calculated in the knock-out drum component, the difference decreases to about 10% for all inter-coolers. This difference can be accounted for by two simplification made during the development of the Modelica components models. First, the heat released related to the dissolving of CO\textsubscript{2} in water is neglected. Second, the heat capacity is assumed constant, whereas in the ASPEN it varies with the temperature.

A deviation in cooling duty is also observed for inter-cooler 6. This deviation cannot be ascribed to the condensation of water because it is located after the dehydrator. This difference is introduced by the different Equation of State (EoS) used to predict the properties of CO\textsubscript{2}. The Modelica model uses the Span Wagner (S&W) EoS which is more accurate for high CO\textsubscript{2} pressures than SRKHV EoS used in ASPEN.

As far as the knock-out drums are considered, a large deviation in the liquid CO\textsubscript{2} mass fraction is observed. This large deviation is due to different mixing rules used to calculate the amount of CO\textsubscript{2} dissolved in the condensed liquid. The ASPEN model uses the more accurate Soave Redlich Kwong EoS with modified Huron Vidal mixing rules (SRKHV) while the Modelica model uses ideal mixing rules. Even though there is a relevant difference indicated between the two models the absolute amount of CO\textsubscript{2} in the liquid is extremely low, and thus, this inaccuracy will have negligible influence on the compressor train performance. The results for the simulation at minimum load of the power plant showed similar deviations which can be explained by the same considerations given above.

### Table 4.1: Steady state results of the compressor train at full load of the power plant

<table>
<thead>
<tr>
<th>Component</th>
<th>Units</th>
<th>ASPEN</th>
<th>Modelica</th>
<th>Difference(%)</th>
<th>Component</th>
<th>Units</th>
<th>ASPEN</th>
<th>Modelica</th>
<th>Difference(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Knockout drum</td>
<td>P\textsubscript{in} [bar]</td>
<td>25,63</td>
<td>25,63</td>
<td>0,0</td>
<td>Knockout drum</td>
<td>P\textsubscript{in} [bar]</td>
<td>25,63</td>
<td>25,63</td>
<td>0,0</td>
</tr>
<tr>
<td></td>
<td>T\textsubscript{in} [°C]</td>
<td>38,00</td>
<td>38,00</td>
<td>0,0</td>
<td></td>
<td>T\textsubscript{in} [°C]</td>
<td>38,00</td>
<td>38,00</td>
<td>0,0</td>
</tr>
<tr>
<td></td>
<td>f\textsubscript{in} [kg/s]</td>
<td>47,37</td>
<td>47,39</td>
<td>0,0</td>
<td></td>
<td>f\textsubscript{in} [kg/s]</td>
<td>47,37</td>
<td>47,34</td>
<td>-0,1</td>
</tr>
<tr>
<td></td>
<td>f\textsubscript{out} [kg/s]</td>
<td>47,31</td>
<td>47,34</td>
<td>-0,1</td>
<td></td>
<td>f\textsubscript{out} [kg/s]</td>
<td>0,06</td>
<td>0,05</td>
<td>14,1</td>
</tr>
<tr>
<td></td>
<td>f\textsubscript{b} [kg/s]</td>
<td>0,06</td>
<td>0,05</td>
<td>14,1</td>
<td></td>
<td>x\textsubscript{H\textsubscript{2}O} [kg/kg]</td>
<td>0,87</td>
<td>0,96</td>
<td>-9,0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0,13</td>
<td>0,04</td>
<td>211,8</td>
<td></td>
<td>y\textsubscript{H\textsubscript{2}O, in} [kg/kg]</td>
<td>0,002</td>
<td>0,002</td>
<td>9,1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0,001</td>
<td>0,001</td>
<td>18,9</td>
<td></td>
<td>y\textsubscript{H\textsubscript{2}O, out} [kg/kg]</td>
<td>0,001</td>
<td>0,001</td>
<td>18,9</td>
</tr>
</tbody>
</table>

4.1.2 Pipeline and storage well verification

Table 4.2 shows the simulation results for the pipeline and storage well simulated with the Modelica and OLGA models. The two models are compared at different mass-flows and reservoir pressures on the basis of the simulation results described at four different stations along the pipeline and well (see Figure 4.1). Namely the pipeline inlet, well head, well small tubing inlet and CO\textsubscript{2} reservoir.

The steady state results show that most values deviate by only a small amount, except
in some cases for the stream temperature. This difference originates in the assumed soil temperature distribution. In the Modelica model it is assumed that the soil temperature linearly increases with depth, from 7°C at the well head to 123 °C at reservoir depth. To calculate the temperature difference for the heat flux from the soil to the gas flowing in the tubes, an average soil temperature is used for each control volume describing the well. In the OLGA program more temperature nodes are used increasing consequently the accuracy of the model. Because the temperature differences are small, it is assumed that they have little impact on the compressor train performance and dynamic behaviour.

4.2 Qualitative Verification of the dynamic model

To assess if the Modelica models can correctly predict the dynamic behaviour of the compressor train, different dynamic simulations are performed. By setting ramp or stepwise changes to selected input variables while other variables are kept constant. The dynamic prediction of the dynamic model are then analysed to check if the depicted trends are consistent with a common understanding of the modelled phenomena. However, when the simulations are performed with the whole dynamic model, it is hard to pinpoint the source of the incorrect physical behaviour. For this reason, different parts where analysed to investigate one-by-one the behaviour of three different sub-systems of the process. The sub-systems of the performed simulations are shown in Figure 4.1.

---

Figure 4.1: Subsystems and pipeline/storage well station points considered for model verification
Table 4.2: Steady-state results of the pipeline and storage well (PI=Pipeline Inlet, WH=Well Head, ST=inlet Small Tubing, BR=Bottom Reservoir)

| Conditions | Units | TNO | Modelica | Difference (%) | Conditions | Units | TNO | Modelica | Difference (%) |
|------------|-------|-----|----------|----------------|------------|-------|-----|----------|----------------|----------------|
| $P_{res} = 20$ bar | $P_{PI}$ [bar] | 82.1 | 81.7 | -0.5 | $P_{res} = 20$ bar | $P_{PI}$ [bar] | 40.0 | 41.2 | 2.9 |
| $\dot{m}_1 = 100\%$ | $T_{PI}$ [°C] | 70.0 | 70.0 | 0.0 | $\dot{m}_1 = 40\%$ | $T_{PI}$ [°C] | 70.0 | 70.0 | 0.0 |
| $P_{WH}$ [bar] | 77.8 | 76.6 | -1.5 | $P_{WH}$ [bar] | 38.3 | 39.3 | 2.6 |
| $T_{WH}$ [°C] | 45.5 | 44.8 | -1.7 | $T_{WH}$ [°C] | 15.7 | 18.6 | 18.0 |
| $P_{ST}$ [bar] | 95.0 | | | $P_{ST}$ [bar] | 50.6 | | |
| $T_{ST}$ [°C] | 71.6 | | | $T_{ST}$ [°C] | 65.1 | | |
| $P_{BR}$ [bar] | 29.2 | 29.4 | 0.5 | $P_{BR}$ [bar] | 23.4 | 23.8 | 1.5 |
| $T_{BR}$ [°C] | 13.5 | 15.8 | 16.9 | $T_{BR}$ [°C] | 47.3 | 47.4 | 0.2 |
| $P_{res} = 100$ bar | $P_{PI}$ [bar] | 88.9 | 87.8 | -1.3 | $P_{res} = 100$ bar | $P_{PI}$ [bar] | 55.0 | 56.7 | 3.2 |
| $\dot{m}_1 = 100\%$ | $T_{PI}$ [°C] | 70.0 | 70.0 | 0.0 | $\dot{m}_1 = 40\%$ | $T_{PI}$ [°C] | 70.0 | 70.0 | 0.0 |
| $P_{WH}$ [bar] | 85.0 | 83.4 | -1.9 | $P_{WH}$ [bar] | 53.7 | 55.6 | 3.5 |
| $T_{WH}$ [°C] | 48.1 | 47.3 | -1.8 | $T_{WH}$ [°C] | 22.4 | 24.7 | 10.4 |
| $P_{ST}$ [bar] | 125.8 | | | $P_{ST}$ [bar] | 103.6 | | |
| $T_{ST}$ [°C] | 83.7 | | | $T_{ST}$ [°C] | 80.3 | | |
| $P_{BR}$ [bar] | 102.1 | 102.4 | 0.3 | $P_{BR}$ [bar] | 100.9 | 100.9 | 0.1 |
| $T_{BR}$ [°C] | 70.5 | 72.2 | 2.3 | $T_{BR}$ [°C] | 79.9 | 80.8 | 1.1 |
| $P_{res} = 200$ bar | $P_{PI}$ [bar] | 106.7 | 107.8 | 1.0 | $P_{res} = 200$ bar | $P_{PI}$ [bar] | 76.6 | 80.3 | 5.0 |
| $\dot{m}_1 = 100\%$ | $T_{PI}$ [°C] | 70.0 | 70.0 | 0.0 | $\dot{m}_1 = 40\%$ | $T_{PI}$ [°C] | 70.0 | 70.0 | 0.0 |
| $P_{WH}$ [bar] | 103.5 | 104.8 | 1.2 | $P_{WH}$ [bar] | 75.3 | 79.7 | 5.9 |
| $T_{WH}$ [°C] | 53.5 | 53.9 | 0.7 | $T_{WH}$ [°C] | 35.1 | 37.1 | 5.8 |
| $P_{ST}$ [bar] | 210.8 | | | $P_{ST}$ [bar] | 195.5 | | |
| $T_{ST}$ [°C] | 97.9 | | | $T_{ST}$ [°C] | 93.4 | | |
| $P_{BR}$ [bar] | 201.3 | 201.1 | -0.1 | $P_{BR}$ [bar] | 200.4 | 200.4 | 0.0 |
| $T_{BR}$ [°C] | 96.9 | 96.6 | -0.3 | $T_{BR}$ [°C] | 96.0 | 95.8 | -0.2 |
| $P_{res} = 300$ bar | $P_{PI}$ [bar] | 133.4 | 133.4 | 0.0 | $P_{res} = 300$ bar | $P_{PI}$ [bar] | 102.0 | 105.7 | 3.6 |
| $\dot{m}_1 = 100\%$ | $T_{PI}$ [°C] | 70.0 | 70.0 | 0.0 | $\dot{m}_1 = 40\%$ | $T_{PI}$ [°C] | 70.0 | 70.0 | 0.0 |
| $P_{WH}$ [bar] | 130.8 | 131.3 | 0.4 | $P_{WH}$ [bar] | 100.3 | 105.3 | 5.0 |
| $T_{WH}$ [°C] | 56.6 | 56.5 | -0.2 | $T_{WH}$ [°C] | 44.1 | 45.3 | 2.8 |
| $P_{ST}$ [bar] | 296.0 | | | $P_{ST}$ [bar] | 290.0 | | |
| $T_{ST}$ [°C] | 92.2 | | | $T_{ST}$ [°C] | 85.0 | | |
| $P_{BR}$ [bar] | 301.0 | 300.9 | 0.0 | $P_{BR}$ [bar] | 300.3 | 300.4 | 0.0 |
| $T_{BR}$ [°C] | 93.7 | 93.7 | 0.0 | $T_{BR}$ [°C] | 88.7 | 87.6 | -1.3 |

