Internship Report [Confidential]

THE DESIGNING PROCESS OF A LONG DISTANCE STEAM TRANSPORT SYSTEM

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Abstract
The combustion of biomass has a large share in the production of sustainable energy. Kara Energy Systems takes part in this branch by designing biomass energy plants which are normally placed inside a construction. For a new project dry steam has to be transported over a distance of 120 meters outside. This change in conditions and the long distance require a different approach regarding expansion and construction. First the requirements of the project are listed. Then these requirements are elaborated upon with calculations. By examining several pipe diameters, based on velocity calculations and accompanying pressure drop and heat loss a pipe diameter is chosen. Using simulations, the mounting structures to the buildings are designed. The pipe support options are then investigated with the calculated expansion. Some necessary components are examined and the forces on the pipes and structures are calculated. Finally, a short comparison analysis is done to be able to choose the support structures. Using Finite Element Method analyses, dimensions for structural parts are found by setting a maximal displacement and stress. All components are put into an assembly which incorporates the requirements and the calculations. The result is a structural design which can be finalized and assembled on site.

Contents

Chapter 1

Introduction

1.1 Company introduction

The combustion of biomass has a large share in the production of sustainable energy. In 2015, 50% of sustainable energy and 83% of the sustainable heat production was produced with biomass [1]. KARA Energy Systems plays a role in this market by supplying custom-made bio-energy and heat installations based on biomass combustion. Depending on the used biomass and required energy and heat output, a solution is engineered. Environmental requirements and accompanying subsidies play a large role, since this can greatly reduce the payback period. With many customers, production waste has no real purpose. It is then very interesting to convert this waste into heat and electricity. Kara Energy Systems is the lead executor of the design, logistics, and the building of the entire custom-made installation.

1.2 Problem definition

ForFarmers Lochem has decided to renew the current natural gas-powered boiler which provides steam to provide for their production process which involves forming presses powered by steam [2]. Electricity is currently purchased from an energy provider. By using a biomass energy plant, which uses wood chips as fuel, this reduces the need for fossil fuels and it also reduces the emission of greenhouse gases [2]. There is not enough space to house a new biomass installation in or near the old boiler room, while still freight traffic across the terrain. An overview of the company grounds is shown in figure 1.1. This brings forth a new problem. Dry steam which is needed for the production processes has to be supplied to the boiler room. It has to be transported from the biomass plant to the boiler room from which it can be distributed to the production processes. This distance is long compared to normal travelling distances within a biomass plant, outside conditions have to be taken into account and this has not been done before by the company.
The high pressure boiler supplies steam at 44 bar(a). This is then sent through the superheater to obtain superheated steam at $400°C$. The superheated steam powers a turbine which induces a pressure drop while producing electricity. For maximum electricity production the pressure drop is at the maximum specification of the turbine. The remaining steam is still superheated and feedwater at $100°C$ is injected in the steam flow which then evaporates over a distance of 15 meters. The amount of feedwater determines the final temperature of the steam flow. This amount is closely monitored to get a higher flow rate while

still remaining sufficiently superheated to be able to supply dry steam to the production processes. Some flow is sent to the deaerator which removes dissolved corrosive gases like oxygen and carbon dioxide from boiler feedwater by bubbling steam through it [4]. The remaining steam flow is transported along the aforementioned steam transportation. The length of this transportation system causes problems with heat loss and pressure drop which have an effect on the state of the steam. Too much heat loss and/or loss of pressure and the steam quality of the remaining steam reduces, which effectively lowers the available steam for the production processes. Material which heats up will expand and at these distances the elongation must be accounted for with the supports or large forces will occur. To allow for free flow of condensate through the pipes during start-up, the pipes should be mounted at a downwards slope so gravity guides the medium through.

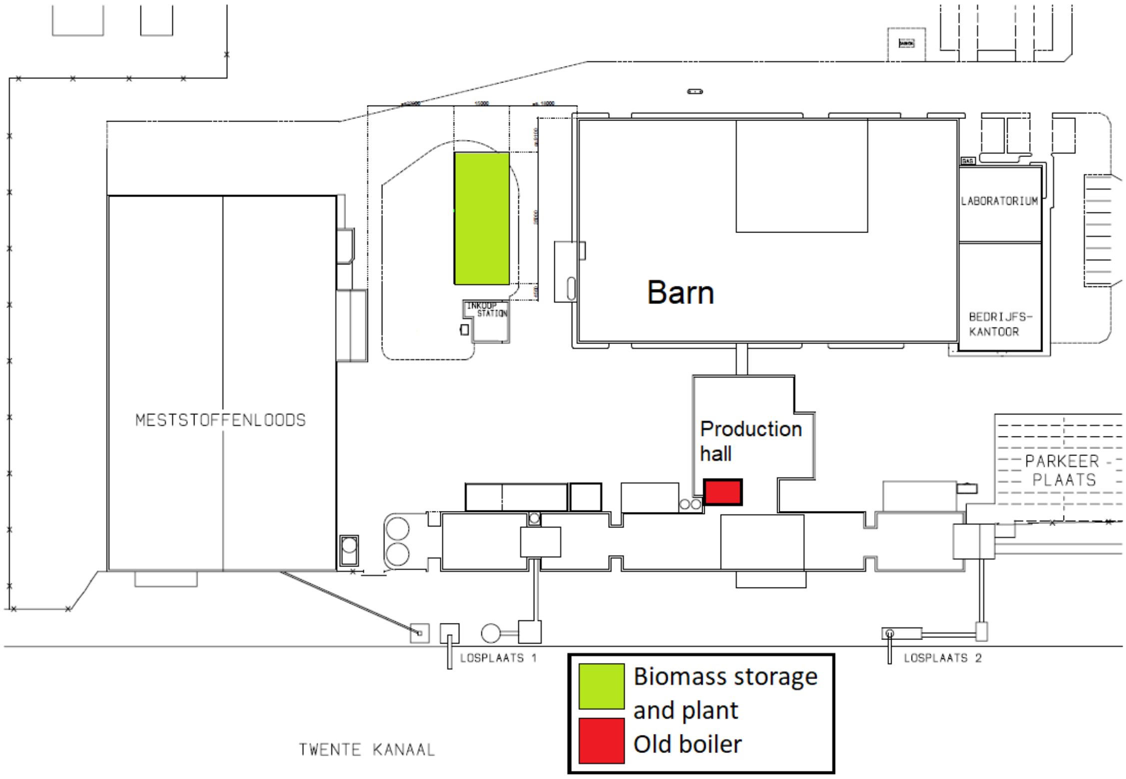

Figure 1.1: A map showing the old boiler room and the new biomass plant location

1.3 Requirements

The resulting design has to conform to a list of requirements which are listed below:

- When the steam exits the transport pipe, it has to be dry steam (+5°C super heated).
- Working pressures of 3 to 10 bar absolute should be allowed.
- The mass flow of steam of 3000-3500 kg/hr should be allowed.
- The steam velocities through the pipes should be below 25-40 m/s to reduce the risk of water hammer.

- The pressure drop should be small.
- The supporting structures have to allow for thermal expansion of the pipes without stress failure.
- The supporting structures should not obstruct traffic.
- The piping should remain above 4 meters to allow for traffic passing underneath.
- The idle system should be able to cope with temperatures below 0°C.
- For economic reasons, standard profiles are preferable.
- Steam pipes should be mounted at a 1:100(0.57°) downward gradient in the direction of the steam flow [3].
- Condensate removal pipes should be mounted at a 1:70(0.82°) downward gradient in the direction of the condensate flow [3].

Chapter 2

Calculations

2.1 Steam velocity and pressure drop

The steam velocity can be calculated by the following equation:

$$u = \frac{\dot{m}}{\rho \cdot \pi r^2} \tag{2.1}$$

The density of steam is dependent on the pressure and the temperature. The maximum mass flow is governed by the mass flow of the boiler which can output a maximum of 3300 kg of steam per hour at 44 bar(a). There is some feed water injection and some outflow to the deaerator and the remaining flow continues through the long distance piping system to the steam distributor which supplies the steam to the production processes. For maximum electricity production while still remaining suffieciently superheated steam, the outgoing pressure is set to 3 bar(a). The steam is slightly superheated and not much feedwater is injected. Some steam is sent to the deaerator and the remaining mass flow is around 3098 kg/hr. The density at this pressure and temperature is known and the velocity can be calculated for various diameters. This density can be found by interpolating from water and steam property tables or by using an industry standard which divides the pressure and temperature into different regions shown in figure 2.1. The properties in these different regions of the water/steam are calculated using basic equations which relate thermodynamic properties to the dimensionless Gibbs g or Helmholtz f free energy number which can be calculated using various summations but this falls outside the scope of this research. For further reading, [5] can be used as reference.

For a lower pressure drop across the turbine a higher pressure superheated steam is obtained. More feedwater can be injected while still remaining sufficiently superheated steam. The case for 10 bar(a) is also calculated. According to Spirax Sarco ltd.[3] the steam velocities should not surpass 40 m/s, because this could risk water hammer and an increased pressure drop. Instead velocities of around 25 to 30 m/s are preferable. By iterating over commercially available pipe sizes this resulted in the following results shown in table 2.1.

Using a larger diameter pipe will result in a lower pressure drop due to the lower velocity. Using a pipe which is too large causes more material and installation costs and the larger surface area causes more heat loss. Using a smaller diameter pipe causes less heat loss, but due to increased velocity, increases the risk of water hammer and erosion and the pressure drop will be higher [3]. Taking this in mind, the diameter size DN150 looks promising, but

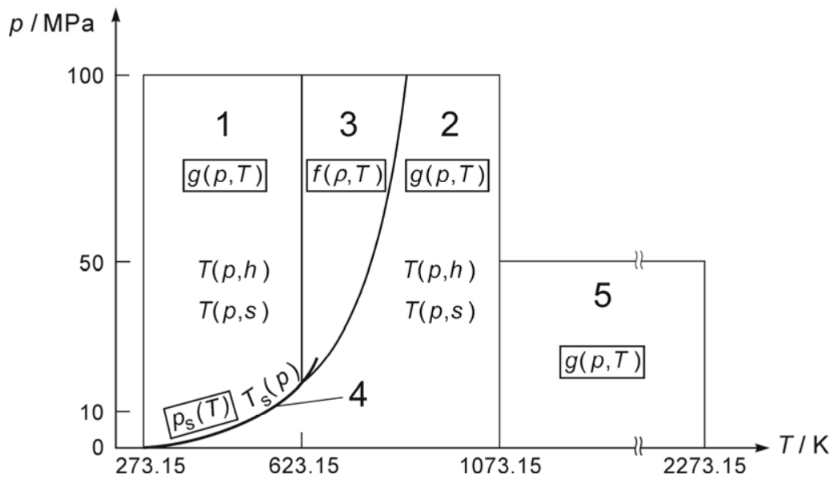

Figure 2.1: A map showing the old boiler room and the new biomass plant location

Table 2.1: The mass flows at certain pressures and resulting velocities for various pipe diameters

	DN125		DN150		DN200	
	3 bar(a)	10 bar(a)	3 bar(a)	10bar(a)	3 bar(a)	10 bar(a)
$\dot{m}[kg/h]$	3098	3500	3098	3500	3098	3500
T $[°C]$	140	185	140	185	140	185
$\rho[kg/m^3]$	1.62	5.07	1.62	5.07	1.62	5.07
r $[mm]$	64.05	64.05	80.75	80.75	105.75	105.75
u $[m/s]$	41.22	14.88	25.93	9.36	15.12	5.46

first the pressure drop has to be checked. This pressure drop can be calculated using the Darcy-Weisbach equation [6]:

$$\Delta p = f\frac{L}{d}\frac{\rho u^2}{2} \tag{2.2}$$

In this equation, f is the friction factor which is dependent on the Reynolds number and the relative roughness, and the length L is taken as 150 meters. The friction factor, which is turbulent since the length is sufficiently high(L≫60d), can either be obtained from a Moody chart or be calculated by iteration or be estimated with equation 2.3 [7]:

$$\begin{aligned} f &= \frac{1}{(0,79ln(Re) - 1,64)^2} \\ Re &= \frac{\rho u d}{\mu} \end{aligned} \tag{2.3}$$

With μ being the dynamic viscosity which can be derived from the thermodynamic properties [5]. Now the Darcy-Weisbach equation can be filled in and the results are shown in table 2.2:

The pressure drop in bends also has to be taken into account. A formula describing this is described by Babcock & Wilcox Co.[8]:

$$\Delta p = \frac{1}{2}f\rho u^2\frac{\pi R_b}{D}\frac{\theta}{180°} + \frac{1}{2}k_b\rho u^2 \tag{2.4}$$

Table 2.2: Pressure drop

	DN125		DN150		DN200	
	3 bar(a)	10 bar(a)	3 bar(a)	10 bar(a)	3 bar(a)	10 bar(a)
u $[m/s]$	41.22	14.88	25.93	9.36	15.12	5.46
μ $[Pas]$	1.37E-05	1.52E-05	1.37E-05	1.52E-05	1.37E-05	1.52E-05
Re [-]	6.26E+05	6.36E+05	4.97E05	5.04E05	3.79E05	3.85E05
f [-]	0.0126	0.0126	0.0131	0.0131	0.0138	0.0138
Δp $[mbar]$	203.2	82.7	66.5	27.1	18.1	7.4
Δp per bend $[mbar]$	4.12	1.68	1.64	0.67	0.56	0.23

In this equation most values are known considering 90 degree bends, and using a bend radius which is 1.5 times the tube diameter. The value for k_b can be found in figure2.2. The results of this calculation are also shown in table 2.2. It can be seen that with DN125, the velocities for 3 bar(a) are exceeding the 40 m/s limit. This also causes pressure drops which are relatively high compared to the working pressure. For 10 bar(a) the resulting velocity and pressure drop look reasonable. For the case with DN150, the velocity and pressure drop for 3 bar(a) are acceptable. And this is similar for the 10 bar(a) case and both pressures for DN200.

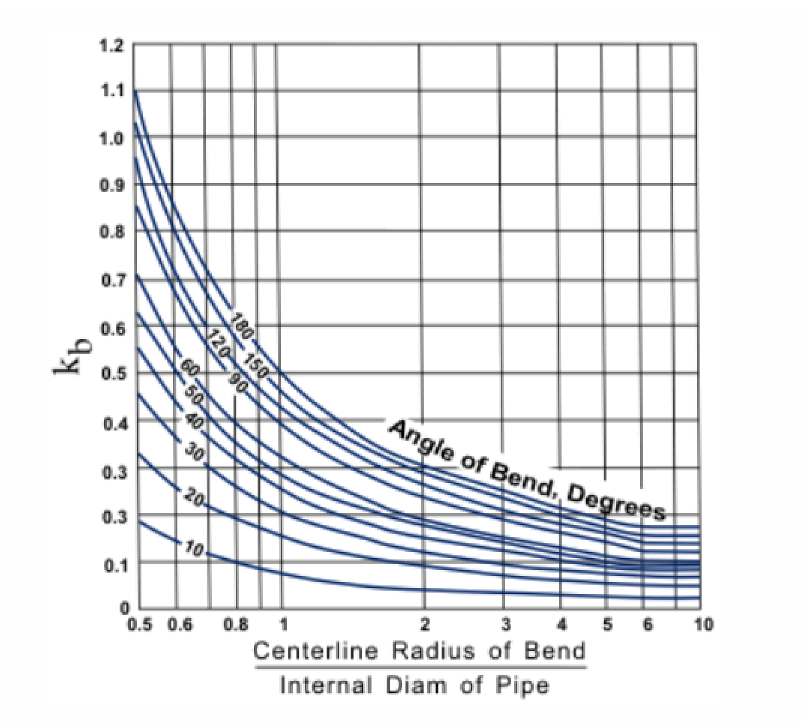

Figure 2.2: Bend loss coefficient for a pipe [8]

2.2 Heat loss and required inlet temperature

The steam is transported through tubing which runs in outside conditions. The superheated steam heats the pipe up to the same temperature and if the pipe is not insulated this heat is passed on to the surrounding air. The pipe material is chosen as P235GH which is commonly used in steam-generating plants which is suitable for this project. It can be used in continuous operation at temperatures up to about $450°C$ [9]. When the plant is started from stand-by, the whole steam transport pipe has to be heated from ambient temperatures. In this process the required heat is taken by condensing the input steam. The amount of condensate can be calculated by taking the required heat from the specific thermal capacity of the material and using the mass of the pipe system and the temperature change and by using the latent heat of vaporization of the steam. The steps are shown in table 2.3. Thus approximately 61-93 liters of condensate need to be removed from the pipes during start-up. The effect of these numbers will be discussed at the end of this subchapter. It needs to be said that the heat loss to the surrounding air has been neglected and the implications will be checked.

