Final Report Combined Heat and Power Plant

Groep B1N 2009/2010 Semester 1

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1 Introduction

This report details the design process for a combined heat and power generation system per the specifications submitted. All design decisions will be motivated by a comparison to any similar alternatives, explaining the pros and cons of the current choice. The design submitted here is by no means the only possible solution to the requirements, but represents the best design based on our understanding of the design requirements. Emphasis during the design was placed on overall efficiency balanced against the financial and environmental concerns expressed.

A hospital is an ideal institution for an independent energy, steam and hot water supply system. Not only for economic reasons, but a reliable and consistent power supply allows the administrators to worry about one less thing when it comes to providing the best possible care to their patients.

There could be environmental savings as well. Heat is produced locally so the transport losses are minimal. Furthermore, production follows demand so the system works at maximum efficiency at most times.

2 Design Requirements and Constraints

The demand of the CHPP is modeled as shown in Table 1. The CHPP will provide steam, warm water and electrical power to the client in 4 variations. The demands on the system vary from a day and night-time requirement and a difference between summer and winter demand periods.

Table 1 Hospital power and water demand

Time \ demand (kW)	Electrical	Warm Water	Steam
Winter daytime	22500	3000	5000
Winter nighttime	15750	2100	3500
Summer daytime	11000	1000	1500
Summer nighttime	7700	700	1050

The system needs to supply the electrical energy in 3 phase AC at 440V with a frequency of 50Hz, the thermal energy in the form of a constant flow of 70ºC water and a separate supply of saturated steam at 165ºC. The focus of the design is on maximum efficiency, maximum reliability and minimal downtime. The base power generator should be either a diesel engine or a gas turbine. A steam cycle will be added to increase the overall efficiency of the system. A natural coolant source (sea water) is available, with a restriction of a maximum temperature increase of the coolant of 5ºC.

3 Design Options

3.1 Diesel Engine vs. Gas Turbine

The first design decision that needed to be made was what the main power source for the system would be. We quickly found the most usable options are a diesel engine or a gas turbine. We set out to compare both possibilities. We took the 'general' pros and cons of the two choices and used this to motivate our choice.

A diesel engine is an internal combustion engine that uses spontaneous combustion of its work medium to drive the reaction. A gas turbine is the main element in the Brayton cycle, using the compression and then expansion (most commonly through internal combustion) of its work medium, air, to generate angular mechanical energy. In a head-to-head comparison, the diesel engine has heavier components, where the gas turbine seems to have more components that require regular maintenance/inspection and could be prone to breaking down (the gas turbine being the more complex of the two). The overall efficiency is hard to determine since it very dependent on the rest of the process, so this will be examined largely based on the examples we could locate. Finally, the diesel engines are reported to be more efficient at constant high RPMs, where the gas turbine is better suited to handling variable loads efficiently.

We looked at our specific design requirements and made them definitive for the selection. Here the use of several Diesel engines won out against the single gas turbine model. The advantages are of maintenance (being able to perform maintenance on components without the system being completely shut down), fewer specialized components (in the spirit of less that could break or go wrong) and performing at peak efficiency during the variable loads by switching off the unneeded engines and keeping RPMs high on the engines that are on compared to throttling the gas turbine.

3.2 Single Heat Recovery System vs. individual components:

A design with one large heat recovery system can be chosen. Or a system with discrete, individual components for the various heat recovery tasks can be taken. The benefits of a large heat recovery system are combined maintenance of its components(maybe more frequent, but shorter overall times), compacter system (though space is not a requirement as such, in the case of heat loss, compactness saves on areas where energy could be lost) versus the potential gain of adding components where they are needed (space economy as well as saving potential heat loss through piping) and the 'mix and match' of optimal components for each task (although this is not by definition impossible with the single system).

3.3 Secondary/auxiliary steam turbine:

The design requirements call for a secondary/auxiliary steam turbine to be used. The question therefore does not become which or what, but where to place it to get maximum efficiency out of our system. We elected to place it in a separate 'loop' where it can provide electrical power from the excess energy in the steam before it is sent to the hospital. Other design possibilities considered placing the steam turbine in a sole 'loop' where the steam was then fed back into the reservoir before going to the hospital or placed into the hospital 'loop' after the steam returned from hospital. These configuration were found to be less efficient or more complicated for no extra gain.

3.4 Back-up systems for supplying the hospital:

The system will be down for maintenance, this is unavoidable. Also, should a mechanical or other form of failure occur, the hospital's supply must not be compromised. On the assumption that the hospital can still provide itself with electrical energy from the national energy net, we provide only a back-up system for the steam and warm water. This system is basically a parallel reservoir of water and steam that doesn't go through the heat exchanger, but goes through a 'simple' boiler that heats the water to the desired temperatures. The efficiencies of this system are considered outside of the scope of this project, but we assume the choice of back up boiler will be one that can provide the required energy at the minimum of waste heat and energy.

3.5 Concepts

The decide the concept of the design, two different concepts are researched. These concepts are explained in the following section.

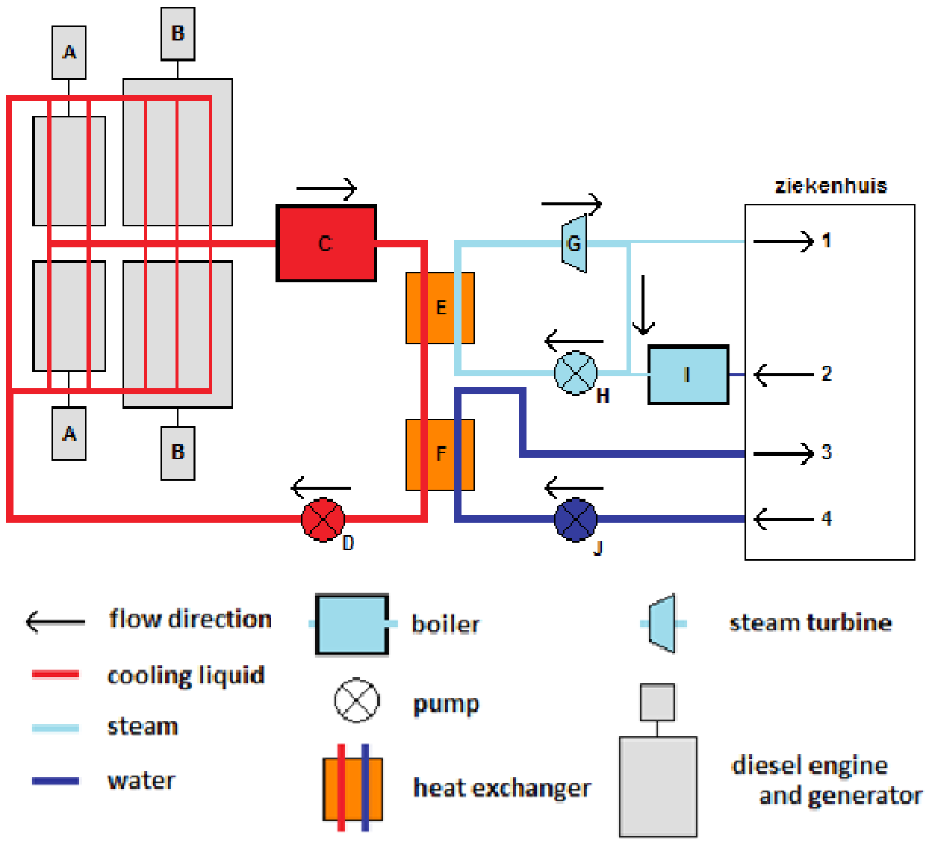


Figure 1 Diesel engine concept

The Diesel Engine concept uses 4 diesel combustion engines to create mechanical energy as well as exhaust gasses with re-usable thermal energy. See Figure 1 for a schematic overview of the Diesel Engine concept.

This configuration has the benefit that the generators can be turned on individually to comply with different demands. The main advantage of this is that the possibility to let all engines run on their most efficient RPM.

The exhaust from the diesel engines is then used to create steam for the hospital supply. The engine coolant is also routed through a heat exchanger to heat water that is also fed to the hospital. Later the coolant's temperature is further decreased by another heat exchanger, this time fed by sea water that ensures optimal cooling of the engines.

The boilers marked as C and K are back-up boilers. In the (unlikely) case that the diesel engines are not functioning, the hospital can still receive it's required supply of water and steam. A possible addition to this concept is a steam turbine, the steam cycle can be combined with that of the hospital's. The diesel engine coolant stream is also required to be cooled further by means of passing through another heat exchanger with the sea water that is available between pump D and heat exchanger F.

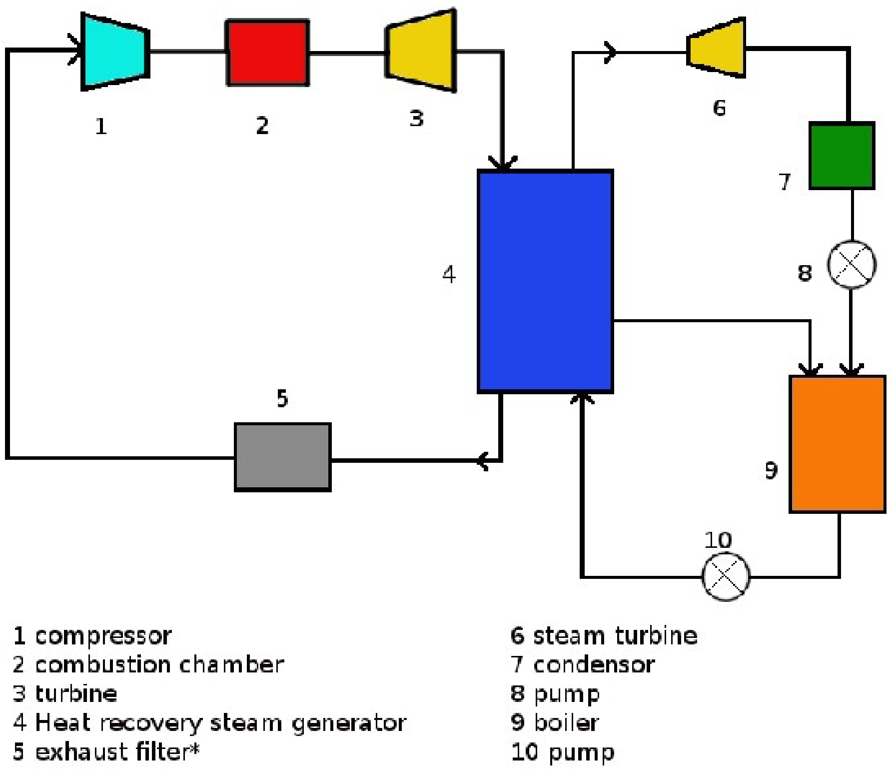


Figure 2 Combined Cycle concept

The Combined cycle concept is akin to a combined-gas-and-steam (COGAS) power plant used in marine propulsion, however it uses turbo electric transmission as opposed to direct drive shaft powering. Figure 2 shows a simple overview of the combined cycle, without the outputs to the hospital in terms of electrical power, steam and heated water.