Notably, the first simulation aims at verifying the recycle valve controller, which modulates the recycle valves, is adequate to prevent compressor surge. This analysis focused on a low pressure compressor in order to also observe the influence of the knock-out drum on the compressor performance. The purpose of the second simulation is to observe the compressor performance when the shaft speed is reduced. Such a test is particularity of interest for the high pressure compressor due to its limited operating range. The third simulation is aimed at the evaluation of the pressure in the pipeline and well when the CO₂ stream is reduced. The results of all three simulations are presented in the next section.

4.2.1 Qualitative analysis of the dynamic simulation results

The reduction of the power plant load from full capacity to 40% of the nominal power requires 15 minutes. To determine if a recycle valve can react rapidly enough to protect a compressor, a larger flow variation is simulated at inlet of the first compressor of the compressor train (see Figure 4.2a). During this simulation the outlet pressure of the compressor is kept constant. The performance of the recycle valve and compressor during the flow variation are shown in Figure 4.2. The flow reduction starts at the inlet of the knock-out drum. Because the
Figure 4.2: (1st) Compressor response to decrease in CO₂ flow-rate: (a) input flow rate, (b) compressor and recycle valve flow rate, (c) beta line compressor, (d) surge controller PID Set-Point(SP) Process Variable(PV) Control Signal(CS), (e) inlet pressure, (f) compressor power
compressor is kept at a constant speed, the compressor will try to continue compressing the same amount of mass-flow. At this point the amount of gas leaving the knock-out drum is larger than the amount that is entering. This difference results in a pressure drop in the knock-out drum. (see Figure 4.2e). Due to this the pressure drop at constant temperature, the density of the gas entering the compressor and so the flow velocity decreases. This increases the angle of the flow entering the impeller and decreases the angle of the flow leaving the impeller into the diffuser. This results in increased frictional forces and flow separation. With a decreased inlet pressure and the increased frictional forces it is harder for the compressor to produce the required static head to overcome the outlet pressure. This increases the risk that the compressor goes into surge and is shown by the increase in beta line in Figure 4.2c. When the beta line reaches a value just above 7.5 the compressor has a 10 % margin from its surge line and the action of the anti-surge PID controller is triggered (see Figure 4.2d). Figure 4.2b shows that at the same time as the control signal increases, the recycle valve starts opening. From this point the pressure in the knock-out drum and mass-flow through the compressor (see Figure 4.2b) remain constant. The compressor power is shown in Figure 4.2f. When the recycle valve opens, the power remains constant. A small undershoot can be observed which is caused by the delay of the PID control action. This delay also causes a small overshoot in the beta line but the compressor operating point remains at a safe distance from the surge line represented by a value of the beta line of 8. It is observed that the knock-out drum has negligible effect on the compressor performance except for a small delay in mass-flow variation in the system.

The goal of the second test case is to analyse the compressor performance when the rotational speed is reduced. A 40% ramp decrease in speed is introduced at $t = 0$ (Fig. 4.3a). During the simulation the outlet pressure and inlet mass-flow of the system are kept constant. The inlet mass-flow resembles that when the power plant operates at 40% of its nominal load and is represented by $\dot{m}_{\text{system}}$ in Figure 4.3b. Due to the limited operating range of the high pressure compressor the recycle valve is open in order to keep the compressor at a minimum flow to prevent surge occurrence. When the speed is reduced the amount of recycled gas decreases (see Figure 4.3f). However, with a slower moving impeller the pressure ratio across the compressor drops. Because the outlet pressure is kept constant the inlet pressure starts to increase (see Figure 4.3d). Despite lowering the compressor speed and the reduced pressure ratio across the compressor, the minimum flow to keep the compressor from surge only drops slightly in contrary to what would be expected when looking at a compressor map. This can be explained because of the reduced volume of the gas stream at inlet of the compressor at a higher inlet pressure. This increases the angle of the gas stream entering and decreases the angle of the gas stream leaving the impeller while the same amount of mass-flow is fed to the system as the conditions before the speed reduction. To prevent surge during these conditions more mass-flow is needed. On the compressor performance map this resembles that all speed-lines are shifted to right. The power reduces only linearly during the speed reduction (see Figure 4.3e).

In the last test case a 60% stepwise decrease in the mass-flow at the inlet of the pipeline is simulated. The aim is to observe the open loop pressure response in the pipeline and the well, without a well control valve. The reservoir pressure simulated is set at 20 bar. The inlet and outlet mass-flow of the pipeline are shown in Figure 4.4a. The outlet mass-flow decreases more sharply than at the pipeline inlet. This is due to the large volume of the pipeline (see Figure 4.4c). The small pressure difference between inlet and outlet of the pipeline is due to
Figure 4.3: (6th) Compressor response to a decrease in shaft speed: (a) normalized shaft speed, (b) compressor and system mass-flow, (c) compressor beta line, (d) inlet pressure, (e) compressor power, (f) recycle valve mass-flow.
Figure 4.4: Pipeline and well response to a stepwise decrease of mass-flow at the pipeline inlet: (a) pipeline inlet and outlet mass-flow, (b) well head and reservoir mass-flow, (c) pipeline inlet and outlet pressure, (d) well head - big tubing and small tubing pressure

the frictional forces in the pipeline. These forces decrease when the mass-flow is reduced and so the difference in pressure between the inlet and outlet. Figure 4.4b shows the mass-flow at the well head an in the well. These variables show the same trend of the mass-flow at the outlet of the pipeline. This shows that the pipeline is the confining delay in the system causing the major pressure variations between in inlet of the pipeline and bottom of the well during load reductions of the power plant. Figure 4.4d shows the pressures at different depths in the well follow the same rate of decrease of the pressure in the pipeline. It takes nearly 15 hours for the pressure to reach steady state conditions in the well because of the large volumes of the pipeline and well.
Chapter 5

Results

As mentioned in Chapter 2, two design cases have been considered. In Case A, the pressure in the pipeline is a direct function of the pressure in the well. In Case B, this pressure in the pipeline is kept constant by a well control valve. For both cases, it is possible to adopt a system configuration with a single or two parallel operating compressor trains. To find the optimal configuration for the ROAD project power plant, different dynamic simulations are performed with the developed Modelica model in order to evaluate the following figures of merit: compression work, impact of control system on stripper pressure, occurrence of surge and choke in the compressors and occurrence of two phase flow in the pipeline and well.

Three different test cases are considered for Case B: (1) the compression section of the power plant consists of a single compressor train configuration, (2) the latter is equipped with a variable speed controller, (3) a configuration with two compressor trains is adopted. The same test cases have been repeated for Case A assuming three different reservoir pressures corresponding to an empty, half-full and full reservoir. In all simulations, the mass-flow of the system is reduced according to the power plant load reduction from full capacity to 40% of the nominal power. All simulations are performed on a computer having a 2.5 GHz Intel Core(i5) processor and employing the Windows 7 Enterprise operating system. The model integration is performed using the DASSL integration method, with a tolerance of 1E-4.

This chapter starts describing the simulation results obtained for a single compressor train configuration (see Section 5.1). The performance achieved with a variable speed controller is evaluated in Section 5.2, while the dynamic behaviour of a configuration with two parallel compressor trains is discussed in Section 5.3.
5.1 Performance of the configuration with one compressor train

Case B is analysed first in Section 5.1.1. In this case the compressor train outlet pressure is decoupled from the pressure in the well. The observations made for Case B, can be partially extended to the more complex Case A (see Section 5.1.2).

![Diagram of the system](image)

Figure 5.1: System PFD of the evaluated points and variables

5.1.1 Constant pressure at well head (Case B)

The simulation starts with the power plant at full load steady state condition. The simulation results are presented in Figure 5.2 and the evaluated points are illustrated in Figure 5.1. Preliminary simulations showed that a two phase flow can occur at the inlet of the last compressor when gas is recycled at minimum load of the power plant. Due to the large pressure drop across this recycle valve, a substantial decrease in temperature was observed when the valve was opened to avoid surge (Joule-Thomson effect). To prevent this effect, the temperature set-point of the inter-cooler after the 6th compressor was increased from 40°C to 70°C. To achieve the required temperature of 40°C at inlet of the pipeline an extra heat-exchanger is placed between the pipeline inlet and recycle line inlet of the 6th compressor. This was also needed for a double compressor configuration. With this adjustment the simulation results presented here where obtained.
5.1 Performance of the configuration with one compressor train

Figure 5.2: System response during mass-flow variation (Case B - single compressor train): (a) mass-flow at stripper outlet and at well control valve, (b) compressor train outlet, pipeline outlet and well head pressures, (c) compressor train inlet and stripper pressures, (d) compressor beta lines, (e) mass-flow of compressor recycle valves, (f) compressor mass-flow
A ramp-like decrease of the stripper mass-flow, starts at $t = 0$ min, while at $t = 180$ min, the mass-flow is increased again to reach the full load conditions (see Figure 5.2a). Figure 5.2a also shows the mass-flow of the gas stream through the well control valve. The mass-flow through the well control valve is slightly lower than that at the system inlet at the beginning of the simulation because of the amount of water removed in the dehydrator and knock-out drums. During minimum load operation the difference between the two streams increases. Because the temperature inside the pipeline starts to drop, consequently the density of the CO$_2$ in the pipeline starts to increase. To prevent that the pressure drops below 85 bar, the control valve at the well head reduces the amount of mass-flow injected into the well until the CO$_2$ stream mass-flow in the pipeline reaches steady state conditions.