Table 2.3: Formed condensate for start-up for two working pressures with DN150 sized pipes

c_p $[kJ/kgK]$	0.46	
Outer diameter $[m]$	0.1683	
Inner diameter $[m]$	0.1593	
Area $[m^2]$	0.002316	
Density $[kg/m^3]$	7850	
Weight per meter $[kg/m]$	18.18	
Total weight $[kg]$	2727	
Ambient temperature $[°C]$	10	
Working pressure $[bar(a)]$	3	10
Steam temperature $[°C]$	140	185
dT [K]	130	175
Required heat [MJ]	124.0	167.0
Specific enthalpy saturated water $[kJ/kg]$	561.4	762.6
Specific enthalpy superheated steam $[kJ/kg]$	2739.0	2790.5
Latent heat of evaporation $[kJ/kg]$	2177.6	2027.9
Formed condensate [kg]	56.96	82.33
Density $[kg/m^3]$	931.0	886.8
Formed condensate $[dm^3]$	61.17	92.84

For the heat loss during operation the thermal resistance is used for the various layers. This is done for the case without insulation, with 50 mm of insulation material and 100 mm insulation material. The thermal resistance can be calculated with the following equation [10]:

$$R_{layer} = \frac{ln(r_{outer}/r_{inner})}{2\pi \cdot \lambda_k} \qquad [m \cdot K/W] \tag{2.5}$$

The λ_k is the thermal conductivity of the corresponding layer with specified radii. The total thermal resistance of the pipe with and without insulation and the surface heat transfer resistances to the steam and external air can be calculated using:

$$R_{tot} = \frac{R_{steam}}{2\pi r_1} + \sum_{n-1}^{k=1} \frac{r_{k+1}/r_k}{2\pi\lambda_k} + \frac{R_{air}}{2\pi r_n} \qquad [m \cdot K/W] \tag{2.6}$$

R_{steam} and R_{air} are the heat transfer resistances, which are equal to the [heat transfer coefficient $h]^{-1}$, between fluid and material. R_{steam} and R_{air} are found to be 1/10000 and 1/8 $[m^2 \cdot K/W]$, respectively [11], [10]. The thermal conductivity for P235GH is found to be 51 $[W/m \cdot K]$. The type of insulation used is Rockwool which has a thermal conductivity of 0.048 $[W/m \cdot K]$ at 150°C and 0.056 $[W/m \cdot K]$ at 200°C [12]. For the non-insulated case this results in:

$$R_{tot} = \frac{1/10000}{2\pi \cdot 0.07965} + \frac{ln(\frac{0.08415}{0.07965})}{2\pi \cdot 51} + \frac{1/8}{2\pi \cdot 0.8415} = 0.096 \quad [m \cdot K/W] \tag{2.7}$$

With 50 mm insulation this becomes:

$$R_{tot} = \frac{1/10000}{2\pi \cdot 0.07965} + \frac{ln(\frac{0.08415}{0.07965})}{2\pi \cdot 51} + \frac{ln(\frac{0.13415}{0.08415})}{2\pi \cdot \lambda_{insulation}} + \frac{1/8}{2\pi \cdot 0.8415} \quad [m \cdot K/W] \tag{2.8}$$

And similarly for 100 mm of insulation:

$$R_{tot} = \frac{1/10000}{2\pi \cdot 0.07965} + \frac{ln(\frac{0.08415}{0.07965})}{2\pi \cdot 51} + \frac{ln(\frac{0.18415}{0.08415})}{2\pi \cdot \lambda_{insulation}} + \frac{1/8}{2\pi \cdot 0.8415} \quad [m \cdot K/W] \tag{2.9}$$

The thermal resistance can be used to calculate the heat flow over the entire piping at a certain temperature with the following equation when using an ambient temperature of 10°C [10]:

$$\Phi_{tot} = \frac{T_{fluid} - T_{air}}{R_{tot}} \cdot L \tag{2.10}$$

Now that we have found the heat loss, the required inlet temperature can be found. The heat loss will be extracted from the steam flow. The required output steam is dry steam(+5°C superheated) for which the enthalpy is known. By dividing the heat loss over the mass flow, the enthalpy margin and the required enthalpy of the inlet can be calculated. This can be used to find the temperature of the incoming steam. The required temperature of the incoming steam is also shown in table 2.4.

To see if the heat loss is relevant during start-up first the duration of the heating of the pipes has to be estimated. During the start-up process, the furnace starts heating up. The gases are passed through the boiler where water starts to boil. Steam is passed through the turbine and deaerator and back into the boiler. The pressure level starts to build up and the valves to the steam transport pipes can be opened which will start to heat up as well. It is estimated that approximately 10% of the maximum flow is used. This means that the previously calculated 124 MJ for 3 bar is supplied at a rate of 0.086 kg/s. The required amount of steam was already calculated and is denoted in table 2.3 as 57 kg. At the rate of 0.086 kg/s this will take around 650 seconds which is approximately 11 minutes. The average power input in the heating of the pipes is thus around $\frac{124000}{651} = 190.5kW$ which is a lot higher compared to the heat loss with some insulation (14 kW or 8.6 kW) and there is fewer heat loss at lower temperatures so disregarding the heat loss is justified.

At the 10 % flow rate the inlet temperature has to be even higher than the values in table 2.3 and if this is not accomplished then the pipes will stay at the transition temperature

Table 2.4: The thermal resistance, heat loss and required inlet temperature for varying insulation thicknesses with DN150 sized pipes

	Thermal resistance $[m \cdot K/W]$	
Insulation	140°C	185°C
0 mm	0.096	0.096
50 mm	1.611	1.390
100 mm	2.648	2.277
dT $[°C]$	130	175
L $[m]$	150	150
Insulation	Heat loss $\Phi[kW]$	
0 mm	203.1	273.3
50 mm	14.0	18.9
100 mm	8.6	11.5
Mass flow at 100 % $[kg/s]$	0.861	0.972
Enthalpy at distributor $[kJ/kg]$	2739.4	2790.9
Insulation	Inlet temperature $[°C]$ (dT)	
0 mm	253.7 (113.7)	309.5 (124.5)
50 mm	147.4 (7.4)	192.7 (7.7)
100 mm	144.6 (4.6)	189.7 (4.7)
Mass flow at 30 % $[kg/s]$	0.258	0.292
Insulation	Inlet temperature $[°C]$ (dT)	
0 mm	518.4 (378.4)	613.2 (428.2)
50 mm	165.2 (25.2)	211.5 (26.5)
100 mm	155.3 (15.3)	200.8 (15.8)

but some condensation will occur and this will have to be removed from the system. This is no problem and the starting of the installation also will not occur frequently. Once a plant starts it is uneconomical to let it cool down and restart so this is only done twice a year for maintenance. During operation the mass flow will not fall below 30% for which also the inlet temperatures are shown in table 2.3.

The heat loss calculations have also been done for the DN125 and DN200. With an additional 20% of heat loss and since it is not necessary regarding the steam velocity, DN200 is a bad candidate. DN125 did have 8% less heat loss compared to DN150 but with the steam velocity of 3 bar(a) steam, at maximum mass flow, exceeding 40 m/s this is also not preferred. DN150 has good steam velocities and the heat loss is manageable and is therefore chosen.

2.3 Wall thickness

At the design temperature of $220°C$, the yield strength can be interpolated from the P235 data sheets [9] resulting in a yield strength $R_{eH,T}$ of 162 MPa. The tensile strength R_m is 360 MPa. The design stress for the pipe according to EN 13480-3 [13] equals:

$$f = min\left[\frac{R_{eH,t}}{1.5}; \frac{R_m}{2.4}\right] = min\left[\frac{162}{1.5}; \frac{360}{2.4}\right] = 108\ MPa \tag{2.11}$$

The minimum required wall thickness can be calculated as:

$$e = \frac{p_c D_0}{2fz + p_c} = \frac{1\frac{N}{mm^2} \cdot 168.3mm}{2 \cdot 108\frac{N}{mm^2} \cdot 1 + 1\frac{N}{mm^2}} = 0.78\ mm \tag{2.12}$$

In this equation, p_c is the design pressure of the medium, D_o is the outer diameter and z is the joint coefficient which is equal to 1. The maximum internal pressure can then be calculated by comparing the required wall thickness with the actual wall thickness after taking into account corrosion tolerance, thinning tolerance and the supplied material tolerance.

$$P_{max} = \frac{p_c \cdot [w - c_1 - c_2 - c_3]}{e} = \frac{1\frac{N}{mm^2} \cdot [4.5 - 0.5 - 0.5 - 0.563]\,mm}{0.78mm} = 3.79\frac{N}{mm^2} = 37.9\ bar(g) \tag{2.13}$$

The steam pipes which are welded together clearly can sustain the internal pressure. The maximum internal pressure for tee fittings is lower. The maximum allowable pressure according to EN 10253-2 [36] is calculated using a pressure factor X which is 33 for Tees Type A with this diameter and wall thickness.

$$P_{max,fitting}\frac{P_{max} \cdot X}{100} = \frac{37.9\ bar(g) \cdot 33}{100} = 12.5\ bar(g) \tag{2.14}$$

The standard wall thickness for DN150 of 4.5 mm is therefore sufficient for the design pressure of 10 bar(g). It has been considered to choose a different wall thickness for the pipes and use a Type B Tee fitting and elbow bends (for use at full service pressure), which have thicker walls in the critical areas. Heavy duty fittings are substantially more expensive and the pipes won't become much cheaper since 4.5 mm is the industry standard for DN150. Additionally, most suppliers will only keep the standard sizes in stock and thus DN150 with 4.5 mm walls is chosen as the pipe size.

2.4 Support interval calculation

Now the pipe diameter is chosen, the interval of the supports can be determined. This is an important parameter in the lay-out of the piping support structure since it decides how many supports are necessary and the where to place them. When these are spaced too far apart, the pipe will deform under its own weight. Formed condensate can concentrate in these valleys and this is called puddle formation which can cause water hammer. For this reason 1 mm is used as deflection threshold. The pipe support spacing is mentioned in Spirax Sarco[3], but covered more extensively in European standard NEN 13480, Part 3 [13]. With the pipe dimensions and properties known, the mass per unit length is also known. The maximum weight is carried during the installation of the biomass plant when the pipes are tested hydrostatically by filling the pipes with water and by increasing the pressure to check for leakage and the ability to resist the working pressure which is described in NEN 1006:2.3 [14]. The weight of the insulation is dependent on the chosen thickness but it still is a small share in the whole as seen in table 2.5. The area moment of inertia is calculated as $\frac{\pi(D^4-d^4)}{64}$ and the deflection follows from a combination of beam deflection formulae as $w = \frac{8}{3}\frac{qL^4}{384EI}$. The maximum spacing is calculated as 4900 mm. This value will be used when designing and locating the supports. When comparing this value to recommended values shown in figures 2.3 and 2.4 it appears to be in the right range and this verified value is used.

Table 2.5: Support spacing calculation

Weight DN150	18.2	kg/m
Weight water content	17.9	kg/m
Weight insulation 100 mm	3.5	kg/m
Total weight	39.6	kg/m
Distributed load	0.389	N/mm
Moment of inertia	7.77E06	mm^4
Elastic modulus (200°C)	207000	N/mm^2
Spacing	4900	mm
Max. deflection	0.98	mm
Reaction force per support	2095	N

Table 10.4.3 Recommended support for pipework

Nominal pipe size (mm)		Interval of horizontal run (metre)		Interval of vertical run (metre)	
Steel bore	Copper outside diameter	Mild steel	Copper	Mild steel	Copper
	15		1.2	2.4	1.8
15		1.8		3.0	
20	22	2.4	1.2	3.0	1.8
25	28	2.4	0.5	3.0	2.4
32	35	2.4	1.8	3.7	3.0
40	42	2.4	1.8	3.7	3.0
50	54	2.4	1.8	4.6	3.0
65	67	3.0	2.4	4.6	3.7
80	76	3.0	2.4	4.6	3.7
100	108	3.0	2.4	5.5	3.7
125	133	3.7	3.0	5.5	3.7
150	159	4.5	3.7	5.5	
200		6.0		8.5	
250		6.5		9.0	
300		7.0		10.0	

Figure 2.3: The allowable support spacing according to Spirax Sarco [3]

Table Q.1 — Allowable spacing of supports for steel pipes (for boundary conditions refer to Explanatory Notes for Table Q.1)

			Empty pipe, without insulation							Water-filled pipe, without insulation							Water-filled pipe, insulation thickness 40 mm							Water-filled pipe, insulation thickness 80 mm						
DN	d_a	s	q	L_1	L_2	L_3	L_4	L_5	L_6	q	L_1	L_2	L_3	L_4	L_5	L_6	q	L_1	L_2	L_3	L_4	L_5	L_6	q	L_1	L_2	L_3	L_4	L_5	L_6
	mm		kg/m			m				kg/m			m				kg/m			m				kg/m			m			
25	33,7	2,0	1,6	2,9	5,6	4,8	2,9	2,8	1,5	2,3	2,7	4,6	4,0	2,5	2,3	1,2	7,0	2,0	2,7	2,3	1,4	1,3	0,7	11,8	1,8	2,0	1,8	1,1	1,0	0,5
25	33,7	4,0	2,9	2,9	5,2	5,2	3,6	2,6	1,8	3,5	2,8	4,8	4,8	3,3	2,4	1,7	8,1	2,2	3,2	3,2	2,2	1,6	1,1	13,0	2,0	2,5	2,5	1,7	1,3	0,9
40	48,3	2,0	2,3	3,6	6,8	5,2	3,2	3,4	1,6	3,9	3,1	5,2	4,0	2,4	2,6	1,2	9,2	2,5	3,4	2,6	1,6	1,7	0,8	14,3	2,3	2,7	2,1	1,3	1,4	0,6
40	48,3	4,0	4,4	3,5	6,5	6,4	3,9	3,3	1,9	5,7	3,3	5,7	5,6	3,4	2,9	1,7	11,0	2,8	4,1	4,0	2,5	2,1	1,2	16,1	2,5	3,4	3,3	2,0	1,7	1,0
50	60,3	2,0	2,9	4,0	7,6	5,4	3,3	3,8	1,7	5,4	3,4	5,6	4,0	2,4	2,8	1,2	11,3	2,9	3,9	2,7	1,7	1,9	0,8	16,6	2,6	3,2	2,3	1,4	1,6	0,7
50	60,3	4,5	6,2	3,9	7,3	6,9	4,2	3,7	2,1	8,3	3,6	6,4	6,0	3,6	3,2	1,8	14,2	3,2	4,9	4,6	2,8	2,4	1,4	19,4	2,9	4,2	3,9	2,4	2,1	1,2
80	88,9	2,3	5,0	5,5	9,3	6,1	3,7	4,7	1,9	10,6	4,6	6,4	4,2	2,5	3,2	1,3	17,8	4,0	4,9	3,2	2,0	2,5	1,0	23,5	3,7	4,3	2,8	1,7	2,1	0,9
80	88,9	5,6	11,5	5,4	9,0	8,0	4,9	4,5	2,4	16,3	5,0	7,6	6,7	4,1	3,8	2,1	23,5	4,5	6,3	5,6	3,4	3,2	1,7	29,2	4,3	5,7	5,0	3,1	2,8	1,5
100	114,3	2,6	7,3	6,3	10,6	6,6	4,0	5,3	2,0	16,6	5,1	7,0	4,4	2,7	3,5	1,3	25,0	4,6	5,7	3,5	2,2	2,9	1,1	31,1	4,4	5,1	3,2	1,9	2,6	1,0
100	114,3	6,3	16,8	6,2	10,3	8,7	5,3	5,1	2,7	24,9	5,6	8,5	7,1	4,4	4,2	2,2	33,3	5,2	7,3	6,2	3,8	3,7	1,9	39,4	5,0	6,7	5,7	3,5	3,4	1,7
150	168,3	2,6	10,8	7,6	13,0	7,1	4,3	6,5	2,2	31,7	5,8	7,6	4,1	2,5	3,8	1,3	42,6	5,4	6,5	3,6	2,2	3,3	1,1	49,5	5,2	6,0	3,3	2,0	3,0	1,0
150	168,3	7,1	28,2	7,5	12,6	9,7	5,9	6,3	3,0	46,9	6,6	9,8	7,6	4,6	4,9	2,3	57,8	6,3	8,8	6,8	4,2	4,4	2,1	64,7	6,1	8,4	6,4	3,9	4,2	2,0
200	219,1	2,9	15,7	8,7	14,8	7,7	4,7	7,4	2,3	51,4	6,5	8,2	4,2	2,6	4,1	1,3	64,7	6,1	7,3	3,8	2,3	3,6	1,2	72,3	5,9	6,9	3,6	2,2	3,4	1,1
200	219,1	7,1	37,1	8,6	14,6	10,2	6,2	7,3	3,1	70,1	7,4	10,6	7,4	4,5	5,3	2,3	83,4	7,1	9,7	6,8	4,2	4,9	2,1	91,0	6,9	9,3	6,5	4,0	4,7	2,0
250	273,0	2,9	19,6	9,8	16,6	8,0	4,9	8,3	2,4	75,6	6,9	8,4	4,1	2,5	4,2	1,2	91,5	6,6	7,7	3,7	2,3	3,8	1,1	99,9	6,5	7,3	3,5	2,2	3,7	1,1
250	273,0	7,1	46,6	9,7	16,4	10,6	6,5	8,2	3,3	99,2	8,0	11,2	7,3	4,5	5,6	2,2	115,0	7,7	10,4	6,8	4,1	5,2	2,1	123,4	7,6	10,1	6,6	4,0	5,0	2,0
300	323,9	2,9	23,3	10,6	18,1	8,2	5,0	9,1	2,5	102,7	7,3	8,6	3,9	2,4	4,3	1,2	120,9	7,0	7,9	3,6	2,2	4,0	1,1	130,1	6,9	7,6	3,5	2,1	3,8	1,1
300	323,9	8,0	62,3	10,6	17,8	11,4	7,0	8,9	3,5	136,8	8,7	12,1	7,7	4,7	6,0	2,4	155,0	8,4	11,3	7,2	4,4	5,7	2,2	164,2	8,3	11,0	7,0	4,3	5,5	2,2
350	355,6	3,2	28,2	11,1	19,0	8,6	5,3	9,5	2,6	123,9	7,7	9,0	4,1	2,5	4,5	1,3	143,6	7,4	8,4	3,8	2,3	4,2	1,2	153,3	7,3	8,1	3,7	2,3	4,1	1,1
350	355,6	8,8	75,3	11,1	18,7	12,0	7,3	9,4	3,7	165,0	9,1	12,7	8,1	4,9	6,3	2,5	184,7	8,8	12,0	7,7	4,7	6,0	2,3	194,3	8,7	11,7	7,5	4,6	5,8	2,3
400	406,4	3,2	32,2	11,9	20,3	8,8	5,4	10,2	2,7	157,9	8,0	9,2	4,0	2,4	4,6	1,2	179,9	7,7	8,6	3,7	2,3	4,3	1,1	190,4	7,6	8,3	3,6	2,2	4,2	1,1
400	406,4	10,0	97,8	11,8	20,0	12,8	7,8	10,0	3,9	215,0	9,7	13,5	8,6	5,3	6,8	2,6	237,0	9,5	12,9	8,2	5,0	6,4	2,5	247,5	9,4	12,6	8,0	4,9	6,3	2,5
500	508,0	4,0	50,4	13,3	22,7	9,9	6,0	11,4	3,0	246,7	8,9	10,2	4,4	2,7	5,1	1,4	273,4	8,7	9,7	4,2	2,6	4,9	1,3	285,4	8,6	9,5	4,1	2,5	4,8	1,3
500	508,0	11,0	134,8	13,2	22,4	13,7	8,4	11,2	4,2	320,3	10,7	14,6	8,9	5,4	7,3	2,7	347,1	10,5	14,0	8,6	5,2	7,0	2,6	359,1	10,4	13,8	8,4	5,1	6,9	2,6

Figure 2.4: The allowable spacing for a pipe supported at both ends according to NEN 13480 [13]

2.5 Larger distance support

For the larger share, the steam pipe and other pipes will be mounted to nearby buildings. This requires the least amount of newly constructed support structures which are costly and can obstruct the present traffic flow. To overcome the distances between buildings a structure has to be designed since the bridging distance is greater than the support spacing distance. The crossing between the biomass installation and the warehouse which is around 18 meters and a crossing between the warehouse and the production hall which is around 8.6 meters. For both these crossings first an estimation is done with standard H beam profiles [15] to give an indication of the dimensions of the supporting structure.
This beam should not deflect too much, but the maximum deflected angle should determine the size of the beam. For designing purposes the steam pipe should be at a fall of at least 1:100(0.57°) in the direction of the steam flow [3]. This way both gravity and steam assist the formed condensate towards drain pots where it can be removed from the system. The condensate removal pipes should be mounted at an even steeper angle of 1:70(0.82°) to maintain free flow of condensate [3]. If the support beam should deform too much this could obstruct the free flow of the steam and condensate. To limit this risk, the maximum angle of deflection is set to be 0.1°. This way the average fall over the beam is maintained while the overall fall is not increased. The overall fall should be minimized to maximize the margin between the piping system and the road. The calculation is similar to the calculation of the deflection of the steam pipes and is shown in table 2.6. For the 8.6 meters crossing, a beam profile with a height of 200 mm and width of 150 mm is sufficient. The maximum stress does not exceed the maximum yield stress and the maximum angle of deflection is close to the aforementioned 0.1°. For some design validation, simulation software [16] was used. To verify the results of this software package and the used beam deflection equation, a simple simulation was performed with a beam with dimensions as in [15] and with a distributed load as stated in table 2.6. The model results are shown in figure 2.5 and differ from the calculations by 0.5% so the results correspond well.

