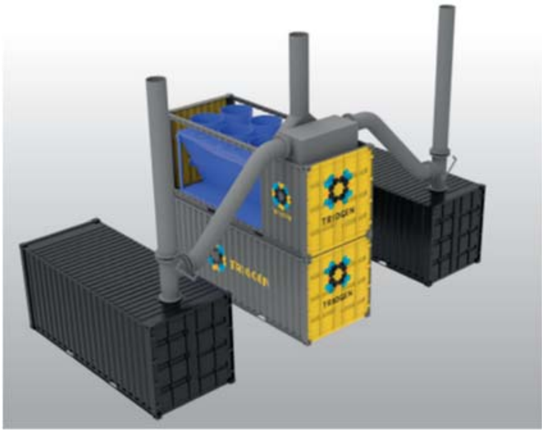
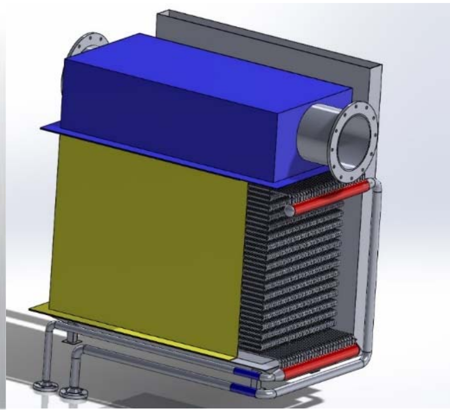
Redesigning the evaporator for the Triogen ORC (Organic Rankine cycle) for use in the containerized version of this ORC (the so called Triogen E-box)



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April 11th 2018- June 28th 2018

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Contents

Nomenclature

Symbol	Quantity
H	Enthalpy
c_p	Specific heat
T	Temperature
P	Pressure
$\dot{m}$	Mass flow
Q	Heat transfer
q	Heat flux
A	Area
h	Heat transfer coëfficient
U	Overall heat transfer coëfficient
E	Energy
v	Velocity
ρ	Density
μ	Viscosity
Re	Reynolds number
f	Friction factor (Darcy)
Nu	Nusselt number
t	Wall thickness
b	Width
L	length
h	Height
D	Diameter
V	Volume
PPR	Pipes per row
N_{rows}	Number of rows
σ	Stress

Introduction

Tri-O-Gen is a company that designs and builds Organic Rankine Cycles for power production. An Organic Rankine Cycle is comparable to a regular steam cycle (Rankine Cycle), where an organic fluid is used as working medium instead of water. The use of an organic fluid allows the system to produce electricity from relatively low temperature residual heat, at a higher efficiency than regular Rankine cycles can. Currently the main applications are fuel saving and power increase of bio fuel electricity generators.

The assignment for this internship will focus on redesigning a heat exchanger for use in the Tri-O-Gen t-box. The Tri-O-Gen system started out as separate components, but is progressively being developed to an integrated system that fits into two 20ft shipping containers, to allow for easy transportation.

The report will go over various aspects of the design process, more or less in the order these were encountered during the internship. First a global analysis of the entire cycle is made, to get a proper feeling of the thermodynamics involved. Then the general design case is studied, followed by the set of design requirements.

1. Thermodynamic cycle

Tri-O-Gen uses an Organic Rankine Cycle (ORC) to generate power from low-temperature heat sources. Much alike the regular Rankine cycle, an ORC generates power by driving vapour through a turbine. This vapour is created by boiling a liquid in an evaporator. Once the fluid leaves the turbine, it is cooled in a condenser and then brought back to pressure using a pump.

In a regular Rankine cycle one of the most severe limitations to the use of the turbine is the formation of water droplets in the turbine, as the water (typical fluid for a regular Rankine cycle) cools to below the saturation curve. In an ORC this does not occur due to the shape of the saturation curve, which allows the fluid to be expanded somewhat further. The exact shape of this curve depends on the fluid used, but the general shape is typical for organic fluids.

The Tri-O-Gen system uses toluene as working fluid. The saturation curve in a logarithmic P-h diagram is displayed in Figure A.1. In this curve the saturated vapour line clearly moves such that the expansion process in the turbine does not cause this line to be crossed.

The previously described thermodynamic behaviour also involves the addition of an extra component to the system. Since the fluid in an ORC leaves the turbine as a superheated vapour (rather than as a mixture), the heat between this point and the saturation point may be recovered and exchanged to the fluid after the pump. The component responsible for this is called a recuperator (see figure A2).

In full, the cycle now goes through the following steps (see appendix A):

Starting before the pump at low temperature, low pressure, in compressed liquid state (1), the toluene is led to two pumps, from which it leaves at low temperature but high pressure (3).

Then, the toluene goes into the recuperator, where the cold stream receives heat from the superheated vapour that exits the turbine. This is a heat exchanger, so no mixing takes place between these streams. This step is the first heating step in the entire heating stage (4).

The second, third and fourth heating step occur in the evaporator. Here the toluene is preheated to the saturated liquid point (5sl) (preheating stage), then boiled to saturated vapour point (5sv) (boiling stage) and lastly superheated slightly (6) (superheating stage).

The toluene then goes through a De Laval nozzle, which regulates the mass flow of the system (7).

Next, the toluene is expanded in the turbine. The toluene cools down and loses pressure, while the turbine produces electricity. The toluene leaves the turbine superheated, but at low pressure and relatively low temperature (8). However, the turbine outlet temperature is higher than the pump inlet temperature, which is typical for ORCs.

The toluene now goes through the hot side of the recuperator, where it transfers the latent heat to the cold stream that comes out of the pumps (9).

The stream then goes into the condenser, where heat from the toluene is transferred to the environment (air), causing the toluene to condense (11).

The toluene exitst the condenser in slightly compressed liquid state (1), such that the entire cycle may now start again.

The power outputs, associated heat etc are also given in Figure A.2. Final calculations are based on slightly different operating conditions.

2. Goal of the assignment

An evaporator must be designed to transfer heat from the flue gases of a diesel generator to the ORC working medium. The goal is to heat up as much toluene as possible with the supply of flue gas.

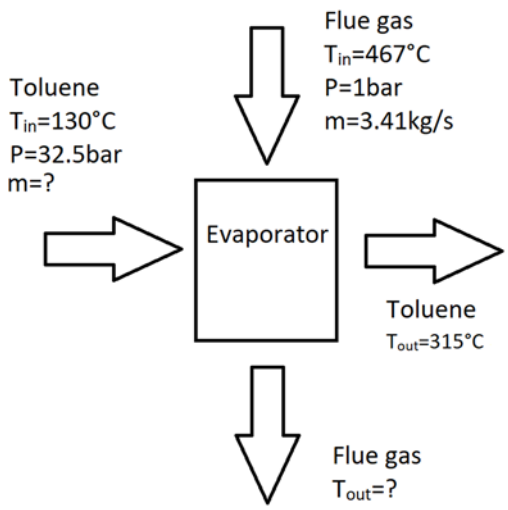

Figure 2.1: Schematic overview of the design case

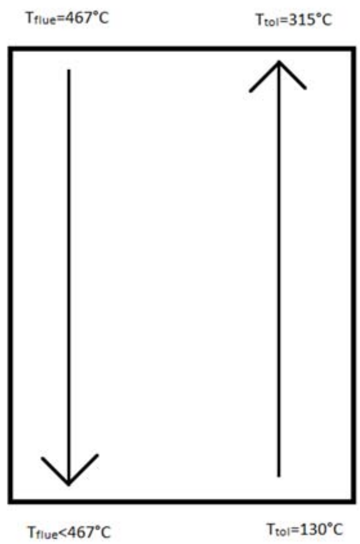

Figure 2.2: Principal temperature gradients

Although the goal is to maximize the toluene mass flow, there are a number of other requirements that complicate this design further. In the new setup Tri-O-Gen fits the entire system into two 20' shipping containers. One holds the recuperator, turbine, pumps, and storage tank, along with a number of safety components. This container is referred to as the e-box. The second container houses an air-cooled condenser and the evaporator. This container is called the t-box.

Since the evaporator must fit in this t-box, the available space is limited. The volume of the fluid inside the evaporator cannot be too high either, since it must be possible to fully drain the evaporator (the fluid then is stored in the storage tank). Minimizing the width of the evaporator (lengthwise direction of the container) hence is also a goal of this assignment.

Another limitation is that toluene degrades when exposed to high temperatures. To prevent the toluene from degrading over time, the toluene should never (at any point in the heat exchanger) exceed a specific temperature.

Furthermore, the toluene may not lose more than 2bar of pressure over the evaporator, whereas the flue gas may only experience a pressure drop of 15mbar.

The heat exchanger will be a cross-flow type heat exchanger, where the toluene passes perpendicular to the flue gas direction. The main flow direction of the toluene and the flue gas is opposite to each other (see Figure 2.2), which effectively makes the heat exchanger a counterflow heat exchanger for calculation purposes.

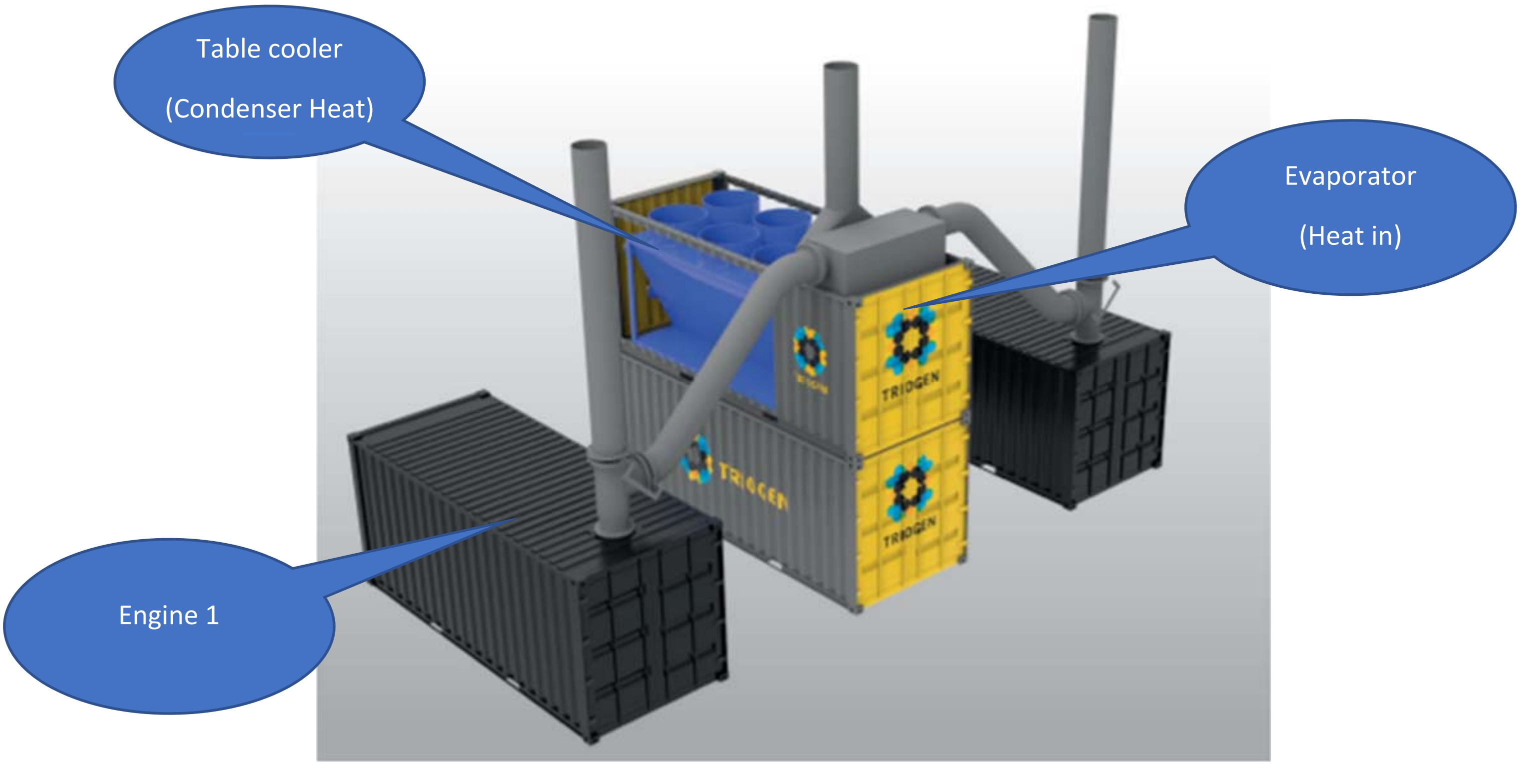

Figure 2.3: Proposed setup of the Tri-O-Gen E-box and t-box

3. Design constraints for the new evaporator

The goal of this assignment is to design a heat exchanger that maximizes the toluene mass flow, while a number of constraints have to be met. These constraints are the following:

- The wall temperature at the toluene may never exceed 330°C
- The flue gas side pressure drop may not be higher than 15mbar. The flue gas is produced by diesel generators which have turbochargers on them. These turbochargers may fail when the pressure drop over the heat exchanger becomes too high.
- The toluene side pressure drop is limited to 2bar.
- The internal volume on the toluene side cannot be higher than 950liter, since the storage tank only has this capacity. A higher volume means the evaporator cannot be fully drained.
- The toluene mass flow must be higher than 1.5kg/s.