The pressure at the pipeline inlet/outlet and well head are shown in Figure 5.2b. The control valve keeps the pipeline outlet pressure at 85 bar while the well head pressure fluctuates with the mass-flow. Figure 5.2c shows the pressure in the stripper and at the inlet of the first compressor, which are separated by the stripper control valve. The pressure at the inlet of the compressor train is slightly lower than the stripper pressure due to over-dimensioning of the compressor train. This over-dimensioning was assumed to prevent that small pressure fluctuations in the system would influence the stripper pressure.

The pressure at the inlet of the compressor train decreases during ramping down of the mass-flow. When the mass-flow reduces through the compressor train the operation point of the compressors move toward their surge lines. Because a constant compressor speed is maintained, the pressure ratio across the compressor increases to the corresponding mass flow. Because at the outlet of the compressor train a constant pressure is maintained, and at the inlet the mass-flow is throttled by the stripper control valve, the inlet pressure starts to decrease.

![Figure 5.3: Compressor train performance during mass-flow variation (Case B - single compressor train): (a) total compressor power, (b) total technical work, (c) total isentropic efficiency](image)

A reduced inlet pressure and lower mass-flow through the compressor train eventually leads to the opening of the recycle valves. The lower mass-flow leads to a smaller flow angle of the flow leaving the impeller into the diffuser and flow separation at the impeller blades. The dissipated energy due friction in the diffuser, lower inlet pressure and possible flow separation at the impeller blades increases the risk that the compressors are unable to generate enough head to overcome the outlet pressure. At this point the compressors operate closer to the surge line. This is represented by the increase in beta line for all compressors in Figure 5.2d, where beta line 8 represents the surge line. When a compressor reaches an operating point
5.1 Performance of the configuration with one compressor train

5.1.1 Performance with mass-flow variation

with a mass-flow only 10% higher than that of the surge line, its recycle valve start to open (see Figure 5.2e) and keeps the mass-flow inside the compressors constant (see Figure 5.2f) avoiding them of going into surge. The simulation results (see Figures 5.2e and 5.2d) show that the sixth compressor is the first that reaches this critical conditions. This is due to the steep speed-lines of this high pressure compressor. When the power plant load is increased again, less mass-flow recirculation is needed to avoid the compressors surge and the recycle valves start closing. When the recycle valves are fully closed, the beta lines of the compressor return to their initial operating point.

The power used by the compressor train is shown in Figure 5.3a. The power gradually decreases with a similar trend of the gas stream mass-flow, and achieves a minimum when the recycle valves keep the compressors at a minimum flow resulting in a constant power consumption. A good way to continue the evaluation of the compressor train performance is to analyse its technical work (see Figure 5.3b). Technical work is defined as the power used by the whole compressor train, divided by the mass-flow leaving the compressor train. The compressor train performance can also be evaluated by analysing the isentropic efficiency, shown in Figure 5.3c.

![Figure 5.4: System response during mass-flow variation for different reservoir pressures (Case A - single compressor train): (a) compressor train outlet pressure, (b) compressor train inlet and stripper pressures](image)

5.1.2 Variable pressure at well head (Case A)

The simulations for Case A where performed assuming a reservoir pressure of 20 bar, 150 bar and 300 bar, respectively corresponding to an empty, half full and full reservoir. Preliminary simulations showed that for each reservoir pressure the compressor train parameters needed to be adjusted to operate the compressor train within the choke and surge line during ramping up and down of the power plant. These different parameters are shown in Appendix A. This was also required for a double compressor train. With the new parameters the simulation results presented here where obtained.

The mass-flow is gradually decreased at $t=0$. As analysed in Section 4.2, the compressor train outlet pressure in the pipeline only slowly decreases (see Figure 5.4a) when the power plant is reduced to a minimum load. In contrary to the simulation of Case B, the mass-flow
is increased at $t = 720$ min to evaluate the compressor train performance during this slower reduction outlet pressure of the compressor train.

Figure 5.4b shows the compressor inlet pressures for all three simulations. During these simulations the stripper pressure is kept constant by the stripper controller and is not influenced by the compressor train performance. The figure shows that during minimum load conditions, the inlet pressures reach a sub-atmospheric state. At these pressures, the liquid water in the first knock-out drum starts to evaporate.

The slow decrease of the compressor train outlet pressure results in a lower pressure ratio across the compressor train. Compared to Case B, this reduces the amount of recycled gas needed to keep the compressors from surge. This is shown in Figure 5.5a, 5.5b and 5.5c, corresponding respectively to the case of an empty, half full and full reservoir. For the
5.2 Results with variable speed control

As explained in Section 2.3.2, the variable speed controller requires a pressure set-point to determine how much speed can be reduced. Preliminary simulations showed that the set-point of the speed controller was not optimal when set at the stripper pressure. To determine the optimal set-point, for each test case additional simulations are performed whereby the performance of the system is analysed with a reduction in the compressor train shaft speed. The starting process conditions during this simulation resemble the power plant at minimum load. The results for Case A and a reservoir pressure of 20 bar are shown in Figure 5.6.

![Figure 5.6: System response to a ramp-wise decrease of shaft speed (Case A - single compressor train): (a) normalized shaft speed, (b) total compressor power, (c) mass-flow compressor recycle valves](image)

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The simulated reduction in the normalized speed of the compressor train is shown in Figure 5.6a while figure 5.6b shows the resulting power consumption of the overall compressor train. The latter shows a minimum value that does not correspond to the end of the speed ramp. This is due to the opening of the recycle valves for the low pressure compressors at the end of the speed ramp (see Figure 5.6c). This can be explained by the following phenomena.

During the conditions at minimum load of the power plant the outlet pressure of the compressor train has dropped because of the reduced frictional forces in the well. This allows the pressure ratio across the compressor to be lowered by reducing the compressor speed which will result in a reduction of compression energy. However, because the gas stream at the inlet of the compressor train is throttled by the stripper control valve, the outlet pressure remains constant when it reached a steady state condition. This means that the accompanied loss of the pressure ratio during the reduction of speed results in an increased pressure at the inlet of the compressor train. For the 6th compressor, this increase in inlet pressure moves its operating point farther from the surge line. This is shown by the reduction in circulating gas for the 6th compressor in Figure 5.6c. For the low pressure compressor the increased inlet pressure moves their operating range to a higher mass-flow range. This is represented by its speed line moving to the right on the performance map. This can be accounted for by the reduction in gas stream velocity because the density of the gas stream increases while the compressed mass-flow in the system remains constant. This changes the flow angels in the impeller. The combination of increased friction in the diffuser, flow separation and a higher outlet pressure increases the risk of surge at the previous safe range of mass-flow. At the point of stagnation in total compressor power, the recycled gas required to keep the low pressure compressors from surge consumes more power than is saved by the reduction in compressor speed and circulated gas for the 6th compressor.

Figure 5.7: System response during mass-flow variation with and without variable speed control (Case A - single compressor train): (a) compressor train inlet and stripper pressure, (b) total compressor power, (c) compressor train shaft speed

The compressor train inlet pressure that allows the minimum power consumption is chosen as set-point for the variable speed controller. For a single compressor train and an empty reservoir this value is equal to 0.9 bar. With this set-point for the variable speed controller, a new simulation is performed in which the gas flow is reduced. This allows to compare the performance of the single compressor train configuration with variable speed, with the previous case where the speed is kept constant (see Figure 5.5). The results show that most process variables follow the same trends those exhibit without a speed controller. The main differences are shown in Figure 5.7. Notably, figure 5.7a shows that the inlet pressure with
5.3 Performance of the configuration with two compressor trains

This configuration consists of two identical compressor trains placed side by side with their outlet and inlet connected (see Figure 5.8). An extra controller is used to control the flow between the both compressor trains (see Section 2.3.2). It is assumed that at minimum power plant load, one compressor train can be gradually shut down in 15 min.

![Diagram of two compressor trains](image)

**Figure 5.8:** System PFD of evaluated points and variables for two compressor train operating in parallel

5.3.1 Constant pressure at well head (Case B)

In Figure 5.9a, the mass-flow from the stripper and the mass-flow at the inlet of both compressor trains is shown. The mass-flow is equally divided between the two trains at full load condition and during the initial decrease of mass-flow. When the flow from the stripper drops below the set-point of the flow controller, the valve of compressor train 1 is progressively closed diverting all the mass-flow to compressor train 2. Afterwards, the compressors in train 1 operate with their recycle valves fully open. When all flow is diverted to compressor train 2, compressor train 1 could be shut down. In the performed simulations this was not done to prevent numerical errors, furthermore a shut down procedure is not incorporated in the dynamic model. During the decrease of mass-flow, only a short opening of the recycle valves of compressor train 2 is observed (see Figure 5.9c). The stripper pressure remained constant during the load variation as shown in Figure 5.9b.
Figure 5.9: System response during mass-flow variation (Case B - two compressor trains in parallel): (a) massflow in the stripper control valve & the gas flow to compressor train 1 and 2, (b) pressure at compressor train inlet and in the stripper, (c) Compressor train 2 recycle valves

Figure 5.10a shows the compressor power consumed by both compressor trains. As the gas mass-flow is ramped down, the power of both trains gradually decrease till around 4 MW. From this point, the recycle valves of compressor train 1 are fully opened and the total train consumes a constant amount of power. Figure 5.10c shows that the efficiency of compressor train 2 remains nearly constant. The efficiency of compressor train 1 drops because there is no mass-flow leaving the compressor train. This is also demonstrated by the large increase of technical work in Figure 5.10b.

To evaluate the whole compression system, it is assumed that compressor train 1 is shut down when the full flow is diverted to compressor train 2. The combined total power is then shown in Figure 5.10d, while the combined technical work and isentropic efficiency are shown in Figure 5.10e and 5.10f, respectively.