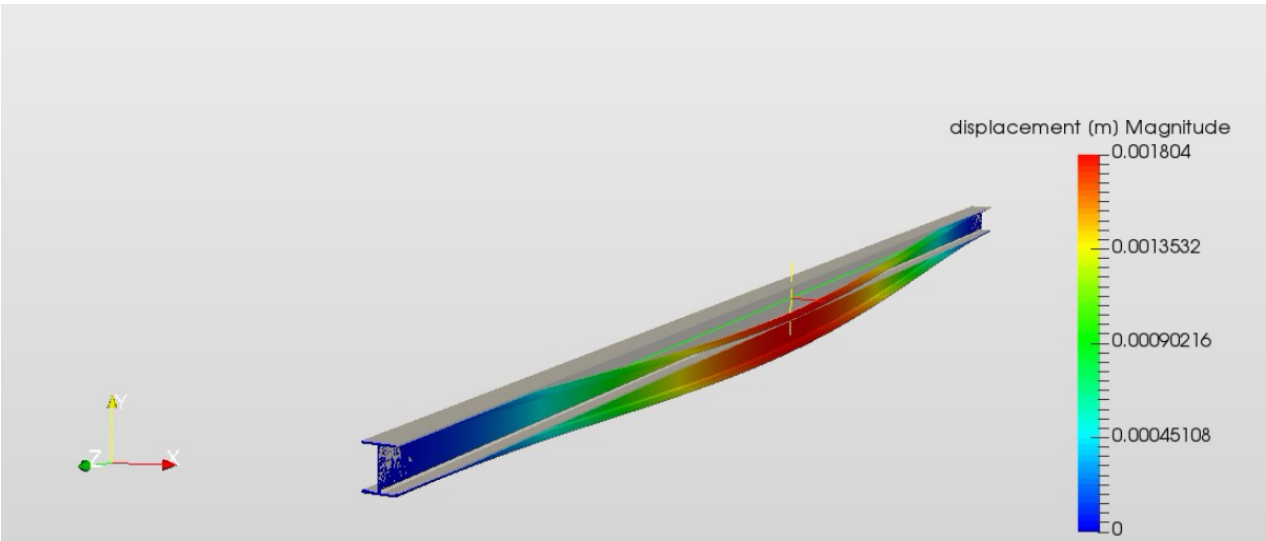

Figure 2.5: The simulated deflection in SimScale Workbench

For the 18 meter crossing logically a much larger profile is needed. As can be seen in table 2.6, the mass of the bridge contributes to a large share of all load upon itself. Using a

Table 2.6: Required h beam profile

Profile	450x300	200x150	height x width
Unit mass			
Bridge	121	29.9	kg/m [15]
Pipe	18.2	18.2	kg/m
Water content	17.9	17.9	kg/m
Insulation 100 mm	3.5	3.5	kg/m
Total weight	160.6	68.5	kg/m
Load	1.57	0.68	N/mm
Length	18000	8600	mm
Elastic modulus	200	200	GPa
Moment of inertia	54700	2630	cm^4 [15]
Section modulus	2490	271	mm^3 [15]
Deflection	3.9	1.8	mm
Max. angle of deflection	0.098	0.095	°
Max. moment	-62806	-6075	Nm
Max. stress	25	22	MPa
Load at sides	14.0	2.8	kN

Figure 2.6: Traditional truss bridge [17]

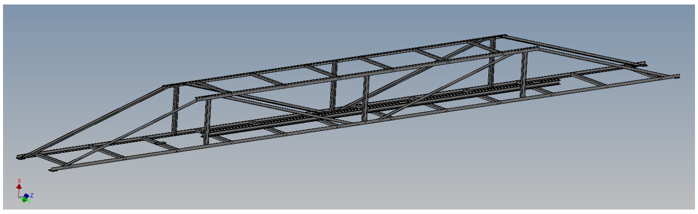

Figure 2.7: Design in Autodesk Inventor

larger profile will hardly increase the performance. Using such a large profile is not desired and an alternative is sought. With angled profiles and rolled steel channels, a bridge truss structure has been designed for specifically this purpose. The design and simulation results are shown in figures 2.6 through 2.9. The displacements are well within bounds, and the maximum angle is at around 0.04° so this is satisfactory albeit overdimensioned. In chapter 3.5 the bridge design will be developed further.

2.6 Thermal expansion

The piping system is installed at around $20°C$. Any deviation from this temperature will cause the material to expand or retract. Since the pipe lengths are long, and the transported steam is at $\approx 190°C$, the pipes will expand in orders that can't be solved by sheer elastic deformation of the supports. To visualize the forces which appear when restricting a thermal expanding pipe, a short example is elaborated on.

A DN150 P235-GH pipe with length 30 m is heated from $20°C$ to $190°C$. The expansion coefficient for this material is 13.0 $10^{-6}K^{-1}$ between $20°C$ and $200°C$ [9]. The elastic mod-

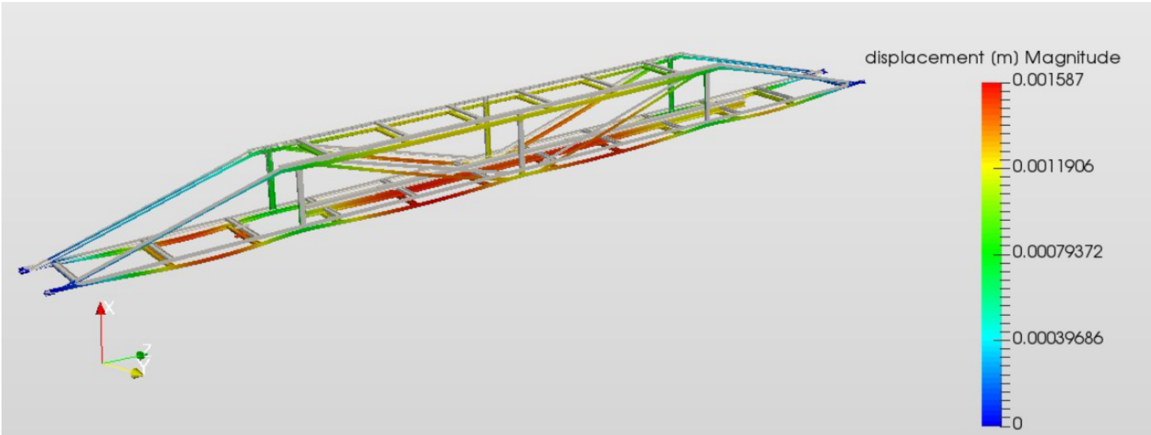

Figure 2.8: The simulated deflection in SimScale Workbench

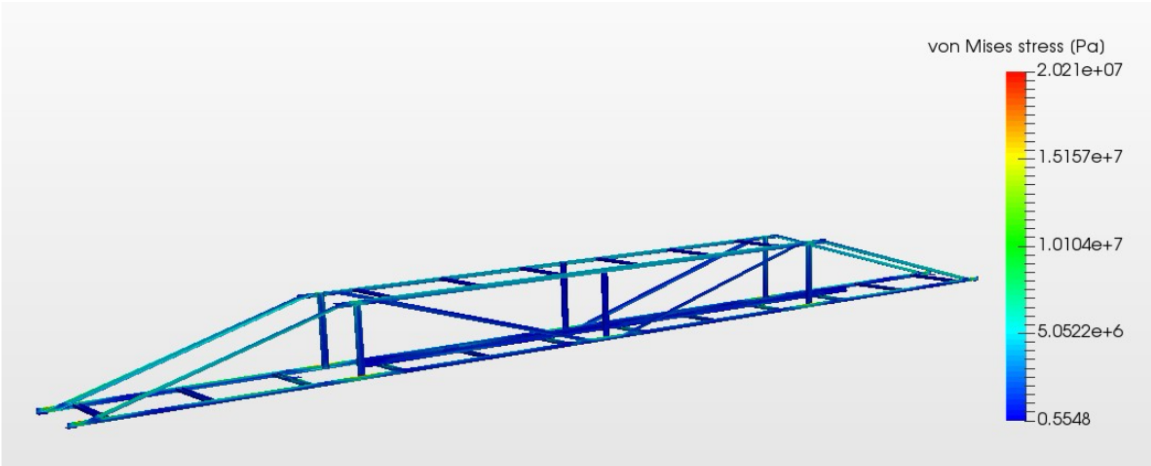

Figure 2.9: The simulated stress concentration in SimScale Workbench

ulus for this material is 208 GPa at $200°C$ [9]. The thermal expansion for an unrestricted pipe is:

$$\begin{aligned} \Delta L &= \alpha \cdot L_0 \cdot dT \\ &= 13 \cdot 10^{-6} \cdot 30 \cdot 180 \\ &= 0.0702m = 70.2mm \end{aligned} \tag{2.15}$$

The thermal stress caused by thermal expansion is as follows:

$$\begin{aligned} \sigma_{dT} &= E \cdot \varepsilon \\ &= E \cdot \alpha \cdot dT \\ &= 208 \cdot 10^9 \cdot 13 \cdot 10^{-6} \cdot 180 \\ &= 486.7MPa \end{aligned} \tag{2.16}$$

The force acting at the ends of the pipe can then be calculated as:

$$\begin{aligned} F &= \sigma_{dT} \cdot A \\ &= 486.7 \cdot 10^6 \cdot 0.001761 \\ &= 857.3kN \end{aligned} \tag{2.17}$$

The immense stress exceeds the yield strength and the occurring forces are best left avoided. This means that the thermal expansion of the pipes must be allowed for. This can be done by using supports which do not take away all degrees of freedom. Some research [18][19][20][21], resulted in the following options which are shown in figures 2.10 through 2.14.

Allowing for the thermal expansion is critical but extra attention has to be paid to perpendicular connections and bends. The thermal axial expansion of a pipe will bend the connected pipe which is less rigid in this direction. In order for the occurring stresses to stay below the maximum allowable stress of the material, the length at which the pipe is fixed laterally is specified in the Metallic industrial piping European Norm [13]. This length is defined as:

$$L = \sqrt{\frac{3 \cdot E \cdot d_a \cdot f \cdot i_x}{10^6 \cdot f_h \cdot v}} \tag{2.18}$$

In this equation E stands for the elastic modulus which is known, d_a is the pipe outer diameter which is known, f is the thermal expansion for a pipe which depends on the temperature change and original length, i_x is a reduction factor which is 1.0 for a bend radius/diameter ratio of 1.5 or larger, f_h is the maximum allowable stress and v is the weld efficiency which is taken to be 0.85 [22]. An accompanying figure is displayed in figure 2.16. In the figure 'FP' is an anchor, 'FL' is a guide and 'LL' is a vertical stop.

As an example the first 18 m crossing with adjacent bend. The pipe length is around 17.5 meter as to keep some margin around the building. The 0.2 % proof stress of the material is 188 MPa at 185°C. The expansion when heated to 185°C can be calculated using equation 2.15 and results in 37.54 mm. This value is f_1 in figure 2.16. The required perpendicular length L_1 according to equation 2.18 equals:

$$L = \sqrt{\frac{3 \cdot 208 \cdot 10^3[MPa] \cdot 168.3[mm] \cdot 37.54[mm] \cdot 1.0[-]}{188[MPa] \cdot 0.85[-]}} \frac{1[m]}{1000[mm]} = 4.97[m] \tag{2.19}$$

This minimal length also thermally expands resulting in an elongation f_2 of

$$4970 \cdot 0.000013 \cdot 165 = 10.65[mm] \tag{2.20}$$

Which in turn requires L_2 to be:

$$L = \sqrt{\frac{3 \cdot 208 \cdot 10^3[MPa] \cdot 168.3[mm] \cdot 10.65[mm] \cdot 1.0[-]}{188[MPa] \cdot 0.85[-]}} \frac{1[m]}{1000[mm]} = 2.65[m] \tag{2.21}$$

The pipe in the bend also has to be supported over a smaller length than the maximum spacing. Therefore at 2 meters from the bend at both sides, a support is placed which can move both in the axial and lateral direction as in figure 2.14. An axial sliding support is not possible since both L_1 and L_2 are greater than 2 meters. The rest of the pipe lengths can be supported using axial supports where intervals of around 4 meters are used. If the length to the fixed point is changed of the perpendicular smaller pipe, obviously f_2 changes which influences L_2.

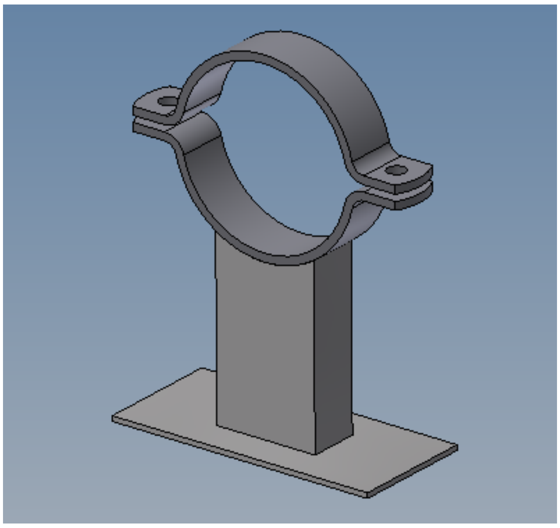

Figure 2.10: Rigid mount

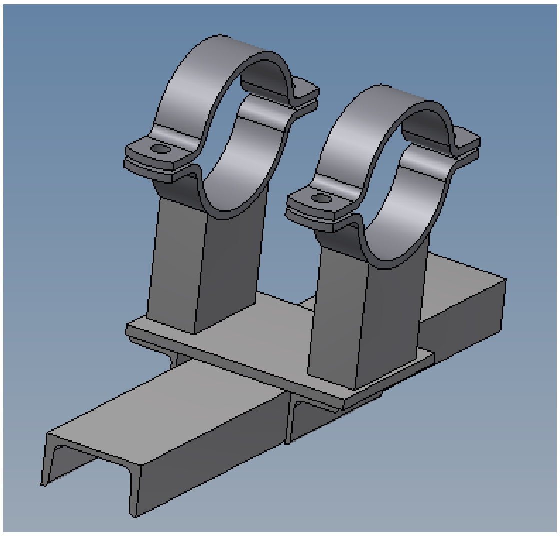

Figure 2.11: Allows for lateral movement

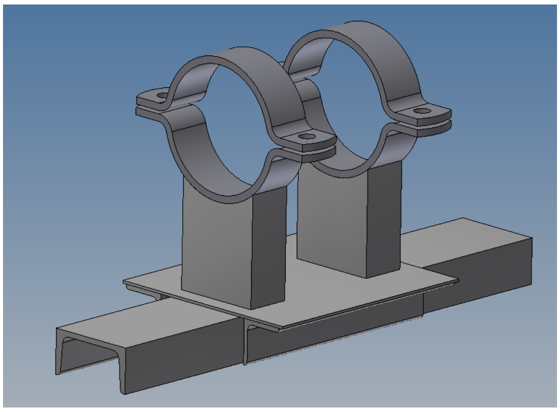

Figure 2.12: Allows for axial movement

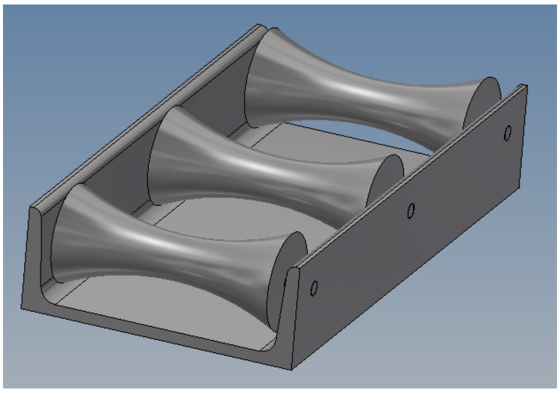

Figure 2.13: Allows for axial motion and some lateral motion

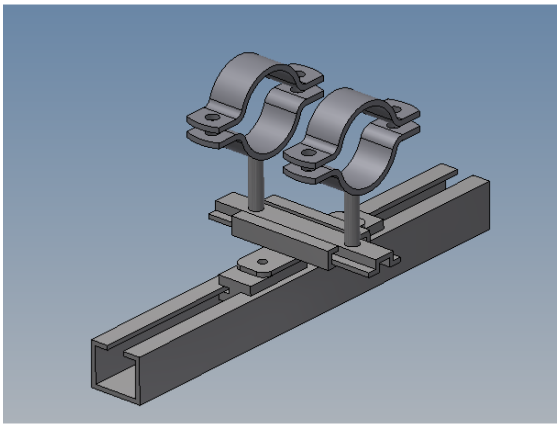

Figure 2.14: Allows for both lateral and axial movement

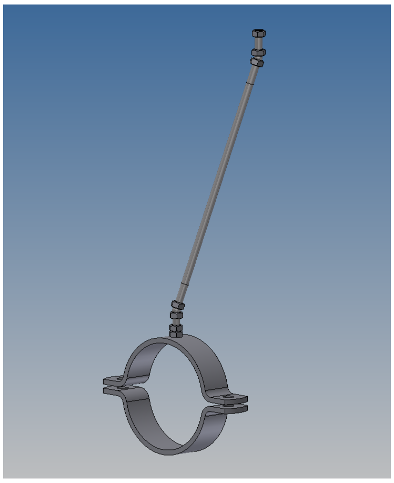

Figure 2.15: Allows for both lateral and axial movement

The entire system has been divided into 8 lengths of piping which are fixed at certain internal points. Along the lengths of pipes, supports have to be placed which limit the degrees of freedom of the pipe with the aforementioned design rule through thermal expansion in mind. A schematic overview of the pipe track is shown in figure 2.17. Dimensions between buildings have been adopted from a structural drawing of the terrain. By implementing the EN for every corner, and without exceeding the 4.7 m spacing distance results from section 2.4 this resulted in the design shown in figure 2.18. To design this, a script has been written which automatically checks if the design satisfies the spacing requirement and whether a support should be able to move in one or two directions. In this design it has also been taken into account that the final length L8 towards the steam distributor should be 4 meters or smaller to enable a good connection to the distributor.