The diagram in Figure 2 shows two distinct loops, one showing the Brayton air cycle (left of the HRSG) and the other showing a variation of the Rankine cycle which is uses water/steam as its work medium. This water and steam will also be branched off at the relevant points to supply the hospital with its required demands. The exhaust filter shown in the Brayton cycle (number 5) is an instance that is provided to show the manner in which the cycle is actually a closed system. In actuality it would be much more practical to vent the hot combusted air to the atmosphere and to intake 'clean' atmospheric air at another point. This ensures optimal conditions for the combustion chamber (number 2). It is known from the literature that the exhaust gasses from a gas turbine are high temperature (approximately 900-1100 degrees C) and low velocity, these can be used to heat up other work mediums. The second medium, water, that flows through the Rankine cycle can be heated through the HRSG to superheated steam, which then flows through a steam turbine and can then be partially syphoned off to fulfill the requirements of the hospital.

The combustion chamber (2) can be a method of warming the air through internal combustion (using the air as one of the components of the combustion with a combustible fluid like diesel, kerosene or gasoline) or external combustion (the combustion of another medium outside of the cycle and then the transfer of this heat to the cycle's work medium). The main benefit of the external combustion also leads to its main drawback. A cleaner work medium allows for less wear of turbine and other cycle components, however this comes at a cost of a less efficiently heated work medium in the combustion chamber. Since space efficiency is not as paramount a requirement here as in the production of aeronautical and marine propulsion systems, the consideration of an external combustion system can be seriously considered.

3.6 Non-technical requirements

Aside from the power requirements needing to be fulfilled, it is important to realize that the choice of concept is also based on economic, environmental and other factors. Knowing this we will briefly compare the two concepts, asserting in advance that the final decision is based on compromise and the specifics of the client's situation.

The diesel engine system is commonly used in emergency generators. It is a system that has been used to generate supplemental power for a number of decades in civilian applications, thus making the technology tried and true. It also has the benefit of being a durable design, meaning that the components are less prone to failure. For example, the engine contains no spark plugs that can cause knocking (the badly timed combustion of the fuel) or failure.

The combined cycle uses two different turbine sets (one for the air/combustion exhaust and one for the steam) as its main source of electrical power. A downside to this concept is the difficulty of providing 50% of the max power output, while maintaining efficiency.

The lifetime of both concepts is approximately 30-40 years. Provided regular maintenance is performed. Diesel engine components are relatively cheap, but bulky compared to the turbine and compressor blades in the combined cycle. The operating costs (looking purely at fuel consumption, since maintenance and personnel costs are assumed to be similar) of the diesel engine however is slightly higher, due to its higher fuel requirement.

Both systems use combustion of fossil fuels as their main source of heating their primary work medium. This means that the use of either system brings with it the knowledge that it is contributing the amount of CO_2 in the Earth's atmosphere. Also, both systems require cooling. As efficient as the excess heat is used, there will still be the need to keep certain components cool and this means the use of sea water to cool them. These two environmental factors are negligible on the global scale in terms of their contribution to global warming, but naturally 'doing your part' and minimizing the so-called carbon footprint is important (currently more from a marketing/PR perspective than a technical one).

3.7 Conclusion

After evaluating the concepts above as a whole and their consistent parts, it is decided that a hybrid Design with diesel engines might serve the client best. The concepts advantage of being able to always run the engines at their most efficient RPM, as well as its overall reliability (which could be a financial benefit in terms of fewer replaceable parts, less downtime and fewer maintenance visits) makes up for the bulkiness and higher average operating costs. The HRSG component of the combined cycle can also be altered slightly to make use of the exhaust and coolant from the diesel engine. Theoretically allowing for a similarly high efficiency in the production of steam and extra electrical power, very comparable to the Rankine cycle section of the combined cycle.

4 Heat recovery system

The heat recovery system is the basis for the CHP plant design. The working principles had to be clear before the design could be worked out further. In this section the outlines of the system is discussed on which the complete CHP plant design is based.

A CHP plant is not complete without a proper Heat Recovery System. Our HRS is based on the system diagram Figure 3 given on page 74 of (1). This system is designed to supply both saturated and superheated steam. Since this system does not completely fulfill our demands, some adjustments have been made:

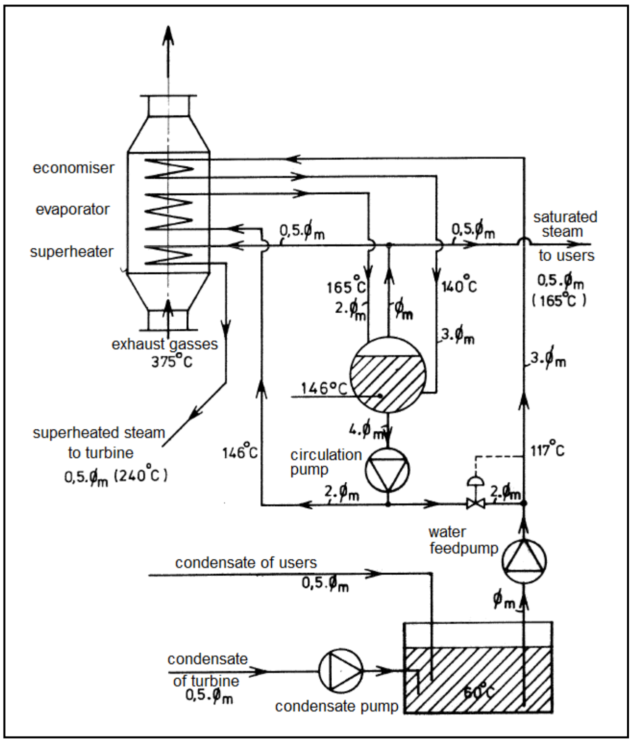


Figure 3 Heat recovery system providing steam

1. The ratio between saturated and superheated steam
2. An additional back-up boiler for saturated steam
3. An additional cycle for hot water
4. The temperature of the primary water tank
5. An additional heat source to heat the primary water tank
6. An additional back-up boiler for hot water
7. In the original system the same amount of saturated as superheated steam is supplied. This does not compete with our demands. However, we only know the mass flow required to fulfill the steam demands of the hospital. Further calculations will show how much superheated steam can be created to power the steam turbine.
8. Since we are designing for a hospital, where lives are at stake, there is no room for error. If, however, the engines do fail the heat supply for the entire HRS will be gone. To make sure steam can always be supplied a boiler will be added for emergency situations.
9. The hospital also requires hot water. Therefore a new cycle will be added to the system. Water will be pumped out of the primary water tank directly to the hospital and then back to the tank.
10. The hot water for the hospital requires a temperature of 70° Celsius and since this water will be directly tapped from the primary water tank, the water tank will be kept at a constant 70° Celsius. This will also slightly increase the temperature of the water heading towards the economizer.
11. Following the changes concerning the primary water tank, an additional heat source has to supply the extra heat required for this system to work properly. We will use the cooling water for the engines to supply this heat. This water will transfer the heat from the engines to the primary water tank and it will pre-heat the water leaving the hospital, before it enters the water tank. This will make sure the water in the tank stays at a constant 70°Celsius.
12. An additional boiler will be installed to provide the required heat to the hot water in case the engines fail to do so.