The goal is to maximize the possible toluene mass flow, which is a slightly different approach from the usual approach where a heat exchanger is designed for a specific thermal load.

The geometrical constraints mentioned before are the maximum sizes the entire evaporator assembly may take up. From these constraints further guidelines can be found, which assume a certain setup and they also use the existing design.

Because the width should be minimized (one of the goals) an orientation is chosen with pipes stretching in the length direction of the evaporator

- The pipe length should be approximately 1750mm, to create space for the toluene connections inside the container.
- In order to integrate the flue gas distribution inlet and the exhaust in the container, the heat exchanger should not exceed 1200mm in height. If this is unrealistic, the distribution inlet cannot be placed inside the container.
- The width of the heat exchanger should not exceed 900mm. If the flue gas distribution inlet cannot be integrated in the container, one of the goals is to minimize the width of the heat exchanger.

The mass flow requirement can, through the thermodynamics discussed in chapter 1 and appendix A, be translated in a duty requirement. The operating conditions of the cycle, more specifically the toluene pressure and inlet and outlet temperature of the evaporator, are somewhat different from those mentioned in appendix A. Instead they are the conditions as given in Figure 2.1.

In Figure 2.1, the energy contents from both streams are not given. These are given by using the specific heat (c_p) of the flue gas and the enthalpy of the toluene:

$$\Delta\dot{E}_{flue} = \Delta T_{flue} * c_{p_{flue}} * \dot{m}_{flue}$$

Equation 3.1

Where the difference in energy per unit time $\Delta\dot{E}$ may also be expressed as the power Q. The specific heat c_p is given to be 1.1kJ/kgK for the gas mixture.

$$\Delta\dot{E}_{tol} = \Delta H_{tol} * \dot{m}_{tol}$$

Equation 3.2

Where the difference in enthalpy ΔH_{tol} is predetermined by the thermodynamics of the cycle. This results in a change of enthalpy of 613.42kJ/kg. This is the difference between point (4) and (6) as described in chapter 1, be it under slightly different working pressures and temperatures.

Assuming adiabatic heat exchange, a new energy balance can be obtained by equating the right hand side of Equation 3.1 and Equation 3.2:

$$\Delta T_{flue} * c_{p_{flue}} * \dot{m}_{flue} = \Delta H_{tol} * \dot{m}_{tol}$$

Equation 3.3

In a perfect (counterflow) heat exchanger heat transfer stops when the temperature of both mediums reach the same temperature anywhere in the heat exchanger. Since the toluene never heats up to the temperature of the flue gas inlet (see Figure 2.2), the logical possibility for this to happen is that the flue gas cools down to the toluene inlet temperature. Due to the boiling platform of the toluene a check should be done whether this is even a valid option. The absolute maximum toluene mass flow then becomes:

$$\dot{m}_{tol,max} = \frac{\Delta T_{flue,max} * c_{p_{flue}} * \dot{m}_{flue}}{\Delta H_{tol}} = \frac{(467°C - 130°C)1.1kJ/kgK * 3.41kg/s}{613.42kJ/kg} = 2.06kg/s$$

Equation 3.4

The associated power is simply the nominator of the previous equation:

$$Q_{max} = \Delta T_{flue,max} * c_{p_{flue}} * \dot{m}_{flue} = (467°C - 130°C)1.1kJ/kgK * 3.41kg/s = 1264.1kW$$

Equation 3.5

The minimum power requirement (based on 1.5kg/s toluene mass flow) is then 920.1kW.

4. Reference design

The assignment involves redesigning an evaporator with a specific optimization criterium and a number of requirements, which are described in chapters 2 and 3. However, a design already exists for an evaporator for use in the Tri-O-Gen t-box. This design does not meet all requirements. The specifications of this design are listed below, as a reference. In addition, appendix C contains photos of this design.

Table 4.1: Design parameters & performance of the reference design

Design parameter	Symbol	Value
Tube setup type	-	U-tubes
Use of fins	-	No
Tube diameter	D_o	22mm
Tube inner diameter	D_i	20mm
Tube length	L	1950mm
Pipes per row	PPR	32
Rows	N_{rows}	36
Horizontal pitch	S_t	27.5mm
Vertical pitch	S_l	27.5mm / 66mm
Height	h	Approximately 1700mm
Toluene side pressure drop	ΔP_{tol}	0.12bar*
Flue gas pressure drop	ΔP_{flue}	14.1mbar*
Internal volume	V	740liter
Toluene side wall temperature	T_{wall}	337.9°C
Toluene mass flow	$\dot{m}_{tol}$	1.78kg/s
Heat transfer	Q	1092kW

* As calculated in the model.

This heat exchanger is designed for use in a container, but has an inlet hood outside the container (as can be seen in Figure C.1and Figure C.2). Toluene pipes also extend out of the container.

5. Design parameters

To efficiently make a design for a heat exchanger that meets all requirements, all design parameters that can be varied must be considered. For all of these parameters an indication can be given of what a generally optimal value would be and what restrictions are relevant for this design parameter.

Some of these parameters need additional research to find optimal values, hence initially only the most important ones are set. These are now discussed. The full list of initially considered parameters can be found in appendix

The pipe length will be 1750mm. In the reference design this is 1950mm, but in the proposed design the toluene piping should not extend outside the t-box.

Furthermore, serpentines will be used (if these prove to provide sufficient heat transfer).

Fins will not be used, since the flue gas side heat transfer is expected to be reasonably well. Using fins takes a lot of space (such that much less tubes can be used in the same space), while it restricts the choice of tube diameters and fins may cause problems with the toluene wall temperature.

The material is for now chosen as stainless steel. This is not expected to influence the performance of the heat exchanger very much.

The width is initially taken as maximally 900mm, based on the reference design.

The height is taken as maximally 1200mm, based on the reference design. Here it should be noted that the entire inlet hood must fit inside the t-box in the new design.

All other parameters (if relevant) will be discussed later, since they need additional optimization.

6. Calculation model

The starting point for the calculations on the heat exchanger is an Excel model. This model employs the add-in FluidProp and the Visual Basics interface to interact between various calculation steps.

FluidProp is an add-in that is able to process thermodynamic data and calculations using the thermodynamic state principle.

During this assignment various modifications have been made to the model, but the core of the model has remained the same.

The core of the model is that an adiabatic heat exchanger is considered, where energy balance is considered between the flue gas and the toluene. For simplicity of the model the pressure changes of the toluene and flue gas have not been considered. Although both have influence on the performance, the influence is small and the added complexity to the model is significant.

7. The model considers a wide range of variables to compare the physical and mathematical design of the heat exchanger. Most of these variables are treated in the chapter Design constraints for the new evaporator

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Design parameters.

On the thermodynamics side of the model, first the relation between temperature and enthalpy for both the flue gas and the toluene are calculated. For the flue gas, a simple linear relation is assumed, where the change in enthalpy with temperature is described with a specific heat $c_{p,flue}$, which is assumed $1.1kJ/kg * K$. The relation for the toluene is obtained by calculating the enthalpy of toluene at the working pressure and the range of temperatures (along with the vapour fraction in the boiling region), according to the state postulate. The flue gas mass flow is also defined.

Apart from temperature, pressure and enthalpy, the density and viscosity of both fluids are also calculated for later use.

A first estimate of the toluene mass flow is made (this is corrected iteratively a number of times at a later point) based on the energy balance of the system. For this, a guess of the flue gas outlet temperature is needed (which is also used to estimate a flue gas composition) such that the total energy available in the flue gas is known. The maximum toluene mass flow then becomes

$$\dot{m}_{tol,max} = a * \frac{Q_{expect}}{\Delta H_{tol}}$$

Equation 8.1

Where a correction factor a (0<a≤1) may be applied to encourage the model to take a conservative initial guess. This causes the actual toluene mass flow to go up if allowable, while it will go down if the initial guess was too optimistic. The latter option may cause the model to give an optimistic guess. In the model a value of 0.9 is chosen, such that the initial guess is slightly conservative if the guess for the flue gas outlet temperature was correct.

The heat exchanger is now divided into several parallel heat exchangers (based on the amount of pipes per row) and serial heat exchangers (based on the amount of rows and the amount of sections per row). The parallel heat exchangers are all calculated at once, since their thermal behavior is equal. Each parallel heat exchanger (a serpentine) gets an equal part of the toluene mass flow and flue gas mass flow assigned to it.

The serial heat exchangers are calculated one after the other.

For each section a heat transfer coefficient is calculated, based on the thermal conduction of the tubes, the heat transfer on the toluene side and the heat transfer on the flue gas side:

$$U_{flue} = \frac{1}{\frac{D_{out}}{h_{tol} * D_{in}} + \frac{t_{eff,cond}}{k_{cond}} + \frac{1}{h_{flue}}}$$

Equation 8.2

Where the effective wall thickness $t_{eff,cond}$ is calculated using

$$t_{eff,cond} = \frac{D_{out} * ln\left(\frac{D_{out}}{D_{in}}\right)}{2}$$

Equation 8.3

In determining the heat transfer coefficients h_{tol} and h_{flue} the Nusselt number is used, which depends on the viscosity and density. The relations used for the heat transfer coefficients are further discussed in the work of Arjen Hiemstra, a student at the UT who created the base model on which this model continues. Since all relations are explained here in detail and these relations are somewhat out of the scope of the assignment, the reader is referred to his report for further information ((Hiemstra, 2011)). The overall heat transfer coefficient (in W/K) multiplied by the temperature difference of the fluids now gives the heat transfer (in W) for this section. The temperature difference is calculated using a logarithmic mean temperature difference (LMTD) between the inlet and outlet on the toluene side and the inlet and outlet on the flue gas side.