5.3.2 Variable pressure at well head (Case A)

Three different reservoir pressures are simulated. The results show that the pipeline pressures have the same behaviour as in the case of a single compressor train. The flow controller acts in the same manner as in the previous test case (see Section 5.3.2). Figure 5.11 shows the total compression power, isentropic efficiency and technical work of the system with the assumption
Figure 5.10: Performance of the configuration with two compressor trains in parallel and constant pressure at the pipeline outlet: (a) compressor train power, (b) technical work, (c) isentropic efficiency of each train, (d) compressor train power, (e) technical work, (f) isentropic efficiency of the overall system.

Compressor train 1 is shut down at the power plant minimum load. During the simulations the stripper pressure remained constant.
Figure 5.11: Compressor train performance during mass-flow variation for different reservoir pressures (Case A): (a) compressor train power, (b) technical work, (c) isentropic efficiency of the overall system.
Chapter 6

Discussion of the results

This chapter discusses the different process configurations simulated with the dynamic model and if they are suited for the power plant of the ROAD project.

6.1 Evaluation of the compressor train performance

The figures of merit used to evaluate the compressor train performance are: impact on stripper pressure, occurrence of surge and choke, occurrence of two phase flow in the pipeline and overall compression work. As far as the first figure of merit is considered, simulations prove that a single compressor train and two compressor trains placed parallel can be designed such that they will have no impact on the stripper pressure.

The second figure of merit is the occurrence of surge and choke. The simulations show that the recycle valves are capable of protecting the single and double compressor train from surge, regardless which control strategy of the well is used. Due to the relative slow mass-flow variations the valves can open in a gradual manner. The time it takes for the controller to open these valves is in the order of minutes, while state-of-the-art recycle valves can be fully opened in few seconds [49]. There is a risk of short compressor choke at low reservoir pressures when the power plant ramps up from minimum load if the pipeline pressure is not controlled. However, it is assumed that this will not influence the other systems in the PCC process because of its short duration.

During the simulations, two phase flow conditions were only observed for a design with a well control valve. This occurred at the inlet of the last compressor. This was prevented by adding an extra heat exchanger before the pipeline inlet and after the recycle line inlet.

To compare the power consumption of the different compressor train configurations and well control strategies, it is assumed that the power plant operates 12 hours at minimum load, and 12 hours at full load on a daily basis. During these simulations the total consumed power by the compressor train is evaluated. Even though these calculations adopted an generalised efficiency map for all the compressors and a hypothetical load profile, it gives an idea of the
order of power required for each solution. The results are shown in Table 6.1. These results are further discussed in the next section.

### 6.2 Optimal process configuration for the ROAD project

Based on the evaluation of the figures of merit and the results, every compressor train configuration meets the required performance and guarantees a stable filling of the well. To identify the optimal solution, the advantages and disadvantages of each design solution are hereafter discussed. Furthermore, possible improvements for increasing the working range and reducing the amount of circulated gas are evaluated.

**Well control valve**

When a configuration with a single compressor train is chosen, a design without a well control valve would be more efficient during minimum load operation of the power plant. But this design will need an extra investment in new impellers for different well pressures (see Section 5.1.2). Furthermore, while two phase flow did not occur in the pipeline during simulations of the normal process conditions, it can develop during emergency shut down procedures. Special care will be needed during these procedures to prevent two phase flow from developing. In addition, safety measures should be taken to prevent the compressor from choking during ramping up of the power plant at low reservoir pressures. For instance, it could be necessary to vent the CO$_2$ at the inlet of the compressor train to the atmosphere when a compressor train is choked.

These complications can be prevented with a control valve at the well head that keeps the pipeline pressure constant. However, at low reservoir pressures there will be a significant pressure drop across the control valve. At the well head, the dense CO$_2$ will undergo a phase change in the control valve. This event will cause large dynamic vibrations which could damage the well head and pipeline valve. Therefore, before this design solution can
be applied, more investigation is needed concerning the hazards, control and additional costs caused by the occurrence of two-phase flows at the well head.

**Single and double compressor train**

Based on the power consumption calculated in Table 6.1, a double compressor train would be more advantageous during minimum load operation. However, an additional compressor train would introduce extra maintenance and capital cost. To get an idea of the costs saved with a double compressor train and the extra investments required, a preliminary calculation is performed. The capital cost for a single compressor train is estimated at 8-10 million\(^1\) Euros. A double compressor train has an estimated capital cost of 12-16 million Euros. On the basis of the results in Table 6.1, the difference in MWh between a single and double compressor train with a well control valve is compared, which has a difference of 45 MWh per day. The average cost of a MWh is estimated at 60 Euros\(^2\). The total amount of money saved annually is calculated with the following formula:

\[
\text{Money saved annually} = \text{MWh saved per day} \times \text{cost per MWh} \times 365 \text{ days} \quad (6.1)
\]

This formula leads to an annual saving of about 1 million Euros. Assuming that the reservoir will be filled within 12 years of operating the power plant, a configuration with two compressor trains parallel could save about 12 million Euros. This figure off-course strongly depends on the power plant utilization and compressor train efficiency. But based on this preliminary calculation, it seems that a double compressor train has more economic advantage. However, a more in-depth investigation of the capital/operational costs and optimisation of the compressor train efficiency is required to confirm this result. Furthermore, the amount of money saved will be lower when a design without a well control valve is chosen.

**Possible operating range improvements**

If placing extra compressor trains in parallel should prove economical disadvantageous, there are still several possibilities to extend the operating range of a single compressor train to save energy at minimum load conditions of the power plant. Possible ways of improving the operating range include using avoidance control strategies (as mentioned in Section 2.3), increasing the number of compression stages and replacing the high pressure stages with reciprocating compressors. The latter two are discussed here but future research should point out which possibility would give an economical advantage.

When variable speed or guide vane control strategies are used to control the compressor train, operating the compressors at a lower pressure ratio extends its operating range (see Figure 6.1). A drawback of this possibility is that the compressors will not operate at the operating point of their optimal efficiency. Moreover, extra compressor stages are needed to meet the required compression ratio.

\(^1\)Based on a proposition of a compressor manufacturer for a 7 stage 13.5 MW CO\(_2\) compressor train
\(^2\)Based on the statistics generated by CBS (Centraal Bureau voor Statistiek)
Another improvement could be to replace the high pressure compressors with reciprocating compressors. Reciprocating compressors have no minimum flow limitation and therefore usable in a wider operating range than the high pressure centrifugal compressors. Liebenthal et al. [5] mentioned this possibility but only analysed it as a possible solution without developing a dynamic model of the system. An illustration of the possible operating range is given in Figure 6.2. The dynamic interaction between the two compressor types during a load reduction of the power plant as well as the possible increase in capital cost, should first be investigated to determine if this solution would give an economical advantage.

Figure 6.1: Extended operating range at off-design pressure ratio of guide vane and variable speed control on normalized performance maps

Figure 6.2: Operating range of an integrally geared centrifugal compressor train combined with a reciprocating compressor [5]
Chapter 7

Conclusions and recommendations

A dynamic model for an integrally geared centrifugal compressor train of a MEA based PCC process has been successfully developed. The various simulations performed with this model give more insights in the performance of the compressor train and its impact on the performance of the capture system and the CO\textsubscript{2} pipeline and storage well. The obtained results lead to the conclusions of Section 7.1 and recommendations for further research of Section 7.2.

7.1 Conclusions

The conclusions are here presented in relation to the objectives defined in Section 1.3.

1. Identify and implement a control strategy for the compressor train

Based on the preliminary analyses and literature review reported in Chapter 2, a speed controller combined with use of recycle valves seems the most suited solution for the compressor train to extend its operating range during minimum load of the power plant. Chapter 5 shows that the recycle valves are able to avoid the occurrence of surge during the load reductions, while the use of a speed controller is only effective for low reservoir pressures and when the pipeline pressure is not kept constant by a control valve situated at the well head.

2 Analyse the dynamic performance of the system with and without a well control valve

Without a well control valve and at low reservoir pressures, the compressor train risks choking when the gas stream is ramped up from minimum load conditions of the power plant. Moreover, different impeller dimensions are needed to meet the outlet pressure for the different reservoir pressures in this case. With a well control valve, an extra heat-exchanger is needed to prevent two-phase flow at the inlet of the last compressor.
3 Analyse the dynamic performance of a single and double compressor train configuration

For a double compressor train, the required power during minimum load operation can be reduced by diverting all the gas mass-flow to a single compressor train without further complications for the Post Combustion Capture (PCC) process. Without a well control valve, at low and medium reservoir pressures, this double configuration consumes less energy at minimum load conditions of the power plant than when the pipeline pressure is kept constant. However for a double configuration similar process issues occur as for a single configuration as mentioned above for designs with and without a well control valve.

4 Analyse the impact of the compression process on the stripper pressure

The simulations show that all process configurations evaluated had no influence on the stripper pressure.

5 Select the best suited process configuration for the PCC system of the ROAD project

A double compressor train uses less energy during minimum load operation of the power plant. The energy reduction it achieves may compensate for the higher capital and operation costs. However, this strongly depends on the power plant utilization. Design solutions with and without a well control valve have different limitations that need further investigation before an optimal design can be proposed for the power plant of the ROAD project.

7.2 Recommendations

As stated in the conclusions, further research is needed to propose an optimal design for the ROAD project. Notably, two future research directions can be distinguished. The first direction can be focused on extending the developed dynamic model with the capture unit model developed at TNO. This allows for analysing the impact on the capture unit during off-design scenarios of the compressor train (like start-up, shut-down and emergency scenarios) as well as heat integration solutions between the models. The second direction should focus on the technology gaps that make the integration of Monoethanolamine (MEA)-based PCC processes with offshore storage challenging.

7.2.1 Further research – Dynamic model

Integration with stripper model The prediction of the thermodynamic properties in the stripper model is carried out by using empirical correlations that have a limited validated range. This makes the model prone to errors during initialisation. To address such a numerical problem very specific initialisation conditions should be found or the validated range of the correlations should be extended. Alternatively, a different modelling approach can be used.
**Heat integration** Before heat integration between the compressor train and the capture unit can be studied, the inter-cooler model should be improved. The current component model neglects the dynamics on the shell side of the heat exchanger. Also the efficiency maps of the compressors should be improved and validated for the specific process conditions assumed.

**Start-up, shut-down and emergency scenarios** During these operations, gas mass-flows in the compression section can have large fluctuations in the order of a few seconds. The surge control systems used to open the recycle valves should be extended with an anti-surge algorithm. This algorithm detects fluctuations that cannot be controlled with the normal control system and sends a signal to fully open the recycle valve instantaneously.