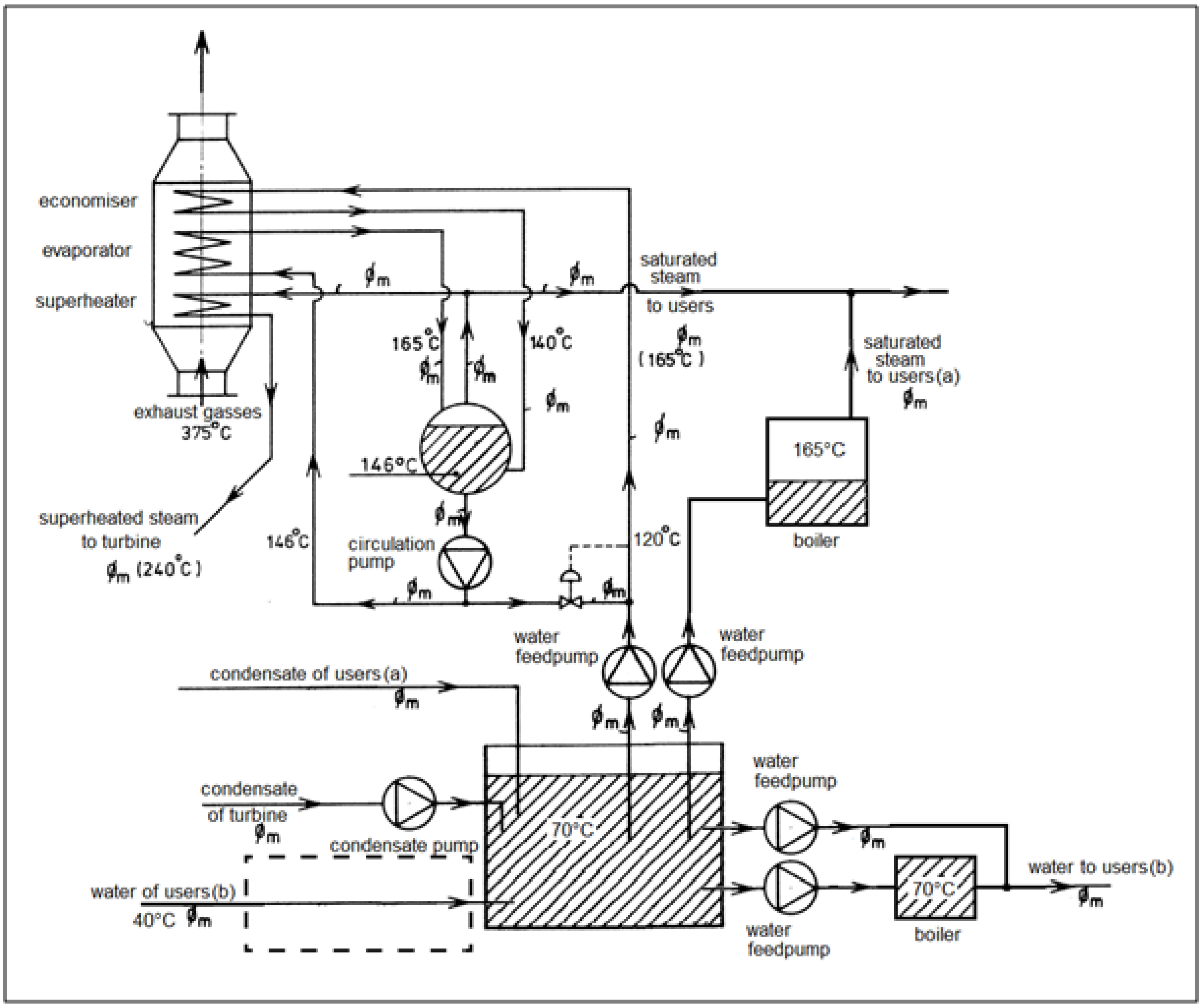


Figure 4 Heat recovery system providing steam and hot water

Figure 4 shows the adjusted system. The mass flows are yet to be calculated and are different for each operating state (winter/summer, day/night). The dotted-lined rectangle on the bottom-left of the image is where the heat exchanger for the cooling water will be placed. How this heat exchanger will work will be clarified later.

Since the only changes to the original system (the other adjustments are additions to the system) are the temperature of the primary water tank and the ratio between saturated and superheated steam, the other ratios are kept the same and therefore the other temperatures in the system remain the same as well.

Bear in mind that the two segments containing a boiler will only be used in case of emergencies, such as engine failure.

5 Diesel engines

To continue the details of the design, a choice had to be made regarding the main engines powering the system. A critical problem regarding engine choice is the deviation in demands between different operating states (summer/winter, day/night). To overcome this problem the choice was made to use multiple engines which can be turned on and off individually. This way all engines, when operative, can run at their most efficient state at all times.

Two different types of engines are chosen to power our plant. The two larger engines will be of the type W16V32 by Wärtsilä (2) and the smaller engines will be Daihatsu's 16DK-28 (3).

Figure 5 Wärtsilä 16V32

Figure 6 Daihatsu 16DK-28

To design a combined heat and power plant (CHP) it is important to know how much energy an engine can deliver and how this energy is divided between multiple mediums. The mediums that are of significant use for a CHP are the electrical output, exhaust gas, high temperature circuit (HTC) and low temperature circuit (LTC). Wärtsilä provides all the data required to create an energy balance for the engine. Daihatsu does not. However, since both engines are in the same order of magnitude it is assumed both engine have a comparable system for the exhaust system and cooling circuits.

5.1 Exhaust

For the Wärtsilä 16V32 diesel engine, the exhaust flow is specified as 14.0 kg/s with a temperature of 350°C. Daihatsu specifies the exhaust temperature at 348°C and the volume flow of the 16DK-28 diesel engine as 28892 Nm^3/h. To transform this value to kg/s the density of the exhaust gas has to be known. To calculate this, the densities of air and diesel are required as well as the air-fuel ratio (AFR). Wärtsilä gives enough data to calculate the AFR and it is assumed the AFR is equal for both engine types. First we need to now the fuel consumption, using the electrical power output, the electrical efficiency and the energy density of diesel as specified by Wärtsilä:

$$\dot{m} = \frac{P_{electric}}{\eta_{electric} \cdot u} = \frac{7124}{0.458 \cdot 42.7} = 0.364 \frac{kg}{s}.$$

Using the mass flows of the fuel input and exhaust output the AFR can be calculated:

$$AFR = \frac{\dot{m}_{exhaust} - \dot{m}_{fuel}}{\dot{m}_{fuel}} = \frac{14 - 0.364}{0.364} = 37.5.$$

Now we can calculate the density of the exhaust gas using the densities of air at 25°C and 1 atm and of the used fuel (diesel oil). These densities are respectively 1.186 kg/m^3 and 836.6 kg/m^3:

$$\rho_{exhaust} = \frac{\rho_{fuel} \cdot 1 + \rho_{air} \cdot AFR}{1 + AFR} = \frac{836.6 + 1.186 \cdot 37.5}{1 + 37.5} = 1.217 \frac{kg}{m^3}.$$

This gives the following mass flow for the exhaust of the Daihatsu engine:

$$\dot{m} = \dot{V} \cdot \rho = \frac{28892}{3600} \cdot 1.217 = 9.77 \frac{kg}{s}.$$

Now the mass flows of both engines are known, the amount of usable energy can be calculated. The HRS is based on the schematic shown in Figure 4. For this design the exhaust fumes will be used in the range of 350/348°C to 160°C. The specific heat of exhaust fumes in the range which the gas is used the average specific heat is $u = 1.07 \frac{kJ}{kg \cdot °C}$. This gives the following energy flows:

Wärtsilä 16V32:

$$\dot{Q}_{exhaust} = u \cdot \Delta T \cdot \dot{m} = 1.07 \cdot (350 \ - 160\ ) \cdot 14.0 = 2816\ kW.$$

Daihatsu 16DK-28:

$$\dot{Q}_{exhaust} = u \cdot \Delta T \cdot \dot{m} = 1.07 \cdot (348 \ - 160\ ) \cdot 9.77 = 1986\ kW.$$

5.2 High temperature circuit

For the Wärtsilä diesel engine the amount of energy released in the high temperature circuit is $\dot{Q}_{HTC} = 2453\ kW$. It is assumed the Daihatsu engine has a similar system and therefor releases a similar amount of energy compared to its electrical output and efficiency:

$$\dot{Q}_{HTC,Daihatsu} = \ \dot{Q}_{HTC,Wärtsilä} \cdot \frac{P_{Daihatsu}}{P_{Wärtsilä}} \cdot \frac{\eta_{Wärtsilä}}{\eta_{Daihatsu}} = 2453 \cdot \frac{7124}{4000} \cdot \frac{0.458}{0.44} = 1433\ kW.$$

5.3 Low temperature circuit

For the LTC the same procedure is used as for the HTC. The LTC energy for the Wärtsilä engine is $\dot{Q}_{LTC} = 1668\ kW$, which gives:

$$\dot{Q}_{LTC,Daihatsu} = \ \dot{Q}_{LTC,Wärtsilä} \cdot \frac{P_{Daihatsu}}{P_{Wärtsilä}} \cdot \frac{\eta_{Wärtsilä}}{\eta_{Daihatsu}} = 1668 \cdot \frac{7124}{4000} \cdot \frac{0.458}{0.44} = 975\ kW.$$

5.4 Energy balance

To finish the energy balance the initial energy loss has to be calculated. Basically this means calculating how much energy is not used of the chemical energy stored in the fuel. For the Wärtsilä engines, we have already calculated the fuel consumption ($\dot{m} = 0.364 \frac{kg}{s}$). For the Daihatsu engine we use the same procedure:

$$\dot{m} = \frac{P_{electric}}{\eta_{electric} \cdot u} = \frac{4000kW}{0.44 \cdot 42.7} = 0.213 \frac{kg}{s}.$$

The chemical energy input for the Wärtsilä 16V32:

$$\dot{Q}_{chemical} = \dot{m} \cdot u = 0.364 \cdot 42.7 = 15.6\ MW.$$

And for the Daihatsu 16DK-28:

$$\dot{Q}_{chemical} = \dot{m} \cdot u = 0.213 \frac{kg}{s} \cdot 42.7 \frac{MJ}{kg} = 9.09\ MW.$$

Table 2 below shows the energy balance for the diesel engines.

Table 2 Energy balance diesel engines

		Wärtsilä 16V32	**Daihatsu 16DK-28**
Fuel	Consumption [kg/s]	0.364	0.213
	Chemical energy [MW]	15.6	9.09
Generator output	Frequency [Hz]	50	50
	Electrical power [kW]	7124	4000
	Electrical efficiency [-]	0.458	0.44
Exhaust gas	Energy [kW]	2816	1986
HT-circuit	Energy [kW]	2453	1433
LT-circuit	Energy [kW]	1668	975
Loss	Energy [kW]	1494	696

6 System calculations

Given the complete engine specification as specified in the previous chapter, further calculations can be made regarding mass flows and required energy transfers. The following section contains the calculations needed for further dimensioning of the system components. An excel datasheet is used to get a clear view of all formulas and dependencies within the system. An overview of the datasheet is given in 11.1.

For the calculations of the heat recovery system, the formulas will be used with the data for the highest state of operation, which is the winter daytime. After each used formula the data for the other states of operations are given in a table. For all heat exchanging steps (heat exchanger, economiser, evaporator and super heater) a loss of 25% is assumed. The system is based on the schematic given in Figure 4 and the following configuration.

