2018

THESIS REPORT:

Selecting alternatives for Viking-Johnson couplings

(Sizes 6” through 24”)

PHASING OUT OF VIKING-JOHNSON COUPLINGS FROM THE PETROCHEMICAL INDUSTRY

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Preface

To the reader,

This report finalizes 17 weeks of hard work and dedication. It proves the level of competence that I have acquired since the start of my studies three and a half years ago. The combination of work and studies has been an exhausting yet valuable experience. During my time working at Vicoma Engineering I have grown both as an individual and as a professional. This has been possible, in part, by the constant support, constructive feedback and guidance that I have received from mentors, team leaders and colleagues. Therefore, I would like to express my most sincere gratitude and appreciation to those involved in the process.

In this part I would like to acknowledge and give special thanks to the following persons:

- **Theo Gunneweg** for his mentoring and guidance during my time working at Vicoma since the start of my studies and during the graduation phase;
- **Peter de Jonge** for giving me the opportunity of working at Vicoma in parallel to my studies and for his constant support throughout the years;
- **Paul Souren** for his guidance and counseling throughout the graduation phase;
- **Chris Kuiper, Rutger Schuddebeurs and Lars Hartemink** for advising me about the calculations for pipe stress and flexibility;
- **Aad Keijzer** and **Alex Scherpehuizen** for taking the time to review my report and provide me with useful feedback.

The selection matrix tool developed within this project should be used as a reference guide to aid in the selection of alternatives in different scenarios. Therefore, I believe that the same approach used within this report, could be applied to other problems that require the selection of alternatives for the discontinuation of piping couplings. In this report the phasing out of the Viking-Johnson couplings are used as an example, since this is a trend that has been developing recently within the industry. The intention of this report is not to boycott these couplings but rather to provide several alternatives and more importantly a useful tool for the quick selection of these alternatives.

My mayor achievement in this dual study is not the degree itself but getting to know myself better, my capabilities and my limits. Over the years I have become a more assertive and professional individual. I will continue to grow professionally as an engineer, through experience and follow-up studies. The knowledge and skills that I have acquired through the years will most definitely help me achieve these personal goals.

Yours truly,

Carlos Spagnol

Nomenclature

No.	Name	Description
1	Anchor	Fixed pipe support that prevents displacements of the pipe in any direction.
2	Bellows	Expansible slides that provide flexibility in a joint.
3	Bund wall	Retaining wall around a tank pit used to retain spilling.
4	Elbow	Pipe bend used to change the direction of a pipe.
5	Equipment	Rotating: e.g. pumps, compressors and turbines. Static: e.g. heat exchangers, boilers and furnaces.
6	Joint	The connection at the end of a pipe that ensures tight sealing.
7	ASME	American Association of Mechanical Engineers
8	API	American Petroleum Institute
9	PGS	Publicatiereeks Gevaarlijke stoffen

List of symbols

Symbol	Description	SI units
SCH	Schedule	[--]
P_{op}	Operational internal pressure	[Pa]
P_d	Design internal pressure	[Pa]
σ_y	Yield stress	[N/m^2]
t_d	Minimal design thickness	[m]
d_0	Outside pipe diameter	[m]
d_i	Inside pipe diameter	[m]
Q	Quality factor	[--]
Y	Thickness correction coefficient	[--]
C	Mechanical corrosion allowance	[m]
Υ	Specific weight	[kg/m]
ρ	Material density	[kg/m^3]
A	Cross-sectional area of the pipe	[m^2]
I_{xy}	Area moment of inertia	[m^4]
y	Resultant of total displacement strains	[m]
L	Developed length of piping between anchors	[m]
U	Straight line between anchors	[m]
E	Modulus of elasticity	[N/m^2]
G	Shear modulus of elasticity	[N/m^2]
F	Force	[N]
M	Moment	{Nm]
ϑ	Angle of rotation	[rad]
k	Beam stiffness	[N/m]
x	Resulting positions of the nodes	[m]
τ	Shear stress	[N/m^2]
S	Longitudinal stress	[N/m^2]
Z	Polan moment of inertia	[m^4]

Summary

The local Dutch legislations (PGS-29) regarding the above-ground storage of flammable products have become stricter. This has led to the phasing out of Viking-Johnson couplings that are located inside of tank pits. These couplings are prone to leakage and they use rubber seals which are now prohibited by PGS-29. In the case of fire, the rubber seals could melt down, allowing the fire to reach the product inside of the pipeline which can lead a full-blown disaster.

The aim of this graduation thesis is twofold:

- To create design options as alternatives for the Viking-Johnson couplings that comply with both the PGS-29 and the international process piping code (ASME B31.3) for the oil- and gas industry;
- To create a selection matrix tool in Excel for selecting the best alternative in any given tank connection scenario.

The thesis approached this problem by first looking into the literature of piping design and pipe stress to find design solutions that could be used as alternatives for the Viking-Johnson couplings. This led to the formulation of design criteria that complied with the PGS-29 guidelines, the ASME B31.3 codes and all underlying codes specified therein. The found solutions can be summarized as following:

- A straight pipeline to replace the coupling;
- The use of elbows to create L-shaped pipelines and pipe loops to increase the flexibility;
- The use of metallic expansion joints to replace the coupling.

A stress- and flexibility analysis was performed with Caesar II and Nozzle Pro to determine the minimum dimensions required of each alternative to prevent excessive nozzle stresses. The FEM analysis of Nozzle Pro concluded that pipelines larger than 16” became too heavy which lead to excessive stresses on the tank nozzle. To address this issue, it was recommended to design special supports to counteract this weight. The metallic expansion joints did not have this issue; they can be used for any tank settlement scenario and for any pipe size because of the inherent flexibility of the bellows. Due to time constraints, the stress- and flexibility calculations cover only pipelines where the point of rotation is located along the longitudinal axis of the tank nozzle. A follow-up project should determine the effects on the tank nozzle in cases where the point of rotation lays at an offset from the longitudinal axis. The effects of tank keeling on the tank connection should also be included in the scope of follow-up projects.

The selection matrix tool was created using Excel. The results of the stress- and flexibility analysis were saved into the database of this model. The scope of the model depends solely on the completeness of the database. Vicoma can fill the database through follow-up projects. To do this, it is essential to follow the analysis procedure established in the thesis for other pipe specs, having different process conditions and with different tank settlements. The selection matrix tool in its current state is able to select the best alternative for tank settlements of 200 mm, with design conditions of 18,66 bar(g) and 68 ^{0}C by using the Vic C-150 pipe specs (carbon steel – 150#).

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Introduction

Since the industrial revolution, oil has become a very lucrative and dangerous business worldwide. Along the years many people have lost their lives due to fires and explosions that have occur inside oil refineries such as the 2015 BP disaster in Texas, USA, where 15 people lost their lives do to an unexpected explosion. Luckily, the lessons learned from previous mistakes have made refineries nowadays much safer. The rules and legislations have become stricter and emphasize on safety above all. An example of this, are the Dutch PGS-29 legislations where it states that materials that are not resistant to fire, such as rubber seals, cannot be used for the transport of flammable product. This is the main reason why the Viking-Johnson couplings are being phased out from the Dutch industry.

The purpose of this thesis is to find design solutions to be used as possible alternatives for the Viking-Johnson couplings that comply with the Dutch PGS-29 legislations and to create a selection matrix tool in Excel for the selection of these alternatives. The scope of the thesis covers only tank pits and flanged connections of vertical cylindrical storage tanks. Vicoma can use the selection matrix as a tool to quickly identify the best possible alternative for the Viking-Johnson coupling(s) currently being used by the client in any given scenario.

This problem will be addressed in three stages:

Stage 1: Literature framework
Literature of piping design and pipe stress for pipelines in the oil- and gas industry will determine which design options can be used as alternatives for the Viking-Johnson couplings. This framework will also determine the design criteria needed to comply with the design codes and legislations.

Stage 2: Research analysis
The second stage is intended for addressing the more technical aspects of the research that are needed for the creation of the selection matrix. Stress- and flexibility calculations are performed using Caesar II and Nozzle Pro to determine the minimum dimensions required for each alternative. The pipe sizes considered range from 6” though 24” since these sizes are common for the transportation of hydrocarbons and for tank connections. The analysis also covers all the technical, maintenance, safety, and financial aspects of each alternative.

Stage 3: Selection matrix set-up
The third stage is the creation of the selection matrix model using Excel. The results of stage two will be saved in the database of the model. In this stage, the benefits of using the model will be elaborated and a list of potential clients within the Netherlands is created.

1. Project background

Vicoma Engineering is a multidisciplinary engineering bureau operating in six locations throughout the Netherlands with headquarters in Rotterdam-Hoogvliet. Vicoma is active in a broad spectrum of industries such as the food sector, pharmaceuticals, oil & gas and the energy industry. The research thesis takes place at the headquarters in the department Mechanical.

The main goal of this research is to establish a standard procedure in the form of a selection matrix tool for selecting alternatives for the Viking-Johnson couplings (see **figure 1.1** for an example). These couplings are being phased out from the oil industry due to strict legislations, such as the Dutch PGS-29, preventing the use of rubber gaskets for the transport of flammable products. In the case of fire, these gaskets can melt down, causing flammable product to leak out and thus fueling the fire further.

Figure 1.1: Viking-Johnson coupling - Dismantling Joint type (Crane Co., 2018)

Vicoma already has some experience in projects regarding the replacement of Viking-Johnson couplings. However, Vicoma does not have a standard method for approaching these types of projects, nor possess a list with possible alternatives for these couplings. Therefore, it would be in the best interest of Vicoma to obtain such a tool, or method, in order to strengthen its market position when dealing with such projects. Viking-Johnson couplings are very popular in the Piping industry for the handling of fluids due to the wide variety of couplings available in the market and their mount simplicity. However, these couplings make use of rubber gaskets to seal off the coupling connections and the pipe.

The following examples provide a better understanding of the failure mechanism (leakage) of these couplings:

- Settlement of the tank can cause bending of the pipe, resulting in improper closing of the seals;
- Thermal expansion of the pipe can cause excessive stress on the coupling, which in turn can lead to material failure;
- Hydraulic shock of the fluid being transported can create high tensile stresses on the coupling rods, which can cause material deformation beyond its yield value, resulting in less pressure being exerted on the seals or even total failure of the coupling;
- Pressure build-up can cause the coupling rods to elongate beyond its yield value;
- The rubber seals are prone to degrade with time and by being in contact with different products through their service lifecycle;
- Rubber seals are not resistant to heat and thus can melt during a fire inside the tank pit.

2. Alternatives for Viking-Johnson couplings

This chapter explains which alternatives can be used to replace the Viking-Johnson couplings. Alternative 1 uses a straight pipe (§**2.1**), alternative 2 uses elbows to increase the flexibility (§2.**2**) and alternative 3 uses metallic expansion joints (§**2.3**). Metallic flexible hoses were considered; however, these are prohibited by the Dutch legislations (§**2.4)**. In **appendix I**, a short summary of the theory of piping is given to better understand the terminology used in this document.

2.1 Alternative 1: Straight pipe

The first alternative uses a straight pipe to replace the Viking-Johnson coupling. The pipeline needs to be flexible enough to prevent overstress of the tank nozzle due to tank settlement; flexibility increases with pipe length. However, thermal expansion also increases with pipe length which leads to higher nozzle stresses. These effects need to be considered when implementing this idea. A schematic representation of alternative 1 can be seen in **Figure 2.1**.

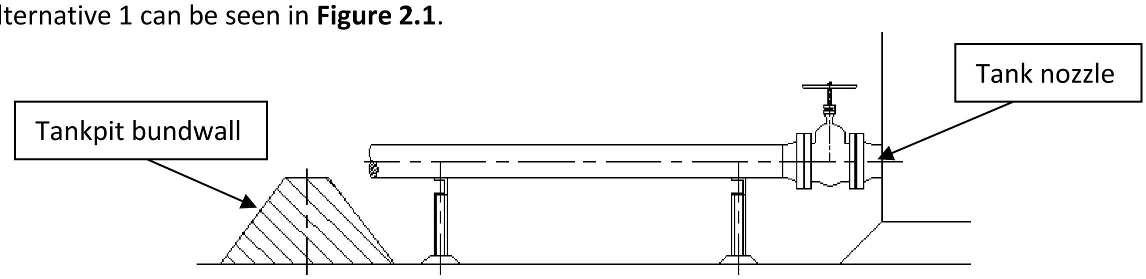


Figure 2.1: Schematic drawing of alternative 1 (Straight pipeline)

2.2 Alternative 2: Increased flexibility with the use of elbows

The second alternative uses elbows to increase the flexibility of the pipeline. The design includes a L-shape pipeline and a pipe loop (U-shape). **Figure 2.2** shows a schematic drawing of the pipe loop. Sometimes the pipe loop is also referred to as an expansion loop due to its ability to absorb thermal expansion of the pipeline. The extra flexibility of this design could be implemented to prevent overstress of the nozzle due to tank settlement. The size of the pipe loop needs to be considered since it requires more space to be installed.

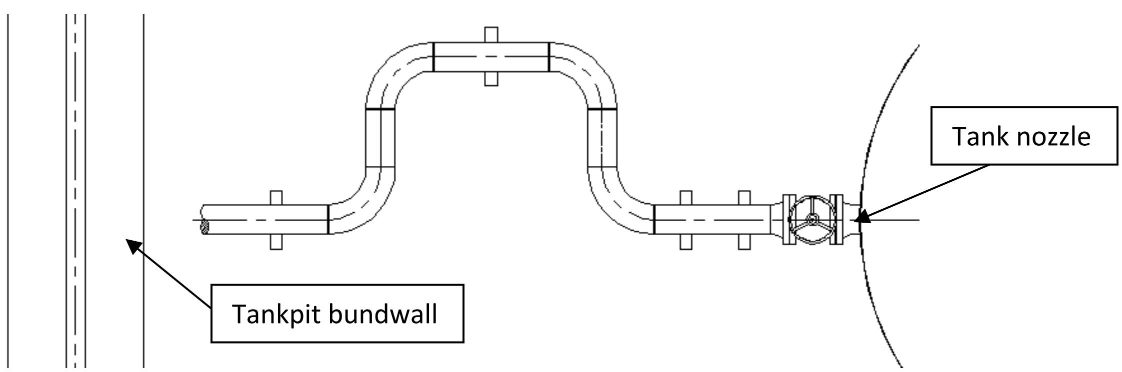


Figure 2.2: Schematic drawing of alternative 2 (Piping-loop)

2.3 Alternative 3: Metallic expansion joints

The third alternative uses metallic expansion joints to increase the flexibility of the pipeline (see **figure 2.3**). There are many different types of expansion joints available, each with its own purpose. For this reason, this section will try to summarize all of these expansion joints and their functions.

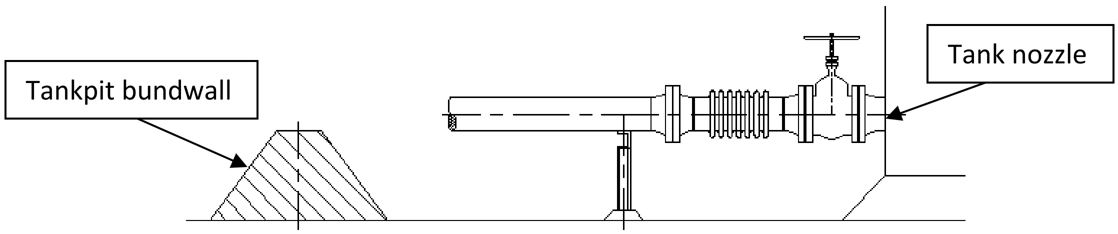
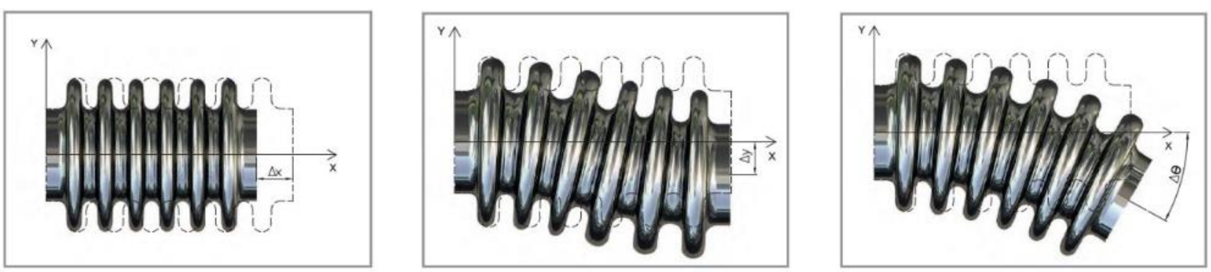


Figure 2.3: Schematic drawing of alternative 3 (Metallic expansion joint)

Metallic expansion joints gain their flexibility from the bellow arrangement within the design (*Belman Group, 2017*). **Figure 2.4** shows the different types of bellow displacements. These displacements are categorized as follows:

- Axial displacements - in the direction of the longitudinal axis of the bellow;
- Lateral displacements - perpendicular to the longitudinal axis of the bellow;
- Angular displacements - rotation of the bellow at the midpoint of the bellow;



(a) Axial displacement (b) Lateral displacement (c) Angular displacement

Figure 2.4: Types of bellow displacements, (Belman Group, 2017)

Axial displacement

Created by thermal expansion (or compression) of the pipeline. A single bellow configuration is used to absorb axial displacements.

Lateral displacement

A parallel offset is created between the ends of the bellows. The bellows are limited to small lateral displacement due to high shearing stresses. A design configuration using two bellows separated by a straight pipe allows for a larger lateral displacement; the displacement gain is proportional to the length of this straight pipe.

Angular displacement

The longitudinal axis of the bellow is curved as an arc from its initial position. This displacement is usually guided with rods connected by a hinge mechanism to prevent combined displacements.

Metallic expansion joints are categorized into two main groups:

- Flanged expansion joints;
- Welded expansion joints.

Flanged expansion joints

Flanged connections allow the expansion joints to be maintained or inspected easily by disconnecting the flanges. The flanges are disconnected by removing the bolts holding the flanges together. However, the gaskets, the bolts and the nuts need to be replaced by new ones every time the flanges are disconnected to ensure the integrity of the connection.

Welded expansion joints

The welded ends prevent the expansion joints to be easily removed. The only way to remove these joints is to cut through the pipe. Welded expansion joints are mainly used for the transportation of dangerous gasses to prevent leakage.

2.4 Alternative 4: Metallic hoses

This idea uses flexible metallic hoses that are specifically designed for industry applications (see **figure 2.5**). Metallic hoses have the highest flexibility when compared to all the alternatives mentioned. They can be used to overlap very short distances and the effects of thermal expansion can be neglected. However, according to PGS-29 (2016), the use of hoses is prohibited inside of tank pits for the transport of product. For this reason, the use of metallic hoses is considered in this report.

Figure 2.5: Metallic hose (Image retrieved from Shutterstock (2018) under a free to share license)

3. Design

This chapter starts with basis of design (BOD) which provides reasoning for the design choices made (§**3.1**). The design criteria is composed according to the ASME B31.3 international code for process pipelines in the oil- and gas industry (§**3.2**). This chapter ends by presenting the designs (§**3.3**).

3.1 Basis of design

3.1.1 Pipe Specs (Material data sheets)

Pipe specs are material data sheets used for the selection of piping components. Every fluid under certain process conditions has its own pipe specs. These were at one point determined through extensive calculations and analysis. Vicoma has its own pipe specs for the transportation of hydrocarbons; Vic C-150 (Vicoma Carbon Steel 150#). These pipe specs are confidential, therefore only a fraction is shown in this report. The following information was retrieved from Vic C-150:

Services:	Hydrocarbons		
Temperature range:	0 – 400 ^{0}C		
Corrosion allowance:	1.6 mm		
Pressure rating:	150#		
Material type:	Carbon Steel	Yield Stress	Young's modulus
- Pipes	ASTM A106-B	240 MPa	203 GPa
- Fittings	ASTM A234-WPB	240 MPa	200 GPa
- Flanges	ASTM A105N	250 MPa	190 GPa

Appendix II shows Ashby diagrams with some material properties. According to these diagrams, Carbon Steel is the best material in terms of modulus of elasticity, price per kg and thermal expansion. Stainless steel has better corrosion properties but has a thermal expansion coefficient twice that of Carbon Steel and is more expensive. These are the main reasons why Carbon Steel is widely used in the oil and gas industry.

Furthermore, depending on the chemical composition of the product being transported and the type of piping component, specific Carbon Steel alloys are chosen. For hydrocarbons, the materials shown in Vic C-150 meet the criteria. These materials are specified by the ASME codes. An example is shown in **table 3.1**. This table was retrieved from ASME B13.5 (2017) and shows the pressure-temperature rating of a 150# Carbon Steel flange (ASTM A105N). As can be seen in the table, a 150# flange made from A105N Carbon Steel can withstand a maximum of 17,7 bar(g) at 100 ^{0}C. The higher the design temperature, the lower the design pressure allowed (and vice versa). Interpolation can be used to obtain intermediate values. The scope of the thesis covers carbon steel for the transport of hydrocarbons with a design temperature of 68 ^{0}C and a design pressure of 18,66 bar(g); this shows that Vic C-150 can be used. This is the usual temperature for transport lines that do not require heating. This temperature also accounts for extreme heating by direct sunrays of the sun.

Temperature [^{0}C]	**-15**	**50**	**100**	**150**	**200**	**250**	**300**	**325**	**350**	**375**	**400**
Pressure [Bar(g)]	19,6	19,2	17,7	15,8	13,8	12,1	10,2	9,3	8,4	7,4	6,5

Table 3.1: Pressure-Temperature rating acc. to ASME B16.5.

3.1.2 Hydrocarbons

A hydrocarbon is a chemical compound composed only of the elements carbon (C) and hydrogen (H). The hydrogen atoms are attached to the carbon framework to form many configurations. Petroleum and natural gas are typical examples of complex hydrocarbon mixtures found in nature. Petroleum is heated in distillation towers to separate the hydrocarbon mixture and form refined products such as fuels, lubricants and raw materials for plastics, fibers, rubbers, solvents, explosives and industrial chemicals (Encyclopedia Britannica, 2018).

Hydrocarbons are classified as aliphatic or aromatic (see in **figure 3.1**). Aliphatic hydrocarbons are obtained from the fats and oils (for example Petroleum). These are divided into alkanes (single carbon bonds), alkenes (carbon-carbon double bonds) and alkynes (carbon-carbon triple bonds). Aromatic hydrocarbons are obtained by chemical breakdown of pleasant-smelling plant extracts. These are classified as arenes (with a benzene ring) or as nonbenzenoids (without a benzene ring).

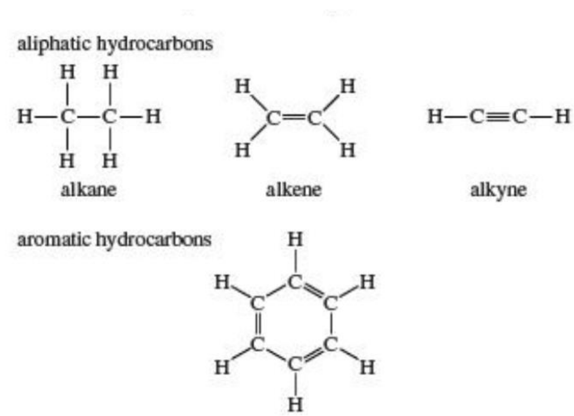


Figure 3.1: Classification of hydrocarbons (Encyclopedia Britannica, 2018)

Physical properties

The more complex the molecule the higher the melting and boiling point. Gasses such as methane, ethane, propane and butane have boiling points below 0 ^{0}C. These gasses are stored in pressurized spheres. The high pressure transforms these gasses into liquids. The majority of the hydrocarbons are liquids at room temperature such as automobile fuels (gasoline and diesel). Those with high melting points are considered solids and are usually used as lubricants or as raw materials for plastics and rubbers.

Relation to PGS-29

Most hydrocarbons are classified as highly flammable. According to PGS-29 (2016), these fall under classification 1 and 2 with flashpoints between 0 and 55 ^{0}C. This means that valves and pipelines need to be made fire resistant by preventing the melt down of gaskets and seals.

Scope of Vic C-150 specs

The pipe specs Vic C-150 covers materials for the transportation of hydrocarbons with operating temperature of 0 through 400 ^{0}C. This temperature range covers all hydrocarbon types except compressed gases. These gasses require a pressurized sphere for storage and the pipelines need to be of a Carbon Steel type that can withstand very low temperatures (<<0 ^{0}C).

3.1.3 Corrosion

In the piping industry, corrosion can be classified in three main groups:

- Internal erosion;
- Environmental corrosion (e.g. Pitting corrosion);
- Galvanic corrosion.

Internal erosion

This type of corrosion occurs inside of the pipeline do to the erosion that results from the movement of substances through the pipe. Each pipeline must have an extra layer of thickness (Mechanical corrosion allowance). This extra thickness must be calculated for every process.

Environmental corrosion

Pipelines and support structures are usually in the open and therefore prone to corrosion do to the environment. Such corrosion can be the cause of moisture, salty environments, oxidation or exposure to industrial chemicals. To delay or prevent the corrosion, certain measures are taken, such as applying corrosion resistant coatings. The corrosion resistant coatings act as a temporary barrier on the metal surface and therefore inhibit contact between the metal surface and the corrosive environment.

Galvanic corrosion

This type of corrosion occurs when two dissimilar metals are coupled in a corrosive electrolyte. For example, if stainless steel comes into contact with carbon steel when there is water present connecting the two metals. In such case the water act as the electrolyte conducting electricity between both metals. When a galvanic couple forms, one of the metals becomes the anode and the other one the cathode. The rate of corrosion of the anode metal will accelerate and the rate of corrosion of the cathode metal will decelerate or even stop (depending on the metal and the situation). There are some methods available to prevent galvanic corrosion:

- An isolating Teflon tape is used to isolate the two metals from each other;
- Galvanic cathodic protection (Galvanic CP) connects a sacrificial anode material, such as a block of Zinc, to the material that needs protection. The block of Zinc is more electrochemical reactive than steel and will corrode at a much faster rate. This method relies on the difference in potential between the anode (Zinc) and the cathode (Carbon steel). The greater the potential difference, the better the protection.
- Application of coatings to isolate the contact surfaces. However, the coating could scrape off due to the friction between the pipeline and the support when the pipe is subjected to thermal expansion.

3.1.4 Stress and flexibility requirements

According to *ASME B31.3 (2016)*, piping systems must have sufficient flexibility to prevent the following failures:

- Failure of piping supports due to overstress or fatigue;
- Leakage at joints;
- High stresses at piping valves, tank nozzles or equipment (e.g. pumps, heat exchangers and turbines).

Stresses due to forces and moments

The stresses created by forces and moments are shown in **figure 3.2**.