### 7.2.2 Further research – Technology gaps

Before an optimal design can be proposed for the power plant of the ROAD project, a techno-economic analyses is required. This study should aim at evaluating the plant utilization and optimising the compressor train efficiency, capital and operational costs. The following technology gaps should be included:

**Compressor control** Variable speed control and guide vane positioning can extend the working range of a compressor train by operating the compressors at a lower pressure ratio than at their original design point. This will reduce the amount of recycled gas needed during part-load operation, however will also reduce the compressor efficiency during full-load operation. Moreover, extra compressor stages are needed to meet the required compression ratio.

**Compressor type** By replacing the high pressure centrifugal compressor with a reciprocating compressor, the working range of the compressor train can be increased. This would possibly increase the efficiency of a single compressor train during minimum load of the power plant. Furthermore the development of shock-wave compressors should be monitored in future research as it can prove to be a cost-efficient replacement of the integrally geared centrifugal compressor.

**Transportation process** Because of the large pressure drop across the well control valve, an additional investigation is needed to assess if the occurrence of two phase can damage the pipeline and well head. Also the possibility of transporting the CO\textsubscript{2} by boat should be investigated. This option allows for CO\textsubscript{2} liquefaction on land, which replaces the high pressure compressors by pumps. This would give the compression train a larger operating range and further complications in the pipeline are prevented. Moreover, the utilization of the capture plant is guaranteed also when the reservoir is filled without extra investments in new pipelines or the compressor train. However, the costs of such a solution and the energy requirements to liquefy the CO\textsubscript{2} stream should be compared with the pipeline solution in future research.
Appendix A

Component parameters and compressor maps

A.1 Component parameters

Table A.1: Knock-out drum parameters for a single train

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Table A.2: Inter-cooler parameters for a single train

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<td>44,5</td>
<td>104</td>
<td>0,19</td>
</tr>
<tr>
<td>Comp 6</td>
<td>50,7</td>
<td>38</td>
<td>38</td>
<td>104</td>
<td>0,14</td>
</tr>
</tbody>
</table>
## Table A.4: Compressor parameters for a double train

<table>
<thead>
<tr>
<th>Case A [P_{res} = 20 \text{ bar}]</th>
<th>(P_d[\text{bar}])</th>
<th>(T_d[\degree C])</th>
<th>(m_d[\text{kg/s}])</th>
<th>(N_T[%])</th>
<th>(D[\text{m}])</th>
</tr>
</thead>
<tbody>
<tr>
<td>Comp 1</td>
<td>1,6</td>
<td>38</td>
<td>24,3</td>
<td>100</td>
<td>0,61</td>
</tr>
<tr>
<td>Comp 2</td>
<td>3,2</td>
<td>38</td>
<td>23,6</td>
<td>100</td>
<td>0,42</td>
</tr>
<tr>
<td>Comp 3</td>
<td>6,3</td>
<td>38</td>
<td>22,3</td>
<td>103</td>
<td>0,29</td>
</tr>
<tr>
<td>Comp 4</td>
<td>12,7</td>
<td>38</td>
<td>22,2</td>
<td>103</td>
<td>0,20</td>
</tr>
<tr>
<td>Comp 5</td>
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<td>38</td>
<td>22,3</td>
<td>104</td>
<td>0,14</td>
</tr>
<tr>
<td>Comp 6</td>
<td>50,7</td>
<td>38</td>
<td>22,5</td>
<td>98</td>
<td>0,11</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Case A [P_{res} = 150 \text{ bar}]</th>
<th>(P_d[\text{bar}])</th>
<th>(T_d[\degree C])</th>
<th>(m_d[\text{kg/s}])</th>
<th>(N_T[%])</th>
<th>(D[\text{m}])</th>
</tr>
</thead>
<tbody>
<tr>
<td>Comp 1</td>
<td>1,6</td>
<td>38</td>
<td>24,3</td>
<td>100</td>
<td>0,61</td>
</tr>
<tr>
<td>Comp 2</td>
<td>3,2</td>
<td>38</td>
<td>23,6</td>
<td>101</td>
<td>0,42</td>
</tr>
<tr>
<td>Comp 3</td>
<td>6,3</td>
<td>38</td>
<td>22,3</td>
<td>103</td>
<td>0,29</td>
</tr>
<tr>
<td>Comp 4</td>
<td>12,7</td>
<td>38</td>
<td>22,2</td>
<td>103</td>
<td>0,20</td>
</tr>
<tr>
<td>Comp 5</td>
<td>25,3</td>
<td>38</td>
<td>21,4</td>
<td>110</td>
<td>0,13</td>
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<tr>
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<td>38</td>
<td>18,5</td>
<td>106</td>
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</table>

<table>
<thead>
<tr>
<th>Case A [P_{res} = 300 \text{ bar}]</th>
<th>(P_d[\text{bar}])</th>
<th>(T_d[\degree C])</th>
<th>(m_d[\text{kg/s}])</th>
<th>(N_T[%])</th>
<th>(D[\text{m}])</th>
</tr>
</thead>
<tbody>
<tr>
<td>Comp 1</td>
<td>1,6</td>
<td>38</td>
<td>23,3</td>
<td>108</td>
<td>0,60</td>
</tr>
<tr>
<td>Comp 2</td>
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<td>21,6</td>
<td>109</td>
<td>0,39</td>
</tr>
<tr>
<td>Comp 3</td>
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<td>20,3</td>
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<td>Comp 4</td>
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<td>38</td>
<td>20,2</td>
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<td>0,16</td>
</tr>
<tr>
<td>Comp 5</td>
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<td>38</td>
<td>22,4</td>
<td>100</td>
<td>0,11</td>
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<tr>
<td>Comp 6</td>
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<td>38</td>
<td>15,5</td>
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<td>0,07</td>
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</table>

<table>
<thead>
<tr>
<th>Case B</th>
<th>(P_d[\text{bar}])</th>
<th>(T_d[\degree C])</th>
<th>(m_d[\text{kg/s}])</th>
<th>(N_T[%])</th>
<th>(D[\text{m}])</th>
</tr>
</thead>
<tbody>
<tr>
<td>Comp 1</td>
<td>1,6</td>
<td>38</td>
<td>24,3</td>
<td>100</td>
<td>0,61</td>
</tr>
<tr>
<td>Comp 2</td>
<td>3,2</td>
<td>38</td>
<td>23,6</td>
<td>101</td>
<td>0,42</td>
</tr>
<tr>
<td>Comp 3</td>
<td>6,3</td>
<td>38</td>
<td>22,3</td>
<td>103</td>
<td>0,29</td>
</tr>
<tr>
<td>Comp 4</td>
<td>12,7</td>
<td>38</td>
<td>22,2</td>
<td>103</td>
<td>0,20</td>
</tr>
<tr>
<td>Comp 5</td>
<td>25,4</td>
<td>38</td>
<td>22,3</td>
<td>104</td>
<td>0,14</td>
</tr>
<tr>
<td>Comp 6</td>
<td>50,7</td>
<td>38</td>
<td>18,5</td>
<td>104</td>
<td>0,10</td>
</tr>
</tbody>
</table>
Figure A.1: Compressor 1 original design point: $P = 2.2\text{bar}$, $T = 31.85^\circ\text{C}$, $m = 96.72\text{kg/s}$, $D = 1.03\text{m}$, $\omega = 5534\text{rad/s}$

Figure A.2: Compressor 2: Original design: $P = 4.48\text{bar}$, $T = 31.85^\circ\text{C}$, $m = 96.72\text{kg/s}$, $D = 0.72\text{m}$, $\omega = 7857\text{rad/s}$

Figure A.3: Compressor 3: Original design: $P = 9.0\text{bar}$, $T = 31.85^\circ\text{C}$, $m = 96.72\text{kg/s}$, $D = 0.50\text{m}$, $\omega = 11158\text{rad/s}$

Figure A.4: Compressor 4: Original design: $P = 18.2\text{bar}$, $T = 31.85^\circ\text{C}$, $m = 96.72\text{kg/s}$, $D = 0.34\text{m}$, $\omega = 15825\text{rad/s}$

Figure A.5: Compressor 5: Original design: $P = 36.9\text{bar}$, $T = 31.85^\circ\text{C}$, $m = 96.72\text{kg/s}$, $D = 0.23\text{m}$, $\omega = 21934\text{rad/s}$

Figure A.6: Compressor 6: Original design: $P = 74.5\text{bar}$, $T = 94.85^\circ\text{C}$, $m = 96.72\text{kg/s}$, $D = 0.20\text{m}$, $\omega = 19178\text{rad/s}$
Appendix B

Documentation of the component models

B.1 Compressor

![Compressor icon from the ThermoPower library](image)

**Figure B.1:** Compressor icon from the ThermoPower library

Component description

The compressor component describes the conversion of mechanical energy into static pressure. The compressor performance is calculated based on tabulated performance maps and its boundary conditions. The compressor component is a module from the ThermalPower library and is adapted for scaling. For every compressor stage in the compressor train a compressor component with a specific performance map is used (see Appendix A).

Relevant phenomena

- Conversion of mechanical energy into kinetic energy by the impeller blade
- Conversion of kinetic energy into static pressure in the diffuser
Hypothesis and assumptions

- Change in potential energy is negligible
- No accumulation of energy and mass
- No reactions take place
- Perfect thermal insulation (adiabatic process)
- Turbulent 1D flow
- No occurrence of surge and stall
- Impeller momentum can be neglected

Sub Models

The compressor is modelled as a single fluid resistive module.