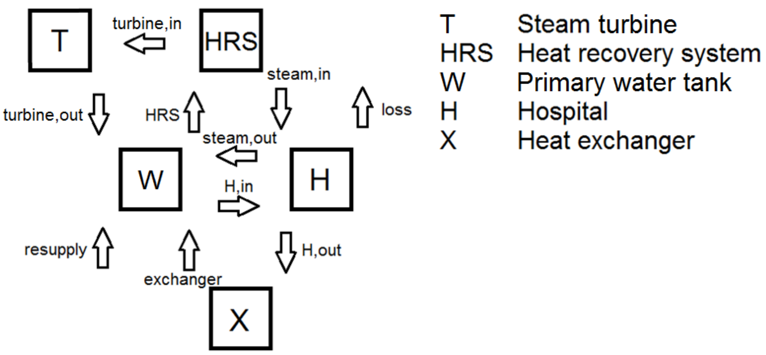


Figure 7 Schematic of system configuration and flows

For this heat recovery system, the critical factor is the amount of steam that can be generated. The exhaust gas from the diesel engines is used to heat water from 70°C to saturated (165°C) and superheated (240°C) steam for use in respectively the hospital and the steam generator. Using the differences in enthalpy between steam going in the hospital and the water coming out and the total amount of energy used, we find the following mass flow:

$$\dot{m}_{steam} = \frac{\dot{Q}_{steam}}{\Delta h} = \frac{5000}{2763.4 - 188.45} = 1.94 \frac{kg}{s}.$$

	Summer day	Summer night	Winter day	Winter night
$\dot{Q}_{steam}\ [kW]$	1500	1050	5000	3500
$\dot{m}_{steam}\ \left[\frac{kg}{s}\right]$	0.58	0.41	1.94	1.36

Now the amount of energy that has to be drawn from the exhaust gas to heat this part can be calculated:

$$\dot{Q} = \dot{m} \cdot \Delta h = 1.94 \cdot (2763.4 - 292.98) = 4797\ kW.$$

	Summer day	Summer night	Winter day	Winter night
$\dot{Q}[kW]$	1439	1007	4797	3358

The rest of the energy stored in the exhaust can be used for heating steam for the steam generator:

$$\dot{m}_{generator} = \frac{\dot{Q}_{exhaust} \cdot \eta_{exchanger} - \dot{Q}_{steam}}{\Delta h} = \frac{9605 \cdot 0.75 - 4797}{2850.4 - 292.98} = 0.94 \frac{kg}{s}.$$

	Summer day	Summer night	Winter day	Winter night
$\dot{Q}_{exhaust}\ [kW]$	4802	2816	9605	5632
$\dot{m}_{generator}\ \left[\frac{kg}{s}\right]$	0.85	0.43	0.94	0.34

6.1 Heat exchanger

With these mass flows the energy balance for the primary tank can be made. The primary tank will be kept at 70°C at all times. There are four mass flows entering the primary tank and two mass flows exiting the tank. To achieve this constant temperature, the water returning from the hospital will be heated before entering the tank. Using the energy balance, the required temperature for this mass flow can be calculated. Energy flow is specified as mass flow times specific enthalpy:

$$\dot{Q} = \dot{m} \cdot h$$

$$\dot{Q}_{exchanger} + \dot{Q}_{turbine,out} + \dot{Q}_{steam,out} + \dot{Q}_{resupply} = \dot{Q}_{H,in} + \dot{Q}_{HRS}$$

$$h_{exchanger} = \frac{(\dot{m} \cdot h)_{H,in} + (\dot{m} \cdot h)_{HRS} - \left((\dot{m} \cdot h)_{turbine,out+}(\dot{m} \cdot h)_{steam,out} + (\dot{m} \cdot h)_{resupply}\right)}{\dot{m}_{exchanger}}$$
$$= \frac{23.92 \cdot 292.98 + 2.88 \cdot 292.98 - 0.94 \cdot 251.13 + 1.90 \cdot 188.45 + 0.04 \cdot 62.99}{23.92}$$
$$= 303.32 \frac{kJ}{kg}.$$

Table 3 Massflows

	Summer day	Summer night	Winter day	Winter night
$\dot{m}_{exchanger} \left[\frac{kg}{s}\right]$	7.97	5.58	23.92	16.75
$h_{exchanger} \left[\frac{kJ}{kg}\right]$	305.24	304.04	303.32	302.52
$\dot{m}_{turbine,out} \left[\frac{kg}{s}\right]$	0.85	0.43	0.94	0.34
$h_{turbine,out} \left[\frac{kJ}{kg}\right]$	251.13	251.13	251.13	251.13
$\dot{m}_{steam,out} \left[\frac{kg}{s}\right]$	0.57	0.40	1.90	1.33
$h_{steam,out} \left[\frac{kJ}{kg}\right]$	188.45	188.45	188.45	188.45
$\dot{m}_{resupply} \left[\frac{kg}{s}\right]$	0.01	0.01	0.04	0.03
$h_{resupply} \left[\frac{kJ}{kg}\right]$	62.99	62.99	62.99	62.99
$\dot{m}_{H,in} \left[\frac{kg}{s}\right]$	7.97	5.58	23.92	16.75
$h_{H,in} \left[\frac{kJ}{kg}\right]$	292.98	292.98	292.98	292.98
$\dot{m}_{HRS} \left[\frac{kg}{s}\right]$	1.43	0.84	2.88	1.70
$h_{HRS} \left[\frac{kJ}{kg}\right]$	292.98	292.98	292.98	292.98

6.2 Economizer

The HRS design is based on the schematic shown in figure 3. The mass flow going through the economizer is three times that of the mass flow entering the HRS:

$$\dot{m}_{econ} = \dot{m}_{HRS} + \dot{m}_{HRS\,feedback} = 2.88 + 5.77 = 8.65\frac{kg}{s}.$$

The temperature at the economizer outlet is set to 140°C, the heat transfer in the economizer is:

$$\dot{Q}_{econ} = \dot{m}_{econ} \cdot h_{econ} - \left(\dot{m}_{HRS} \cdot h_{HRS} + \dot{m}_{HRS\,feedback} \cdot h_{HRS\,feedback}\right)$$
$$= 8.65 \cdot 589.13 - (2.88 \cdot 292.98 + 5.77 \cdot 614.97) = 704.77\ kW.$$

	Summer day	Summer night	Winter day	Winter night
$\dot{Q}_{econ}[kW]$	349	205	705	415
$\dot{m}_{evap}\left[\frac{kg}{s}\right]$	4.28	2.52	8.65	5.09

6.3 Evaporator

In the evaporator half of the mass flow going in is evaporated, the amount of energy required to do this is:

$$\dot{Q}_{evap} = \left(\dot{m}_{evap,g} \cdot h_{evap,g} + \dot{m}_{evap,l} \cdot h_{evap,l}\right) - \dot{m}_{evap} \cdot h_{evap}$$
$$= (2.88 \cdot 697.38 + 2.88 \cdot 2763.4) - 5.77 \cdot 614.97 = 6431.18\ kW.$$

	Summer day	Summer night	Winter day	Winter night
$\dot{Q}_{evap}[kW]$	3186	1873	6431	3788
$\dot{m}_{evap}\left[\frac{kg}{s}\right]$	2.86	1.68	5.77	3.40

6.4 Superheater

In the super heater the steam going to the steam turbine is heated to 240°C and 25bar:

$$\dot{Q}_{super} = \dot{m}_{super} \cdot \Delta h_{super} \quad = 0.94 \cdot (2850.4 - 2763.4) = 81.87\ kW.$$

	Summer day	Summer night	Winter day	Winter night
$\dot{Q}_{super}[kW]$	73.6	37.6	81.9	29.5
$\dot{m}_{super}\left[\frac{kg}{s}\right]$	0.85	0.43	0.94	0.34

6.5 Steam Turbine

The output of the steam turbine is water at 60°C. An electrical efficiency of 35% can be achieved with the steam turbine, which gives the following electrical output:

$$\dot{Q}_{turbine} = \dot{m}_{turbine} \cdot \Delta h_{turbine} \quad = 0.94 \cdot (2850.4 - 251.13) = 2446\ kW$$

$$P_{turbine} = \dot{Q}_{turbine} \cdot \eta_{electric} \quad = 2446 \cdot 0.35 = 856\ kW.$$

	Summer day	Summer night	Winter day	Winter night
$\dot{Q}_{turbine}[kW]$	2198	1123	2446	880
$P_{turbine}[kW]$	769	393	856	308

7 Heat Exchanger Design

Heat exchanger design choices:
The order in which components are discussed is done from the 'perspective' of the exhaust gas. It first passes through the superheater, then the evaporator and finally through the economizer. This order was chosen because it makes the design of the tubing and ducts that both medium need to follow simpler as well as the amount of energy needed by each section of the HRSG is now in decreasing order.

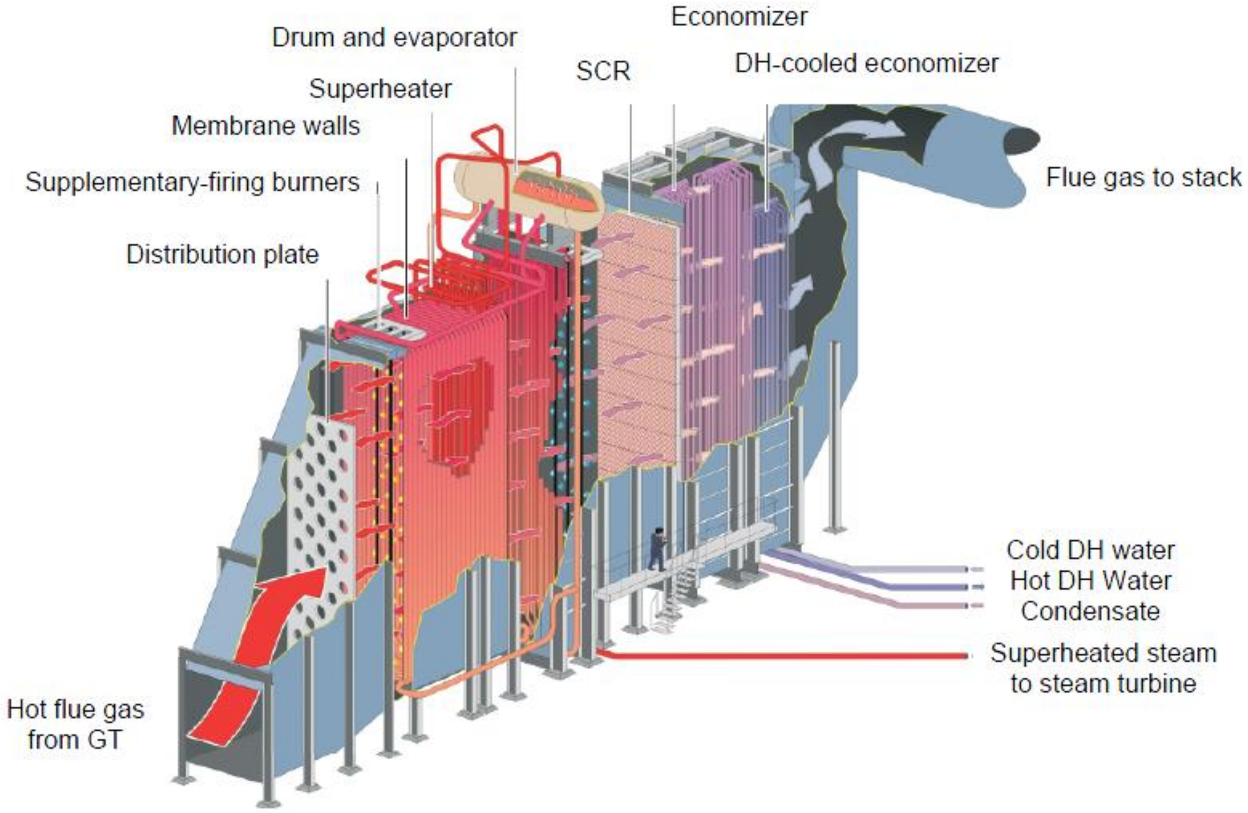


Figure 8 Image only to illustrate order of processes, courtesy of Siemens, Finsprong, Sweden

The superheater design decision was closely linked to that of the evaporator. Due to the evaporator's higher variability between designs, it was decided that the final design of the superheater by made to complement that of the evaporator. Therefore the superheater follows the design of an I-frame heat exchanger.