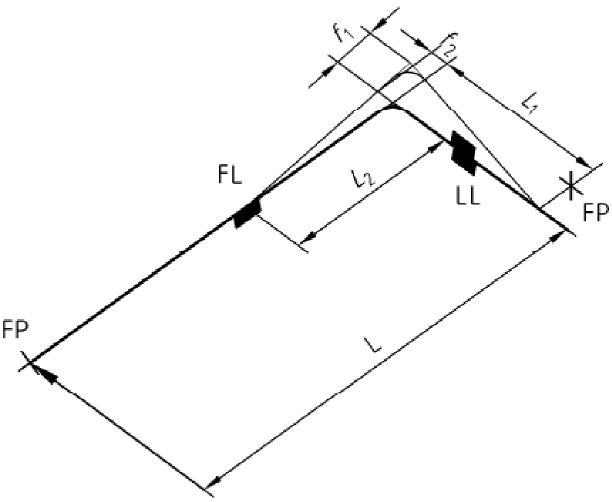

Figure 2.16: Specification of the required pipe lengths in bends [13]

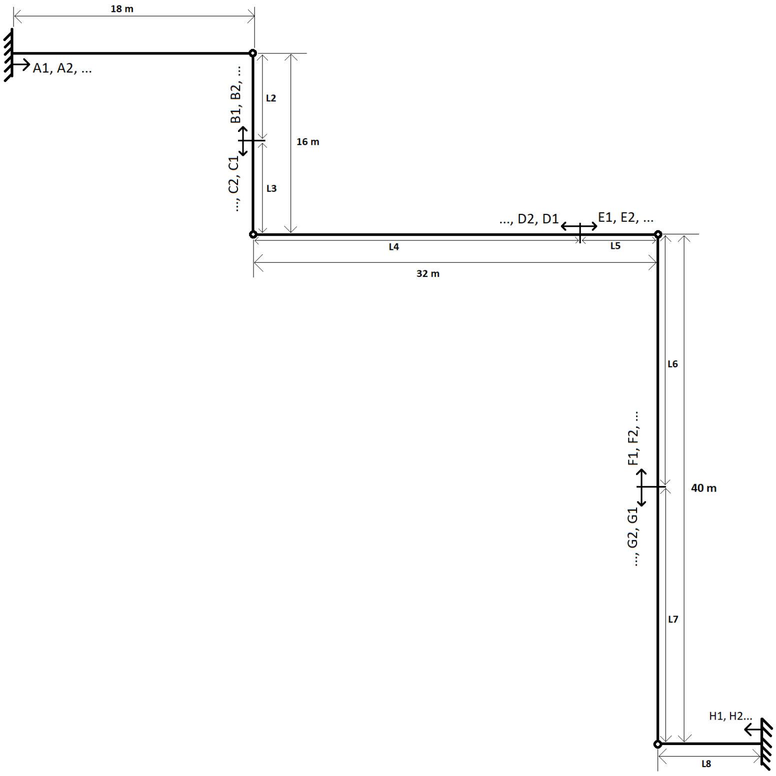

Figure 2.17: A schematic overview of the pipe design

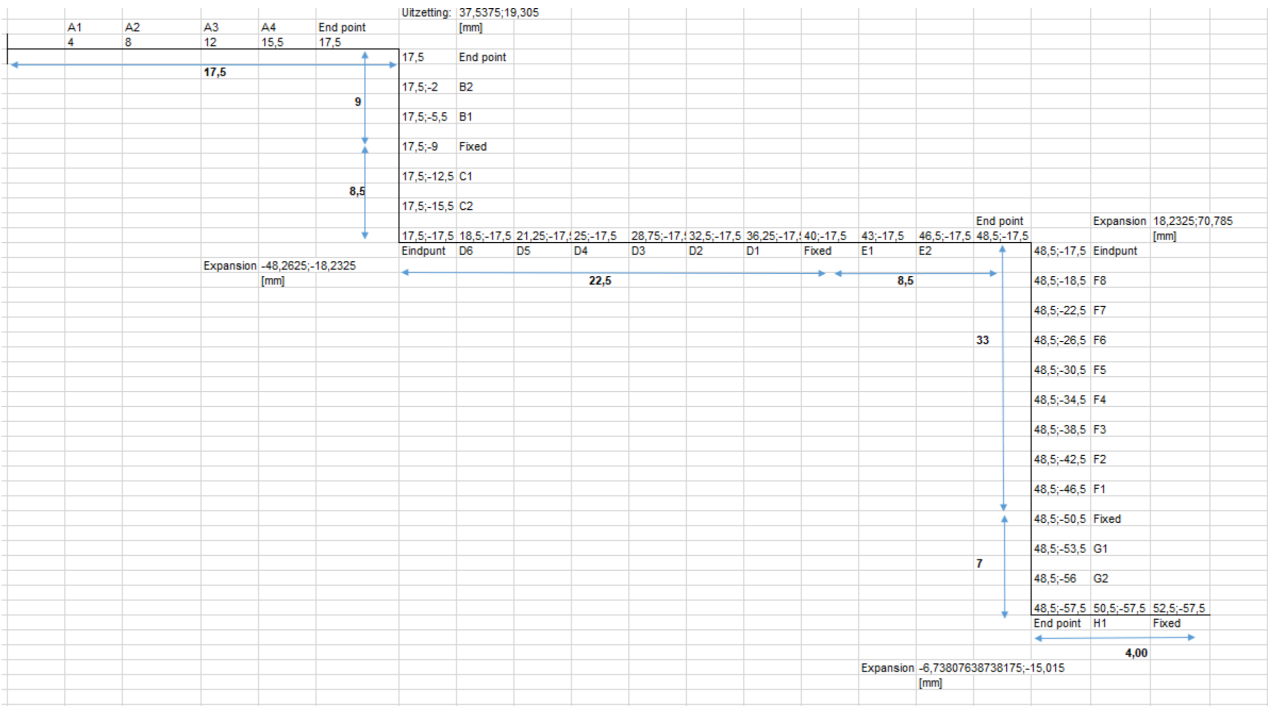

Figure 2.18: Locations of the supports for the design

2.7 Steam main components and condensate removal

A steam main installation has certain components and design rules which are needed for operation. In this section, the main components are described.

- Steam trap
  During start-up of the installation, condensate is formed and this needs to be removed from the pipes. For this purpose steam traps are used. These are automatic valves which can discharge condensates with negligible consumption of steam [3].

- Check valve
  Part of the steam trap set is a check valve. This is a non-return valve which ensures that flow is only allowed in one direction. These are placed to prevent reverse flow on shutdown, to prevent flooding and, to protect equipment which can be affected by reverse flow [3].

- Strainer
  Strainers are used to remove debris from circulating through the piping system. Rust and weld metal is usually left in the system and has to be removed before damaging equipment. In some steam traps this can be integrated into the steam trap as can be seen in figure 2.20 [3].

- Separator
  Wet steam can cause damage to turbines and reduce efficiency of the system. Separators are designed to remove moisture from the steam flow by diverting the flow direction vertically and by changing the direction numerous times. Suspended water droplets have greater inertia and mass and are collected on barriers whereas the steam flows around these barriers [3].

- Control or isolation valve
  Valves are used to reroute steam or water to facilitate plant shutdown, maintenance or equipment removal. They are used in fully open or closed position. Within this category linear and rotary movement valves are used [3].

In the figures 2.19 through 2.22 typical set-ups of condensate removal tracks are shown. Also the required fall of the steam and condensate track, which was previously mentioned in section 2.5, is shown. In the steam transport design the fall of both pipes has to be taken into account. During start-up condensate is formed which has to be removed. During operation, the steam at the steam distributor needs to be dry, and therefore no condensate can be formed in the steam main.
The formed condensate during start-up needs to be removed through steam traps. These have to be placed at 50 meter intervals. When passing through a steam trap, a pressure drop happens since the pressure in the condensate line is lower. At this lower pressure the saturation point of water is much lower. The condensate contains too much energy to remain completely liquid and some of it becomes steam. Condensate at 10 bar(a) has a saturation temperature of 179.9°C, whereas the saturation point of 1 bar (a) water lies at 100°C. The excess heat converts a part of the condensate to steam. This is shown in table 2.7. The mass fraction of flash steam is calculated using [23]:

$$\text{Flash steam mass}[\%] = \frac{h_{fluid1} - h_{fluid2}}{h_{evaporation2}} \tag{2.22}$$

Table 2.7: Flash steam mass and volume fraction

	1 bar(a)	3 bar(a)	10 bar(a)
Specific enthalpy saturated water	417.4	561.4	762.6
Latent heat of evaporation	2257.5	2177.6	2027.9
Flash steam mass %	0	6.4	15.3
Flash steam volume %	0	99.1	99.7

Using the density of the fluid and gas at 1 bar (a), the volume fraction can then also be calculated. Clearly the condensate line should be designed to transport the formed condensate in gaseous form. The formed condensate is transported along the steam main so multiple condensate removal sets can be joint in one condensate removal pipe. For economical reasons it can be interesting to recover the condensate. This reduces water charges, reduces fuel costs, and less water needs to be chemically treated [3]. Since the condensate only forms during start-up when the superheated steam heats up the pipes, which is planned to be only two times per year, the formed condensate in the long distance steam transport is considered to be negligible. This condensate is transported to a flash vessel where the flash steam can be vented and the fluid can be cooled after which it can be discharged to the sewers. The steam which is sent to the steam distributor and enters the production process is collected in a condensate mix tank. Some of the condensate in this tank is used by the old gas boiler which can be used when the steam demand is too high for the biomass installation. The rest of the condensate is pumped back to the biomass installation along the same construction the steam transport runs along. There is one more pipe which needs to be transported back to the biomass installation building. This is the cooled feedwater pipe. This water has been treated to reduce the mineral content. It is transported to the biomass installation building where the mineral content is reduced even further after which it can be used in the steam process.

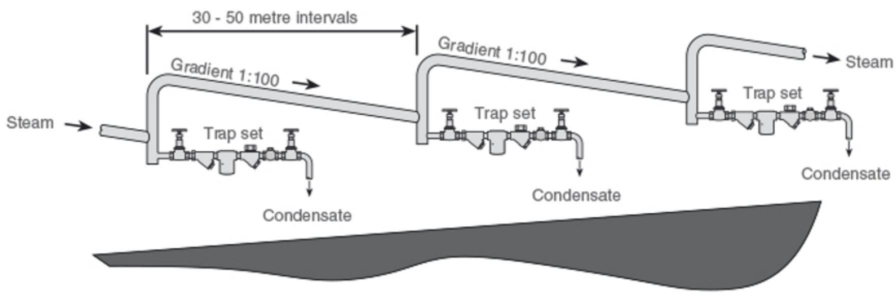

Figure 2.19: Typical steam main installation with an isolation valve, strainer, flow meter, thermodynamic steam trap, check valve and isolation valve on the condensate removal set [3]

2.8 Steam flow and resulting forces

The pressurized steam flow through the steam main exerts forces on the pipe. A tube or cylinder is subjected to a hoop and longitudinal stress in the wall. The hoop stress acts

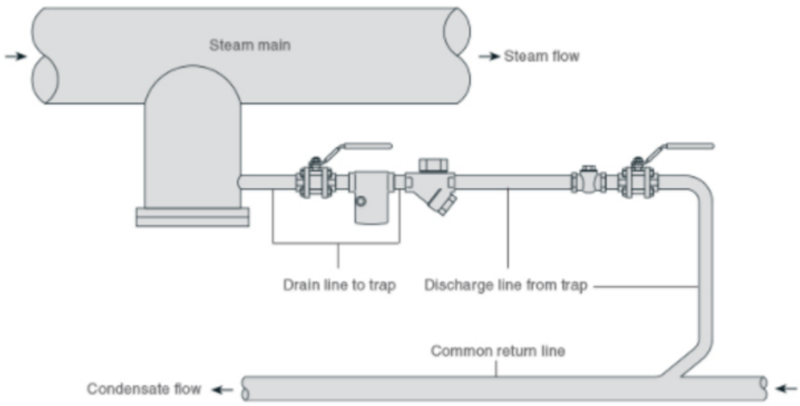

Figure 2.20: Condensate removal set with f.l.t.r. an isolation valve, flow meter, thermodynamic steam trap with integrated strainer, check valve and isolation valve [3]

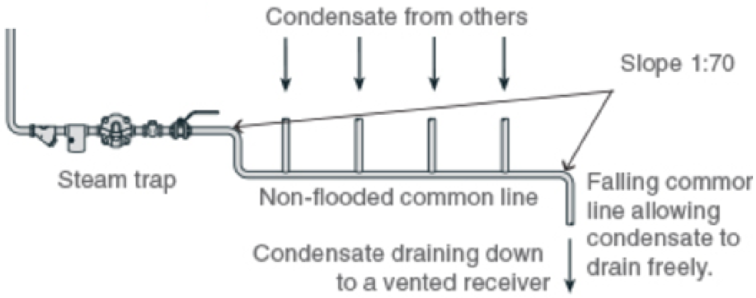

Figure 2.21: Condensate removal which drains freely because of the mounted angle. The used steam trap is a ball float steam trap [3]

circumferential along the tube and the longitudinal stress acts along the axis of the cylinder and they can be calculated as:

$$
\begin{aligned}
\sigma_h &= \frac{P \cdot r}{t} = \frac{1 \cdot 10^6 \cdot 0.08075}{0.0034} = 21.375 \text{ MPa} \\
\sigma_{xx} &= \frac{P \cdot r}{2t} = \frac{1 \cdot 10^6 \cdot 0.08075}{2 \cdot 0.0034} = 10.688 \text{ MPa}
\end{aligned}
\tag{2.23}
$$

If there is condensate formation, this slug flow through the pipes will exert a significant force on the construction as shown in figure 2.24. The slug force is calculated as [24]:

$$
\begin{aligned}
F_{axial} &= (\rho_{slug} \cdot A \cdot U^2)(1 - cos(\theta))DLF \\
F_{orthogonal} &= (\rho_{slug} \cdot A \cdot U^2)(sin(\theta))DLF \\
F_{axial} = F_{orthogonal} &= (931.8 \cdot 0.0205 \cdot 25.93^2) \cdot 2 = 25.6 \text{ kN}
\end{aligned}
\tag{2.24}
$$

For the Dynamic Loading factor (DLF) a factor of 2 has been found to be sufficient [24]. In the calculation the velocity of the slug is taken as the maximum velocity of the steam with the entire inner diameter surface filled with condensate. This will not happen for two reasons. At higher velocities the slug disperses and dispersed bubble flow is apparent as shown in figure 2.25. And when the steam condenses to reach densities as in equation 2.24

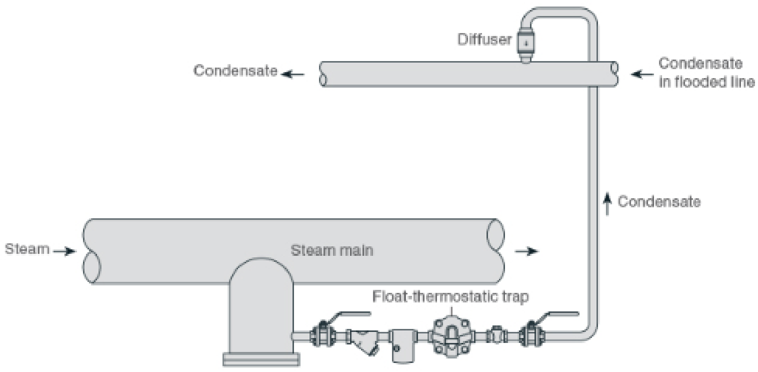

Figure 2.22: Condensate riser after steam trap operating by pressure difference [3]

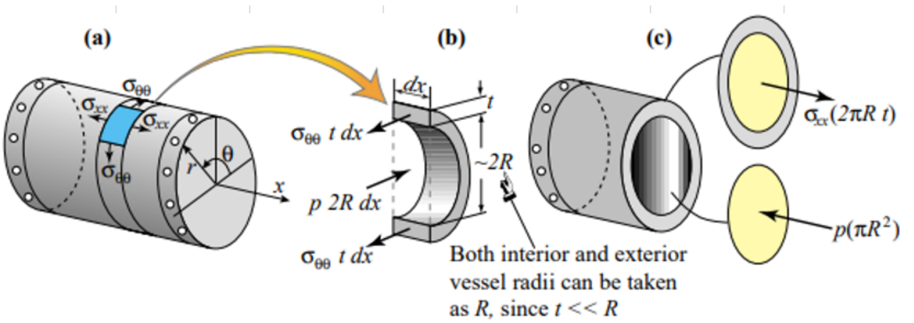

Figure 2.23: Circumferential and axial stress caused by internal pressure

the velocity drastically reduces. The calculation depicts an extreme for a situation which is definitely unwanted and will not happen for dry steam. It is therefore important that the temperature is checked constantly. Water hammer is known to destroy valves, pipes and entire supporting structures so this should be avoided at all cost. For higher working pressures, the velocity will be lower and the density will also be slightly lower. The resulting force will also be smaller. The elongation and stress of a pipe which is exposed to these forces is:

$$\delta = \frac{FL}{EA} = \frac{25.6 \cdot 10^3 \cdot 40}{200 \cdot 10^9 \cdot 0.001761} = 2.9 \text{ mm} \tag{2.25}$$

$$\sigma = \frac{F}{A} = \frac{25.6 \cdot 10^3}{0.001761} = 14.6 \text{ MPa} \tag{2.26}$$

The pipe material can withstand this stress and the elongation is also small compared to the thermal expansion. Still, the effect of this sudden force on the supports will be substantial and the outgoing temperature at the current pressure should be monitored constantly to minimize the risk of water hammer.

