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Thesis: Design and Engineering of Main Gas Compression Piping.

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I am proud to present to you the final project of my HBO studies in Mechanical Engineering. I am happy that my project has captured a few essentials subjects from mechanical engineering: designing, material science and statics; combined with knowledge from the field of piping design and engineering, learnt on the job.

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1

Abbreviations

The following list serves to clarify the abbreviations used in this document.

BOE	Barrels of Oil Equivalent
BOE/D	Barrels of Oil Equivalent per Day
FPSO	Floating Production Storage and Offloading Vessel
LOI	Letter of intent
VLCC	Very Large Crude Carrier
CO2	Carbon Dioxide Gas
HP	High Pressure
LP	Low Pressure
BOD	Basis of Design
CES	Corporate Engineering Standards
PMC	Pipe Material Classes
P&ID	Process and Instrumentation Diagram
PFD	Process Flow Diagram
Fore	Front side of a ship
Aft	Back side of a ship
ASME	American Society of Mechanical Engineers
API	American Petroleum Institute
DNV	Det Norske Veritas
FEA	Finite Element Analysis
CAESAR2	FEM based Pipe Stress Analysis Software

2

Summary

SBM Schiedam is active in the design, engineering and construction of Floating Production Storage and Offloading (FPSO) ships. Currently SBM has a few projects for Petrobras. Purpose of an FPSO is to produce oil from a subsea oilfield. The FPSO production process is split over different modules which can be built simultaneously. A few of these modules are gas processing modules, where gas is treated and compressed. This compression process requires a centrifugal compressor which is connected to scrubbers and heat exchangers by piping. Because centrifugal compressors are sensitive rotating machines, they can take little nozzle loading. Industry supports this view and API subscribes a recommendation for allowable nozzle loads. These nozzle loads need to be adhered to as a minimum to guarantee proper functioning of equipment.

In order to stay below allowable nozzle loads. The piping system must be self supporting, that is to say it does not rest on the compressor system. This is just the start of a good design, in fact the compressor piping near the machine must be designed so that it is isolated from the effects of the rest of the piping system. This forms the investigation of this report. To find the solution a basis of design is made, a design is proposed and this design is tested by means of a finite element model.

The proposed solution is to design 'isolation loops' around the compressor, which allow controlled thermal growth in the horizontal direction. The isolation loops are separate from the system by placing an axial stop in line with the fixed point of the centrifugal compressor. Furthermore a spring hanger is fitted for vertical support but allowing vertical growth. Isolation loops create an additional problem for some conditions to minimize this effect a snubber device is added to the loops.

The proposed solution works in theory and needs to be implemented in a working model and if proven effective, tried on a real module.

3

Project backgrounds

In this chapter a brief background is given to understand the context of the assignment.

3.1 Brazilian oil developments

Only a decade ago, the notion that Brazil would become self-sufficient in energy, let alone a major exporter, seemed far-fetched. This changed with the discovery of ultra-deepwater pre-salt reservoirs like Tupi, estimated to be the western hemisphere's largest oil discovery of the last 30 years. [Subsea7] The pre-salt oil reserves are estimated to contain a measure of 56 billion barrels of oil equivalent (BOE).

Although the upstream oil sector was fully liberalized in 2000 Petrobras, Brazil's national oil and gas company, still retains a dominant position in the industry. Petrobras has set an ambitious growth target to be the largest oil producer by 2020 outperforming other internatinal oil companies. By 2014 Petrobras wants to produce 3,9 million (BOE/D) and by 2020 5,4 million (BOE/D) to reach this target 1 million (BOE/D) will eventually be expected from the pre-salt fields. [Petrobras]

Figure 3.1: Santos Pre-Salt Basin

The Santos Basin pre-salt fields are located 300km offshore South-East of Sao Paulo, Brazil and consist of an area of 352,260 square kilometers. (See figure 3.1) The fields are called pre salt because the oil is buried below as much as 2000m of salt. The wells in the Santos Basin start at a water depth of 2km. The total measured drilling depth is almost 6km.[Halliburton] The waterdepth and drilling depth are historical challenges that need to be addressed. Due to the ultra-deepwater conditions conventional fixed production platforms are not feasible. Therefore Petrobras has chosen to use floating production facilities, some of which are FPSOs. In December 2009 Petrobras employed a total of 41 production platforms and FPSOs by 2020 Petrobras expects a total of 84.

In August 2011 SBM Offshore received a letter of intent (LOI) from a consortium headed by Petrobras, for the 20 year charter and operation of an FPSO for the Guara Norte block development. The Guara Norte field is located in block BM-S-9 in the Santos Basin.

The FPSO will include topside facilities to process 150.000 BPD of production fluids, associated gas treatment for 6,000,000 m^3/d with compression and carbon dioxide removal, hydrogen sulphide removal, and a water injection facility for 180.000 BPD. The project schedule foresees delivery of the FPSO in 35 months from LOI. In the near future this FPSO will be referred to as FPSO 'Cidade de Ilhabela.' This will be the ninth FPSO for the Brazilian market and the largest FPSO (capacity) to date.

3.2 FPSO overview

This section gives a brief overview of floating production storage and offloading vessels (FPSOs). Reference will be made to the Ilha Bela, SBMs latest FPSO EPCM project. For this project SBM Offshore is refurbishing and converting a very large crude carrier (VLCC) into a FPSO. The ship is refurbished and prepared for integration of the modules in China, processing modules are constructed in Brazil and placed on the vessel at a later stage. To understand the FPSO concept one needs to be familiar with the development of a typical offshore oil and gas field.

1. Exploration
2. Exploratory drilling
3. Development drilling
4. Production
5. Storage and offloading
6. Transportation
7. Decommisioning

The FPSO is involved in the exploitation of field from the production stage onwards. The FPSO is connected to the oil field and stores produced oil in storage tanks. Flowlines connected to flexible risers link the subsea development wells to the FPSO after the development wells have been drilled by other types of offshore units. Produced oil is periodically offloaded to shuttle tankers using a flexible hose reel.

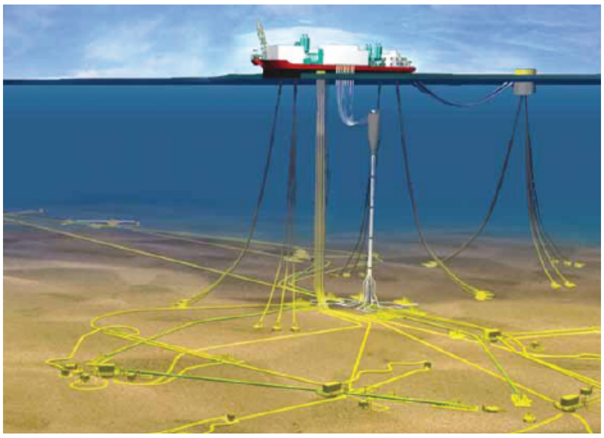

Figure 3.2: Spread Moored FPSO

FPSOs can be moored using different methods depending on prevailing ocean and weather conditions. The Ilha Bela will make use of a spread mooring arrangemenent. This is a mooring system that consists of multiple lines terminating at different locations on the floating structure, extending outwards and anchored to the seabed. This provides an almost constant vessel heading and secures the FPSO in one location. These lines are made of heavy chains which are tensioned onboard the vessel by special winches. This mooring arrangement is an alternative for SBMs patented turret mooring system, for rough environmental conditions.

The incoming risers carrying the produced hydrocarbons enter the vessel on the side of the ship by means of a dedicated riser balcony. There are also outgoing risers hung from this balcony for enhancing well production by reinjecting water, gas and CO_2. On the balcony is a riser pulling system that provides the right amount of tension on the risers.

Figure 3.3: Riser Balcony

A current trend in the FPSO industry is that vessel's production capacity is increasing both in size and in terms of scope. FPSOs are becoming full offshore production systems where a saleable crude oil is made onboard the facility. The hydrocarbon well stream coming from subsea production is separated into water, oil and gas. These products are further treated for carbon dioxided and other contaminants. Produced water and carbon dioxide is re-injected in the reservoir to maintain pressure. This trend calls for larger vessels with a high equipment density on the deck. The mentioned processing systems are seperately built on modules and integrated on board of the vessel. Modules are arranged so that low pressure, non hydrocarbon processes occur at aft of the ship and high pressure processes are located at the bow. Living quarters, accommodation and utilities are located near aft and the flare system is located at the bow, this with respect to safety. (See Drawing DTT001 and DTT006 located in the appendix)

Utilities	**Function**
Power generation	Generate electricity for vessel and processes.
Seawater treatment	Produce desalinated water for processes.
Production process	
Chemical injection	Enhance recovery of the oil field.
Water injection	Pressurize oil field and dispose produced water.
Production manifold	Couples incoming/outgoing risers to the FPSO.
Oil processing	Separation of well streams into water, oil and gas.
Main gas compressors	Provide pressure differential in the gas stream.
CO2 compressors	Compress CO2 gas.
Gas treatment	Removal of CO2, removal of sulphur, dehydration.
Injection gas compressors	Compress gas for injection into well.
Flare system	Burn off toxic, hazardous, flammable gasses in emergencies.
Vent system	Depressurize the system.
Offloading system	Offload produced crude into shuttle tanker

The FPSO is a complex processing facility, the focus of this document is on the piping of the Main Gas Compression Module B.