For the evaporator we could choose between a D-frame, an O-frame, an A-frame, an I-frame and a horizontal tube evaporator.

The d-frame was discarded because it's main advantage was its compact size. This was not a highly weighted design constraint over modularity (the heat exchanger needs to function at optimum capacity in all four reactor configurations) and overall efficiency. The A-frame design was also discarded quickly because it was mainly suitable for a heating medium that contained ash, which could be removed in the hopper between the two lower drums. Since our exhaust won't be causing fouling or particulates to a degree that this design makes sense, we discarded it. Also, the horizontal tube evaporator has similar advantages and limitations to the o-frame, but without the advantage of fitting into our vertically stacked HRSG, so it too had to go.

The o-frame came in second to our final choice. It was relatively efficient and modular. It had as a main advantage that it's upper header can be configured as the steam separation drum. It also allowed for the use of multiple risers, and the closing of any number of risers via valves, in order to keep the amount of heat transfer to our medium at the desired level. It lost out to the I-frame because the I-frame has the similar modularity required for our HRSG, but the tube bundles are connected in such a way that they are easier to shut off individually (should that be required), and also the cost per unit is lower than a similar o-frame.

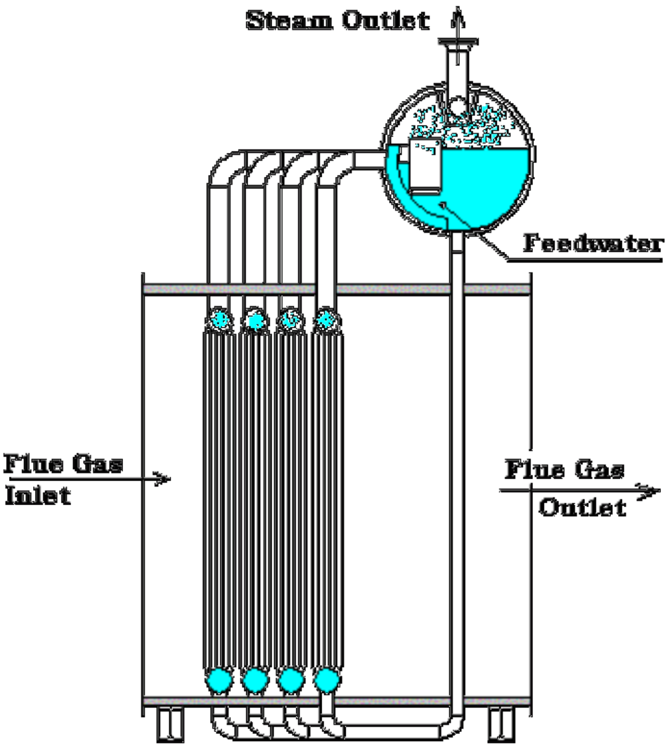


Figure 9 I-frame evaporator

The economiser follows a very similar set up to the other two sections of the HRSG, with one small exception. It is not an I-frame, but a vertical tube heat exchanger. This means it does not have lower drums, but simple curved tubes (see Figure 10 and Figure 11)

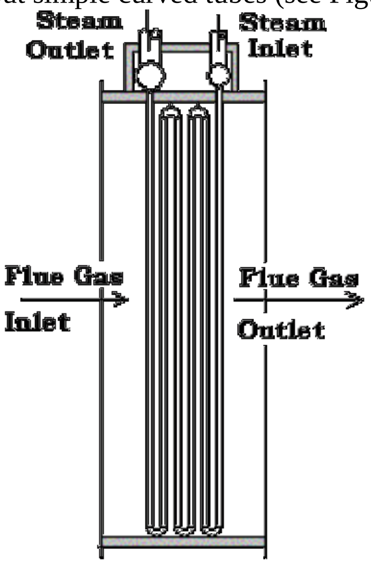


Figure 10 Vertical tube superheater

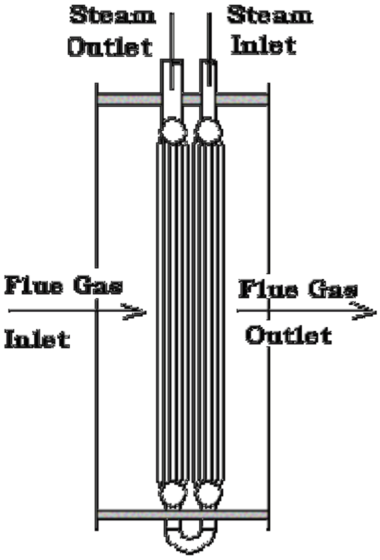


Figure 11 I-frame superheater

This was done because the pressure differential is now lower along the length of the tubing and no phase-change in the water is desirable at this point.

Order of tubes:

The decision of whether to have a counter-current or co-current set up of heat exchangers, and the order of which the above three heat exchangers is set up, though not trivial, was not difficult for this application. Putting the superheater first in contact with the exhaust gasses makes sense, because it requires the largets amount of heat energy. This allows for a relatively smaller contact surface needed to make the heat exchanger work according to requirements. Since the efficient and practical transfer of heat from the exhaust gasses trumps the need for the exhaust gasses to converge on the temperature of their counterpart medium a counter-current set up is chosen.

Single, double or triple pressure arrangements:
Another design decision that needed to be made is whether the HRSG is kept as a 'simple' single pressure pressure-device or not. Meaning the pressure throughout the HRSG is not increased at various points with pumps, but maintained at an even level. This allows for a natural circulation of both media. This does not mean the use of pumps to counteract pressure drops is not required, but merely that they increase the pressure back to the inlet pressure, no more. A visual aide to help with this decision can be seen from the bottom two graphs.

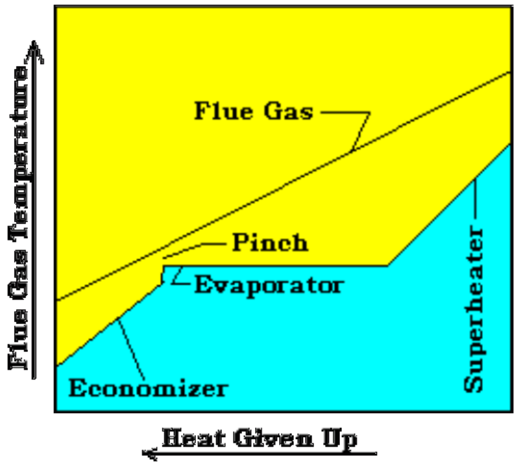


Figure 12 Superheater, evaporator & economizer

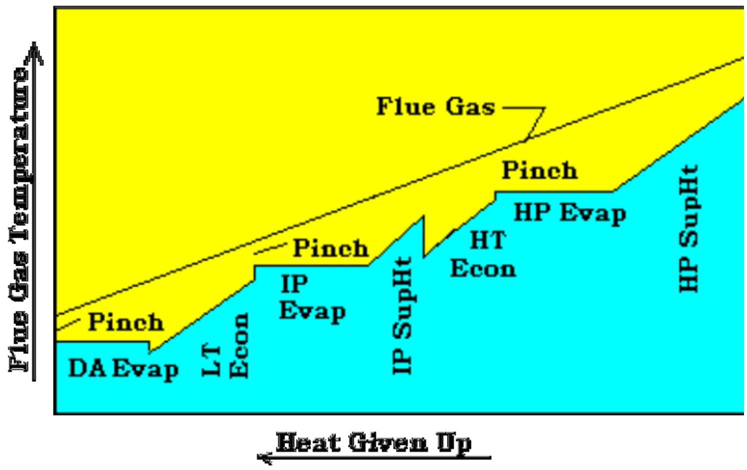


Figure 13 Triple pressure with SH IP & integral DA

The first image shows single pressure set up, where the second shows a complex, assisted circulation set up. The area under the straight line labeled 'flue gas' is also the potential heat energy that can be recovered from the system. The area shaded in cyan is the amount currently recovered, labeled for each separate component. It is again a balancing act, weighing the complexity of the system (and thus increasing potential downtown and parts that could fail, not to mention expenses) against the theoretical maximum efficiency attainable.

The calculations below show the relative efficiency of a single pressure set up and that of an industry standard triple pressure set up. Though in theory the addition of even more components could create a higher-efficiency system, but it is the opinion of this group that the triple-pressure set up is indeed the best choice, weighing complexity versus efficiency.

Calculations and dimensions:

Single vs triple pressure:
Assumptions: we make a considerable amount of assumptions to ensure that the calculations are manageable. This leads to simplifications and inaccuracies, but as long as these are kept in mind and kept to a minimum, they are acceptable. First off we assume a constant performance factor for our heat exchangers, 75%, we believe this is erring on the cautious side. Second we assume that the decreases in efficiency and heat transfer due to scaling the size of the heat exchanger is so small that it is negligible.

Due to the iterative nature of the calculations required, the formulae will be stated once and each subsequent use will just feature the final answer. The calculations were made using MATLAB scripts, featured in the appendix.