2.9 Wind load on structure

In the crossings between buildings the piping system is more prone to wind load. To estimate the forces which arise during a storm, an analysis is made. The longest crossing

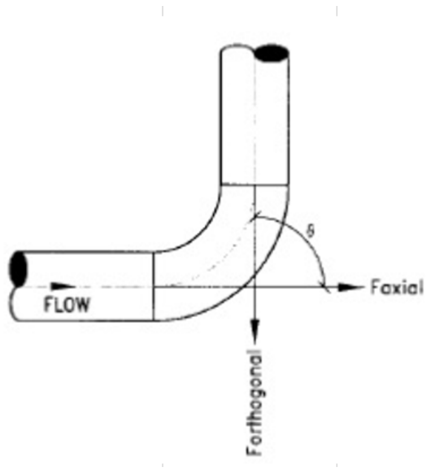

Figure 2.24: Axial and orthogonal force caused by fluid flow [24]

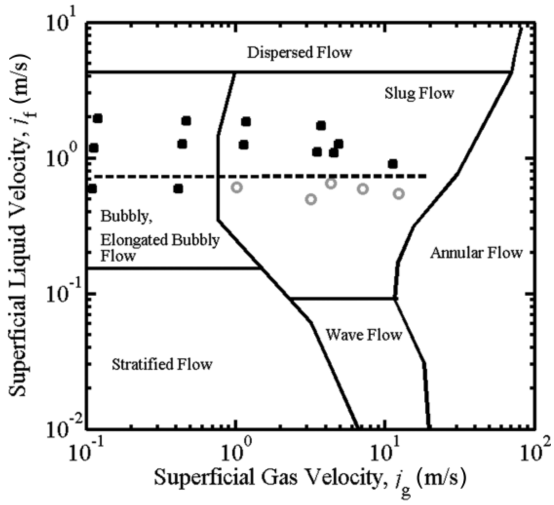

Figure 2.25: Two phase flow velocities and resulting flow pattern [25]

of around 18 meter is picked as the forces will be the highest here. The DN150 pipe with 100 mm insulation (diameter $= 0.168 + 2 \cdot 0.1 = 0.368$) here rests at a height of around 6 meters. The projected area of the pipe is the diameter times the length which equals $A_{ref} = 0.37 \cdot 18 = 6.66m^2$. The wind velocity v_b is taken as 28 m/s since when gusts of wind reach these speeds, the weather alarm is put in action [27]. The terrain roughness factor is dependent on the terrain category which in this case is category III. [28] describes category III as "*an area with regular cover of vegetation and buildings or with isolated obstacles with separations of maximum 20 obstacle heights (such as villages, suburban terrain permanent forest)*". An illustration of the categories is shown in figure 2.26. For this category corresponding values for z_0 and z_{min} are 0.3 m and 5.0 m, respectively [28]. The value for $z_{0,II}$ equals 0.05 m. Since the pipe is mounted at 6 meters which is higher than z_{min} this is the relevant reference height z_e. The terrain factor k_r is calculated as [29]:

$$k_r = 0.19 \cdot \left(\frac{z_0}{z_{0,II}}\right)^{0.07} = 0.19 \cdot \left(\frac{0.3}{0.05}\right)^{0.07} = 0.2154m \quad (2.27)$$

The roughness factor c_r at reference height z_e equals:

$$c_r(z_e) = k_r \cdot ln\left(\frac{z_e}{z_0}\right) = 0.2154 \cdot ln\left(\frac{6.0}{0.3}\right) = 0.6452 \quad (2.28)$$

The mean wind at the reference height, with $c_0 = 1.00$ as there are no hills/cliffs, is calculated as:

$$v_m = c_r \cdot c_0 \cdot v_b = 0.6542 \cdot 1.00 \cdot 28 = 18.07m/s \quad (2.29)$$

The wind turbulence intensity I_v at reference height z_e with turbulence factor k_l taken as 1.0 is calculated as:

$$I_v = \frac{k_l}{c_0 \cdot ln\left(\frac{z_e}{z_0}\right)} = \frac{1.0}{1.0 \cdot ln\left(\frac{6}{0.3}\right)} = 0.3338 \quad (2.30)$$

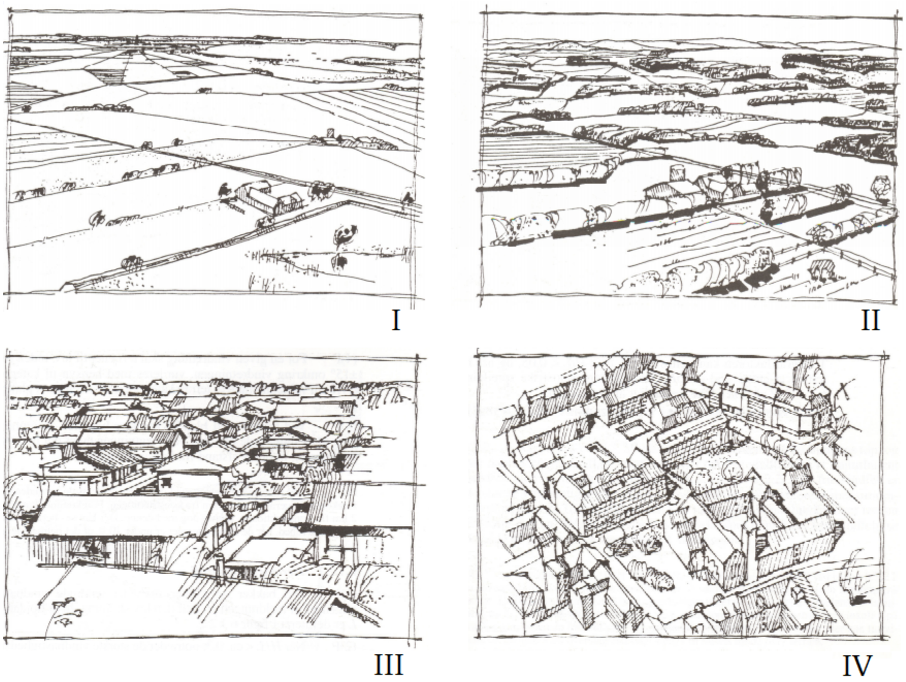

Figure 2.26: Illustrations for terrain categories [28]

The basic velocity pressure q_b is the pressure corresponding with the basic wind velocity v_b.

$$q_b = \frac{1}{2} \cdot \rho \cdot v_b^2 = \frac{1}{2} \cdot 1.25 \cdot 28^2 = 0.490 kN/m^2 \tag{2.31}$$

The peak velocity pressure q_p is defined as:

$$q_p = (1 + 7 \cdot I_v) \cdot \frac{1}{2} \cdot \rho \cdot v_m^2 = (1 + 7 \cdot 0.3338) \cdot \frac{1}{2} \cdot 1.25 \cdot 18.07^2 = 0.681 kN/m^2 \tag{2.32}$$

The exposure factor c_e is defined as the ratio between the peak and basic velocity pressure:

$$c_e = \frac{q_p}{q_b} = \frac{0.681}{0.490} = 1.3892 \tag{2.33}$$

The peak wind velocity v is calculated with the peak velocity pressure:

$$v = \left(2 \cdot \frac{q_p}{\rho}\right)^{0.5} = \left(2 \cdot \frac{0.681 \cdot 10^3}{1.25}\right)^{0.5} = 33 m/s \tag{2.34}$$

The Reynolds number can be calculated using the diameter, the peak wind velocity and the kinematic viscosity of air which is taken to be $15 \cdot 10^{-6} m^2/s$:

$$Re = \frac{b \cdot v}{\nu} = 0.8141 \cdot 10^6 \tag{2.35}$$

The slenderness follows from the aspect ratio of the structure and can be calculated from

$$\lambda = \frac{L}{b} + \frac{-0.3L}{b} \frac{L - 15}{50 - 15} = 47.398 \tag{2.36}$$

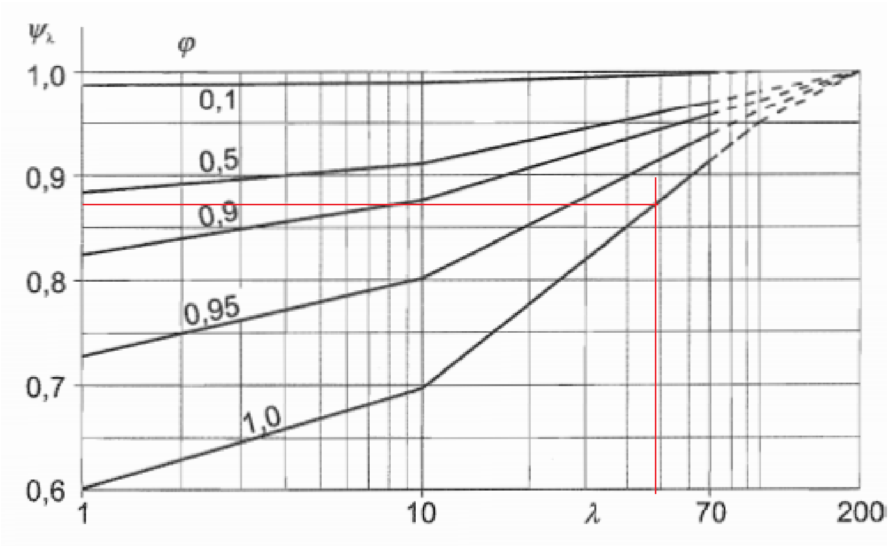

Figure 2.27: End-effect factor ψ_λ as a function of ϕ versus λ [29]

With a solidity ratio $\phi = A/A_c = 1$, which is a solid structure, the end-effect factor ψ_λ is be found from figure 2.27 to be 0.87.

The surface roughness k is assumed to be 0.2 which corresponds with galvanized steel which is similar to the insulation cover. The force coefficient for infinite cylinders can be calculated from:

$$c_{f,0} = 1.2 + \frac{0.18 \cdot log\left(\frac{10 \cdot k}{b}\right)}{1 + 0.4 \cdot log\left(\frac{Re}{10^6}\right)} = 1.2 + \frac{0.18 \cdot log\left(\frac{10 \cdot 0.2}{0.37}\right)}{1 + 0.4 \cdot log\left(\frac{0.8141 \cdot 10^6}{10^6}\right)} = 0.777 \tag{2.37}$$

For finite cylinders, the force coefficient can be calculated as:

$$c_f = c_{f,0} \cdot \psi_\lambda = 0.777 \cdot 0.87 = 0.68 \tag{2.38}$$

The structural factor $c_s c_d$ is chosen as 1.000 which is appropriate for buildings or structures at a height below 15 meter. Now all components are available to calculate the total wind force and effective wind pressure on the structure:

$$\begin{aligned} F_w &= c_s c_d \cdot c_f \cdot q_p \cdot A_{ref} = 1.000 \cdot 0.68 \cdot 0.681 \cdot 10^3 \cdot 6.66 = 3.08kN \\ w_{eff} &= F_w/A_{ref} = 3.08/6.66 = 0.463kN/m^2 \end{aligned} \tag{2.39}$$

This force was used to design the supporting structure for the bridge and as validation for the bridge design.

Table 2.8: My caption

	Slide guide	Swivel hanger
Advantages	In-plane motion Large deflection Both supporting and suspending	Cheap Simplistic design Small footprint Simple mount
Disadvantages	Expensive Many parts Large footprint	Vertical movement when the pipe expands axially Possible swinging

2.10 Support comparison

For the supports allowing expansion a short comparison is done between the supports shown in figures 2.14 and 2.15. The difference between the two designs is whether the supporting structure is supporting from below or above (slide guide allows for both). The main supporting structure is similar for both designs so the comparison is done between the two pipe supports assemblies. For the costs, the inventory of support system supplier Müpro [38] is used. Several options for sliding shoes and swivel hangers are available. What is apparent is that slide guides become really expensive for increasing load capacity. Whereas a 1 kN slider costs 11 euros, 1.75 kN costst 18 euros and 4 kN is already 50 euros per slider. Comparing this to the swivel hangers, The small deflection hangers can support up to 2.4 kN and cost less than 2 euros. The larger deflection hanger can hold up to 3 kN and is around 3 euro per hanger. The main steam pipe can exert up to 2.1 kN per support as shown in table 2.5. More advantages are shown in table 2.8. For the sliding guide, the assembly in figure 2.14 was just an idea which has not been found at manufacturers. By combining two linear rails, planar motion is acquired. By not being able to purchase this part from a manufacturer as is, a lot more attention has to be paid to the design to ensure a fail-safe part. This will require time and the resulting assembly incorporates many parts and is bulky which is also not preferred since several pipes have to be mounted alongside one another. The swivel hanger does have many upsides. It is available, can allow for large deflection, and several pipes can be mounted next to each other. The vertical movement when the pipe is expanded can be analyzed. The largest expansion is around 80 mm with an adjacent expansion of 20 mm. With a threaded rod of 300 mm in between, the end point will move 11.6 mm vertically. The pipe will rise this vertical motion over the length of the adjacent pipes. The shortest pipe will have a steeper angle. This angle can be calculated by taking $tan^{-1}(11.6/10000) = 0.066°$. The 10000 is the length of the adjacent pipe. This angle does not pose a problem. Based on the up- and downsides, the swivel hanger is preferred. It is easier to implement at a fraction of the cost.

Chapter 3

Design

3.1 Buildings and mounting points

First the buildings, to which the piping will be supported, are modelled, based on blueprints supplied by the customer shown in figure 3.1. The accompanying model is shown in figure 3.2. The vertical dimensions were first estimated using photographs of the location and later these dimensions were adjusted using measured distances. The structural parts to which support structures can be mounted on the barn were also measured. This is shown in figure 3.3. Since these distances are greater than the largest allowable support interval mentioned in chapter 2.4, additional supports need to be mounted besides just at every barn support. Therefore additional steel beam profiles are mounted to the barn support structures. The design is shown in figure 3.4. The translucent part is corrugated sheet which has to be cut away at some points to allow for the assembly to the barn structure. The steal beams allow for mounting of the steam pipes at arbitrary positions in between the barn support structures. The dimensions of the steel profile are determined using FEM simulations with the displacement as limiting factor. Similar to chapter 2.5, an angle of deflection of 0.1° is set as the upper limit. This resulted in a 140x140 beam profile. For the set of return pipes also some support needs to be available. This can be mounted from structural steel t-shapes which are welded to the beam profiles. These are placed such that water won't collect in the profile. FEM analyses on the stress and displacement are shown in figures 3.6 through 3.8.

By using hangers, the pipes will hang below the steel mounts. The height of the steel mounts determine the height of the steam pipes which in turn determine the clearance height for traffic. The steel mounts should therefore be placed as high as possible to maximize the overall clearance height. This is limited at the southern wall of the barn by the gutter which is mounted at 5.7 meter. The steel mounts are mounted just below this gutter and the steam pipe has to hang at the angle mentioned in chapter 2.5. This poses a problem since the steam pipe including insulation will hang below 4 meters at the end of the steam transport. $4.96 - tan(0.82°) * 72 = 3.93$. The initial height is shown in figure 3.5. The height could be increased slightly, but the margin is too small which does not allow for small errors in construction. Therefore a solution is sought which will be explained in chapter 3.3.

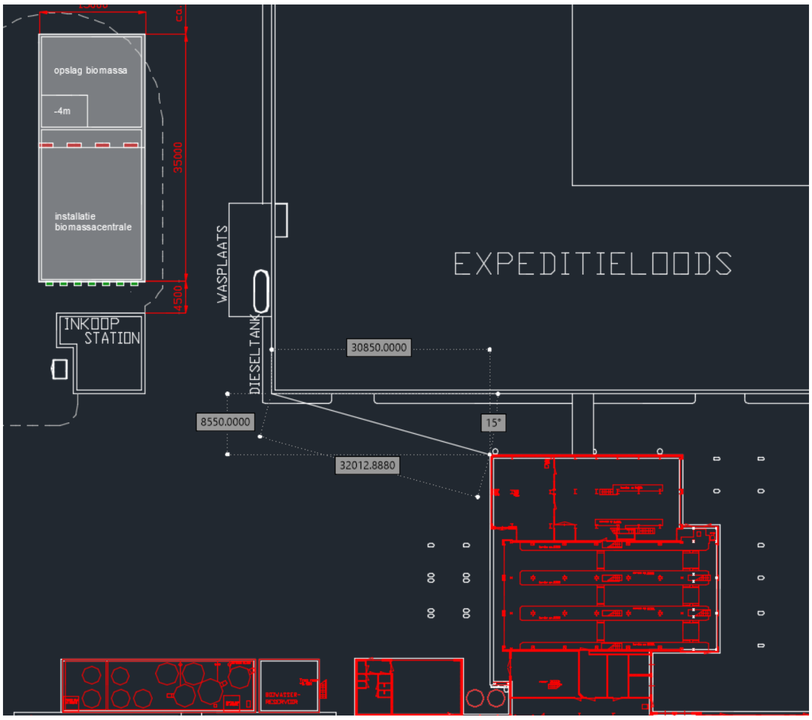

Figure 3.1: Blueprint of the company grounds supplied by the customer showing the distance between barn and production hall

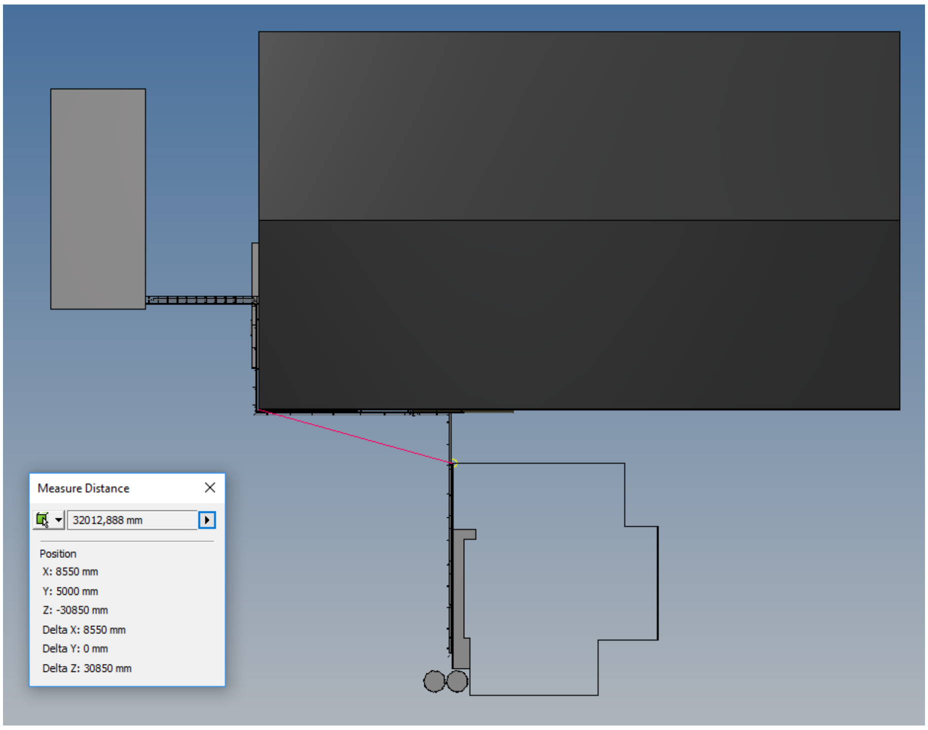

Figure 3.2: Modelled buildings in Autodesk Inventor with the distance from figure 3.1 shown in Delta X and Delta Z