3.3 Main gas compression process description

To understand the context of the compressor piping system a brief description of the process. These processes are typically divided over different modules.

After the crude enters the vessels it is treated and split into three main streams: water, oil and gas. This separation happens in vessels called separators. Gas from the High Pressure (HP) separator and the vapor recovery unit is routed to Main Gas compressor A. Here the gas is pressurized to a level sufficient to feed the gas treatment system. Main Gas Compressor A shall also be able to handle regeneration fas flow from the downstream Gas Treatment system. The gas used for regeneration of the molecular sieves is recycled to Main Gas Compressor A. Since this gas is available at a higher pressure, the gas stream shall be introduced to the compressor as a side stream at a level higher than the suction pressure. Main Gas Compressor A shall have a 2 times 100% configuration and is single stage. For spare capacity and back up purposes.

Main Gas Compressor B is fed with treated gas by the CO2 removal membranes system or with non treated gas by the bypass connection around the membranes system, following production modes. The gas is further pressurized to a sufficient level for export and for further re-injection into the reservoir, to pressurize the field. Main Gas Compressor B shall also have a 2 times 100% configuration and is two stage.

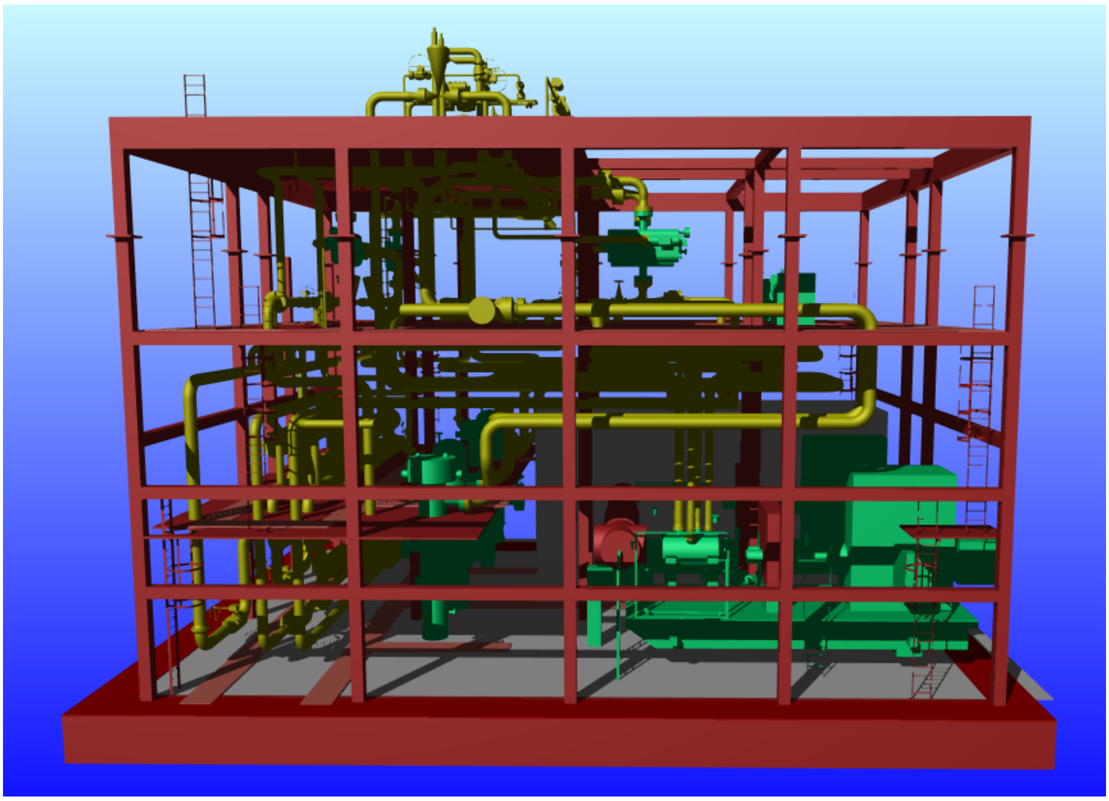

Figure 3.4: Module Main Gas Compressor B

4

Project definition

4.1 Main Objective

Design and engineering of compressor piping systems are a recurring challenge on SBM Offshore's projects. Compressors modules are inherently one of the toughest modules to design due to many basis of design criteria. Because this system transports high pressure, high temperature, flammable gasses the compressor piping systems requires sound engineering. Failure of the compressor piping system can lead to large scale disasters. Besides physical danger, poor engineering of compressor modules can threaten financial success of the project in terms of construction difficulty and maintenance costs. By setting a good standard for the critical lines of the compressor, engineering lead time can be reduced and troubleshooting for poor engineering in a later phase can be avoided.

The main objective of the project is to set a standard philosophy for the compressor interface pipe routing and supporting to be used as a reference on future projects.

4.2 Goals

The main objective is decomposed into the following goals:

- Generate a basis of design (BOD) for gas compressor piping systems.
- Design a piping system that meets BOD requirements.
- Verify design by means of a pipe stress analysis.
- Set a standard for gas compressor modules.

Because there are multiple gas compression modules on the FPSO, Main Gas Compressor B is chosen. This is the most challenging compressor because it is a two stage compressor with the four nozzles and because gas is at relatively high pressure and temperature. In operation this compressor raises the gas pressure from 130 to 324 bar g. with an associated end temperature of 101 °C. These pressures and temperatures are not much of a problem for the piping system, however due to the large wall thickness of pipe to contain this pressure, the nozzles are subject to large loads and moments. Goal of the assignment is to reduce these loads by flexible routing and choosing adequate, practical supports.

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Basis of design

The basis of design (BOD) has been compiled based on SBM's Corporate Engineering Standards (CES) (see appendix) and is further based on interviews with experienced piping professionals. The BOD serves to be an exhaustive list of criteria that the compressor piping system should meet.

5.1 Main Gas Compression Process

The main gas compressor piping is routed between three pieces of equipment: suction scrubber, centrifugal compressor and heat exchanger. Compressor B is usually located on a dedicated module in a 2x100% configuration for back up and sparing operation modes. We will focus on one compressor configuration that can be duplicated on the module. Where information such as wall thickness, material, process data etc... is needed, data from a existing FPSO is used.

5.1.1 Suction scrubber

Suction scrubbers are placed upstream of the compressor's inlet nozzle. The suction scrubber removes liquids from the gas stream that could be damaging to the compressor impeller. The internals of the scrubber are configured so that gas streams through a dense mesh to remove liquid contaminants. These liquid contaminants in the gas stream, exit the scrubber at the bottom, gas flows to the compressor from the top nozzle of the scrubber. See figure 5.1 for an impression.

Figure 5.1: Suction Scrubber

5.1.2 Centrifugal compressor

The centrifugal compressor is used to create a pressure differential to enable gas flow, the compressor does this by means of a high RPM impeller. On this project a choice is made for a radially split compressor, where the compressor is split perpendicular to the shaft. This allows the compressor bundle to be axially removed from the compressor for maintenance purposes. See figure 5.2 for an impression.

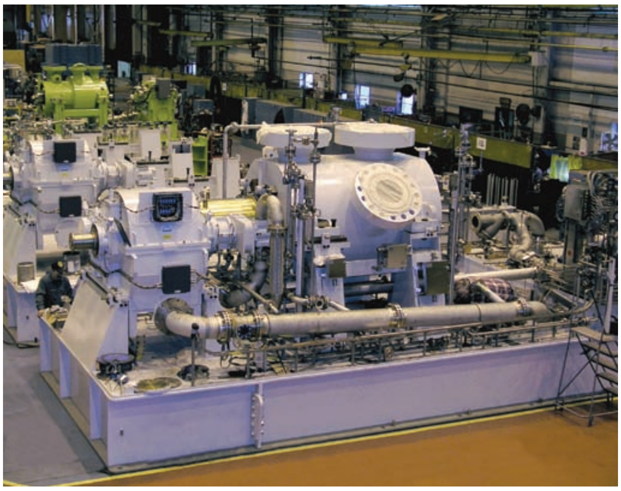

Figure 5.2: Dresser Rand Two Stage Centrifugal Compressor

5.1.3 Heat exchanger

The compressor conducts work on the gas raising temperature and pressure. The heat exchanger is a piece of equipment built for efficient heat transfer from one medium to another. The heat exchanger is installed downstream from the compressor to cool the gas. The heat exchanger is a printed circuit type and cools using inhibited water. These heat exchangers are also referred to as discharge coolers. See figure 5.3 for an impression.