Conservation equations

The conservation of mass is formulated as

\[
\frac{dM}{dt} = \dot{m}_{in} - \dot{m}_{out} \tag{B.1}
\]

Without mass accumulation

\[
\dot{m} = \dot{m}_{in} = \dot{m}_{out} \tag{B.2}
\]

The conservation of energy is formulated as

\[
\frac{dU}{dt} = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} + \dot{Q} - \dot{W}_S - P\frac{dV}{dt} \tag{B.3}
\]

Without energy accumulation, adiabatic process and constant volume the equation reduces to

\[
\dot{W}_S = \dot{m}(h_{in} - h_{out}) \tag{B.4}
\]

The conservation of momentum is formulated as

\[
\frac{dG}{dt} = \dot{m}_{in}v_{in} - \dot{m}_{out}v_{in} + (S_{in}P_{in} - S_{out}P_{out} - F_{ff}) + S\rho g(z_{in} - z_{out}) \tag{B.5}
\]

Without momentum accumulation and negligible change in potential energy the formula is written as

\[
\dot{m}(v_{in} - v_{out}) = S_{in}P_{in} - S_{out}P_{out} - F_{ff} \tag{B.6}
\]

The momentum formula is replaced by the performance maps which lead to

\[
\dot{m} = f(P_{in}, P_{out}, N_T) \tag{B.7}
\]
Constitutive equations

In Section 3.1.5 is explained how the performance maps are fitted for numerical modelling. This method of modelling reduces the performance characteristics of the compressor to three tables. The first two tables determine the normalized flow and compressor efficiency based on the beta line and referred speed.

\[ \phi_c = f(\beta, N_T) \] (B.8)

\[ \eta = f(\beta, N_T) \] (B.9)

The beta line is determined by the third table and is a function of the pressure ratio and referred speed.

\[ \beta = f(\Psi, N_T) \] (B.10)

The pressure ratio is formulated as

\[ \Psi = \frac{P_{out}}{P_{in}} \] (B.11)

To gain the mass-flow and scale the compressor to the right process conditions the Buckingham II theorem is applied (see Section 3.1.5). This leads to the following formula

\[ \dot{m} = \phi_c P_{in} \sqrt{T_{in}} \left( \frac{\dot{m}_d \sqrt{T_d}}{P_d} \right) \] (B.12)

where

\[ T_{in} = f(P_{in}, h_{in}) \] (B.13)

For the high pressure compressors with the SW EoS, the scaling equations are adapted for non-ideal behaviour. This only effects the manner of calculating the mass-flow and is given by

\[ \dot{m} = \phi_c a_{in} \rho_{in} \left( \frac{\dot{m}_d}{a_d \rho_d} \right) \] (B.14)

where

\[ a_{in} = f(P_{in}, h_{in}) \] (B.15)

\[ \rho_{in} = f(P_{in}, h_{in}) \] (B.16)

\[ a_d = f(P_d, T_d) \] (B.17)

\[ \rho_d = f(P_d, T_d) \] (B.18)
The torque $T_q$ follows from

$$T_q = \frac{1}{\eta_m} \frac{W_S}{\omega}$$  \hspace{1cm} (B.19)

The energy equation can then be written as

$$\dot{m} (h_{\text{out}} - h_{\text{in}}) = \frac{1}{\eta_m} T_q \omega$$  \hspace{1cm} (B.20)

The outgoing enthalpy is calculated with

$$h_{\text{out}} = h_{\text{in}} + \frac{h_s - h_{\text{in}}}{\eta}$$  \hspace{1cm} (B.21)

where

$$h_s = f(P_{\text{out}}, P_{\text{in}}, h_{\text{in}})$$  \hspace{1cm} (B.22)

The referred speed is given by

$$N_T = \frac{\omega}{\omega_d}$$  \hspace{1cm} (B.23)

### Model input and output variables and parameters

| Input variables | $P_{\text{in}}, P_{\text{out}}, h_{\text{in}}, \omega$ |
| Output variables | $\dot{m}_{\text{in}}, \dot{m}_{\text{out}}, h_{\text{out}}, T_q$ |
| Parameters | $\dot{m}_d, T_d, P_d, \eta_m, \omega_d$ |

#### B.2 Knock-out drum

![Knock-out drum icon](image)

**Figure B.2:** Knock-out drum icon
Component description

In a knock-out the condensed water from the feed stream is removed. The gravitational pull is used to settle condensed liquid at the bottom. The liquid level is controlled by a control valve at the bottom of the drum.

Relevant phenomena

- Accumulation of mass and energy in the gas and liquid phase
- Mass transfer between the gas and liquid phase
- Hold up of liquid
- Energy release due to condensation of H_2O and dissolution of CO_2

Hypothesis and assumptions

- CO_2 fraction in liquid is neglected
- Energy accumulation is small and can be neglected
- Ideal gas and liquid behaviour
- No accumulation momentum
- Joule Thomson effect of entering recycle stream is neglected
- Constant heat of evaporation for H_2O
- Condensation takes place before the gas is entering the knock-out drum
- No reactions take place
- Perfect thermal insulation (adiabatic process)

Model equations

Conservation equations

The mass conservation equation for the knock-out drum is given by

\[
\frac{dM}{dt} = \dot{m}_{in1} + \dot{m}_{in2} - \dot{m}_{vap} - \dot{m}_{liq} \tag{B.24}
\]

This is combined with the component balance

\[
\frac{dM}{dt} \cdot X + M \cdot \frac{dX}{dt} = \dot{m}_{in1} X_{in1} + \dot{m}_{in2} X_{in2} - \dot{m}_{liq} x_{Mass} - \dot{m}_{vap} y_{Mass} \tag{B.25}
\]

The conservation of energy equation is given by

\[
\frac{dU}{dt} = \dot{m}_{in1} h_{in1} + \dot{m}_{in2} h_{in2} - \dot{m}_{vap} h_{vap} - \dot{m}_{liq} h_{liq} + \dot{Q} - \dot{W}_S - P \frac{dV}{dt} \tag{B.26}
\]

There is no work done and the volume remains constant. This transforms the equation into

\[
\frac{dU}{dt} = \dot{m}_{in1} h_{in1} + \dot{m}_{in2} h_{in2} - \dot{m}_{vap} h_{vap} - \dot{m}_{liq} h_{liq} + \dot{Q} \tag{B.27}
\]
The conservation of momentum is given by
\[
\frac{dG}{dt} = \dot{m}_{in}v_{in} - \dot{m}_{out}v_{in} + (S_{in}P_{in} - S_{out}P_{out} - F_f) + S\rho g (z_{in} - z_{out})
\] (B.28)

With the assumptions made it reduces to
\[
P = P_{in1} = P_{in2} = P_{iq} = P_{vap}
\] (B.29)

**Constitutive equations**

To determine the vapour liquid equilibrium the Raoult’s and the Henry’s law are used, respectively given by
\[
\frac{P_{sat}}{P} = \frac{y_{H_2O}}{x_{H_2O}}
\] (B.30)
\[
P_{CO_2} = H \cdot x_{CO_2}
\] (B.31)

The Henry’s constant for low pressures is correlated from Ref. [] and given by:
\[
H = (4.615T + 61.538) \cdot 1e5
\] (B.32)

The saturated water pressure is a function of the temperature
\[
P_{sat} = f(T)
\] (B.33)

The mole fractions in the vapour and liquid phase relate by:
\[
y_{CO_2} + y_{H_2O} = 1
\] (B.34)
\[
x_{CO_2} + x_{H_2O} = 1
\] (B.35)

The partial pressure of CO\(_2\) is given by
\[
\frac{P_{CO_2}}{P} = y_{CO_2}
\] (B.36)

The liquid level is calculated with
\[
LL = \frac{V_{iq}}{\pi * (D/2)^2}
\] (B.37)

The liquid volume is obtained from
\[
M = \rho_{vap} \cdot V_{vap} + \rho_{iq} \cdot V_{iq}
\] (B.38)

Where
\[
V = V_{vap} + V_{iq}
\] (B.39)

The CO\(_2\) in the liquid is neglected so the density of the liquid is a function of the temperature and pressure of pure H\(_2\)O
\[
\rho_{iq} = f(P, T)_{H_2O}
\] (B.40)
The density of the vapour is calculated with the ideal gas law

\[ \rho_{\text{vap}} = \frac{P \cdot \text{MM}_{\text{vap}}}{R \cdot T} \]  

(B.41)

The mass balance between the liquid and vapour is given by

\[ M_{\text{vap}} \cdot y_{\text{H}_2\text{O}} = M_{\text{H}_2\text{O}} - M_{\text{liq}} \cdot x_{\text{H}_2\text{O}} \]  

(B.42)

In the inter-cooler component only a gas phase can be calculated, which neglects the heat of condensation. For correcting this error the assumption is made that the water is condensed between the inter-cooler and knock-out drum. To calculate the correct inlet enthalpy in this case, the following formula is used for both inlets

\[ h_{\text{in,real}} = q_v \cdot h_{\text{vap, in}} + (1 - q_v) \cdot h_{\text{liq, in}} \]  

(B.43)

The inlet vapour quality is calculated with the inlet mass fractions. These are calculated with the Raoult’s law based on the inlet conditions.

\[ X_{\text{in,H}_2\text{O}} = q_v \cdot y_{\text{in,H}_2\text{O}} + (1 - q_v) \cdot x_{\text{in,H}_2\text{O}} \]  

(B.44)

The inlet vapour enthalpy is a function of the inlet temperature and inlet composition.

\[ h_{\text{vap, in}} = f(P, T_{\text{in}}, y_{\text{in}}) \]  

(B.45)

The liquid enthalpy is calculated based on the amount of condensed water and dissolved CO\(_2\). For the CO\(_2\) the specific heat is used to calculated its part of the liquid enthalpy

\[ h_{\text{liq, in}} = x_{\text{in,H}_2\text{O}} \cdot h_{\text{in,H}_2\text{O}} + x_{\text{in,CO}_2} \cdot C_{\text{p,CO}_2} \cdot T_{\text{in}} \]  

(B.46)

Where

\[ h_{\text{in,H}_2\text{O}} = f(P, T) \]  

(B.47)

The inlet temperature from the inter-cooler is used as isothermal temperate along the knock-out drum. Given by

\[ T = f(P, h_{\text{in}}, X_{\text{in}}) \]  

(B.48)

The total energy in the drum is given by

\[ U = M_{\text{vap}} u_{\text{vap}} + M_{\text{liq}} u_{\text{liq}} \]  

(B.49)

\[ u_{\text{liq}} = f(P, T) \]  

(B.50)

\[ u_{\text{vap}} = f(P, T) \]  

(B.51)

The evaporated water is calculated by

\[ \dot{m}_{\text{evap}} = \frac{y_{\text{H}_2\text{O}} \cdot M_{\text{vap}}}{dt} - q_{\text{vap, in}} m_{\text{in}} \cdot y_{\text{in,H}_2\text{O}} - q_{\text{vap, in2}} \cdot m_{\text{in2}} \cdot y_{\text{in2,H}_2\text{O}} + m_{\text{out}} y_{\text{H}_2\text{O}} \]  

(B.52)

The total heat of condensation is calculated with

\[ Q_{\text{cooler}} = m_{\text{in}} \left[(h_{\text{in,ideal}} - h_{\text{in1}}) - (1 - q_{\text{vap, in}})(x_{\text{in,H}_2\text{O}} \cdot H_{\text{vap}})\right] \]  

(B.53)
Model input and output variables and parameters

<table>
<thead>
<tr>
<th>Input variables</th>
<th>$h_{in1}, \dot{m}<em>{in1}, X</em>{in}, h_{in2}, \dot{m}<em>{in2}, X</em>{in2}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output variables</td>
<td>$h_{out}, P_{in}, P_{out}, X_{out}, m_{bottom}, h_{bottom}, P_{bottom}$</td>
</tr>
<tr>
<td>Parameters</td>
<td>$C_{pCO_2}, V, H_{vap}$</td>
</tr>
</tbody>
</table>

B.3 Valve

![Valve icon from the ThermalPower library](image)

Figure B.3: Valve icon from the ThermalPower library

Component description

The valve component calculates a mass-flow based on the provided valve opening and pressure drop. The component is taken from the ThermalPower library. Different flow characteristics can be chosen for the flow coefficient i.e.: linear, quadratic, equal percentage.