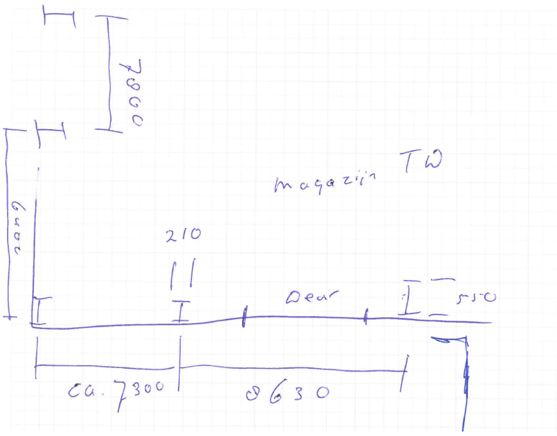

Figure 3.3: Distances between the steel beam profiles and the steal beam profile size

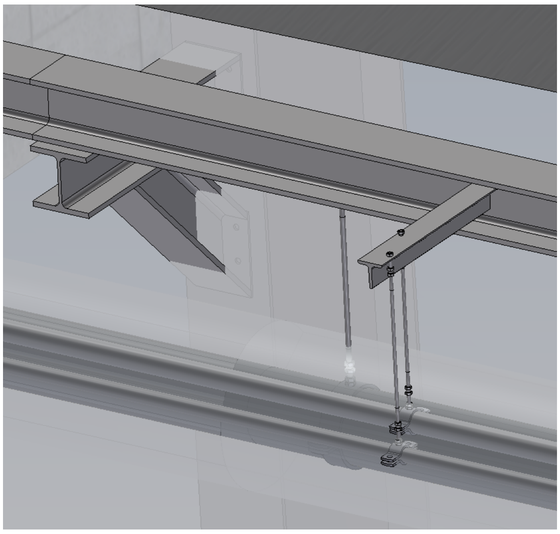

Figure 3.4: A steel support assembly bolted to the building

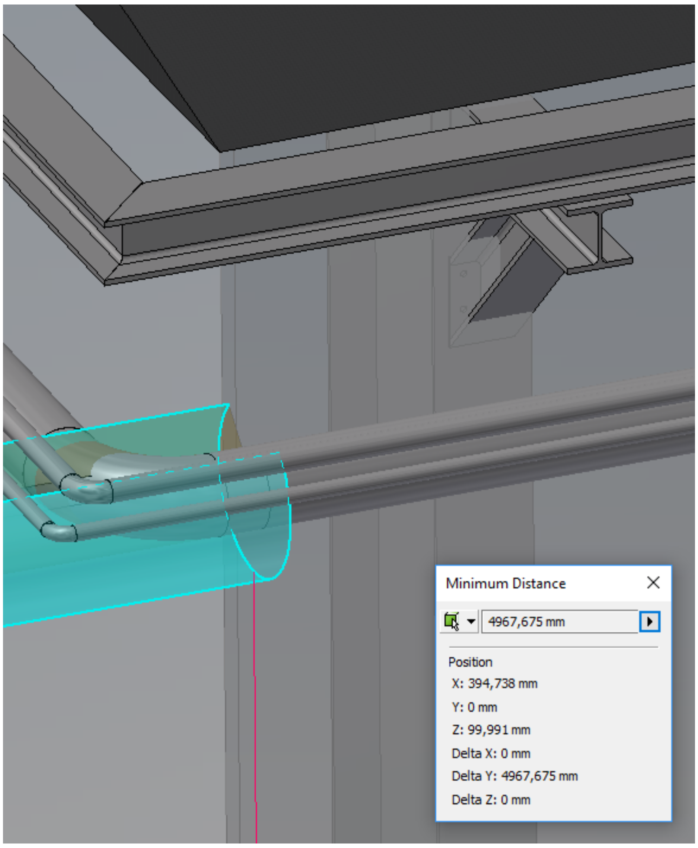

Figure 3.5: Height of the insulation and the steel mount mounted just below the gutter height (black roof)

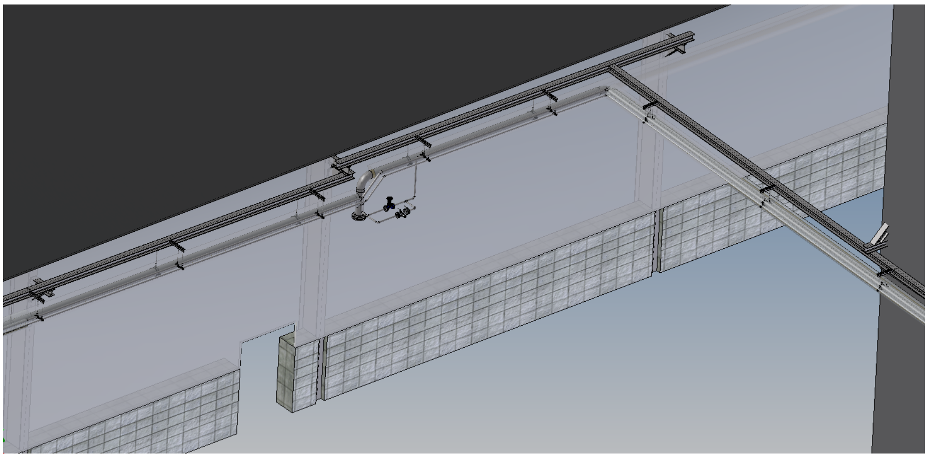

Figure 3.6: Corner at the crossing between the barn and the production hall

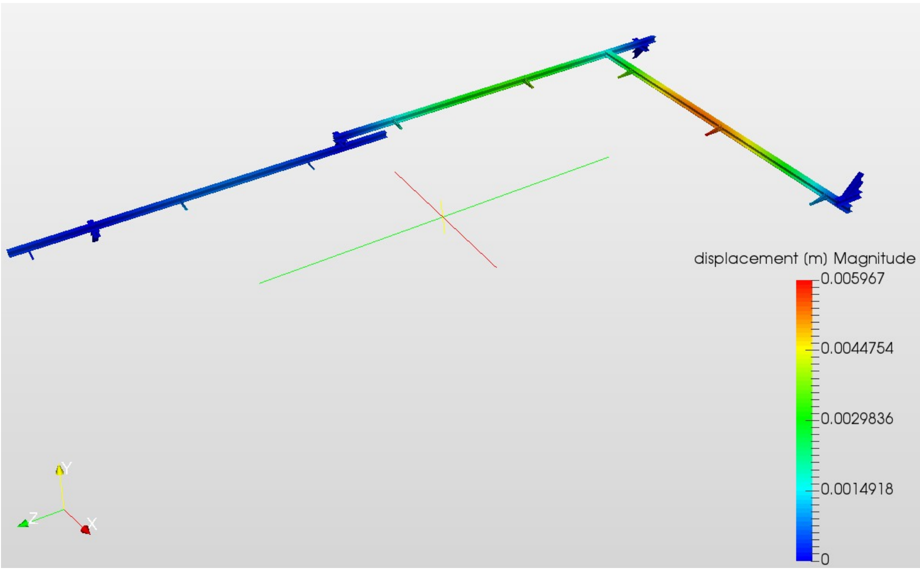

Figure 3.7: FEM displacement analysis on the crossing and mounts

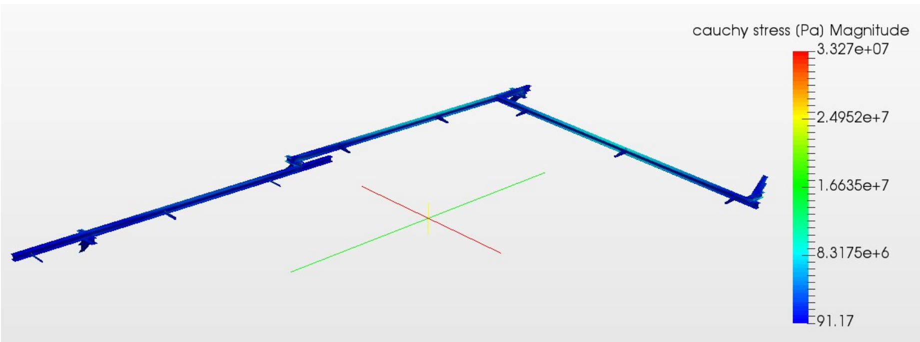

Figure 3.8: FEM stress analysis on the crossing and mounts

3.2 Supports

Based on the cost and flexibility, hanging supports are chosen. These are based on commercially available swivel hangers which make use of two spheroid joints. The hanger is shown in figure 3.9 and the assembly is shown in figure 3.10. Depending on the model, the maximum deflection differs from 12 to 22° of the vertical axis. For the steam main, the maximum deflection is around 80 mm. To facilitate this movement while remaining within the 12 or 22° domain, the rotating part of the threaded rod must be either 376 or 198 mm. By reducing this length, the effective height above the ground is increased, which in turn gives more margin for traffic underneath. Therefore for the steam pipe, the greater deflection swivel hanger is preferred. The smaller condensate removal pipe and the return pipes for condensate and purified water have a smaller temperature difference and will expand less. The condensate return pipe will expand at most around 30 mm. The required threaded rod needs to be 141 or 74 mm long for either the 12 or 22°. The lighter pipes have less deflection and the load is also less. The small deflection swivel hanger is sufficient for these pipes so these are used. In the figures 3.11 through 3.13 the modelled swivel hangers are shown in rest and in deflected state. The structural tee to which the hangers are mounted, is mounted this way to prevent water from collecting in recesses during rain.

Figure 3.9: Swivel hanger allowing for axial and lateral movement [30]

Figure 3.10: An example of a swivel hanger assembly [30]

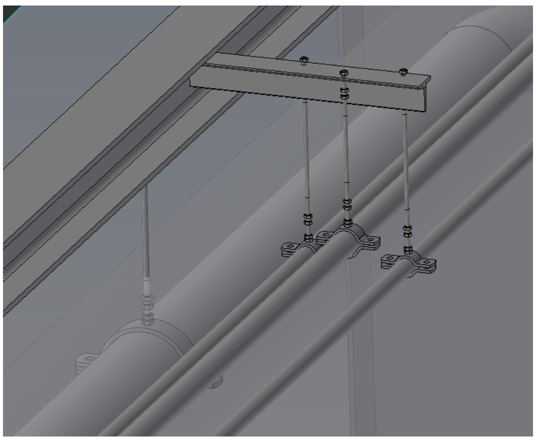

Figure 3.11: Swivel hangers for all four pipes

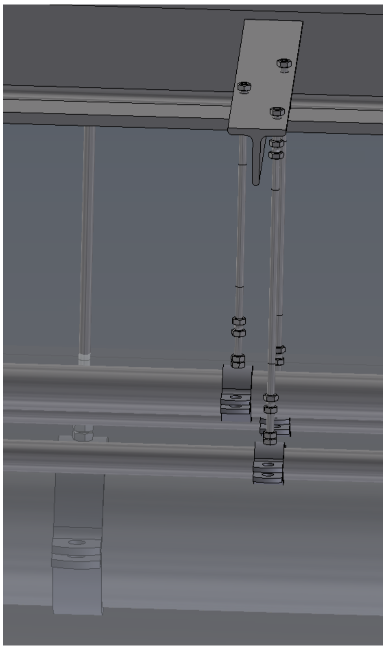

Figure 3.12: Swivel hangers in the relaxed state

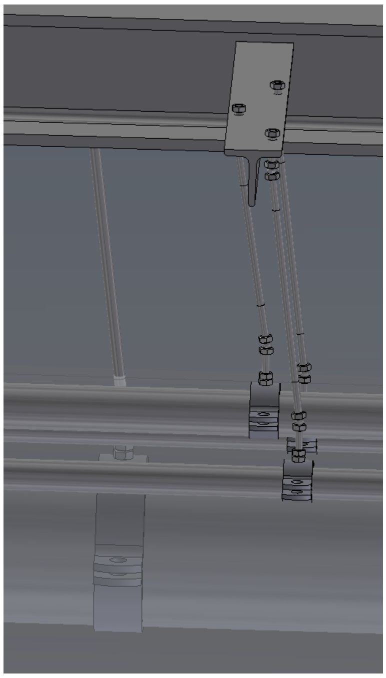

Figure 3.13: Swivel hangers in the deflected state

3.3 Condensate removal and riser

During start-up condensate will form when the steam heats up the pipes. This has to be removed quickly from the system for performance. This is done using steam traps which are some kind of automatic valves which can differentiate live steam from condensate and non-condensable gases such as air and CO_2. For superheated steam two types of traps are advised which are bimetallic steam traps and inverted bucket style steam traps [31].

A bimetallic steam trap works by using several elements of dissimilar metal strips welded together [3]. When heated the elements will deflect and close a valve. A spring washer is incorporated to increase the required force at higher pressures as to follow the saturation curve more closely [3]. As can be seen in figure 3.14, the bimetallic steam trap requires condensate below saturated temperature to open but will be closed when confronted with steam. When increasing the steam temperature the seal will be stronger which makes it a good candidate for superheated steam. This quality combined with the ability to handle large start-up loads, makes this type of steam trap a good candidate.

The inverted bucket steam trap consists of an inverted bucket which is attached by a lever and a valve. When the entire trap is filled with condensate the bucket will be held down and the valve will be open. Incoming steam, being less dense than steam, will remain in the top portion of the bucket and it will lift the bucket and close the valve. When superheated steam enters the trap chamber, this could possibly cause the trap to lose its water seal. Therefore a check valve in front of the trap is required.

The bimetallic steam trap has better start-up conditions which are important for this application. It has better air venting capabilities and because the trap requires condensate below the saturation temperature, fewer flash steam is formed. Therefore this steam trap is chosen. A sufficiently large drip leg is required to refrain the condensate from backing

up into the system. Having superheated steam run through the pipes will evaporate the remaining condensate in the drip leg. An increase in pressure will increase the saturation temperature so the pipes will have to heat up as well which could cause condensate to form. Since the travelling steam is superheated and the pipes are also at this temperature during operation this will not pose a problem if the pressure increase is gradual so the pipes have time to heat up without forming condensate.
A substantial pressure drop occurs over the steam trap since the condensate removal end in a flash vessel at ambient pressure. This pressure difference aids to force the condensate through the steam trap along with the formation of flash steam. This pressure difference can be used to gain height which was previously mentioned to be a problem. This principle is shown in figure 2.22. According to [3] the back pressure is increased from a rising discharge line by 0.1 bar(g) per meter. Since the minimum operating pressure is 3 bar(a) and the condensate line pressure is 1 bar(a) this will not pose a problem, because only 1 meter of height gain is required. The resulting riser is shown in figures 3.15 and 3.16. In parallel to the steam trap is also a bypass valve in blue. All sorts of debris are in the pipes such as weld metals, rust and other jointing compounds. These will mostly be removed from the system during start-up. It is undesirable to pass all this debris through the steam trap and therefore a bypass valve is introduced which will close when the pipes are at around 80-90°C at which point the steam trap will take over. Based on the pressure and temperature conditions the used isolation valve is chosen to be a ARI-FABA-Plus 12.046 PN16 DN25. This is a standard isolation valve with sufficient capacity. Behind the steam trap is a check valve which will prevent the condensate which is in the rising leg from running back into the system.

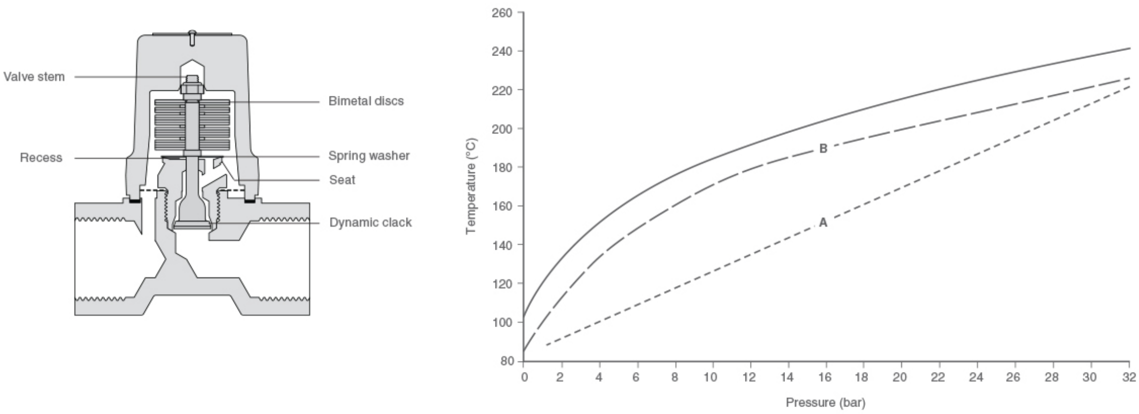

Figure 3.14: Section view and operating temperature of a multi-leaf(B) bimetallic trap [3]

3.4 Return pipes

The return pipes transport both condensate which is formed and collected from the production processes and the water supply to the biomass plant. Pumps are used to transfer these fluids from the old boiler room to the biomass plant and therefore these can be run at an inclining angle along the steam main support structure. As mentioned previously these are mounted using hangers which allow for the small expansion relative to the steam main. The pipes have also been mounted at such a distance from one another so the expansion won't

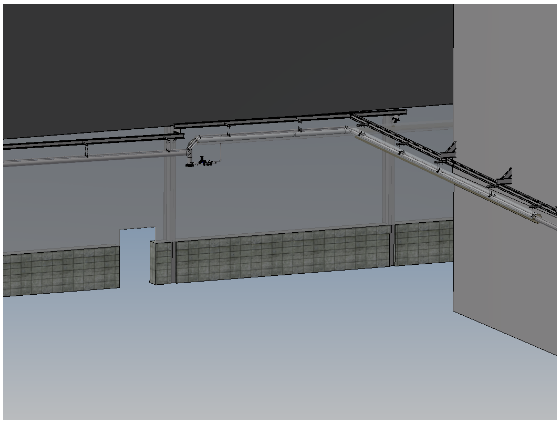

Figure 3.15: Location of riser and steam trap

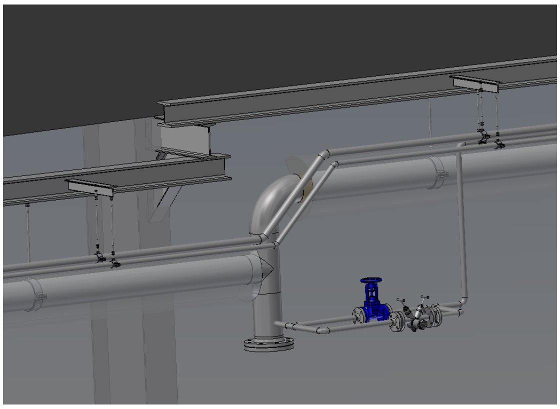

Figure 3.16: Riser and steam trap in the steam main and return lines

cause the pipes to collide. Having the pipes mounted outside will expose them to ambient temperatures. This will cause the pipes to lose heat. When the ambient temperature drop below zero degrees Centigrade, the fluids in the pipes might freeze. This is no issue for the condensate return line when the biomass plant is in production, but for the water supply pipe which is at 10°C this can become a problem. Also when the plant is on standby and some water is left in any of the pipes, this can freeze over. For this purpose heating cable or heat trace cable is run along all of the pipes. This is a cable which gets hot when current is run through. When ambient temperatures drop below 0° the heat tracing is used for the water supply pipe and for the condensate removal pipe. When the plant is in standby, all pipes are heated with heat trace cables. It is therefore important that the heat supplied by the trace cable is sufficient. In other words that the heat lost, which is radiated by the pipes to the surroundings by the temperature difference, is smaller than the heat gain by the heat trace cable. This is checked for all pipes similar to the heat loss calculation in chapter 2.2. With heat trace cables with a power of 10 W/m this resulted in insulation thicknesses for the condensate removal, water supply and condensate return pipe of 20 mm. Figure 3.17 shows the scale of the pipes and insulation.