The process to find this temperature difference is iterative and somewhat complex. First of all, the need for an iterative solution must be clarified. The logarithmic mean temperature difference is dependent on the flue gas inlet temperature, the flue gas outlet temperature, the toluene inlet temperature and the toluene outlet temperature. For each section, the toluene outlet temperature and the flue gas inlet temperature are known (output values from the previous section). Note that the toluene is actually being considered as cooling down rather than heating up in this way, since the enthalpy difference with a lower temperature is being calculated. The problem now is that the heat transfer cannot be calculated, since it requires the LMTD, and the LMTD cannot be calculated, since it requires the temperature differences, for which the heat transfer is required.

So, a guess must be made to initiate the iteration. By setting the initial LMTD (used for calculating the heat transfer) equal to the flue gas inlet temperature minus the toluene outlet temperature, a first guess can be made for the heat transfer. This is then used to calculate the flue gas outlet temperature and toluene inlet temperature (both based on enthalpy difference), after which an improved value of the LMTD is calculated. With this value the process can be done again. As little as three iterations yield highly accurate (and converging) results, with more iterations barely adding accuracy.

Since the actual mass flow is not correct, this must be corrected. This is done by taking the ratio between the actual enthalpy difference of the toluene and the enthalpy difference under process conditions (as defined). Multiplying the mass flow by that ratio for the next iteration causes the transferred heat to be (nearly) equal, while the enthalpy values for the toluene are now correct throughout the calculations. A small deviation in the transferred heat may be present between the first and second iteration due to a change in heat transfer at the toluene side (different mass flow causes the velocity to be different, which leads to the Nusselt number being different). However, since the initial toluene mass flow guess is already substantiated (i.e. not a blind guess) and the heat transfer is mainly limited by the flue gas side heat transfer, the actual mass flow converges rapidly, even after the first correction step. In the model, four correction steps are used.

A condition for this procedure to work accurately is that the estimated flue gas outlet temperature should be close to the actual (calculated) outlet temperature. Due to the ease of convergence of this method, “close” mainly means that the estimated outlet temperature should be lower than or roughly equal to the actual outlet temperature (the model malfunctions with estimated flue gas outlet temperatures much higher than actual outlet temperatures, since it thinks too many rows have been used). The easiest way of using this is hence to run the model twice; the first time the flue gas outlet estimate is set to the toluene inlet temperature (maximal achievable) and the second time the calculated flue gas outlet temperature is used as estimate. When this procedure is used the difference between the two is generally less than 0.01°C, making it a highly effective method, although the need for the second step can even be questioned.

With the corrected mass flow the entire process is gone over again, including the iteration of the LMTD for each section. This means that in effect the calculations are done multiple times, for various mass flows, where the mass flow converges to the mass flow that corresponds to adiabatic operation of the heat exchanger.

After the final mass flow is calculated, the pressure drop over each segment is calculated for the toluene, while for the flue gas the pressure drop per row is calculated. It should be noted here that both pressure drops are not taken into account in the working pressure of the fluids, they are only calculated to see what the pressure drop in the end becomes. On the toluene side an additional pressure drop for a 180° bend is calculated after each row. The total pressure drop on the toluene side is obtained by adding these pressure drop effects along with the inlet and outlet pressure drop effects. At the flue gas side the pressure drop is averaged between the sections of a row to obtain the pressure drop of that row. This is valid for small deviations in the pressure drop between different sections. The overall pressure drop on the flue gas side is then calculated by summing the average pressure drop of each row.

For the pressure drop on the toluene side various effects of pressure drop are taken into account: dynamic pressure drop, momentum pressure drop, static pressure drop and pressure drop through bends and expansion and contraction.

Next to the pressure drops, the wall temperature on the toluene side is also calculated. This is done using the Tri-O-Gen model:

$$T_{wall,tol} = T_{tol} + \frac{q_{local}}{h_{tol,local}}$$

Equation 8.4

The subscript local is used here since the wall temperature will vary from point to point.

All geometric parameters are also related to obtain final sizes of the bundle and the internal volume.

The heat exchanger has so far been considered adiabatic. However, a non-adiabatic heat exchanger can also be modelled by simply taking the heat transferred to the toluene side to be lower than the heat that the flue gas transfers, where the imbalance is "lost to the environment":

$$Q_{tol} = Q_{flue}(1 - losses_{\%})$$

Equation 8.5

For small losses the effect is that the flue gas outlet temperature remains roughly equal, while the toluene mass flow drops by about the same amount as the defined losses and the wall temperature goes up a bit due to the lowered toluene mass flow. Whether the increased wall temperature is correct or not is debatable, since the heat flux on the toluene side could be taken to decrease, but this is currently not done.

Since the total heat transfer is defined as the exchanged heat, this rating is based on the heat transfer on the flue gas side. If there are heat losses, the heat transfer to the toluene side is less, meaning the duty of the heat exchanger becomes

$$Q_{tol} = Q * (1 - losses_{\%})$$

Equation 8.6

Obviously the relations used for heat transfer coefficients and pressure drops are not detailed in this section, they are instead described in appendix **Error! Reference source not found.** to keep this section comprehensive.

Discrepancies in the model

The model uses the estimated flue gas outlet temperature to obtain a flue gas composition. This is required to get a reference for the density and viscosity at various temperatures (the pressure is assumed constant). Each mixture should have a specific heat of approximately 1.1kJ/kgK. However, between various estimated temperatures tiny differences may occur in this value, leading to slightly different thermal behaviour of the flue gas.

These differences are very minor, but it is good to note that the model introduces a small discrepancy here between the “wanted” specific heat value and the specific heat value that is actually used in the model.

One solution here is to increase the accuracy of the mixture composition calculation: right now the best of 500 options is chosen, but one could also increase this to 5000 options. This would cause the influence of different temperature estimates to decrease for the used specific heat value.

9. Design choices

Based on the calculation model and some other argumentation a number of design variables can now be chosen. For each variable the reasoning and result is discussed.

9.1. Pipe length

A pipe length of 1750mm will be used. Previous designs use a pipe length of 1950mm, the gained clearance can be used to connect the toluene piping inside the container.

9.2. Use of fins

Although fins can help increase the heat transfer per pipe, a design without fins can be made much more compact than a design with fins. In addition fins increase the heat transfer to the toluene per unit area, which will lead to an increased wall temperature. Initial study suggests that the wall temperature will already be close to 330°C, so using fins is not recommended (instead a low tube diameter should be used, which is already concluded in the previous section).

9.3. Use of serpentines

Serpentines will be used, since this allows for full draining of the evaporator and they offer significant production advantages over the classic U-tube design. The U-tube design is kept as secondary option. Now that the pipe length of the serpentines is known, the direction of the serpentines is also known due to geometrical constraints.

9.4. Choice of tube diameter

From appendix D, it can be seen that in general a lower outer diameter leads to a higher heat transfer, a lower wall temperature and higher pressure drops. For the flue gas pressure drop mainly the inlet area is of importance (and not so much the diameter of each tube).

When choosing a certain diameter, the robustness of the design is also of importance. A certain diameter may not exceed the toluene pressure drop limit in the calculations, but if reality differs a bit from the calculations it may. Therefore it is best to take a diameter that provides results that are not too sensitive to deviations. This also allows the final design to be revised by changing the initial conditions without immediately having to choose a new diameter. This is a good approach since appendix D holds for one diameter.

Taking this in mind, a logical choice would be to take the diameter at the highest as 14mm, where the wall temperature can just be corrected for by changing the pitch, but preferably somewhat lower to have better heat transfer. On the other hand, the pressure drop of the toluene increases really fast when going from 10mm back to 8mm diameter. To prevent high sensitivity of the system an outer diameter of 12mm is chosen.

It should be noted here that the actual flue gas and toluene pressure drop are difficult to predict accurately. So, a design with 10mm tubes will most likely work slightly better, but it is not chosen due to the sensitivity of such a design.

The wall thickness is chosen as 1mm (for each considered outer diameter). The strength of this profile is examined in appendix G.

9.5. Remaining parameters

The remainder of the design parameters has to be determined further based on the geometrical constraints and thermodynamical performance of the heat exchanger. The design sequence for these parameters is described in the next section.

10. Design of the evaporator

In the previous section a tube size has been chosen. With this, the actual design of the heat exchanger begins. At first the design for 1200mm of vertical space is done, then a short study is done into the effects of safety margins and the extension of the vertical space by placing the inlet hood outside the container.

10.1. Design for 12mm tubes

Now that an inner- and outer diameter for the tubes are chosen, a design proposal should be made specifically for this tube size. Although a design methodology has been made for any set of design variables, the set of constraints discussed in chapter 3 along with the tube diameter makes this approach redundantly complex.

Therefore, a different approach may now also be taken. One can simply fill up the available space with tubes and check all restrictions (internal volume, flue gas pressure drop, toluene pressure drop and wall temperature of the toluene). The only true variables are now the pitch ratios. However, since the amount of space available in vertical direction is very limited (see chapter 3), the vertical pitch should be kept at the minimum of 3 (for serpentines). The horizontal pitch may now be varied to gain optimal results while still meeting all restrictions.

In practice, the latter means that the pitch should be chosen as low as possible. When lowering the pitch, the flue gas heat transfer increases, leading to a higher possible toluene mass flow. This goes at the cost of an increased wall temperature and increased pressure drops.

In this method, no free choice is made in the amount of rows and pipes per row. It seems evident that, in the limited available vertical space, as many as possible rows should be used. This can be verified by lowering the row count: although the pressure drops do lower slightly, so does the performance of the heat exchanger.

In general it can be said that the difficulty of this design is that the available space in vertical direction is very limited, which restricts the amount of rows. This means that a high performance must be partially obtained by using a high amount of pipes per row and a low pitch ratio, both of which contribute to a higher wall temperature.

Based on this approach, the row count is now set to 34 (with a vertical pitch of 36mm this is the maximum that fits in a 1200mm space when there are no inlet and outlet to and from the last row). This is the absolute maximum. From a performance point of view, the best setup in the horizontal plane would be to have a minimum pitch ratio (1.25) combined with a maximum amount of pipes per row. In the width available, currently set at 900mm, this yields 60 pipes per row as the maximum. In this scenario an output of 1132kW (89.5%) can be achieved, with a wall temperature of 330.8°C and pressure drops of 13.1mbar on the flue gas side and 0.32bar on the toluene side respectively. The internal volume is estimated at around 300 liter, which is far less than the maximum.

While the pressure drops and internal volume immediately satisfy the restrictions, the maximum wall temperature is exceeded in this design. In order to get a design that satisfies the 330°C restriction, the pitch ratio and the amount of pipes per rows may be changed.

These two variables are linked to each other through the maximum inlet width, which was set at 900mm. It may also be chosen to lower the inlet width in order, to retain high performance while lowering the maximum wall temperature. While a decreasing amount of pipes per row decreases the wall temperature, it also lowers the performance. The same goes for an increase in pitch ratio. Decreasing the amount of pipes per row also increases pressure drops, while increasing the pitch ratio lowers the pressure drops. Based on this, a combination of the two could be a proper solution to keep the pressure drop roughly equal (since the pressure drop on the flue gas side is near its restriction) while lowering the wall temperature. Below an overview is given for the performance along with whether restrictions are met or not.

Table 10.1: feasibility and performance

		pitch ratio							
	Q (kW)	1,25	1,3	1,35	1,4	1,45	1,5	1,55	1,6
PPR	30	1065	1042	1021	1003	986	970	955	941
	35	1081	1060	1040	1023	1006	991	977	963
	40	1095	1074	1056	1039	1023	1009	995	982
	45	1106	1087	1069	1053	1038	1024	1011	998
	50	1116	1098	1081	1065	1051	1037	1024	1012
	55	1125	1107	1091	1076	1062	1049	1036	1025
	60	1132	1115	1099	1085	1071	1059	1047	1036

The bright red indicates that one or more of the restrictions are not met, while the numbers indicate the thermal performance in kW. This performance can be directly translated to a mass flow of the toluene by dividing by 613.4kJ/kg, although it is better to calculate the mass flow itself. This is due to a small discrepancy in the model, which only influences the translation between the heat transfer and the mass flow. The model actually does employ adiabatic heat transfer, so the calculated mass flow always agrees with the flue gas outlet temperature, mass flow, and specific heat.