Relevant phenomena

- Flow is a function of the pressure difference over the valve and valve opening

Hypothesis and assumptions

- Perfect thermal insulation (adiabatic process)
- No work
- No accumulation of energy, mass or momentum
- Turbulent 1D flow
- No molecule reactions

Model equations

Conservation equations. The conservation of mass equation is given by

$$\frac{dM}{dt} = \dot{m}_{in} - \dot{m}_{out}$$  \hspace{1cm} (B.54)

There is assumed no mass accumulation which gives

$$\dot{m} = \dot{m}_{in} = \dot{m}_{out}$$  \hspace{1cm} (B.55)
The conservation of energy equation is given by
\[
\frac{dU}{dt} = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} + \dot{Q} - \dot{W}_S - P \frac{dV}{dt} \tag{B.56}
\]
With the assumptions made it reduces to
\[
0 = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} \tag{B.57}
\]
The conservation of momentum is given by
\[
\frac{dG}{dt} = \dot{m}_{in}v_{in} - \dot{m}_{out}v_{in} + (S_{in}P_{in} - S_{out}P_{out} - F_{ff}) + S \rho g (z_{in} - z_{out}) \tag{B.58}
\]
There is assumed no momentum accumulation and negligible change in potential energy and cross-section.
\[
F_{ff} = S (P_{in} - P_{out}) \tag{B.59}
\]

**Constitutive equations**  
The Darcy-Weisbach law for pressure loss is given by
\[
\delta P = \xi \cdot \rho \cdot \frac{L}{D} \cdot \frac{v^2}{2} \tag{B.60}
\]
Where \( \xi \) is the dimensionless Darcy-friction coefficient given by
\[
\xi = f(Re) \tag{B.61}
\]
The friction force is than given by
\[
F_{ff} = S\delta P = S\xi \cdot \rho \cdot \frac{L}{D} \cdot \frac{v^2}{2} \tag{B.62}
\]
The velocity is given by
\[
v = \frac{\dot{m}}{\rho S} \tag{B.63}
\]
Substituting B.63 in B.62 and rearranging it to the mass-flow gives
\[
\dot{m} = \sqrt{\frac{2\rho S^2}{C} \delta P} \tag{B.64}
\]
All constants can now be replaced by a flow coefficient \( K_v \), and when a valve opening is introduced it can be written as
\[
\dot{m} = f(\theta)K_v \sqrt{\rho \delta P} \tag{B.65}
\]
The valve opening \( \theta \) depends on the chosen flow characteristic.
Model input and output variables and parameters

<table>
<thead>
<tr>
<th>Input variables</th>
<th>Output variables</th>
<th>Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{in}$, $P_{out}$, $h_{in}$, $\theta$</td>
<td>$\dot{m}<em>{in}$, $\dot{m}</em>{out}$, $h_{out}$</td>
<td>$K_v$</td>
</tr>
</tbody>
</table>

B.4 Inter-cooler

![Inter-cooler model with icons from the ThermalPower library](image)

Figure B.4: Inter-cooler model with icons from the ThermalPower library

Component description

The inter-cooler model cools the gas to a desired temperature. Based on its outlet temperature a PID controller controls the amount of heat subtracted from the gas-flow. For the inter-cooler model different components are taken from the ThermoPower library.

Relevant phenomena

- Heat transfer from gas (hot side) through tube wall to liquid (cold side)
- Accumulation of mass and thermal energy
- Frictional losses

Hypothesis and assumptions

- Cold side is neglected
- No accumulation of energy in metal parts
- Ideally mixed fluid
- Single phase flow
- No change in potential energy
- Convective heat transfer
- No shaft work
- Turbulent 1D flow
Sub Models

The model consists of a fluid storage module connected to a heat resistive module with controllable heat sink.

Model equations

Conservation equations

The conservation of mass equation is given by

$$\frac{dM}{dt} = \dot{m}_{in} - \dot{m}_{out}$$  \hspace{1cm} (B.66)

Because there is a fixed volume this can be rewritten as

$$\frac{d\rho}{dt} = \frac{1}{V} (\dot{m}_{in} - \dot{m}_{out})$$  \hspace{1cm} (B.67)

The conservation of energy equation is given by

$$\frac{dU}{dt} = \dot{m}_{in} h_{in} - \dot{m}_{out} h_{out} + \dot{Q} - \dot{W}_S - P \frac{dV}{dt}$$  \hspace{1cm} (B.68)

There will be no work done and the volume remains constant which reduces the equation to

$$\frac{dU}{dt} = \dot{m}_{in} h_{in} - \dot{m}_{out} h_{out} + \dot{Q}$$  \hspace{1cm} (B.69)

The conservation of momentum is given by

$$\frac{dG}{dt} = \dot{m}_{in} v_{in} - \dot{m}_{out} v_{in} + (S_{in} P_{in} - S_{out} P_{out} - F_{ff}) + S \rho g (z_{in} - z_{out})$$  \hspace{1cm} (B.70)

Which reduces to

$$P_{in} = P_{out}$$  \hspace{1cm} (B.71)

Constitutive equations

For a single storage module the total mass is determined with

$$M = \rho \cdot V$$  \hspace{1cm} (B.72)

The total internal energy is calculated with

$$U = M \cdot u$$  \hspace{1cm} (B.73)

Where $\rho$ and $u$ are functions of

$$\rho = f(P, T)$$  \hspace{1cm} (B.74)

$$u = f(P, T)$$  \hspace{1cm} (B.75)

When heat is subtracted from the storage module $Q$ is determined by the thermal port

$$q_i \cdot A = -Q$$  \hspace{1cm} (B.76)

And $q_i$ by the thermal source

$$q_i = -\frac{Q_s}{L \cdot \dot{N}_t}$$  \hspace{1cm} (B.77)
Model input and output variables and parameters

<table>
<thead>
<tr>
<th>Input variables</th>
<th>$Q_s, m_{in}, m_{out}, h_{in}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output variables</td>
<td>$P_{in}, P_{out}, h_{out}$</td>
</tr>
<tr>
<td>Parameters</td>
<td>$V, N_t, L, A$</td>
</tr>
</tbody>
</table>

B.5 Pipeline

![Pipeline section model with icons from the ThermalPower library](image)

**Figure B.5**: Pipeline section model with icons from the ThermalPower library

Component description

Relevant phenomena

- Heat transfer to surrounding
- Accumulation of mass and thermal energy
- Frictional losses

Hypothesis and assumptions

- All frictional forces are summed in one friction coefficient per section
- No work
- Inviscid barotropic flow
- No accumulation of momentum
- Turbulent 1D flow

Sub Models

The pipeline model consists of several components from the ThermalPower library. A fluid storage module and a fluid resistive model for the volume of the pipeline and frictional forces. Thermal resistive modules are connected to the fluid storage module to account for the heat loss to the surrounding. The fluid storage module and thermal resistive module are similar to
the inter-cooler and therefore not described here. The pressure drop component has a similar equation derivation as the valve component without the valve opening variable.

**Model input and output variables and parameters**

<table>
<thead>
<tr>
<th>Input variables</th>
<th>$Q_s, m_{in}, h_{in}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output variables</td>
<td>$P_{in}, m_{out}, h_{out}$</td>
</tr>
<tr>
<td>Parameters</td>
<td>$T_{sur}, V, C_f_{avg}, N_t, L, A$</td>
</tr>
</tbody>
</table>

**B.6 Well**

![Well model with icons from the ThermalPower library](image)

**Figure B.6:** Well model with icons from the ThermalPower library

**Component description**

The well model describes the pressure and temperature dynamics in the well when it is filled with CO$_2$. 
Relevant phenomena

- Heat transfer to surrounding
- Accumulation of mass and thermal energy
- Frictional losses

Hypothesis and assumptions

- All frictional forces are summed in one friction coefficient per section
- No work
- Inviscid barotropic flow
- No accumulation of momentum
- Turbulent 1D flow

Sub Models

The well is divided in different sections. (1) Big tubing, (2) small tubing, (3) bottom hole, (4) reservoir. Each section consists of a fluid storage module to account for the volume and a resistive module for the frictional forces. In the resistive modules also the pressure increase due to gravity is calculated. To gain a good model accuracy the heat loss to the soil is calculated in the resistive module. This module is newly created and the equations are derived below. The fluid storage components is taken from the ThermalPower library.