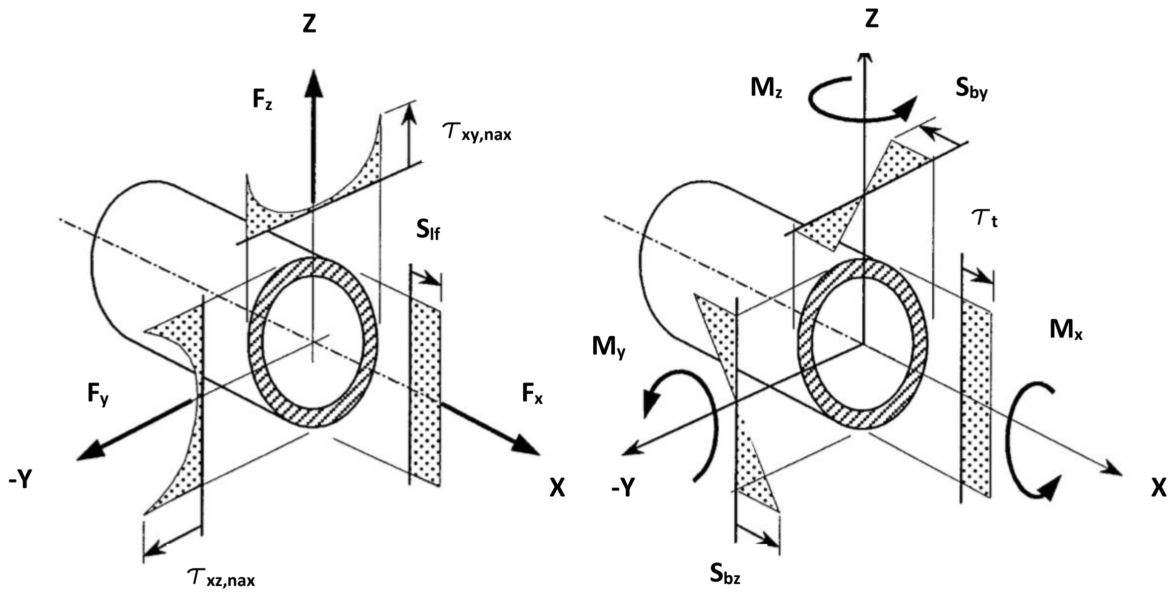


Figure 3.2: Stresses due to forces and moments (Pipe stress engineering, 2017)

The combined stresses due to forces are calculated with equations (**3.1**) and (**3.2**) respectively. The combined moment stresses created by moments are calculated with equation (**3.3**) and the shear stress created by torsion is calculated by equation (**3.4**).

$$S_{lf} = \frac{F_x}{A}, \quad with \quad A = 2\pi r_m t \qquad \textbf{(3.1)}$$

$$\tau_f = 2\frac{\sqrt{F_y^2+F_z^2}}{A} \qquad \textbf{(3.2)}$$

$$S_b = \sqrt{S_{by}^2 + S_{bz}^2} = \frac{1}{Z}\sqrt{M_y^2 + M_z^2} \qquad \textbf{(3.3)}$$

$$\tau_t = \frac{M_t}{2Z}, \quad with \quad Z = \pi r_m^2 t \qquad \textbf{(3.4)}$$

S_{lf} = longitudinal stress [Pa]

F_x = longitudinal force [N]

A = pipe cross-sectional area [m^2]

r_m = radius of thin-walled pipe [m]

t = thickness of thin-walled pipe [m]

τ_f = combined shear stress [Pa]

F_y and F_z = shear forces [N]

S_b = combined longitudinal stress [Pa]

S_{by} and S_{bz} = longitudinal stresses [Pa]

Z = polar moment of inertia [m^4]

M_y and M_z = moments [Nm]

τ_t = shear stress [Pa]

M_t =torsion [Nm]

According to Liang-Chuan and Tsen-Loong (2017), ASME B31.3 states that the sum of the tensile stresses due to pressure, weight and other loads can reach as much as 1.33 times the basic allowable tensile stress of the material at the design temperature. ASME B31.3 does not provide any formula to calculate the magnitude of these stresses. The values of the allowable stresses are retrieved from ASME B31.3, Appendix A, Table A-1M. These tables are grouped by materials and product forms. **Table 3.2** summarizes the basic allowable tensile stresses for pipes, elbows and flanges for Carbon Steel according to the pipe specs Vic C-150.

Component	Material specification	Basic allowable stress, S_h [MPa]	Max. allowable tensile stress, 1.33S_h [MPa]	Max. allowable shear stress, 0.8S_h [MPa]
Pipe	A106 Gr. B	138	183	110
Elbows	A234 Gr. WPB	138	183	110
Flanges	A105N	156	207	124

Table 3.2:Basic allowable stresses for carbon steel at design temperature of 68 ^{0}C (ASME B31.3, 2016)

Equation (**3.5**) summarizes the allowable stress calculation according to ASME B31.3 (2016).

$$S_{lf} + S_b + S_{occc} \leq 1{,}33S_h \quad \textbf{(3.5)}$$

S_{lf} = longitudinal stress created by combined forces [Pa]

S_b = longitudinal stress created by combined moments [Pa]

S_{occ} = occasional longitudinal stress created by an earthquake, wind loads and hydraulic shock[Pa]

S_h = Basic allowable stress [Pa]

According to ASME B31.3 (2016), the maximum allowable shear stress is 0.8 times the basic allowable longitudinal stress of the material, see equation (**3.6**).

$$\tau_f + \tau_t + \tau_{occ} \leq 0{,}8S_h \quad \textbf{(3.6)}$$

τ_f = combined shear stresses created by shear forces [Pa]

τ_t = shear stress caused by torsion [Pa]

τ_{occ} = shear stresses caused by occasional loads [Pa]

The allowable flexibility of a pipeline can be approached with the Kellogg's equation for pipe flexibility as shown in equation **(3.7)**.

$$\frac{d_o y}{(L-U)^2} \leq 208 \quad \textbf{(3.7)}$$

d_o = Outside pipe diameter [mm]

L = Total length of piping between anchors [m]

y = Resultant of total displacement strains [mm]

U = Straight line between anchors [m]

3.1.5 Pipe supports

According to *ASME B31.3 (2016)*, the design of pipe supports shall be able to withstand all concurrently acting loads transmitted into them. Such loads include weight effects, loads by service- pressures and temperatures, vibration, wind, earthquake, hydraulic shock and displacement strains. In any case, the supports must be able to withstand the weight of the pipe plus the weight of water that fills the volume of the pipe during hydrotesting.

The layout and design of pipe supports shall be able to prevent the following:

- Excessive piping stresses (beyond the yield values);
- Leakage at joints;
- Excessive loads and moments at equipment connections;
- Excessive stresses in the supporting elements;
- Resonance with induced vibrations;
- Excessive interference of with thermal- expansion and contraction;
- Unintentional disengagement of the pipeline from the support;
- Excessive sag in the pipeline;
- Excessive distortion or sag of piping subject to creep during repeated thermal cycling conditions;
- Excessive heat flow, exposing the supporting elements to temperatures outside of their design limits.

According to ASME B31.3 (2016), the design and locations of pipe supports may be based on simple calculations and engineering judgements. For this reason, the buckling calculations of the pipe support needs to be calculated with a safety factor of 0.75 on the material yield stress.

3.1.6 Storage tanks and tank connections

According to the *API 653 (2003)* standard from the American Petroleum Institute, settlement of tanks and tank tilts are caused mainly by the following factors:

- Lack of supporting;
- Non-homogeneous geometry or compressibility of the soil;
- Non-homogeneous distribution of loads on the soil;
- Lack of foundation quality;
- Liquefaction phenomenon caused by earthquakes.

Settlement measurements needs to be taken regularly, at planned intervals depending on soil data and settlement predictions, to ensure safe operations of the tank during service. The amount of allowable tank settlement depends on the size of the tank, as shown in **Figure 3.3**.

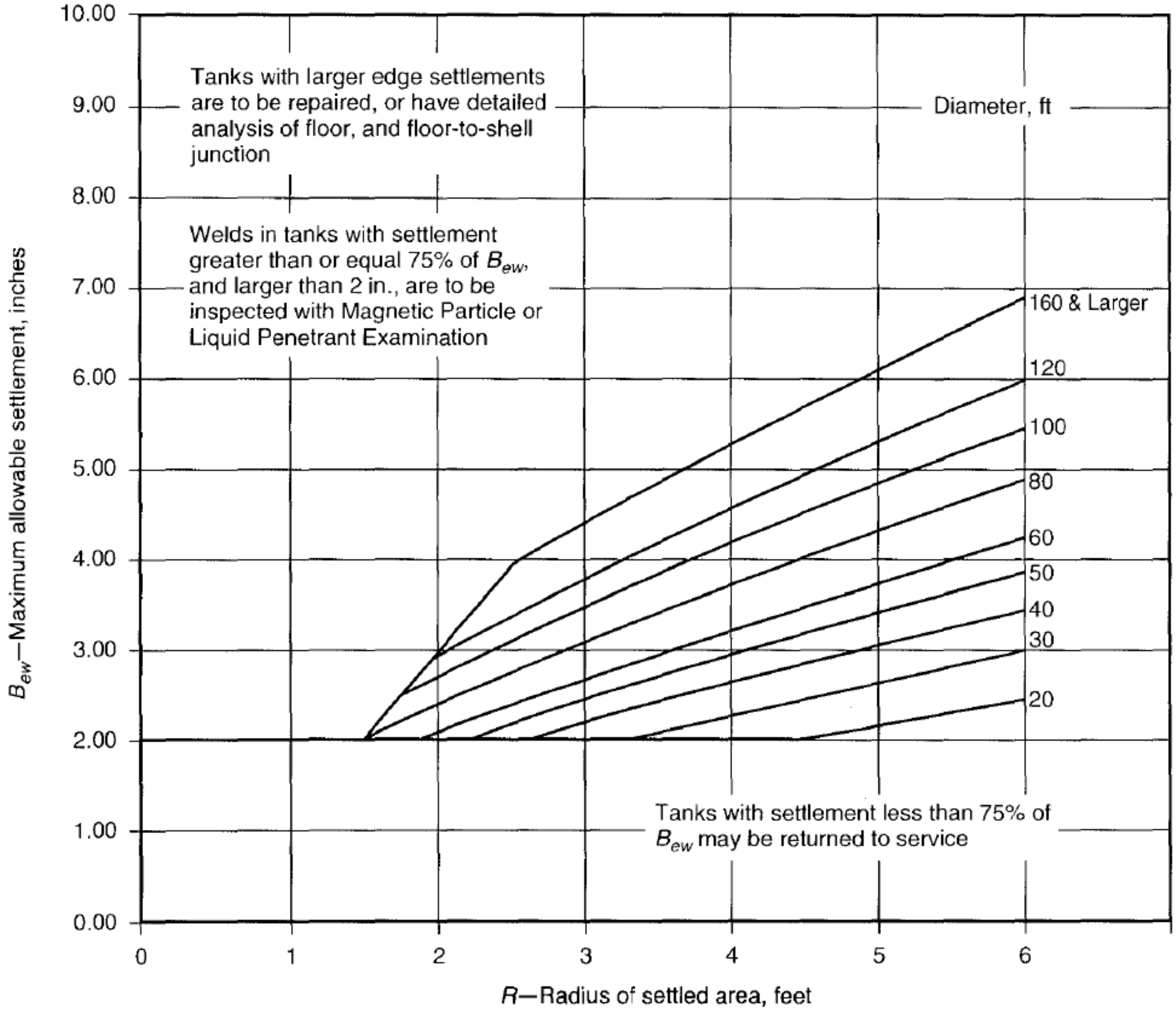


Figure 3.3: Maximum allowable edge settlements of tanks, (API 653, 2003).

As can be seen in the figure, small diameter tanks are allowed to settle around 2.1/2" (~64mm) vertically into the soil and larger sizes have a maximum allowable settlement below 7" (<178mm). Therefore, this range of settlement values will be used as a base of reference for flexibility and stress calculations throughout this report.

3.2 Design criteria

Each one of the three alternatives for the Viking-Johnson couplings needs to be integrated into a complete piping design. The following criteria applies for the design of these alternatives:

Pipeline and piping components

- Carbon steel shall be used for the pipelines. The required types of Carbon Steel are described in the Vicoma pipe specs Vic C-150:
    - Pipes (A106 Gr. B);
    - Elbows (A234 Gr. WPB);
    - Flanges (A105N)
- The pipe flanges must be able to withstand a pressure of 18,66 bar(g) at a temperature of 68 ^{0}C;
- Only fire-resistant seals and gaskets shall be used, rubber products shall be avoided;
- The pipeline flexibility must be in accordance with the Kellogg flexibility equation;
    - $\frac{d_o y}{(L-U)^2} \leq 208$
- The pipeline shall be able to withstand stresses caused by tank settlement. According to API-653, the maximum allowed settlement is 178mm for a large tank;
- According to ASME B31.3, the stress limits of the pipeline must comply with the following equations:
    - $S_{lf} + S_b + S_{occ} \leq 1{,}33 S_h$ (allowed tensile stresses);
    - $\tau_f + \tau_t + \tau_{occ} \leq 0{,}8 S_h$ (allowed shear stresses;
- Nozzle stresses need to be within acceptable values. These values shall be obtained by performing a FEM analysis with Nozzle Pro;
- Pipe thickness shall contain a corrosion allowance of 1,6mm as stated in the Vicoma pipe specs Vic C-150;
- Metallic expansion joints with flanges shall be used instead of welded ends to increase the ease of maintenance and for inspection purposes.

Pipe supports

- The type and locations of pipe supports need to be specified by performing a Caesar II stress and flexibility analysis;
- The pipe supports must be designed with a yield stress safety factor of 0,75;
- Simple profiles shall be used in the design of pipe supports to simplify construction;
- Distances between pipe supports need to comply with the safe spans of supported pipelines (see **appendix III**). The safe spans are the maximum distances between supports to prevent the pipeline from sagging due to its own weight. These distances increase with pipe size.

3.3 Design models using Caesar II

3.3.1 Alternative 1: Straight pipe

This option provides the shortest route to connect the pipeline to the tank (see **Figure 3.4**). This is also the simplest way to replace the Viking-Johnson coupling.

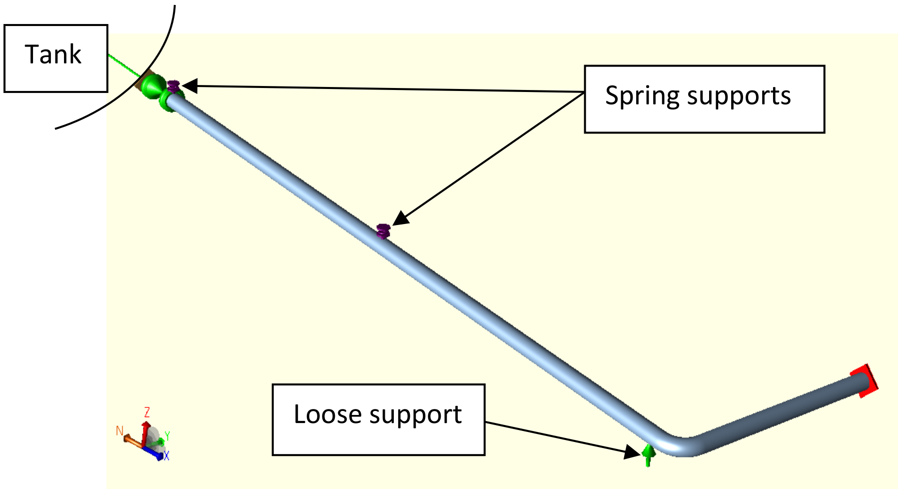
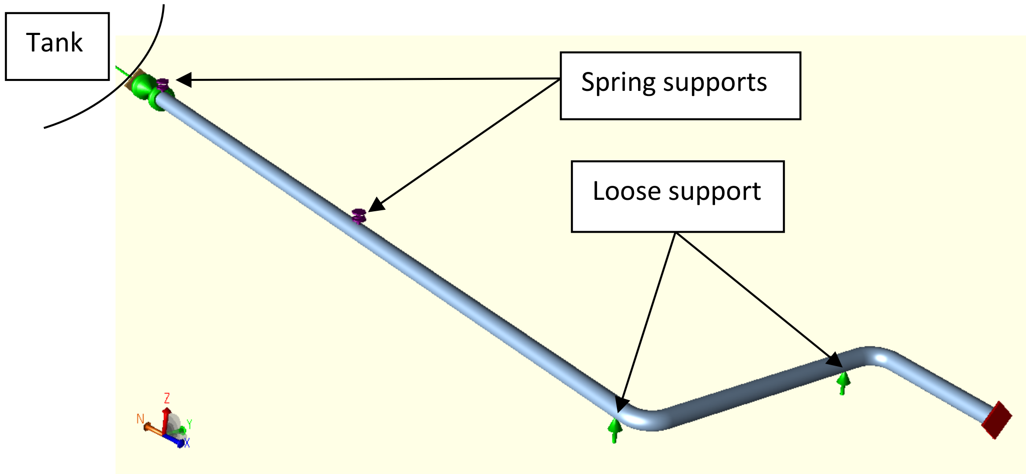


Figure 3.4: Alternative 1 - straight pipe.

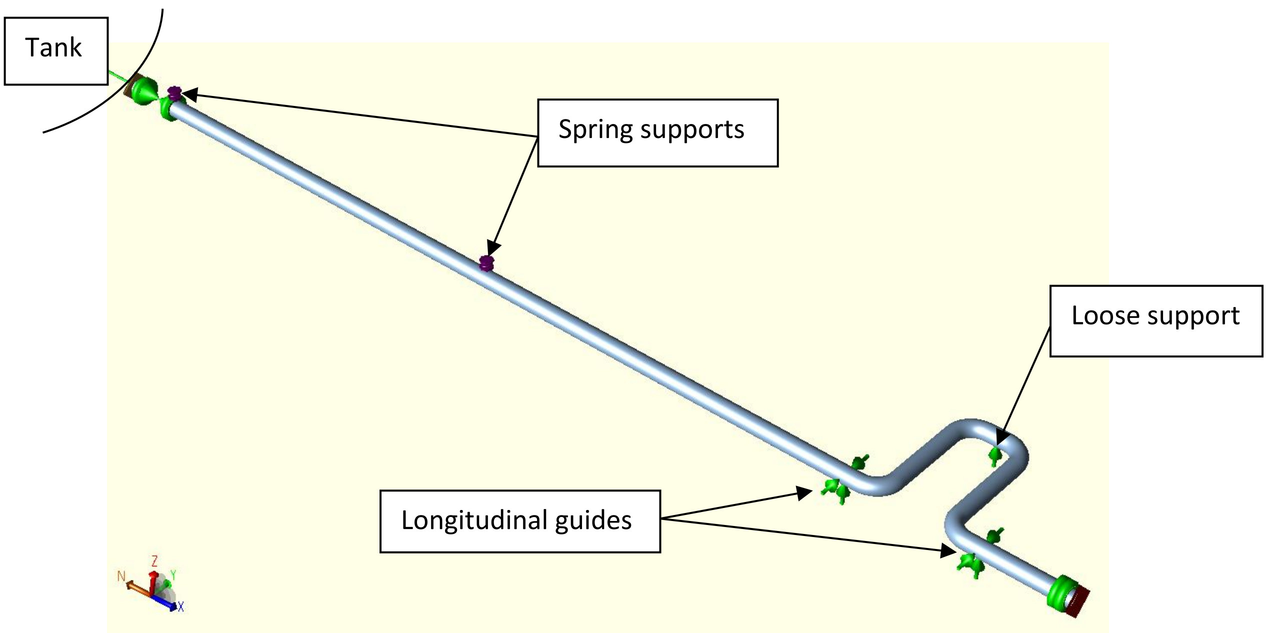
Spring supports need to be used to ensure the flexibility of the pipe due to the tank settlement and to support its own weight. One spring support needs to be located at the tank nozzle to allow the pipeline to deflect vertically with the tank settlement. A second spring support should be positioned between the loose support at the existing elbow and the tank nozzle. This arrangement allows the pipe to bend gradually and therefore minimize stresses. Also, the location of the supports needs to be in accordance with the maximum allowable safe spans of the pipeline.

3.3.2 Alternative 2: Increased flexibility with the use of elbows

The flexibility of a pipeline can be increased by introducing elbows into the design, as can be seen in **Figure 3.5** (a and b). In the case of a pipe loop, it is necessary to utilize a longitudinal guide just before the first elbow and after the last one so that the thermal expansion of the pipe can be absorbed optimally by the loop. The own weight of the pipe loop needs to be supported to avoid torsion moments caused by sagging of the pipe loop. Spring supports are also necessary for the tank settlement and the distances of the supports are in accordance with the maximum allowable safe spans.



(a) Compensating leg (L-pipeline)



(b) Pipe-loop

Figure 3.5: Alternative 2 – Increased flexibility with the use of elbows

3.3.3 Alternative 3: Flanged metallic expansion joints for lateral displacements

Metallic expansion joints gain their flexibility from their bellows arrangement. This design makes use of flanged metallic expansion joints (see **figure 3.6**). Due to the high pressure inside of the pipeline (18,66 bar(g)), the bellows need to be safeguarded by longitudinal rods. These rods limit the bellows from expanding longitudinally, therefore preventing rupture of the bellows. These rods are connected by hinges to allow lateral displacements.

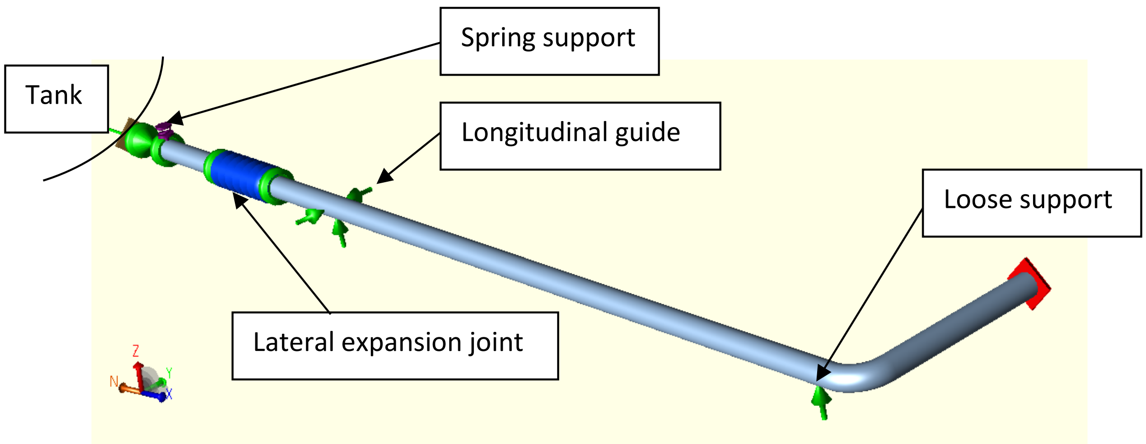


Figure 3.6: Option 3 – metallic expansion joints

The pipe supports are specifically chosen to increase the metallic joints range of motion. A spring support is placed at the tank nozzle to support the weight of the pipeline and to allow for vertical displacement in the case of tank settlement. Behind the expansion joint, a guide will hold the rest of the pipeline in place. This configuration allows the expansion joint to deflect vertically only. The expansion joint offers the maximum flexibility compared to the other designs. This means that the nozzle stresses are minimized and the design is kept compact.

4. Stress- & Flexibility analysis

This chapter starts by explaining how Caesar II and Nozzle Pro are used (§**4.1**). Subsequently, the approach used to analyze the stress and flexibility is explained (§**4.2**). Thereafter, the stress and flexibility results are presented (§**4.3**). Then a calculation check is performed by using cantilever equations (§**4.4**). Lastly, this chapter ends by calculating the supporting for buckling and the spring supports are elaborated (§**4.5**).

4.1 Calculations methodology of software

Caesar II

The stress and flexibility of the pipeline is calculated using Caesar II. This software is a system analysis tool that uses beam stiffness to simplify pipelines. Instead of modelling an actual pipeline, this tool calculates using the stiffness of a beam. This is further simplified by equation (**4.1**).

$$F = kx \tag{4.1}$$

F = Load applied at each end of a pipe [N]
k = Beam stiffness representing a pipe between two ends (node points) [N/mm]
x = Deflection of the nodes [mm]

The internal loads on a pipe are calculated based on the final position of each node in the system. These loads are then converted into code-defined stresses of the system. The limits of these code-defined stresses comply with the standard piping codes and legislations. Caesar II has more than 35 codes in its database to be chosen from.

The following work procedure explains how Caesar II is used to calculate a pipeline for flexibility and stress (for illustrations see **appendix IV**):

1. Start of new project and selection of units;
2. Modelling of pipeline and tank connection;
3. Selection of materials and process conditions;
4. Selection and location of pipe supports;
5. Specification of loads and tank settlement;
6. Selection of static load cases according to the latest ASME B31.3 code;
7. Error check, analysis run and generate reports;

When the analysis run is finished, Caesar II indicates if the stress and flexibility of the pipeline passes the stress criteria of ASME B31.3. The reports show the allowable stresses and the actual stresses of each node in the model. If one of these stresses is too high, then the design will have to be adjusted. These steps are repeated for all pipe sizes (6” through 24”) and for every alternative.

Nozzle Pro

Caesar II is not a finite element analysis tool. Therefore, the resulting loads acting on the tank nozzle that were calculated using Caesar II will serve as input for the finite element analysis tool (Nozzle Pro). This tool has over 37 load cases in its database. According to ASME B31.3 (2016), these load cases are necessary to ensure the integrity of the tank nozzle. If the stresses are too high, then the Caesar II model needs to be adjusted to create more flexibility into the design. This iterative method is performed for all pipe sizes and for every alternative. **Appendix V** explains how the tank is modelled using Nozzle Pro.

4.2 Analysis approach

Tank settlement

As explained previously in chapter 3, the allowable edge settlements of tanks may vary between 64mm for small diameter tanks and 178mm for large diameter tanks (API 653, 2003). This report uses a large diameter tank (50m) as a reference to perform the stress- and flexibility calculations. When a tank is filled, it will temporarily settle a few millimeters into the ground. When the tank is empty, the tank will return to its original position. For this reason, an additional settlement of 20 mm is used in the calculations.

In practice, pipelines are installed to tank nozzles with a small deflection upwards to compensate for the tank settlement (see **figure 4.1**). The tank used as a reference in this report has a maximum settlement of approximately 200 mm. The pipeline is installed with a pre-deflection upwards to reduce the stresses of the nozzle due to tank settlement. As the tank settles, the pipeline will return to a neutral position and then deflect 100 mm further downwards. This method decreases the nozzle stress significantly. General cantilever equations prove that the reaction loads due to deflection depend solely on the deflection regardless on the direction (in the case of a pipe). This means that a pre-deflection upwards will reduce the maximum nozzle bending load by half.

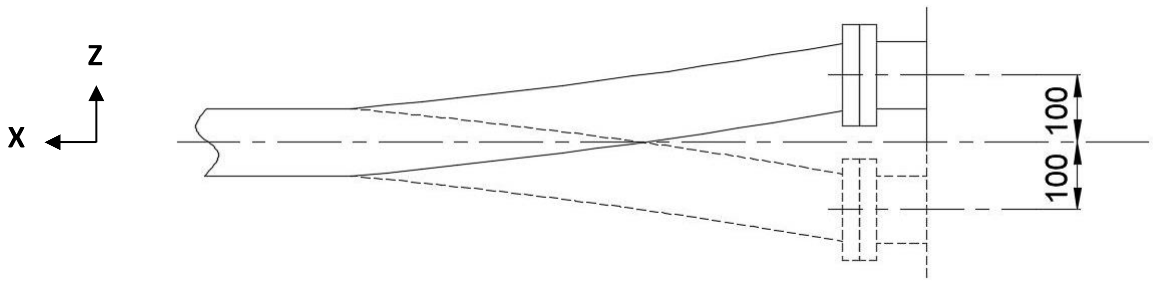


Figure 4.1: Pre-deflection of pipeline during installation

Point of rotation

The pipeline uses the first pipe support as a point of rotation as the tank settles (see **figure 4.2**). Note that the designs considered in this report have the point of rotation along the nozzle axis (the x-axis). When this point is located at an offset from the nozzle axis, the loads acting on the tank nozzle becomes more complicated to calculate. An offset increases torsion loads on the nozzle and the effect of thermal expansion becomes more difficult to predict (depends on the pipe routing). For the scope of this project it is not necessary to research the effects of the point of rotation located at an offset from the nozzle axis. The minimum distance required from the tank nozzle to the point of rotation along the x-axis is sufficient. The distance to this point of rotation is found with Caesar II and Nozzle Pro.

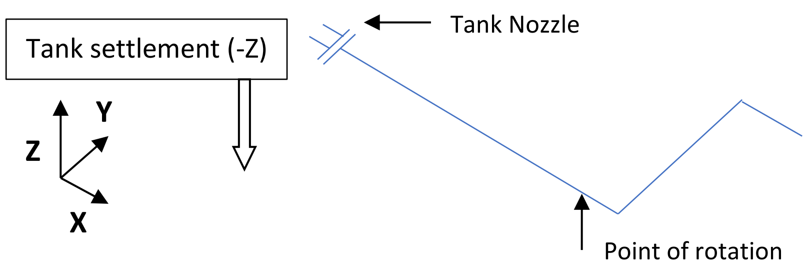


Figure 4.2: Tank settlement and point of rotation

4.3 Stress- and flexibility results

4.3.1 Results of alternative 1: straight pipe

Figure 4.3 provides a schematic representation of alternative 1. This alternative introduces a straight pipe for replacing the Viking-Johnson coupling. As the tank settles, the pipeline bends and rotates about the point of rotation. The results of the stress and flexibility analysis are shown in **table 4.1**.