All options that meet all restrictions are coloured according to performance, the best being green and the worst being pale red. A trend may be observed (also in the red coloured area) that better performance is achieved by increasing the pipes per row count, while decreasing the pitch. However, a pitch ratio of 1.25 results in a too high wall temperature, while using 60 pipes per row will always cause the maximum width to be exceeded except for the case that the pitch ratio is 1.25. A detailed overview of the restrictions can be found in appendix E.

Based on this analysis, the best performing option among those considered is a setup with a pitch ratio of 1.3 and 55 pipes per row. The associated toluene mass flow is 1.80kg/s.

Based on Table 10.1 and appendix E, it is wise to interpret the bright red boxes correctly. While all of the restrictions must be met, some are more easy to design for than others. The internal volume and the width are known exactly for a given design, while the pressure drops and the wall temperature may differ somewhat from done calculations. This also goes for the toluene mass flow, but this is no restriction. All options meet the requirement for the toluene pressure drop with ease, which comes from the choice for an outer diameter of 12mm.

By taking a somewhat higher safety towards the allowable flue gas pressure drop and wall temperature, a solution more to the lower right of the table would be desirable. This yields the possibilities of 55 pipes per row and a pitch ratio of 1.35, or 50 pipes per row and pitch ratios of 1.35, 1.4, 1.45, or 1.5.

Checking the wall temperature and flue gas pressure drop for these cases, starting with the best performing, we see that a considerable margin is already achieved for the option of 55 pipes per row and a pitch ratio of 1.35. The thermal performance is still at 98.5% of that of the best suitable option (1.78kg/s).

This option also provides a robust design for operating modes other than full load. Even in lower loads the maximum wall temperature is exceeded only by 0.3°C (assuming the flue gas mass flow halves, which is not necessarily completely true).

It should be noted that these designs are done to optimize the toluene mass flow. This does not mean that the designed heat exchanger will in fact be able to deal with exactly this mass flow. For instance, no thermal entrance region has been considered in the heat exchanger (not even the vertical pitch at the front and back has been considered, but this may be countered by either adding these pitches or ignoring the performance of these rows. Another point of interest is that while most options have some clearance towards the minimum power, the option with 30 pipes per row and a pitch ratio of 1.6 comes very close to this minimum. If it performs even just over 2% worse than calculated, this minimum may not be met.

10.2. Effect of additional vertical space and safety

In the proposed design the flue gas inlet hood is placed inside the container. In previous designs it was placed on top of the container. By doing so in a new design, the evaporator could be higher than it currently is, up to around 1700mm instead of 1200mm. This would also mean that a restriction to the width will no longer be caused by the flue gas inlet flanges (if this is currently the case). The 1200mm version (casing height 1210mm, with 5mm added height at either side) has 34 rows.

$$N_{rows} = \frac{L_{vert} - D_o}{S_L} + 1 = \frac{1188mm}{36mm} + 1 = 34$$

Equation 10.1

If the available vertical space is increased by the entire height of the top box (545mm), the maximum amount of rows becomes

$$N_{rows} = \frac{L_{vert} - D_o}{S_L} + 1 = \frac{1733mm}{36mm} + 1 = 49.138$$

Equation 10.2

However, a full, even, amount of rows should be used, so 48 rows would be used instead in this case, which takes up a vertical space of 1704mm (excluding added casing).

A comparison will be made between the thermodynamical performance of both options, as well as options that consider a safety margin. The safety margin is considered by evaluating the thermal load and/ or toluene mass flow for a heat exchanger with less rows than the actual design, while the pressure drops are still calculated using the actual heat exchanger. This is done to get a conservative estimate of the thermal load. From the safe option, the real option has about 10% added area (in the from of added rows).

Table 10.2: Effect of considering safety and increasing the evaporator height

Amount of rows calculated/ actual	Safety (added area)	Toluene mass flow (kg/s)	Thermal load (kW)	Flue gas pressure drop (mbar)	Toluene pressure drop (bar)	Internal volume (liter)	Wall temperature (°C)
30/ 34	13.33%	1.7299	1061	7.4	0.35	275.831	329.1
34/ 34	0%	1.7799	1092	7.4	0.35	275.831	328.8
42/ 48	14.29%	1.8522	1136	10.0	0.50	389.643	328.4
48/ 48	0%	1.8905	1160	10.0	0.50	389.643	328.2

Adding an additional 14 rows to the design improves the thermal performance by nearly 70kW (the safe design even improves somewhat more than 70kW), while the pressure drops and internal volume are still within margin. Since the only thing that changes in the heat exchanger at the inlet is the toluene mass flow, the wall temperature at the toluene side actually decreases, but this is safe in both designs.

Comparing the ideal designs between each other or the safe designs between each other, the increase in mass flow by increasing the evaporator with 14 rows is expected to be around 0.11-0.12kg/s (or around 70kW in thermal load).

Due to further complications in the system it is somewhat difficult to say that an increase in mass flow leads to a colinear increase in power output of the turbine. Also, a higher toluene mass flow in the evaporator also means that other components of the system may have to be resized.

With the safety taken into account the minimum mass flow for the proposed design becomes 1.73kg/s, while it becomes 1.85kg/s for the case that the inlet hood is positioned outside the container.

11. Component design

Now that the global design of the heat exchanger is done, all other components involved should be designed to see if the design as a whole fits in the available space. For naming of these components see Figure F.1.

The starting point here is the set of serpentines. The serpentines, the connections to other piping and casings for the heat exchanger will first be designed. Then an inlet for the heat exchanger (the top box) will be designed, along with a bottom box and an exhaust. Distributors to divide and combine the flows to and from the serpentines must also be designed.

Piping, insulation, and other practical issues influence this process along the way.

During the design of these components it may occur that certain components must be redesigned completely and/or that the heat exchanger itself still has to change size.

11.1. Serpentines

The design of the serpentines is largely determined by the heat exchanger design. The tube diameter, combined with a pipe length and bend radius, gives the serpentine geometry per pass. For a set amount of rows (currently 34) the internal tube banks may then be designed. It should be noted here that the ends of each serpentine go straight through rather than that they make a bend, such that the toluene may enter and leave the heat exchanger.

11.2. Casing

The casing is made such that it encloses the whole set of serpentines neatly. A clearance of 2mm is used at the sides (0.5*c) and a thickness of 5mm is assumed. Tri-O-Gen generally uses casings slightly thinner than 5mm, but 5mm can also be used. The casing must hold all 55 serpentines next to eachother, with the required spacing and clearance. The height is determined by the serpentine height (1200mm) plus a small margin such that the serpentines are supported in the casing as well. This margin is taken as 5mm, such that the height of the casing becomes 1210mm.

11.3. Flue gas inlets/ top box

The horizontal cross section of the top box will be the same as that of the casing. This helps in evenly distributing the flue gas flow over the heat exchanger. Again, a thickness of 5mm is used, which will also form the top part of this box. The height of this box is determined mainly by the inlets that have to be connected to it; two 16” flue gas inlet flanges (BS10 table D) are used to direct the flue gas into the heat exchanger.

These flanges are rather large, as is the 16” pipe. First of all, neither the flange nor the pipe can stick out above the rest of the design. Initially, the top of the flange could be laid equal to the top of the flue gas inlet. However, 100mm insulation will be added on top of the inlet, which allows the flange to stick out by 100mm. In practice, that would mean that part of the pipe does not end up in the inlet. Therefore, pipe is placed as high as reasonably possible (with some margin).

On the bottom side of the flue gas inlet a similar problem occurs, yet now the problem is that the flue gas inlets should not interfere with the serpentine outlets. On one side this is no problem, since the serpentine outlets are all located at the same side. To minimize this problem, the flue gas inlets are hence made of different lengths, such that the flange is positioned farther away from the serpentine outlets. This does not solve the vertical spacing issue however, which is why the top box will extend somewhat further downward. The overall height becomes 545mm, 5mm of which is the top plate, 17mm the clearance at the top side, 117mm clearance at the bottom side and the remainder is the hole for the flue gas inlets.

11.4. Exhaust

In designing an outlet and exhaust for the flue gas two main factors must be considered. The first is that the velocity in the outlet and exhaust should not exceed 20m/s. The second is that the pressure drop should be nearly equal over each part of the heat exchanger. If this is not true, the flow going through various sections may differ significantly, which is not beneficial to the heat transfer.

In a perfect scenario, the inlet and outlet of the heat exchanger would both be exactly in line with the heat exchanger. When placing the outlet perpendicular to this direction and on one side, the effect of this on the flow pattern should be taken in consideration. The inlets are placed symmetrically at the top and for now these are assumed to be placed in the middle.

First of all, the exhaust will be designed to obtain a reference design for the heat exchanger outlet. The design criterium for the outlet is that the flue gas velocity should not exceed 20m/s. In addition, the pressure drop over the exhaust should not be significant since this influences the overall pressure drop of the system.

A minimum area for the exhaust may easily be calculated from the flue gas volume flow and maximum velocity:

$$A_{exhaust,min} = \frac{\dot{V}_{flue}}{v_{flue,max}} = \frac{\frac{3.41kg/s}{0.777kg/m^3}}{20m/s} = 0.22m^2$$

Equation 11.1

Using the same width as the heat exchanger to keep the added space down as much as possible, the width becomes 1802mm and a rectangular profile should be used. The minimum profile height then becomes:

$$h_{min} = \frac{A_{exhaust,min}}{b} = \frac{0.22m^2}{1.802m} = 0.122m$$

Equation 11.2

To be somewhat on the secure side, a profile height of 0.135m is used. This gives a 10% safety margin while it only takes up 13mm additional space.

The hydraulic diameter of this profile is

$$D_{h,exhaust} = \frac{4 * A}{p} = \frac{4 * 1.802m * 0.135m}{2 * (1.802m + 0.135m)} = 0.251m$$

Equation 11.3

The Reynolds number of this flow is

$$Re_{flue,exhaust} = \frac{\rho_{flue} * v_{flue} * D_h}{\mu_{flue}} = \frac{\frac{0.777kg}{m^3} * \frac{20m}{s} * \frac{0.122m}{0.135m} * 0.251m}{2.67 * 10^{-5} Pa * s} = 132020$$

Equation 11.4

The correction factor (0.122m/0.135m) is applied to the velocity because the actual area is higher than the area required to get a velocity of 20m/s.

From a Moody chart the Darcy friction factor becomes (assuming a roughness of 0.045) 0.07.

With this, the pressure drop per meter of straight exhaust (not considering the 90° bend) becomes

$$\frac{\Delta P}{L} = f * \frac{\rho * v^2}{2 * D_h} = \frac{0.07 * \frac{0.777kg}{m^3} * 18.074^2}{2 * 0.251m} = \frac{35.4Pa}{m} = \frac{0.354mbar}{m}$$

Equation 11.5

The pressure drop of the exhaust itself will hence not be very high (since the exhaust will be less than 3m high this will become at most 1mbar).

11.5. Bottom box

A bigger issue is the distribution of pressure drops over the flue gas outlet to the exhaust. If there are larger differences in this outlet, the back-pressure of each channel of the heat exchanger may vary. Since all channels are combined, this back-pressure must flatten out. The way this happens is that the actual flow over different channels will vary. Whether this is problematic or not depends on the magnitude of this effect.

As a starting point, the flue gas outlet will have the same shape as the exhaust (1802mm x 135mm).