Figure 5.3: Compact Plate Heat Exchanger

5.1.4 Equipment Summary

The following table displays the equipment at the interface of the piping system, reference is made to the appendix for vendor drawings of the equipment.

Equipment	**Tag-numbers**
Compressor B	1K-T7141
Stage 1 suction scrubber	1V-T7141
Stage 2 suction scrubber	1V-T7142
Stage 1 discharge cooler	1E-T7141
Stage 2 discharge cooler	1E-T7142

5.2 Material Selection

The material selection standard provides general principles, engineering guidance and requirements for material selection and corrosion protection for all parts of the offshore installations;

- Corrosion and material selection evaluations.
- Specific material selection where appropriate
- Corrosion protection
- Design limitations for candidate materials
- Qualification requirements for new materials or new applications

Based on these factors and process conditions a material selection is made for the compressor piping. In the case of the compressor piping two pipe materials A333 Gr.6 and API 5L X52, which have the same mechanical properties.

5.3 Material Classes

The material classes specfication covers Piping Material Classes (PMC) with material requirements for all FPSO Topsides piping systems. The PMCs have been setup with respect to the ABSM B31.3 process piping code. Data from the assigned pipe class is used to construct and model the compressor piping system.

The PMC consists of piping components selected according to the requirements in the applicable rules and regulations. Within the design limits of the PMC the individual components of the piping system are fit for purpose. Each line on the P&ID is assigned a PMC which is specified further in the PMC. Below is a summary of the lines subject of study. For more information reference is made to the appendix.

Line	**Nominal size [inches]**	**Pipeclass**
Scrubber 1 to Compressor	12	15C04
Compressor to Discharge Cooler 1	12	15C04
Scrubber 2 to Compressor	12	15C04
Compressor to Discharge Cooler 2	10	25C06

5.4 Flushing and testing

The flushing and testing specification covers flushing, testing and non destructive examination (NDE) of piping systems. The safety of personnel shall be of prime importance during these procedures. For the compressor lines this implies that all lines will be tested and need to be designed to allow for this. Especially the hydrotest creates an additional design challenge because the system needs to be examined closest to its intended state. Thus the final spool, connected to the compressor is fitted with a flanged connection to allow rotation and fitting of a blind flange.

5.4.1 Non Destructive Examination

A complex visual inspection and dimensional check shall be carried out for all fabricated piping systems in accordance with ASME B31.3, Chapter VI. Each piping system shall be checked against the isometric drawings for material identification, dimensions and fittings. A record shall be retained of all such checks and inspections.
Inspection and testing shall be carried out after fabrication, welding and heat treatment has been finalized but prior to painting, coating and lining. If any welding is performed after inspection and testing, a retest will be required.
Various examination techniques are used depending on the pressure class, wall thickness and pipe material. The following examination techniques are used:

- Visual inspection
- Radiographic testing
- Ultrasonic testing
- Magnetic particle examination
- Liquid penatrant examination

5.4.2 Preparation for flusing and pressure testing

All items that could be damaged resulting from flushing/testing shall be blanked off, or removed from the system.
All piping shall be adequately supported; spring supports shall have the pin insterted/blocked to prevent movement. Temporary supports shall be provided if additional stability is deemed necessary. Special consideration shall be given to vapor lines.
Hydrotest and flushing medium (usually water) shall not exceed 200ppm chloride ions and shall contain a corrosion inhibitor to protect the piping system. Lines need to be thoroughly dried after testing. For stainless steel, hydrotest and flushing medium shall not exceed 50ppm.
All lines shall be checked to ensure they are clear of debris that may have accumulated during facbrication, installation and erection. Wherever practical, all the lines shall be blown down with compressed air prior to flushing. All necessary precautions shall be taken to ensure debris is not flushed into associated equipment or "dead ends."
All in-line pressure sensitive equipment shall be substituted with hydrotest spools. All spools shall be pressure rated to the applicable PMC, consistent with equipment pressure rating.

5.4.3 Flushing

Flushing is required to clean the system from debris.When lines are broken for flushing, on completion of the flushing, broken connections shall be reinstated using new gaskets.All lines that require pressure testing shall be flushed prior to system testing. The medium used for flusing shall be same as that required for the pressure test. For corriosion resistant alloys, seawater may be used as the flusing medium as long as the line is properly cleaned after flushing.
For piping systems where the flusing medium is water, flow velocities of 1.5 to 2 times the normal operating velocity or 5 m/s, whichever is greater must be achieved wherever possible.

Piping systems which are blown-out using compressed air, shall be oil free and dry, the flow velocity should not be less than 35 m/s.
The main headers shall be flushed out first and then all the branches, which are connected to any equipment. All necessary precautions shall be taken to ensure that debris is not flused into associated equipment or dead ends. The main headers shall be flushed for 30minutes at least, branches 15 minutes respectively. The system shall be flushed or blown down from the highest point in the system.

5.4.4 Pressure testing

Pressure testing is necessary to verify that the system is suitable to hold 1,5x design pressure without leaking. In-line equipment such as control valves, shall be isolated during the hydrostatic test. If this is impractical then a vendor agreement shall be obtained to satisfy points listed. Pressure vessels are usually tested separately.

- The equipment test pressure is equal to or greater than the piping system.
- The supporting steelwork should not be over stressed if the pipework and equipment were tested together due to combined weights.
- Equipment shall not suffer damage due to testing meidum.

The thickness of blinds used for pressure testing shall be in accordance with ASME B16.5 Pipe Flanges and Flanged Fittings. Blinds cut from plate shall have the thickness calculated in accordance with ASME B31.3.

5.5 Piping Plant Layout

5.5.1 General Layout Considerations

This specification in addition to other specifications and codes, defines the general piping requirements for the plot plan and plant layout design to provide the required operational, maintenance and safety measures. Layout shall be consistent with prevailing atmospheric and site conditions and accepted engineering practices. It is important to assess the escape route requirements at a very early stage to ensure escape areas are not compromised during layout development. The arrangement of modules shall provide the maximum separation between hazardous and safe areas in descending order, from fore to aft.

Lower deck module steel (pancakes) are aligned with columns, which are attached to the deck at ship webframes. Major equipment (pumps, drums, exchangers, compressors etc.) should be arranged so that their supoorts are on primary steel work where possible, to avoid need for heavy secondary steel. Rotating machinery such as compressors preferably shall be positioned with the centerline being parallel to that of the vessel, the steadiest axis of the ship.

The primary consideration in the piping layout of areas shall be to provide an economical facility that is safe, fit for purpose, logical, easy to operate and maintain. Piping arrangments shall favor compactness but shall allow sufficient room for escape routes. Piping shall be routed to ensure the shortest practical length without affecting flexibility. A minimum number of flanges, fittings and other components are to be used.

Valves, instruments and equipment requiring inspection, maintenance or servicing shall be accessible from major platforms or walkways where possible. When access from major platforms on walkways is not possible, then additional platforming, walkways or ladders shall only be provided when economically feasible.

Piping around equipment shall be designed to allow maintenance, servicing, testing and removal of equipment with minimum dismantling and disruption of piping systems. This can be accomplished by providing valves, spades and spectacle blins and sufficient flanged connections. Provision shall be made for using mechanical means of lifiting, where weight exceeds 25kg.

5.5.2 Compressor Module Layout

The following design criteria are more specific to the compressor system and need to be met as a minimum.