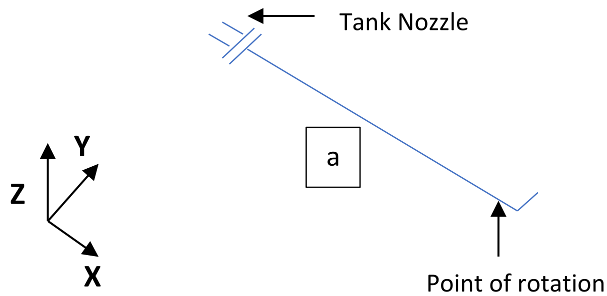


Figure 4.3: Schematic representation of Alternative 1 – Straight pipe

STRAIGHT PIPE									
d_0 [inch]	**6”**	**8”**	**10”**	**12”**	**14”**	**16”**	18”	20”	24”
a [mm]	9000	11000	13000	16500	19000	24750	…	…	…
Nozzle stress [%]	79	86	97	98	99	100	PIPELINE TOO HEAVY		
Pipeline stress [%]	96	90	83	73	69	74			

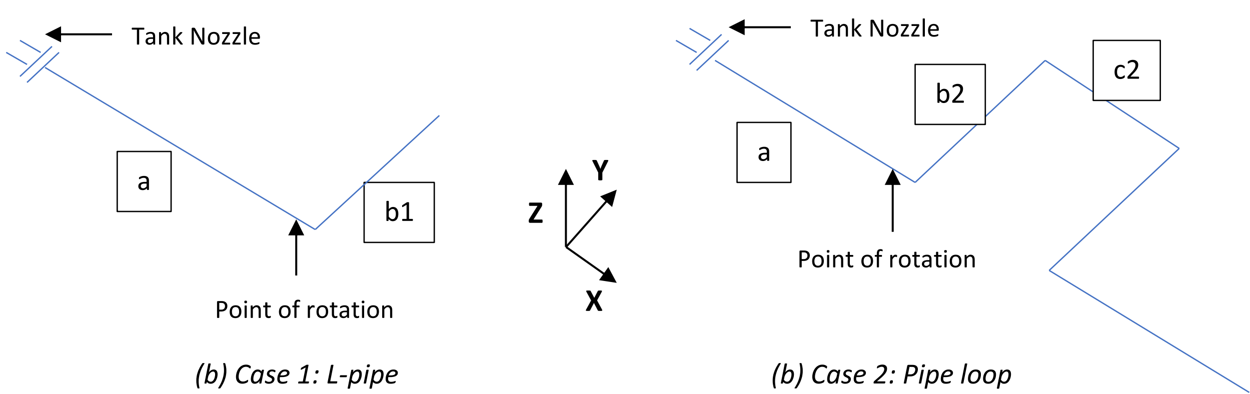
Table 4.1: Stress- and flexibility results for alternative 1

It can be seen in the table that the pipe length increases the flexibility of the pipeline. This is necessary to ensure acceptable nozzle stresses. However, sizes above 16” become too heavy for the nozzle. At 16”, the nozzle reaches its maximum allowable stress.

4.3.2 Results of alternative 2: Increasing flexibility with elbows

Figure 4.4 provides a schematic representation of alternative 2. This alternative increases the flexibility by introducing elbows in the design. Two cases were considered:

- Case 1: L-pipeline
- Case 2: Pipe loop



(b) Case 1: L-pipe

(b) Case 2: Pipe loop

Figure 4.4: Schematic representation of Alternative 2 – Increasing flexibility with elbows

The variables a, b and c were found iteratively with Caesar II and Nozzle Pro. In each iteration one variable was held constant while the others were changed. The effects of these changes to the applied loads on the nozzle were analyzed, which lead to length choices based on load predictions. The results are minimum lengths required to ensure the integrity of the nozzle for a tank settlement of 200 mm. **Table 4.2** show the results for Cases 1 and 2.

Pipe detail		CASE 1: L-pipe			CASE 2: Pipe loop				Stress reduction	
d_0	a	b1 (min)	Nozzle stress	Pipeline stress	b2 (min)	c2 (min)	Nozzle stress	Pipeline stress	Nozzle stress	Pipeline stress
[inch]	[mm]	[mm]	[%]	[%]	[mm]	[mm]	[%]	[%]	[%]	[%]
6"	9000	2200	79	96	1500	750	73	92	6	4
8"	11000	2700	86	90	1750	875	79	88	7	2
10"	13000	3300	97	83	2000	1000	91	83	6	0
12"	16500	4000	98	73	2750	1375	94	70	4	3
14"	19000	4500	99	69	3500	1750	99	64	0	5
16"	24750	5500	100	74	3750	1875	102	65	-4	9
18"	…	PIPELINE TOO HEAVY			PIPELINE TOO HEAVY				NOZZLE STRESS INCREASES	
20"	…									
24"	…									

Table 4.2: Stress- and flexibility results for alternative 2 (Cases 1 and 2)

These results show that for alternatives 1 and 2, pipelines larger than 16” become too heavy for the tank nozzle to handle. The Caesar II and Nozzle Pro results are shown in **appendix VI.** To put things in perspective, a calculation is made to compare the relative weights of the pipeline. The pipeline is assumed to be filled with water for hydrotest and maintenance purposes; water is heavier than hydrocarbons. **Table 4.3** shows the relative weights of the pipelines.

d_0 [inch]	d_i [mm]	Specific weight [kg / m]	Pipe length “a” [mm]	Total weight [kg]
6”	161	25	9000	408
8”	213	31	11000	733
10”	267	39	13000	1235
12”	318	47	16500	2085
14”	349	55	19000	2862
16”	400	63	24750	4668
18”	451	71	24750	5709
20”	502	79	24750	6851
24”	604	95	24750	9439

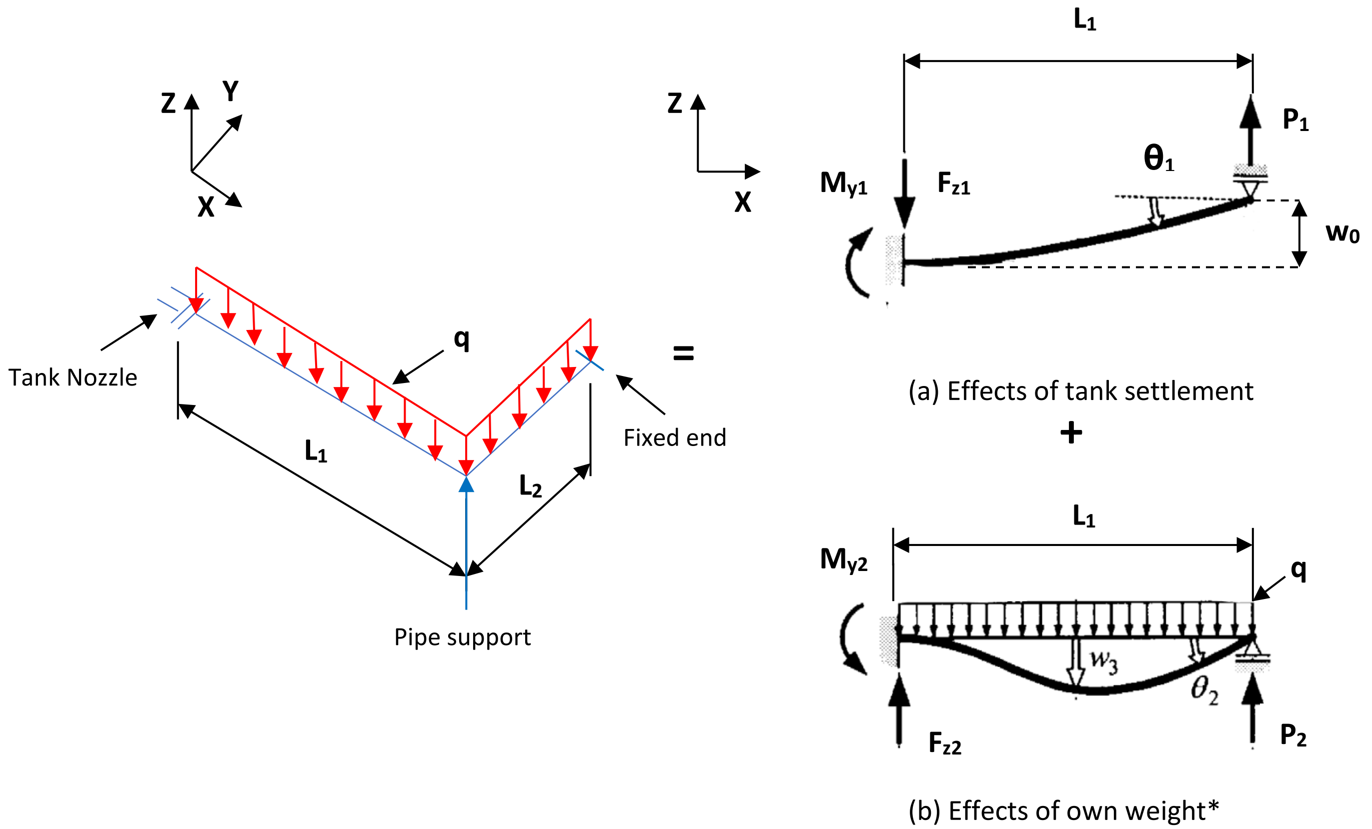
Table 4.3: Weight comparison

As can be seen in the table, an 18” pipe filled with water weighs 1041 kg more than a 16” having the same length. Note that these weights are partially absorbed by spring supports, however the more spring supports in the design the less flexible the pipeline becomes. Therefore, it is recommended that pipe sizes larger than 16” be calculated using special mechanisms that support the weight and yet allow the pipeline to be flexible. An example of this is the use of counterweight supports. Further elaboration of this support is omitted from this report because it falls outside of the scope.

Results of alternative 3: Metallic expansion joints with flanges (lateral displacements)
The stress and flexibility results from Caesar II show that the acting loads on the tank nozzle are very low and the pipe stresses are near zero. For this reason, it was concluded that metallic expansion joints can be used with any pipe size and length configuration. For the results of Caesar II regarding the expansion joints can be found in appendix VI. In these results it can be seen that the reaction forces due to tank settlement equals 0.

4.4 Control check of Caesar II results

To validate the results of Caesar II, a manual calculation is performed by applying superposition to the theory of cantilever beams (Hibbeler, 2015). The cantilever results are then compared to the results obtained with Caesar II. Alternative 2 (Case 1 - 6") is used as example for these calculations. **Figure 4.5** shows the free body diagram of Case 1 (L-pipe). **Appendix VII** show a list of equations and free body diagrams for different cantilever beam situations.



**The same free body diagram is used for L_2 to calculate the effect of torsion along the x-axis of the nozzle.*

Figure 4.5: Free body diagram of alternative 2 (Case 1) – (Source: TUDelft, 2013)

Reaction loads on tank nozzle due to tank settlement:

Tank settlement occurs along the -Z axis. This settlement results in a reaction force $\mathbf{F_{z1}}$ and moment $\mathbf{M_{y1}}$. These reactions can be calculated with equation (**4.2**) and (**4.3**) respectively. The angle of rotation $\boldsymbol{\theta_2}$ is needed to calculate the torsion $\mathbf{T_y}$ of the compensating leg (L_2). This angle can be found with equation (**4.4**). Note that the compatibility equation ($\delta = \delta_1 + \delta_2 = 0$) to solve the '*singular statically indeterminate problem*' is already included in the beam equations. This compatibility equation is essential for finding unknown terms in the equations do to over constrain of the pipe.

$$F_{z1} = P_1 = \frac{3EIw_0}{L_1^3} \qquad \textbf{(4.2)}$$

$$M_{y1} = \frac{3EIw_0}{L_1^2} \qquad \textbf{(4.3)}$$

$$\theta_1 = \frac{3w_0}{2L_1} \qquad \textbf{(4.4)}$$

F_{z1} and P_1 = Reaction forces [N]

M_{y1} = Internal moment [Nm]

E = Young's modulus of elasticity [Pa]

I = Moment area of inertia [m^4]

L_1 = Length of pipe [m]

w_0 = Tank settlement [m]

θ_1 = Angle of rotation [rad]

<u>Calculation:</u>

Given:	L_1 = 9,74 m	I = $1,17*10^{-5}$ m^4 (6" pipe)
	E =203,39 GPa	w_0 = 100 mm = 0,1 m

$$F_{z1} = \frac{3EIw_0}{L_1^3} = \frac{3*203{,}39*10^9*1{,}17*10^{-5}*0{,}1}{9{,}74^3} = 774\ N$$

$$M_{y1} = \frac{3EIw_0}{L_1^2} = \frac{3*203{,}39*10^9*1{,}17*10^{-5}*0{,}1}{9{,}74^2} = 7533\ Nm$$

$$\theta_1 = \frac{3*0{,}1}{2*9{,}74} = 0{,}0154\ rad$$

θ

Reaction loads on tank nozzle due to weight effects:

The reaction force $\mathbf{F_{z2}}$ and moment $\mathbf{M_{y2}}$ act on the nozzle due to weight effects of the pipe filled with water. These reactions are calculated with equations (**4.5**) and (**4.6**) respectively. Water is used instead of hydrocarbons because the pipeline is subjected to hydrotests prior to usage; water has a higher specific weight than hydrocarbons. The effect of weight also creates torsion on the compensating leg (L_2), therefore the angle of rotation $\boldsymbol{\theta_2}$ needs to be calculated with equation (**4.7**). The distributed load $\mathbf{q}$ is obtained with equation (**4.8**).

$$F_{z2} = \frac{5qL_1}{8} \qquad \textbf{(4.5)}$$

$$M_{y2} = \frac{qL_1^2}{8} \qquad \textbf{(4.6)}$$

$$\theta_2 = \frac{qL_1^3}{48EI} \qquad \textbf{(4.7)}$$

$$q = \frac{(m_{pipe}+m_{water})*g}{(L_1+L_2)} \qquad \textbf{(4.8)}$$

F_{z2} = Reaction force [N]

M_{y2} = Internal moment [Nm]

E = Young's modulus of elasticity [Pa]

I = Moment area of inertia [m^4]

L_1 and L_2 = Length of pipe [m]

θ_2 = Angle of rotation [rad]

q = distributed weight [N/m]

m = mass [kg]

g = gravity [m/s^2]

Calculation:

Given:	L_1 = 9,74 m	L_2 = 2,5 m
	E =203,39 GPa	m_{pipe} = 455 kg
	I = 1,17*10^{-5} m^4 (6" pipe)	m_{water} = 230 kg

$$q = \frac{(m_{pipe} + m_{water}) * g}{(L_1 + L_2)} = \frac{(455 + 230) * 9{,}81}{(9{,}74 + 2{,}5)} = 549\ N/m$$

$$F_{z2} = \frac{5qL_1}{8} = \frac{5 * 549 * 9{,}74}{8} = 3342\ N$$

$$M_{y2} = \frac{qL_1^2}{8} = \frac{549 * 9{,}74^2}{8} = 6510\ Nm$$

$$\theta_2 = \frac{qL_1^3}{48EI} = \frac{549 * 9.74^3}{48 * 203.39 * 10^9 * 1{,}17 * 10^{-5}} = 4{,}44 * 10^{-3}\ rad$$

θ

Calculations to find torsion T_y of the compensating leg:
The compensating leg undergoes a torsion moment $\mathbf{T_y}$ due to the angular rotation $\boldsymbol{\theta_T}$. The torsion can be calculated with equation (**4.9**). Furthermore, **figure 4.6** shows a free body diagram of the compensating leg.

$$T_y = \frac{\theta_T IG}{L_2} \tag{4.9}$$

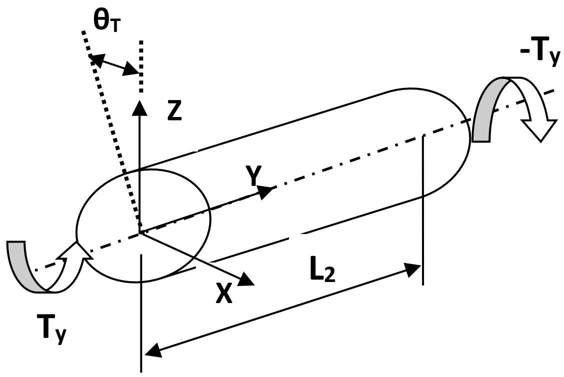
T_y = Torsion of compensating leg [Nm]

θ_T = Total angle of rotation [rad]

I = Area moment of inertia [m^4]

G = Shear modulus of elasticity [Pa]

L_2 = Pipe length of compensating leg [m]



This is a statically determinate problem.

Given:

L_2 = 2,5 m

E = 203,39 GPa

I = 1,17*10^{-5} m^4 (6" pipe)

v = 0,292 (Poisson ratio)

G = E/(2*(1+ v) = 78,7 GPa

Figure 4.6: Free body diagram of the compensating leg

Calculation:

$$T_y = \frac{\theta_T IG}{L_2} = \frac{(0{,}0154 + 4{,}44 * 10^{-3}) * 1{,}17 * 10^{-5} * 78{,}7 * 10^9}{2{,}5} = 7307\ Nm$$

Calculation to account for thermal expansion:
Thermal expansion causes the pipeline to deform according to **figure 4.7**. This deformation results in the following loads and moment on the tank nozzle: $\mathbf{R_x}$, $\mathbf{R_y}$, $\mathbf{M_z}$. According to the theory of pipe stress engineering (Liang-Chuan & Tsen-Loong, 2017), the equations of simple beam deflection are used to approximate the values that were obtained with Caesar II.

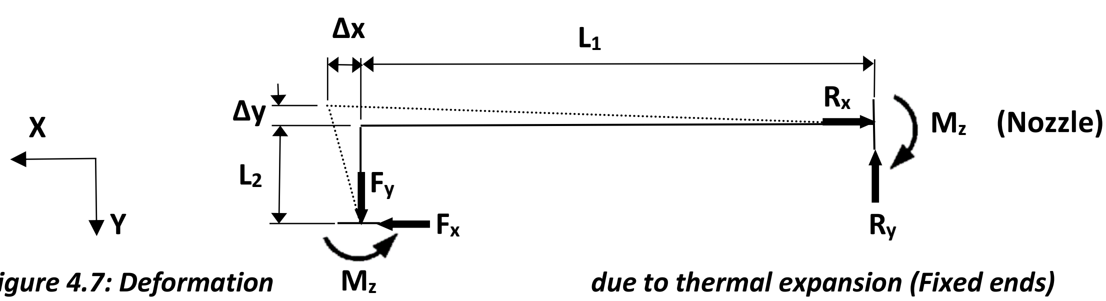


Figure 4.7: Deformation due to thermal expansion (Fixed ends)

The values for thermal expansion (**Δx** and **Δy**) can be found with equation (**4.10**). The reaction forces $\mathbf{R_x}$ and $\mathbf{R_y}$ can be found with equation (**4.11**) and the reaction moment $\mathbf{M_z}$ with equation (**4.12**).

$$\Delta x = c_T L_1 \Delta T, \quad \Delta y = c_T L_2 \Delta T \tag{4.10}$$

$$R_x = \frac{12EI\Delta x}{L_2^3}, \quad R_y = \frac{12EI\Delta y}{L_1^3} \tag{4.11}$$

$$M_z = \frac{3EI\Delta y}{L_1^2} \tag{4.12}$$

Δx and Δy = thermal expansion [mm]

c_T = thermal expansion coefficient [mm/(m*^{0}C)]

L_1 and L_2 = pipe lengths [m]

ΔT = temperature difference [^{0}C]

R_x and R_y = Reaction forces on the nozzle [N]

M_z = Reaction moment on the nozzle [Nm]

E = Young's modulus of elasticity [N/m^2]

I = Moment area of inertia [m^4]

<u>Calculation:</u>

Given: L_1 = 9,74 m

L_2 = 2,5 m

c_T = 1,2*10^{-2} mm/(m*^{0}C)

E = 203,39 GPa

I = 1,17*10^{-5} m^4 (6"pipe)

$$\Delta x = c_T L_1 \Delta T = 1{,}2 * 10^{-2} * 9{,}74 * 48 = 5{,}67\ mm$$

$$\Delta y = c_T L_2 \Delta T = 1{,}2 * 10^{-2} * 2{,}5 * 48 = 1{,}44\ mm$$

$$R_x = \frac{12EI\Delta x}{L_2^3} = \frac{12 * 203{,}39 * 10^9 * 1{,}17 * 10^{-5} * 5{,}67 * 10^{-3}}{2{,}2^3} = 15206\ N \;\rightarrow$$

$$R_y = \frac{12EI\Delta y}{L_1^3} = \frac{12 * 203 * 10^9 * 1{,}17 * 10^{-5} * 1{,}44 * 10^{-3}}{9{,}74^3} = 45\ N \;\uparrow$$

$$M_z = \frac{3EI\Delta y}{L_1^2} = \frac{3 * 203 * 10^9 * 1{,}17 * 10^{-5} * 1{,}44 * 10^{-3}}{9{,}74^2} = 108\ Nm \;\curvearrowright$$

Calculations to account for torsion along the x-axis:

Torsion along the x-axis is created by the own weight of the compensating leg as can be seen in **figure 4.8**.

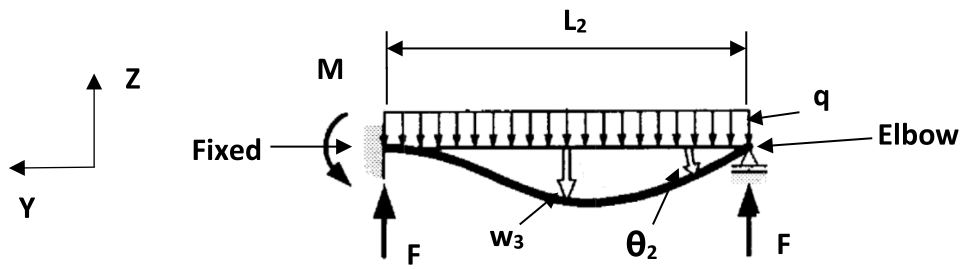


Figure 4.8: Free body diagram of compensating leg

In the free body diagram above, it can be seen that only the rotation angle **θ_3** is needed to calculate the torsion **M_x**. These values are obtained with equations (**4.13**) and (**4.14**) respectively.

$$\theta_3 = \frac{qL_2^3}{48EI} \qquad \textbf{(4.13)}$$

$$M_x = \frac{\theta_3 IG}{L_1} \qquad \textbf{(4.14)}$$

θ_3 = rotation angle [rad]
q = distributed weight [N/m]
L_2 = length of compensating leg [m]
E = Young's modulus of elasticity [N/m^2]

M_x = torsion along the x-axis of the nozzle [Nm]
G = shear modulus of elasticity [Pa]
L_1 = Length of pipe along the x-axis [m]
I = area moment of inertia [m^4]

Calculation:

Given:

L_1 = 9,74 m
L_2 = 2,5 m
q = 549 N/m
E = 203,39 GPa
I = 1,17*10^{-5} m^4 (6"pipe)
G = 78,7 GPa

$$\theta_3 = \frac{qL_2^3}{48EI} = \frac{549 * 2{,}5^3}{48 * 203{,}39 * 10^9 * 1{,}17 * 10^{-5}} = 7{,}51 * 10^{-5}\,rad$$

$$M_x = \frac{\theta_3 IG}{L_1} = \frac{7{,}51*10^{-5}*1{,}17*10^{-5}*78{,}7*10^9}{9{,}74} = 7{,}1 \text{ Nm}$$

Comparing the calculation results with the results of Caesar II:
The results obtained with the cantilever beam calculations and the results obtained in Caesar II are compared in **table 4.4**. From this table it can be concluded that the beam equations can be used to calculate the reaction forces of the tank settlement. The difference between Caesar II and the beam equations is less than 6%. However, as can be seen in table, the effects of thermal expansion are too complex to be approached solely by the beam equations and the equations for thermal expansion. Possible reasons that explain these huge deviations are:

- The effect of spring supports was not included in the beam calculations;
- Caesar II calculates numerically by using a complex stiffness matrix that includes all the nodes in the model;
- The beam equations do not take into account the effect of stress intensification factors nor the effects of rotation of the elbows;
- Under a bending moment, a curved pipe (elbow) behaves differently than a curved beam; the circular cross-section of the pipe becomes oval. Caesar II approximates the effects ovalization numerically with the stiffness matrix.

	Nozzle (Node 25)						Fixed support (Node 70)
	Rx [N]	Ry [N]	Rz [N]	Mx [Nm]	My [Nm]	Mz [Nm]	Ty [Nm]
Beam equations	15206	45	2568	7,1	14043	108	7307
Caesar II (Node 25)	18038	1152	2424	308	13209	4461	5435
Difference (%)	15,7	96,1	5,6	97,7	5,9	96,1	25,6

Table 4.4: Comparison of results between the beam calculations and Caesar II

Also, according to the theory of pipe stress and flexibility (Liang-Chuan & Tsen-Loong, 2017), beams under thermal stresses behave differently from sustained stress caused by weight and pressure. This is shown in **figure 4.8**. A pipe subjected to a sustained load will have a displacement that increases accordingly when the load is increased. When the stress reaches the yield value, the stress magnitude remains constant but the displacement increases by a large amount. In the case of thermal expansion, the cantilever beam deflects differently; it depends solely on the material strain (elongation divided by original length) and is therefore self-limiting. This means that yielding or deformation of the material reduces stress without further displacement. The effect of these differences in deflection can be seen in the figure.

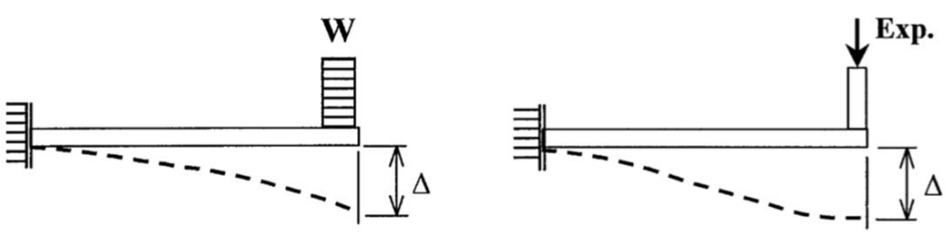


Figure 4.8: Sustained stress versus thermal stress

4.5 Analysis of pipe supports

There are many different types of supporting that can be used to support the pipeline, each with a specific function. For simplicity, this report makes use of the following pipe supports:

- Regular supports
- Spring supports

Regular supports

According to ASME B31.3, (2018), the piping engineer is not constrained in the design of pipe supports. However, the use of simple steel profiles is highly recommended when designing the pipe support. Simple profiles like a HEB-beam can withstand very high loads without buckling. For this reason, simple profiles are recommended by ASME B31.3. Equation (**4.15**) is used to determine the critical load for buckling to occur for a given beam length.

$$F_{cr} = \frac{\pi^2 EI}{(KL)^2} \tag{4.15}$$

F_{cr} = critical buckling load [N]

E = Young's modulus of elasticity [N/m^2]

I = Moment area of inertia [m^4]

K = coefficient for type of end connection [--]

L = Length of the beam [m]

However, it is empirical to check whether the critical material stress of the beam is greater or less than the allowed material yield stress (according to ASME B31.3, 2016). See equation (**4.16**).

$$\sigma_{cr} = \frac{\pi^2 E}{(KL/r)^2} \leq \sigma_y \tag{4.16}$$

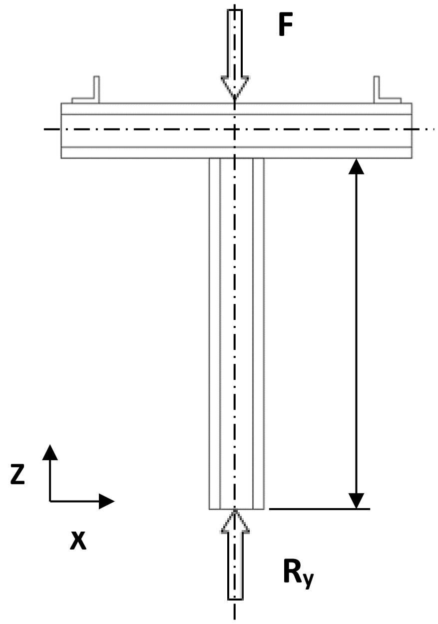
σ_{cr} = Critical material stress [N/m^2]

σ_y =Material yield stress [N/m^2]

$(KL/r)^2$ = beam slenderness factor [--]

r = radius of gyration [m]

Figure 4.9 shows the free body diagram and calculation for the support having a vertical HEB-100 beam with length 1,5 m.



This is a statically determinate problem with $F = R_y$

Given:

L = 1,5 m

E = 200 GPa

σ_y = 250 MPa,

(allowable yield stress = 0,75 *250 = 187 Mpa

$I_x = 1{,}67 \cdot 10^{-6}$ m^4

$I_z = 4{,}5 \cdot 10^{-6}$ m^4

$A = 2{,}6 \cdot 10^{-3}$ m^2

$r = (I/A)^{(1/2)} = 0{,}0253$ m

K = 2 (Hibbeler, Strength of materials, 8th edition)

Calculation:

$$F_{cr} = \frac{\pi^2 EI}{(KL)^2} = \frac{\pi^2 * 200 * 10^9 * 1{,}67 * 10^{-6}}{(2 * 1{,}5)^2} = 366\ kN$$

$$\sigma_{cr} = \frac{\pi^2 E}{\left(\frac{KL}{r}\right)^2} = \frac{\pi^2 * 200 * 10^9}{(\frac{2 * 1{,}5}{0{,}0253})^2} = 140\ MPa < \sigma_y(187\ MPa)$$

Figure 4.9: Free-body-diagram (FBD) and buckling calculation

The results of the calculations show that this simple support can withstand a critical load of 366 kN before bucking. This means that the support can withstand a weight of 37309 kg without buckling. The critical yield stress is lower than the material yield stress (accounted for safety factor 0.75). Thus, the critical load is calculated using the critical yield stress.