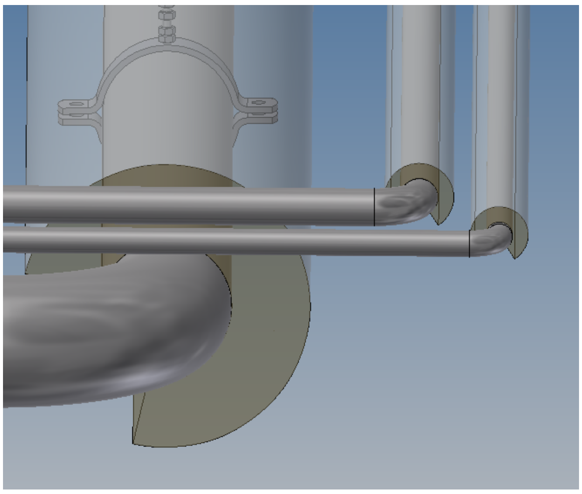

Figure 3.17: Three supported pipes with some insulation for size comparison

3.5 Redesign bridge

Using standard profiles a design was made in chapter 2.5. There were however some flaws. For one thing the bridge was not designed for production. The bridge had to be welded into one assembly and then had to be transported to the customer. This truss structure will be placed outside and so has to be protected from rust. This means that there should be no crevices or recesses where water can stay behind. The u steel profiles on the lower tier have been rotated so water won't be able to trapped here. Also the vertical structures have been moved to the outside of the structure. For corrosive protection, the structure should be galvanized after which no welds are performed or these parts will be unprotected. This hot-dip galvanization is done by lowering it into a bath of molten zinc. This bath allows for parts of size 8.25x2.00x2.90 meter [34] and therefore the entire bridge won't be able to fit in one piece. Also this would be inconvenient for transportation this way. The bridge is therefore split into three parts which can be galvanized individually and then be connected using some bolt connections. It is important that holes for bolt connections are drilled according to ISO 20273 [32] while keeping the effects of the galvanization process in mind. [33] states that an additional 1.5 mm should be added to the diameter which was defined by the ISO 20273. This concerns the holes which are made for the bridge bolt connection and the M8 and M12 hangers. ISO 20273 states that for a middle tight fit, for M8 a 9 mm hole should be used and 13.5 mm for M12. So 10.5 mm and 15 mm holes should be drilled in the concerning parts. Figures 3.18 and 3.19 show the division and the bolted connection.

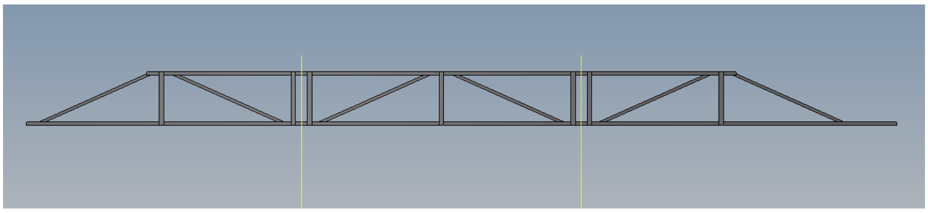

Figure 3.18: The bridge divided into 3 parts of 5.9, 6.0 and 6.8 meter from left to right

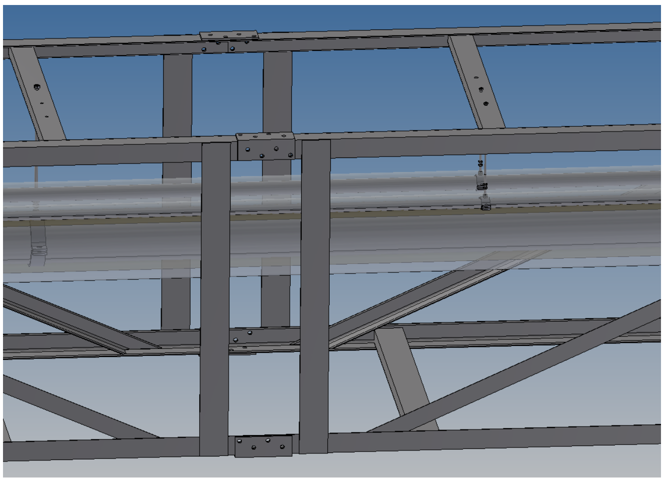

Figure 3.19: The division in the bridge connected with plates and M12 bolts

3.6 Overview

All parts are shown in figure 3.20. This assembly contains the steam main which runs from the biomass installation to the old boiler room. Halfway the riser and steam trap branch off and continue to the old boiler room. A condensate return line and treated water line run back from the old boiler room to the biomass installation along the same support structure. The doors and traffic entrances are kept accessible. Also the platform mounted to the barn is kept accessible. The lowest point of the bridge is still 2.36 meter above the platform. The resulting heights are sufficiently high to allow for traffic passing underneath. The bridge is mounted under a downward angle of 1° with a lowest point at 4.89 meter. The steam main at the barn truck entrance hangs 4.76 meter above the ground. After the riser, the pipes cross-over to the production hall and the lowest point is the insulation of the steam main which hangs at 4.83 meter. Finally the lowest point of the pipe system where traffic passes underneath is at the loading dock next to the old boiler room. The insulation hangs at a height of 4.37 meter at the far right truck entrance. The height requirement of 4 meters is met. The pipes and supports are mounted at a downward angle of 1° to allow for gravity assisted flow of steam and condensate. This is slightly higher than required to allow for imprecise structural placement and slight sagging of the structures. Mounting structures are welded together and mounted to the existing structures using bolts shown previously.

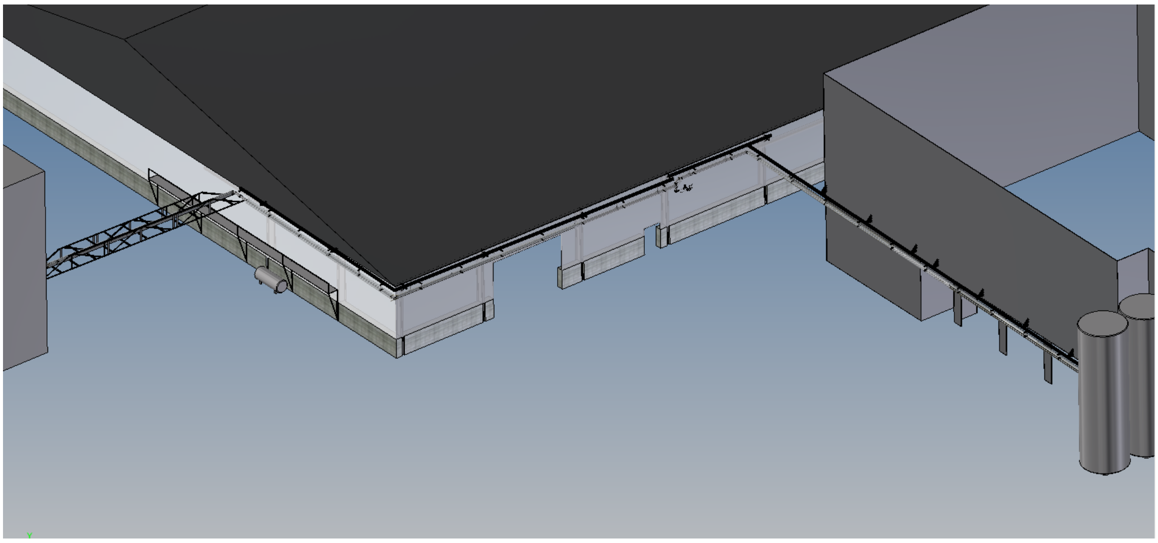

Figure 3.20: Overview of the steam transport system

Chapter 4

Required drawings and norms

4.1 Pressure Equipment Derivative

When designing and manufacturing pressure equipment this could potentially be hazardous. The PED(Pressure Equipment Directive) offers Essential Requirements to which the designing, installation, testing and running of the pressure equipment must comply. This ensures that the system is adequately checked and that the right components are used. Using a PED calculator the category of the steam pipe is determined. The steam transport pipe is designed for 10 bar(g) and the dimension of the pipe is DN150. This puts the pipe in category I. The result of the PED calculator is shown in figures 4.1 and 4.2. The other pipes outside have a smaller diameter and lower pressures and fall in Art. 4, par. 3 which equates to sound engineering practice. For the Netherlands also an Inspection before commissioning should be done if the pipe diameter DN size times the design pressure bar(g) is larger than 5000. Since this is not the case, this inspection is not mandatory.

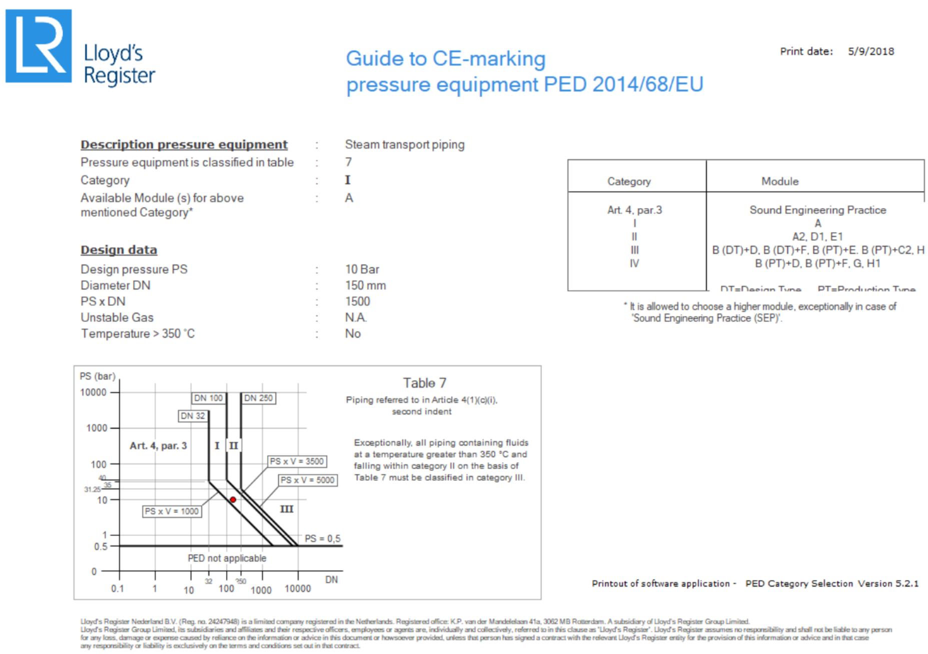

Figure 4.1: Lloyd's Register PED calculator result

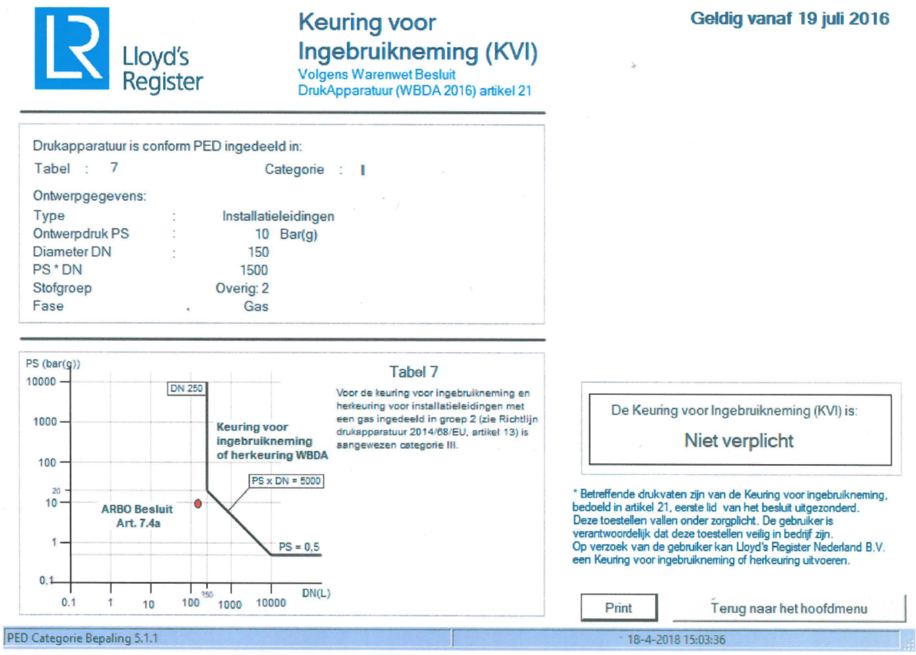

Figure 4.2: Lloyd's Register Inspection before Commissioning Result

4.2 Welding checks

Since the steam transport pipe is in category I some inspection has to be done on the weld quality of all circumferential, branch fillet and seal welds. The material P235GH falls in material group 1.1 according to ISO 15608 [35]. On the steam pipe there are mainly circumferential welds connecting elbows and lengths of pipe. At the steam trap location there is a branch weld which connects the drainage leg to the pipe leading to the steam trap and bypass. Consulting EN 13480-5 Metallic Industrial Piping - Part 5: Inspection and testing gives the following table which is shown in figure 4.3. For the material group and category I it shows that all circumferential welds should be checked visually. And for every welder at least 5% of the welds have to be checked using Radiographic Testing or Ultrasonic Testing. For this purpose welding drawings have to be made which denote every weld and an accompanying sheet with the weld number, welder, and whether the weld was found to be acceptable after visual and possibly non-destructive testing.

4.3 Hydrostatic testing

As mentioned in chapter 2.4, the pressure piping has to be tested hydrostatically. As mentioned in chapter 2.3, the yield strength at the design temperature of 220°C equals 162 MPa. The required test pressure for the superheated steam pipe is described in EN 13480-5 as:

$$P_t = max\left[1.25 \cdot p_c \cdot \frac{R_{eH}}{R_{eH,T}}; 1.43 \cdot p_c\right] = max\left[1.25 \cdot 10\ bar(g) \cdot \frac{235}{162}; 1.43 \cdot 10\ bar(g)\right] = 18.1\ bar(g) \tag{4.1}$$

<table>
<tr><th rowspan="3">Material group [a]</th><th rowspan="3">Category</th><th rowspan="2">All welds</th><th colspan="3">Circumferential welds</th><th colspan="6">Branch welds</th><th colspan="2">Socket/fillet welds</th><th colspan="2">Seal welds</th></tr>
<tr><th colspan="2">Surface testing</th><th>Volumetric testing [b]</th><th colspan="3">Surface testing</th><th colspan="3">Volumetric testing [b,k]</th><th colspan="2">Surface testing</th><th colspan="2">Surface testing</th></tr>
<tr><th>VT %</th><th>e_n mm</th><th>MT/PT [c] %</th><th>RT/UT %</th><th>Branch diameter</th><th>e_n [h] mm</th><th>MT/PT [c] %</th><th>Branch diameter [i]</th><th>e_n [h] mm</th><th>RT/UT %</th><th>e_n mm</th><th>MT/PT %</th><th>e_n mm</th><th>MT/PT %</th></tr>
<tr><td rowspan="3">1.1, 1.2, 8.1</td><td>I</td><td rowspan="3">100</td><td colspan="2" rowspan="3">0 (5) [f,g]</td><td rowspan="2">5 (10) [g]</td><td colspan="2" rowspan="3">All</td><td rowspan="2">0 (5) [f,g]</td><td colspan="2" rowspan="2">All</td><td rowspan="2">0</td><td rowspan="3">All</td><td rowspan="2">0</td><td rowspan="3">All</td><td rowspan="2">0</td></tr>
<tr><td>II</td></tr>
<tr><td>III</td><td>10</td><td>10</td><td>> DN 100</td><td>> 15</td><td>10</td><td>10</td><td>10</td></tr>
<tr><td rowspan="6">1.3, 1.4, 1.5, 2.1, 2.2, 4.1, 4.2, 5.1, 5.2, 8.2, 8.3, 9.1, 9.2, 9.3, 10.1, 10.2</td><td rowspan="2">I</td><td rowspan="6">100</td><td>≤ 30</td><td>5</td><td>10</td><td colspan="2" rowspan="4">All [e]</td><td rowspan="6">10 (25) [g]</td><td colspan="2" rowspan="4">All</td><td rowspan="4">0</td><td rowspan="4">All [e]</td><td rowspan="4">10</td><td rowspan="4">All [e]</td><td rowspan="4">5</td></tr>
<tr><td>> 30</td><td>10</td><td>10</td></tr>
<tr><td rowspan="2">II</td><td>≤ 30</td><td>5</td><td>10</td></tr>
<tr><td>> 30</td><td>10</td><td>10</td></tr>
<tr><td rowspan="2">III</td><td>≤ 30</td><td>5</td><td>10 (25 [d]) [f,g]</td><td colspan="2" rowspan="2">All</td><td rowspan="2">> DN 100</td><td rowspan="2">> 15</td><td rowspan="2">10</td><td rowspan="2">All</td><td rowspan="2">25</td><td rowspan="2">All</td><td rowspan="2">25</td></tr>
<tr><td>> 30</td><td>10</td><td>10 (25 [d]) [f,g]</td></tr>
<tr><td rowspan="6">3.1, 3.2, 3.3, 5.3, 5.4, 6.1, 6.2, 6.3, 6.4, 7.1, 7.2</td><td rowspan="2">I</td><td rowspan="6">100</td><td>≤ 30</td><td>10</td><td>25</td><td colspan="2" rowspan="6">All</td><td rowspan="4">25</td><td rowspan="6">> DN 100</td><td rowspan="6">> 15</td><td rowspan="4">25</td><td rowspan="6">All</td><td rowspan="4">25</td><td rowspan="6">All</td><td rowspan="4">10</td></tr>
<tr><td>> 30</td><td>25</td><td>25</td></tr>
<tr><td rowspan="2">II</td><td>≤ 30</td><td>25</td><td>25</td></tr>
<tr><td>> 30</td><td>25</td><td>25 (25 [d]) [f,g]</td></tr>
<tr><td rowspan="2">III</td><td>≤ 30</td><td>100</td><td>25 (100) [f,g]</td><td rowspan="2">100</td><td rowspan="2">100</td><td rowspan="2">100</td><td rowspan="2">100</td></tr>
<tr><td>> 30</td><td>100</td><td>25 (100 [d]) [f,g]</td></tr>
</table>

[a] Material group, see CEN ISO/TR 15608.
[b] For the selection of the appropriate NDT-method for volumetric testing, see 8.4.4.3.
[c] See 8.4.4.2.
[d] Additional testing for transverse defects from weld surface (see EN ISO 17640:2010, testing level C).
[e] Only if PWHT has been carried out.
[f] Value in brackets applies to piping where creep or fatigue is the controlling factor in design.
[g] Value in brackets applies to piping with pneumatic pressure test with 1,1 times the maximum allowable pressure.
[h] e_n is the nominal thickness of the branch pipe at the weld (see W3, W3.1 and W6 in EN 13480-4:2017, Figure 9.14.4-1 and Figure 9.14.4-2).
[i] For parts without DN designation d_i > 120 mm may be used instead of DN > 100.
[k] Volumetric testing is required if both criteria (branch diameter and nominal thickness) are satisfied.