- Piping local to the compressor will be supported using spring supports to allow thermal expansion of the nozzle but take the weight of the compressor nozzle.
- Temporary or permanent strainers shall be provided in compressor suction lines for start up and continuous operation. The strainer shall be located as close as possible to the compressor inlet nozzle. Piping arrangement shall allow removal of temporary strainer without the need to dismantle piping, supports or affect alignment.
- All gas compressor suction piping between the scrubber and compressor shall be kept to a minimum length and routed to avoid trapping or collecting liquid and permit draining of condensate back to the knock-out vessel. Therefore the suction lines will slope back toward the scrubbers. Heat exchangers should be elevated above the scrubbers.
- Reducers located upstream of compressor should be located 5D minimum from nozzle to induce an even inflow, important for equal loading of the impeller blades. For flow measurement purposes there must be a straight length of 4D upstream and downstream 20D, where an orrifice flange will be fitted.
- A check valve (non-slam) shall be located on each compressor discharge line, as close to the discharge nozzle as is practically possible. The check valve prevents medium flowing back to the compressor in case of a shut down.
- From a maintenance perspective the layout of the compressor must allow for mechanical handling. For the gas compressor body and bundle a lifting arrangement is provided in the surrounding steel structure for maintenance.
- Removable spools shall be provided for all equipment that needs maintenance and shall be as short and light as possible for handling purposes. The compressor is fitted with a removable spool to allow easy assembly. Furthermore the spool needs to be rotatable (clash free) to allow for pressure testing.
- The compressor bundle (approximately 3000 kg) is extractable and in case of maintenance, put to rest on a vendor supplied maintenance sledge. This sledge is mounted onto the compressor casing. The compressor bundle is pulled from the compressor onto the sledge. If local maintenance is not feasible, the sledge is mounted on a trolley with a dedicated beam and moved to a location that is within crane reach.
- For supporting reason multiple layers of pipe above each other are to be avoided, the bottom of pipe (BOP) elevation is to be kept the same.
- All equipment especially the compressor is to be "isolated" from rest of the system to minimize nozzle loading under all circumstances by using bends and adequate supports.

5.6 Pipe Stress

Within the piping discipline, structural analysis of piping systems is commonly referred to as pipe stress analysis or just stress analysis. To validate the structural integrity of the compressor piping system different analysis are performed. All piping shall be designed in such a way to optimize loading on equipment nozzles to satisfy vendor allowable criteria and to prevent disturbance of alignment and internal clearance. Changes in direction and built-in loops shall be used to increase flexibility of the system. All piping shall be designed taking taking into account vessel motion and environmental conditions. Normally the piping system is also evaluated for blast but this is disregarded in the scope of this project. Fatigue is also excluded in the analysis as the system is assumed to have less than 7000 start up/shutdown cycles during its lifetime. The system will be evaluated using the following standards.

- API Standard 617 Axial and centrifugal compressors and expander-compressors for petroleum, chemical and gas industry services.
- AMSE B31.3 Process Piping
- ASME B16.5 Pipe Flanges and Flanged Fittings
- DNV Recommended Practice DNV-RP-D101L Structural Analysis of Piping Systems
- SBM Allowable Nozzle Loads

The ASME B31.3 code is the basic design code for all process piping on the FPSO modules. To ensure the structural integrity of the piping systems, the code has assembled a set of procedures and specifications covering the minimum requirements for material, design, fabricaton, erection, inspection and testing. The piping system is ensured of proper safety factor on structural integrity when all code requirements are followed and satisfied. For the purpose of this project the code will be used to evaluate the sustained loads and displacement strains on the system. To conduct this analysis use is made of COADE CAESAR2, a dedicated pipe stress software based on beam element theory, which contains a piping code-check module. [DNV] A hand calculation will be used to verify accuracy of the software this will be demonstrated during the defense of this thesis.

5.6.1 API617 nozzle loads

The layout of the piping system is governed by the allowed interface reaction loads. API 617 specifies that "the design of each compressor body must allow for limited piping loads on the various casing nozzles. For maximum reliability, nozzle loads imposed by piping should be as low as possible regardless of the compressor's machines' load carrying capability." The API load is increased with a factor of 3,5 after mutual agreement between SBM Offshore and vendor, allowing higher loads. The compressor cannot endure slight deformations due to the risk of shaft misalignment. To maintain smooth operation of the rotating equipment, the shaft needs to be kept in perfect alignment. [PENG]

The total resultant force and total resultant moment imposed on the compressor at any single connection (individual nozzle) should not exceed the values shown in equation (5.1)

$$F_r + 1.09M_r \leq 189D_e \tag{5.1}$$

where

F_r = resultant force in Newtons

$$F_r = \sqrt{F_x^2 + F_y^2 + F_z^2} \tag{5.2}$$

M_r = resultant moment in Newton-meters

$$M_r = \sqrt{M_x^2 + M_y^2 + M_z^2} \tag{5.3}$$

For sizes up to 200 mm (8 in.), use a value of:

$$D_c = \frac{400 + D_{nom}}{3}(mm) \tag{5.4}$$

where

D_c =equivalent pipe diameter of the connection, in mm.

D_{nom} = nominal pipe diameter, in mm.

The combined resultants of the forces and moments of the two inlets and two discharge connections resolved at the centerline of the largest connection shall not exceed the following:
1. The resultants shall not exceed:

$$F_x + 1.64M_c \leq 141.4D_c \tag{5.5}$$

where

F_c = combined resultant of inlet and discharge forces, in Newtons.

M_c = combined resultant of inlet and discharge moments, and moments resulting from forces, in Newton-meters.

D_c = diameter (in mm) of one circular opening equal to the total areas of the inlet and discharge openings. Because the equivalent nozzle diameter is greater than 230 mm (9 in.) a value of D_c equal to:

$$D_c = \frac{460 + \text{Equivalent Diameter}}{3}(mm) \tag{5.6}$$

The individual components of these resultants should not exceed:

$F_x = 56D_c \qquad M_x = 86D_c$
$F_y = 142D_c \qquad M_x = 43D_c$
$F_z = 114D_c \qquad M_x = 43D_c$

where
F_x = horizontal component of F_c parallel to the compressor shaft, in Newtons,
F_y = vertical component of F_c, in Newtons,
F_z = horizontal component of F_c at right angles to the compressor shaft, in Newtons,
M_x = component of M_c around the horizontal axis, in Newton-meters,
M_y = component of M_c around the vertical axis, in Newton-meters,
M_z = component of M_c around the horizontal axis at right angles to the compressor shaft, in Newton-meters.

Figure 5.4 shows the orientation of the nozzle loads for the API617 calculation. The loads necessary for this calculation are obtained from the CAESAR 2 calculation.

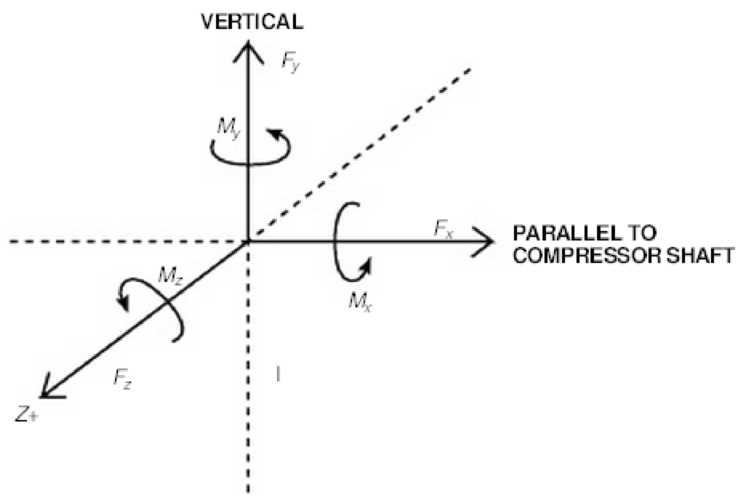

Figure 5.4: API 617 Nozzle Loads Orientation

5.6.2 Equipment Nozzle Loads Calculation

Nozzle loads on the scubbers and heat exchangers are subject to allowable nozzle loads as indicated by SBM specification ES45000 SKF92064C1. These loads can be found in table beneath. Figure 5.5 shows the description of the nozzle load components.

Equipment	Size in.	Orientation	Class	F_x (N)	F_y F_z (N)	F_r (N)	M_x (Nm)	M_y M_z (Nm)	M_r(Nm)
1V-T7141	16”	-Y	900	38345	46965	76690	67450	47660	95325
1V-T7142	14”	+Y	1500	41830	51235	83665	64465	45585	91710
1E-T7141	10”	+Z	2500	22920	28075	45845	31555	22315	44628
1E-T7142	8”	+Z	2500	17865	21880	35730	19950	14110	28217

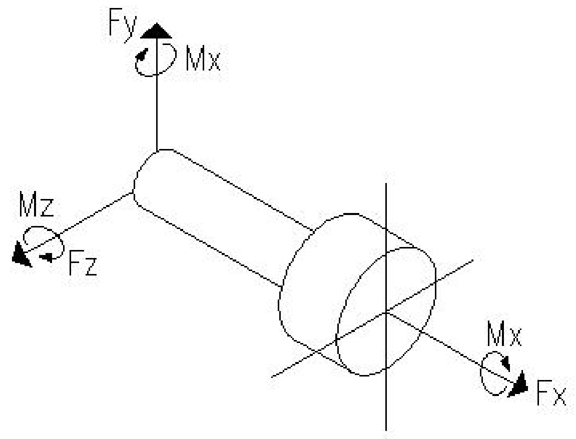

Figure 5.5: Equipment Nozzle Loads

6

Design

In this chapter a design is proposed for the main gas compression B piping. Please refer to drawings DTT002 'Overall Plan' and 'DTT005 Concept Side View'in the appendix when reading this section. As stated in the basis of design the API 617 states certain allowable loads on the compressor, this has large implications for the routing and supporting of piping local to the compressor. The most important goal of the assignment is to generate a philosophy for the routing and supporting of compressor piping. The rest of the design is also of importance but is to a lesser extent the focus of the assignment.