Equations used:
$Q_s = W_s(H_{s2} - H_{s1}) = Q_c + Q_n + Q_r$ is the energy balance of the superheater, Qs is the amount of energy absorbed by the steam in the superheater.

$Q = \eta * U * A * \Delta T_{\text{log mean}}$ gives the heat transfer [W] as a function of a given performance factor, the heat transfer coefficient, area of heat transfer and the log mean temperature. Useful for dimensioning various parts of the heat exchanger.

Using the above equations we calculated the required minimal surface area of the piping in the evaporator. Assuming the overall heat transfer coefficient between steel piping, water and air/exhaust to be approximately 10 W/m^2 k. We get a required surface area of 1.8206 m^2. In the winter day condition. In the summer night condition (other extreme) we require a factor of 3.5 less. You shall see in the course of these calculations that we have both kept that in mind and with the proper adjustments this has negligible effect on the overall efficiency and the component dimensions. The extra length of the piping does not endanger the equipment (keeping the exhaust gasses safely over the minimal temperature of 130 where H2SO4 condenses and starts to affect the materials.

Using the formula: $Q = \eta * U * A * \Delta T_{\text{log mean}}$ we can reformulate it to calculate the required area for the heat exchanger components for each condition. What we find is that the area required of the evaporator scales with approximately the same factor as the mass flows differ in the various states (ie during the day in the winter the heat exchanger needs 3-4 times as much surface area than at night in the summer). This scaling can be accomplished by closing a portion of the tubes in the heat exchanger components. This seems counter intuitive at first glance because it decreases the theoretical max efficiency of the first heat exchanger component, ie the superheater. But when considered with the auxhiliary design requirements in mind it makes sense. Keeping the heat exchangers working at the same efficiency means the temperature profile of the exhaust gasses won't change between operating states. Allowing for overall efficiencies to remain as they were designed. Increasing the efficiency of the superheater lowers the rest heat that can be drawn by the evaporator and economiser. Lowering their respective efficiencies. Not only this, but it keeps the exhaust gasses over the critical 130 degrees required to prevent sulphuric acid precipitation and formation which would damage the equipment at an unacceptable rate.

$\Delta T_{im} = \frac{(\Delta T_A - \Delta T_B)}{(\ln \frac{\Delta T_A}{\Delta T_B})}$ is the log mean temperature given between the temperatures of pipes A and B at a given point in the HRSG.

$$Q = \eta * U * A * \Delta T_{\text{log mean}}$$

Where Q is the (desired) heat transfer, η is the performance factor, U is the overall heat transfer coefficient, A is the surface area, and ΔT_{lm} is the log mean temperature of the two streams.

$$A = \frac{Q}{U * \Delta T_{\text{log mean}}}$$

NB eta is omitted because the value of Q used in the following formulae already has taken into account the inefficiency of the process.

Since Q is given for each state (ie we know how much energy we want to transfer), we only need to find the value for U in our heat exchanger. From literature we know the order of magnitued should be between 300-3000 [W/m^2 k].

$$U = \frac{1}{\frac{1}{h_1} + \frac{d}{k} + \frac{1}{h_2}}$$

Where h_1 is the heat transfer coefficient of steam, h_2 is the heat transfer coefficient of the exhaust gasses (taken to be approximately equal to air), d is the thickness of the piping in our exchanger and k is the heat transfer coefficient of these walls.

Filling in these values from literature:

$$\frac{1}{\frac{1}{150} + \frac{0,001}{15} + \frac{1}{6000}} = 407[W/m^2k]$$

Calculating the area required during each state gives us:

NB these values are taken from the excel sheet that holds our calculations and the above calculations

Winter night:

$$A = \frac{3788.07*10^3}{(38.711*407)} = 240.43[m^2]$$

Winter day:

$$A = \frac{6431.18*10^3}{(38,711*407)} = 408.19[m^2]$$

Summer night:

$$A = \frac{1873.4*10^3}{(38,6156*407)} = 119.20[m^2]$$

Summer day:

$$A = \frac{3186.12*10^3}{(38,6156*407)} = 202,72[m^2]$$

We have the luxury of choosing a pipe diameter for the heat exchanger that makes the speed at which the steam and water mixture passes through the heat exchanger acceptable for us. The relation between the speed and piping area is as follows: $V = \frac{\rho * m}{A}$

Judging 10 m/s as a good upper bound for speed, we test out it's effect on our components. Giving us a required piping area, using the data from the winter daytime, of 0.577 m^2. Factoring this maximum area into our calculations, we can set 100 pipes of radius 4.2cm or 10000 pipes of radius 4.2mm. So choosing a pipe of radius 4.2mm, which we bunch into groups of 100 pipes (for control purposes), we have the required area to maintain flow velocity. This flow velocity can be kept approximately constant by shutting of certain groups of pipes. This has the added benefit of lessening the surface area exposed to the exhaust gasses that absorb heat energy (therefore not throwing our heat exchanger's efficiency out of proportion).

Using this diameter, and the unit length, we can calculate what length of these pipes we need for our heat exchanger to work as we designed it.

$$0,0084*\Pi*1 = 0,0264[m^2]$$

We need a maximum amount of piping through our evaporator of 408.19/0.0264 = 15461.74m. Which we then divide by the 10000 pipes, giving us a length of 1.55m of piping.

Under different circumstances (such as winter night or summer) a smaller amount of surface area is also needed. The pumping mechanism and opened piping groups can be controlled to optimize flow velocity and usable heat exchanger area.

An evaporator wouldn't be more than a heat exchanger if it didn't allow for the expansion of its primary medium into steam. A reservoir is therefore present at the top of the pipes (since the water enters at the bottom and exits at the top), where it is allowed to expand into steam and the liquid is also captured. This reservoir connects directly to the feed supply for the evaporator (see diagram above).

Knowing the maximum possible mass flow for our primary medium (5.77 kg/s), and assuming our evaporator generates the required 50% steam and 50% liquid water by weight. We know the required expansion area needed for this steam. 2.88 kg/s of steam at 165 degrees gives a specific volume of 0.273 m^3/kg, which means our reservoir receives 0.786 m^3 of steam per second. Which is the maximum area required in the steam catching reservoir on top of the evaporator. Ideally, this reservoir is not a single vat, but a series of vats. The use of smaller vats facilitates maintenance, but also allows for varying the volume used for the expansion of steam. This means when the mass flow is lower, the other vats are closed off using valves so the expansion doesn't lower the pressure of the steam too much under the designed parameters.

The superheater is dimensioned using a similar rationale:

Using the familiar equation from above; $A = \frac{Q}{U * \Delta T_{\text{logmean}}}$ we will calculate the required surface area of our superheater. We will also use a modular approach, just like the evaporator. This means when the superheater is being used in a state that requires less heat transfer (see rationale above), it does not start using more of the heat energy in the exhaust gasses than the design parameters require of it. This gives us the following areas:

Winter night:

$$A = \frac{29.48*10^3}{(142.15*92)} = 2.25[m^2]$$

Winter day:

$$A = \frac{81.87*10^3}{(142.15*92)} = 6.26[m^2]$$

Summer night:

$$A = \frac{37.58*10^3}{(142.15*92)} = 2.87[m^2]$$

Summer day:

$$A = \frac{73.57*10^3}{(142.15*92)} = 5.63[m^2]$$

Once again assuming a maximum allowable fluid velocity of 10 m/s, we find winter day gives us an area of 18.8m^2 over our series of pipes. Choosing a series of 100 pipes we are given a radius of 2.4cm per pipe. By the same method used above, we find the area per unit area of piping to be 0.15 [m^2], leading to a maximum required length of piping of 41.73m, which amounts to approximately 42cm of piping along the entire set. This shows that this approach yields a relatively small section of heat exchanger.

24mm radius pipes, grouped as 10x10, allowing for equal areas of piping and exhaust gasses, yields a cross sectional area:
(Pipe radius + pipe thickness) * 2 *(10 pipes, plus 11 areas of exhaust flow)
(24+2)*2*21 = 1.09m on a side, so the heat exchanger as seen from the exhaust flow is 1.09m x 0.42m and 1.09m deep.

8 Exergy Analysis

Summarizing all the energy flows an exergy analysis can be made. In this section the total system efficiency and energy losses are calculated.

During winter days a total of 1.154 kg of diesel is used every second. With an energy density of 42.7 MJ/kg the chemical energy put into the system is 49291 kW.

The engines generate a total of 22248 kW of electricity. This is 45.1% of the input energy. The excess energy is transferred to heat. This heat can be divided in oil circuit and radiation (4380 kW), exhaust gas (9605 kW), and low (5286 kW) and high (7772 kW) temperature cooling circuit. The exhaust gas and high temperature cooling circuit can be used for heat recovery. The other energies cannot be restored and are considered a loss: $4380 + 5286 = 9666\ kW \equiv 19.6\%$.

From the high temperature cooling water 3247 kW (6.6% of total energy) will be effectively transferred to the hot water system. The rest has is lost to radiation or has to be drawn using sea water. This is a loss of $4528\ kW \equiv 9.2\%$.

Over the heat recovery system, which is 'fueled' with the exhaust gas, a loss of 25% is assumed, this gives a loss of 2401 kW (4.9% of total energy). 5000 kW is transferred to steam (10.2%) and 3000 kW to hot water (6.1%). The rest is used for the steam turbine. The steam turbine generates 856 kW of electricity (1.7% of total energy), this means another loss of 1590 kW (3.2%).