The pressure drop for each specific channel number n to the next (note, this is NOT the exhaust!) is the following:

$$\Delta P_n = f * \frac{v_n^2 * \rho}{2} * \frac{L_n}{D_h}$$

Equation 11.6

While the total pressure drop from the first channel to the outlet becomes

$$\Delta P_{outlet} = \sum_{n=1}^{56} \Delta P_n$$

Equation 11.7

In both cases the friction factor used is the earlier mentioned 0.07, which is valid in the turbulent part of the flue gas outlet. In the laminar and transition parts the friction factor will be lower, so the 0.07 is on the high side and should be corrected if it yields problematic results

This total pressure drop to the exhaust may also be calculated for each channel individually if relevant. However; calculating the summation for the first channel (maximum pressure drop difference) yields a pressure drop of 0.14mbar. The pressure drop over the heat exchanger is calculated as 7.4mbar, which means that the effect of this non-symmetric pressure distribution is very low. In comparison; if the mass flow through the channel the closest to the exhaust would be 101% of the nominal mass flow and the mass flow through the furthest away is 99% of the nominal mass flow, this would cause a pressure drop difference of 0.31mbar.

Although it is possible to calculate the expected flow per channel, the variations are hence predicted to be within small and acceptable margins, such that the effect of the flue gas outlet can be neglected.

This also means that the inlets will be placed centrally, to prevent uneven pressure distributions at the top of the heat exchanger.

The bottom box can also be extended somewhat further than the previous mentioned 135mm, to allow the placement of an inspection hatch and to further lower the pressure drop difference. Eventually the front of the bottom box will be adjusted somewhat, causing the pressure drop in the section the furthest away from the outlet to increase, so a slight increase in height (and width that hydraulic diameter) is desirable to make sure the effect of this adaptation will not cause uneven pressure distributions. The entire box is increased in height by 100mm, leading the pressure drop difference (between the front and the back) of the non-adjusted box to become 0.08mbar. Adjusting it will have some effect, but only ever so slightly.

11.6. Distributors

The distributors are pipes to which all serpentine inlets or outlets are welded. There is one distributor for the inlet and one for the outlet. The distributors are connected to a pipe of the size of the corresponding toluene inlet and outlet that connects the heat exchanger to the rest of the system.

To establish an even flow, it is desired that the velocity in the distributors is lower than that in the serpentines. At the inlet this ensures proper distribution over all channels, while at the outlet it allows all flows to be combined properly before exiting the distributor.

$$A_{distributor,min} = PPR * A_{tol} = 55 * \frac{\pi}{4} * 10mm^2 = 4320mm^2$$

Equation 11.8

$$D_{distributor,min} = \sqrt{\frac{4 * A_{distributor,min}}{\pi}} = 74.16mm$$

Equation 11.9

Since the resulting diameter is not commonly used in industry (and a somewhat larger diameter is preferred), a pipe of 77.9mm inner diameter (88.9mm outer diameter) is used. Although it seems the effect here is limited, the effect will increase when moving further away from the inlet (since part of the fluid is in the serpentines then).

The distributor at the inlet has a connection to a DN40 pipe, while the outlet has a DN50 connection.

11.7. Inlet & outlet pipelines

The outlet toluene piping is sized as DN 50, the inlet as DN40. Both are pressure rated as PN100 (100bar of pressure at room temperature, which is required to safely use stainless steel at 350°C and 45bar, which is the design pressure).

This piping must be connected to the E-box (under the t-box) through the floor of the t-box. With respect to fixation and transport the following must be considered:

- During transport all piping elements must stay within the respective containers
- During operation the pipes must be connected in a way such that the pipes are connected. This connection must be accessible.

Since this design focus on the t-box side, a solution from this side will be sought. The first solution is to let the pipes from the t-box extend a tiny bit into the E-box, where a flange connection is made. During transport these pipes are bent away (by force). In this case (and the second) a displacement of around 100mm is needed to make the connection during operation while the pipes would also stay in the container during transport.

Initial studies showed that various setups (results are not included in the report) all exceed or go eerily close to the maximum allowable von Mises stress, where welds in the pipes have not yet been considered (all components are considered to be one solid piece). Even with additional S-bends to accommodate easier bending the stresses go quite high. This option is therefore not considered as suitable.

The second option is to use piping components that allow movement. Three different options arise here: axial compensators, angular compensators and swivel joints. The latter is not a valid option, since swivel joints of this size cannot be found in a PN100 rating.

Axial compensators are also impractical for the large displacement of 100mm, this can much easier be achieved by employing angular compensators at the right position. A design that shows the pipe connections in operation and transport state is shown in Figure 11.1. Two side notes should be placed here: compensators of this size and pressure rating are no standard components, but can be ordered from companies like Eriks and Hanwel ((Eriks, 2018), (Hanwel, 2018)). Furthermore, it is somewhat counterintuitive to use a component specifically designed to allow movement while the system does not require this freedom during normal operation. An advantage of this is that this design does allow thermal expension and vibration absorption.

This design requires the lower box to be revised slightly, such that the flanges on the toluene piping have a free movement path to inside the container.

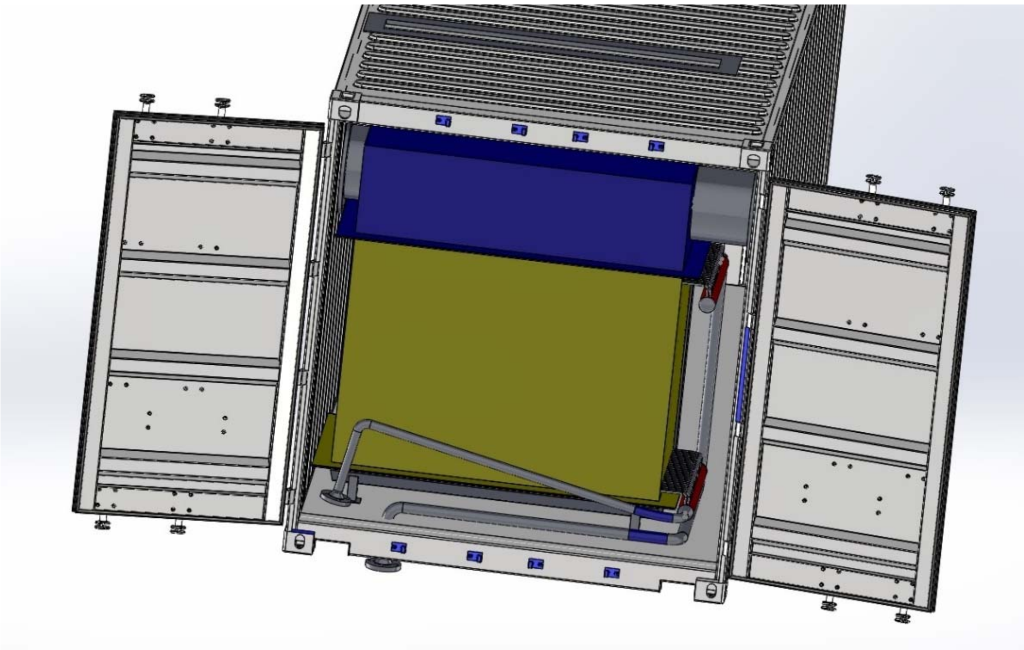


Figure 11.1: Toluene piping in transport state (L) and operation (R)

The third design is to use separate pieces of pipe that is mounted with flanges in a J-shape (upside down) between the E-box and the t-box. These pieces can be removed during transport.

This design would require the same general layout of the piping as the second design, since the piping must be accessible. The cut-out in the lower box must also be made.

11.8. Iterative design

By changing various components it may be needed to change the next component, which leads to another component being redesigned and so on. Because of this, multiple design versions have been made which all finetune the components as described above. For instance, the hot toluene piping in Figure 11.1 passes at the back of the cold toluene piping, such that it can pass underneath it at the front. That in turn allows this pipe to end up right of the cold toluene piping, while both pipes have enough length to allow movement. This configuration of connection points is chosen because it is easier to connect on the E-box side than a configuration where the cold side and hot side connection are switched.

11.9. Proposed design overview

The proposed design may now be

Table 11.1: Design parameters & performance of the proposed design

Design parameter	Symbol	Value
Tube setup type	-	Serpentines
Use of fins	-	No
Tube diameter	D_o	12mm
Tube inner diameter	D_i	10mm
Tube length	L	1750mm
Pipes per row	PPR	55
Rows	N_{rows}	34
Horizontal pitch	S_t	16.2mm
Vertical pitch	S_l	36mm
Height	h	1200mm
Toluene side pressure drop	ΔP_{tol}	0.34bar
Flue gas pressure drop	ΔP_{flue}	7.4mbar
Internal volume	V	276liter
Toluene side wall temperature	T_{wall}	328.8°C
Toluene mass flow	$\dot{m}_{tol}$	1.7776kg/s
Heat transfer	Q	1090kW

12. Production feasibility

The mechanical support and producibility has largely been ignored when coming up with the design, although it has played a role in the background of some design choices. However, once the proposed design was done, a consultant from an external company has looked into the design to see if this is feasible.

Based on this meeting three main conclusions were drawn:

- The design can be produced, in proper consultation with a manufacturer
- Additional support of the bundle is needed, see below
- The piping connection with compensators is possible, but separate connection pieces seem more logical.

The support of the bundle could be done by welding the serpentines to stainless steel slabs with a thickness of the horizontal gap width. In these slabs holes can be made, through which a pen is then ran, such that the rows are supported. See Figure 12.1 for a visualization.

For convenience it would be practical to make the gap width 4.0mm rather than 4.2mm, corresponding to a pitch ratio of 1.333. Looking at Table B.E.6 this option falls in between the thermally best option and the currently chosen option, meaning it will have a higher heat transfer at the cost of slightly increased pressure drops and wall temperature.

In Table 10.1 the performance of the proposed design and a design with 4.0mm gap width are compared.

Table 10.1: Performance of the proposed and more practical design

Parameter	Symbol	Proposed design	Practical option
Horizontal pitch	S_t	16.2mm	16.0mm
Toluene side pressure drop	ΔP_{tol}	0.34bar	0.34bar
Flue gas pressure drop	ΔP_{flue}	7.4mbar	8.3mbar
Toluene side wall temperature	T_{wall}	328.8°C	329.0°C
Toluene mass flow	$\dot{m}_{tol}$	1.7776kg/s	1.7862kg/s
Heat transfer	Q	1090kW	1096kW

Since the latter option is more practical to produce and has better performance while still meeting the requirements it seems wise to investigate that option further.

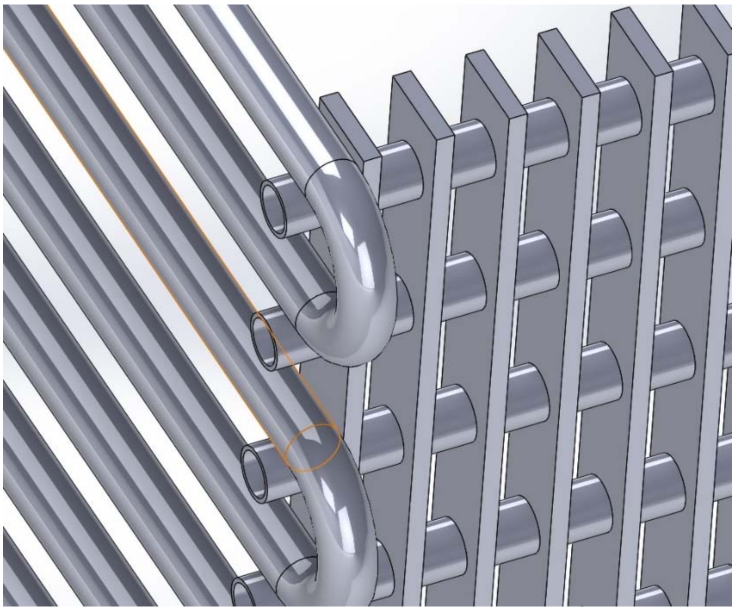


Figure 12.1: Possible support structure

Conclusions

The starting point was to design an evaporator that fits in a container, is small, allows the inlet hood to fit inside the container, and have a good thermal performance.