6.1 Routing of the compressor piping

The routing and supporting of compressor piping is designed to minimize loading of the compressor nozzles. The main philosophy behind the compressor piping system is to 'isolate' the piping that is connected to the compressor. This can be achieved by designing a loop around the compressor as shown in (figure 6.1). As the temperature of the piping system increases, there is thermal growth of the pipe. If this thermal growth differs across the legs of the loop, a moment is built up on the compressor nozzle. By installing an axial stop in line with the fixed point of the compressor two equal legs are created, because these legs are unrestrained they can grow equally in the same direction, preventing development of a moment. Furthermore this axial stop prevents displacements occuring elsewhere in the system from being transmitted on the compressor nozzle. The legs are also fitted with a guide so that sideward movement is restricted, expansion only happens in the fore direction of the vessel.

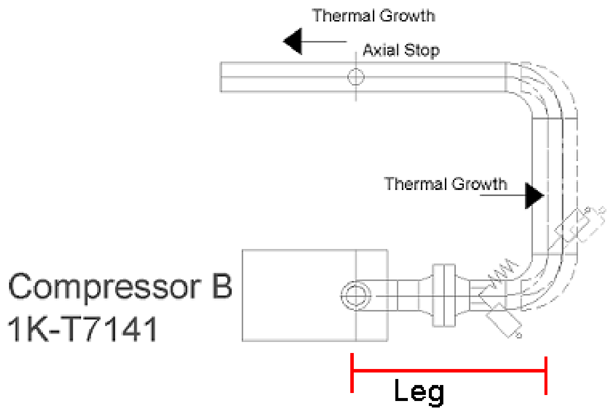

Figure 6.1: Isolation loop

The loop is fitted with a variable spring hanger (see figure 6.2). The variable spring hanger takes the load of the compressor nozzle by providing an upward force. This force is also present during thermal growth of the piping system, thus the piping remains supported. The underlying objective for using a spring hanger is to minimize the change of load applied by the piping on the connected equipment. Internally a spring hanger consists of a spring coil which resists the force which compresses it. To calculate the required spring the sustained vertical load (dead load) is considered.

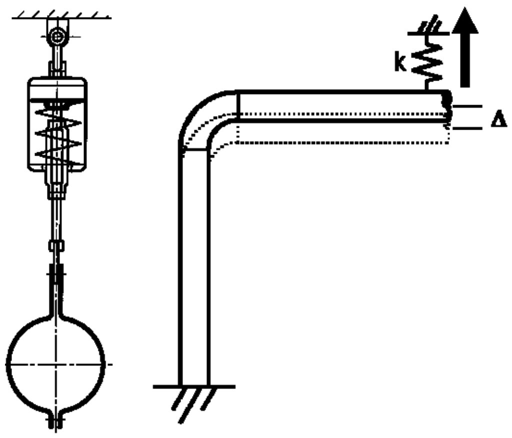

Figure 6.2: Variable spring hanger

During the stress analysis phase it was observed that the loops work for mostønditons. However there are some cases, such as a storm (heavy winds and high vessel accelerations) where the loops create excessive loading of the compressor nozzle. An ideal solution for this problem is installing a device called a snubber. A snubber is used when unrestrained movement must be allowed, but acts as a restraint during impulsive or cyclic disturbances. The unit is nof effective against low amplitude, high frequency movement. As can be seen in (figure 6.3) most hydraulic snubbers have a piston which is relatively unconstrained at low displacement rates. At high displacement rates the piston 'locks-up', that is, the force required to move the piston increases substantially, usually as a result of closing of a valve. The snubber is mounted at a 45°angle with the pipe to restrict motion in both vertical and fore/aft direction.

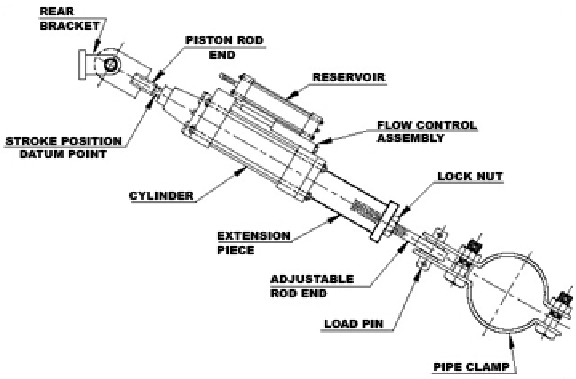

Figure 6.3: Hydraulic snubber

The loops have been designed so that they do not reach over the driver, leaving room for maintenance. Each of the connections has been designed using the following philosophy. The final spool is disconnectable (for maintenance), rotatable (to hydrotest the system in one go) and it can be field welded (alignment with the compressor). The idea is that the system can be installed up to the first flange after the spring hanger, the spring hanger will keep the system in place. The final spool is made on site to achieve alignment requirements with the compressor. After achieving correct alignment, the spool is rotated at the flange and fitted with a blind flange.

6.2 Plot Plan

After establishing that the compressor piping system is to be designed with loops, ways were examined to fulfill the other basis of design requirements. A plot plan establishes the location of the equipment and orientation of the nozzles. In designing the plot plan account was taken of typical module sizing. Modules typically consist of main beams spaced at 5(m) intervals. The compressor is placed on a skid that is about 10m in length (shortest). There is a requirement to have a 20D and 4D straight length requirement for placing a measurement device in the suction lines. This gives a module length of about 20m. The scrubber nozzles are given a sideways orientation to create so that they can also be designed with an 'isolation loop.' By positioning the scrubbers at the back of the module, incoming piping can enter and leave the module in the center. The heat exchangers are positioned at a higher level in the module, they are spaced as as far apart as possible to allow handling of the bundles. Furthermore piping can be routed under deck, improving access on the deck. The heat exchangers are also designed to have an 'isolation loop.'

6.3 Supports

In designing the piping, pipe supports were taken into consideration at an early stage. All of the loops have the same bottom of pipe elevation and there is no routing of pipe above them. The idea behind this is that pipes are easily reached and supports can be suspended from the deck above. Furthermore it was the aim to keep supports in line with each other in the sideward direction and close to primary steel, to avoid additional pipe support steel in the modules.

6.4 Isometrics

Based on the proposed overall plan isometrics of the piping system have been generated which serve as an input for the pipe stress analysis, subject of the next chapter.

7

Pipe Stress & Nozzle Loads

In this chapter the design proposed in the previous chapter is analysed. In order to do this a brief explanation of the pipe stress subject is given for the unfamiliar reader. The CAESAR2 model is discussed and the various load cases for which the system is analysed. After that a summary is given of the most important CAESAR2 output. Detailed output can be found in the appendix.

7.1 Pipe stress

As stated earlier, pipe stress analysis checks the structural integrity of piping systems. Pipe stress analysis has some commonalities with structural analysis sustained loads on the system, but differs due to temperatures and pressure to which the system is exposed. When the pipe is exposed to temperature, it will expand, when this expansion is restrained it causes stresses.
Within pipe stress analysis a distinction is made between primary stresses and secondary stresses. The ASME B31.3 code for Process Piping also makes this distinction for judgment of the piping system. The code specifies an allowable stress for sustained loads (primary stresses) and for displacement stresses (secondary stresses). "Primary stresses are those developed by the imposed loading and are necessary to satisfy the equilibrium between external and internal forces and moments of the piping system. Typical loads are dead weight and internal pressure." [DNV] Primary stresses are sustained stresses and are not self limiting. "Secondary stresses are those developed by constraining the free displacement of piping subjected to thermal loads or imposed displacements from movements of anchor points. Secondary stresses are self limiting." Each of these stresses is associated with a different kind of failure mode. Primary stresses can cause gross plastic deformation and rupture. Failure of the piping system may occur due to single application of the load. Allowable loads for secondary stresses are based upon cyclic and fatigue failure modes. A single application of the load never produces failure. Failure only occurs after a high number of applications of the load.