These values are combined in the following chart:

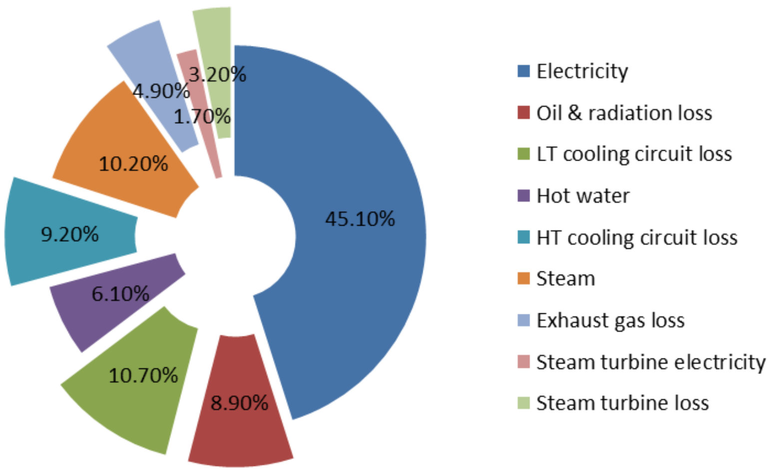


Figure 14 Exergy chart

Table 4 Exergy analysis

	Summer day	Summer night	Winter day	Winter night
Energy input (Fuel) [kW]	24645	15555	49291	31109
Electricity drawn from engines [kW]	11124	7124	22248	14248
Electricity drawn from turbine [kW]	769	393	856	308
Total electricity [kW]	11893	7517	23104	14556
Hot water usage [kW]	1000	700	3000	2100
Steam usage [kW]	1500	1050	5000	3500
Excess heat LT cooling circuit [kW]	2643	1668	5286	3336
Excess heat HT cooling circuit [kW]	2422	1437	3442	1893
Loss heat exchanger	366	254	1082	753
Loss heat recovery system	1203	705	2406	1411
Loss steam turbine	1429	729	1590	572

Efficiency [%]	58.4	59.6	63.1	64.8
Electricity overcapacity [kW]	893	-183	604	-1194

The excess heat of the cooling water will be contracted using a cooling installation with seawater.
As follow from Table 4, the amount of electricity generated is either more or less than the demand. Since the different operating states are equally divided over the years, the 'Electricity shortages' can be summed to calculate the total under- or overcapacity of electricity:

$$P_{avg} = \frac{893kW - 183kW + 604kW - 1194kW}{4} = 30kW.$$

The hospital's power plant has an average overcapacity of 30kW, which is only 0.2% of the total capacity. Yearly this comes to a 262.8MWh overproduction. An additional advantage is that the overcapacity is high during the day time, when the amount of energy used in urban areas are high and during night time, when urban energy usage is low, the system can easily draw electricity from the public net. An arrangement can be made with the local electricity supplier.

9 Maintenance

9.1 Diesel-engines

This section is based on a CAT marine diesel-engine. It is assumed that our engines are of a similar type.

It is also assumed that the winter period comprises one third of the year. This correspondents to 120 days, 2880 hours.

All engines can run for 24 hours straight without maintenance. During a winter night, when one small and one large engine is in operation, this enables maintenance on the other two engines.

Every 250 hours, small parts like belts should be checked or replaced. It is assumed that this can be done in one part of the day, the winter night. This way, the engines are back at full capacity during the day.

Every 3000 hours, some bigger maintenance projects such as the replacement of cooling fluid should be done. This takes three days. This should be done just before the winter starts so that all four engines are always available during the winter.

Every 10000 hours, the engines need a major revision. This should of course be done during the summer. This takes 250 hours per engine during which electricity is to be taken from the public grid (4).

9.2 Heat-exchanger

When a heat-exchanger is in use for a period of time, solid particles may contaminate the walls. This has two disadvantages for the efficiency of the heat-exchanger. Firstly: the boundary between the hot and the cold medium becomes larger. Secondly: the material of which the particles are made of most likely conducts heat worse than the material of the heat-exchanger (made of a metal). See Figure 15.

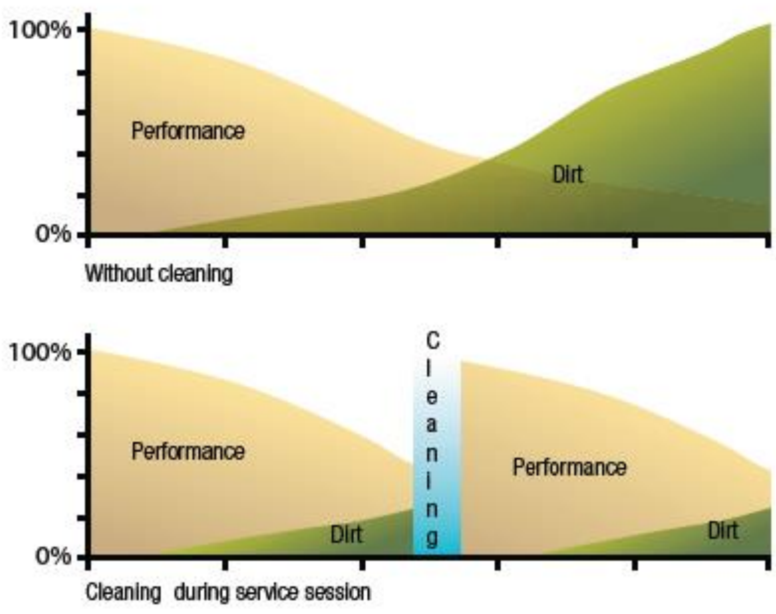


Figure 15 Performance of a heat-exchanger related to fouling

This phenomenon can be detected by a bigger than expected pressure drop over the heat-exchanger or a loss of efficiency. The heat-exchanger should then be cleaned. The effort required to clean a heat-exchanger strongly increases with a thicker contamination layer so the cleaning should be done in time. The worst case scenario is a damaged heat-exchanger due to overheating.

Hot exhaust fumes from the diesel-engines flow into the hot part of heat-exchanger H2. These fumes contain ashes from the engines en thus will contaminate the heat-exchanger wall. Cooling fluid and water flow through heat-exchanger H1. This water contains calcium and can lead to a similar phenomenon as what happens in a washing-machine. The deposition of lime scale (5) (6).

9.3 Steam turbine

To improve the reliability of a steam turbine, it is important to prevent liquid water from entering. This will lead to corrosion and a lower efficiency. (7)

10 Conclusion

To provide electricity, hot water and steam to a hospital a Combined Heat and Power plant is designed. For the main power source 4 diesel engines are used which can be turned on and off to comply with different states of demand. The diesel engines create mechanical energy which is directly converted to electricity and thermal energy which is re-used using a Heat Recovery System. Excess heat from the cooling system is used to heat water and the exhaust gasses are captured to create steam for both the hospital and an additional steam turbine. With an overcapacity at daytime and under capacity at night the system is self-sustaining on average. The average efficiency is just over 61%, which makes the system environmental friendly compared to using the national power supply.

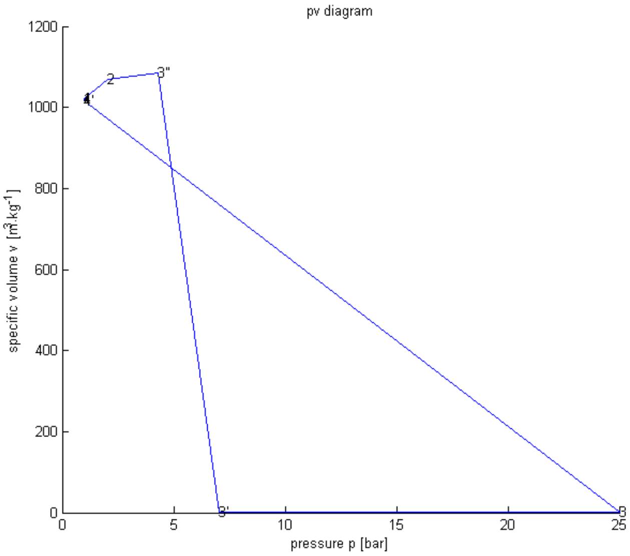
11 Appendices

11.1 Excel datasheet

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W
1	3	Fixed	State	Calc.	Calc. dep.																		
2			winter day		Fuel	Electricity	Exhaust			HT cooling water			LT cooling water				Mass flows [kg/s]				T	enthalpy	
3					m [kg/s]	P [kW]	Q [kW]	m [kg/s]	T_out [°C]	Q [kW]	T_in [°C]	T_out [°C]	Q [kW]	T_in [°C]	T_out [°C]		HT cooling water		115,54		WATER		
4		2	Wärtsilä	16V32	0,729	14248	5632,48	28,0	348	4906	80	96	3336	38	49		LT cooling water		115,05		15	62,99	
5		2	Daihatsu	16DK-28	0,426	8000	3972,482	19,5	350	2866	80	96	1950	38	49		Hot water demand		23,92		38	159,21	
6			Total		1,154	22248	9604,962	47,54	348,82	7772	80	96	5286	38	49		Steam demand		1,94		40	167,57	
7																	Steam demand returr		1,90		45	188,45	
8		Demands	[kW]	Power	22500	Steam	5000	Hot water	3000		Heat exchanger efficiency			0,75			Resupply		0,04		49	205,15	
9						T_in [°C]	165	T_in [°C]	70		Steam turbine efficiency			0,35			Steam turbine		0,94		60	251,13	
10						T_out 98% [°C]	45	T_out [°C]	40								HRS		2,88		70	292,98	
11						T_out 2% [°C]	15														72,47	303,32	
12																					80	334,91	
13		Fase 1	HXCH		Fase 2	TNK1		Fase 3	ECON		Fase 4	TNK2		Fase 5	EVAP						87,09	364,70	
14		Hot water demand return			Hot water demand return			HRS			ECON (out)			EVAP			Seawater T [°C]		10,00		96	402,18	
15		m	23,92		m	23,92		m	2,88		m	8,65		m	5,77		Flow rate [kg/s]				120,67	507,64	
16		T_in	40		T	72,47		T	70		T	140		T_in	146						140	589,13	
17		h_in	167,57		h	303,32		h	292,98		h	589,13		h_in	614,97		Components				146	614,97	
18		T_out	72,47		Steam demand return			HRS feedback			EVAP (out)			m_l_out	2,88		Heat exchanger		HXCH		165	697,38	
19		h_out	303,32		m	1,90		m	5,77		m_l	2,88		T_l_out	165		Primary tank		TNK1		165	2763,40	
20		Q	3247,23		T	45		T	146		T_l	165		h_l_out	697,38		Economizer		ECON		240	2850,40	
21		HT cooling water			h	188,45		h	614,97		h_l	697,38		m_g_out	2,88		Secundairy tank		TNK2		EXHAUST GAS		
22		m	115,54		Steam turbine return			ECON			m_g	2,88		T_g_out	165		Evaporator		EVAP		160	171,20	
23		T_in	96		m	0,94		m	8,65		T_g	165		h_g_out	2763,40		Superheater		SHTR		178,10	190,57	
24		h_in	402,18		T	60		T_in	120,67		h_g	2763,40		Q	6431,18		Steam turbine		TRBN		346,68	370,94	
25		T_out	87,09		h	251,13		h_in	507,64		TNK2			Exhaust							348	372,36	
26		h_out	364,70		Resupply			T_out	140		m_l	14,41		m	47,54		Diesel Oil				348,82	373,24	
27		Q	4329,640904		m	0,04		h_out	589,13		T_l	146		T_in	346,68		Energy density [kJ/kg]				350	374,50	
28		HXCH			T	15		Q	704,77		h_l	614,97		h_in	370,94			42700					
29		Q_loss	1082,41		h	62,99		Exhaust			m_g	2,88		T_out	178,10		Density [kg/m3]						
30					TNK1			m	47,54		T_g	165		h_out	190,57			836,6					
31					m	26,80		T_in	178,10		h_g	2763,40		Q	8574,90								
32					T	70		h_in	190,57					EVAP			Dimensions:						
33					h	292,98		T_out	160					Q_loss	2143,73		Heat exchanger						
34								h_out	171,20									Tube (kg/s)	23,92				
35		Fase 6	SHTR		Fase 7	TRBN		Q	939,69									Shell (kg/s)	115,54				
36		Steam turbine			Steam turbine+condensor			ECON			Excess heat HT cooling water [kW]				3442,359			Q (kW)	4329,641				
37		m	0,94		m	0,94		Q_loss	234,92		Excess heat LT cooling water [kW]				5286		Economizer		exhaust	(kg/s)	47,54		
38		T_in	165		T_in	240					Electricity drawn from engines [kW]				22248			m (kg/s)	8,65				
39		h_in	2763,40		h_in	2850,40					Electricity drawn from turbine [kW]				856,13			Q (kW)	939,69	Q_eff	704,77	Q_loss	234,92
40		T_out	240		T_out	60					Total electricity generated [kW]				23104,13		Evaporator		exhaust	(kg/s)	T_in [°C]		
41		h_out	2850,40		h_out	251,13					Electricity shortage [kW]				-604,13			m (kg/s)	5,77				
42		Q	81,87		Q	2446,07985					Efficiency [%]				63,1			Q (kW)	8574,90	Q_eff	6431,18	Q_loss	2143,73
43		Exhaust			Q_eff	856,13											Superheater		exhaust	(kg/s)	0		
44		m	47,54		Q_loss	1589,95												m (kg/s)	0,94				
45		T_in	348,82															Q (kW)	109,1634	Q_eff	81,87	Q_loss	27,29
46		h_in	373,24														Condensor						
47		T_out	346,68															m (kg/s)	0,94				
48		h_out	370,94																				
49		Q	109,1634431																				
50		SHTR																					
51		Q_loss	27,29																				
52																							