A proposed design is made, which performs nearly as well as the reference design (or even slightly better when the practical design is considered), while it now covers all requirements and it allows both the inlet hood and the toluene piping to fit inside the t-box. The design is also quite robust in the sense that the parameters that may be sensitive due to the way they are calculated have significant margin to their limit.

The design takes limited space in the lengthwise direction of the container, such that the condenser might take over that space.

The main gains of the design are size reduction and meeting all requirements. Also, the design offers a significantly higher toluene mass flow than the minimum requirement, even though this is nearly equal to the mass flow in the reference design.

Two viable solutions have been provided to the challenge of coupling the toluene piping between the e-box and t-box. Using compensators requires little work and is completely maintenance free on the t-box side. Separate connection pieces may also be used, which requires somewhat more work and inspection of the t-box may periodically be needed.

The complete design is also deemed feasible in the sense that it could be produced by suppliers.

Recommendations

A final choice should be made between the proposed design and the more practical design, as discussed in chapter 12.

Also, a decision has to be made on the piping connections to the e-box. Using separate connection pieces seems to be the most valid option here.

The mechanical design of the heat exchanger needs further attention, but the CSS, Solidworks designs and the design considerations from chapter 11 and 12 form the basis for this.

Due to the sensitivity of the pressure drops, particularly the flue gas pressure drop, additional attention should be given to this issue. The proposed design is made robust to ensure the maximum pressure drop is not exceeded, so if a more accurate model can be employed the maximum flue gas can be more closely approached, which potentially allows to choose thermally better setups.

References

Eriks. (2018, 06). *Slangen en Toebehoren*. Retrieved from Eriks: http://eriks.nl/documentatie/industri%C3%ABle-slangen-en-compensatoren/compensatoren/rx-compensatoren.pdf

Hanwel. (2018, 06). *Hanwel RVS Compensatoren*. Retrieved from Hanwel: https://www.hanwel.com/en/productcategorie/stainless-steel-expansion-joints/

Hiemstra, A. (2011). *A steady state evaporator model (for calculating and designing a standardised configuration).* Tri-O-Gen (internship company) / UT.

General use:

Cengal, Y.A. *Heat and Mass Transfer: a Practical Approach*
4th Edition in SI Units, ISBN-13: 9780071311120
Publisher: Mcgraw Hill Higher Education

A. The cycle

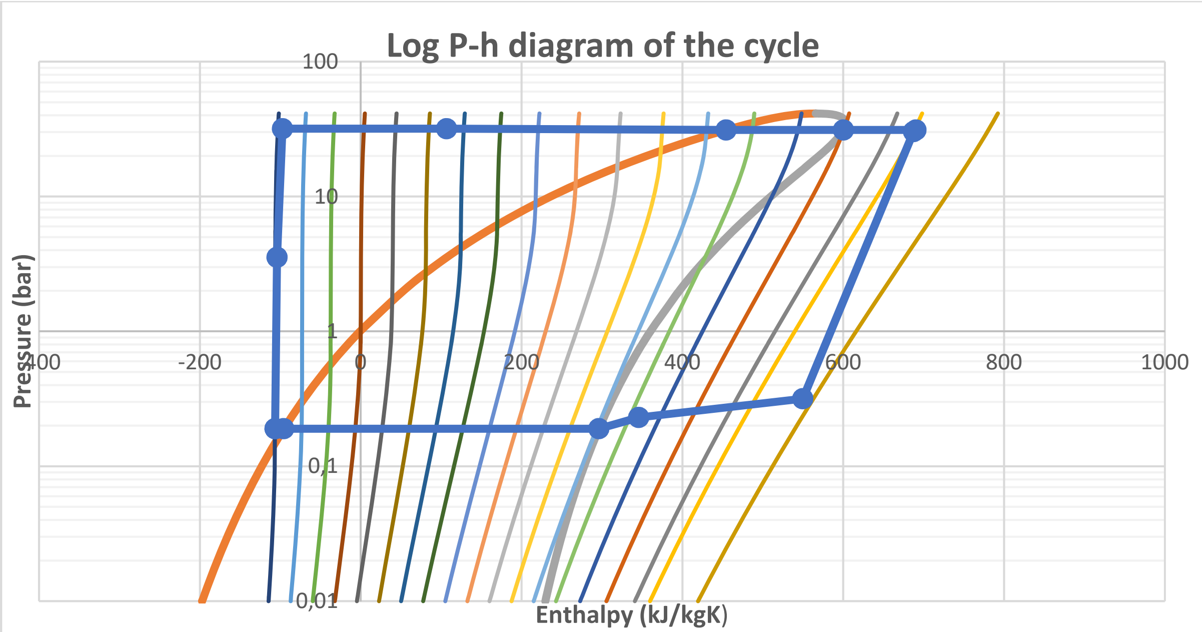


Figure A.1: log P-h diagram of the Organic Rankine cycle. Working fluid: toluene

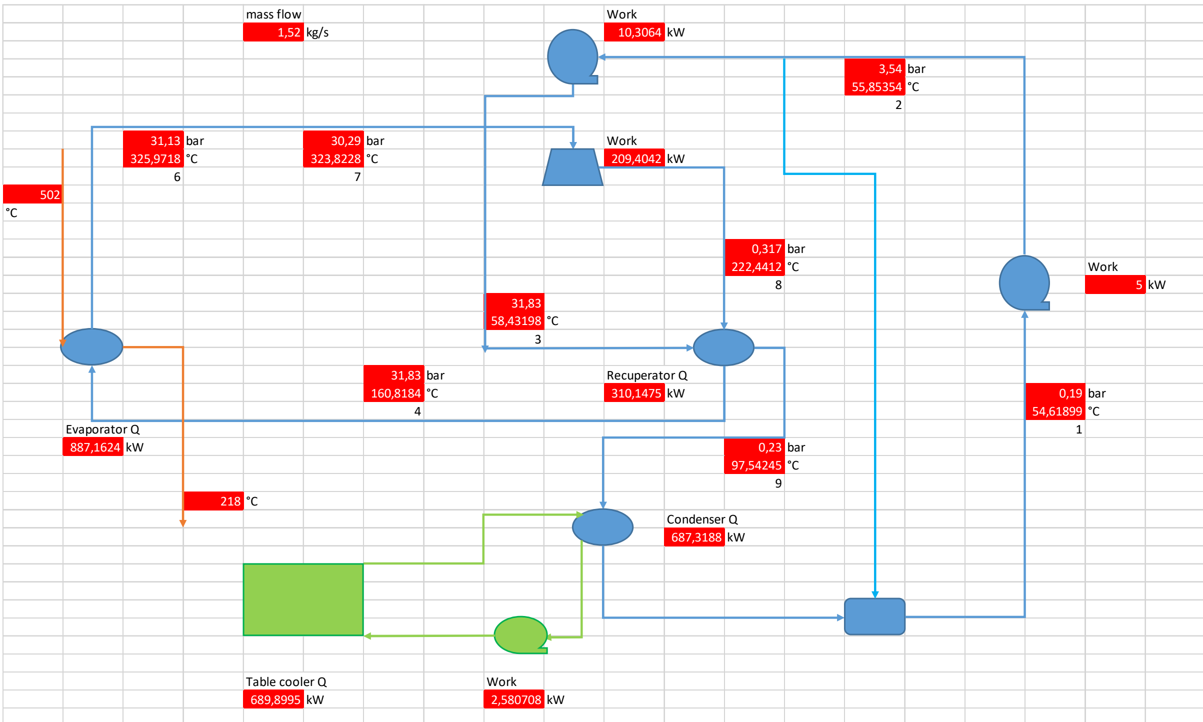


Figure A.2 Schematic overview of the cycle

Table A.1

Point	P (bar)	T (°C)	H (kJ/ kg)	S (kJ/kgK)	x	
1	0,19	54,6190	-106,4013	-0,2990	-	
2a	3,54	55,8535	-103,9260	-0,2927	-	
3a	31,83	58,4320	-97,1455	-0,2824	-	
4	31,83	160,8184	106,8989	0,2516	-	
5sl	31,13	295,9718	454,3649	0,9428	0	
5sv	31,13	295,9718	600,4277	1,1994	1	
6	31,13	325,9718	690,5583	1,3540	-	
7	30,29	323,8228	687,5925	1,3507	-	
8a	0,317	222,4412	549,8265	1,4739	-	
9	0,230	97,5424	345,7821	1,0313	-	
10	0,19	60,6190	296,2455	0,9077	1	
11	0,19	60,6190	-95,5407	-0,2662	0	
1	0,19	54,6190	-106,4013	-0,2990	-	
2s	3,54	55,8535	-103,9260	-0,2990	-	
3s	31,83	56,5538	-100,5358	-0,2927	-	
8s	0,317	189,4654	490,7840	1,3507	-	

B. Design parameter overview

Table B.1: Overview of design parameters that have been considered for the proposed design

Parameter	Design parameter	Theoretically optimal	Restrictions	Correlations
Pipe length	No/ partially	As long as possible	Container dimensions & HEX inlets	Increasing the pipe length decreases the pressure drop and Nusselt number.
Pipes per row	Yes	Complex	Too high may cause toluene degradation; too high gives low heat transfer. Too few pipes causes high DP flue	Toluene velocity and pressure drop, heat exchanger width (along with pitch).Flue gas velocity and pressure drop.
Pipe diameter	Yes	As small as needed.	Practical minimum, also affects	Toluene velocity, heat transfer, pressure drops.
Wall thickness	No/ semi	As small as possible	Pressure, production.	Ratio influence internal and external area.
Pitch configuration	Yes	Highest effective diameter, lowest mass velocity (rotated square)	Practical fabrication & serpentine layout	Flue gas velocity, effective diameter, horizontal & vertical pitch
Pitch ratio	Yes	Varies with PPR and diameter	Normally 1.25-1.4	HEX width and height, flow
Friction factor (material finish)	Difficult/ No	0	Producability & cost. 0.045mm/D for carbon steel	Flue gas pressure drop
Tube material	No	Proper thermal conduction (not infinitely high but properly high)	Toluene & flue gas use, temperature use,	Conduction in the tube wall
Total width/ frontal area	Over defining (PPR, pitch)	As high as possible for HEX performance, as low as possible for space optimization	Container dimensions	Follows from pitch, PPR and pipe length
Use of serpentines	Yes	Yes	Can this be supported?	The use of serpentines eliminates any “dead spots” on the toluene side and it would seem less costly.
Serpentine direction	Yes, but practicality must be considered	Depends on connections and dimensions, but rows should be horizontal	Vertical direction should be approx 2m, while other should be as small as possible.	Available space
Fins	Yes	To be used only if the flue gas side heat transfer is problematically poor.	The wall temperature at the toluene side may still not be too high, so the heat transfer at the flue gas side is limited by that at the toluene side.	Optimization of space use compared to no fins