7.2 CAESAR II analysis

7.2.1 Methodology

In modern engineering practice most pipe stress analysis is carried out using Finite Element Analysis (FEA) software that is based on the beam element theory in combination with stress intensity and stress concentration factors. For the analysis of the system the piping system is divided into many small bodies that consist of nodes and elements. In piping analysis, these bodies are actually fairly large compared to the general sense of a finite element analysis. In piping FEA analysis the system is broken down into two types of beam elements: straight pipe and curved pipe. By decomposing the model into these elements, the direct stiffness method algorithm can be employed to solve for unknown displacements and forces, which can then be used to further calculate stresses. This paper will not go into further detail on this subject but will assume that accurate results are given by this type of analysis. Please refer to stress isometrics in appendix for further information. A CAESAR input echo for the piping system is also given in the appendix. For modeling purposes the system has been broken down into four distinct lines.

Line	**Drawing(s)**
Suction 1	DTT007-DTT008
Discharge 1	DTT009-DTT010
Discharge 2	DTT011
Suction 2	DTT012

7.2.2 Temperatures and Pressures

The system will be tested at three different temperatures and four different pressures. These temperatures and pressures are found on the isometrics and are based on process conditions.

- Design Maximum Temperature (T1) with corresponding pressure (P1)
- Design Minimum Temperature (T2) with corresponding pressure (P2)
- Operational Temperature (T3) with corresponding pressure (P3)
- Hydrotest Pressure (HP)

7.2.3 Displacements

Just as the pipe system undergoes thermal growth when subjected to different temperatures, equipment such as scrubbers and heat exchangers also undergo thermal growth. The result is that the nozzles of the equipment, interface with the piping system undergoes displacement. To take this effect into account, nozzle displacements are modeled according to following tables. Displacements are calculated based on tabulated data on the expansion of carbon steel.

Scrubber 1

Case	Temperature (°C)	Z-displacement (mm)	X-displacement (mm)
Operational	20	0	0
Design 1	80	2.01	0.74
Design 2	-29	-1.70	-0.62

Scrubber 2

Case	Temperature (°C)	Z-displacement (mm)	X-displacement (mm)
Operational	40	0.5	0.18
Design 1	80	1.8	0.66
Design 2	-29	-1.51	-0.55

Heat Exchanger 1

Case	Temperature (°C)	Z-displacement (mm)	X-displacement (mm)
Operational	92	0.73	1
Design 1	140	1.33	1.84
Design 2	-29	-0.20	-0.28

Heat Exchanger 2

Case	Temperature (°C)	Z-displacement (mm)	X-displacement (mm)
Operational	101	1.01	1.06
Design 1	141	1.40	1.47
Design 2	-29	-0.57	-0.59

7.2.4 Loads

An important step in the pipe stress analysis is to define the load cases to which the piping system will be subjected. Just like the stress categories, loads can be divided into primary loads and expansion loads. Primary loads can be divided into two categories based on the duration of the loading: sustained loads and occasional loads. Sustained loads are expected to be present throughout the operation: pressure and weight. Occasional loads are present at infrequent intervals vessel motion (inertia load) and (wind load) from different directions. These load cases are taken from SBM's Piping Stress Specification from the Cidade de Paraty project.

7.2.5 Inertia Load

The inertia load has been considered during transportation, operation and survival. For stress analysis the transit case has a neglible effect on equipment and support loading as the condition is short term with piping empty and not subject to additional thermal loads. Maximum acceleration from the survival condition is therefore used for stress calculation purposes (Design Environmental Condition). For the process deck the following maximum accelerations are used. That is not to say that the gas processing module will be in operation with these kind of accelerations, but if the piping system stays intact during these conditions it is bound to survive in normal operating conditions. When developing load cases it will be assumed that these accelerations can come from two directions and thus two vectors are created named U1 and U2.

Direction	**Accelerations [g's]**
Longitudinal	0.10
Transverse	0.26
Vertical	0.28

7.2.6 Wind Loads

The ten minute wind velocity at a reference elevation of 10m with a return period of 100 years is given as 34.0 (m/s). Based on statistics for the intended area. This value is used as the maximum design wind speed for stress analysis on exposed pipe at elevation 100.000 (process deck). For higher elevations the wind speed is profiled in accordance with the following table. Where **Ch** is the height coefficient. It is assumed that the direction of the wind can be in the X (longitudinal) and Y (transverse) direction and that a reversal can take place. This creates four options for the wind load WIN1, WIN2, -WIN1 and -WIN2.

Velocity (m/s)	**Elevation (mm)**	**Ch**
34.00	100,000	1.00
34.00	104,800	1.00
40.12	120,000	1.18
44.54	135,500	1.31

Using this table the pressure exerted on the piping is given by the following formula.

$$P_{wind} = 0.610 C_s C_h V_{ref}^2 (N/m^2) \tag{7.1}$$

7.2.7 Load Case Table

The following table displays the load cases to which the piping system is subjected. Load cases are built up using the prementioned loading conditions, using different combinations the system is exposed to all loads that can be experienced during lifetime of the system.

Loadcase	Combination	Snubbers	Description
1	(HGR) W	-	
2	(HGR) W+D1+T1+P1	-	
3	(OPE) W+D1+T1+P1+H	-	
4	(OPE) W+D2+T2+P1+H	-	
5	(OPE) W+D3+T3+P1+H	-	
6	(OPE) W+D1+T1+P1+H+U1+WIN1	active	
7	(OPE) W+D1+T1+P1+H+U1+WIN2	active	
8	(OPE) W+D1+T1+P1+H+U2-WIN1	active	
9	(OPE) W+D1+T1+P1+H+U2-WIN2	active	
10	(OPE) W+D2+T2+P1+H+U1+WIN1	active	
11	(OPE) W+D2+T2+P1+H+U1+WIN1	active	
12	(OPE) W+D2+T2+P1+H+U2-WIN1	active	
13	(OPE) W+D2+T2+P1+H+U2-WIN2	active	
14	(OPE) W+D3+T3+P3+H+U1+WIN1	active	
15	(OPE) W+D3+T3+P3+H+U1+WIN2	active	
16	(OPE) W+D3+T3+P3+H+U2-WIN1	active	
17	(OPE) W+D3+T3+P3+H+U2-WIN2	active	
18	(SUS) W+P1+H	-	sustained loadcase
19	(HYD) WW+HP	-	hydrotest loadcase
20	(OCC) L20=L6-L3	-	occasional effect
21	(OCC) L21=L7-L3	-	occasional effect
22	(OCC) L22=L8-L3	-	occasional effect
23	(OCC) L23=L9-L3	-	occasional effect
24	(OCC) L24=L18+L20	-	occasional code stress
25	(OCC) L25=L18+L21	-	occasional code stress
26	(OCC) L26=L18+L22	-	occasional code stress
27	(OCC) L27=L18+L23	-	occasional code stress
28	(EXP) L28=L3-L18	-	expansion design hot
29	(EXP) L29=L4-L18	-	expansion design cold
30	(EXP) L30=L5-L18	-	expansion operating
31	(EXP) L31=L3-L4	-	expansion range

7.3 Results Analysis

This section presents the results of the analysis based on the CAESAR2 model. Due to the large quantity of data that is computed by the model, the most important results will be summarised. For a detailed presentation of the model results please refer to the appendix of the report, an overview of the model is given below.