Figure 16 Excel datasheet for winter day scenario

11.2 System diagrams



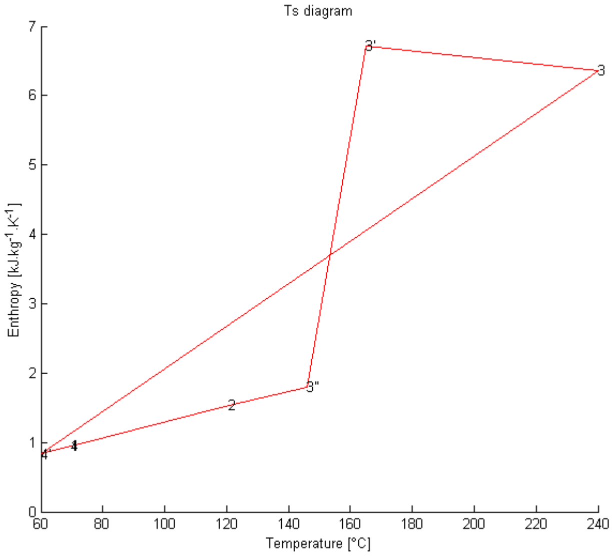


Figure 17a P-V diagram for steam cycle *b T-S diagram for steam cycle*

11.3 Mfiles

```
function watt = hxt(U,A,deltaT)

%this function calculates the heat transfer for a heat exchanger (in watts)
%based on the heat transfer coeff of the material, area of heat transfer,
%assumed efficiency (75%) and the log mean temperature difference.

watt = 0.75 * U * A * deltaT;

%mu = 75%
%U = [W/k*m^2]
%A = [m^2]
%deltaT = [k]

end


function area = hxtarea(q,U,deltaT)
%this function uses the same equation at hxt, but uses the other knows to
%find the area needed to achieve the required effect.

area = q / (U * deltaT);

end


function deltaTlm = logmean(a,b)
%Calculate Log mean temp
%where a is the temp diff at point 1 in the exchanger, b is the temp diff
%at the other end of the exchanger

deltaTlm = (a - b)/(log(a/b));

end
```

11.4 Dimensions of components

Table 5 Dimensions of components

wat	van	naar	mass winter	mass zomer	specific vol	snelheid winter	snelheid zomer	diameter	lengte	d_hoogte	rechte bochten	overige bochten
pi=	3,14159265		kg/s	kg/s	m^3/kg	m/s	m/s	m	m	m	m	m
exhaust												
1	engines	superheater	32,9	11,1	0,050	23,272	7,85	0,3	11,7	+0,5	2x (r=0,5)	
2	superheater	evaporator	32,9	11,1	0,050	23,272	7,85	0,3	7,1	-0,2	2x (r=0,5)	
3	evaporator	economizer	32,9	11,1	0,050	23,272	7,85	0,3	6,0	+0,9	1x (r=0,5)	
4	economizer	exhaust tower	32,9	11,1	0,050	23,272	7,85	0,3	5,8		2x (r=0,5)	
cooling water												
1	heat exchanger	pump	23	7,7	0,001	2,928	0,98	0,1	2,8	-0,6	3x (r=0,25)	
2	pump	seawater tank	23	7,7	0,001	2,928	0,98	0,1	2,1	-0,7	1x (r=0,25)	
3	seawater tank	engines	23	7,7	0,001	2,928	0,98	0,1	17,4	+0,7	3x (r=0,25)	
4	engines	heat exchanger	23	7,7	0,001	2,928	0,98	0,1	22,1	+0,4	7x (r=0,25)	
water/stoom												
deel 1												
1	tank_1	pomp_1	2,78	0,93	0,001	1,416	0,47	0,05	1,9	-1	1x (r=0,1)	
2	pomp_1	splitsing_1	2,78	0,93	0,001	1,416	0,47	0,05	2,1		1x (r=0,1)	
3	splitsing_1	economizer	8,34	2,79	0,001	4,248	1,42	0,05	5,6	+3,9	3x (r=0,1)	
4	tank_2	pomp2	11,12	3,72	0,001	5,663	1,89	0,05	1,4	-0,9	1x (r=0,1)	
5	pomp_2	splitsing_2	11,12	3,72	0,001	5,663	1,89	0,05	0,4			
6	splitsing_2	splitsing_1	5,56	1,86	0,001	2,832	0,95	0,05	5,7		2x (r=0,1)	
7	splitsing_2	vaporizer	5,56	1,86	0,001	2,832	0,95	0,05	6,8	+2,6	5x (r=0,1)	
deel 2												
1	economizer	tank_2	8,34	2,79	0,001	4,248	1,42	0,05	14,7	-2,3	6x (r=0,1)	
deel 3												
1	vaporizer	tank_2	5,56	1,86	0,138	10,857	3,63	0,3	9,1	-1	3x (r=0,5)	
deel 4												
1	tank_2	splitsing_3	2,78	0,93	0,275	10,813	3,62	0,3	2,5	+0,7	1x (r=0,5)	
2	splitsing_3	superheater	0,8	0,51	0,275	7,002	4,46	0,2	6,3	-0,4	2x (r=0,5)	2x 45 graden (r=1)
3	splitsing_3	splitsing_4	1,98	0,42	0,275	7,702	1,63	0,3	5,9		1x (r=0,5)	
4	splitsing_4	hospital	1,98	0,42	0,275	7,702	1,63	0,3	2			
5	backup_2	splitsing_4	1,98	0,42	0,275	7,702	1,63	0,3	9,8	-0,2	3x (r=0,5)	1x 30 graden (r=1)
deel 5												
1	tank_1	pomp_3	26,6	6,2	0,001	3,387	0,79	0,1	4,2	-3,5	1x (r=0,25)	
2	pomp_3	splitsing_5	26,6	6,2	0,001	3,387	0,79	0,1	1,1			
3	splitsing_5	splitsing_6	26,6	6,2	0,001	3,387	0,79	0,1	2,2		1 (r=0,25)	
4	splitsing_6	hospital	26,6	6,2	0,001	3,387	0,79	0,1	2,6			
5	splitsing_5	backup_1	26,6	6,2	0,001	3,387	0,79	0,1	6,2	+1,4	4 (r=0,25)	
6	backup_1	splitsing_6	26,6	6,2	0,001	3,387	0,79	0,1	12,8	-1,4	5 (r=0,25)	
deel 6												
1	superheater	steam turbine	0,8	0,51	0,088	4,000	2,55	0,15	1,8	-0,8	2x (r=0,25)	
2	steam turbine	pomp_4	0,8	0,51		0,000	0,00	0,15	1,6	+0,4	2x (r=0,25)	
3	pomp_4	tank_1	0,8	0,51	0,001	0,407	0,26	0,05	13,3	+2,3	2x (r=0,1)	2x 45 graden (r=0,25)
deel 7												
1	tank_1	pomp_5	1,98	0,42	0,001	1,008	0,21	0,05	1,5	-1	1x (r=0,1)	
2	pomp_5	backup_2	1,98	0,42	0,001	1,008	0,21	0,05	11	+2,6	4x (r=0,1)	
deel 8												
1	hospital	tank_1	1,98	0,42	0,001	1,008	0,21	0,05	5		1x (r=0,1)	
deel 9												
1	hospital	pomp_6	26,6	6,2	0,001	3,387	0,79	0,1	9,9		2x (r=0,25)	
2	pomp_6	tank_1	26,6	6,2	0,001	3,387	0,79	0,1	10,3	+4,9	2x (r=0,25)	2x 30 graden (r=0,5)

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