Model equations resistive module well

Conservation equations

The conservation of mass is formulated as

\[ \frac{dM}{dt} = \dot{m}_{in} - \dot{m}_{out} \]  \hspace{1cm} (B.78)

Without mass accumulation

\[ \dot{m} = \dot{m}_{in} = \dot{m}_{out} \]  \hspace{1cm} (B.79)

The conservation of energy is formulated as

\[ \frac{dU}{dt} = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} + \dot{Q} - \dot{W}_{S} - P \frac{dV}{dt} \]  \hspace{1cm} (B.80)

With the assumptions made it is rewritten as the Bernoulli equation

\[ \frac{1}{2} \rho_{in} \dot{v}_{in}^2 + \rho_{in} g z_{in} + P_{in} = \frac{1}{2} \rho_{out} \dot{v}_{out}^2 + \rho_{out} g z_{out} + P_{out} + Q \]  \hspace{1cm} (B.81)

The conservation of momentum is formulated as

\[ \frac{dG}{dt} = \dot{m}_{in} \dot{v}_{in} - \dot{m}_{out} \dot{v}_{in} + (S_{in} P_{in} - S_{out} P_{out} - F_{ff}) + S \rho g (z_{in} - z_{out}) \]  \hspace{1cm} (B.82)

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Without momentum accumulation and no change in pipeline diameter

\[ \dot{m}v_{in} + SP_{in} + S\rho_{in}z_{in} = \dot{m}v_{out} + SP_{out} + S\rho_{out}gz_{out} + F_{ff} \]  

(B.83)

**Constitutive equations** The inlet and outlet velocities are given as

\[ v_{in} = \frac{m}{\rho_{in} A} \]  

(B.84)

\[ v_{out} = \frac{m}{\rho_{out} A} \]  

(B.85)

Where

\[ S = A = \pi(D/2)^2 \]  

(B.86)

The densities are a function of

\[ \rho_{in} = f(P_{in}, h_{in}) \]  

(B.87)

\[ \rho_{out} = f(P_{out}, h_{out}) \]  

(B.88)

The heat loss to the surface is given as

\[ Q = UA_{outer}(T_{soil} - T_{avg}) \]  

(B.89)

Where the outer surface of the pipeline is given by

\[ A_{outer} = 2\pi D L \]  

(B.90)

The average soil temperature is given by

\[ T_{soil} = \left(\frac{T_{bottom} - T_{top}}{z_{out}}\right) z_{mid} + T_{top} \]  

(B.91)

The average temperature of the CO\(_2\) is given by

\[ T_{avg} = \frac{T_{in} - T_{out}}{2} \]  

(B.92)

The frictional forces are calculated with

\[ F_{ff} = \tau_{avg}A_{outer} \]  

(B.93)

Where

\[ \tau_{avg} = \frac{cf_{avg}}{6} \rho_{avg} v_{avg} \]  

(B.94)

The average velocity is given by

\[ v_{avg} = \frac{v_{in}^2 + v_{out}^2 - v_{in}v_{out}}{2} \]  

(B.95)

The average density is given by

\[ \rho_{avg} = \frac{\rho_{in} + \rho_{out}}{2} \]  

(B.96)
Model input and output variables and parameters

<table>
<thead>
<tr>
<th>Input variables</th>
<th>$P_{in}, P_{out}, h_{in}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output variables</td>
<td>$\dot{m}<em>{in}, \dot{m}</em>{out}, h_{out}$</td>
</tr>
<tr>
<td>Parameters</td>
<td>$c_{f_{avg}}, z_{mid}, D, L, T_{top}, T_{bottom}$</td>
</tr>
</tbody>
</table>

B.7 Dehydrator

![Dehydrator icon](image)

**Figure B.7:** Dehydrator icon

**Component description**

The dehydrator component separates $\text{H}_2\text{O}$ from $\text{CO}_2$. In addition it enables the transition from the ideal EoS to the Span Wagner (S&W) EoS.

**Relevant phenomena**

- Adsorption of water molecules
- Accumulation of mass and energy
- Frictional losses

**Hypothesis and assumptions**

- Heat released due to adsorption is low compared to the total energy of the flow
- Instant adsorption of water molecules
- Switching between dehydrator columns has negligible impact on compressor performance
- Change in potential energy is negligible
- No reactions take place
- Perfect thermal insulation (adiabatic process)
- Turbulent 1D flow
Sub Models

The dehydrator is modelled with two fluid storage modules with in between a resistive module. Components are taken from the ThermalPower library. The resistive module is adapted so it can model the transition from the ideal EOS to the S&W EOS. The equations of the fluid storage module and resistive module are already derived before.

Model input and output variables and parameters

<table>
<thead>
<tr>
<th>Input variables</th>
<th>$m_{in}, m_{out}, h_{in}, X_{in}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output variables</td>
<td>$h_{out}, P_{in}, P_{out}$</td>
</tr>
<tr>
<td>Parameters</td>
<td>$V, C_{f_{avg}}$</td>
</tr>
</tbody>
</table>

B.8 Stripper

![Figure B.8: Stripper model icons from the ThermoPower library](image)

Component description

The stripper model replaces the connection of the compressor train with the capture unit model. The pressure in the stripper is controlled by a valve. The feed stream resembles the outlet stream of the capture unit model.

Relevant phenomena

- Heat and mass accumulation
- Mass transfer between liquid and gas
- Heat release due to the absorption reaction
- Liquid hold up

Hypothesis and assumptions

- When designing the compressor train with sufficient head the stripper dynamics can be omitted
– Feed stream is already condensed by the condenser. Liquid is captured in downstream knock-out drum
– Feed stream only consist of H$_2$O and CO$_2$. Other components have negligible impact on compressor performance

**Sub Models**

The stripper model consists of a flow source, fluid storage module and a stripper control valve. The components are taken from the ThermalPower library. Their equations are already derived in the previous sections.

**Model input and output variables and parameters**

<table>
<thead>
<tr>
<th>Input variables</th>
<th>$f(m_{in})$, $h_{in}$, $P_{out}$, $X_{i_{in}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output variables</td>
<td>$m_{out}$, $h_{out}$, $X_{i_{out}}$</td>
</tr>
<tr>
<td>Parameters</td>
<td>$V$, $K_v$</td>
</tr>
</tbody>
</table>


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Glossary

List of Acronyms

IEA    International Energy Agency
IPCC   Intergovernmental Panel on Climate Change
IEA    International Energy Agency
CCS    Carbon Capture and Storage
PCC    Post Combustion Capture
IPCC   Intergovernmental Panel on Climate Change
IEA    International Energy Agency
MEA    Monoethanolamine
EoS    Equation of State
S&W    Span Wagner
SRKHV  Soave Redlich Kwong EoS with modified Huron Vidal mixing rules
LKP    Lee Kesler Plocker
List of symbols
## List of symbols

<table>
<thead>
<tr>
<th>Symbols</th>
<th>Greek letters</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Area $[m^2]$</td>
</tr>
<tr>
<td>$a$</td>
<td>Speed of sound $[m/s]$</td>
</tr>
<tr>
<td>$Cf$</td>
<td>Friction coefficient $[-]$</td>
</tr>
<tr>
<td>$D$</td>
<td>Diameter $[m]$</td>
</tr>
<tr>
<td>$G$</td>
<td>Total momentum $[kg.m/s]$</td>
</tr>
<tr>
<td>$g$</td>
<td>Gravitational acceleration $[m/s^2]$</td>
</tr>
<tr>
<td>$h$</td>
<td>Enthalpy $[J/kg]$</td>
</tr>
<tr>
<td>$H$</td>
<td>Height of column $[m]$</td>
</tr>
<tr>
<td>$k$</td>
<td>Constant $[-]$</td>
</tr>
<tr>
<td>$K_v$</td>
<td>Flow coefficient $[-]$</td>
</tr>
<tr>
<td>$MM$</td>
<td>Molar mass $[kg/mol]$</td>
</tr>
<tr>
<td>$M$</td>
<td>Total mass $[kg]$</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Mass-flow rate $[kg/s]$</td>
</tr>
<tr>
<td>$N$</td>
<td>Rotational speed $[rad/s]$</td>
</tr>
<tr>
<td>$N_T$</td>
<td>Normalized rotational speed $[%]$</td>
</tr>
<tr>
<td>$N_q$</td>
<td>Number of notes $[-]$</td>
</tr>
<tr>
<td>$P$</td>
<td>Pressure $[bar]$</td>
</tr>
<tr>
<td>$\dot{Q}$</td>
<td>Heat added to system $[J/s]$</td>
</tr>
<tr>
<td>$q_i$</td>
<td>Heat transfer $[W/K]$</td>
</tr>
<tr>
<td>$q_v$</td>
<td>Vapour quality $[-]$</td>
</tr>
<tr>
<td>$R$</td>
<td>Ideal gas constant $[J/mol.K]$</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number $[-]$</td>
</tr>
<tr>
<td>$S$</td>
<td>Surface $[m^2]$</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature $[\circ C]$</td>
</tr>
<tr>
<td>$T_q$</td>
<td>Torque $[Nm]$</td>
</tr>
<tr>
<td>$t$</td>
<td>Time $[s]$</td>
</tr>
<tr>
<td>$U$</td>
<td>Total internal energy $[J]$</td>
</tr>
<tr>
<td>$u$</td>
<td>Internal energy $[J/kg]$</td>
</tr>
<tr>
<td>$U_c$</td>
<td>Overall heat transfer coefficient $[W/m^2.K]$</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume $[m^3]$</td>
</tr>
<tr>
<td>$v$</td>
<td>Velocity $[m/s]$</td>
</tr>
<tr>
<td>$W$</td>
<td>Work done by the system $[J/s]$</td>
</tr>
<tr>
<td>$y$</td>
<td>Vapour fraction $[mol/mol]$</td>
</tr>
<tr>
<td>$x$</td>
<td>Liquid fraction $[mol/mol]$</td>
</tr>
<tr>
<td>$X$</td>
<td>Component mass fractions $[kg/kg]$</td>
</tr>
<tr>
<td>$y$</td>
<td>Vapour molar composition $[mol/mol]$</td>
</tr>
<tr>
<td>$z$</td>
<td>Height $[m]$</td>
</tr>
</tbody>
</table>

### Abbreviates

| Avg | Average |
| Bottom | Bottom of component |
| C | Normalized |
| D | Design |
| E | Entering |
| FF | Frictional force |
| In | Inlet |
| L | Leaving |
| LL | Liquid level |
| LIQ | Liquid |
| Mechanical | Mechanical |
| Out | Outlet |
| S | Shaft |
| Isotropic | Isotropic |
| Sat | Saturated |
| Vap | Vapour |
| Top | Top of component |

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