However, in cases where the critical yield stress is larger than the material yield stress, then the lowest value should be used to calculate the critical load that the support can withstand by multiplying the cross-sectional area of the beam with the yield stress (F=A*σ). As a control check, this scenario will be calculated:

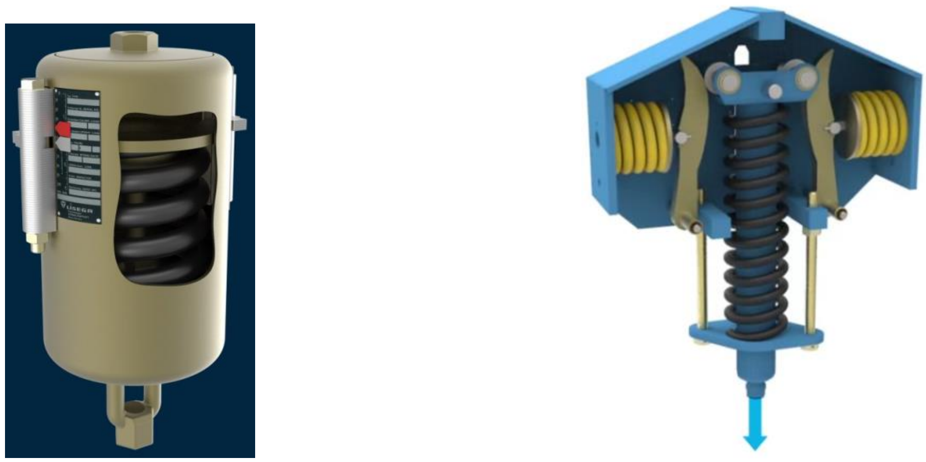
$$F_{cr} = A * \sigma_y = 2.6 * 10^{-3} * 187 * 10^6 = 486\ kN$$

As can be seen by this calculation check, buckling occurs at a much lower critical load (366 kN < 486 kN).

Spring supports

Spring supports can be categorized into two basic types (see **figure 4.10**):

- Variable load - Reaction forces vary linearly with the spring travel (constant spring rate).
- Constant load - Reaction force is always a constant value regardless of the spring travel.



(a) Variable load (single-spring mechanism) (b) Constant load (triple-spring mechanism)

Figure 4.10: Basic spring support types (Lisega, 2016).

Lisega spring supports manufacturing company provides over 73 different types of constant load spring supports, each with its own specific spring travel. The sheer number of variable spring supports is much higher, in the range of hundreds. However, Lisega provides special tables for selecting the most appropriate spring support. To acquire the necessary spring rate for variable spring supports, see equations (**4.17**) and (**4.18**). These values are necessary to select the specific type of pipe supports in the tables of Lisega. The operating load and spring travel are determined from support calculations made with Caesar II, these are the vertical reaction forces and vertical displacements acting on the node where the support is located.

$$c_1 \leq \frac{p_1 F}{100s} \tag{4.17}$$

$$p_2 = \frac{100sc_2}{F} \leq p_1 \tag{4.18}$$

c_1 = theoretical spring rate required [N / mm]

c_2 = available spring rate acquired from the spring rate tables [N / mm]

p_1 = permissible load deviation, usually < 25 [%]

p_2 = actual load deviation [%]

F = operating load [N]

s = working spring travel [mm]

5. Technical aspects

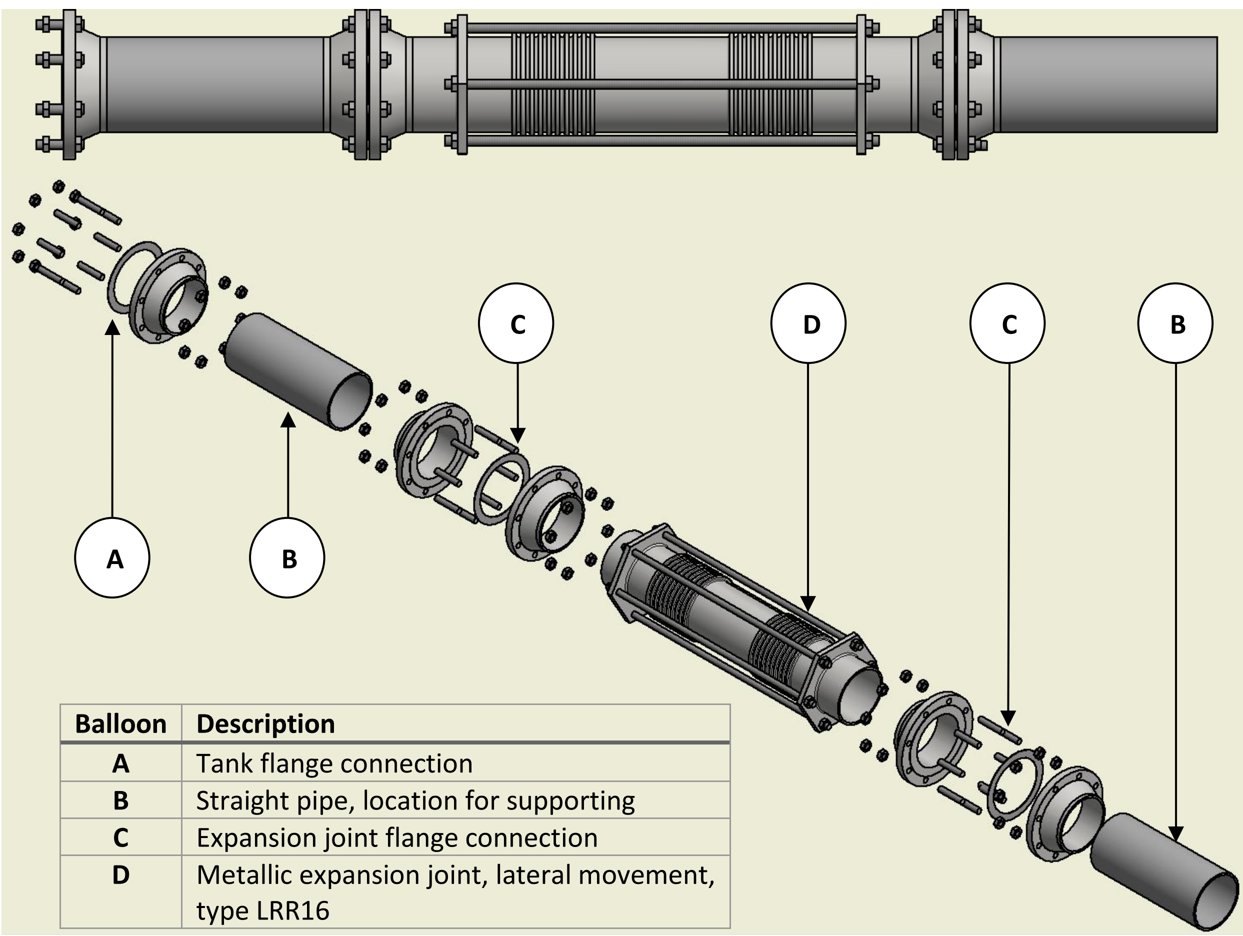
This section explains the technical aspects of the design alternatives for the replacement of Viking-Johnson couplings. These aspects are categorized as follows:

- Weight and complexity (§**5.1**)
- Support fabrication and installation (§**5.2**)
- Pipeline installation procedure (§**5.3**)

5.1 Weight and complexity

Piping components are very heavy, for example a 6” flange weighs 10.6 kg whereas an 8” flange weighs almost double (17.6 kg). The maximum weight that a man is allowed to lift without the use of mechanical help is 23kg (Dutch Ministry of Social Affairs and Employment, 2018). In the case of increased lift frequency or tougher work conditions, the maximum weight allowed decreases substantially (NIOSH-method). For this reason, the complexity of each design alternative is measured by the sheer number of components to be installed. In **appendix VIII**, a set of tables can be found with the precise number of components to be used for each alternative.

The first alternative is the simplest solution, it uses only a flange connection and a straight pipe. The second alternative makes use of extra piping components to increase the pipe flexibility and is thereby more difficult to install. The third alternative is the most compact solution but also the most complex one. **Figure 5.1** depicts an exploded view of this alternative.



Balloon	Description
A	Tank flange connection
B	Straight pipe, location for supporting
C	Expansion joint flange connection
D	Metallic expansion joint, lateral movement, type LRR16

Figure 5.1: Exploded view of alternative 3 (Expansion joint - made with Inventor)

5.2 Support fabrication and installation

The installation and fabrication of supporting will be split into two categories:

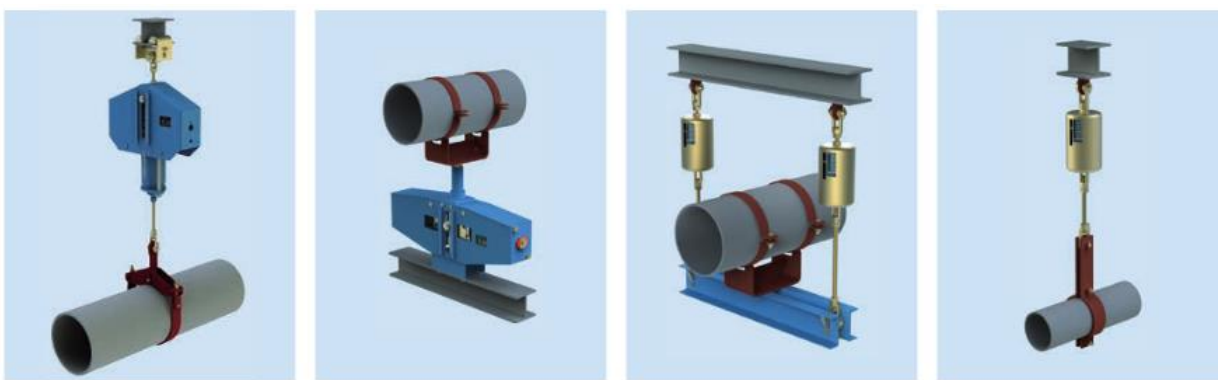
- Regular supports
- Spring supports

Regular supports

These supports are prefabricated in the shop and later transported on-site for installation. The profiles are welded together to form simple, yet tough, constructions. Usually, clients in the oil- and gas industry have their own standard drawings for pipe supports and support foundations. Existing foundations are frequently re-used for the installation of pipe supports. When this is not possible, new foundations are placed into the ground. For these reasons, the installation of regular pipe supports is usually relatively easy, given that there is enough workspace available to work on the support and the foundation.

Installation of spring supports

Each type of spring support comes with an installation manual from the manufacturers. Usually, the spring assembly needs to be either bolted or welded to a regular supporting structure (see **figure 5.2** for an example).



(a) Constant load hanger (b) Constant load support (c) variable spring hangers

Figure 5.2: Spring support installation examples (Source: Lisega, 2016)

5.3 Pipeline installation procedure

The existing pipeline (including the Viking-Johnson coupling) inside of the tank pit is disconnected from the tank Nozzle flange. Depending on the alternative chosen, the existing pipeline is cut through with a special pipe cutter tool. Then the supporting is placed and the elevations are checked. After this stage, the pipeline can be transported to the location for installation. Pipe lengths, are transported with trucks and lifted with cranes. The type of transportation and lifting cranes are specifically chosen depending on the available space to move around and to place the pipelines. The pipes are placed on top of the pipe supports and the butt ends of the pipes are welded together. This is called a field weld. Then all the flanged connections are tightened with the correct torque requirements for each bolt. After the pipeline is fully installed, a non-destructive test (X-ray) is performed to check the weld quality of the pipeline. If the test passes the quality check, a hydrotest is performed. The pipeline is fully filled with water and pressurized at 1.5 times the design pressure (ASME B31.3, 2016). This test is performed at ambient temperature. If the pipeline passes this test then the pipelines is painted with a protective paint layer and is ready for use.

6. Maintenance aspects

When the construction of the pipeline is finished the life cycle of the pipeline starts. During this period, the pipeline undergoes a series of maintenance. To put the life cycle into perspective, the following categories are introduced:

- Commissioning testing
- Service life
- De-commissioning and recycle

Commissioning testing

During commissioning of a project, the pipeline undergoes a series of rigorous tests and checks to ensure that the pipeline is safe for operations before officially handing over the pipeline to the client. One of the tests is the hydrotest that was mentioned in the previous chapter.

Service life

The service life of the pipeline forms the core of the pipeline life-cycle. Every oil refinery has its own maintenance plan and procedures for every process within the refinery. Examples are production and storage of hydrocarbons. The type and frequency of maintenance depends on the type of process. The expansion joint, however, needs to be removed for separate cleaning and inspection. A list of possible maintenance types is listed below:

- Pipeline chemical cleansing - The use of chemicals to remove product from the pipe walls.
- Pipeline pigging - The use of a pig to clean and inspect the inside of the pipeline (see **figure 6.1**).
- Pipeline flooding - Using water to flood and flush the pipeline to remove dirt and debris.

Figure 6.1: Pig modules for cleaning and inspection (Source: Fraser Engineering Company, Inc., 2013)

De-commissioning and recycle

At the end of the service life of the pipeline (~20 years), the pipeline is decommissioned from service, it is cleaned and thrown away or can be sold for recycling purposes.

7. Safety aspects

The safety aspects of the three alternatives are analyzed using the FMEA method (Failure mode Effect Analysis). The FMEA method prioritizes failures according to their significance (S), risk probability (O) and their risk detection probability (D). The result is a risk priority number (RPN), see equation (**7.1**).

$$RPN = SOD \tag{7.1}$$

S = significance coefficient [--]

O = Risk probability coefficient [--]

D = Detection probability coefficient [--]

The RPN is calculated for the entire design with the goal to identify potential risks and to take mitigating action to prevent critical failures. According to a research paper by the Canadian Center of Science and Education (2015), a FMEA analysis in the Oil- and Gas industry can be performed using a calculated scale for evaluating the probability and impact of risks to quantify the factors (S), (O) and (D). These scales can be found in **appendix IX**. The calculation of RPN can be found in **appendix X**. Only the failures that constitute a high risk due to a high RPN are explained in this section (see **table 7.1** in the next page).

Mitigation actions need to be taken to reduce or eliminate the risk of potential failures described in table 7.1. The mitigation actions are summarized as follows:

Hydraulic shock
Operators need to properly handle the transport of product through pipelines. This means that valves cannot be opened or closed abruptly. The proper use of pumps is also recommended to prevent large changes in momentum of the fluid inside of the pipeline. During detail design of the alternatives, a flexibility analysis should determine if there is enough flexibility in the pipeline to (partially) absorb the effects of hydraulic shock.

Dirt in the spring housing of supports
Proper maintenance and higher check intervals should decrease the chances of dirt accumulating inside of the spring housing.

Internal pipe erosion
Proper maintenance of the pipeline and increased pipeline intervals should decrease the chances of grime or dirt accumulating inside of the pipeline which decreases the effect of erosion.

Torsion of bellows
Proper tank settlement inspection should reveal in a timely manner if the tank settlement is too large or if improper tank settlement is occurring. If such cases are predicted to happen, then corrective measures should be taken such as tilting the tank and reinforcing the ground. These corrective actions will prevent the bellows of the lateral expansion joints from rupturing due to large shear stresses created by torsion.

No.	Type of failures	Significance	(S)	Risk probability	(O)	Detection probability	(D)	RPN
1	Hydraulic shock	Hydraulic shock can lead to rupture of piping components such as elbows and flange connections.	10	Hydraulic shock occurs when the fluid inside of the pipe undergoes an abrupt change in momentum.	3	Very difficult to detect. Operators need to be very careful when opening or closing valves and handling pumps.	4	120
2	Excessive dirt in the spring housing of support	Dirt can cause high friction of the springs and can cause a jam which can lead to increased nozzle stresses.	6	Infrequent checks and improper maintenance can increase the risk of dirt accumulating.	5	Dirt inside of the spring housing can only be detected during a check or during maintenance.	3	90
3	Internal erosion of the pipe	Erosion causes thinning of the pipe thickness and can lead to weakening of pipeline strength.	7	Grime inside of the pipeline can increase the chances of erosion if maintenance frequency is low.	4	The NDT-(X-ray) test is meant to detect the quality of the weld and thus detect any imperfection.	3	84
9	Excessive torsion of the bellows	Asymmetric tank settlement can cause excessive torsion of the bellows which can create high material shear stresses.	9	Settlement of tanks occur slowly over time and is monitored by engineers.	4	Tanks inspection occurs periodically. Tanks settlements are monitored and in case of improper settlement, corrective actions are taken.	2	72

Table 7.1: Risk priority number (PNR) – All design options

8. Financial aspects

To analyze each alternative in terms of costs, the DACE book (Dutch Association of Cost Engineers, 2012) is used as a reference. This book gives the estimated prices for each piping component and an estimation for the installation of the complete piping system. These estimations already include the costs of man labor, transportation and welding. The costs are split into component costs and installation costs. See **tables 8.1** and **8.2** for a cost estimate of each alternative.

Design type	Unit	Pipe sizes					
		6”	8”	10”	12”	14”	16”
Straight pipe (2m)	€	82	122	166	228	264	296
L-pipe	€	1200	1700	2500	3800	4800	7000
Pipe loop	€	1500	2300	3400	5000	6600	NA
Expansion joint	€	3200	3900	4700	6000	6800	7700

Table 8.1: Cost estimate of piping components

Design type	Unit	Pipe sizes					
		6”	8”	10”	12”	14”	16”
Straight pipe (2m)	€	711	936	1245	1633	2258	3067
L-pipe	€	4000	6400	10200	16800	26600	46400
Pipe loop	€	4600	7200	11200	19100	31400	NA
Expansion joint	€	1900	2400	3100	3900	5000	6400

Table 8.2: Cost estimate of pipeline installation

As can be seen in the cost estimate tables above, the expansion joints are the most financially feasible due to the relatively low installation costs. The reason for the high cost estimate values is that the overall length of the pipelines is quite large. These lengths are necessary to ensure the pipe flexibility and thus keep the nozzle stresses low. The design of the expansion joints is very compact and therefore do not requires long pipelines.

The need for more compact solutions

It is possible that another pipe routing could make the pipeline more compact while still maintaining its flexibility. This can be achieved by introducing more pipe bends into the design and by using the third dimension (height) to route the pipeline. This in turn would make the design more complex to calculate but it could also reduce the installation costs dramatically if the overall pipe length is kept short. According to DACE (2012), the installation costs per meter for a 6” pipeline is approximately €500,- euros. This price increases with pipe size, e.g. for a 12” size the installation costs are approximately €1000,- per meter.

9. Selection matrix method

This chapter explains how the selection matrix model is constructed and how it is used (§**9.1**). Also, the benefits of the selection matrix model are introduced in this chapter (§**9.2**).

9.1 Selection matrix model

The selection matrix model serves as an Excel tool to quickly identify the best alternative for a given tank connection scenario. A flowchart can be found in **appendix XI**. The scope of this scenario is determined at the start sheet of the model.

START sheet

This sheet contains dropdown menus with variables to choose the scope of the model (see **figure 9.1**):

- Tank settlement [mm]
- Pipe size [inch]
- Pipe specs [--]
- Design temperature [^{0}C]
- Design pressure [bar(g)]

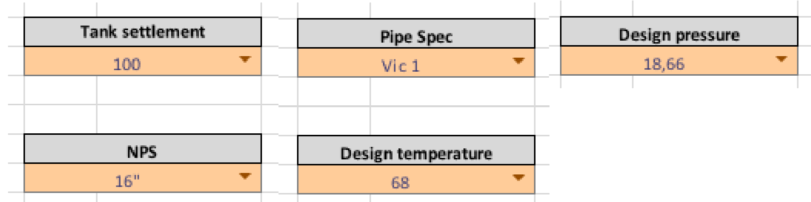


Figure 9.1: Selecting the scope of the model

All the calculations performed with Caesar II and the FEM analysis with Nozzle Pro are related to this scope. Calculations need to be performed again for a different scenario, for example a different tank settlement or different pipe specs. The results of these calculations are stored in a database sheet in the model.

DATABASE sheet

The database contains the results of the calculations in terms of minimum pipe lengths required for each design alternative and for each pipe size. These values are linked with the sheets containing the design options. The database also keeps track of all the pipe components required for each design.

Design option sheets

These option sheets are separated into alternatives 1, 2 and 3. The dimensions of the design can be typed in the input fields of the sheet (see **figure 9.2**). A "PASS" or "FAIL" message indicates if the dimensions are long enough according to the values in the database. This message is achieved with simple logic codes (IF, THEN, ELSE) and cross-sheet references. There is also the option to use a Macro scripts to find the minimum lengths of compensating leg required for a given pipe length. The macro scripts are linked to buttons in the sheet and uses the "Solver" add-in. This add-in must be manually turned on when the Excel model is opened for the first time and the macros need to be allowed to run.

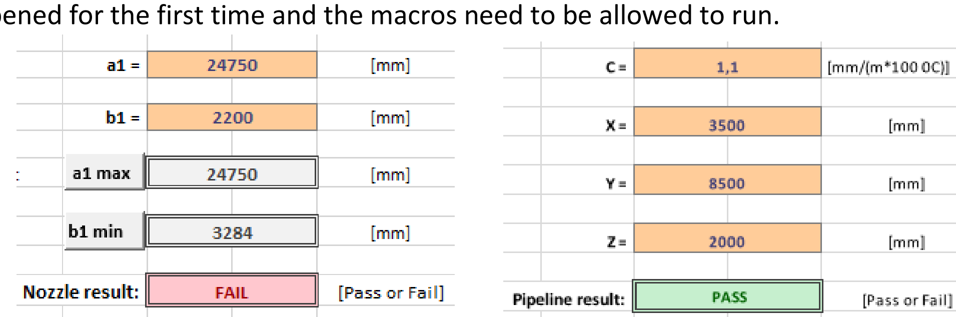


Figure 9.2: Input fields

The total flexibility of the existing pipeline inside the tank pit is checked with the Kellogg equation. The sheets containing the design options have input fields to type in the sum of the lengths of the pipeline dimensions in the X-, Y- and Z-axis. These input fields are linked with the database sheet were the calculations occur automatically in the background. The user sees only a “PASS” or “FAIL” message when the total pipeline dimensions are typed in.

Selection table

The selection matrix table uses criteria and weighing factors to determine the best alternative (see **figure 9.3**). The relative scores are determined in the database. For example, a straight pipe is the easiest to install, therefore it scores a 10 for pipeline installation. A pipe loop on the other hand has more pipe components and is longer, therefore it scores a 4. The metallic expansion joint is simpler to install than a pipe loop but more difficult than a straight pipe, therefore it scores a 7. This reasoning is repeated for all aspects.

SELECTION MATRIX TABLE												
	OPTIONS	**Criteria**								**Results**		
		Technical		**Maintenance**		**Safety**		**Financial**				
		Pipeline installation	Support installation	Pipeline cleansing	Inspection	Leakage prevention	Flexibility and stress	Components	Installation	**RAW SCORE**	**WEIGHTED SCORE**	**RANK**
		***Weighing factors* [1 - 10]**										
		5	**3**	**7**	**5**	**5**	**7**	**3**	**7**			
1	**Straight pipe**	3 4 5 6 7 8 9 10	27	56	40	50	14	30	70	67	1851	**1**
2	**Extra flexibility**		15	56	40	50	28	18	14	47	1335	**3**
3	**Expansion joint**	[illegible]	30	28	20	5	70	6	42	44	1388	**2**

Figure 9.3: Selection matrix table

On the results section in the selection table, the scores are grouped and ranked. In this part there is a distinction made between raw scores, weighted scores and ranking. The ranks are based on weighted scores. The highest score is ranked first; this is the best option. In the example above, the straight pipe is the best option. This option is the simplest one and the most economical. The second best is the expansion joint. Although the cost of the expansion joint is higher than the other options, the installation costs is substantially lower than a pipe loop. If the weighing factors are changed, for example if all the aspects are equally important, then option 3 (expansion joint) ranks third. This can be seen in the results section under raw scores.

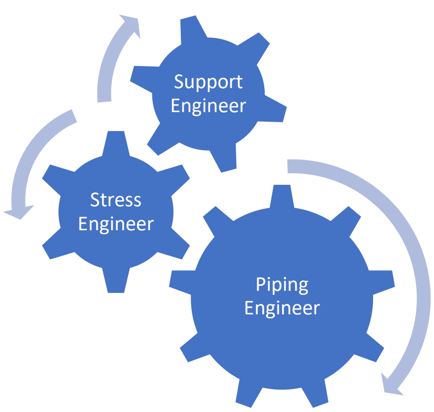
9.2 Benefits of using the selection matrix

The selection matrix is meant to be used as a pre-selection tool during feasibility studies to quickly identify the most suitable alternative for the replacement of the Viking-Johnson couplings. The selection matrix connects perfectly with feasibility studies because the stress- and flexibility has already been calculated for the different design alternatives. Furthermore, this thesis maps the alternatives that can be used to replace Viking-Johnson couplings and therefore results in time saving.

The benefits of using the selection matrix are categorized as follows:

- A quick way to select an alternative for any tank scenario (given that the scope of the selection matrix is further expanded);
- Time reduction of the feasibility studies since the alternatives have already been mapped;
- Increased chance of closing a deal with the client due to faster response time;
- During the detail engineering phase, the stress engineers have to perform only check calculations to the detail designs. The chances of a failed design due to high nozzle stresses are substantially reduced because stress- and flexibility has already been taken into account in the selection matrix, which can save time;
- Reduction of project lead time can result in:
  - Increased profit margin if the contract has a fixed price;
  - Better market position if the project is sold at a lower price;
- The selection matrix can be expanded cheaply by intern students if they follow the same analysis steps shown in the thesis. A win-win situation is created where intern students get better acquainted with the stress side of piping and the relation with design choices.

To better understand the relations between a piping engineer and a stress engineer, the following workflow is shown:



Workflow

The Piping Engineer designs the piping system. When the piping system is critical and needs to be analyzed for safety or load on equipment nozzles, the Piping Engineer hands over the design to the Stress Engineer. In this stage, the stress engineer calculates the stresses on the pipeline and on nozzles by using Caesar II and Nozzle Pro. If the stresses are too high, the design is handed back over to the Piping Engineer for design revision. The supports are designed by the support engineer. In smaller engineering bureaus the Piping Engineer also have the task of designing the pipe supports. This iterative type of workflow can be time consuming. With the selection matrix, the chances of a good design straight off the bat are increased during the detail design phase.

10. Potential clients

Potential clients in the Netherlands

The Europoort and the Botlek are part of the Rotterdam municipality. These locations form the core of the petrochemical industry of the Netherlands. For this reason, the majority of the companies that stores flammable and dangerous products are located in the Europoort and Botlek. Other potential clients in the Netherlands are tank terminals for storage of oil and dangerous chemicals. These are mainly located in Amsterdam. **Table 10.1** shows a list of potential clients with large storage tanks with flammable or dangerous products.

No.	Company name	Type	Location	Website
1	Maasvlakte Olie Terminal	Storage	Maasvlakte	www.mot.nl
2	Neste	Refinery	Maasvlakte	www.neste.nl
3	BP refinery	Refinery	Europoort	www.bp.com
4	Shell	Refinery	Europoort Botlek	www.shell.com
5	Gunvor	Refinery	Europoort	www.gunvorgroup.com
6	Exxonmobil	Chemical plant Refinery	Europoort Botlek	www.exxonmobil.be
7	Rotterdam Antwerpen Pijpleiding	Storage	Europoort	www.rapl.nl
8	Vitol Processing Rotterdam	Refinery	Europoort	www.vprenergy.com
9	Vopak	Storage	Europoort Botlek	www.vopak.com
10	Lyondell Chemical Netherlands	Chemical plant	Botlek	www.lyondellbasell.com
11	Emerald Kalama Chemical	Chemical plant	Botlek	www.emeraldmaterials.com
12	LBC Rotterdam	Storage	Botlek	www.lbctt.com
13	Odfjell Terminals	Storage	Botlek	www.odfjell.com
14	Koole Terminals	Storage	Botlek	www.koole.com
15	Tankstrorage Amsterdam	Storage	Amsterdam	www.tankstorage-amsterdam.nl
16	Eurotank Amsterdam	Storage	Amsterdam	www.vtti.com
17	Alkion Terminal Amsterdam	Storage	Amsterdam	www.alkion.com
18	NuStar Terminals	Storage	Amsterdam	www.interterminals.com
19	Standic	Storage	Dordrecht	www.standic.com
20	Zeeland refinery	Refinery	Zeeland	www.zeelandrefinery.nl

Table 10.1: List of potential clients

Potential clients in the EU

Oil refineries and storage companies in the EU operate according to the local legislations of their country. Due to time constraints, it is not possible to determine if Viking-Johnson couplings are being phased out from the oil- and gas industry of other countries. For this reason, only potential clients in the Netherlands were listed.