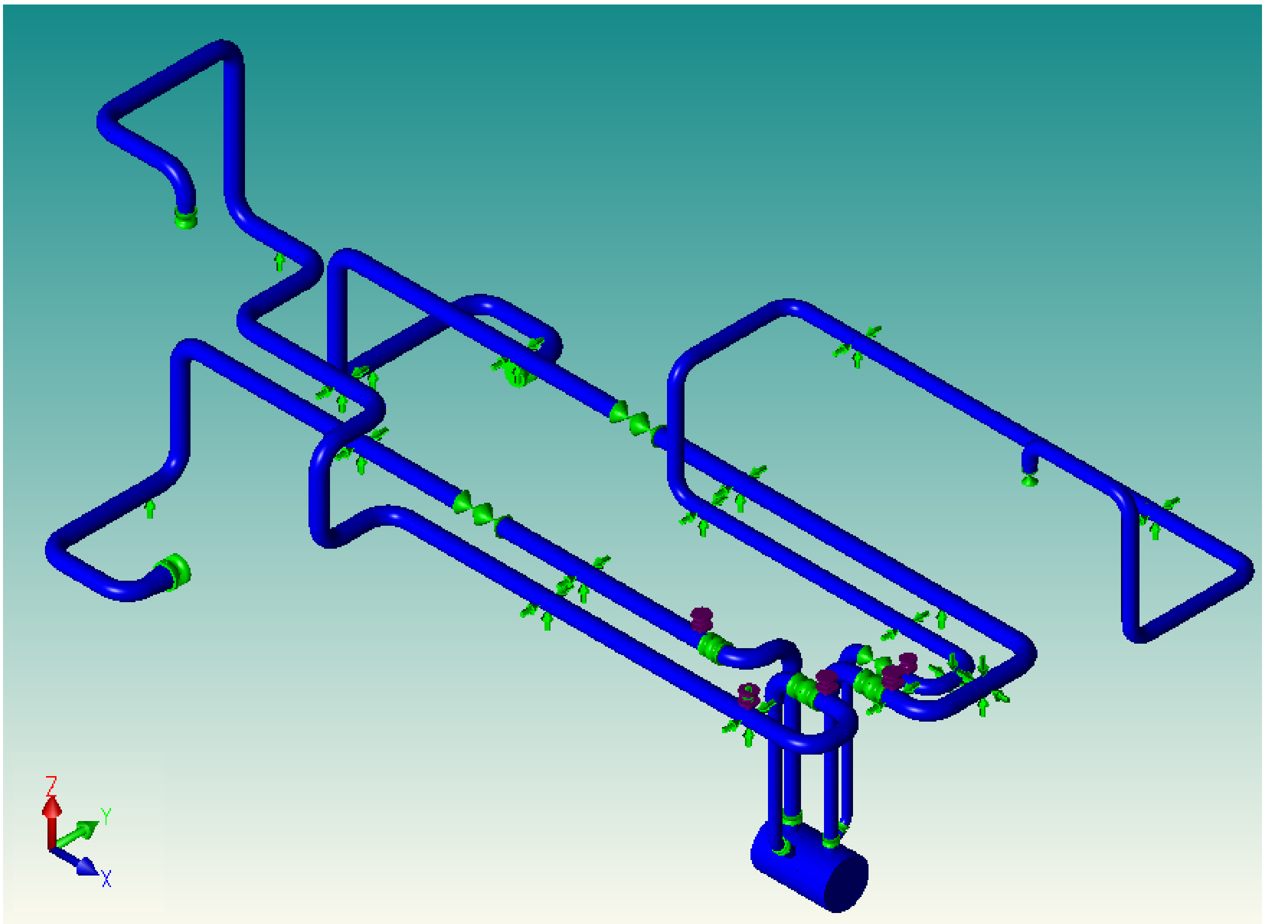

Figure 7.1: CAESAR2 Model

As stated earlier lines were modeled individually to reduce complexity of the model. Figure 7.2 is an image the second discharge line. Nozzle loads are combined later in the API617 calculation.

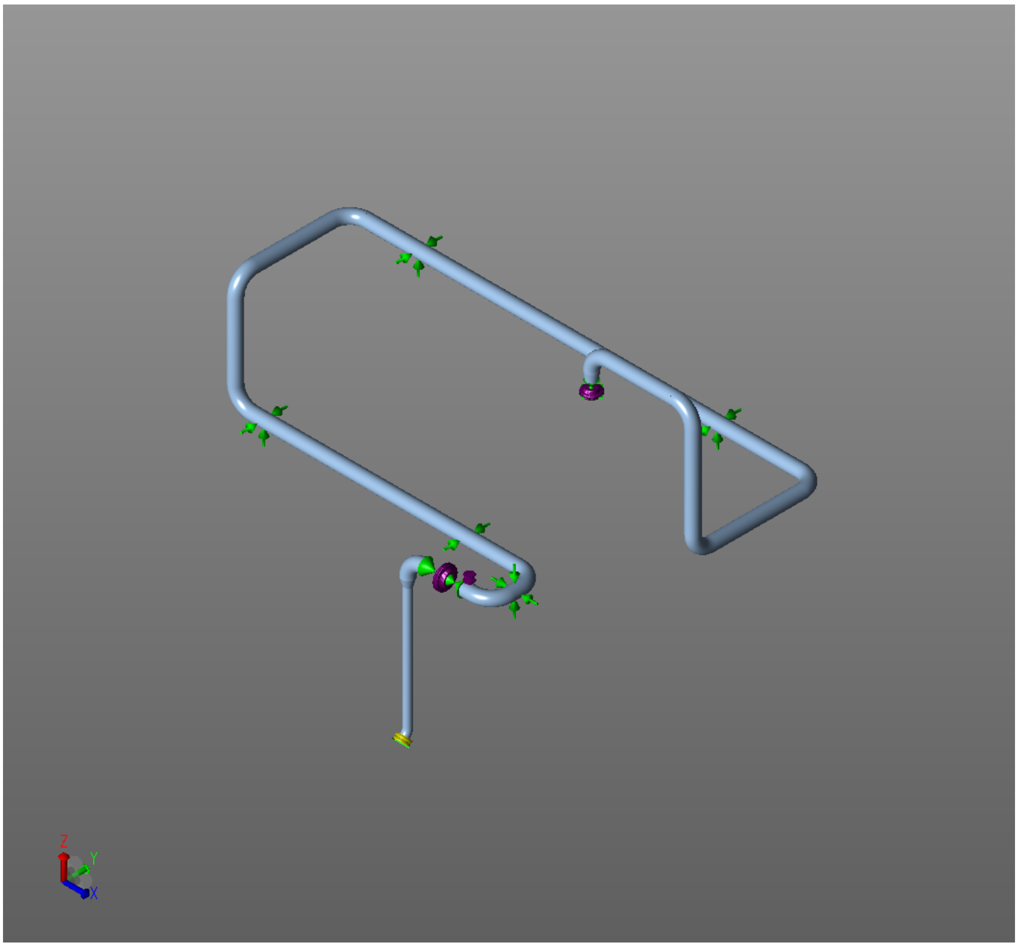

Figure 7.2: Discharge 2 Line

7.3.1 ASME B31.3 Code Compliance

To comply with ASME B31.3 code requirements the beneath stated conditions need to be met. CAESAR2 will test the resulting stresses generated by each load case against the following criteria, depending on the load case category. For every load case the stresses were below the allowable stress, thus the system complies to the code for stress. This is not surprising, since the system isolation loops creates freedom for thermal growth, stress relief, and is properly supported. Therefore secondary, expansion stresses are not very high also the system is relatively 'cold' from a pipe stress point of view.

Sustained Stress
$S_{sustained} < Sh$ (Allowable stress at design temperature)
Displacement Stress
$S_{expansion} < Sa$ (Allowable stress range)
Occasional Stress
$S_{occasional} < 1.33Sh$

The table below gives the code stress for the highest temperature and pressure line to give the reader an idea of the stresses present. As stated earlier the three types of stresses are assessed independently. The first stresses in the table are the sustained stresses, HYD is for the hydrotest of the system water weight and hydrotest pressure. Then there are the occasional stress components due to environmental conditions. Lastly we see the expansion stresses. The expansion stresses are quite low in comparison with the occasional and sustained stresses.

Case	Node	Code Stress N/mm^2	Allowable Stress N/mm^2	Code Stress Ratio
18 &(SUS) W+P1+H	4715	64.9	137.9	47.1
19 &(HYD) WW+HP	4715	81.6	241.3	33.8
24 &(OCC) L24=L18+L20	4560	67.1	183.4	36.6
25 &(OCC) L25=L18+L21	4560	67.5	183.4	36.8
26 &(OCC) L26=L18+L22	4560	68.9	183.4	37.6
27 &(OCC) L27=L18+L23	4560	69	183.4	37.6
28 &(EXP) L28=L3-L18	4720	19.9	282.1	7.1
29 &(EXP) L29=L4-L18	4399	9.9	290.8	3.4
31 &(EXP) L31=L3-L4	4399	27.8	282.1	9.9

7.3.2 API617 Nozzle Loads

A bigger challenge for the piping system is the interface with the centrifugal compressor, to meet the API nozzle load requirements. For all relevant load cases the individual and combined load cases are in the allowable range. Table beneath shows the load cases and the results percentage wise of the combined nozzle loads. Under normal operating conditions combined nozzle loads are in the 'safe' zone. Loadcases 14 to 17 are all occasional load cases based on the design environmental conditions. Though the compressor will not be operating under these conditions, nozzle load compliance under these conditions proves that the system is properly designed.

Loadcase	**Combination**	**Combined Nozzle Loads %**
5	(OPP) W+D3+T3+P1+H	34,55%
14	(OPE) W+D3+T3+P3+H+U1+WIN1	91.29%
15	(OPE) W+D3+T3+P3+H+U1+WIN2	93,35%
16	(OPE) W+D3+T3+P3+H+U2-WIN1	79.70%
17	(OPE) W+D3+T3+P3+H+U2-WIN2	90.43%

7.3.3 Equipment Loads

According to SBM general Piping Stress Specification, equipment nozzle loads are checked at design temperatures. If loads for an individual direction are overstressed, the resultant forces F_r $\sqrt{F_x^2 + F_y^2 + F_z^2}$ and moments M_r $\sqrt{M_x^2 + M_y^2 + M_z^2}$ are checked against the allowable resultants. If the resultants are below the allowables the nozzle load evaluation is passed. Following tables present results for nozzle loads.