C. Reference design

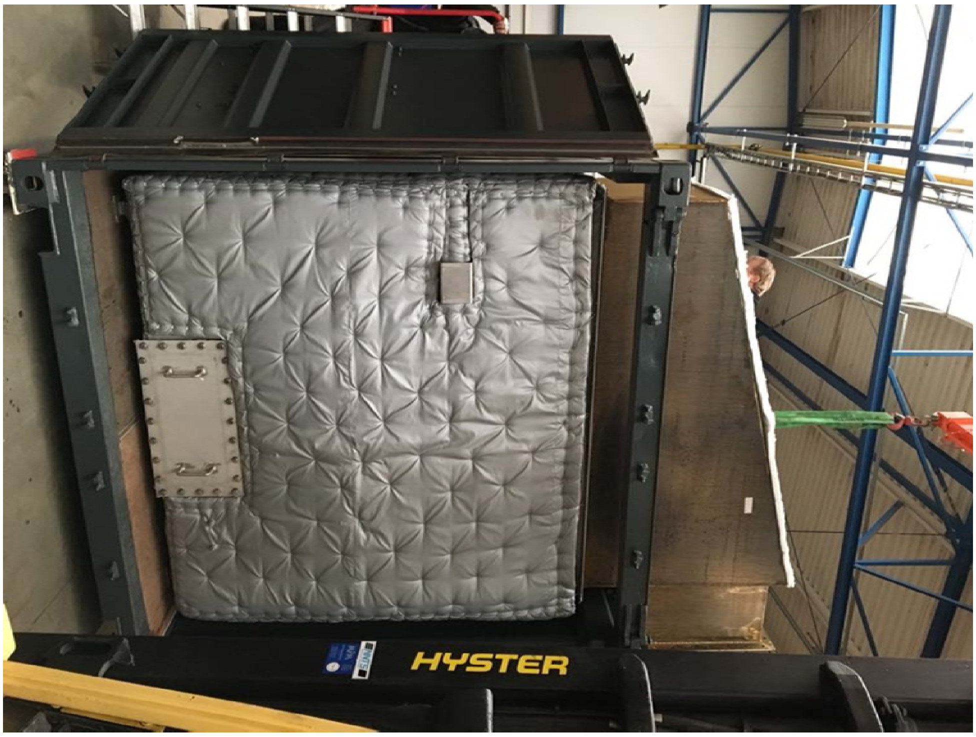


Figure C.1: Insulated reference design in the container (sideways). The inlet hood is clearly visible on top.

Figure C.2: Reference design without insulation in the container.

D. Optimal diameter determination

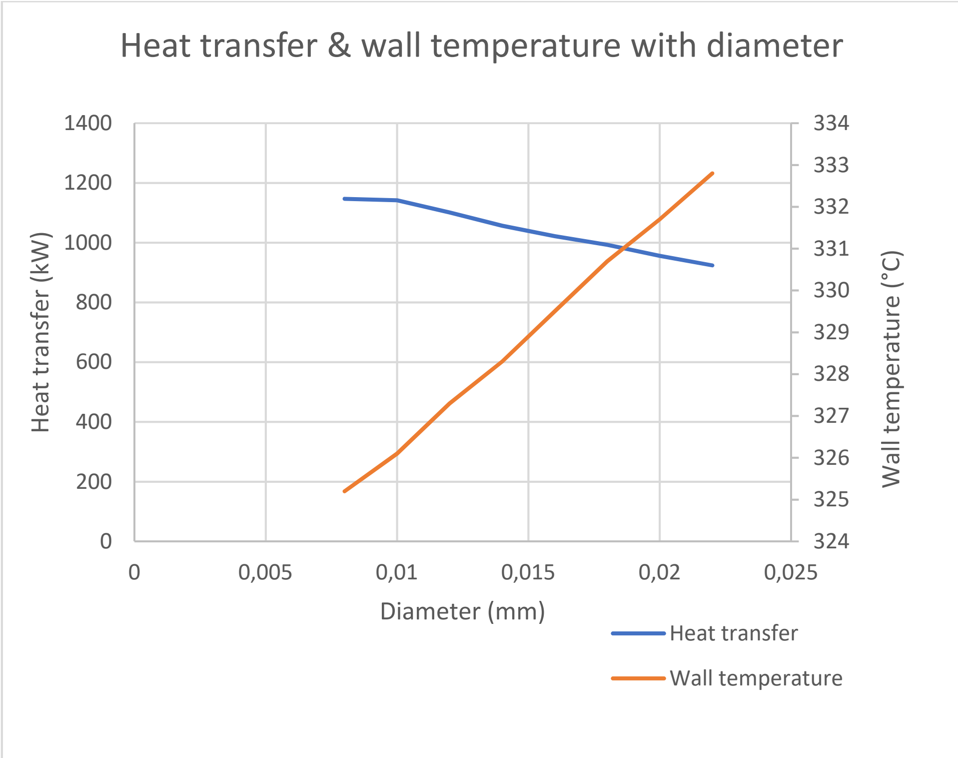


Figure D.1: Heat transfer and wall temperature for constant flue gas inlet area

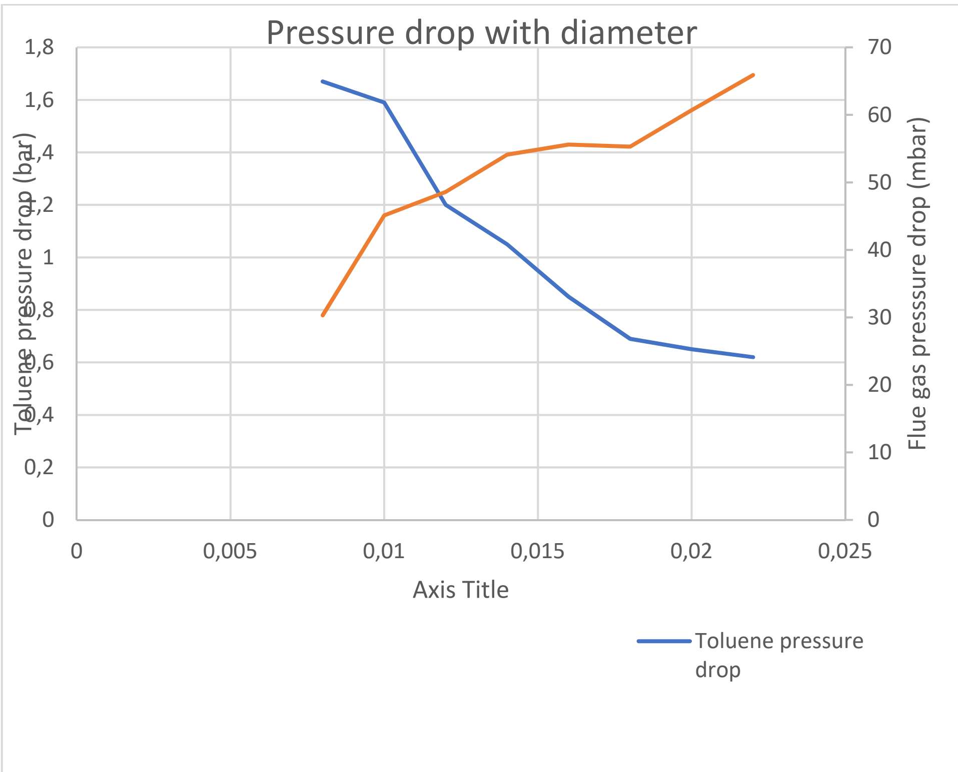


Figure D.2: Toluene and flue gas pressure drop for constant flue gas inlet area

E. Performance and feasibility of various designs for 12mm tubes

Table B.E.1

T_{wall} (°C)	→ pitch ratio			T_{max} = 330°C				
↓ PPR	1,25	1,3	1,35	1,4	1,45	1,5	1,55	1,6
30	329,3	328,4	327,7	327,1	326,6	326,2	325,9	325,5
35	329,6	328,7	327,9	327,4	326,9	326,4	326,1	325,7
40	329,9	328,9	328,2	327,6	327,1	326,6	326,3	325,9
45	330,1	329,2	328,4	327,8	327,3	326,8	326,4	326,1
50	330,4	329,4	328,6	328	327,4	327	326,6	326,2
55	330,6	329,6	328,8	328,2	327,6	327,1	326,7	326,4
60	330,8	329,8	329	328,3	327,8	327,3	326,9	326,5

Table B.E.2

Flue gas pressure drop (mbar)	→ pitch ratio			Maximum pressure drop = 15mbar				
↓ PPR	1,25	1,3	1,35	1,4	1,45	1,5	1,55	1,6
30	54	36,7	26,1	19,3	14,6	11,3	9	7,2
35	39,5	26,7	18,9	13,9	10,5	8,1	6,4	5,1
40	30,1	20,3	14,3	10,5	7,9	6,1	4,8	3,8
45	23,7	15,9	11,2	8,2	6,2	4,8	3,7	3
50	19,1	12,8	9	6,6	4,9	3,8	3	2,4
55	15,7	10,5	7,4	5,4	4	3,1	2,4	1,9
60	13,1	8,8	6,2	4,5	3,4	2,6	2	1,6

Table B.E.3

Toluene pressure drop (bar)	→ pitch ratio			Maximum toluene pressure drop = 2bar				
↓ PPR	1,25	1,3	1,35	1,4	1,45	1,5	1,55	1,6
30	0,9	0,88	0,87	0,85	0,84	0,83	0,81	0,8
35	0,71	0,7	0,68	0,67	0,67	0,66	0,65	0,64
40	0,58	0,57	0,56	0,55	0,54	0,54	0,53	0,53
45	0,48	0,48	0,47	0,46	0,46	0,45	0,45	0,44
50	0,41	0,41	0,4	0,4	0,39	0,39	0,39	0,38
55	0,36	0,36	0,35	0,35	0,35	0,34	0,34	0,34
60	0,32	0,32	0,31	0,31	0,31	0,3	0,3	0,3

Table B.E.4

Width (mm)	→ pitch ratio			Maximum width =900mm				
↓ PPR	1,25	1,3	1,35	1,4	1,45	1,5	1,55	1,6
30	451	468,4	485,8	503,2	520,6	538	555,4	572,8
35	526	546,4	566,8	587,2	607,6	628	648,4	668,8
40	601	624,4	647,8	671,2	694,6	718	741,4	764,8
45	676	702,4	728,8	755,2	781,6	808	834,4	860,8
50	751	780,4	809,8	839,2	868,6	898	927,4	956,8
55	826	858,4	890,8	932,2	955,6	988	1020,4	1052,8
60	901	936,4	971,8	1007,2	1042,6	1078	1113,4	1148,8

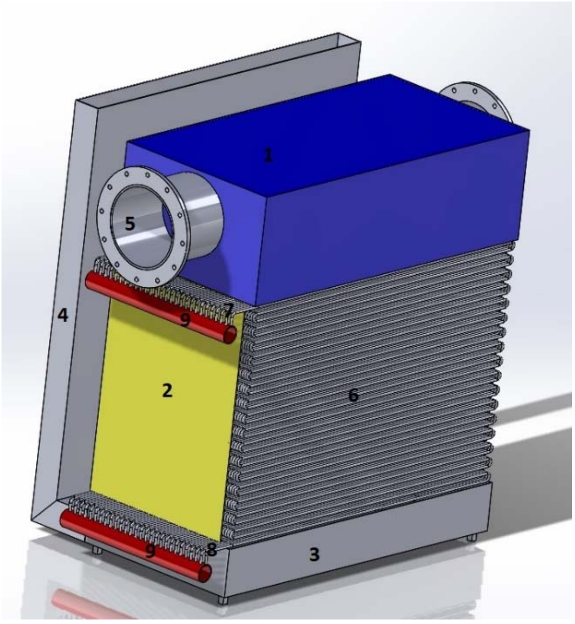
Table B.E.5

Volume (liter)	→ pitch ratio			Maximum volume =950 liter				
↓ PPR	1,25	1,3	1,35	1,4	1,45	1,5	1,55	1,6
30	150,453	150,453	150,453	150,453	150,453	150,453	150,453	150,453
35	175,529	175,529	175,529	175,529	175,529	175,529	175,529	175,529
40	200,604	200,604	200,604	200,604	200,604	200,604	200,604	200,604
45	225,68	225,68	225,68	225,68	225,68	225,68	225,68	225,68
50	250,755	250,755	250,755	250,755	250,755	250,755	250,755	250,755
55	275,831	275,831	275,831	275,831	275,831	275,831	275,831	275,831
60	300,906	300,906	300,906	300,906	300,906	300,906	300,906	300,906