11. Competence development

At the start of the thesis, the level of competence is determined by the final report of the competence development assessment made at the end of the 5th study semester (see **appendix XII** for the assessment).

Competence level at the end of the thesis period

Table 12.1 shows the competence level that was maintained during the effectuation of the thesis.

Competence type	Level	Official description	Proof in the thesis report
Professional skills	3	Skills needed to effectively carry out the engineering competence levels.	Skills acquired by learning how to use Caesar II and Nozzle Pro. These skills are necessary to carry out the research and stress analysis. (See paragraph §**4.1**)
Research	2	The engineer has a critical investigative attitude and uses suitable methods regarding the gathering and judgement of information to perform an applied research.	A calculation check based on literature calculations is performed to evaluate the results obtained with Caesar II. This shows a critical attitude towards the realization of the research. (See paragraphs §**4.2**, §**4.3** & §**4.4**)
Design	3	The realization of an engineering design and the collaboration with engineers and non-engineers. The design could be a device, a process or a method. The impact on aspects such as social, health, safety, environment, durability and commercial are taken into account. The design is based on predetermined design criteria.	The design consists of two parts: • Three design alternatives for the Viking-Johnson couplings; • An Excel tool for selecting the best alternative. The designs cover technical, safety, maintenance and financial aspects and is performed according to design criteria based on local legislations and international codes. (see chapters **3**, **5**, **6**, **7, 8** & **9**)
Analysis	3	The analysis of an engineering problem includes the identification of the problem, consideration of solution directions and the mapping of requirements. Methods such as mathematical analysis, computer models, simulations and experiments can be used. Boundary conditions regarding financial, social, safety, environment and durability are considered.	The analysis consists of identifying the problem: safety issues of Viking-Johnson couplings according to PGS-29. The solution directions are the design alternatives found by a literature and market search. The stress- and flexibility analysis is performed with Caesar II, Nozzle Pro and the theory of cantilever beams. The boundary conditions are specified by legislation (PGS-29) and international codes (ASME B31.3 & API 653). (See chapters **1**, **2**, **4** & **10** and paragraphs §**3.1.4**, §**3.1.5**, §**3.2** & §**9.2**)

Table 12.1: *Competence level for the graduation thesis*

Conclusions and recommendations

Conclusions

Viking-Johnson couplings are being phased out from the oil- and gas industry within the Netherlands due to stricter Dutch legislations regarding the above ground storage of flammable fluids (PGS-29). At the moment, it is unknown whether the phasing out of the Viking-Johnson couplings also apply to countries outside of the Netherlands because every EU-country operates under their own local legislations.

The purpose of this thesis was to establish a procedure for the selection of alternatives for Viking-Johnson couplings. This procedure must hold true for any type of tank connection scenario, that involves the handling of petrochemicals. With this purpose in mind, a selection matrix tool was created in Excel which contains a database with pre-calculated alternatives using Caesar II and Nozzle Pro. This tool also has a selection table to select the best alternative in any scenario. The scope of this tool can be selected in the front page and it is fully determined by the completeness of the database. The thesis describes and tests a procedure for calculating and analyzing alternatives. This procedure is essential for adding more scenario cases to the database and therefore increasing its scope.

According to literature findings and Dutch legislations, it can be concluded that the following alternatives are currently the only options available to replace the Viking-Johnson couplings from the oil- and gas industry in the Netherlands:

- A straight pipe to replace the coupling if the inherent flexibility of the pipeline allows it;
- The use of elbows to create L-shaped pipelines and pipe loops if extra flexibility is needed;
- Metallic expansion joints (flanged or welded) provide the highest flexibility range possible and therefore can be used in any tank settlement scenario and for any pipe size.

The nozzle stress FEM analysis using Nozzle Pro show that pipelines above 16” become too heavy for alternatives concerning a straight pipe or a pipe using elbows due to the large length of the pipe. For this reason, it is recommended to design special pipe supports to counteract the weight of the pipeline for sizes beyond 16”. Expansion joints are not subjected to these limitations.

Recommendations

- Due to time constraints, the stress- and flexibility calculations procedure in the thesis only cover pipelines where the point of rotation is located along the x-axis of the tank nozzle. Further research is recommended to map the effects and boundaries of pipe routings where the point of rotation is located at an offset from the nozzle x-axis.
- Vicoma should use the analysis procedure described in the thesis to expand the scope of the selection matrix tool to other pipe specs, having different process conditions and with different tank settlement values; the thesis used a worst-case scenario (largest allowed tank settlement) as a starting point.
- Further research is needed to map the effects of tank keeling (tilting about the x-axis) on the connected pipeline.

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Appendix list

Appendix I: Theory of process piping and storage

PIPELINES

According to the Merriam-Webster dictionary (pipeline, 2018), a pipeline is defined as a pipe system connected to pumps, valves and control devices for the transportation of substances such as fluids, gasses or grains. Also, in the process and piping industry there is a distinction to be made between a pipe and a tube. A pipe is used as the main transportation object for substances. The pipe sizes can range from 1/2" up to sizes larger than 24”. Whereas a tube is the term used for small diameter connections, usually 3/4" or less in diameter, used to connect instruments to the pipeline for monitoring purposes.

Note that in the oil industry, the use of inches is usually preferred to specify the nominal size of the pipe. However, in Europe and in other parts of the world, there are companies that use the metric notation instead to specify the nominal size of pipes. For this reason, the use of inches shall be used throughout this report to specify the nominal size of pipes, whereas all other units shall be in accordance with the international system of units (SI metric notation).

Pipe dimensions

There are several methods to indicate pipe dimensions like nominal pipe size and wall thickness. The most popular methods used in Europe are the DIN and ANSI methods (listed below). **Table A.1** provides a standard comparison table that integrates these methods into one overview (*Tioga Pipe Inc., 2013*).

- DIN - Deutsche Institut fur Normung - [*Metric units*];
- ANSI - American National Standards Institute - [*English units*].

U.S./METRIC

NOMINAL PIPE SIZE	OD	SCHEDULE DESIGNATIONS			WALL THICKNESS		WEIGHT		ID	
INCH MM	INCH MM	ASME			INCH	MM	LBS/ FOOT	KG/ METER	INCH	MM
5	5.563	5		5S	0.109	2.77	6.36	9.46	5.345	135.76
125	141.3	10		10S	0.134	3.40	7.78	11.56	5.295	134.50
		STD	40	40S	0.258	6.55	14.63	21.77	5.047	128.20
		XS	80	80S	0.375	9.53	20.80	30.97	4.813	122.24
		120			0.500	12.70	27.06	40.28	4.563	115.90
		160			0.625	15.88	32.99	49.12	4.313	109.54
		XX			0.750	19.05	38.59	57.43	4.063	103.20
6	6.625	5		5S	0.109	2.77	7.59	11.31	6.407	162.76
150	168.3	10		10S	0.134	3.40	9.30	13.83	6.357	161.50
					0.188	4.78	12.94	19.28	6.249	158.74
		STD	40	40S	0.280	7.11	18.99	28.26	6.065	154.08
		XS	80	80S	0.432	10.97	28.60	42.56	5.761	146.36
		120			0.562	14.27	36.43	54.21	5.501	139.76
		160			0.719	18.26	45.39	67.57	5.187	131.78
		XX			0.864	21.95	53.21	79.22	4.897	124.40

Table A.1: Pipe size chart (DIN vs ANSI), (Tioga Pipe Inc.,2013)

Nominal Pipe Size vs Diametre Nominal

The nominal pipe size (NPS) refers to the American notation to specify the rough average (or nominal) diameter of the pipe for commercial purposes and its dimensions are given in inches. The DIN however, denotes these measures as DN (Diametre Nominal) instead of NPS and uses the metric units. In **figure A.1**, the location of the NPS dimension can be seen (dimension d_n). The exact location of the NPS may vary depending on the pipe size. Here, the inside- and outside dimeters are denoted d_i and d_o respectively.

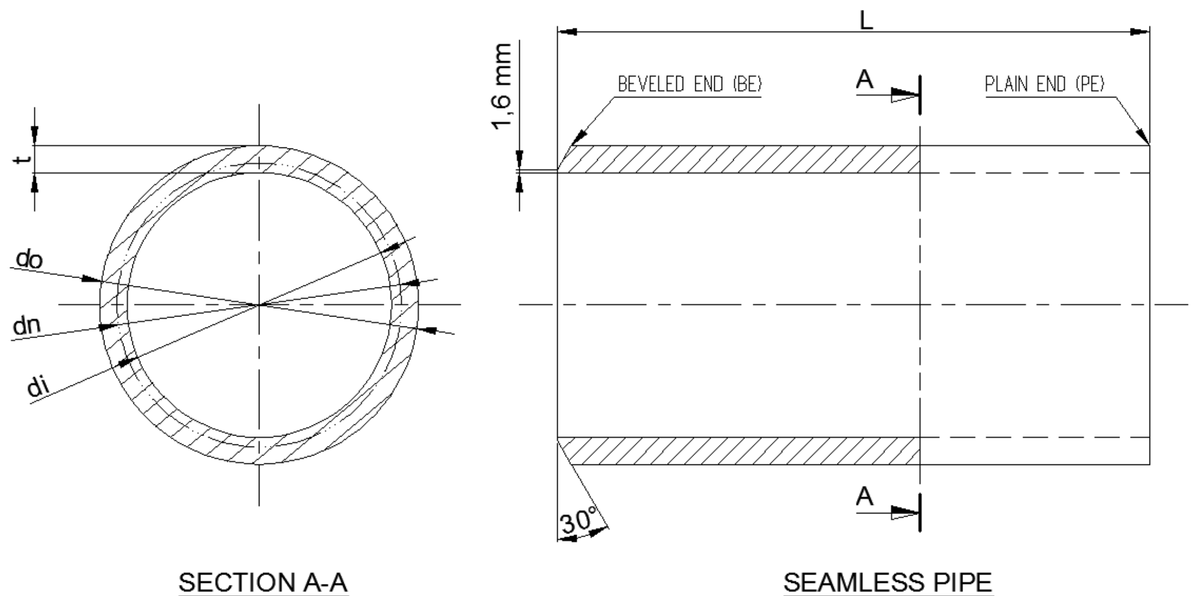


Figure A.1: Dimensions of a seamless pipe

Note that the ends of the pipe, as seen in this figure, can vary depending on the type of welding required. The beveled ends types are required for welding two pipe ends directly to each other. Whereas the plain end is used in socked weld connections; the end of the pipe is inserted inside of a socket and then welded together. These types of pipe ends do not have any influence on the location nor size of the NPS.

Pipe diameter, Schedule designation and weight

The dimensions of a pipe consist mainly on three variables, the inside- and outside diameters and the length (L) of the pipe. The outside diameter differs from the NPS in that it is an exact dimension, whereas the NPS is an approximate value. The thickness of the pipe is the difference between the inside- and outside diameter. However, in the piping world, an extra designation is used to describe the pipe thickness – the pipe schedule (*ASME / ANSI B31.10, 2018*). Through the years, there have been many different notations to specify the schedule of a pipe. Therefore, the ASME has provided a list of equivalent notations to specify the pipe schedule as can be seen in the pipe size chart. Equation **(A.1)** gives the correlation between the thickness and schedule of a pipe.

$$SCH = \frac{P_{op}}{\sigma_y}\, x\; 1000 \qquad \textbf{(A.1)}$$

SCH = Pipe schedule [--]

σ_y = Yield stress [Pa]

P_{op} = Internal operational pressure [Pa], In principle, the higher the schedule designation, the more pressure is allowed inside the pipe, thus the thicker the pipe must be. The only variable in the equation is the pipe thickness, since the allowable pipe stress is a material property (Yield stress) that remains constant for a given material. For example, a schedule 80 can withstand a pressure that is twice the pressure of schedule 40 for the same pipe size and material. Also, according to *ASME B36.19* (*2018*), schedules with the suffix "*S*" after the schedule number is used to designate pipe schedules for stainless steel pipes, as can be seen in the pipe size chart.

Wall thickness and Pipe weight

The exact pipe thickness of a pipe is calculated by taking several mechanical variables into account (*ASME B31.3, 2018*), as shown by equation **(A.2)**.

$$t_d = \frac{P_i d_o}{2(\sigma_y Q + PY)} + \mathrm{C} \qquad \textbf{(A.2)}$$

t_d = Minimal design thickness [mm]

P_i = Design internal pressure [MPa]

d_o = Outside diameter [mm]

σ_y = Yield stress [MPa]

Q = Quality factor [--]

Y = Thickness correction coefficient [--]

C = Mechanical corrosion allowance [mm]

The pipe specific weight is obtained with equation **(A.3)**.

$$\gamma = \rho A, \quad \text{with } A = \frac{\pi(d_o^2 - d_i^2)}{4} 10^6 \qquad \textbf{(A.3)}$$

γ = Specific weight [kg/m]

ρ = Material density [kg/m^3]

A = Cross-sectional area of the pipe [m^2]

d_o = Outside diameter [mm]

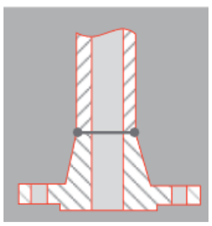
d_i = Inside diameter [mm]

FLANGES

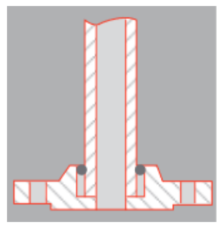
In the piping industry, pipelines are connected to valves, equipment and storage tanks typically via a flange connection, unless otherwise specified. A flange provides easy access for maintenance and inspection. For example, a valve can be taken out of the pipeline system by removing the bolts that are holding the flanges together. However, to ensure the integrity of the connection, the bolts, nuts and gaskets need to be replaced by new ones every time that a flange is disconnected.

Type of flanges

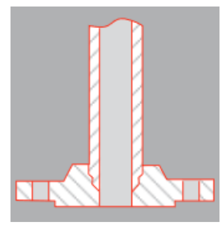
All flanges that are found in the market that fall in the category of 1/2" through 24” in size, need to be in accordance with the requirements specified in ASME B16.5 (*Pipe flanges and Flanged Fittings, 2017*). The most commonly used flanges in the petrochemical industry are shown in **figure A.2 (**a through f).



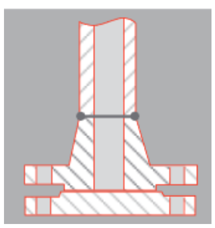
(a) Welding neck



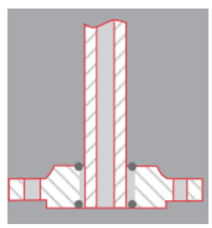
(b) Socket weld



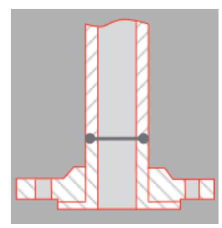
(c) Threaded



(d) Blind



(e) Slip on



(f) Lap joint

Figure A.2: Commonly used flange types (ULMA Technical Handbook, 2018)

Welding neck flange

The neck of the flange is directly welded to the beveled end of the pipe. The flanges are bored to match the inside diameter of the pipe to eliminate product flow restriction. This in turn prevents a turbulent flow which reduces material erosion. These flanges can be used in applications that involves high- pressures and temperatures.

Socket weld flange

The pipe plain end is inserted into the socket of the flange for welding. The shoulder of the flange - where the pipe end meets the flange - has the same thickness of the pipe so that product flow is not restricted. These flanges were initially developed to be used in small diameter, high-pressure pipelines for applications involving chemical processes, hydraulics and steam distribution.

Threaded flange

These types of flanges use threads that are tapered specifically to create a seal between the flange and pipe. However, such sealing mechanism does not allow applications that involve high- or cyclic pressures. These flanges can be used in low pressure hazardous environments where welding is restricted.

Blind flanges

These flanges have no bore and are typically used to (temporarily) seal ends of pipelines. Another function of these flanges is to provide access to pressure vessels, such as manholes in storage tanks. Usually, blind flanges are subjected to higher stresses than other flange types due to internal pressure and bolt loadings. This creates high bending stresses at the center of the flange. However, the design of the flange allows the bending stresses to be absorbed safely.

Slip on flanges

These flanges are designed to slide over the pipe. Welding is performed at end of the pipe inside the flange and at the hub (back end) of the flange. Therefore, permanently fixing the flange to the pipe. However, this method results in high shear stresses in the weld connections, thereby restricting the use of these flanges in applications that involves high pressures.

Lap joint flanges

These flanges are used in conjunction with a Lap Joint Stub End pipe. Basically, the flange slides over this piece of pipe which has a lap sticking out at one end to restrict the movement of the flange in that direction. The beveled end of the Lap Joint Stub End pipe is directly welded to the beveled end of the pipeline. This allows the flange to slide unrestricted over the pipeline after welding (up to the lap joint). These flanges are usually used in applications where frequent maintenance is required or in situations where bolt alignment is difficult.

Pressure ratings

Flanges are classified in several classes (or pressure ratings), depending on how much pressure they can withstand. These classes are usually denoted with a # symbol (*pronounced [pounds]*) and they depend on the material type of the flanges. For example, according to *ASME B16.5 (2017)*, 150# flanges that are fabricated out of forged carbon steel (A105) can withstand 17.7 bar(g) at 100 ^{0}C. Whereas, a 300# flange of the same material can withstand 46.6 bar(g) at 100 ^{0}C (see **table A.2**).

Nominal Designation	Forgings	Castings	Plates
C–Si	A 105 (1)	A 216 Gr. WCB (1)	A 515 Gr. 70 (1)
C–Mn–Si	A 350 Gr. LF2 (1)		A 516 Gr. 70 (1), (2)
C–Mn–Si–V	A 350 Gr. LF6 Cl. (4)		
$3\frac{1}{2}$ Ni	A 350 Gr. LF3		A 537 Cl. 1 (3)

Working Pressure by Classes, bar							
Class Temp., °C	150	300	400	600	900	1500	2500
−29 to 38	19.6	51.1	68.1	102.1	153.2	255.3	425.5
50	19.2	50.1	66.8	100.2	150.4	250.6	417.7
100	17.7	46.6	62.1	93.2	139.8	233.0	388.3
150	15.8	45.1	60.1	90.2	135.2	225.4	375.6
200	13.8	43.8	58.4	87.6	131.4	219.0	365.0

Table A.2: Pressure-Temperature Ratings for Group 1.1 Materials (ASME B16.5, 2017)

STORAGE TANKS

There are different regulations worldwide for the specifications regarding storage tanks. Within the petroleum industry, most tanks are designed according to the American Petroleum Institute code API-650. This code establishes the minimum requirements for material, design, fabrication, erection, and testing of aboveground cylindrical storage tanks. Furthermore, the Dutch PGS-29 guidelines for the storage of flammable products aboveground in vertical cylindrical tanks refers to the standards found in the API-650, among other codes.

Types of storage tanks

Storage tanks come in a wide variety of shapes and sizes, depending on the type of product to be stored and the conditions for storage. According to *Sölken (2018)*, there are essentially eight types of tanks for the storage of liquids:

- Fixed-roof tanks
- External floating roof tanks
- Internal floating roof tanks
- Domed external floating roof tanks
- Horizontal tanks
- Pressure tanks
- Variable vapor space tanks
- Liquified natural gas tanks (LNG)

Fixed-roof tanks

These tanks consist of a cylindrical steel shell erected vertically, with a fixed cone- or dome shaped rooftop. This tank is fully welded and is designed to retain both liquids and vapors without leaks. A breather valve is placed on the top of the fixed roof to prevent the tank from imploding if vacuum occurs inside of the tank. This valve allows outside air or inert gasses to flow into the tank so that the tank is always at atmospheric pressure.

External floating roof tanks

These cylindrical tanks make use of a special roof that floats on the surface of the fluid stored, rising and falling with the liquid level inside of the tank. External floating roof tanks are equipped with a rim seal system that slides against the tank wall as the roof is raised or lowered to prevent leakage.

Internal floating roof tanks and domed external floating roof tanks

The internal floating roof tank consists of both a fixed rooftop and a floating roof inside of the tank. The fixed rooftop can either be supported by vertical columns located inside of the tank or can be supported by the tank shell. This last one usually occurs when an external floating roof tank is converted to an internal floating roof tank. Such a tank is called a domed external floating roof tank.

Horizontal tanks

These types of tanks are usually small storage tanks compared to the tanks described above and are used for the storage of liquids above- and underground. To ensure the structural integrity of the tank, the length is not greater than six times its diameter.

Pressure tanks

According to *ASME Section VIII, (2018)*, a pressure vessel is a container of any shape for the containment of internal- or external pressure. An example of a pressure vessel is a sphere for the storage of fluids under pressure.

Variable vapor space tanks
These types of tanks use expandable reservoirs, e.g. flexible membranes, to accommodate vapor volume fluctuations that occur due to temperature or pressure changes. The loss of vapor only occurs if the vapor capacity of the tank is exceeded.

Liquified natural gas tanks (LNG)
These tanks are specifically designed for the storage of liquified natural gas -162 ^{0}C and therefore need to be insulated. At this temperature the natural gas condenses into a liquid. According to *Shell, (2018)*, LNG is a clear, colorless and non-toxic liquid that is shrunk 600 times the volume of the gas, making it easier and safer to be stored. In its liquid state, the LNG will not ignite. However, to conserve this state, the LNG needs to be kept under constant cooling.

Tank pits
The function of a tank pit is to retain the spills caused by a tank structural failure. These tank pits range in size, depending on the storage capacity and number of tanks. According to *PGS-29, (2016)*, tank pits must have the following volume available in case of a calamity:

- The tank pit must be able to retain 100% of the volume of the tank inside of the tank pit;
- If applicable, the volume of the tank pit must account for a foam layer to prevent the damp of toxic gasses;
- If applicable, the tank pit must account for the volume of fire- or cooling water in the case of a full-blown tank pit fire;
- The volume of rain water must be accounted for;
- Additional bund wall height of 15 cm in the case of waves created by wind unless it can be demonstrated numerically that this additional height is not necessary;

CODES and LEGISLATIONS

ASME B31.3: Process Piping
Since the year 1880, the American Society of Mechanical Engineers (*ASME*) has been writing codes to be used as standards within the engineering communities for the purpose of improving lives and livelihoods worldwide. One of these codes is the ASME B31.3, which is used for process piping found typically in petroleum refineries. Furthermore, this code is also used in other industries like pharmaceutical, textile, paper, semiconductor and cryogenics. This code covers all the rules regarding the design, materials and fabrication of piping systems (*ASME, 2018*).

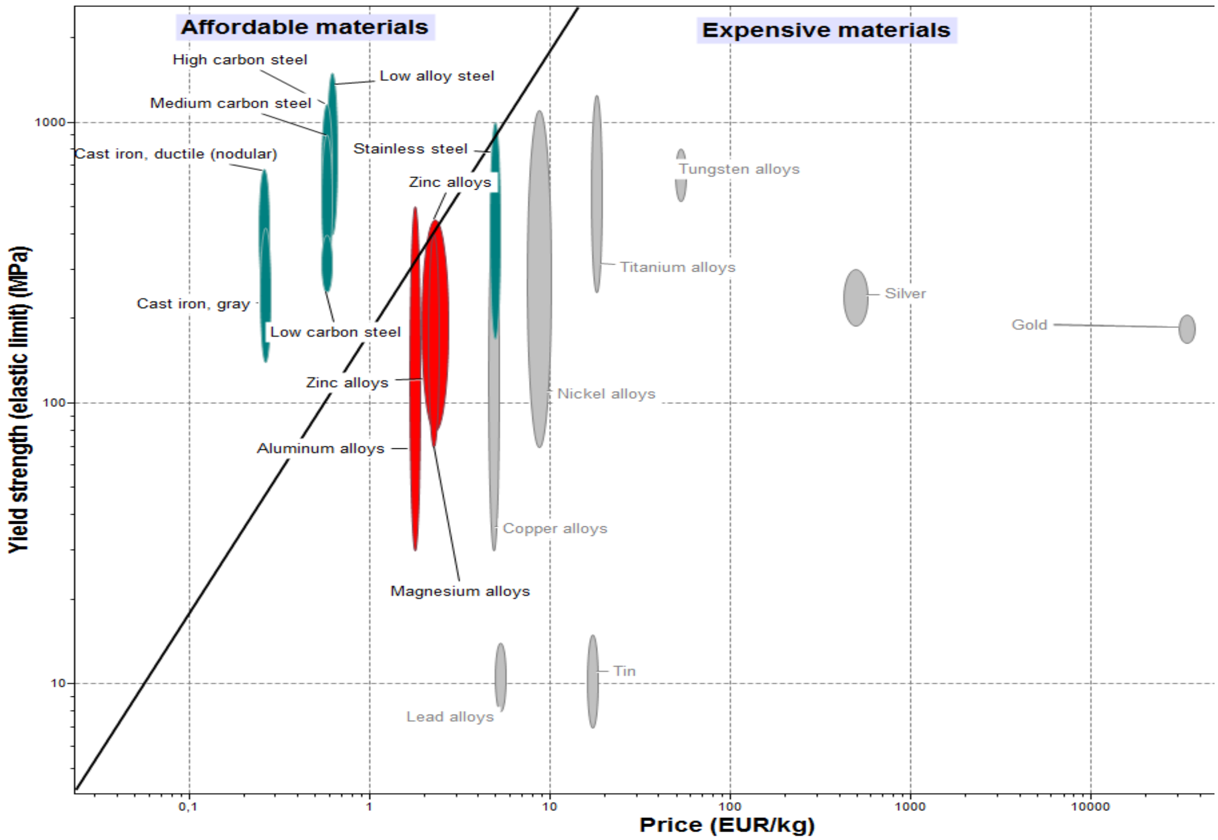
ASME B36.10, B36.19: Welded and seamless wrought steel pipe and stainless-steel pipes, respectively.
These codes deal with the rules for the design and construction of carbon steel, alloys and stainless-steel pipes. The pipe chart used throughout this thesis and other pipe charts throughout the world were created according to the rules found in these documents (*ASME, 2018*).

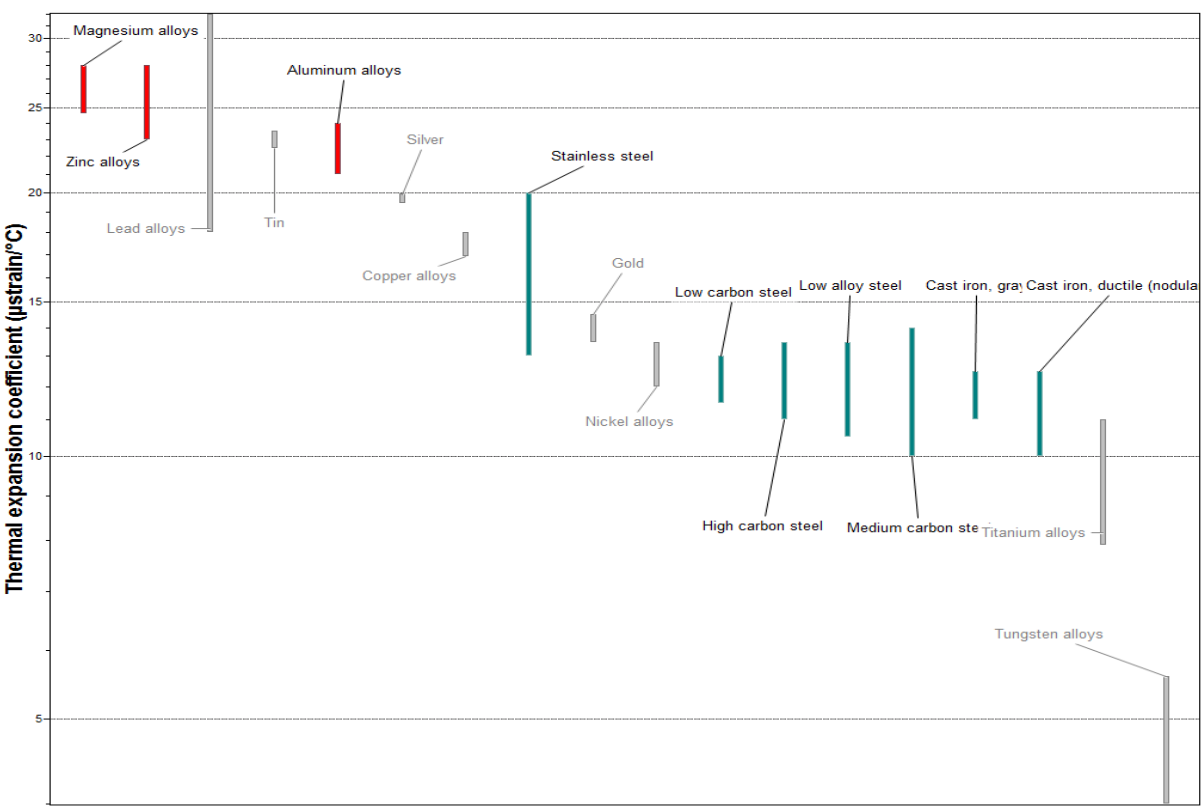
ASME B16.5: Pipe flanges and flanged fittings.
This standard deals with the rules regarding de design and fabrication of pipe flanges and flanged fittings for sizes 1/2" through 24” in size. This code also includes pressure-temperature rating tables for different material groups. Within these tables the maximum allowable pressure of a flange connection can be found for any given temperature and flange rating (*ASME B16.5, 2003*).