Scrubber 1

Condition	Fx (N)	Fy (N)	Fz (N)	Fr(Nm)	Mx(Nm)	My(Nm)	Mz(Nm)	Mr(Nm)
Allowable	38345	46965	46965	76690	67450	47660	47660	95325
3 (OPE)	21724	-15336	-14450	30264	-15034	-2026	-27053	31016
4 (OPE)	-21577	14790	-5517	26735	18407	1887	24874	31002
6 (OPE)	24036	-13011	-16115	31729	-13330	-3022	-25489	28922
7 (OPE)	24386	-13707	-15909	32182	-12971	-3394	-26158	29394
8 (OPE)	19279	-17715	-19695	32763	-15355	93	-28625	32483
9 (OPE)	18928	-17020	-19902	32312	-15715	466	-27956	32074
10 (OPE)	-19204	17024	-6990	26598	20380	592	26330	33301
11 (OPE)	-18854	16328	-6784	25848	20739	219	25660	32994
12 (OPE)	-23961	12320	-10571	28942	18354	3707	23193	29808
13 (OPE)	-24312	13016	-10777	29608	17995	4079	23862	30164
	passed 63.60%	passed 36.25%	passed -11.75%	passed 42.72%	passed 30.75%	passed 8.56%	passed 55.25%	passed 34.93%

Scrubber 2

Condition	Fx (N)	Fy (N)	Fz (N)	Fr(Nm)	Mx(Nm)	My(Nm)	Mz(Nm)	Mr(Nm)
Allowable	41830	51235	51235	83665	64465	45585	45585	91169
3 (OPE)	-33611	-23599	-7953	41831	-17830	4546	38924	43054
4 (OPE)	27013	17794	980	32362	21819	-1977	-30608	37641
6 (OPE)	-30742	-21841	-10691	39197	-18240	2725	37315	41624
7 (OPE)	-30912	-22450	-11168	39803	-18730	2099	37876	42306
8 (OPE)	-36419	-25803	-7761	45303	-14145	7322	40923	43913
9 (OPE)	-36249	-25194	-7285	44741	-13654	7948	40362	43344
10 (OPE)	29882	19553	-1758	35754	21410	-3798	-32218	38869
11 (OPE)	29882	19553	-1758	35754	21410	-3798	-32218	38869
12 (OPE)	24205	15590	1172	28815	25505	800	-28609	38336
13 (OPE)	24375	16199	1649	29313	25995	1426	-29171	39099
	passed 71.44%	passed 38.16%	passed 3.22%	passed 54.15%	passed 40.32%	passed 17.44%	passed 89.77%	passed 48.17%

HEX 1

Condition	Fx (N)	Fy (N)	Fz (N)	Fr(Nm)	Mx(Nm)	My(Nm)	Mz(Nm)	Mr(Nm)
Allowable	22920	28075	28075	45845	31555	22315	22315	44628
3 (OPE)	-8850	2633	-16336	18765	22519	-1976	5107	23175
4 (OPE)	-15522	142	2999	15810	-5942	-21689	-3228	22719
6 (OPE)	-15079	11182	-14268	23579	7841	-8267	-1291	11467
7 (OPE)	-15457	12426	-15632	25252	7841	-9910	-2314	12847
8 (OPE)	-10280	-5458	-19652	22840	38175	-4737	10933	39991
9 (OPE)	-9901	-6702	-18289	21850	38174	-3094	11955	40122
10 (OPE)	-21916	8814	5035	24153	-20796	-28281	-9741	36430
11 (OPE)	-22295	10058	3671	24733	-20796	-29924	-10763	37997
12 (OPE)	-17117	-7826	-349	18824	9538	-24751	2483	26641
13 (OPE)	-16739	-9070	1015	19065	9537	-23108	3506	25243
	passed	passed	passed	passed	overstressed	overstressed	passed	passed
	-38.61%	44.26%	17.93%	55.08%	120.98%	-8.86%	53.57%	89.90%

HEX 2

Condition	Fx (N)	Fy (N)	Fz (N)	Fr(Nm)	Mx(Nm)	My(Nm)	Mz(Nm)	Mr(Nm)
Allowable	17865	21880	21880	35730	19950	14110	14110	28217
3 (OPE)	-9350	-7490	4700	12869	-15609	8105	-2226	17728
4 (OPE)	-15211	1618	1244	15347	699	19075	1039	19116
6 (OPE)	-12742	-4944	7595	15636	-12865	11190	-2985	17310
7 (OPE)	-12668	-4246	7061	15112	-12375	10995	-3512	16922
8 (OPE)	-12481	-10603	2771	16610	-20021	11622	-791	23163
9 (OPE)	-12555	-11302	3305	17213	-20511	11816	-263	23673
10 (OPE)	-19110	4196	4253	20022	3230	23276	3	23499
11 (OPE)	-19110	4196	4253	20022	3230	23276	3	23499
12 (OPE)	-18728	-1471	-598	18795	-3876	23441	2263	23867
13 (OPE)	-18800	-2170	-64	18925	-4365	23631	2792	24192
	fail	passed	passed	passed	fail	overstressed	passed	passed
	-52.34%	19.18%	34.71%	56.04%	16.19%	167.48%	19.79%	85.74%

7.3.4 Flange calculations

In addition to thermal and dead weight loading, the influence of vessel motion and acceleration increase the risk of flange leakage. Special attention is given to the compressor lines as these contain flammable gasses.
Flange leakage is checked using the quivalent pressure method derived from:

$$P_e = \frac{4F}{\pi G^2} + \frac{16M_b}{\pi G^3} \tag{7.2}$$

Where:
F= Axial Force (N)
G= Diameter of effective gasket reaction (mm)
M_b= Maximum bending moment (Nmm)

As per DNV-RP-D1014 section 3.8.2, the total pressure shall not exceed following criteria:

$$P_t(TotalPressure) = P_{equivalent} + P_{design} < 1.5 \cdot \text{ANSI B16.5 Rating} \tag{7.3}$$

This analysis shows that all flanges are below this allowed value, thus no further analysis is required for the flanges in the system.

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Conclusion

Here we shall review if the main objective and goals of the assignment have been accomplished. As a reminder the objective of the project is to set a standard philosophy for the compressor pipe routing to be used as a reference for future projects. During the assignment an appropriate method has been found to route and support the piping which is directly at the interface of the compressor. The key to achieving this is by isolating the compressor from the rest of the piping system. This has been achieved by designing an 'isolation loop' round the compressor that is separated from the rest of the system by an axial stop. This axial stop has been positioned in line with the fixed point on the compressor to create equal growth on both sides of the loop. This isolation loops functions well for the normal load cases and absorb thermal growth of the system. However it can reate an additional problem for the compressor nozzles under harsh environmental conditions. To solve this problem, inertia load on the loops has been reduced by installing snubbers. Variable spring hangers are always required to allow vertical growth but allow continuous vertical support. Outcome of this routing is that the API617 Centrifugal Compressor Nozzle Loads are satisfied, even for design environmental conditions.

Design-wise and in CAESAR2 this approach works, but it is recommended that this approach is to be tried in a PDMS model and on a physical module, to find out if it works in practice. The removable spool necessary for maintenance, hydrotesting and compressor nozzle alignment is still not ideal. Recommendation is made for further study and investigation, to see if a more practical solution can be reached. This spool could be pressure tested separately and installed afterwards. Further study could also be done into optimising the rest of the compressor module layout. Biggest lesson from this investigation is that engineering of FPSO modules requires a multidisciplinary approach. Ideally one would not make compromises on his design but sometimes this necessary to fulfil requirements of another discipline. Nevertheless safety requirements should never be compromised due to higher engineering and construction costs.

9

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Appendix

The following list shows the content of the appendix. The reader is strongly advised to refer to the appendix whilst reading the report. For practical reasons large A3 size drawings are found in this appendix. Preview of the contents of the calculations are given behind cover sheets. Considering the environment and cost of paper, the reader is kindly asked to refer to the included CD-ROM for full calculation reports (hundreds of pages). Most important results from calculations have been included within the report in a summarised format.

1. Vendor Drawings
2. Design Drawings
3. Stress Isometrics
4. Code Compliance and Restraints
5. API617 Reports
6. CAESARII Input Echo
7. CAESARII Files