Table B.E.6

Duty (kW)	→ pitch ratio			Theoretical maximum: 1226kW				
↓ PPR	1,25	1,3	1,35	1,4	1,45	1,5	1,55	1,6
30	1065	1042	1021	1003	986	970	955	941
35	1081	1060	1040	1023	1006	991	977	963
40	1095	1074	1056	1039	1023	1009	995	982
45	1106	1087	1069	1053	1038	1024	1011	998
50	1116	1098	1081	1065	1051	1037	1024	1012
55	1125	1107	1091	1076	1062	1049	1036	1025
60	1132	1115	1099	1085	1071	1059	1047	1036

F. Solidworks models



Component number	Description
1	Top box (inlet hood)
2	Bundle casing
3	Bottom box
4	Exhaust
5	Flue gas inlets
6	Bundle/ serpentines
7	Toluene outlet side
8	Toluene inlet side
9	Distributors

Figure F.1: Global evaporator layout and component naming

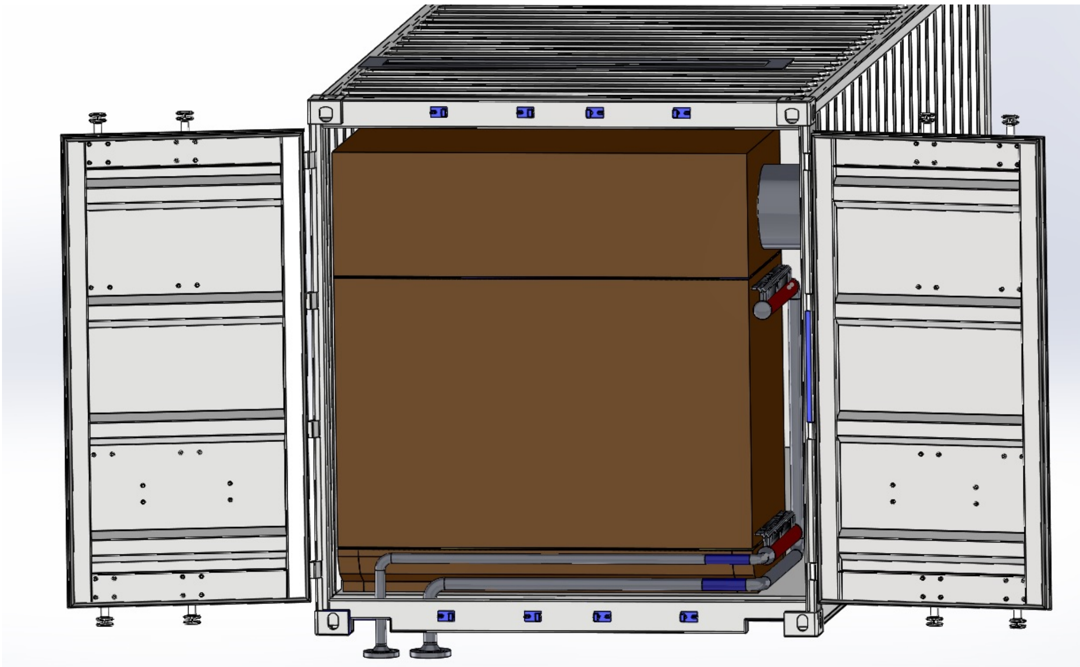


Figure F.2: Final design inside a shipping container

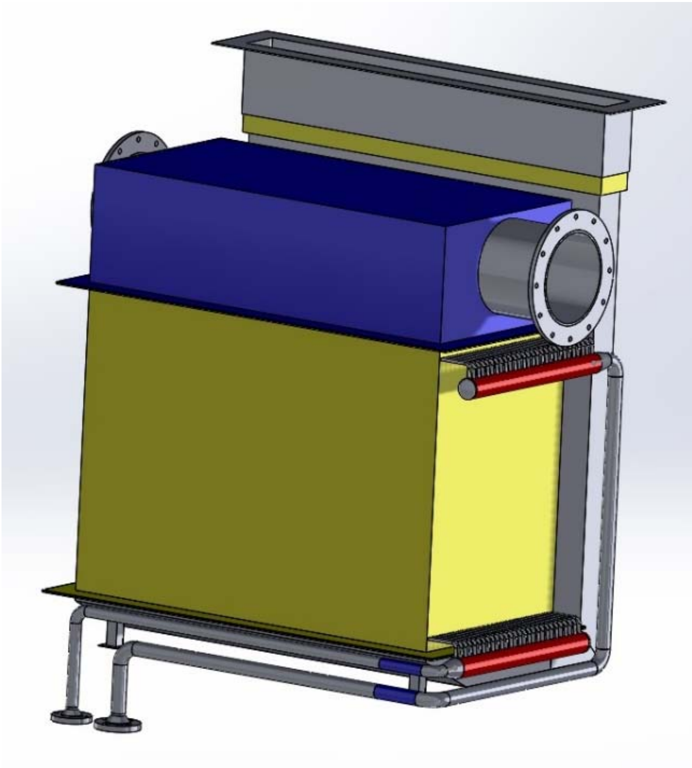


Figure F.3: Final design without insulation

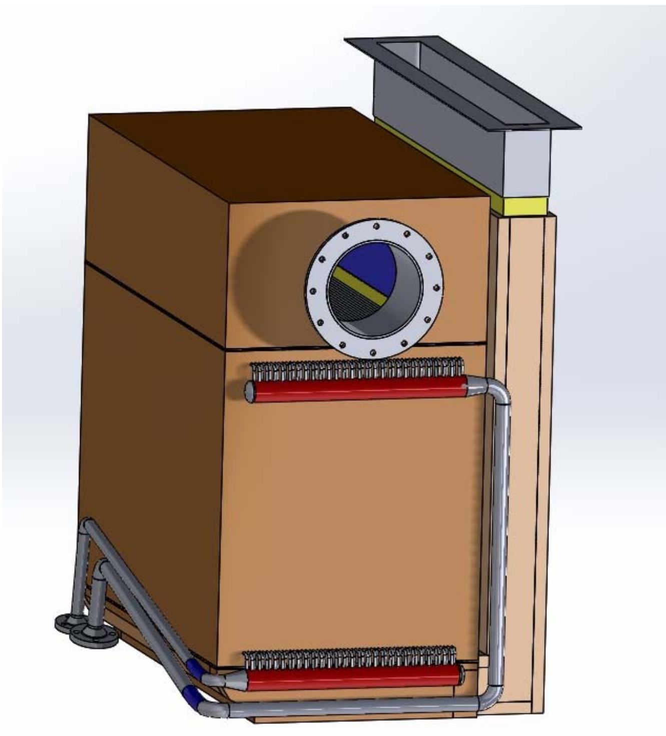


Figure F.4: Final design with insulation. The piping is shown in transport state to demonstrate that the flanges fit next to the insulation

G. Pipe strength

One of the design criteria is that the chosen tubes (12mm outer diameter, 1mm wall thickness) are able to withstand the pressure difference between the toluene on the inside and the flue gas on the outside. The bursting pressure and normal working pressure can be calculated using Barlows formula:

$$P = \frac{2 * \sigma_y * t}{D_o * f_s}$$

The bursting pressure can be found by using the ultimate tensile strength for σ_y and using a safety factor f_s of 1. The working pressure can be found by using the yield strength for σ_y and applying a safety factor (generally 1.5).

Using a tensile strength of 515MPa and a yield strength of 205MPa, the following pressures are obtained:

$$P_{burst} = \frac{2 * 515MPa * 1mm}{12mm} = 85.83MPa = 858.3bar$$

$$P_{working} = \frac{2 * 205MPa * 1mm}{12mm * 1.5} = 22.77MPa = 227.7bar$$

All pipes must be rated PN100, due to the combination of high temperature and high pressure. Along with a slight safety margin this causes the first PN (pressure nominale) rating of PN64 to be insufficient.

The distributor pipes, serpentines and the toluene inlet and outlet piping must all be rated to this pressure. The serpentines have been calculated to withstand a pressure of 227.7bar, which is widely sufficient. For the distributor and inlet and outlet piping a minimum wall thickness can be calculated to see if the PN100 standard is met.

$$t_{min} = \frac{D_o * f_s * P_N}{2 * \sigma_y}$$

$$t_{min,dist} = \frac{88.9mm * 1.5 * 10MPa}{2 * 205MPa} = 3.26mm$$

$$t_{min,DN40} = \frac{48.3 * 1.5 * 10MPa}{2 * 205MPa} = 1.77mm$$

$$t_{min,DN50} = \frac{60.3mm * 1.5 * 10MPa}{2 * 205MPa} = 2.21mm$$

A safety factor of 1.5 has been taken here in the thickness calculation, along with the safety on the PN choice itself.

H. Company profile & self reflection

Company profile

Tri-O-Gen company profile

Tri-O-Gen is a company that designs and builds Organic Rankine Cycles. These ORCs, which function as small power plants, use relatively low temperature heat to produce electricity.
Tri-O-Gen mainly focusses on fuel save and on power increase of bio fuel generators.

Tri-O-Gen was founded in 2001, in Goor, the Netherlands. Approximately fifteen people work at the company, in various functions. The company develops their product themselves, while most of the components are bought from suppliers. Assembly is done by Tri-O-Gen again.

The business model mainly consists of leasing ORCs, while selling is also possible. Products get sold or leased directly to end users (that use the electricity themselves), although the option of supplying to 'middlemen' is currently also being explored, where Tri-O-Gen would sell the ORCs to companies that provide power supply for end users.

Tri-O-Gen also provides maintenance and support for the ORCs during operation.

Self reflection

Self reflection

The assignment I've been doing at Tri-O-Gen was due it is nature quite individualistic. I've had contact with my supervisor as well as other colleagues and interns about the assignment, but I've also been doing quite a lot on my own. I think that I could have been more extrovert in my communication towards my colleagues, especially my supervisor, but when necessary I have made sure to communicate openly about problems, interactions to other projects and background information.

An interesting aspect of this assignment is that, during my internship, two other interns were also working on other parts of the e-box and t-box. Since I started last, I have tried to adapt my design such that it would not interfere with their ideas, such that Tri-O-Gen can eventually try to combine these three designs.

For me, the meeting with an external party to see if the proposed design was feasible was an exciting and interesting meeting. This meeting gave me the opportunity to confirm that what I had done up until then was useful, and that the proposed design may eventually be picked up by Tri-O-Gen.

An issue I encountered during the internship is that I have somewhat little practical technical know-how. The use of piping flanges, reinforcement of the evaporator, and so on, is somewhat lesser known territory for me, whereas it is of great value in business.

Furthermore, when I started the internship I was not sure whether I should be doing an internship for forty hours a week (especially due to my epilepsy). However, I decided to give it a try and see how long I could maintain this. This did force me to really change my pattern, and at some points it has been quite heavy on me, but in the long run I'm glad I've (successfully) tried to maintain a full workweek rhythm.

Perhaps the most positive note on this assignment for me is that I feel like I have delivered useful work, in multiple aspects, that has potential to help Tri-O-Gen in the future. I really enjoy having results at the end of such a process.