API-653: Tank inspection, repair, alteration and reconstruction.
The American Petroleum Institute (API) has been writing operating standards for the handling of petroleum, natural gas and petrochemical equipment and has more than 90 years of experience in the field. The goal of these standards is to promote operational safety across the globe. The API-653 deals with the engineering activities of existing storage tanks. This code also includes information regarding the allowable settlements of tanks (*API, 2018*).

PGS-29: Aboveground storage of flammable liquids in vertical cylindrical storage tanks.
The PGS-29 is a Dutch official document that contains guidelines for the storage of flammable liquids. PGS stands for *'Publicatiereeks Gevaarlijke Stoffen'* that translates into '*Series of publications regarding hazardous substances*'. The PGS-organization was officially created in 2008 with the intend to increase the safety within the Netherlands when dealing with hazardous substances (*PGS, 2018*). The PGS-29 in particular refers to international accepted codes such as the ASME B31.3 and the API-653 among other codes.

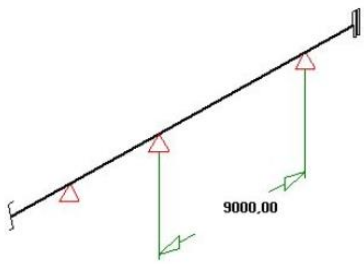
Appendix II: Ashby diagrams for material selection

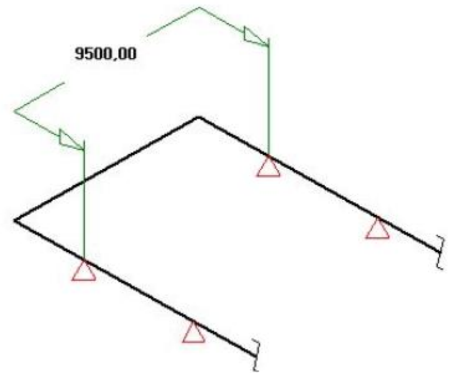




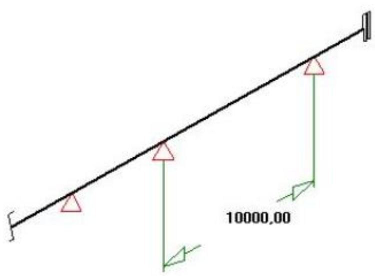
Appendix III: General safe spans of pipe supports

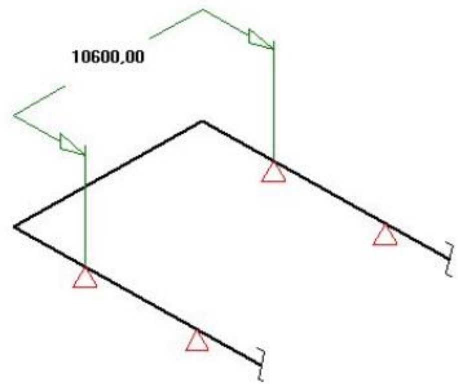
6" Sch-40 General safe spans



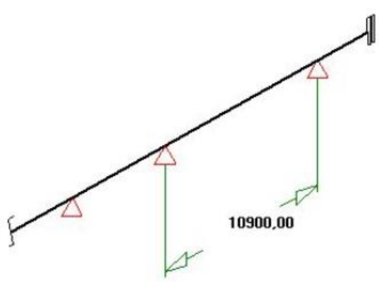


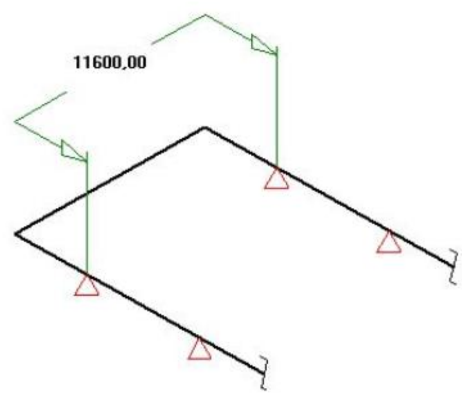
8" Sch-20 General safe spans



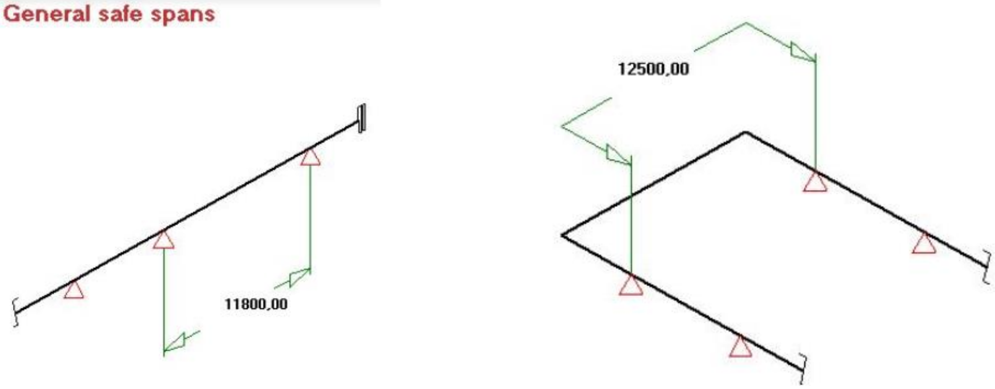


10" Sch-20 General safe spans

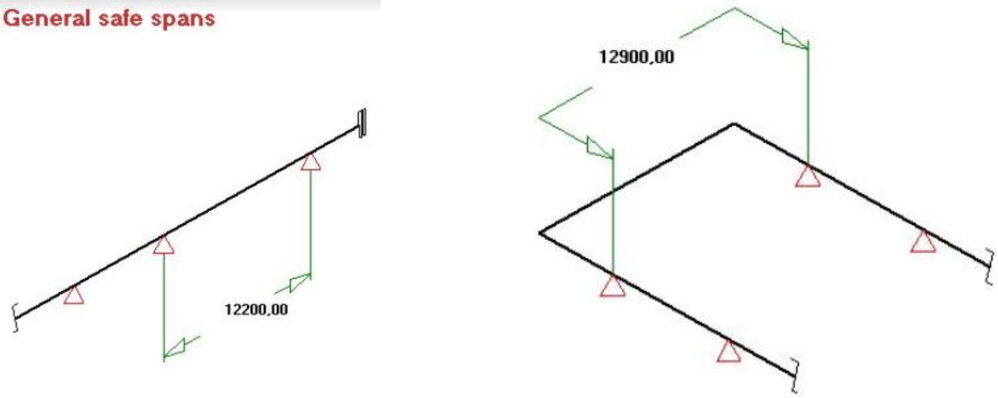




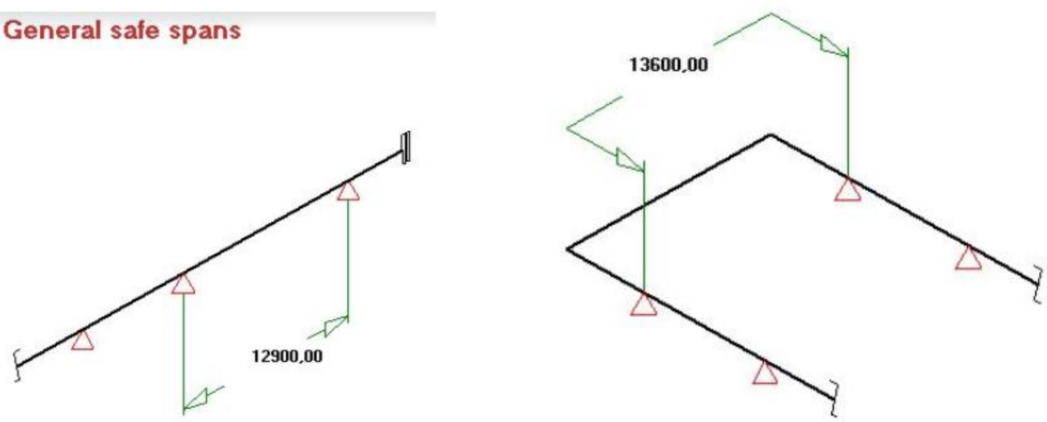
12" Sch-20 General safe spans



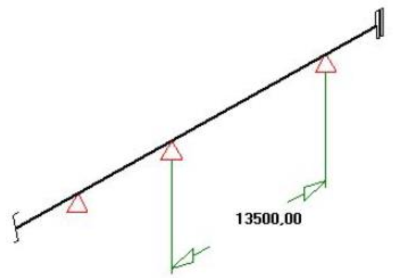
14" Sch-10 General safe spans

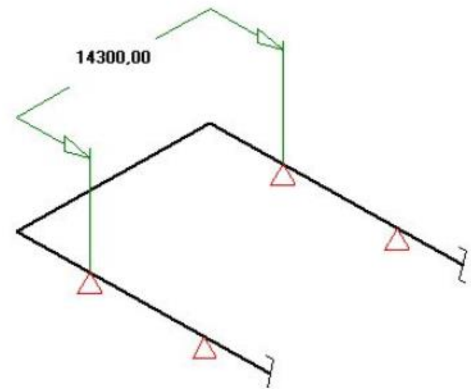


16" Sch-10 General safe spans

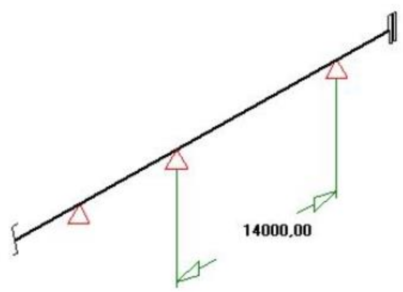


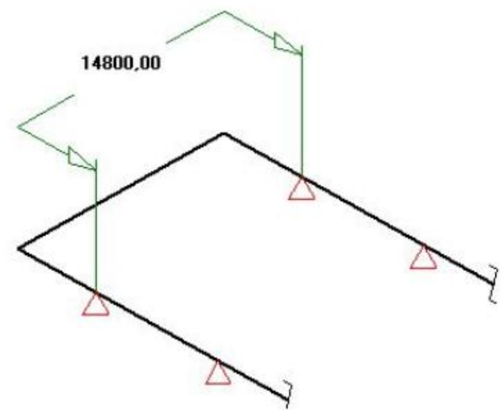
18" Sch-10 General safe spans



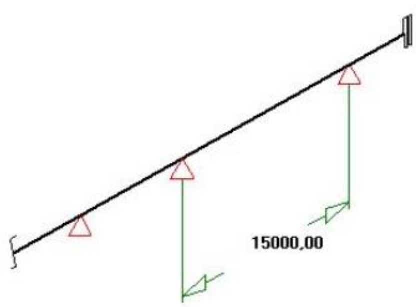


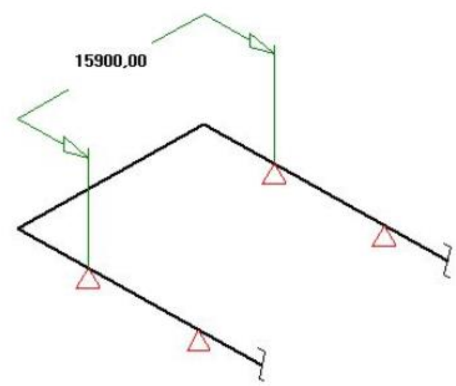
20" Sch-10 General safe spans



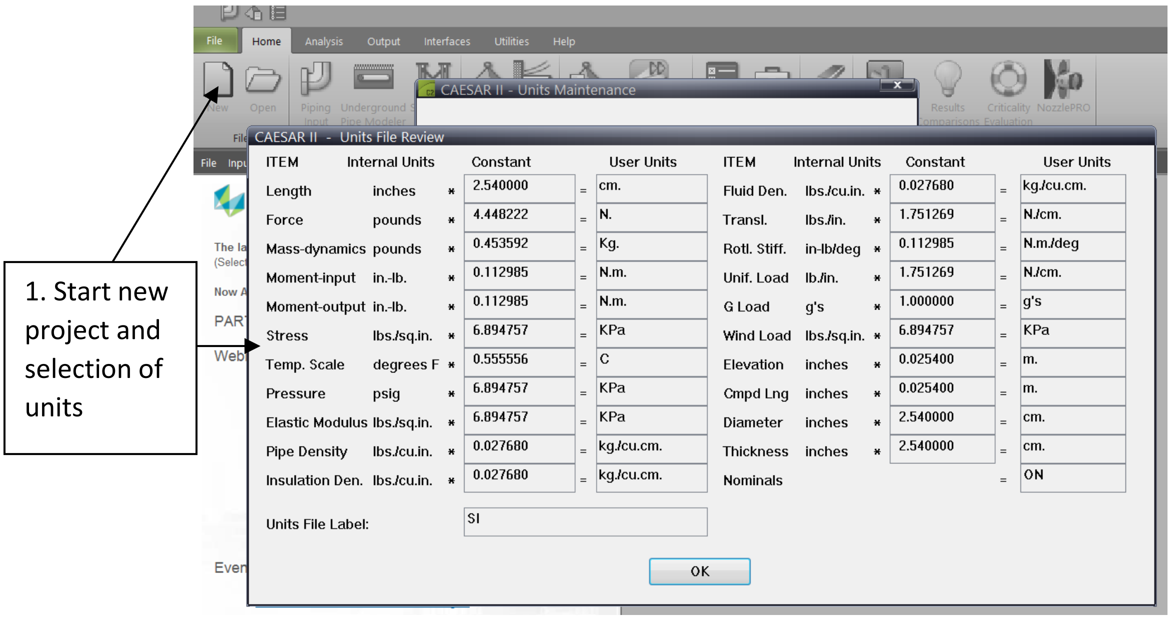


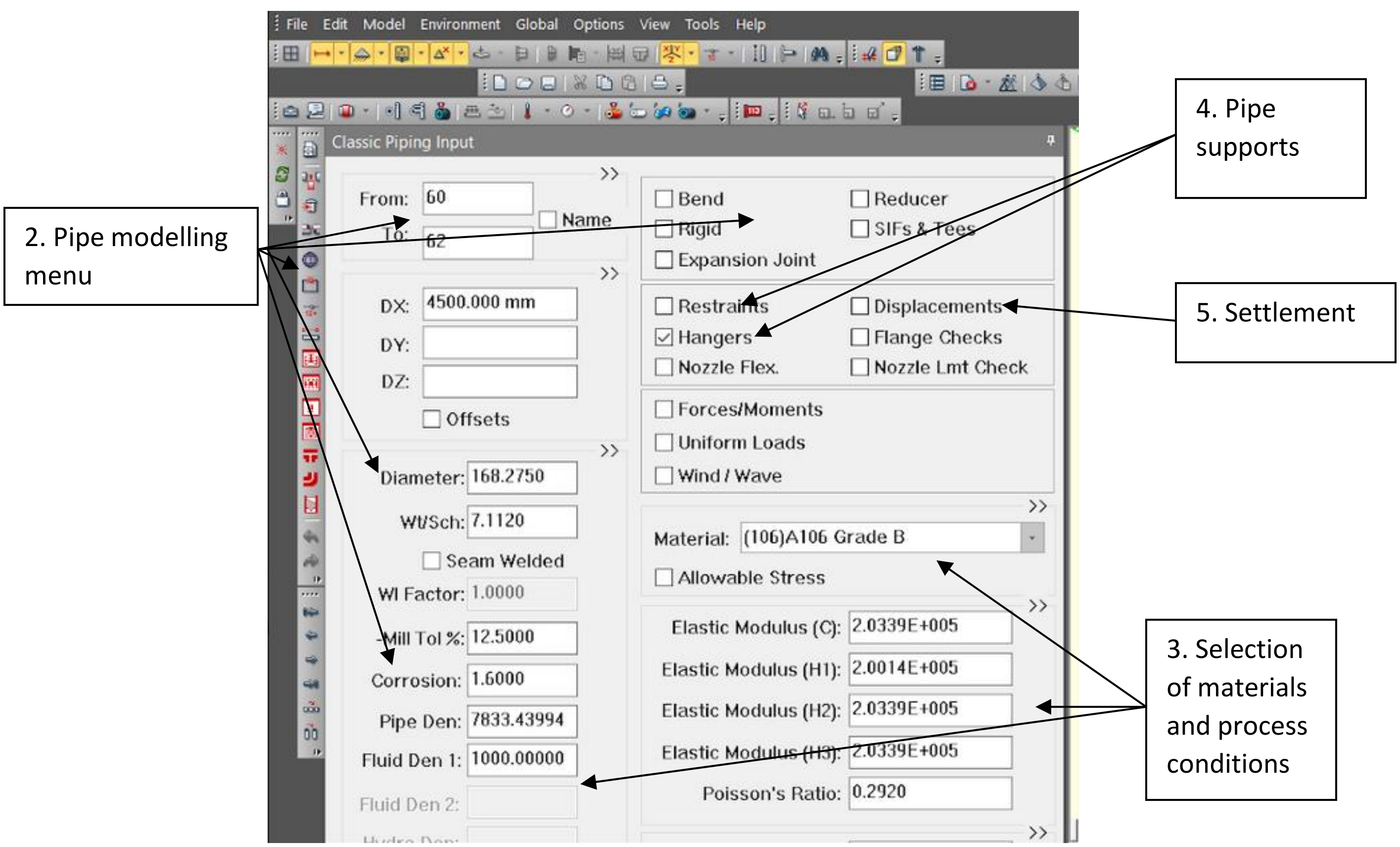
24" Sch-10 General safe spans



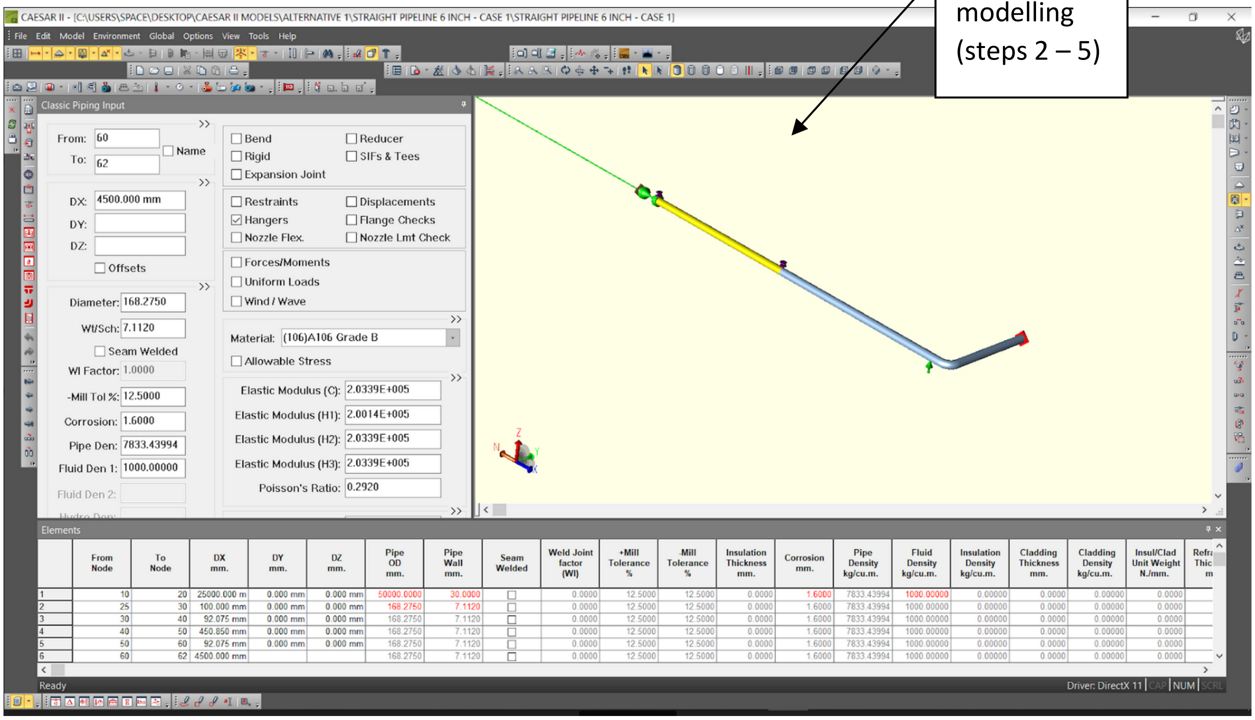


Appendix IV: Caesar II visual procedure

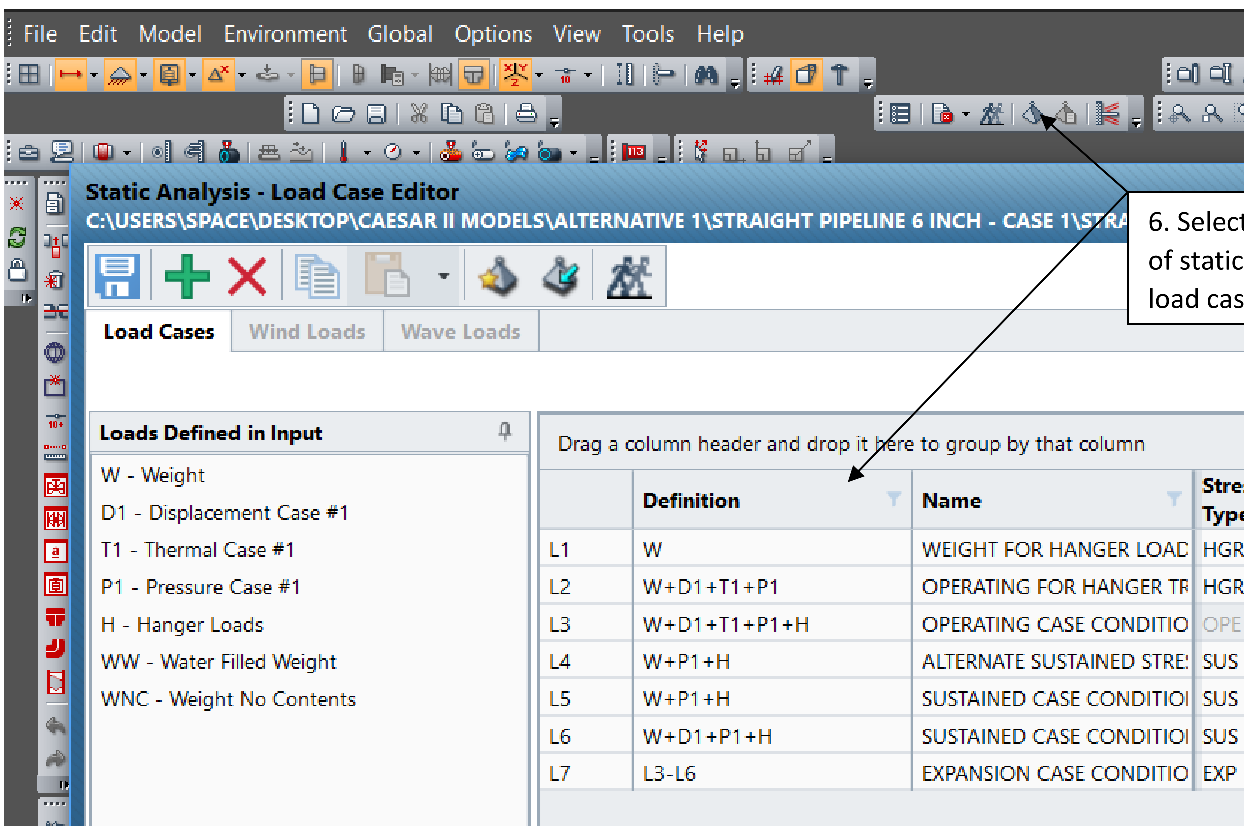




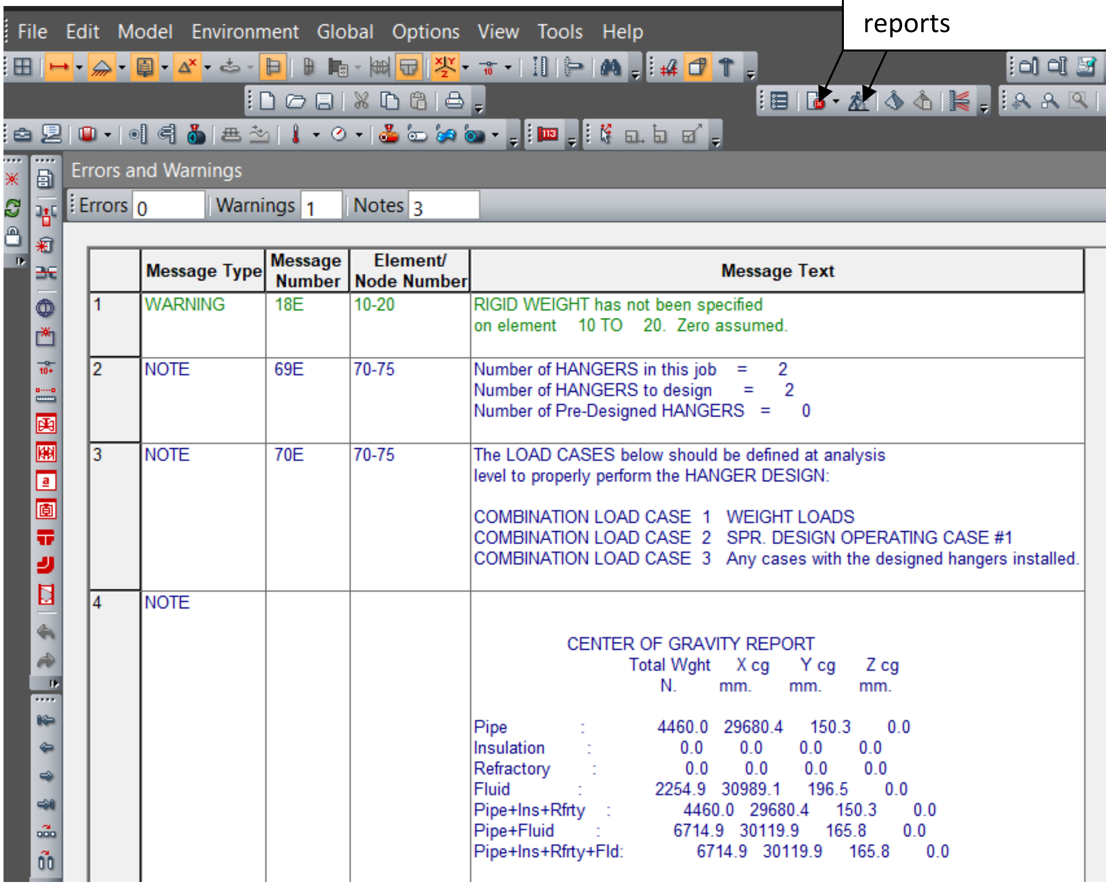
Overview of pipe modelling (steps 2 – 5)