Figure 4.3: Extent of testing for various welds

This pressure has to be kept for at least 30 minutes during which the entire piping system should be checked for leakage [37]. The medium in the pipes is water at 20 $°C$. The joints should be left uninsulated so inspection can take place. A testing form has to be filled in which is shown in Appendix B.

Chapter 5

Conclusions

What has been delivered is a well documented basis. The mounts and where to mount them, the supporting structure which rests on the mounts, the supports and the pipes that they carry, the insulation, have all been designed for the medium which the pipes carry across the distance outside. Whereas the company normally builds and over-dimensions parts greatly to be on the safe side, all parts have been calculated and simulations have been run to select parts which fit the application best. Norms have been inspected in order to make a model for designing supports for expansion. Some of the used calculations can be re-used for other projects and some simulations are used to validate designs. Additionally some industry standards have been checked which are mandatory for such projects.
The resulting design is not yet ready for production. The parts in the assembly have to be drawn for production and ordered. The entrance to the old boiler room does not have good drawings and there are already many pipe sections for the existing boiler. Therefore the additional piping will be cut and welded on-site.

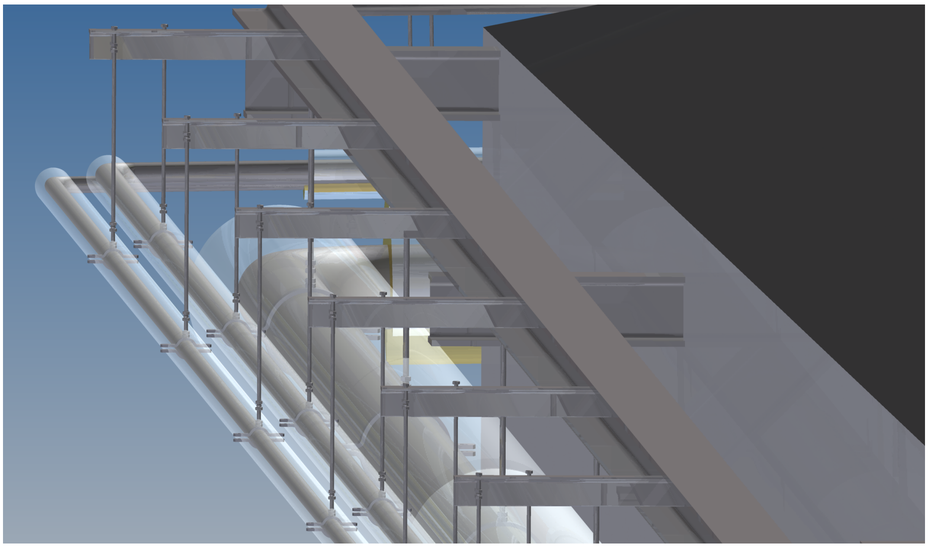

Figure 5.1: Pipe supports overview

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Appendices

Appendix A

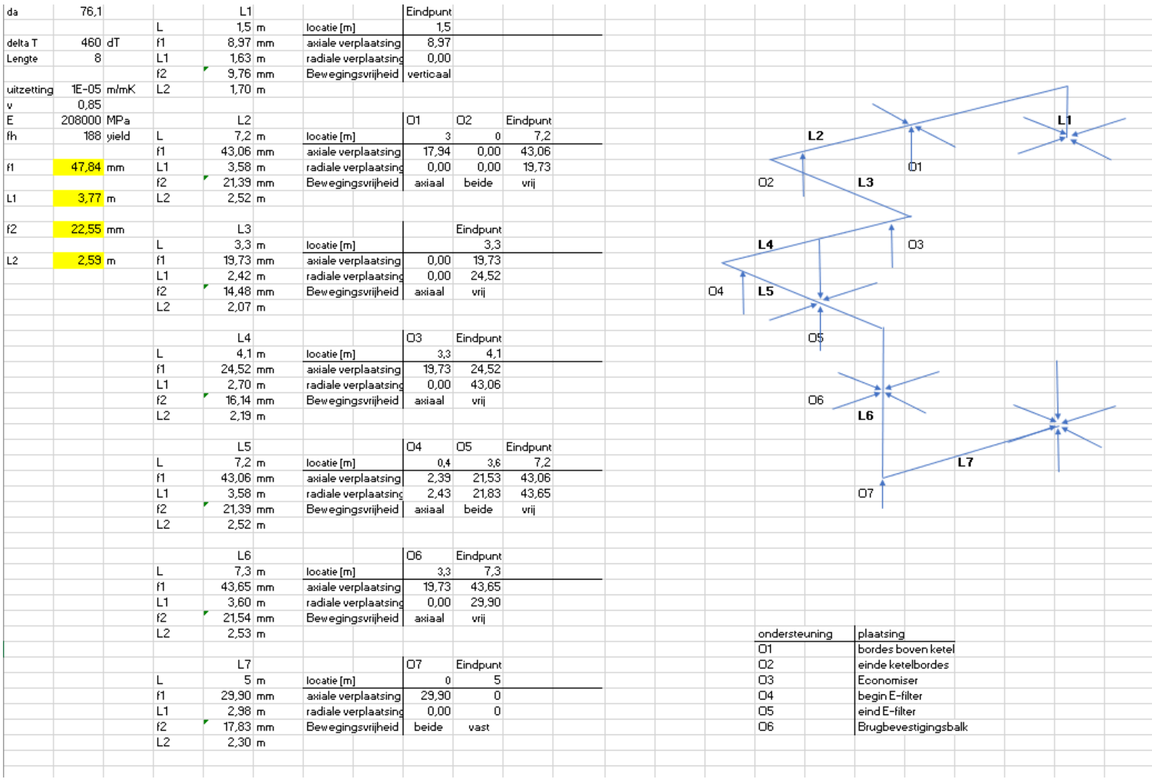
Reflection

A.1 Side projects

Besides the steam transport project I also worked on other side projects during my internship. I was in contact with the sales department for whom I would do calculations which were required for the sale of a new installation. These included basic fluid velocity, pressure drop and heat loss calculations as described in the first sections of chapter 2. I also supplied some figures for a presentation. For the ForFarmers project I joined in discussions regarding the layout inside the biomass installation building. I then mapped the pipes inside the building and modelled the expansion. This resulted in a schematic overview for which supports were simple to choose. I then went through older projects to design supports. An example for one section of pipe is shown in figure A.1. The resulting pipe with some supports is shown in blue in figure A.2

A.2 Evaluation

During my time at KARA Energy Systems I worked full-time on the ForFarmers project. During my first weeks I became familiar with the working environment. I started getting to know the modelling software, the tools used for sharing files and preserving the most recent ones. I gained insight through observing older projects and a guided tour through one of their new biomass installation. This allowed me to quickly grasp the concepts of the biomass installations as well as grant me some know-how as to take part in discussions regarding design choices. There were some calculations available which were relevant to my research but for the larger share I was trying to find new methods to design, compute and validate components which were not previously used. Complicate assemblies are not so easily computed by hand and this required me to find some simulation software to validate designs. The version of Autodesk Inventor which was provided did not include simulation features, since these often offered more problems than that they offered results. Through this project I was able to master a small part of the project while still being able to lend a hand in the neighbouring processes.
I got to know the company, which is a family business with a touch of Twente. The employees are all very down to earth and kind. The manager himself came by to check up on the employees daily for an update on an assignment or just for a friendly conversation. Besides this I also had some intermittent overview presentations with my mentor, the lead engineer and some others to show the progress on the design. This always resulted in some



da	76,1			L1			Eindpunt		
			L	1,5	m	locatie [m]	1,5		
delta T	460	dT	f1	8,97	mm	axiale verplaatsing	8,97		
Lengte	8		L1	1,63	m	radiale verplaatsing	0,00		
			f2	9,76	mm	Bewegingsvrijheid	verticaal		
uitzetting	1E-05	m/mK	L2	1,70	m				
v	0,85								
E	208000	MPa		L2			O1	O2	Eindpunt
fh	188	yield	L	7,2	m	locatie [m]	3	0	7,2
			f1	43,06	mm	axiale verplaatsing	17,94	0,00	43,06
f1	47,84	mm	L1	3,58	m	radiale verplaatsing	0,00	0,00	19,73
			f2	21,39	mm	Bewegingsvrijheid	axiaal	beide	vrij
L1	3,77	m	L2	2,52	m				
f2	22,55	mm		L3				Eindpunt	
			L	3,3	m	locatie [m]		3,3	
L2	2,59	m	f1	19,73	mm	axiale verplaatsing	0,00	19,73	
			L1	2,42	m	radiale verplaatsing	0,00	24,52	
			f2	14,48	mm	Bewegingsvrijheid	axiaal	vrij	
			L2	2,07	m				
				L4			O3	Eindpunt	
			L	4,1	m	locatie [m]	3,3	4,1	
			f1	24,52	mm	axiale verplaatsing	19,73	24,52	
			L1	2,70	m	radiale verplaatsing	0,00	43,06	
			f2	16,14	mm	Bewegingsvrijheid	axiaal	vrij	
			L2	2,19	m				
				L5			O4	O5	Eindpunt
			L	7,2	m	locatie [m]	0,4	3,6	7,2
			f1	43,06	mm	axiale verplaatsing	2,39	21,53	43,06
			L1	3,58	m	radiale verplaatsing	2,43	21,83	43,65
			f2	21,39	mm	Bewegingsvrijheid	axiaal	beide	vrij
			L2	2,52	m				
				L6			O6	Eindpunt	
			L	7,3	m	locatie [m]	3,3	7,3	
			f1	43,65	mm	axiale verplaatsing	19,73	43,65	
			L1	3,60	m	radiale verplaatsing	0,00	29,90	
			f2	21,54	mm	Bewegingsvrijheid	axiaal	vrij	
			L2	2,53	m				
				L7			O7	Eindpunt	
			L	5	m	locatie [m]	0	5	
			f1	29,90	mm	axiale verplaatsing	29,90	0	
			L1	2,98	m	radiale verplaatsing	0,00	0	
			f2	17,83	mm	Bewegingsvrijheid	beide	vast	
			L2	2,30	m				

ondersteuning	plaatsing
O1	bordes boven ketel
O2	einde ketelbordes
O3	Economiser
O4	begin E-filter
O5	eind E-filter
O6	Brugbevestigingsbalk

Figure A.1: Schematic design for expansion inside pipes

constructive criticism based off of years of experience which then could be implemented in the project.

The company is slowly expanding its knowledge and abilities but is limited by the fact that there are many ongoing projects. This meant that not much regarding design rules and tutorials was available and it was mostly up to me to try to reach out to colleagues. I found this hard at first and I had many calculations and simulations going on, on which I focused first. Later in the internship I walked around more with mostly immediate result. This is something that was very useful for me, since I'm used to doing things on my own without reaching out much and getting results after many hours of tedious research and trial and error. This company gave me some much needed practical experience which resulted in some necessary developments of myself. Having this experience also gave me a better insight in my future career as an engineer. I do like doing simulations and the designing aspect but I feel like projects at KARA do tend to look similar with only minor changes. The project I was on had some new aspects regarding high pressures and the outside conditions but on the long-term I would not feel at home in such a work environment. I did however enjoy the working staff very much and such had an enjoyable time here. I look back on the internship as hard work, friendly colleagues, and a good result. Hopefully I get to see the design in person once it gets installed at the end of this year.

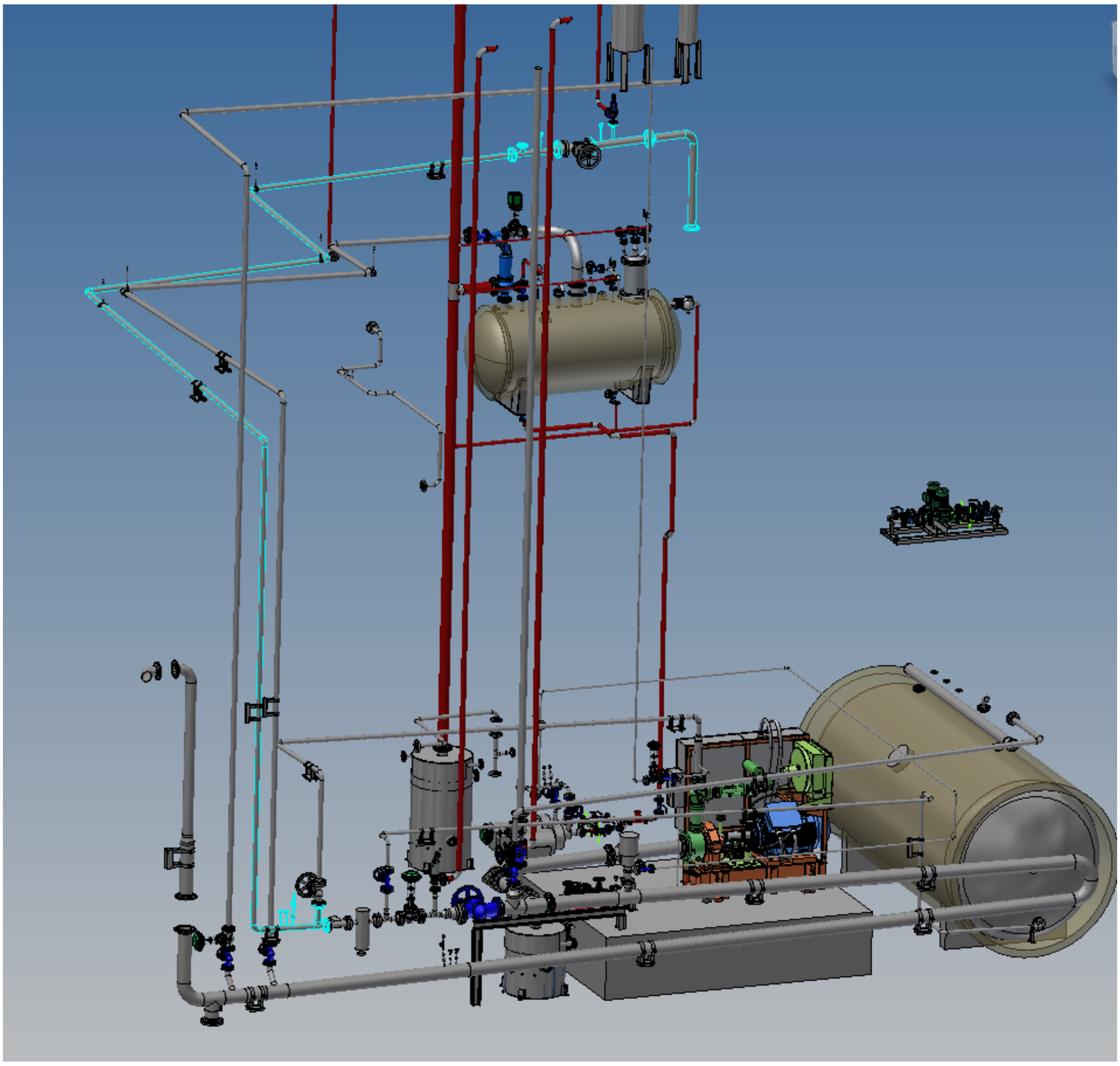


Figure A.2: Steam pipe as in figure A.1 with supports

Appendix B

Hydrostatic testing

KARA Energy Systems B.V.
Plesmanweg 27
NL 7602 PD; Almelo
The Netherlands

KARA ENERGY SYSTEMS

Pressure Test

(According to NEN EN 13480; T_{test}=20ºC)

Minimum required test pressure:	18,13	Bar (g)
Minimum required test time:	30	Minutes

Hereby we declare that the:

Product : **Pipe feedwater**
Project : **712**
Serial number : **810-001/2**
Design pressure : **10** Bar (g)
Design Temperature : **220** ºC

Has been tested at the following conditions

Testing pressure : ____________ Bar (g)
Testing medium : **water**
Serial no manometer : ____________________________
Test date : ____________
Start pressure : ____________ Bar (g)
Time & Date (Start) : ____________________________
Finish pressure : ____________ Bar (g)
Time & Date (Finish) : ____________________________

Tested by:	Supervised by:
____________________	____________________
Location & Date:	Location & Date:
____________________	____________________
Signature:	Signature:
____________________	____________________

SINCE 1910

Figure B.1: Form to be filled in for hydrostatic testing