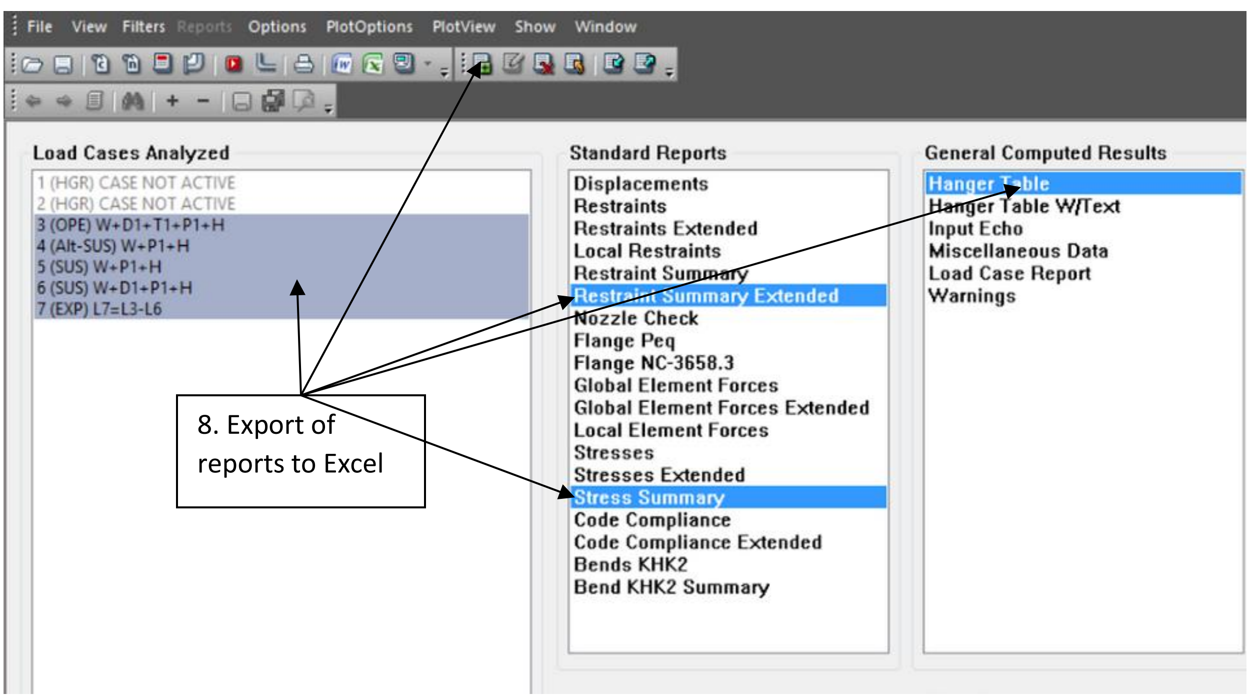


6. Selection of static load cases

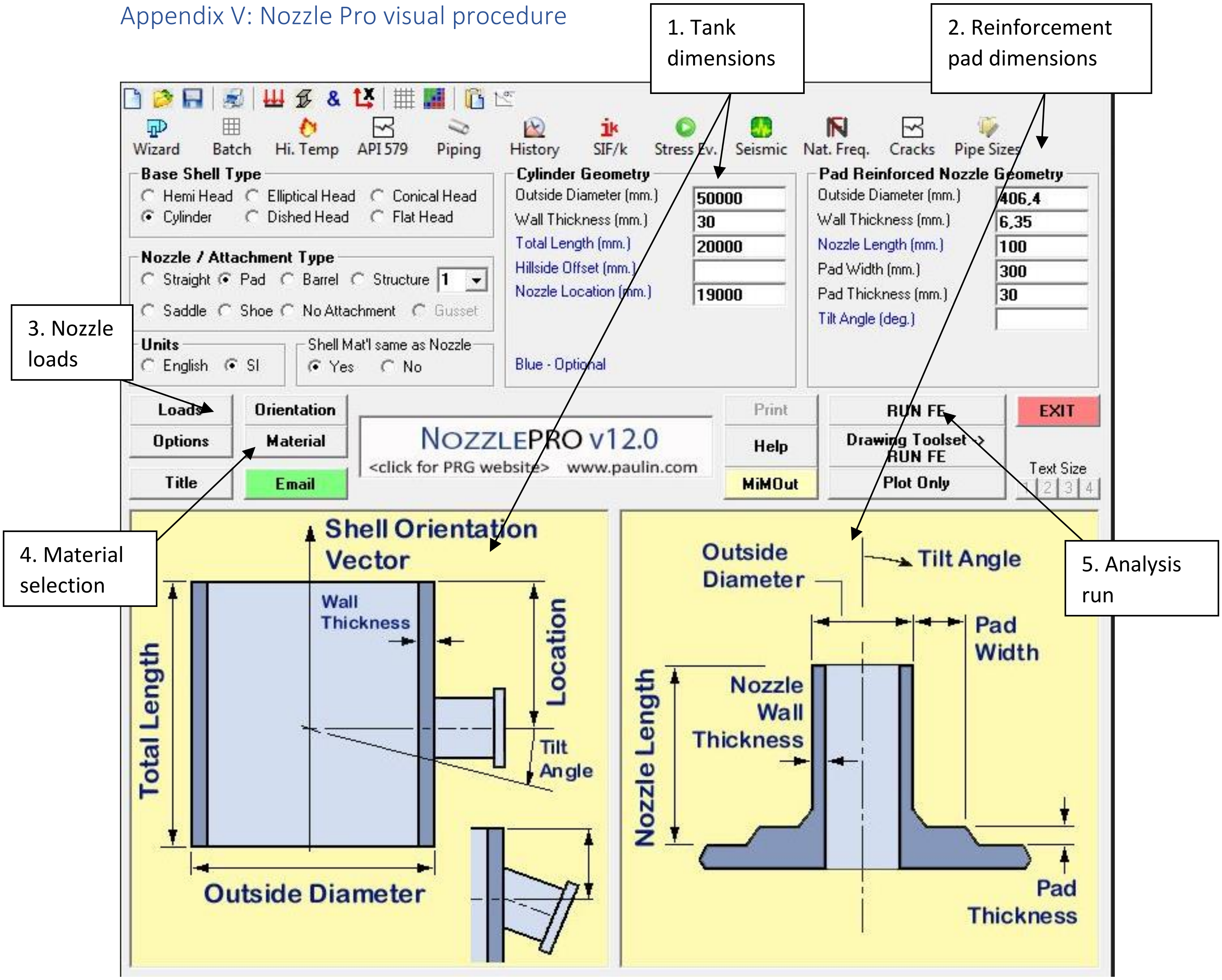


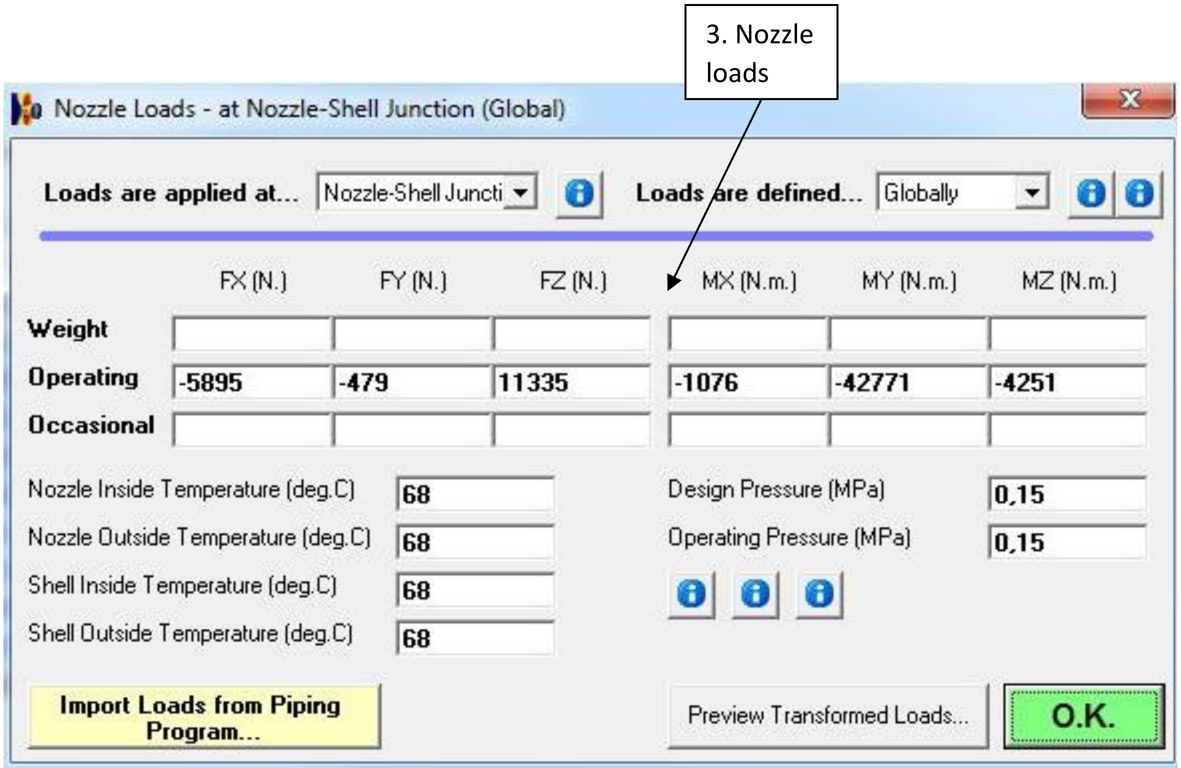
7. Error check, analysis run and generation of reports

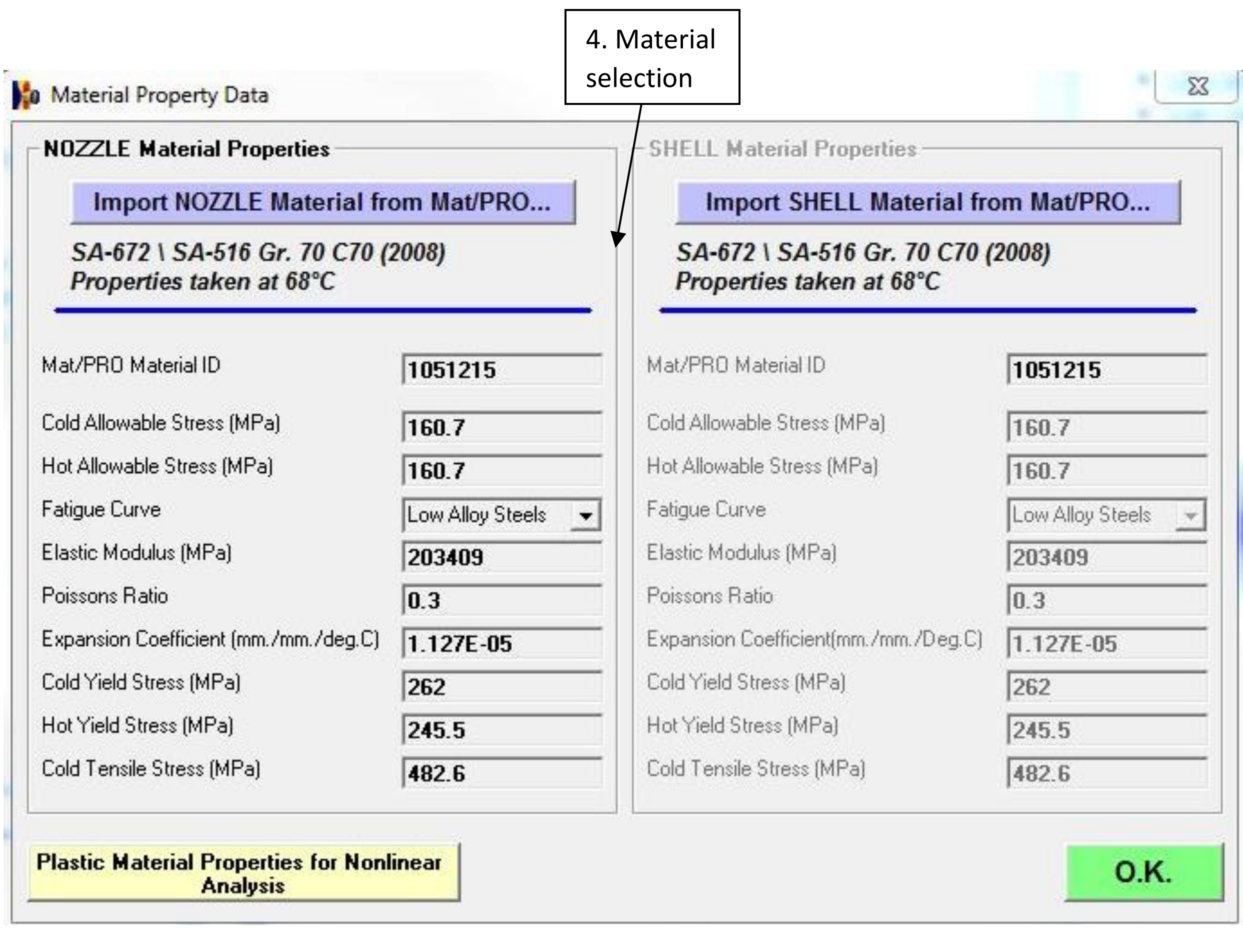




Appendix V: Nozzle Pro visual procedure




3. Nozzle loads
Nozzle Loads - at Nozzle-Shell Junction (Global)
Loads are applied at... Nozzle-Shell Juncti
Loads are defined... Globally
FX (N.) FY (N.) FZ (N.) MX (N.m.) MY (N.m.) MZ (N.m.)
Weight
Operating -5895 -479 11335 -1076 -42771 -4251
Occasional
Nozzle Inside Temperature (deg.C) 68
Nozzle Outside Temperature (deg.C) 68
Shell Inside Temperature (deg.C) 68
Shell Outside Temperature (deg.C) 68
Design Pressure (MPa) 0,15
Operating Pressure (MPa) 0,15
Import Loads from Piping Program...
Preview Transformed Loads...
O.K.


4. Material selection
Material Property Data
NOZZLE Material Properties
Import NOZZLE Material from Mat/PRO...
SA-672 \ SA-516 Gr. 70 C70 (2008)
Properties taken at 68°C
Mat/PRO Material ID 1051215
Cold Allowable Stress (MPa) 160.7
Hot Allowable Stress (MPa) 160.7
Fatigue Curve Low Alloy Steels
Elastic Modulus (MPa) 203409
Poissons Ratio 0.3
Expansion Coefficient (mm./mm./deg.C) 1.127E-05
Cold Yield Stress (MPa) 262
Hot Yield Stress (MPa) 245.5
Cold Tensile Stress (MPa) 482.6
SHELL Material Properties
Import SHELL Material from Mat/PRO...
SA-672 \ SA-516 Gr. 70 C70 (2008)
Properties taken at 68°C
Mat/PRO Material ID 1051215
Cold Allowable Stress (MPa) 160.7
Hot Allowable Stress (MPa) 160.7
Fatigue Curve Low Alloy Steels
Elastic Modulus (MPa) 203409
Poissons Ratio 0.3
Expansion Coefficient(mm./mm./Deg.C) 1.127E-05
Cold Yield Stress (MPa) 262
Hot Yield Stress (MPa) 245.5
Cold Tensile Stress (MPa) 482.6
Plastic Material Properties for Nonlinear Analysis
O.K.

Appendix VI: Caesar II and Nozzle Pro results

Note that the results of alternative 2: L-shape pipe is the same as Alternative 1: Straight pipe. The reasoning behind this is that alternative 1 checks the minimum length required for the **existing** compensating leg and alternative 2 checks the minimum length required for a **new** L-shape pipe design to be installed. This means that the installation required for alternative 1 is only the short section of the pipe (~2 m) but the minimum distance that must be available on site is equal to the length of the L-shape pipe. Therefore, for calculations Alternative 1 is equal to the L-shape pipe calculations.

Alternative 2: L-shape pipe

Node	Load Case	FX N.	FY N.	FZ N.	MX N.m.	MY N.m.	MZ N.m.	DX mm.	DY mm.	DZ mm.
LOAD CASE DEFINITION KEY										
CASE 3 (OPE) W+D1+T1+P1+H										
CASE 5 (SUS) W+P1+H										
CASE 6 (SUS) W+D1+P1+H										
CASE 7 (EXP) L7=L3-L6										

NPS 6" - Case 1									
	TYPE=Rigid ANC;								
3(OPE)	-18038	-1152	2424	-308	-13209	-4461	13.913	0	-99.999
5(SUS)	0	0	-539	-25	1228	0	0	0	0
6(SUS)	0	0	2424	-308	-13209	0	0	0	-99.999
7(EXP)	-18038	-1152	0	0	0	-4461	13.913	0	0
MAX	-18038/L	-1152/L3	2424/L3	-308/L3	-13209/L	-4461/L3	13.913/L	-0.000/L3	-99.999/L3

NPS 8" - Case 1									
	TYPE=Rigid ANC;								
3(OPE)	-20373	-1187	3559	-147	-17315	-5456	13.913	0	-99.999
5(SUS)	0	0	889	-34	1684	0	0	0	0
6(SUS)	0	0	3559	-147	-17315	0	0	0	-99.999
7(EXP)	-20373	-1187	0	0	0	-5456	13.913	0	0
MAX	-20373/L	-1187/L3	3559/L3	-147/L3	-17315/L	-5456/L3	13.913/L	-0.000/L3	-99.999/L3

NPS 10" - Case 1									
	TYPE=Rigid ANC;								
3(OPE)	-22800	-1317	5595	95	-23460	-7031	13.913	0	-99.999
5(SUS)	0	0	513	-40	4774	0	0	0	0
6(SUS)	0	0	5595	95	-23460	0	0	0	-99.999
7(EXP)	-22800	-1317	0	0	0	-7031	13.913	0	0
MAX	-22800/L	-1317/L3	5595/L3	95/L3	-23460/L	-7031/L3	13.913/L	-0.000/L3	-99.999/L3

Node	Load Case	FX N.	FY N.	FZ N.	MX N.m.	MY N.m.	MZ N.m.	DX mm.	DY mm.	DZ mm.
LOAD CASE DEFINITION KEY										
CASE 3 (OPE) W+D1+T1+P1+H										
CASE 5 (SUS) W+P1+H										
CASE 6 (SUS) W+D1+P1+H										
CASE 7 (EXP) L7=L3-L6										

NPS 12" - Case 1

	TYPE=Rigid ANC;								
3(OPE)	-23178	-1226	8885	327	-23785	-8074	13.913	0	-99.998
5(SUS)	0	0	2390	-14	9787	0	0	0	0
6(SUS)	0	0	8885	327	-23785	0	0	0	-99.998
7(EXP)	-23178	-1226	0	0	0	-8074	13.913	0	0
MAX	-23178/L	-1226/L3	8885/L3	327/L3	-23785/L	-8074/L3	13.913/L	-0.000/L3	-99.998/L3

NPS 14" - Case 1

	TYPE=Rigid ANC;								
3(OPE)	-22984	-1161	12360	12	-23273	-8768	13.913	0	-99.997
5(SUS)	0	0	4084	-111	14782	0	0	0	0.001
6(SUS)	0	0	12360	12	-23273	0	0	0	-99.997
7(EXP)	-22984	-1161	0	0	0	-8768	13.913	0	0
MAX	-22984/L	-1161/L3	12360/L3	-111/L5	-23273/L	-8768/L3	13.913/L	-0.000/L3	-99.997/L3

NPS 16" - Case 1

	TYPE=Rigid ANC;								
3(OPE)	-21144	-978	11494	54	-19681	-9347	13.913	0	-99.997
5(SUS)	0	0	2744	-138	23808	0	0	0	0
6(SUS)	0	0	11494	54	-19681	0	0	0	-99.997
7(EXP)	-21144	-978	0	0	0	-9347	13.913	0	0
MAX	-21144/L	-978/L3	11494/L3	-138/L5	23808/L5	-9347/L3	13.913/L	-0.000/L3	-99.997/L3

NPS 18" - Case 1

	TYPE=Rigid ANC;								
3(OPE)	-24749	-1177	13853	198	-25255	-11656	13.913	0	-99.997
5(SUS)	0	0	8663	-189	27219	0	0	0	0.002
6(SUS)	0	0	13853	198	-25255	0	0	0	-99.997
7(EXP)	-24749	-1177	0	0	0	-11656	13.913	0	0
MAX	-24749/L	-1177/L3	13853/L3	198/L3	27219/L5	-11656/L	13.913/L	-0.000/L3	-99.997/L3

Alternative 2: CASE 2

Node	Load Case	FX N.	FY N.	FZ N.	MX N.m.	MY N.m.	MZ N.m.	DX mm.	DY mm.	DZ mm.
LOAD CASE DEFINITION KEY										
CASE 3 (OPE) W+D1+T1+P1+H										
CASE 5 (SUS) W+P1+H										
CASE 6 (SUS) W+D1+P1+H										
CASE 7 (EXP) L7=L3-L6										

NPS 6" - Case 2									
	TYPE=Rigid ANC;								
3(OPE)	-8647	-566	2220	-705	-12481	-2054	13.913	0	-99.999
5(SUS)	0	0	-558	-46	1283	0	0	0	0
6(SUS)	-8647	-566	-558	-46	1283	-2054	13.913	0	0
7(EXP)	0	0	2779	-659	-13764	0	0	0	-99.999
MAX	-8647/L3	-566/L3	2779/L7	-705/L3	-13764/L	-2054/L3	13.913/L	-0.000/L3	-99.999/L3

NPS 8" - Case 2									
	TYPE=Rigid ANC;								
3(OPE)	-8390	-617	3291	-829	-16241	-2649	13.913	0	-99.999
5(SUS)	0	0	844	-90	1791	0	0	0	0
6(SUS)	-8390	-617	844	-90	1791	-2649	13.913	0	0
7(EXP)	0	0	2447	-739	-18032	0	0	0	-99.999
MAX	-8390/L3	-617/L3	3291/L3	-829/L3	-18032/L	-2649/L3	13.913/L	-0.000/L3	-99.999/L7

NPS 10" - Case 2									
	TYPE=Rigid ANC;								
3(OPE)	-9440	-741	5254	-1016	-21985	-3675	13.913	0	-99.999
5(SUS)	0	0	427	-165	4968	0	0	0	0
6(SUS)	-9440	-741	427	-165	4968	-3675	13.913	0	0
7(EXP)	0	0	4827	-851	-26953	0	0	0	-99.999
MAX	-9440/L3	-741/L3	5254/L3	-1016/L3	-26953/L	-3675/L3	13.913/L	-0.000/L3	-99.999/L

NPS 12" - Case 2									
	TYPE=Rigid ANC;								
3(OPE)	-6659	-568	8622	-1272	-22231	-3475	13.913	0	-99.998
5(SUS)	0	0	2316	-377	10151	0	0	0	0
6(SUS)	-6659	-568	2316	-377	10151	-3475	13.913	0	0
7(EXP)	0	0	6307	-895	-32382	0	0	0	-99.998
MAX	-6659/L3	-568/L3	8622/L3	-1272/L3	-32382/L	-3475/L3	13.913/L	-0.000/L3	-99.998/L7

Node	Load Case	FX N.	FY N.	FZ N.	MX N.m.	MY N.m.	MZ N.m.	DX mm.	DY mm.	DZ mm.
LOAD CASE DEFINITION KEY										
CASE 3 (OPE) W+D1+T1+P1+H										
CASE 5 (SUS) W+P1+H										
CASE 6 (SUS) W+D1+P1+H										
CASE 7 (EXP) L7=L3-L6										

NPS 14" - Case 2

	TYPE=Rigid ANC;								
3(OPE)	-5754	-530	11936	-1254	-21343	-3688	13.913	0	-99.997
5(SUS)	0	0	3840	-540	15410	0	0	0	0.001
6(SUS)	-5754	-530	3840	-540	15410	-3688	13.913	0	0.001
7(EXP)	0	0	8097	-714	-36753	0	0	0	-99.998
MAX	-5754/L3	-530/L3	11936/L:	-1254/L3	-36753/L	-3688/L3	13.913/L	-0.000/L:	-99.998/L7

NPS 16" - Case 2

	TYPE=Rigid ANC;								
3(OPE)	-5895	-479	11335	-1076	-17930	-4251	13.913	0	-99.997
5(SUS)	0	0	2657	-669	24841	0	0	0	0
6(SUS)	-5895	-479	2657	-669	24841	-4251	13.913	0	0
7(EXP)	0	0	8677	-407	-42771	0	0	0	-99.998
MAX	-5895/L3	-479/L3	11335/L:	-1076/L3	-42771/L	-4251/L3	13.913/L	-0.000/L:	-99.998/L7

NPS 18" - Case 2

	TYPE=Rigid ANC;								
3(OPE)	-6511	-574	13641	-1412	-22902	-5235	13.913	0	-99.997
5(SUS)	0	0	8560	-889	28480	0	0	0	0.002
6(SUS)	-6511	-574	8560	-889	28480	-5235	13.913	0	0.002
7(EXP)	0	0	5081	-523	-51382	0	0	0	-99.999
MAX	-6511/L3	-574/L3	13641/L:	-1412/L3	-51382/L	-5235/L3	13.913/L	-0.000/L:	-99.999/L7

Alternative 3: Expansion joint

Node	Load Case	FX N.	FY N.	FZ N.	MX N.m.	MY N.m.	MZ N.m.	DX mm.	DY mm.	DZ mm.
LOAD CASE DEFINITION KEY										
CASE 3 (OPE) W+D1+T1+P1+H										
CASE 5 (SUS) W+P1+H										
CASE 6 (SUS) W+D1+P1+H										
CASE 7 (EXP) L7=L3-L6										

NPS 6" - Expansion joints

	TYPE=Prog Design	CSH;							
3(OPE)	0	0	-2021	0	0	0	11.538	0	-199.864
5(SUS)	0	0	-2021	0	0	0	0	0	0
6(EXP)	0	0	0	0	0	0	11.538	0	-199.864
MAX			-2021/L3				11.538/L3		-199.864/L

NPS 8" - Expansion joints

	TYPE=Prog Design	CSH;							
3(OPE)	0	0	-2655	0	0	0	11.538	0	-199.921
5(SUS)	0	0	-2655	0	0	0	0	0	0
6(EXP)	0	0	0	0	0	0	11.538	0	-199.921
MAX			-2655/L3				11.538/L3		-199.921/L

NPS 10" - Expansion joints

	TYPE=Prog Design	CSH;							
3(OPE)	0	0	-3452	0	0	0	11.538	0	-199.901
5(SUS)	0	0	-3452	0	0	0	0	0	0
6(EXP)	0	0	0	0	0	0	11.538	0	-199.901
MAX			-3452/L3				11.538/L3		-199.901/L

NPS 12" - Expansion joints

	TYPE=Prog Design	CSH;							
3(OPE)	0	0	-7044	0	0	0	11.538	0	-199.932
5(SUS)	0	0	-7044	0	0	0	0	0	0
6(EXP)	0	0	0	0	0	0	11.538	0	-199.932
MAX			-7044/L3				11.538/L3		-199.932/L

NPS 14" - Expansion joints

	TYPE=Prog Design	CSH;							
3(OPE)	0	0	-8113	0	0	0	11.539	0	-199.921
5(SUS)	0	0	-8113	0	0	0	0	0	0
6(EXP)	0	0	0	0	0	0	11.539	0	-199.921
MAX			-8113/L3				11.539/L3		-199.921/L

Node	Load Case	FX N.	FY N.	FZ N.	MX N.m.	MY N.m.	MZ N.m.	DX mm.	DY mm.	DZ mm.
LOAD CASE DEFINITION KEY										
CASE 3 (OPE) W+D1+T1+P1+H										
CASE 5 (SUS) W+P1+H										
CASE 6 (SUS) W+D1+P1+H										
CASE 7 (EXP) L7=L3-L6										

NPS 16" - Expansion joints

	TYPE=Prog Design CSH;								
3(OPE)	0	0	-9979	0	0	0	11.539	0	-199.94
5(SUS)	0	0	-9979	0	0	0	0	0	0
6(EXP)	0	0	0	0	0	0	11.539	0	-199.94
MAX			-9979/L3				11.539/L3		-199.940/L

NPS 18" - Expansion joints

	TYPE=Prog Design CSH;								
3(OPE)	0	0	-12029	0	0	0	11.539	0	-199.951
5(SUS)	0	0	-12029	0	0	0	0	0	0
6(EXP)	0	0	0	0	0	0	11.539	0	-199.951
MAX			-12029/L3				11.539/L3		-199.951/L

NPS 20" - Expansion joints

	TYPE=Prog Design CSH;								
3(OPE)	0	0	-18390	0	0	0	11.539	0	-199.958
5(SUS)	0	0	-18390	0	0	0	0	0	0
6(EXP)	0	0	0	0	0	0	11.539	0	-199.958
MAX			-18390/L3				11.539/L3		-199.958/L

NPS 24" - Expansion joints

	TYPE=Prog Design CSH;								
3(OPE)	0	0	-29736	0	0	0	11.539	0	-199.971
5(SUS)	0	0	-29736	0	0	0	0	0	0
6(EXP)	0	0	0	0	0	0	11.539	0	-199.971
MAX			-29736/L3				11.539/L3		-199.971/L

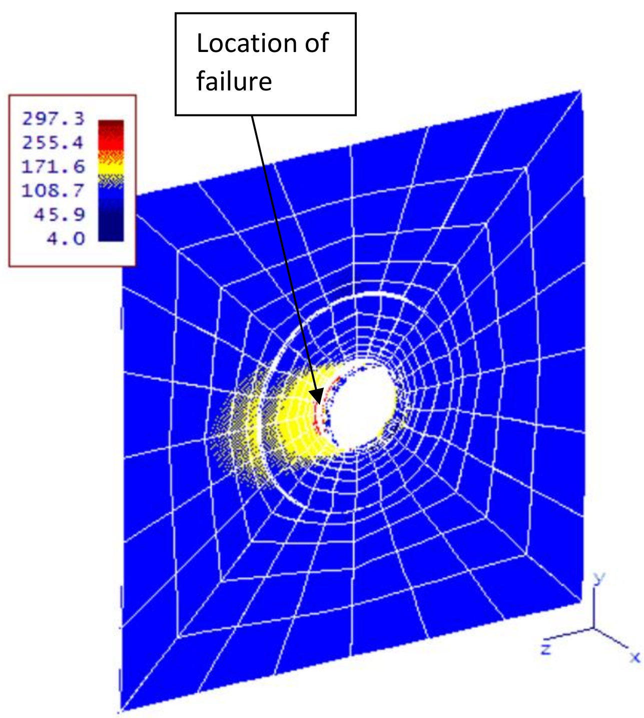
Alternative 2: Nozzle pro results

NPS 14" – Pipe loop

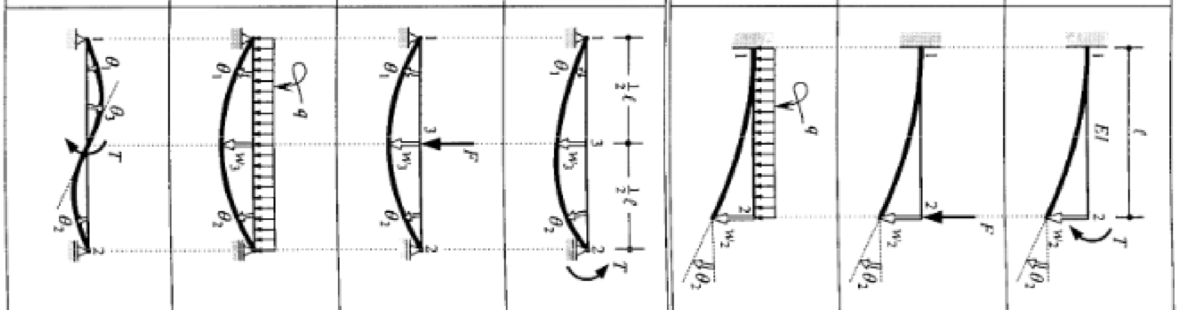
```
ASME Overstressed Areas                                   $X


          *** NO OVERSTRESSED NODES IN THIS MODEL ***
```

NPS 16" – Pipe loop

```
ASME Overstressed Areas                                   $X


Branch at Junction

Pl+Pb+Q+F    Damage Ratio    Primary+Secondary+Peak (Inner) Load Case 2
      297     1.077 Life     Stress Concentration Factor = 1.350
      MPa     1.025 Stress   Strain Concentration Factor = 1.000
                             Cycles Allowed for this Stress = 6,500.
Allowable                    "B31" Fatigue Stress Allowable = 401.8
    289.9                    Markl Fatigue Stress Allowable = 287.5
      MPa                    WRC 474 Mean Cycles to Failure = 98,205.
                             WRC 474 99% Probability Cycles = 22,814.
     102%                    WRC 474 95% Probability Cycles = 31,674.
                             BS5500 Allowed Cycles(Curve F) = 8,084.
                             Membrane-to-Bending Ratio = 0.253
                             Bending-to-PL+PB+Q Ratio = 0.798
                             Plot Reference:
                             8) Pl+Pb+Q+F < Sa (EXP,Inside) Case 2
```



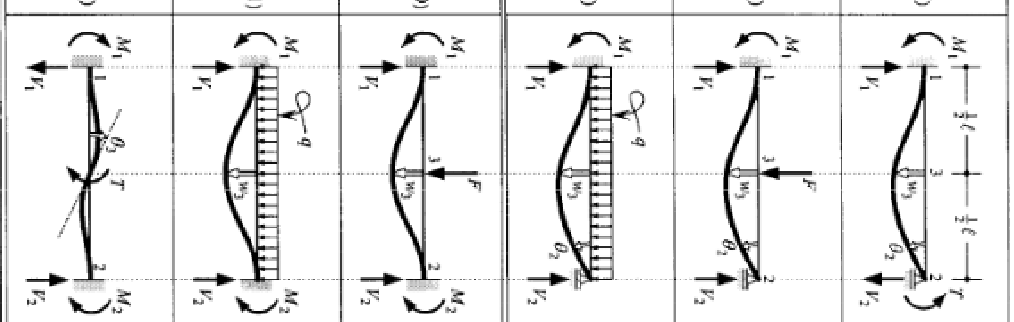
Appendix VII: Equations and FBD of cantilever beams



Case	Formulas
(1)	$\theta_2 = \frac{T\ell}{EI}$; $w_2 = \frac{T\ell^2}{2EI}$
(2)	$\theta_2 = \frac{F\ell^2}{2EI}$; $w_2 = \frac{F\ell^3}{3EI}$
(3)	$\theta_2 = \frac{q\ell^3}{6EI}$; $w_2 = \frac{q\ell^4}{8EI}$
(4)	$\theta_1 = \frac{1}{6}\frac{T\ell}{EI}$; $\theta_2 = \frac{1}{3}\frac{T\ell}{EI}$; $w_3 = \frac{1}{16}\frac{T\ell^2}{EI}$
(5)	$\theta_1 = \theta_2 = \frac{1}{16}\frac{F\ell^2}{EI}$; $w_3 = \frac{1}{48}\frac{F\ell^3}{EI}$
(6)	$\theta_1 = \theta_2 = \frac{1}{24}\frac{q\ell^3}{EI}$; $w_3 = \frac{5}{384}\frac{q\ell^4}{EI}$
(a)	$\theta_1 = \theta_2 = \frac{1}{24}\frac{T\ell}{EI}$; $\theta_3 = \frac{1}{12}\frac{T\ell}{EI}$; $w_3 = 0$

vergeet-mij-nietjes (1)–(3); *vrij opgelegde ligger (statisch bepaald)* (4)–(a)

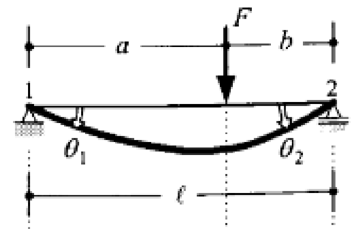
statisch onbepaalde ligger (enkelzijdig ingeklemd) (7)–(9); *statisch onbepaalde ligger (tweezijdig ingeklemd)* (10)–(b)



Case	Formulas
(7)	$\theta_2 = \frac{1}{4}\frac{T\ell}{EI}$; $w_3 = \frac{1}{32}\frac{T\ell^2}{EI}$; $M_1 = \frac{1}{2}T$; $V_1 = V_2 = \frac{3}{2}\frac{T}{\ell}$
(8)	$\theta_2 = \frac{1}{32}\frac{F\ell^2}{EI}$; $w_3 = \frac{7}{768}\frac{F\ell^3}{EI}$; $M_1 = \frac{3}{16}F\ell$; $V_1 = \frac{11}{16}F$; $V_2 = \frac{5}{16}F$
(9)	$\theta_2 = \frac{1}{48}\frac{q\ell^3}{EI}$; $w_3 = \frac{1}{192}\frac{q\ell^4}{EI}$; $M_1 = \frac{1}{8}q\ell^2$; $V_1 = \frac{5}{8}q\ell$; $V_2 = \frac{3}{8}q\ell$
(10)	$w_3 = \frac{1}{192}\frac{F\ell^3}{EI}$; $M_1 = M_2 = \frac{1}{8}F\ell$; $V_1 = V_2 = \frac{1}{2}F$
(11)	$w_3 = \frac{1}{384}\frac{q\ell^4}{EI}$; $M_1 = M_2 = \frac{1}{12}q\ell^2$; $V_1 = V_2 = \frac{1}{2}q\ell$
(b)	$\theta_3 = \frac{1}{16}\frac{T\ell}{EI}$; $w_3 = 0$; $M_1 = M_2 = \frac{1}{4}T$; $V_1 = V_2 = \frac{3}{2}\frac{T}{\ell}$

Enkele formules voor prismatische liggers met buigstijfheid EI.
T, F en q zijn belastingen door resp. een koppel, kracht en gelijkmatig verdeelde belasting.
M_i en V_i zijn het buigend moment en de dwarskracht op eindoorsnede i van de ligger ten gevolge van de oplegreacties.

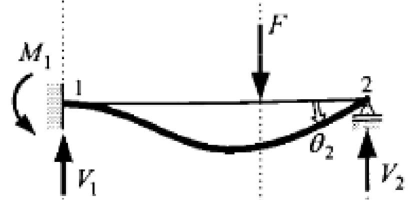
(c)



$$\theta_1 = \frac{Fab(\ell + b)}{6EI\ell} = \frac{F\ell^2}{6EI}\left(2\frac{a}{\ell} - 3\frac{a^2}{\ell^2} + \frac{a^3}{\ell^3}\right)$$

$$\theta_2 = \frac{Fab(\ell + a)}{6EI\ell} = \frac{F\ell^2}{6EI}\left(\frac{a}{\ell} - \frac{a^3}{\ell^3}\right)$$

(d)



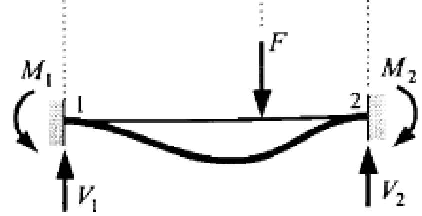
$$M_1 = \frac{Fb(\ell^2 - b^2)}{2\ell^2} = F\ell\left(\frac{a}{\ell} - \frac{3}{2}\frac{a^2}{\ell^2} + \frac{1}{2}\frac{a^3}{\ell^3}\right)$$

$$V_1 = \frac{Fb(3\ell^2 - b^2)}{2\ell^3} = F\left(1 - \frac{3}{2}\frac{a^2}{\ell^2} + \frac{1}{2}\frac{a^3}{\ell^3}\right)$$

$$V_2 = \frac{Fa^2(3\ell - a)}{2\ell^3} = F\left(\frac{3}{2}\frac{a^2}{\ell^2} - \frac{1}{2}\frac{a^3}{\ell^3}\right)$$

$$\theta_2 = \frac{Fa^2b}{4EI\ell} = \frac{F\ell^2}{4EI}\left(\frac{a^2}{\ell^2} - \frac{a^3}{\ell^3}\right)$$

(e)



$$M_1 = \frac{Fab^2}{\ell^2} = F\ell\left(\frac{a}{\ell} - 2\frac{a^2}{\ell^2} + \frac{a^3}{\ell^3}\right)$$

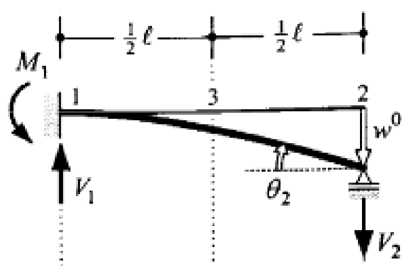
$$V_1 = \frac{Fb^2(\ell + 2a)}{\ell^3} = F\left(1 - 3\frac{a^2}{\ell^2} + 2\frac{a^3}{\ell^3}\right)$$

$$M_2 = \frac{Fa^2b}{\ell^2} = F\ell\left(\frac{a^2}{\ell^2} - \frac{a^3}{\ell^3}\right)$$

$$V_2 = \frac{Fa^2(\ell + 2b)}{\ell^3} = F\ell\left(3\frac{a^2}{\ell^2} - 2\frac{a^3}{\ell^3}\right)$$

drie bij-de handjes

(f)

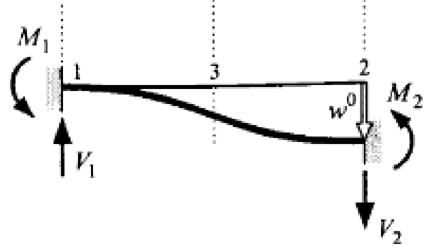


$$M_1 = \frac{3EI}{\ell^2}w^0; \quad V_1 = V_2 = \frac{3EI}{\ell^3}w^0$$

$$\theta_2 = \frac{3}{2}\frac{w^0}{\ell}$$

$$\theta_3 = \frac{9}{8}\frac{w^0}{\ell}; \quad w_3 = \frac{5}{16}w^0$$

(g)



$$M_1 = M_2 = \frac{6EI}{\ell^2}w^0; \quad V_1 = V_2 = \frac{12EI}{\ell^3}w^0$$

$$\theta_3 = \frac{3}{2}\frac{w^0}{\ell}; \quad w_3 = \frac{1}{2}w^0$$

zettingen

Appendix VIII: Quantity of components for each alternative

Option 1: Straight pipe

Component type		Qty.	Nominal pipe size [inch]					
			6”	8”	10”	12”	14”	16”
No.	Flange	[pcs]	1	1	1	1	1	1
1	Gasket	[pcs]	1	1	1	1	1	1
2	Bolt + 2 Nuts	[pcs]	8	8	12	12	12	16
3	Pipe	[m]	2	2	2	2	2	2

Option 2: Increased flexibility with extra Piping (Case 1)

Component type		Qty.	Nominal pipe size [inch]					
			6”	8”	10”	12”	14”	16”
No.	Flange	[pcs]	1	1	1	1	1	1
1	Gasket	[pcs]	1	1	1	1	1	1
2	Bolt + 2 Nuts	[pcs]	8	8	12	12	12	16
3	Pipe	[m]	11,2	13,7	16,3	20,5	23,5	30,3
4	Elbow 90^0	[pcs]	1	1	1	1	1	1

Option 2: Increased flexibility with extra Piping (Case 2)

Component type		Qty.	Nominal pipe size [inch]					
			6”	8”	10”	12”	14”	16”
No.	Flange	[pcs]	1	1	1	1	1	NA
1	Gasket	[pcs]	1	1	1	1	1	NA
2	Bolt + 2 Nuts	[pcs]	8	8	12	12	12	NA
3	Pipe	[m]	12,8	15,4	18,0	23,4	27,8	NA
4	Elbow 90^0	[pcs]	4	4	4	4	4	NA

Option 3: Metallic expansion joint

Component type		Qty.	Nominal pipe size [inch]								
			6”	8”	10”	12”	14”	16”	18”	20”	24”
No.	Flange	[pcs]	5	5	5	5	5	5	5	5	5
1	Gasket	[pcs]	3	3	3	3	3	3	3	3	3
2	Bolt + 2 Nuts	[pcs]	8	8	12	12	12	16	16	20	20
3	Pipe	[m]	1	1	1	1	1	1	1	1	1
4	Expansion joint	[mm]	965*	1150*	1170*	1320*	1360*	1500*	1560*	1770*	1540**

* Type LRR 16 (Witzemann, 1990)
** Type LRK 16 (Witzemann, 1990)

Appendix IX: Scales of risk significance and probability and detection for FMEA-analysis

Rating	Significance	Description
1	Very insignificant	Failures and defects do not constitute a problem
2		
3	Insignificant	Defects can be repaired and easily removed
4		
5	Significant	Failures can cause loss of structural safety
6		
7	Critical	Defects can cause ruptures and accidents on pipelines
8		
9	Catastrophic	Failures can cause hazard to life and health
10		

Rating	Probability	Description
1	Very low	Risk of defect very unlikely, almost zero
2		
3	Low	Very insignificant probability
4		
5	Medium	Medium probability of defect
6		
7	High	Past experiences forsee high risk of failures
8		
9	Very high	Defects are inevitable
10		

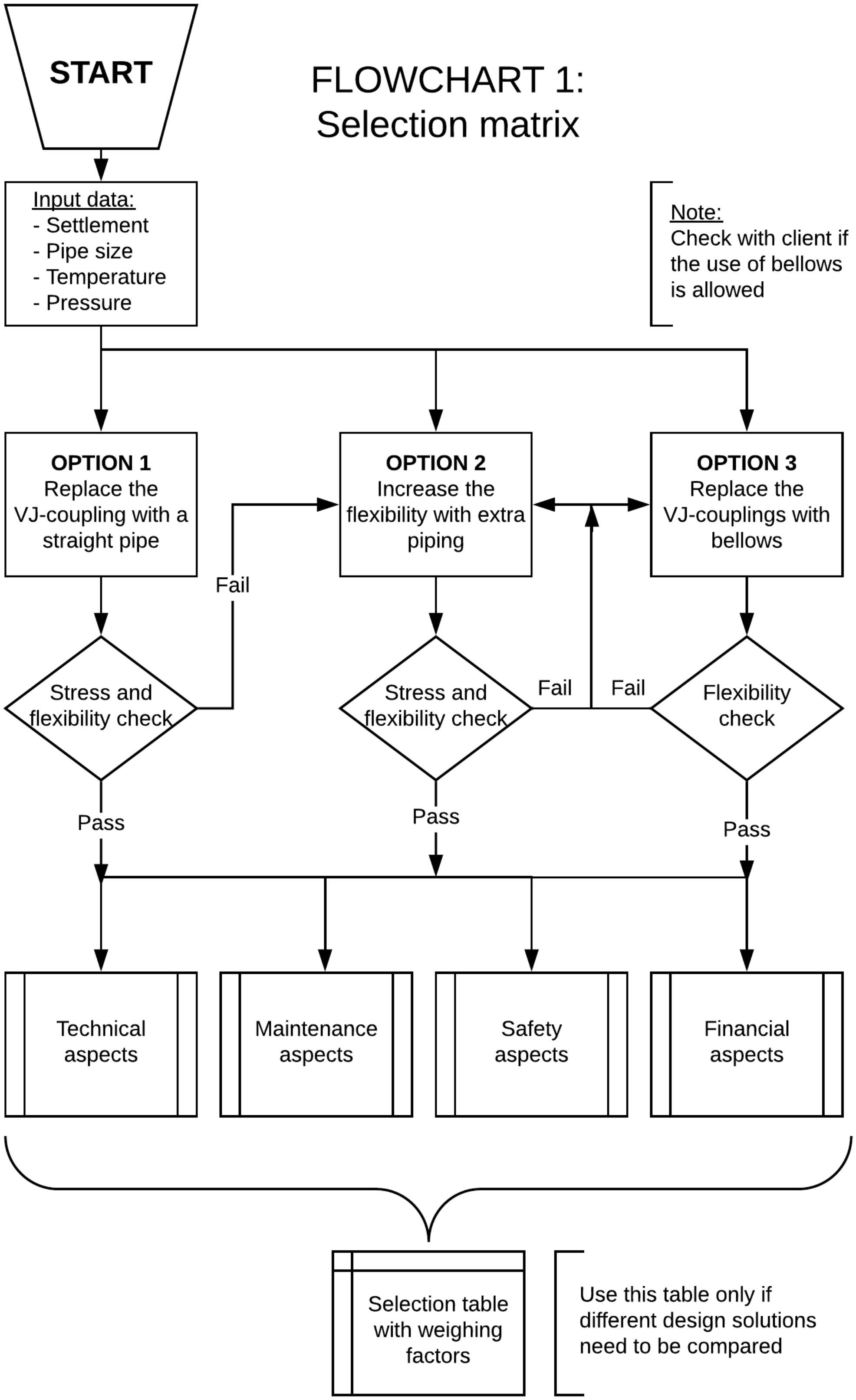
Rating	Probability	Description
1	Guaranteed	Failures are always detected
2		
3	High	Failures are easy to detect
4	Moderate	Unlikely to detect inconsistencies
5		
6	Medium	Failures are difficult to detect during control and test
7		
8	Low	Very difficult, technological checks ineffective
9		
10	Very low	Impossible to detect

Appendix X: RPN calculation for FMEA-analysis

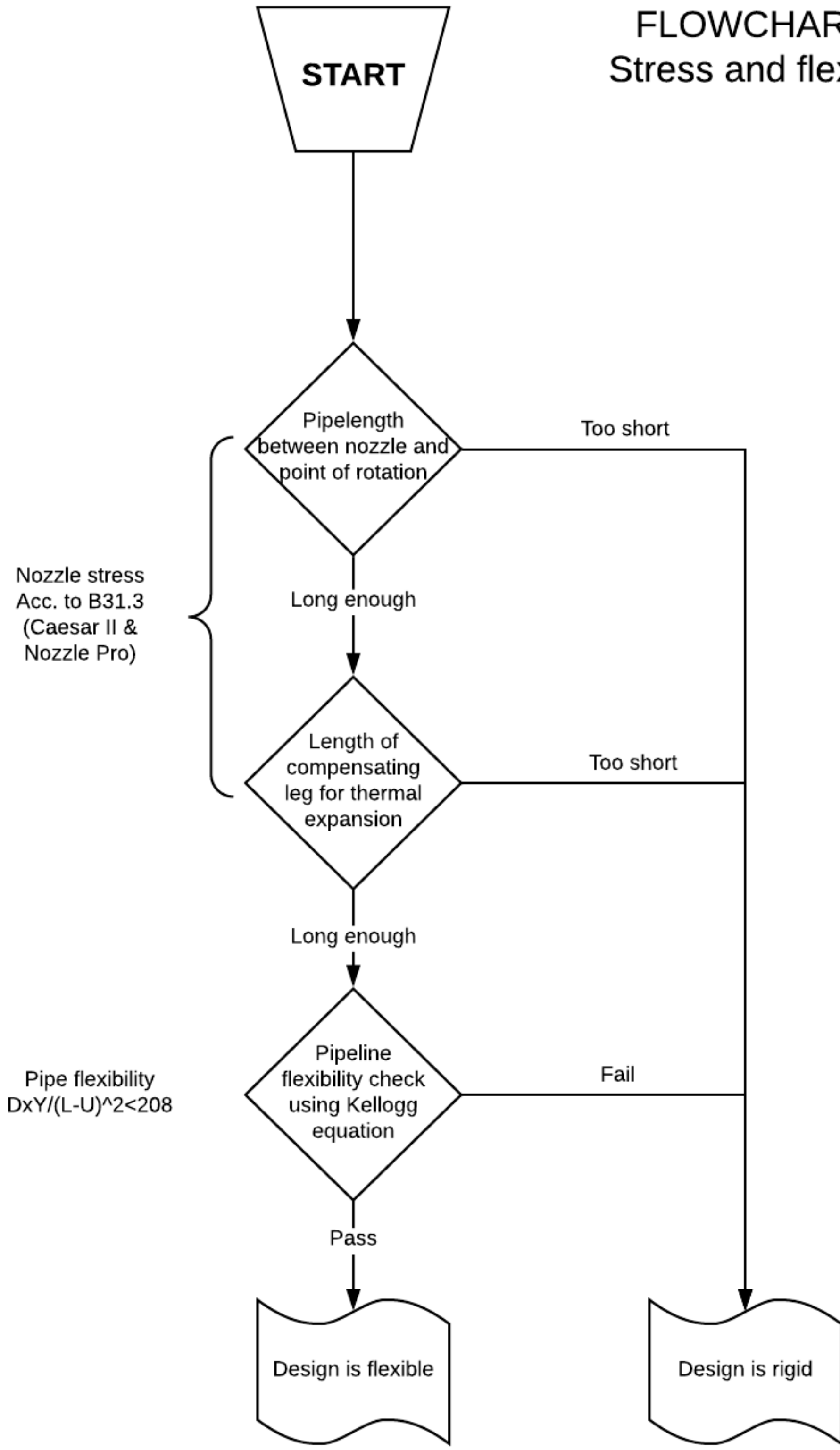
Risk priority number (PNR) - Option 1 & Option 2								
No.	Types of failures	Consequences	S	Risk probability	O	Detection probability	D	RPN = S*O*D
1	Bolts of flanges are improperly tightened	High operational pressures can lead to product leakage.	5	Increased probability in the case of new or inexperienced personnel.	4	A leakage can be easily detected at startup when a check is performed	3	60
2	Welding impurities	High operational pressures and bending of the pipeline can lead to rupture of the weld.	8	Very low probability because the NDT-(X-ray) test is performed by qualified personnel.	2	The NDT-(X-ray) test is meant to detect the quality of the weld and thus, detect any imperfection.	3	48
3	Internal pipe corrosion (erosion)	Erosion causes thinning of the pipe thickness and can lead to weakening of pipeline strength.	7	Dirt inside of the pipeline can increase the chances of erosion if maintenance frequency is low	4	The sensors of a pigging module measures the inside diameter of the pipeline during maintenance.	3	84
4	Excessive dirt in the spring housing of support	Dirt can cause high friction of the springs and can cause a jam which can lead to high nozzle stresses.	6	Infrequent checks and improper maintenance can increase the risk of dirt accumulating.	5	Dirt inside of the spring housing can only be detected during a check or during maintenance.	3	90
5	Pitting corrosion of the supports	Corrosion of the supporting can lead to loss of support strength and increase pipe stresses.	6	Corrosion takes time to set in therefore the risk of corrosion remains low.	4	Corrosion is easily detected with the naked eye during checks.	2	48

6	Excessive thermal expansion in the summer	Excessive thermal expansion can cause extra stresses on the tank nozzle which can be catastrophic.	9	The design temperatures of the pipeline accounts for the temperature increase during hot summer days.	1	Thermal expansion is difficult to detect with the naked eye since the expansion is relatively small.	7	63
7	Excessive cold shrinkage during the winter	Excessive cold shrinkage can cause extra stresses on the tank nozzle which can be catastrophic.	9	The outside temperatures would have to decrease to < -48 deg C to have any significant effect.	1	Cold shrinkage is difficult to detect with the naked eye.	7	63
8	Hydraulic shock	Hydraulic shock can lead to rupture of piping components such as elbows and flange connections.	10	Hydraulic shock occurs when the fluid inside of the pipe undergoes an abrupt change in momentum.	3	Very difficult to detect. Operators need to be very careful when opening or closing valves and handling pumps.	4	120
9	Excessive torsion of the bellows	Asymmetric tank settlement can cause excessive torsion of the bellows which can create high material shear stresses.	9	Settlement of tanks occur slowly over time and is constant monitored by engineers, therefore the chances of occurring are very low.	4	Tanks are inspected periodically. Settlements of the tank are monitored and in case of excessive or improper settlement, corrective measures are taken.	2	

Appendix XI: Flowcharts for selection matrix model (Excel)



FLOWCHART 2: Stress and flexibility



Appendix XII: Competence assessment “*Werkend Leren*” (2018)

DE HAAGSE HOGESCHOOL

Adviesbeoordelingsformulier

Haagse Hogeschool
Faculteit voor Technologie Innovatie en Samenleving, Delft
Opleiding werktuigbouwkunde duaal
Versie 2017-2018

Bedrijf:	Vicoma Engineering
Bedrijfsbegeleider:	Theo Gunneweg
Schoolbegeleider:	Dennis Mulder
Student:	Carlos Spagnol
Studentnummer:	13114301
Semester:	WDH3

Te verwerven competenties	Niveau			Beoordeling ****				
	Aanvang*	Doelstelling **	Gerealiseerd***	Onvoldoende	Matig	Voldoende	Goed	Uitmuntend
1. Analyseren	2	3	3				X	
2. Ontwerpen	2	3	3			X		
3. Realiseren	2	-	-					
4. Beheren	1	2	2			X		
5. Managen	0	1	1			X		
6. Adviseren	2	-	-					
7. Onderzoeken	2	-	-					
8. Professionaliseren	2	3	3			X		

Opmerkingen :
Ontwikkeling is goed verlopen, waak ervoor dat je niet lui wordt en denkt dat alles komt aanwaaien. In de toekomst kunnen er moeilijkere projecten voorbij komen.

Handtekening :
Bedrijfsbegeleider
T. Gunneweg

Handtekening :
Student
C. Spagnol

Datum : 25-01-2018

Datum : 25-01-2018

Appendix XIII: Bedrijfsbeoordeling uitvoering afstudeeropdracht (2018)

DE HAAGSE HOGESCHOOL

Bedrijfsbeoordeling uitvoering afstudeeropdracht

De Haagse Hogeschool, Faculteit voor Technology, Innovation & Society Delft

Naam student	Carlos Spagnol
Studentnummer	13114301
Opleiding student	Werktuigbouwkunde
Afstudeer periode	Shift 1 2018-2019
Naam bedrijf	Vicoma Engineering
Naam bedrijfsbegeleider	Theo Gunneweg

Wat is uw oordeel over de prestatie van de student tijdens de afstudeerperiode, onderscheiden naar de volgende punten:

O = onvoldoende; T = twijfelachtig; V = voldoende; G = goed; U = uitstekend

	O	T	V	G	U
Inzicht			X		
Systematische aanpak			X		
Kwaliteit gegevensverzameling				X	
Kwaliteit (tussen)rapportage				X	
Theoretische kennis				X	
Praktische inzicht			X		
Nauwkeurigheid/zorgvuldigheid				X	
Houding en optreden				X	
Zelfstandigheid				X	
Werktempo			X		
Totale beoordeling				X	

Paraaf bedrijfsbegeleider ....

Paraaf student ....

Z.O.Z.

DE HAAGSE HOGESCHOOL

Verdere op- en aanmerkingen over de student en/of de afstudeeropdracht.

Carlos heeft tijdens het duaal traject grote ontwikkelstappen gemaakt zowel op persoonlijk als professioneel vlak. Dit is ook tot uiting gekomen tijdens de afstudeerperiode.
De afstudeerperiode is zwaar geweest, omdat er ook nog deeltijd gewerkt moest worden. Carlos heeft zich goed staande weten te houden tijdens deze drukke periode.
Aan het einde van zijn duaal traject willen we Carlos graag in dienst nemen bij Vicoma.

Datum: 18-12-2018

Voor akkoord bedrijfsbegeleider:

Datum: 18-12-2018

Voor gezien student: