Analysis of efficient hydrogen fuelled steam cycle for power production

Master's thesis in Sustainable Energy Systems

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CHALMERS UNIVERSITY OF TECHNOLOGY
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Abstract

The share of renewable energy sources such as wind and solar are increasing rapidly in our society due to harder carbon capture policies and an increased request for green energy. Due to the fluctuating nature of these energy sources, reliable storage is needed and additionally highly efficient power cycles able to convert energy from storage back to useful energy. Hydrogen is a promising energy carrier than can be produced when energy is high, stored in gaseous or liquid phase and converted back to energy in a hydrogen fuelled power cycle when energy is low. The power cycle needs to be highly efficient to be competitive with fossil fuels and reach a high round trip-efficiency from energy to hydrogen to energy.

A hydrogen fuelled power cycle similar to a Rankine cycle (steam cycle) with direct combustion of hydrogen and oxygen has been modelled and evaluated in IPSEpro in this project. Combustion of hydrogen and oxygen results in high temperature steam functioning as the working media. The aim of the project is to investigate the thermodynamics and critical parts of the steam cycle and quantify its efficiency.

The modelled hydrogen cycle resulted in an efficiency based on lower heating value, LHV, of 67.7% which is lower than for other similar hydrogen power cycles studied in previous literature. These are the Graz cycle, Toshiba cycle, Westinghouse cycle and MNRC cycle. In this model, hydrogen and oxygen were assumed to be supplied ”freely” at required pressure to the combustion chamber. A sensitivity analysis of three critical parameters showed that the efficiency is increased with increased pressure and temperature of the cycle.

To expand the system boundary of the power cycle, and not assume ”free” hydrogen and oxygen to the combustion chamber, the substances are instead assumed to be supplied from its production site at 40 bar. A model of liquefaction of hydrogen and oxygen together with pumping up to combustion chamber pressure was then added to the cycle. This enables liquid storage of hydrogen. The resulting model including hydrogen power cycle and liquefaction and pumping of hydrogen and oxygen resulted in an efficiency of 31.1%. The decrease in efficiency depends mainly on the high energy consumption to liquefy hydrogen as a result of its low boiling point of -252°C.

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Sofia Wiell, Gothenburg, June 21

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Introduction

1.1 Background

With growing energy demand and harder carbon dioxide capture policies, renewable energy sources such as wind and solar power needs to play a bigger role in the futures energy demand [1]. As renewable, intermittent, energy source increases, the need to be able to store energy increases as well. Renewable energy sources has a fluctuating nature which results in times where there is too much energy on the energy net and times when there is lack of energy. To be able to provide energy on demand, 24 hours a day, a high efficient, reliable energy storage is needed [2]. Hydrogen is a promising energy carrier that can be stored and used as fuel for reconversion of hydrogen to electricity. Hydrogen can be produced when there is an energy abundance and be used as fuel when converting back to energy [2]. Once hydrogen is produced, an ecological friendly fuel, free from pollutants is produced since combustion of hydrogen in pure oxygen results in high temperature steam [3], see Equation 1.1. The concept sets high demand on the hydrogen fuelled power cycle (gas to power) to reach a high overall efficiency from power to gas to power [2].

$$H_2 + \frac{1}{2}O_2 \rightarrow H_2O \tag{1.1}$$

Hydrogen has many advantages as a fuel, it is free from pollutants, has a high energy content and is non-toxic. The major drawback with hydrogen is its low density and low boiling point. The low density makes hydrogen storage in gaseous phase very space consuming and the low boiling point makes liquefaction of hydrogen very energy consuming [4].

Hydrogen is produced mainly by three processes, reforming of natural gas, gasification of biomass and electrolysis of water. Electrolysis of water is the most sustainable and environment friendly. It is done by splitting water into hydrogen and oxygen [5]. Schmidt et al. [6] have investigated different electrolysis methods where polymer electrolyte membrane (PEM) is concluded to be the most promising process in the future. The process is most effectively run when hydrogen and oxygen are delivered pressurized around 40 bar [6].

1.1.1 Hydrogen power cycles

A number of hydrogen power cycle concepts have been investigated during this theses. The concepts are done by Sanz et al, Weiliang, Milewski and Platzer et al. What all cycles have in common is direct combustion of hydrogen in pure oxygen to generate high temperature steam as the working media in the cycles. Sanz et al. studied a combined cycle [2], Weilang a concept combining hydrogen production with a hydrogen turbine and a steam turbine [7]. Milewski summarised previous studies of three power cycles more similar to a Rankine cycle [8]. Platzer et.al also modelled a cycle very similar to the combined cycle by Sanz et al. but included a fuel cell to increase the efficiency [9]. The cycles have shown efficiencies between 55% and 77%.

1.1.2 Liquefaction of gases

Hydrogen can either be stored as compressed gas or in liquid phase [3]. If liquid storage is desirable, liquefaction of hydrogen needs to be done. This is an energy intensive process, partly due to its low boiling point of -252°C. It is most often made industrially with the so called Claude process. Liquefaction of oxygen can also be made with the Claude process but is far from as energy intense as hydrogen liquefaction as its boiling point is only -183°C [10].

1.2 Objectives

The aim of the project is to investigate the thermodynamics and critical parts of a hydrogen fuelled power cycle and the potential of energy production. The goal is to investigate the possible efficiency of a hydrogen steam cycle similar to a Rankine cycle with pre-heat and how it can be optimised.

The specification of issue includes the following statements:

- Quantify the efficiency potential of a hydrogen fuelled steam cycle.
- First assessment to identify the critical aspects and limiting factors of the hydrogen cycle.

The long-term goal of the project is to provide information and an idea on how to decrease the use of fossil fuels in power generation and replace it by hydrogen.

1.3 Limitations

Limitations in the project includes:

- The project is a pure thermodynamic study based on heat and mass balance analysis.
- The study will not included any component design such as turbo machinery design or flow path analysis.
- The project will not include any economic aspect.

2

Theory

The following section covers the basic theory of heat engines and some specific cycles together with the combustion in a power cycle. Followed by a section about hydrogen, its production process and liquefaction of gases.

2.1 Heat engines

A heat engine builds on the idea of convert heat to work. Heat is received from a high temperature source and part of the heat is converted into work, most often by a rotating shaft. The remaining heat is rejected to a heat sink, most often a condenser. There are always losses acquiring in this process. A heat engine operates on a thermodynamic cycle where a system is taken through a series of different states by different components. There are both closed and open cycles. In the open one, new working media is continuously added to the cycle and passed to the surroundings afterwards. In a closed cycle, the working media is enclosed and going round in the cycle multiple times. The simplest thermodynamic cycle consist of a combustion chamber, turbine, compressor and/or a pump [11].

The net work output, W_{net} in Watt, from a heat engine is defined as the difference between work output, W_{out}, and work input, W_{in}. It can also be described as heat input, Q_{in} in Watt, minus heat output, Q_{out}, according to Equation 2.1. The thermal efficiency, η_{th}, for such a cycle is the net work output divided by the total heat input, Equation 2.2 [11].

$$W_{net} = W_{out} - W_{in} = Q_{in} - Q_{out} \tag{2.1}$$

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} \tag{2.2}$$

In a thermodynamic cycle, the energy balance of each component can be achieved from the first law of thermodynamics; energy can neither be created nor destroyed, it can only change form. The energy balance is therefore based on the idea that the total energy entering the system minus the total energy leaving the system is equal to the net change in energy of the system during a process. This is applicable to any kind of system undergoing any kind of process [11]. A T-s-diagram (temperature-entropy) is often used to analyse energy transfer in thermodynamic cycles. Work done by and on the system can easily be visualised in the diagram. By definition, from the first law of thermodynamics, the area under the graph in a T-s-diagram represents the net energy output of the system [12]. Thus, if the area under the graph increases when steam parameters of the cycle are changed, but the energy input is still the same, the area increase also results in an increase in efficiency of the cycle.

2.1.1 Carnot cycle

The Carnot cycle is a theoretical, closed heat engine that operates at four reversible processes and is therefore an idealisation of a real heat engine. The four processes are isothermal expansion, adiabatic expansion, isothermal compression and adiabatic compression. For this reversible heat engine, the thermal efficiency is shown in Equation 2.3 [11]. Since the Carnot heat engine is a reversible, ideal cycle, this efficiency is always higher than for an irreversible, real cycle, even though the cycles operates between the same hot and cold temperatures. The Carnot cycle is thus the upper limit on the efficiency that any thermodynamic engine with temperatures T_{hot} and T_{cold} can achieve during the conversion of heat into work [11].

$$\eta_{th} = 1 - \frac{T_{cold}}{T_{hot}} \tag{2.3}$$

From the equation of thermal efficiency for the Carnot cycle it is clear that the temperatures at heat source, T_{hot}, and heat sink, T_{cold}, in a cycle is highly important for the efficiency. To optimise the efficiency, T_{hot} should be as high as possible and T_{cold} as low as possible, i.e. the temperature gap between these two temperatures should be as big as possible [11].

2.1.2 Rankine cycle

The most common heat engines are gas turbines and steam cycles. A gas turbine is an open cycle which generally consist of a compressor, a combustion chamber with direct combustion and cooled turbines. Cooling is needed because to high temperatures are created during combustion for the material to handle. The working medium in a gas turbine is normally air [11].

A steam cycle, also called Rankine cycle, is a closed cycle that generally consist of a combustion chamber with indirect combustion, a turbine, a condenser and a pump, see Figure 2.1 for cycle configuration and its corresponding T-s-diagram. Water is the working fluid in the cycle and the engine depends on the phase change of water. Steam is a key resource for heat engines because of its wide availability, advantageous properties and nontoxic nature [13].

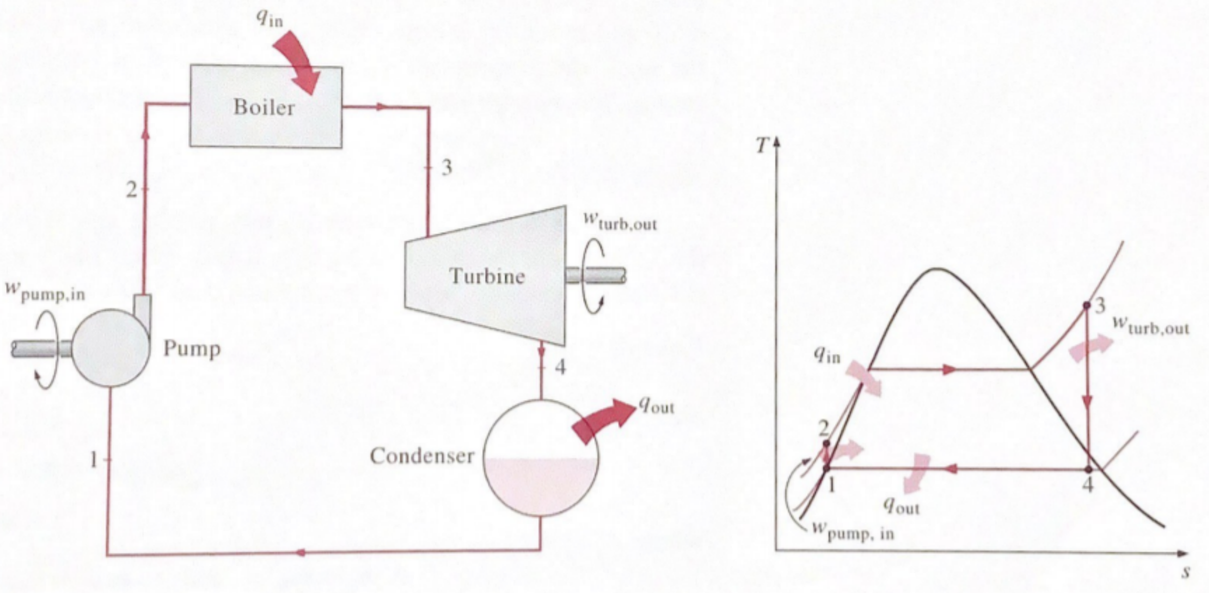


Figure 2.1: Cycle configuration of a simple Rankine cycle and its corresponding T-s-diagram [11]

2.1.3 Heating value and combustion energy balance

Standard enthalpy of formation, $h_f°$, of a substance is by definition the enthalpy change (positive or negative) of the reaction creating one mole of the substance (and solely one product in the reaction) from its elements in their most stable form. All reactants in the reaction should be stochiometric to the product. Enthalpy of formation is expressed in J/mole or J/kg. When presenting the enthalpy of formation of a substance, it is included what state the substance is at (solid, liquid, gas). Tabulated values of standard enthalpy of formation are determined experimentally and are given at a so called reference state, normally at 1 bar, 25°C and liquid phase. The reference state is marked with a °(e.g. $h_f°$for standard enthalpy of formation at reference state). From this, it follows that the standard enthalpy of formation for a substance in its most stable form, is zero, as there is no change involved in their formation cause they are at their purest form [14].

From the standard enthalpy of formation, one can calculate the standard reaction enthalpy for a specific reaction. This can either be positive or negative which is a result of whether the reaction absorbs or releases energy (as heat). All combustion reactions releases energy. The reaction enthalpy is highly depended on the state of the product since it releases or absorbs different amount of energy when the phase of a substance changes [14].

In a combustion reaction, enthalpy of reaction is often called heating value. It is the energy released as heat during fully combustion of one unit of the fuel. It is presented in J/kg or J/mole and is based on standard enthalpy of formation for the combusted substances and has two definitions: [15]

- High heating value (HHV). Energy released as heat when a unit of the fuel is fully combusted. Both incoming and outgoing streams should be at reference state, and in particular condensing any water vapour produced during combustion to liquid phase. The combustion should be done under constant pressure. HHV is determined from the difference in enthalpy of formation between products and reactants at their reference state, Equation 2.4. It can also be determined experimentally. See Figure 2.2 for graphical explanation of HHV where octane is combusted in air, reference state 25°C, 1 atm and water product is in liquid phase [15].

  $$HHV = \sum(n_p h_{f,p}{}^{\circ}) - \sum(n_r h_{f,r}{}^{\circ}) \qquad (2.4)$$

  where n_p is the number of moles of product per unit of fuel and n_r is the number of moles of reactants per unit of fuel. $h_{f,p}{}^{\circ}$and $h_{f,r}{}^{\circ}$is the enthalpy of formation at reference state for products respectively reactants [15].

- Low heating value (LHV) is defined in the same way as the HHV but with all water from the combustion process as product in gas phase. LVH is therefore the HHV subtracted by the enthalpy of vaporization, ΔH_{vap} for water (2,4 MJ/kg at 25°C) since the energy required to vaporize the water therefore is not released as heat, see Equation 2.5 [15]. The concept is described graphically in Figure 2.3.

  $$LHV = HHV - \Delta H_{vap} \qquad (2.5)$$

  Important to notice is that HHV and LHV depends on the chosen reference state due to its definition based on $h_f{}^{\circ}$and will thus be a different value if another reference state is chosen.

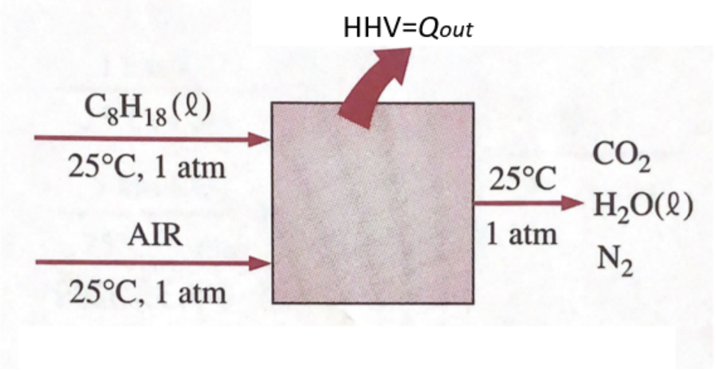


Figure 2.2: Combustion process describing HHV where substances are burned at reference state and products returned at reference state, water is in liquid phase [11]

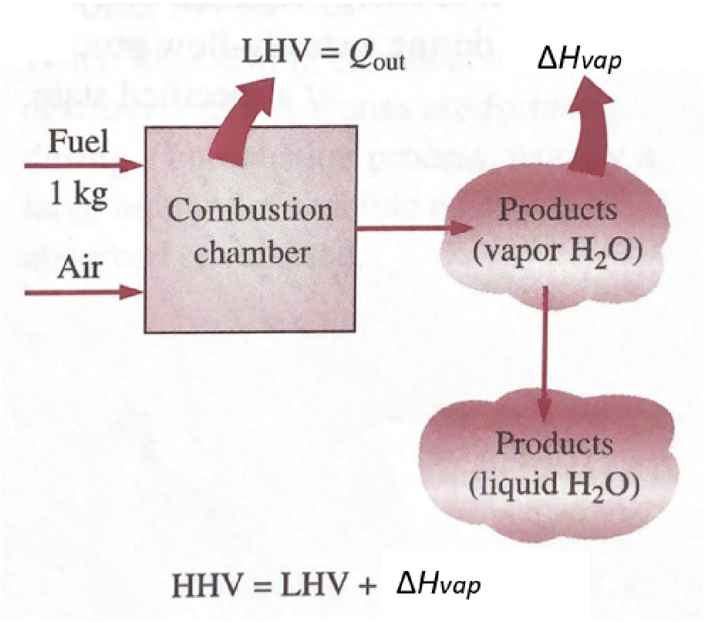


Figure 2.3: Combustion process showing LHV and the relation to HHV, water as product is returned as vapor which results in LHV. If the water is condensed HHV is achieved [11]

When setting up the energy balance for a combustion process and using heating value to represent the amount of heat released during the process, the reference state needs to be used to refer the enthalpies of incoming and outgoing streams to this state as they normally not are at the reference state. This is done by subtracting the enthalpy at the reference state, $h°$, from all terms. This can be seen in Equation 2.6 where $(\bar{h} - h°)$ represents the sensible enthalpy referred to its reference state. As mentioned, 1 bar, 25 °C, liquid phase is a common used reference state [11].If incoming and outgoing stream are not referred to the reference state, heating value is not usable in the energy balance since it is defined by inlet and outlet streams at their reference state [11]. Important to notice for an energy balance of a combustion reaction is that the energy balance should give the same result independent of what reference state is chosen. The heating value will be different for different reference states, but when incoming and outgoing streams are referred to the reference state it should give the same result.
When setting the energy balance based on the first law of thermodynamics for a reacting system, e.g. combustion, where heat is released or absorbed the following is given, Equation 2.6 [11]:

$$\sum(\dot{m}_{in}(h_f{}^\circ + (\bar{h} - h^\circ))_{in}) = \sum(\dot{m}_{out}(h_f{}^\circ + (\bar{h} - h^\circ))_{out}) \tag{2.6}$$

where *in* represents all substances entering the combustion and *out* the products leaving the combustion chamber after reaction. Mass flows, $\dot{m}$, is in kg per unit of fuel. It is important in all energy balances that $h_f°$and h°is taken at the same reference state.

This equation can be simplified using the heating value definition. Combining Equation 2.6 and 2.4 one can obtain Equation 2.7

$$\sum \dot{m}_{in}(\bar{h} - h^{\circ})_{in} = HHV + \sum \dot{m}_{out}(\bar{h} - h^{\circ})_{out} \tag{2.7}$$

Since HHV is negative for combustion processes, one can move HHV to the left side and always but is as positive, Equation 2.8.

$$\sum \dot{m}_{in}(\bar{h} - h^{\circ})_{in} + HHV = \sum \dot{m}_{out}(\bar{h} - h^{\circ})_{out} \tag{2.8}$$

If water produced in a reaction is condensed, and the condensation energy is utilized, one can use HHV to gain the thermal efficiency, see Equation 2.9.

$$\eta_{th} = \eta_{HHV} = \frac{W_{net}}{\dot{m}_{fuel} * HHV} \tag{2.9}$$

where $\dot{m}_{fuel}$ is the mass flow of fuel to the combustion chamber in kg/s.

Although, if the produced water is in gaseous phase and no condensation enthalpy is utilized, one would like to use LHV instead. Equation 2.8 then becomes 2.10 when the definition of LHV is used.

$$\sum \dot{m}_{in}(\bar{h} - h^{\circ})_{in} + LVH = \sum \dot{m}_{out}(\bar{h} - h^{\circ})_{out} - \Delta H_{evap} \tag{2.10}$$

The thermal efficiency can be define using LHV, according to Equation 2.11

$$\eta_{th} = \eta_{LHV} = \frac{W_{net}}{\dot{m}_{fuel} * LHV} \tag{2.11}$$

2.1.4 Pre-heat of feedwater

Heat losses in the condenser of a steam cycle is directly proportional to the quantity of working media entering the condenser. The quantity can be reduced by extracting parts of the working media at intermediate pressure of the turbines and using the extracted heat of the media to heat the feed water after the condenser. Pre-heating also reduces the heat supply during combustion as the feed water is already partly heated. This result is a higher thermal efficiency since it results in a higher average temperature of heat supply but the same temperature of heat removal. The electric power output of the turbines in a steam cycle is substantially dependent on the number of extractions and the mass flow passing through the turbines. The highest power output for a steam turbine including feed water heaters is achieved at an infinite number of extractions. This is because, the smaller temperature difference between the hot and the cold stream in a heat exchanger, the more efficient heat transfer and less losses can be obtained. With infinite number of extractions the streams can be matched perfectly. In practice this is limited by techno-economical considerations. The extractions are often designed to reach the same heat drop in all extractions to reach the maximum efficiency [13].

To find the optimum conditions (pressure and temperature) where extractions takes place, Laupichlers method can be used. It divides the enthalpy difference between the turbine extractions equally [13]. See Equation 2.12.

$$\Delta H = \frac{H_{initial} - H_{final}}{n_{extractions} + 1} \tag{2.12}$$

where $H_{initial}$ is the enthalpy of the inlet to the first turbine and H_{final} is the enthalpy of the outlet of the last turbine.
In a heat exchanger, the pinch point can be set to achieve the temperature difference wanted. Pinch point is the location in heat exchanger where the temperature difference between hot and cold fluid is minimum at that location, this is where the design is most constrained [11].

2.2 Hydrogen

With growing demand for energy and harder carbon dioxide capture policies, other energy source than fossil fuel needs to play a bigger role in the energy demand. Energy production also needs to be more efficient than ever and renewable energy sources such as solar and wind needs to be used in a more effectively way by store abundance energy in some way. Hydrogen is an energy source (or energy carrier) that is seen as a sustainable and environmental friendly since burning hydrogen results in only water and no emissions according to Reaction 2.13 [1].

$$H_2 + \frac{1}{2}O_2 \longrightarrow H_2O \tag{2.13}$$

The reaction makes it possible to burn hydrogen in pure oxygen (instead of air which is normally used) in a direct combustion chamber and high temperature steam is formed. Much higher temperatures than what can be achieved when burning coal or fossil fuels indirect in a Rankine cycle [1]. Combustion with pure oxygen in called oxy-combustion. Sanz et. al states than an excess on 3% oxygen to the stochiometric ratio is needed for full combustion of hydrogen [2]. Hydrogen has the ability to be stored and used when needed, used as an energy carrier. In this case, when wind or solar power is at its maximum production, the energy excess can be used to produce hydrogen that can be stored, in gas och liquid phase [2].

Hydrogen as a fuel has many advantages, for instance its high heating value per kilogram and its high energy converting efficiency. Hydrogen has a low heating value, LHV, of 120 MJ/kg and a high heating value, HHV, of 142 MJ/kg [1]. This can be compared to natural gas that has LHV of 40 MJ/kg and HHV of 45 MJ/kg [27] which gives hydrogen an energy content almost three times as much as natural gas, on a mass basis [4]. Hydrogen is more to be seen as an energy carrier than a fuel in this context. From that point of view it needs to be able to be stored efficiently but there are challenges when storing hydrogen. Hydrogen has the highest energy value per kilogram, but hydrogen is a challenging substance on a volume basis. Due to hydrogens light density and molecular weight, it occupies a big volume, liquid hydrogen has a density of 8 MJ/L whereas gasoline has a density of 32 MJ/L [4].

As an example, the volume of 324 g hydrogen is 3.9 L while 1 kg gasoline occupies 1.3 L [5]. Independent on the pressure, hydrogen has less energy content per volume than methane or other hydrocarbons which leads to challenges in gaseous storage [5]. This gives liquid hydrogen an advantage against gaseous hydrogen when storing.

Hydrogen can be produced by three main processes. Reforming of natural gas, gasification of biomass and electrolysis of water where the last one is regarded as the most sustainable one. Electrolysis is currently regarded as very expensive but the interest in the process is big and the process is seen as the futures sustainable way of hydrogen production [5].

2.2.1 Water electrolysis for hydrogen production

There are three main water electrolysis processes for hydrogen production, solid oxide electrolyzer cell (SOEC), polymer electrolyte membrane (PEM) and alkaline water electrolysis. The three methods have been compared and discussed by Schmidt et al. and PEM was concluded to be the futures electrolysis method [6]. PEM can be carried out under various conditions and pressures where high-pressure PEM is shown to be an effective electrolysis method and preferable compared to atmospheric PEM with subsequent compression. The optimal working pressure was proven by Marangino to be between 30 and 45 bar. PEM usually operates between 50 and 80 °C [16].

2.2.2 Liquefaction of gases

Liquefaction of gases can be made though multiple processes where two of the most common ones are the Linde-Hampson and the Claude cycle. When condensation of gases requires temperatures below -153°C to condense, one call the liquefied gas a cryogenic gas and more complex processes than normal refrigeration processes are needed to achieve liquefaction [10].

To reach below the boiling point of cryogenic gases, they are first compressed to a high-pressure level, cooled and then expanded in an expansion valve down to a low-pressure level, often ambient pressure. When the right pressures and temperatures are used during expansion, the gas condensates. The expansion depends on the Joule-Thompson (J-T) effect. The expansion through an expansion valve can be seen as isenthalpic as no work is added or removed and outer heat transfer and changes in kinetic energy normally can be neglected. Expansion of a non-ideal fluid will hence result in a change in temperature. This is the J-T effect and makes it possible to reach the cold temperatures needed for liquefaction of gases by expansion from high-pressure level to low-pressure level. The J-T effect can be explained by inter molecular forces for non-ideal fluids. The J-T coefficient is used to predict whether the temperature will increase or decrease when the gas is expand, it represents the rate of change of temperature with respect to pressure. Positive J-T coefficient results in an increase in temperature while negative coefficient results in a decrease in temperature during expansion. For an ideal-gas the coefficient is zero and the temperature is unchanged during expansion [10].

To achieve liquefaction, i.e. to lower the temperature during expansion, the gas needs to be at conditions (temperature and pressure) to the left of the curves in Figure 2.4 at the inlet of the expansion valve, where the J-T coefficient is negative. In other words, the gas needs to be pre-cooled ”into the negative coefficient area” before expansion. As can be seen in Figure 2.4, temperatures and pressures where air can be expanded and liquefied is higher than for hydrogen. The liquefaction of air is thus simpler to achieve than liquefaction of hydrogen and also less energy intensive [10].

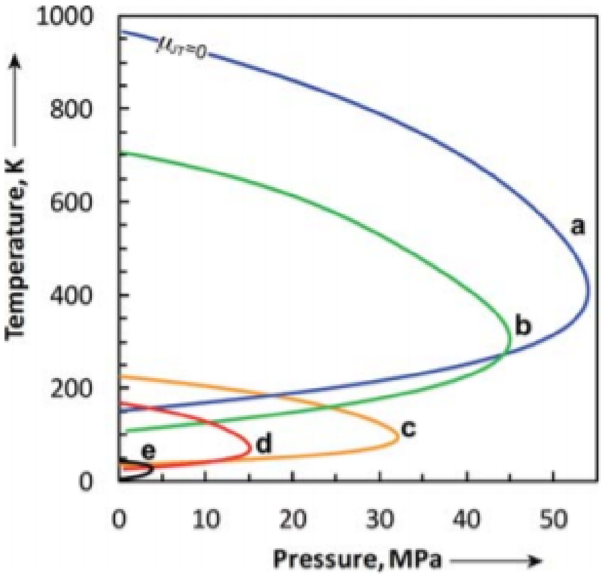
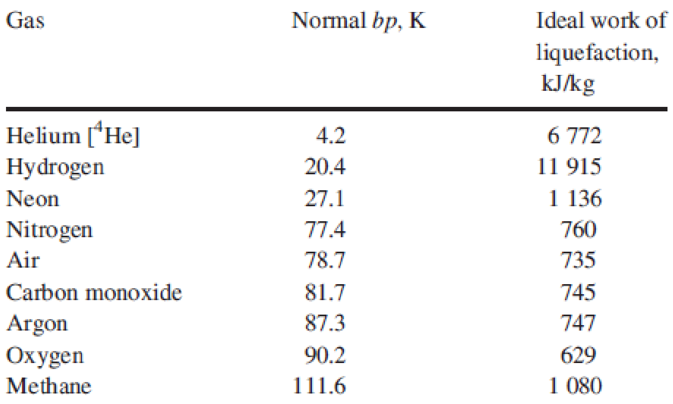


Figure 2.4: J-T inversion curves of common cryogenic fluids a) Methane, b) Air, c) Neon, d) Hydrogen, e) Helium. Cooling of isenthalpic expansion is only possible left (inside) of the respective curve [10].

The energy consumption for liquefaction of a number of cryogenic gases can be seen in the table in Figure 2.5. The figure shows that ideal work for liquefaction of oxygen is 82% of the ideal work for nitrogen. Ideal work for liquefaction of hydrogen is very energy consuming which is partly a result of its low boiling point of 20.4 K, -252°C.



Gas	Normal *bp*, K	Ideal work of liquefaction, kJ/kg
Helium [^{4}He]	4.2	6 772
Hydrogen	20.4	11 915
Neon	27.1	1 136
Nitrogen	77.4	760
Air	78.7	735
Carbon monoxide	81.7	745
Argon	87.3	747
Oxygen	90.2	629
Methane	111.6	1 080

Figure 2.5: Boiling points and ideal work requirements for liquefaction of gases beginning at 300 K and 1 bar [10]

2.2.2.1 Linde-Hampson liquefaction process

The Linde-Hampson process is the least complex cycle for liquefaction of cryogenic gases, see Figure 2.6 for schematic picture and 2.7 for T-s-diagram of the process. It consists of compressor (point a) compressing the gas to the high-pressure level, heat exchanger (point b) cooling the gas, expansion valve (point c) expanding to low-pressure level and separation unit (point d) separating liquid and vapor. The expansion valve is the unit needed to liquefy a gas using the J-T effect described in previous section. For ideal liquefaction, compression is ideally made isothermal (infinite number of compressor units with inter-cooling in between) and the expansion in the valve should be made isentropic [10]. The high-pressure level after compression depends on the gas. Normally around 200 bar for liquefaction of air. Most gases are then expand to a pressure between 1-2 bar, and stored in atmospheric pressure tanks [10].

Air and oxygen can be liquefied by the Linde-Hampson process thanks to its wide range of temperatures and pressures where the J-T coefficient is negative, see Figure 2.4. This results in the fact that not that much pre-cooling is required to be able to expand the gas down to cooler temperatures. Air and oxygen is able to ”cool itself” with returning vapor in the Linde-Hampson cycle. On the other hand, for hydrogen it is impossible to ”cool itself” down to such low temperatures only using returning vapor in the Linde-Hampson cycle [10].

Both high- and low-pressure levels during liquefaction can be chosen by the operator, although, the high-pressure level needs to be chosen, and cooled, to the point (point 3 in Figure 2.6) where isenthalpic expansion ends up in the two-phase region where the J-T coefficient is negative. It is also important to choose a low-pressure level that lies within the two-phase region (below the critical pressure) to be able to liquefy the gas. The optimum pressures to minimize the work for liquefaction is found numerical [10].

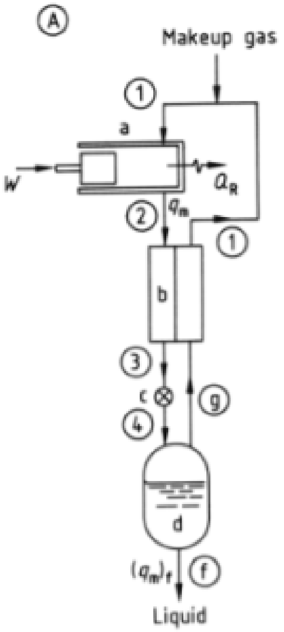


Figure 2.6: Schematic figure of Linde-Hampson liquefaction process for liquefaction of gases. a) Compression, b)Heat exchanger, c)J-T expansion valve, d) separation unit [10]

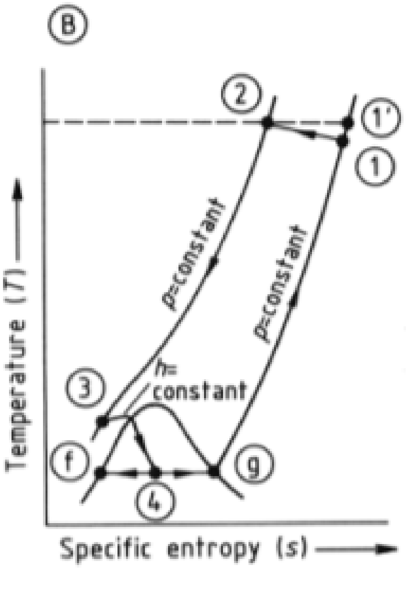


Figure 2.7: T-s-diagram of Linde-Hampson liquefaction process. 2 to 3: isobaric cooling, 3 to 4:isenthalpic expansion using J-T effect, point 4: shows fraction of liquid and vapor, 4 to 1: isobaric heating, 1 to 2: compression with intercooling [10]

2.2.2.2 Claude liquefaction process

The Claude cycle is similar to the Linde-Hampson cycle but with a turbine, see Figure 2.8. A portion of the high-pressure flow (point 3), is send thorough a turbine

(point c), expanding the flow and mixing it back (between point 7 and 8). This increases the amount of returning cold flow and the cycle is able to liquefy even hydrogen since it can ”cool itself” without need for external cooling. The addition of a turbine improves the performance significantly when partly replacing the isenthalpic expansion with an isentropic one. Gained energy in the turbine can also be used to drive the compressor [10].
The operational pressure is lower in the Claude cycle compared to the Linde-Hampson cycle. When comparing air liquefaction, 17 bar can be used in the Claude cycle instead of 200 bar as in the Linde-Hampson cycle [17]. Hydrogen is industrially liquefied using the Claude cycle [18].

Hydrogen is normally compressed to around 50 bar [19] and oxygen to around 60 bar [10] in the Claude cycle. The hydrogen Claude cycle consumes between 10.8-12.7 kWh/kg LH_2 (39-46 MJ/kg LH_2) [20] but can be minimised theoretically to 3.9 kWh/kg LH_2 (14 MJ/kg LH_2) when liquefied from ambient temperature to ambient pressure [21]. This is 10 times the more energy than ideal liquefaction of LNG [22].

No energy consumption for liquefaction of oxygen has been found for the Claude cycle, although it is clear from Figure 2.5 that the energy consumption for liquefaction of oxygen is 82% of liquefaction for nitrogen. Liquefaction of nitrogen using the Claude cycle is 2.2 MJ/kg LN_2. Consequently, the energy consumption for oxygen could be assumed to be 1.8 MJ/kg LO_2

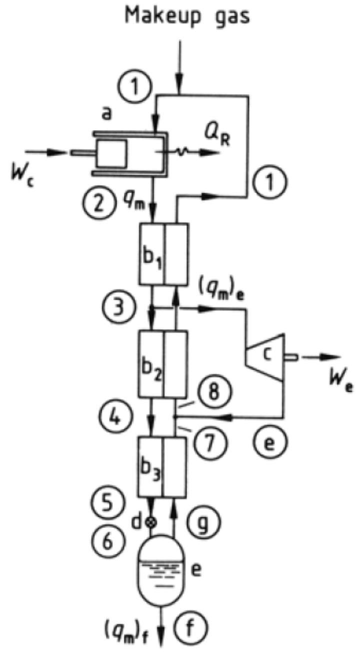


Figure 2.8: Schematic figure of Claude liquefaction process for liquefaction of gases. a) Compression, b1-3) Heat exchanger, c) Turbine d) Expansion valve e) Separation unit [10]

2.2.3 Hydrogen power cycles

To convert hydrogen to energy, so called gas-to-power, mainly two processes can be used. Fuel cells or power cycles. For power units with capacities above 10 MW it is shown that hydrogen fuelled power cycles are preferable over fuel cells, it also has a shorter start up time [23].

A number of concepts of hydrogen fuelled power cycles, both steam cycles and combined cycles have been studied. For instance the Graz cycle [2], the Toshiba cycle [8], the Westinghouse cycle [8] and the MNRC cycle [8]. Multiple configurations were created during a Japanese program, WE-NET, started in 1993 where the goal was to develop high efficiency technologies converting hydrogen to energy. An article by Milewski [8] is a summary of configurations created during this project and the goal of these cycles has thus been to reach a very high efficiency [8]. The common design for the power cycles is oxy-combustion that produces high temperature steam, a condenser changing the phase of the water, a heat exchanger that in some way heat exchanging between the hot steam, and condensed, returning feed water and a pump or compressor.

The Graz cycle is an example of a combined cycle with oxy-combustion, see Figure 2.9 for cycle configuration. The Graz cycle was originally designed by Jericha in 1985 for oxy-combustion of fossil fuels with separation of carbon dioxide to retain carbon dioxide generated by combustion. The concept has evolved through the years and in 2016, Sanz et al. redesigned and adapted the cycle for combustion of hydrogen [2]. The combustion chamber pressure of the cycle is 40 bar. The cycle has an efficiency, η_{LHV}, of 68.43%, producing 205 MW of net electricity [2].

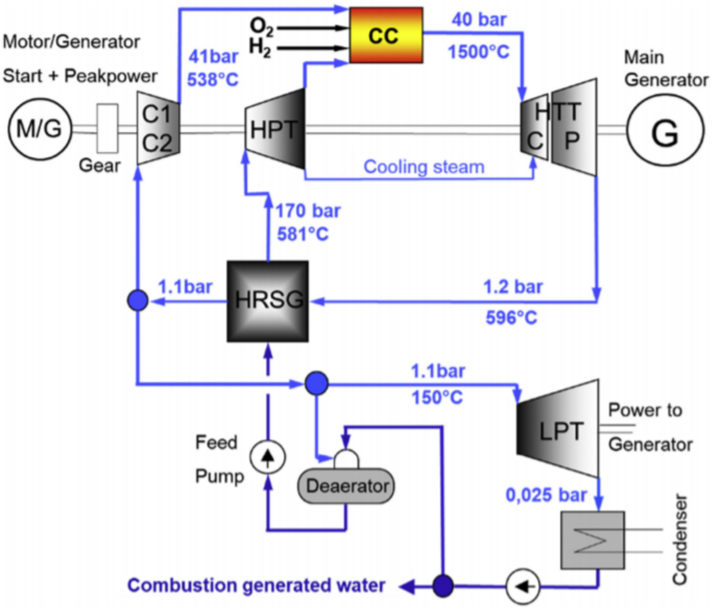


Figure 2.9: Hydrogen fuelled cycle configuration of the Graz cycle by Sanz et al. [2]

Milewski summarised and compared three hydrogen cycles in 2015, the Graz cycle, the Toshiba cycle and the Westinghouse cycle. The cycles are designed for 500 MW net electrical output and the resulting efficiencies, η_{LHV}, are 69% 69.2% and 72.2% [8]. The Toshiba cycle and the Westinghouse cycle are similar to a conventional Rankine cycle with double combustion and with a heat recovery steam generation (HRSG). The Toshiba cycle has a pressure of 73 bar in the combustion chamber and Westinghouse cycle 250 bar. The difference between the cycles is an extra turbine in the Toshiba cycle. See Figure 2.10 and 2.11 for cycle configurations [8].

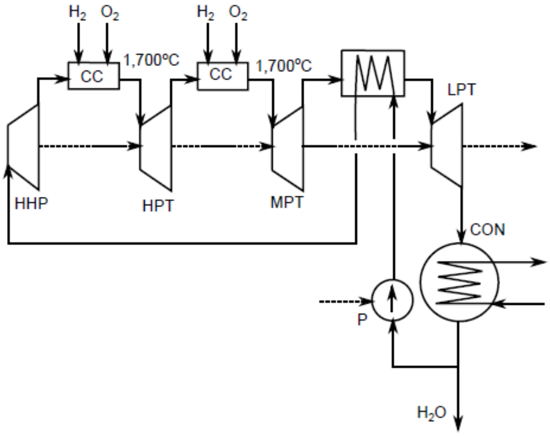


Figure 2.10: Hydrogen fuelled cycle configuration of the Toshiba cycle by Milewski [8]

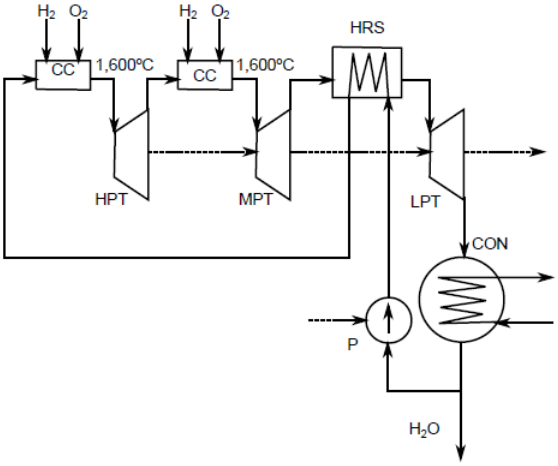


Figure 2.11: Hydrogen fuelled cycle configuration of the Westinghouse cycle by Milewski [8]

Milewski also proposed a new cycle, the MNRC cycle, which is an optimisation based on the Toshiba and the Westinghouse cycle. The MNRC cycle is the Toshiba cycle with an additional combustion chamber and turbine. The cycle has a pressure of 250 bar at the combustion chamber. The efficiency, η_{LHV}, of the cycle is 77% for 500 MW net electrical output [8]. See Figure 2.12

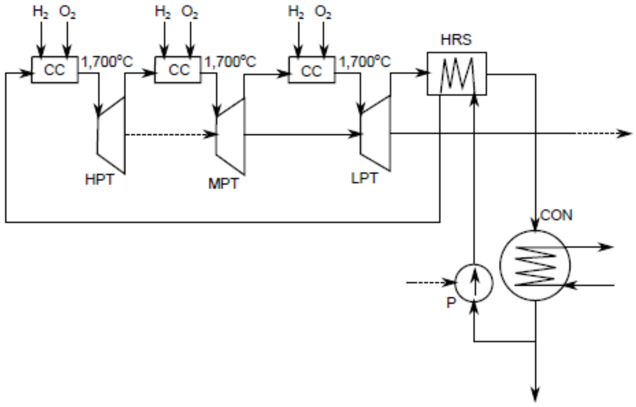


Figure 2.12: Hydrogen fuelled cycle configuration of the MNRC cycle by Milewski [8]

Comparing the cycles, the Graz cycle has a relatively complex design compared to Toshiba, Westinghouse and MNRC cycle. It can also be concluded that the cycles with best efficiency (Toshiba, Westinghouse and MNRC cycle) are the cycles similar to the Rankine cycle and pumps are used for increasing the pressure of the steam instead of compressors as in the Graz cycle. Compressors are energy intensive but with the ability to compress gas. In pumps, one should avoid gas and have the working media at liquid phase. This is because one need to make sure to avoid cavitation in the pump as it can damage the unit. This can be achieve by letting sub-cooled liquid enter the pump. All the different cycles also consists of a HRSG utilizing the whole mass flow. This results in big and expensive heat exchangers compared to if only a fraction of the mass flow is led though the heat exchanger [13].

It is also clear form the article by Milewski that multiple combustion chambers increases the efficiency. Although, the number of combustion chambers in relation to increase in efficiency is a techno-economic question. What all the mentioned hydrogen cycles also have in common is that hydrogen and oxygen are expected to be supplied freely at ambient temperature and required pressure to the combustion chamber, which is between 40 and 250 bar for the different cycles [8][2].

3

Methods

In the following section, the method of the project is describes. It consists of a literature review, modelling and sensitivity analysis.

3.1 Literature review

An initial literature review was done to cover information about heat engines and how to optimise them. Additional literature study about hydrogen, hydrogen power cycles and liquefaction of gases have been made to cover previous work and to gain knowledge about existing processes and configurations. Previous work with hydrogen power cycles is also important to include in the work to be able to compare and evaluate results.

3.2 Modelling

The modelling in the project has been divided into three parts. Modelling of a hydrogen power cycle, re-modelling of the Graz cycle by Sanz et al. [2] and modelling of liquefaction of hydrogen and oxygen. The models of hydrogen power cycle, the Graz cycle and liquefaction processes has been made gradually in combination with literature study and adjustments and developments of some components.

3.2.1 IPSEpro

Modelling of thermodynamic cycles have been made in IPSEpro which is a component based program. A component library is included in the program where components such as combustion chambers, turbines, pumps etc. are possible to choose. The components are programmed based on energy balances. Components are connected via streams where the composition of substances in each stream can be chosen. The system of components and streams creates a set of thermodynamic equations where a number of input data of temperatures and pressures are needed to solve the whole system. For the thermodynamic calculations, REFPROP has been used to obtain necessary fluid data. Some components in the library of IPSEpro has been adjusted and developed to fit the project. For instance, the combustion chamber has been adjusted and programmed to be able to burn hydrogen instead of methane and the energy balance of the combustion chamber has been programmed to be Equation 2.10.

Microsoft Visual Studio 2017, with the programming language C++, has been used to connect IPSEpro to REFPROP data. This connection has been made previous by the performance department at Siemens Energy where the thesis has been carried out.

3.2.2 Hydrogen power cycle

The hydrogen power cycle is modelled as a Rankine cycle with pre-heat. Pressures at turbine extractions has been chosen to achieve equal enthalpy difference over all turbines, according to Laupichlers rule [11]. Atmospheric pressure and temperature are assumed to be at 1 bar and 15°C. Water used for cooling or heating to ambient temperature are assumed to be supplied freely. Pressures and temperatures used in simulation are based on existing steam power plant data by GE Steam Power, from ultra-super critical steam power technology [24]. Pressure and temperature at the outlet of combustion chamber are thus decided to be 300 bar and 1400°C. Temperature at the first turbine extraction was decided to be 650°C [24]. These parameters are identified as key parameters in the cycle and are later on evaluated in a sensitivity analysis to analyse their impact on the efficiency. 300 bar, 1400°C and 650°C are named the default case during sensitivity analysis. The first turbine in the power cycle needs to be a cooled turbine since temperatures above 650°C is to hot for the material to handle without cooling. Remaining turbines can be regarded as normal steam turbines.

Mixed temperature is used at the inlet to the first turbine, according to the ISO standard for gas turbines [25]. It is used as a simplification for thermodynamic calculations of cooled turbines, since the cooling of turbines is a rather complex area. This implies that the turbine inlet temperature is defined arbitrary as a theoretical flow-weighted mean temperature before the first turbine stage. In most cases it means that exiting stream from combustion is mixed with cooling fluid prior to the turbine inlet, instead of continuously throughout the turbine. In this model, the mixing is done inside the combustion chamber and included in the energy balance of the combustion. Thus, the temperature at the combustion outlet in this model is the mixed temperature. The efficiency of a turbine needs to lowered compared to the real isentropic efficiency when mixed temperature is used to include losses occurring for a cooled turbine. The efficiency also needs to be lowered to make up for the extra energy output achieved when adding all cooling fluid before the inlet instead of continuously. This is because a bigger mass flow creates more energy output. The efficiency is thus a fictional one for a turbine with mixed temperature.

Condenser pressure is given from saturation pressure and pinch point in the condenser. The cooling water used in the condenser is 15°C, which gives the temperature of the condensed steam at the outlet of the condenser, assumed to be at saturation, using the pinch point in the condenser. The saturated temperature is correlated to its saturation pressure and the condenser pressure is achieved.

Fixed parameters, efficiencies and losses used in the hydrogen cycle is presented in

Table 4.1. These values have been concluded by internal discussion with Siemens Energy and the performance department [26] except for temperatures and pressure achieved from GE Steam Power [24] marked in the table with its reference.

Parameter	**Value**
Ambient temperature	15 °C
Ambient pressure	1 bar
Condensing pressure	0.034 bar
Combustion pressure	300 bar [24]
Turbine inlet temperature	1400°C [24]
First turbine extraction temperature (Highest possible temperature without cooling of material)	650°C [24]
Isentropic efficiency turbine	88% HPT, 92% IPT, 90% LPT
Combustor pressure loss	4.5%
Condenser pinch point	11°C at hot outlet, 3°C at saturation hot inlet and cold outlet
Pump efficiency	76%
Heat exchanger pinch point	25°C
Pre-heater pinch point	3°C
Generator electrical efficiency, η_g	98,5%
Overall mechanical efficiency, η_m	99%
Extraction pressure drops	1%

Table 3.1: Parameters for modelling of hydrogen power cycle [26]

A pressure drop of 1% added to all turbine extractions represents the pressure drop occurring in heat exchangers in the pre-heating chain.

Efficiency of the cycle, including mechanical and generator losses is shown in Equation 3.1. The efficiency is based on LHV since most of the vaporisation energy is not utilized in the condenser, and also to easily be able to compare with efficiencies from previous work that are all based on LHV.

$$\eta_{LHV} = \frac{\eta_g \eta_m W_{net}}{\dot{m}_{fuel} LHV} \tag{3.1}$$

where

$$W_{net} = W_{turbines} - W_{pumps} \tag{3.2}$$

Optimisation of the cycle has been made with the aim to improve the thermal efficiency. Although, with an economic view in the background. For instance if the efficiency is not increased enough when adding and extra turbine extraction it can be seen as unnecessary as the material cost increases with more units.

The hydrogen power cycle model is valid for reference temperatures below saturation point. Since the energy balance in the combustion chamber component is based on LHV, as explained in Section 2.1.3, and the reference state is 25°C, 1 bar, liquid phase, the energy not received by condensation of exiting water, ΔH_{vap}, needs to be subtracted from the LHV in the energy balance since water leaves the combustion chamber i gaseous phase. If the reference state would be above saturation, the evaporation enthalpy should be removed from the energy balance, this is not included in the programming, the evaporation term is always present.

3.2.2.1 Graz cycle

The Graz cycle modelled by Sanz et al. [2] has been modelled in IPSEpro as well during the project. The aim of this was to be able to evaluate the method used when modelling in this project compared to the method used by Sanz et al. In this way, it could be ensured that the method and components used in this project is the same, or at least similar, as the method by Sanz et al. Since the Graz cycle by Sanz was modelled in IPSEpro as well, the comparison between the model by Sanz et al. and the newly made model was easy. The new Graz cycle was modelled using the same steam parameters and values as in the Graz cycle by Sanz et al. Especially, it was made sure that enthalpies and massflows in all streams were the same. Pressures and temperatures could differ sligtly in some streams.

In the model by Sanz et al. a turbine component was developed for the calculation of cooled turbine stages. A simple stage-by-stage approach was used in the model that made it possible to calculating the amount of cooling steam needed per stage. The turbine component assumed that half of the cooling mass flow was mixed to the main flow at stage inlet and the rest was added at the stage exit [2].

The development of cooled turbine components in IPSEpro is outside the scope of this project, therefore the mixing of cooling water to the main stream has been made prior to the inlet of each turbine, as a so called mixed temperatures described in Section 3.2.2. This affects the energy output of the turbine since the mass flow through the turbine is bigger in this case than in the model by Sanz et al. To do the models comparable, the efficiencies of the turbines was adapted to result in the same enthalpy differences as in the model by Sanz. As an example, Sanz et al. uses an isentropic efficiency of 92% at the HPT, this efficiency was decreased to 90% in the new Graz model. The same mass flows and enthalpies of both main stream and cooling water at the inlet to each turbine was made sure to be the same as in the model by Sanz. As the mass flows are the same, and the enthalpy difference, the energy output is also equal between the two models and the overall efficiency of the model was similar.

When the main stream out of the combustion chamber in the new Graz cycle at 1500°C was mixed with the cooling stream (at 364°C) the mixed temperature at the turbine inlet to the HPT turbine became 1370°C in the new model of the Graz cycle. Since the turbine inlet temperature to the first turbine is an important parameter for the efficiency, the mixed temperature in the Graz cycle was increased to be 1400°C, same as in the modelled hydrogen cycle. In this way the Graz cycle and the hydrogen cycle can be fair compared. To increase the mixed temperature in the Graz cycle, the outlet temperature of the combustion chamber was increased to 1540°C. The difference between the outlet temperature of the combustion chamber and the mixed temperature is thus 150°C. This new model of the Graz cycle with mixed temperature of 1400°C is henceforth called ”New modelled Graz cycle”.

The 150°C between mixed and ”real” temperature found when modelling the Graz cycles was later on used as the temperature difference between ”real” temperature and mixed temperature to be able to calculate the mixed temperature of the Toshiba, Westnighouse and MNRC cycle to be able to compare all the cycles. Although, important to notice is that the amount of cooling, and thus also the temperature difference, increases as the turbine inlet temperature increases. 150°C might thus be an underestimation when the real turbine inlet temperature is 1700°C.

3.2.2.2 Liquefaction

As hydrogen and oxygen is optimal delivered by PEM electrolysis between 30 and 45 bar and 50 and 80 °C, 40 bar and 65°C was decided to be the parameters where hydrogen and oxygen is added to the liquefaction process. As no reliable source of liquefaction of oxygen in the Claude cycle has been found, the energy consumption is compared to liquefaction of nitrogen using the Claude cycle. The energy consumption for liquefaction of oxygen is assumed to be 82% of the energy consumption for nitrogen, as explained in Section 2.2.2.

The aim of modelling the liquefaction processes in not to optimise the processes but rather to reflect the commercial liquefaction processes and include the processes in the efficiency. Pressures, temperatures and cycle configurations are therefore based on that information. Literature presenting energy consumption of liquefaction processes has in many cases not presented data of what pinch points or how many compressors and inter-coolers have been used. Therefore, 3 compressors with inter-cooling and pinch points of 10°C was decided to be used in these models because energy consumption close to literature was achieved then. Fixed parameters, efficiencies and losses are presented in Table 3.2, these values have been concluded by internal discussion with Siemens Energy and the department of performance [26].

Parameter	**Value**
Ambient temperature	15 °C
Ambient pressure	1 bar
Inlet temperature hydrogen	65°C
Inlet temperature oxygen	65°C
Inlet pressure hydrogen	40 bar
Inlet pressure oxygen	40 bar
Compressor isentropic efficiency	85%
Turbine isentropic efficiency	88%
Pump efficiency	76%
Heat exchanger pinch point	10°C

Table 3.2: Parameters for modelling of liquefaction process [26]

3.3 Sensitivity analysis

Sensitivity analyzes was performed on the hydrogen power cycle to analyse how some steam parameters affects the performance and the efficiency of the cycle. Pressure of the combustion chamber, mixed turbine inlet temperature, temperature of the first extraction and number of extractions was analysed. When changing one parameter, all other parameters were set to constants. Default parameters for sensitivity analyses have been 300 bar and 1400°C at the combustion chamber outlet and 650°C at the first turbine extraction. For instance, when the pressure was analysed, the mixed turbine inlet temperature and the first extraction temperature was kept constant at 1400°C and 650°C. What the actual, highest possible temperatures and pressure are at the different components is outside the scope of this thesis. The sensitivity analysis shows how different temperatures and pressures will affect the efficiency but what is possible in the future is not included. Temperatures and pressures used in the sensitivity analysis was chosen to be 300-400°C higher and lower than the default temperatures and 300 bar higher and lower than the default pressure.

No sensitivity analysis was done for the liquefaction processes since the goal of those models were to reflect industrial liquefaction. Although, some basic parameter analysis have been done to investigate what parameters affects the energy consumption of the liquefaction process.

4

Results

In the following section, the resulting models are presented. First the modelled hydrogen power cycle, which is compared to the other power cycles and also the new modelled Graz cycle with adapted temperature. The sensitivity analysis of the power cycle is also presented. Second, liquefaction processes for liquefaction of hydrogen and oxygen are presented and third, the modelled Graz cycle is presented and compared to the model of the Graz cycle by Sanz et al. to conclude what differences could be found between the methods used when modelling in IPSEpro.

4.1 Hydrogen power cycle layout

A simplified, schematic picture of the hydrogen cycle modelled in this project can be seen in Figure 4.1 and its corresponding T-s-diagram in Figure 4.2. Note that turbine extractions at different pressure-levels are not included in the T-s-diagram. Also, only 4 out of 8 turbine extractions are presented in the simplified schematic picture of the model. Default values of 300 bar, 1400°C and 650°C can be seen in the cycle configuration. The cycle is modelled assuming that hydrogen and oxygen can be supplied ”freely” at ambient temperature and required pressure as in Sanz et al. [2] and Milewski [8]. In this way, the cycles can easily be compared. The full and exact model of the hydrogen cycle is seen in Appendix, Figure A.1.

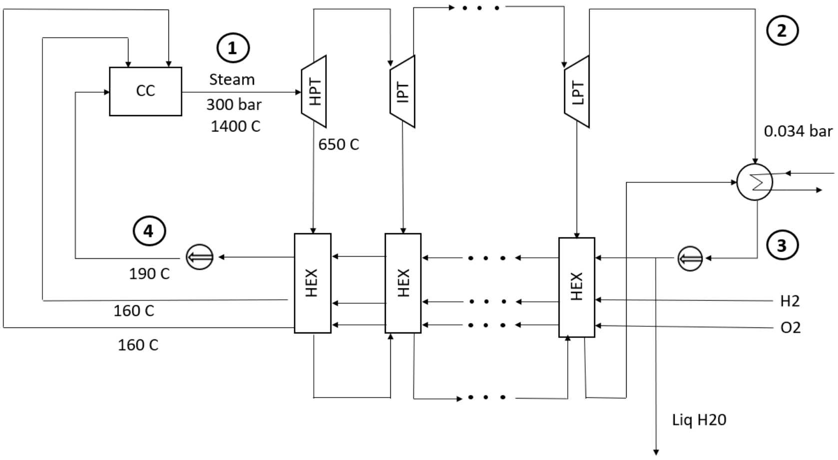


Figure 4.1: Simplified schematic picture of modelled hydrogen power cycle. A few important steam parameters are presented in the figure. 1 is high temperature at the outlet of the combustion chamber and the inlet to the first turbine, 2 is the last turbine expansion, 3 is condensed water, 4 is pre-heated feed-water

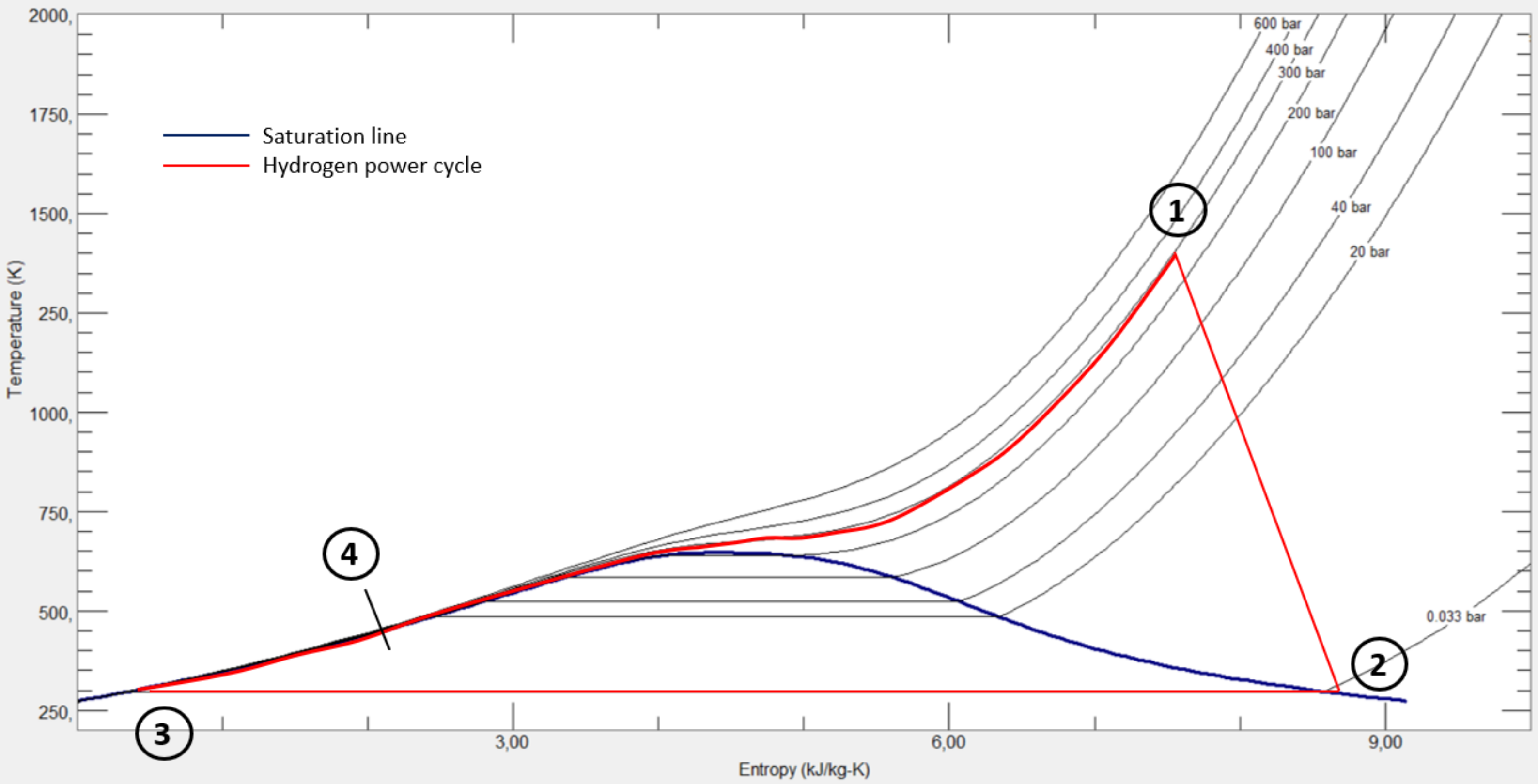


Figure 4.2: T-s-diagram for hydrogen power cycle. 1 to 2 expansion, 2 to 3 condensation, 3 to 4 pre-heating, 4 to 1 combustion.

The model has direct combustion where hydrogen is burned in pure, stochiometric oxygen to produce high-temperature water steam. For each kg of hydrogen, 7.94 kg of oxygen is combusted. The returning feed water works as cooling media in the combustion chamber to keep the temperature down. The model is a Rankine cycle with pre-heating. The first turbine is a cooled turbine where the mixing of cooling water and hot steam is modelled inside the combustion chamber component, as described in Section 3.2.2. The temperature entering the HPT is therefor higher than 1400°C, possibly around 150°C higher as found when modelled the mixed temperature for the Graz cycle, see Section 3.2.2.1

The model contains 8 turbine extractions. The extracted streams are condensed in pre-heaters. Then either pumped forward to the previous pre-heater or led backward to the next pre-heater as shown in Figure 4.1 where all streams are led backward. The last extraction is led trough the condenser, then pumped and led through the pre-heating chain. Around 80% of the mass flow exiting combustion chamber is led through all turbines and also thorough the condenser. The feedwater is pumped in multiple stages, prior and after pre-heaters, see Appendix, Figure A.1 for exact configuration. Feed water is heated to 190°C before entering combustion chamber. Oxygen and hydrogen are also led though the pre-heating chain to increase their inlet temperatures to the combustion chamber, they are increased to 160°C.

Since water is produced during combustion by the reaction of hydrogen and oxygen, water needs to be removed from the cycle to keep the amount of water constant. Water is removed in liquid phase at the nearest pressure above atmospheric pressure. I.e. after the pump after the condenser.

The efficiency of the power cycle with hydrogen and oxygen supplied freely at 300 bar is 67.58%. The efficiency is compared to other hydrogen power cycles in Table 4.1. It is assumed that the difference between mixed temperature and "real" temperature is 150°C to be able to compare temperatures.

Power cycle	$\eta_{\mathbf{LHV}}$	Pressure combustion chamber [bar]	Extraction temperature [°C]	Turbine inlet temperature [°C]	Mixed temperature [°C]	N.o. combustion chambers
Graz by Sanz et al.	68.4%	40	596	1500	1370	1
New modelled Graz	69.8%	40	596	1540	1400	1
Toshiba	69.4%	73	-	1700	1550?	2
Westinghouse	72.2%	250	-	1700	1550?	2
MNRC	77%	250	-	1700	1550?	3
Hydrogen cycle	67.6%	300	650	1550	1400	1

Table 4.1: Summation of hydrogen power cycles parameters and efficiencies

The modelled hydrogen cycle has the highest pressure of all cycles but lower turbine inlet temperature than all except for the Graz cycle. It also has lower efficiency than all other cycles. The cycle has 2.2%-units lower efficiency than the new Graz cycle which has been modified to have the same mixed temperature. The efficiencies of the Toshiba, Westinghouse and MNRC cycles are 1.8-9.4%-units higher than the hydrogen cycle. But they all have 150°C higher mixed temperature. They also have two or three combustion chambers which is commonly known to increase the efficiency significantly. The goal for the cycles in the article that Milweski summarised was to reach very high cycle efficiencies as a part of the WE-NET program, which forced the models to use up to three combustion chambers and high turbine inlet temperatures. Increased number of combustion chambers means an increase in hot units that also increases the costs. The hydrogen cycle modelled in this project is more realistic and cheaper with only one combustion chamber. It is also less complex than both the Graz, Toshiba, Westinghouse and MNRC cycle which is an advantages itself. A cheaper and simpler cycles is more likely to be build in the near future. There is an additional advantages in space when heat exchanging only 20% of the flow with two phase flow on one side and liquid on the other compared to exchanging heat between the entire flow at gas phase which is the case in the HRSG. In that case, the HRSG occupies a lot of space compared to the pre-heater and will also be more expensive. The phase change in the pre-heaters makes the heat exchange much more effective than heat exchanging with gas. The big advantages for the Graz cycle is its low pressure level of only 40 bar.

To increase the efficiency of the cycle, to reach more similar efficiencies as the Westinghouse and MNRC cycle, additional combustion chamber would be interesting to investigate. It would definitely increase the efficiency as the high temperature is reached twice. This increases the area in a T-s-diagram and would increase the efficiency. This is more of a techno-economical question handling the relation between efficiency and cost.

4.1.1 Sensitivity analysis of hydrogen cycle

Four sensitivity analyzes was done on the hydrogen cycle to evaluate how different steam parameters affect the performance of the cycle. Temperature and pressure at the outlet of the combustion chamber, the temperature of the first extraction and the number of extractions was evaluated. Figure 4.3 shows number of extractions against efficiency. The turbine extraction temperature at the first turbine is 650°C in this analysis. The result of the analysis resulted in a model with 8 turbine extractions as the increase in efficiency decreases with increased number of extractions. The increase between 7 and 8 extraction is only 0.07 %-units and 8 extraction was therefore decided to be enough.

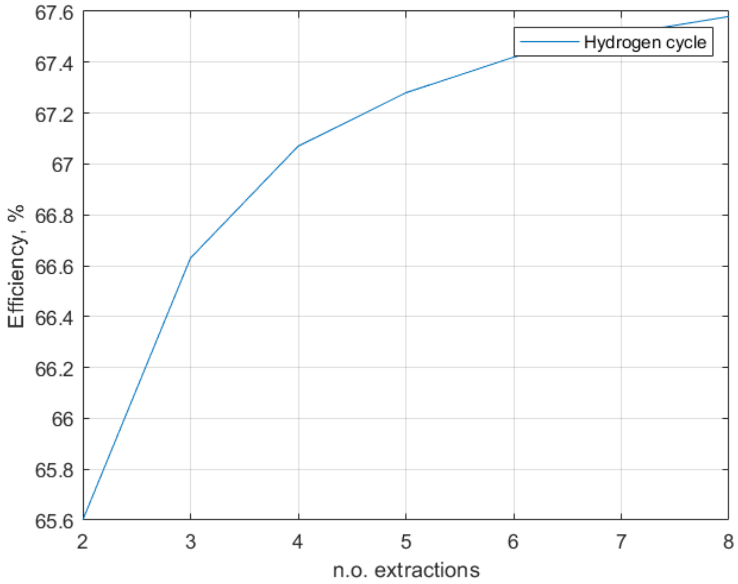


Figure 4.3: Sensitivity analyse showing number of extractions versus efficiency for the hydrogen cycle

The explanation to the decreasing slope in Figure 4.3 is that independent on how many extractions are used, around 20% of the total mass flow is extracted from the flow. So when the number of extractions are increased, some more mass flow is expanded further to a lower pressure and, some more energy output is thus achieved and the efficiency increases. But as more and more extractions are added to the model, the extra energy output achieved when expanding a bigger share of the mass flow to a lower pressure get smaller in comparison to the total energy output which results in the decreasing slope of the curve seen in Figure 4.3.

Temperature of the first turbine extraction against efficiency is shown in Figure 4.4. The slope is almost linear and the efficiency increases as temperature increases.

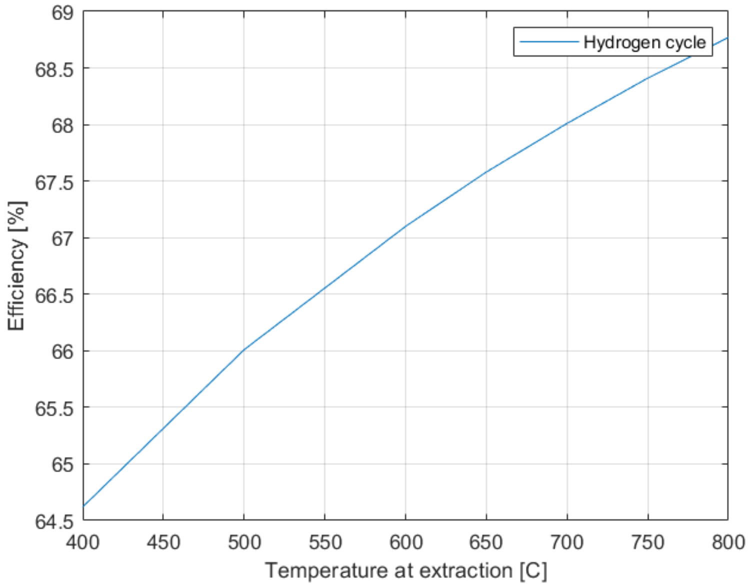


Figure 4.4: Sensitivity analysis showing mixed temperature of first turbine extraction versus efficiency in hydrogen cycle

It can be seen in Figure 4.4 that the efficiency is driven by temperature. As this curve is linear compared to the curve for number of extraction which has a decreasing slope, one can conclude that high turbine extraction temperature is more important than number of extractions. Especially as the number of extraction is above 4 because that is where the slope is decreasing fast. Of course, a combination of both of them is optimal but as mentioned previous, the increase in %-units when adding an extraction above 4 extraction is relatively small.

Pressure at the outlet of combustion chamber against efficiency is shown in Figure 4.5. It clearly shows that an increased pressure increases the efficiency but the slope of the curve decreases as pressure increases. The efficiency increases 6.8%-units from 40 to 300 bar and 2.35%-units from 300 to 600 bar. It can be seen that all other cycles has higher efficiencies than the hydrogen cycle at pressures up to 500 bar. At 500 bar the efficiency of the hydrogen cycle is similar to the Graz and Toshiba cycle, but that is probably an unrealistic pressure. But most of these cycles also has higher turbine inlet temperature.

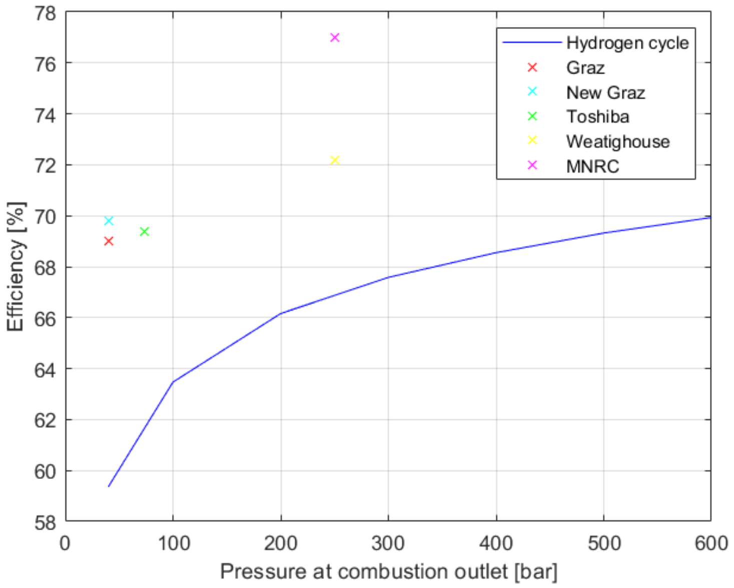


Figure 4.5: Sensitivity analysis showing pressure at outlet of combustion chamber versus efficiency for hydrogen cycle

The mixed temperature at the outlet of the combustion chamber, i.e. the turbine inlet temperature, is shown against efficiency in Figure 4.6. The curve is almost linear and increased temperature increases the efficiency. This shows that efficiency is very driven by temperature. Again, the efficiency is lower in most cases compared to the other cycle, one need to increase the mixed temperature to around 1600°C to reach the same efficiency as the Graz cycle and 1700°C to reach the efficiency of the Toshiba cycle.

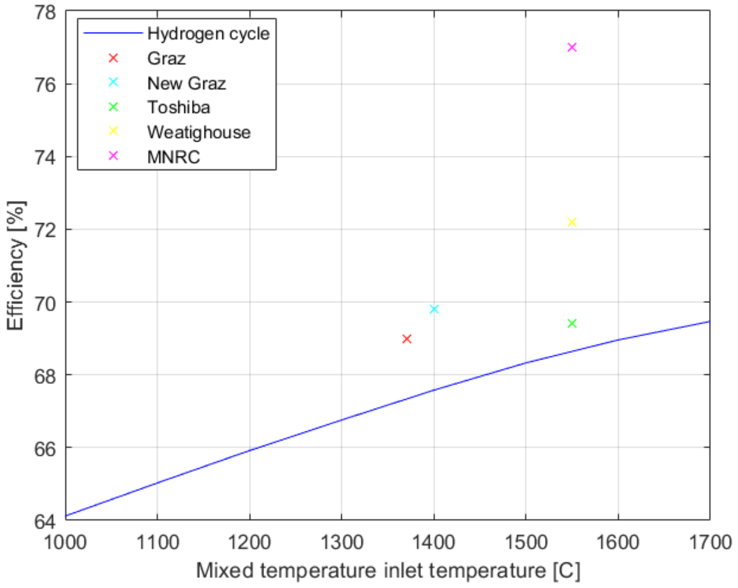


Figure 4.6: Sensitivity analysis showing mixed turbine inlet temperature versus versus efficiency for hydrogen cycle

From the sensitivity analysis it can be consulted that increased pressure and temperatures increases the efficiency. This is simply because higher steam parameters increases the difference between the high temperature source and the low temperature sink and according to the carnot efficiency the efficiency of the cycle is increased. In a T-s-diagram this is explained by an increase of the area under the graph. Figure 4.7 shows a T-s-diagram of the default hydrogen steam parameters (red curve) together with some cases presented in the sensitivity analysis.

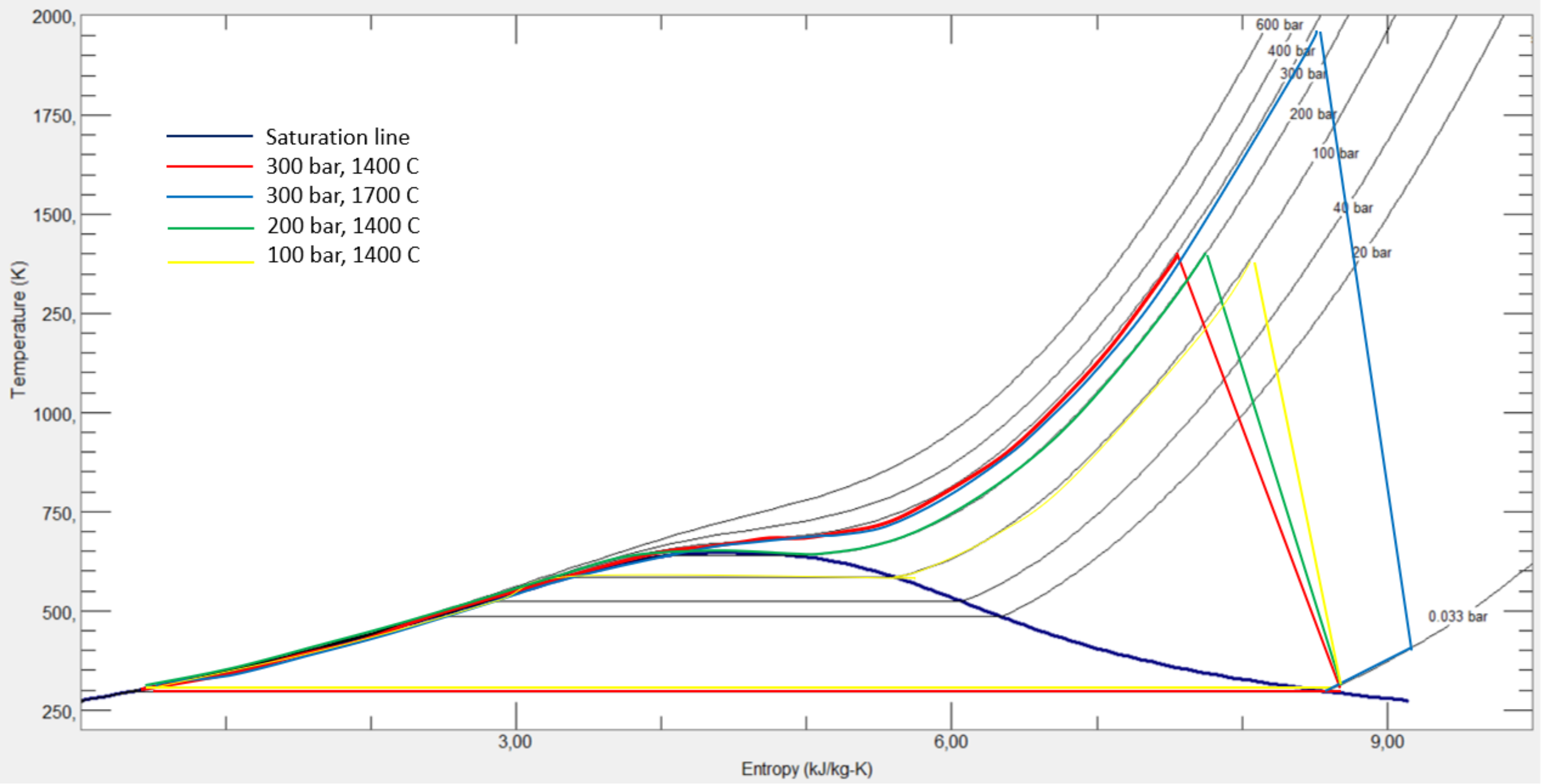


Figure 4.7: T-s-diagram for hydrogen power cycle showing how different cycle parameters affect the T-s-diagram and its area

As can be seen in Figure 4.7, the isobars gets closer to each other as the pressure increases. This explains the decreasing slope of the curve in the sensitivity analysis of the pressure, Figure 4.5. One can see in Figure 4.7 that the area increases as the pressure is increased from 100 bar (yellow curve) to 200 bar (green curve), which gave an increase of efficiency of around 2%-units. Although, one can clearly see that the area increase from 100 bar to 200 bar (yellow curve to green curve) is bigger than the area increase from 200 to 300 bar (green curve to red curve), since the pressure lines converges when pressure increases. This explains the decreasing slope in Figure 4.5. One can conclude that the most important pressure increase to achieve a high efficiency is to increase up to 300-350 bar. After that the pressure lines converges and the increase of efficiency in %-units decreases.

The increase in efficiency when mixed temperature is increased, Figure 4.6, is almost linear. As can be seen in Figure 4.7, the area will increase linearly when the temperature is increased as the constant pressure lines are linear in relation to temperature (holds for temperatures above 750°C), this can especially be seen in the transition from the red curve to the blue curve. To achieve a high efficiency cycle, there are no specific temperature that one really want to reach to increase the efficiency as there is for pressure.

When increasing the mixed turbine inlet temperature, for instance to 1700°C which can be seen in Figure 4.7 in the blue curve, the outlet of the last turbine is superheated when entering the condenser. This can be concluded since the vertical line does not end up on the saturation line but to the right. Super-heated steam is thus entering the condenser, instead of saturated steam, and some energy loss occurs. The energy loss is equal to the temperature difference between saturated steam and the super-heated steam at the condenser pressure times the mass flow going through the condenser.

The temperature of the first extraction versus efficiency can not be explained in the T-s-diagram and the area under the curve. But, as the temperature of the extraction increases, the pressure of the extraction increases as well. This increases the temperature of the returning feedwater which increases the efficiency of the whole cycle as less energy in the combustion chambers is needed to heat the returning feed water, thus less fuel is needed in the combustion chamber. As an example, when the turbine extraction temperature are increased from 650 to 750°C, the returning feedwater is increased by 38°C.

To conclude, to achieve the highest possible efficiency for a power cycle, the highest possible pressure and temperature should be used. Although, one always needs to have material limits in consideration. It can also be concluded that the modelled hydrogen cycle needs increased pressure and temperatures to reach the same efficiencies as other hydrogen cycles. It is possible to get the same or similar efficiency as the Graz cycle by Sanz et al, the new Graz cycle and the Toshiba cycle but to reach efficiency close to Westighouse cycle and MNRC, additional combustion chambers is probably needed.

4.2 Liquefaction process

To expand the system boundary of the model, liquefaction and pumping of hydrogen and oxygen are included. This, of course lower the efficiency of the cycle but gives a more realistic view of. The idea of liquefy hydrogen and oxygen comes from the idea that pumping up to 300 bar is less energy intense than compressing to the same pressure. Also, hydrogen is more easily stored as liquid than gas.

The liquefaction processes of hydrogen and oxygen are modelled as the Claude cycle. Hydrogen and oxygen leaves the liquefaction process in liquid phase at atmospheric pressure. After, pumping of oxygen and hydrogen is needed to increase the pressure to 300 bar which is the pressure in the combustion chamber. When liquefaction and pumping is included the model does thus not assume that hydrogen and oxygen are supplied "freely". After pumping the liquid substances, water at ambient pressure and temperature is used to heat the liquid to 10°C. Hydrogen and oxygen enters the liquefaction processes at 40 bar, 65°C which is the parameters at the outlet of the PEM electrolyser.

The Claude cycles are modelled with 3 compressors including inter-cooling by water at ambient temperature and pressure in between each compressor. The pressure increase in each compressor is divided to achieve equal work in all compressors. Hydrogen and oxygen are added to the liquefaction cycle between the compressors where the pressure is the closes to but still below 40 bar. Schematic figure of the liquefaction process can be seen in Figure 4.8 where both steam parameters for hydrogen and oxygen is presented in blue respectively green.

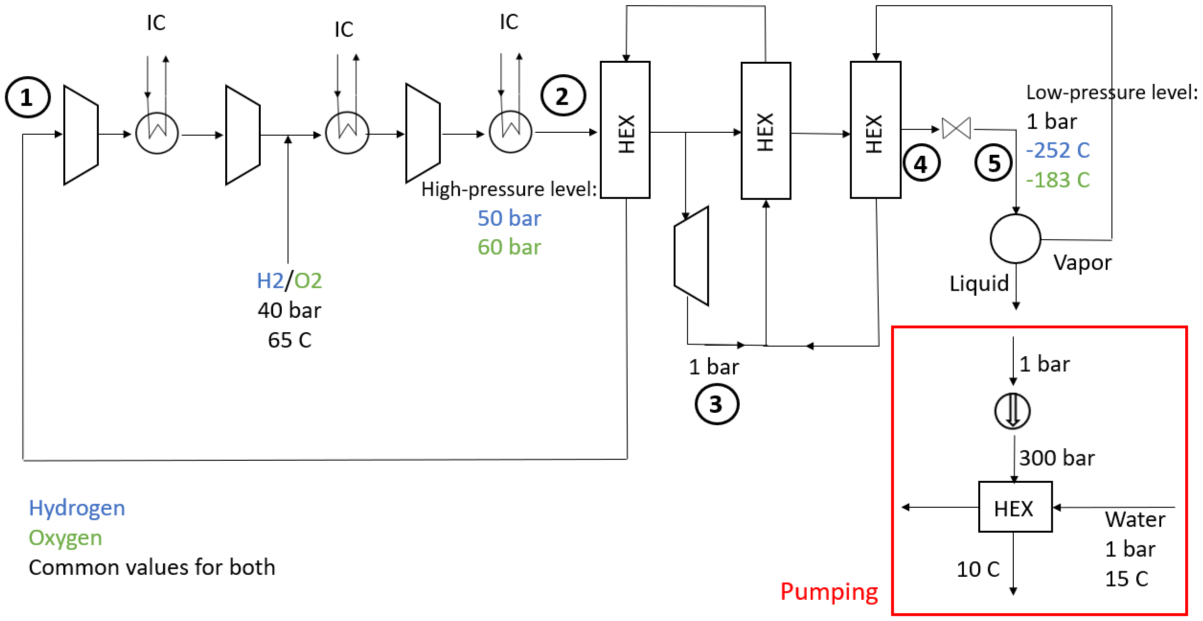


Figure 4.8: Claude liquefaction process for oxygen and hydrogen. Values in blue represents hydrogen and green oxygen. 1 is returning vapor at 1 bar, 2 is the substance at its high-pressure level, 3 is expanded gas to 1 bar, 4 is cooled substance but still in gas phase, 5 is expanded and condensed substance at 1 bar and its boiling point.

4.2.1 Oxygen liquefaction and pumping

A detailed figure of the oxygen liquefaction process and pumping is found in Appendix, Figure A.2. The gas is compressed to 60 bar. It resulted in an energy consumption of 1.13 MJ/kg LO_2 without pumping. Including the pump work needed to increase the pressure to 300 bar, it consumes 1.67 MJ/kg LO_2. The liquefaction process is presented in a T-s-diagram in Figure 4.9. The energy consumption of oxygen is around 1.8 MJ/kg LO_2 industrially so a decrease is found here which is partly thanks to the fact that oxygen is delivered to the liquefaction process at 40 bar, 65°C from the PEM electrolysis in this model instead of at atmospheric pressure and temperature. From modelling it can also be concluded that the pinch point used in the model highly affects the energy consumption. Values of pinch point have been hard to find in literature and can thus be another explanation to the difference in energy consumption.

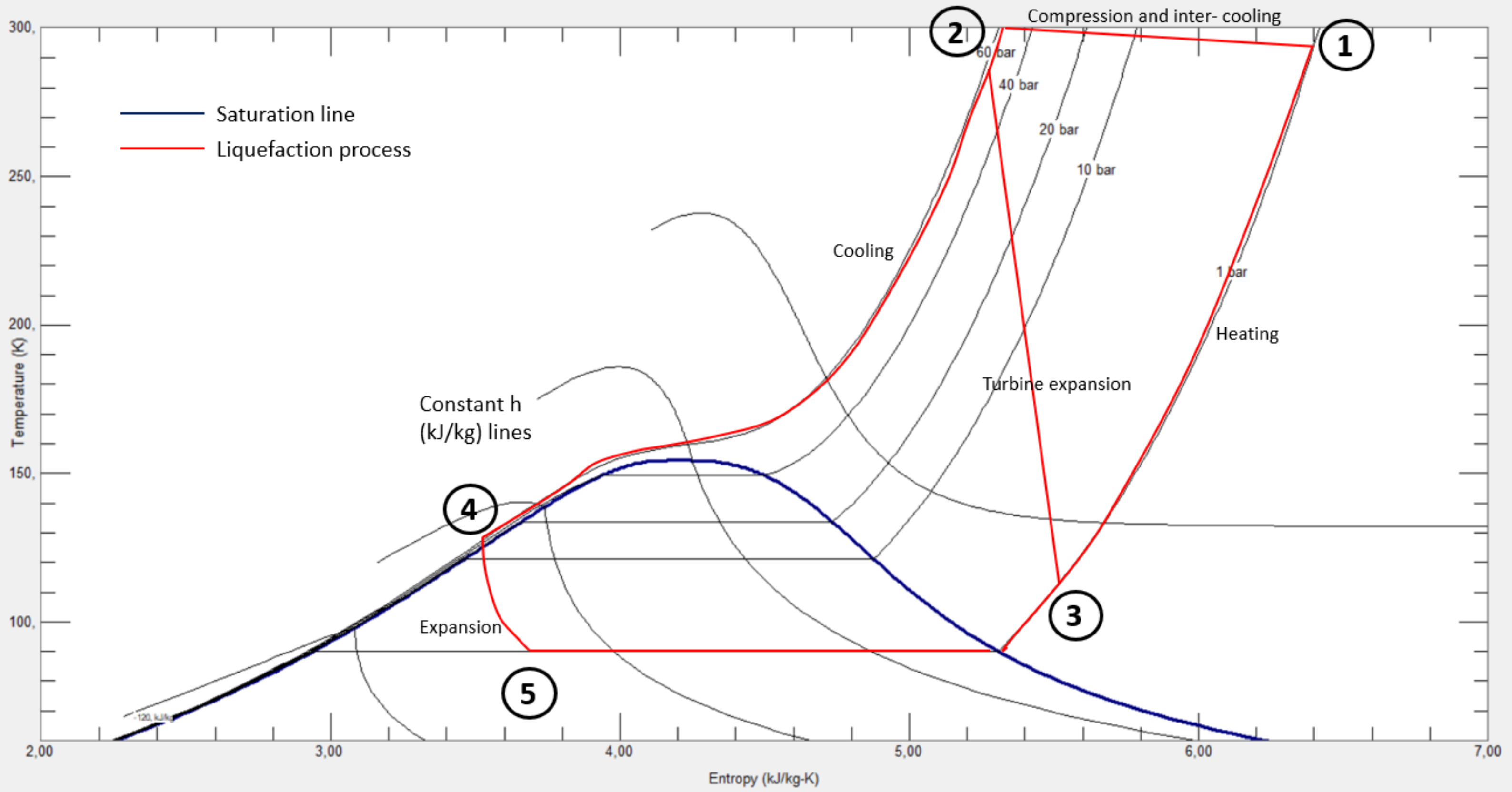


Figure 4.9: Liquefaction process for oxygen in T-s-diagram. 1 to 2: compression with inter-cooling, 2 to 3: expansion in turbine, 2 to 4: cooling in heat exchangers, 4 to 5: expansion in valve, 5 to 3 to 1: heating in heat exchangers.

4.2.2 Hydrogen liquefaction and pumping

The more detailed schematic figure of hydrogen liquefaction and pumping is found in Appendix, Figure A.3. Hydrogen is compressed to 50 bar which resulted in an energy consumption of 35.84 MJ/kg LH_2 without pump work and 36.37 MJ/kg LH_2 with pump work included. The liquefaction process is presented in a T-s-diagram in Figure 4.10. Comparing this to industrial liquefaction of hydrogen, which is around 40 MJ/kg LH_2, the decrease in energy consumption can be explained by hydrogen entering the process at 40 bar instead of 1 bar and the pinch point used.

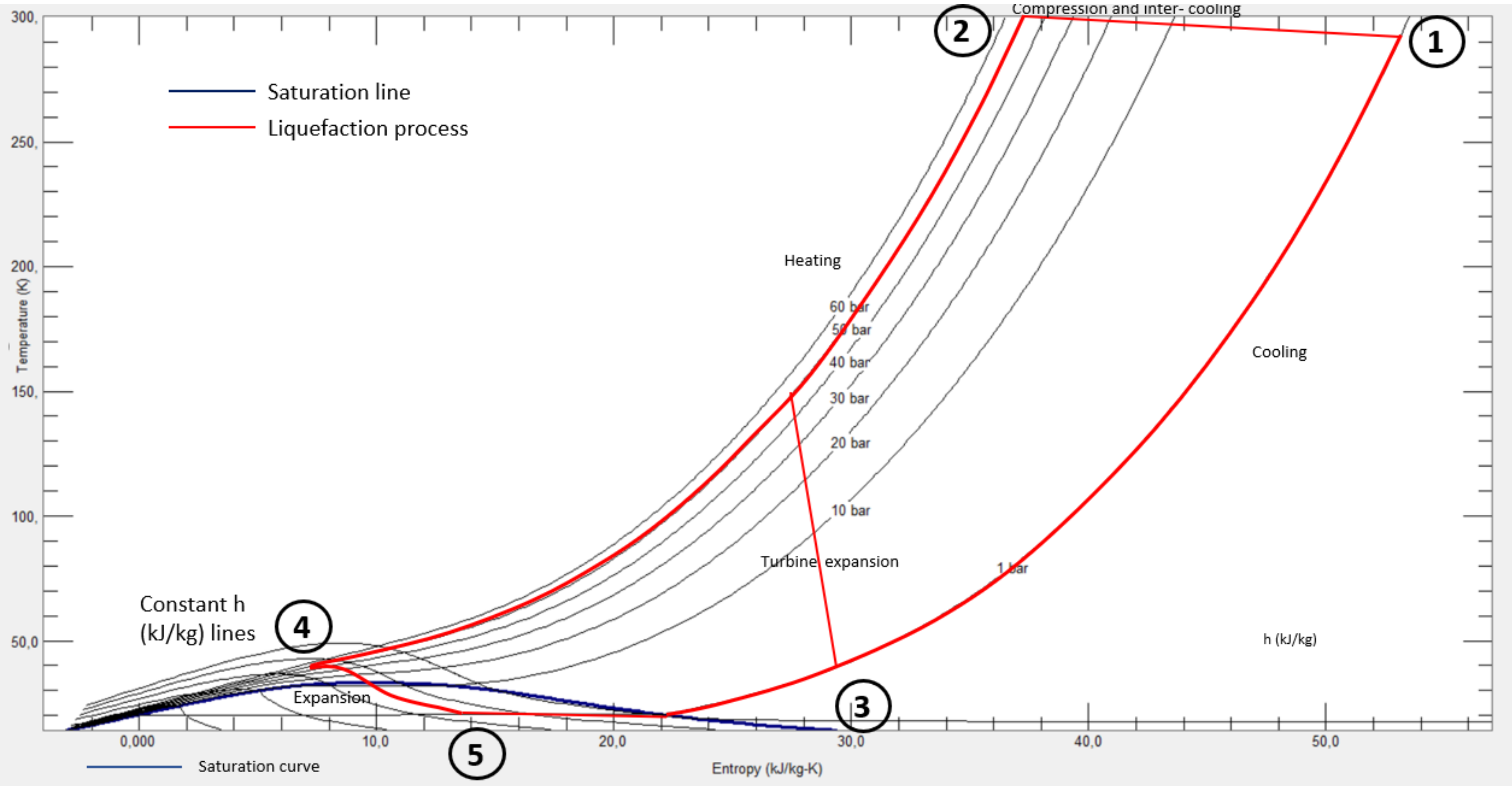


Figure 4.10: T-s-diagram of hydrogen liquefaction modelled as Claude cycle. 1 to 2: compression with inter-cooling, 2 to 3: expansion in turbine, 2 to 4: cooling in heat exchangers, 4 to 5: expansion in valve, 5 to 3 to 1: heating in heat exchangers

4.2.3 Summation of liquefaction processes

From modelling the liquefaction processes, values of pinch points used in previous literature have been hard to find. The chosen pinch point for the liquefaction processes is in this study 10°C. That gave energy consumption relatively similar to literature.

Some critical parameters have been concluded that can be varied to optimise the energy consumption of the liquefaction processes:

- The mass fraction extracted to the turbine.
- The fraction of liquid versus vapor entering the separation unit.
- The high pressure level
- The low pressure level

They all depend on each other and for each high- respectively low pressure level simulated, there are an optimum fraction extracted to the turbine and an optimum fraction liquefied. The fraction of liquid and vapor entering the separation unit is decided by the enthalpy at the expansion outlet. The optimisation work for the liquefaction processes has not been done in this project but could definitely be done to decrease the energy consumption.

4.2.4 Overall efficiency for hydrogen power cycle including liquefaction and pumping

Adding the energy consumption needed for liquefaction and pump work for oxygen and hydrogen to the hydrogen power cycle lowered the efficiency to 31.1%. Hydrogen liquefaction and pumping decreased the efficiency with 29.27%-units and oxygen liquefaction and pump work 7.22%-units. It can be concluded that liquefaction of hydrogen is extremely energy intensive and lower the efficiency of the hydrogen cycle a lot. If this efficiency of 31.1% would be combined with the energy losses at electrolysis when hydrogen is produced and losses during storage, the round-trip-efficient from electricity-hydrogen-electricity will be decreases even further. The liquefaction of hydrogen is one of the breakpoints for the hydrogen society to work. This gives an advantaged for the Graz cycle where hydrogen and oxygen output of the electrolysis at 40 bar could be inserted directly to the power cycle. Although, the hydrogen storage needs to be at gaseous phase in this case. Important to notice is that additional liquefaction and pumping or compression would be needed for the Toshiba, Westinghouse, MNRC and Graz cycles as well if liquid storage wanted.

4.3 Comparison between Graz cycle models

The modelled Graz cycle can be seen in Appendix, in Figure A.4.
After modeling the Graz cycle in IPSEpro one difference could be concluded. The value used for ΔH_{vap} of water, which is used in the energy balance of the combustion chamber to subtract the energy not received from condensing the water to the reference temperature, explained in Section 2.1.3. By calculating backwards it could be conclude that Sanz et al. uses an enthalpy of vaporization of approximately 2350 kJ/kg. While 2441 kJ/kg is the value given by REFPROP. The value of 2441 kJ/kg can be validated from "EngineeringToolbox" [27] and NIST [28]. It is clear that Sanz et al. uses the same reference state as in this project since the LHV is the same, 1.2 MJ/kg, which depends solely on the reference state. One explanation to the difference in ΔH_{vap} can be that one reference states have been used for ΔH_{vap} and LHV and another state as the reference state referred to by subtraction in the energy balance ($h°$). The problem with different reference states in the same energy balance has been discovered at other occasion during literature study in this thesis.

When using 2350 kJ/kg (same as Sanz et al.) as enthalpy of vaporisation in the new model of the Graz cycle, the efficiency became 68.7%. This is 0.27%-units from the efficiency modelled by Sanz et al. Additionally, all components gives the same results. This was a proof for the modelling of the hydrogen cycle that the method and components used was correct and that the results could be compared with the results from the Graz cycle by Sanz et al.

When using 2441 kJ/kg instead of 2350 kJ/kg as enthalpy of vaporisation, the energy removed from lost energy by evaporation is increased and the amount of recirculated hot water in the cycle needs to be increased to reach the same given outlet temperature of 1500°C. This results in an increased mass flow through the entire cycle and 2.3 MW extra turbine energy can be achieved. This increases the efficiency from 68.43% to 69.22% in the model of the Graz cycle.

5

Discussion

5.1 Hydrogen power cycles

Since hydrogen is produced when the amount of electricity is high and converted back to electricity when the amount is low, a part-time power cycle is a likely configuration of suchc a cycle. For a power cycle to be adapted to such conditions, for instance it needs fast start and stop time. There are possibilities that these cycles need to be started multiple times a day when the energy level on the net is low. This can be a challenging configuration but possibly a necessary one. If hydrogen fuelled power cycles like this serves more as a peak-load energy source than base-load, which is a likely scenario, the cycle might get away with the low efficiency of 31%. This is because peak-load units are in general units with low investment cost, high operational costs and not the most efficient plants. The cost of the cycle has not been studied in this thesis and is thus at the moment not known.

The modelled power cycle gives a good first evaluation of a model with this configuration. Further investigations are now needed to do more in-depth calculations to calculate the size of turbine blades etc. This can help motivate what temperatures and pressures are possible for the cycle. To improve the cycle further it would be interesting to look at various options to increase the temperature of the returning feed-water. In one configuration, mentioned in the introduction by Plazer et al. [9], a fuel cell is added to the cycle which is placed prior to the combustion chamber and the temperature of the returning feedwater is increased significantly, this is almost working as a second combustion chamber. There are possibly other configurations that can increase the temperature and thus also the efficiency. Because as could be seen in the sensitivity analysis of the extraction temperature, the efficiency of the cycle increases significantly as the temperature of the returning feedwater increases. If the power cycle is placed close to an industry, perhaps some hot waste stream could be utilized to heat the feed water further, or combustion of waste material prior to the ”real” combustion chamber could be an idea. It would also be interesting to add more combustion chambers to the model to see how that affects the efficiency. Since both Toshiba, Westinghouse and MNRC cycles have multiple combustion chambers and they all have higher efficiencies than this cycle. It would be of value to the report to add the same number of combustion chambers as used in these models and compare the efficiencies to see if the increase in efficiency is thanks to the number of combustion chambers or if there are other explanations.

The hydrogen fuelled power cycles studied in this thesis will face different problems compared to other high temperature cycles. For instance high temperature N_2 is not a factor since there is no presence of nitrogen in the combustion and thus no NOx will be formed. The same holds for SOx as there is no sulfides in the fuel. Furthermore, if true stoichiometric combustion can be achieved, oxidation of turbine blades should not be a problem since no oxygen is present in the turbines. It is thus very important to have the right amount of oxygen surplus in the combustion chamber to ensure full combustion but no superfluous oxygen after combustion. Since cycles like these are not present at the moment, the surplus of oxygen needed for full combustion is at the moment not known. Sanz et al. [2] states that a surplus of 3% should results in full combustion.

5.2 Sensitivity analysis

During sensitivity analysis, some high temperatures and pressures was investigated. State of the art pressure and temperatures are the same at the so called ”default paramters”, namely 300 bar and 650°C. The exact maximum possible mixed temperature has not been found in literature. Even though higher temperatures and pressure are unreasonable at the moment, it is interesting to investigate how it affects the efficiency. If materials used in combustion chambers and turbines are developed further, a turbine extraction temperature of 800°C and pressures at 500 bar might not be unreasonable. And as could be shown in this project, the efficiency increases pretty much at these higher steam parameters. Important to have in mind when increasing the pressure is that also compression work will increase. What the highest possible turbine inlet temperature is at the moment not clear, 1700°C as used in the Toshiba, Westinghouse and MNRC cycle might not be unreasonable, but as it not clear if that is a reasonable temperature a lower temperature was chosen as the default temperature in this project. The effect of higher turbine inlet temperatures can although be seen in the sensitivity analysis.

5.3 Liquefaction

The liquefaction of hydrogen is the major drawback with the configuration of this cycle. Since it reduces the efficiency of the cycle from 67.7% to 31.1%, some improvements for this part of the cycle would be of value. The idea of liquefying hydrogen and oxygen is to be able to pump the substances up to 300 bar instead of using a compressor. This reduces the energy consumption of the compression itself, but as a result liquefaction is needed. On the other hand, it was very clear in the results of the liquefaction processes that pumping up to 300 bar was only a small share of the energy consumption for liquefaction, around 0.5 MJ/kg LH_2 for hydrogen. It would be interesting to compare gaseous compression of hydrogen and oxygen with liquefaction and pumping to see if some improvement could be done. Since compression of gaseous hydrogen and oxygen would require storage in gas phase it would also be of value to investigate the pros and cons of hydrogen storage in gaseous phase versus liquid phase. To be able to store and convert hydrogen back to energy effectively is definitely a deal-breaker for the hydrogen society. It needs to be done in a very efficient way without big losses to be competitive with fossil fuels. As additional investigation to the power cycle it would therefore be of great value to calculate the round-trip efficiency, namely from electricity to hydrogen to electricity. The efficiency of hydrogen production by electrolysis and the losses occurring during storage could be added to the efficiency of the cycle including liquefaction to achieve the round-trip-efficiency. Although, it is clear that it in this case will be lower than 31.1%, meaning that most of the abundance energy at the energy net will be lost during storage and conversion back to electricity.

It would also have been interesting to optimise the liquefaction process. Since hydrogen and oxygen can be added to the power cycle at any pressure, perhaps the low-pressure level could have been much higher (as long as it is under the critical pressure for each substance) and the liquefaction energy consumption would have been lower. When the low-pressure level is increased the boiling point of the substance is increased as well and the liquefaction process is easier to achieve. This would mean that liquid hydrogen would be stored at other pressures than 1 bar, which possibly is energy consuming as pressurised tanks would be needed.

5.4 Ethical and environmental aspects

As the aim of this project is to decrease pollution and negative climate effects, there are mainly positive ecologic and ethical effects of such a cycle. The most positive effect of it is that it enables storage of energy and that renewable energy sources are favoured by this. But our energy consumption on this planet will always have negative effects on the climate though and the effects of the transition towards more sustainable and renewable energy sources will also be seen in the future. We already know that wind and hydro power for instance might have a negative effect for some wildlife. A dilemma in the energy transition may be to weight pros and cons of these renewable energy sources against the damage it might make in the nature.

There is constantly the discussion about the fact that hydrogen is explosive. But power production from hydrogen can be defended by the fact that power production sites are normally enclosed area and not in public place near crowds. In the same way as refineries are strictly enclosed with rules that no electronics are allowed inside the refinery area.

6

Conclusion

The aim of the project has been to investigate and quantify the efficiency of a modelled hydrogen fuelled power cycle similar to a Rankine cycle with pre-heat. Also to analyse its critical parameters and how the cycle can be optimised. The results have further been compared with other hydrogen power cycles found in literature. The result of this study can be used for further, more in-depth investigations covering component design for instance, to optimise the cycle further.

A hydrogen power cycle with relatively high efficiency, 67.6%, has been modelled and investigated. The cycle has lower efficiency than all four other cycles included in this study, the Graz, Toshiba, Westinghouse and MNRC cycles. The critical parameters differs slightly between the studied cycles but are relatively similar which makes the different cycles comparable.

Sensitivity analysis of the cycle shows that increased temperatures and pressure increases the efficiency and can reach the same efficiency as the Graz cycle and the Toshiba cycle. The mixed turbine inlet temperature needs to be increased to around 1550°C and pressure to around 500 bar (which is unrealistic with todays material). The efficiency can not be as high as for the Toshiba and MNRC cycle. Sensitivity analysis also showed that increased temperature at the turbine inlet is linearly against the efficiency while pressure at the turbine inlet has a decreasing slope as pressure increases. From this it can be concluded that if a pressure between 300-350 bar can be achieved, the efficiency increases a lot. At higher pressures the increase in %-units are not that big. Temperature at the first turbine extraction is also linear against efficiency.

Even though the cycle has lower efficiency than the other cycles, the cycle has advantages in its simple design, only one combustion chamber is used and no expensive HRSG is used which occupies big space. It is commonly known that additional combustion chambers increases the efficiency of a power cycle. Although, it also increases the number of units and thus the cost of the cycle. As further investigation of the cycle it would be interesting to investigate the efficiency of the cycle if two or three combustion chambers where used. Perhaps it could be competitive with the Westinghouse and the MNRC cycle.

The big drawback with the cycle is when liquefaction of hydrogen and oxygen is included. The process of liquefying hydrogen is extremely energy consuming and reduce the efficiency to 31.1%. Although, this energy consumption needs to be added to all the other hydrogen cycles as well if liquid hydrogen storage is desirable. Westinghouse, Toshiba and MNRC cycles needs to liquefy and pressurise oxygen and hydrogen as well before entering the cycles as they have pressure between 73 and 250 bar in the combustion chamber. From this point of view the Graz cycle has a clear advantage as the pressure is only 40 bar in its combustion chamber. If one would chose gaseous hydrogen storage in combination with the Graz cycle, no compression or liquefaction of hydrogen and oxygen are need (since the outlet of the electrolysis is 40 bar). Liquid hydrogen storage can be a big advantages since it occupies is less space than gas.

To conclude the project, a high efficient hydrogen fuelled power cycles has been modelled and investigated. It does not reach the same efficiency as other hydrogen power cycle concepts but has advantages in its simple design, only one combustion chamber and storage of hydrogen in liquid phase. For further studies it would be interesting to optimise the liquefaction process and also include the electrolysis and storage energy consumption to get the round-trip efficiency of electricity to hydrogen to electricity.

Bibliography

[1] Mehrpooya. M, Karimi, M. *Hydrogen production using solid oxide electrolyzer integrated with linear Fresnel collector, Rankine cycle and thermochemical energy storage tank.* Energy Conversion and Management, Volume 224, 2020, https://doi.org/10.1016/j.enconman.2020.113359.

[2] Sanz. W, Braun, M. Jericha, H. Platzer, M. F. *Adapting the zero-emission Graz Cycle for hydrogen combustion and investigation of its part load behavior.* International Journal of Hydrogen Energy, Volume 44, Issue 21, Page 11161-11167, 2019, https://doi.org/10.1016/j.ijhydene.

[3] *Fundamentals of Renewable Energy Processes.* `https://eds.b.ebscohost.com/eds/results?vid=0&sid=88f4c004-b1b0-4e97-a356-b5ecdb5538b8%40pdc-v-sessmgr01&bquery=Fundamentals%2Bof%2BRenewable%2BEnergy%2BProcesses&bdata=JmNsaTA9RlQxJmNsdjA9WSZ0eXBlPTAmc2VhcmNoTW9kZT1BbmQmc2l0ZT1lZHMtbGl2ZSZzY29wZT1`

[4] GE Steam Power, *Hydrogen Storage-Hydrogen and Fuel Cell Technologies Office.* [Online] Available: `https://www.energy.gov/eere/fuelcells/hydrogen-storage`. Accessed: 2021-05-07.

[5] Sunden, B. *Hydrogen, Batteries and Fuel Cells: Chapter 3 - Hydrogen.* Academic Press, 2019.

[6] Smidth. O, et.al *Future cost and performance of water electrolysis: An expert elicitation study.* International Journal of Hydrogen Energy, Volume 42, Issue 21, Page 30470-30492, 2017, https://doi.org/10.1016/j.ijhydene.2017.10.045.

[7] Weiliang. W, Shusheng. Z, Ce. Z, *Study of the Hydrogen-Steam Turbine Composite Cycle.* Procedia CIRP, Volume 26, Page 735–739 , 2015, https://doi.org/10.1016/j.procir.2015.01.014.

[8] Milewski. J, *Hydrogen utilization by steam turbine cycles.* Journal of Power Technologies, Volume 95, Issue 4, Page 258–264, 2015.

[9] Platzer, M. F, Sanz. W, Jericha, H. *Renewable power via energy ship and graz cycle.*15th International Symposium on Transport Phenomena and Dynamics of Rotating Machiner, ISROMAC-15, 2014.

[10] Windmeier. C, Barron. R. F, *Ullmann's Encyclopedia of Industrial Chemistry: Cryogenic Technology.* Wiley-VCH Verlag GmbH & Co. KGaA.

[11] Cengel, Y. A. Boles. M. A. *Thermodynamics: An engineering approach.* 4th Edition, New York, USA: McGraw-Hill, 2002.

[12] Engineers Edge, *Temperature-entropy (T-s) Diagram- Thermodynamics.* 2021 [Online] Available: `https://www.engineersedge.com/thermodynamics/temp_enthalpy_th_diagram.htm`. Accessed: 2021-03-10.

[13] Kostyuk, A. Frolov. V. *Steam and Gas Turbines.* Russia: Mir Publishers, 1988.

[14] Atkins, P. Jones, L. Laverman, L. *Chemical Principles: The Quest for Insight.* 6th Edition, New York, USA: W.H. Freeman and Company, 2013.

[15] Normann, F *Energiteknik.* Gothenburg, Sweden, 2017.

[16] Marangio. F, Pagani. M, Santarelli. M, Calì. M, *Concept of a high pressure PEM electrolyser prototype.* International Journal of Hydrogen Energy, Volume 36, Issue 13, 2011, Pages 7807-7815, https://doi.org/10.1016/j.ijhydene.2011.01.091.

[17] Sharma R.G, *Superconductivity.* Springer Series in Materials Science, Volume 214, 2015, Pages 1-12, `https://doi.org/10.1007/978-3-319-13713-1_1`.

[18] Linde Kryotechnik AG, *Hydrogen liquefiers.* 2021 [Online] Available: `https://www.linde-kryotechnik.ch/en/references/hydrogen-liquefiers/`. Accessed: 2021-04-20.

[19] Matsuda. H, Nagami. M, *Study of large hydrogen liquefaction process.* [Online] Available: `hhttp://www.enaa.or.jp/WE-NET/ronbun/1997/e5/sanso1997.html`. Accessed:2021-05-18

[20] Decker. L, *The Latest Developments and Outlook for Hydrogen Liquefaction Technology.* 2013, DOI: 10.1063/1.4860858.

[21] Gardiner. M, *Energy requirements for hydrogen gas compression and liquefaction as related to vehicle storage needs.* DOE Hydrogen and Fuel Cells Program Record, `2009,https://www.hydrogen.energy.gov/pdfs/9013_energy_requirements_for_hydrogen_gas_compression.pdf`.

[22] Kawasaki *Technologies of hydrogen liquefaction, transport and storage — Paving the way to a hydrogen fueled future.* Kawasaki Technical Review No.176, 2016, https://global.kawasaki.com/en/corp/rd/magazine/176/pdf/n176en10.pdf.

[23] Malyshenko. S. P, Gryaznov. A. N, Filatov. N. I, *High-pressure H 2/O 2steam generators and their possible applications.* International Journal of Hydrogen Energy, Volume 29, Issue 6, Page 589-596, 2019, DOI: 10.1016/j.ijhydene.2003.08.004

[24] U.S. Department of Energy, *Ultra-Supercritical & Advanced Supercritical Technology.* [Online] Available: `https://www.ge.com/steam-power/coal-power-plant/usc-ausc`. Accessed: 2021-05-04.

[25] *Gas turbines — Acceptance tests*, ISO 2314:2009, International Organization for Standardization, 2009. Available: `https://www.iso.org/standard/42989.html`.

[26] Performance data for power cycles concluded from internal discussion with Siemens Energy AB, MGT (Medium Gas Turbines) Performance department, Sweden, 2021.

[27] Engineering ToolBox, *Fuel Gases Heating Values.* 2005 [Online] Available: `https://www.engineeringtoolbox.com/heating-values-fuel-gases-d_823.html`. Accessed: 2021-03-15.

[28] Wikipedia, *Water.* 2021 [Online] Available: `https://en.wikipedia.org/wiki/Water_(data_page)`. Accessed: 2021-05-11.

A

Appendix

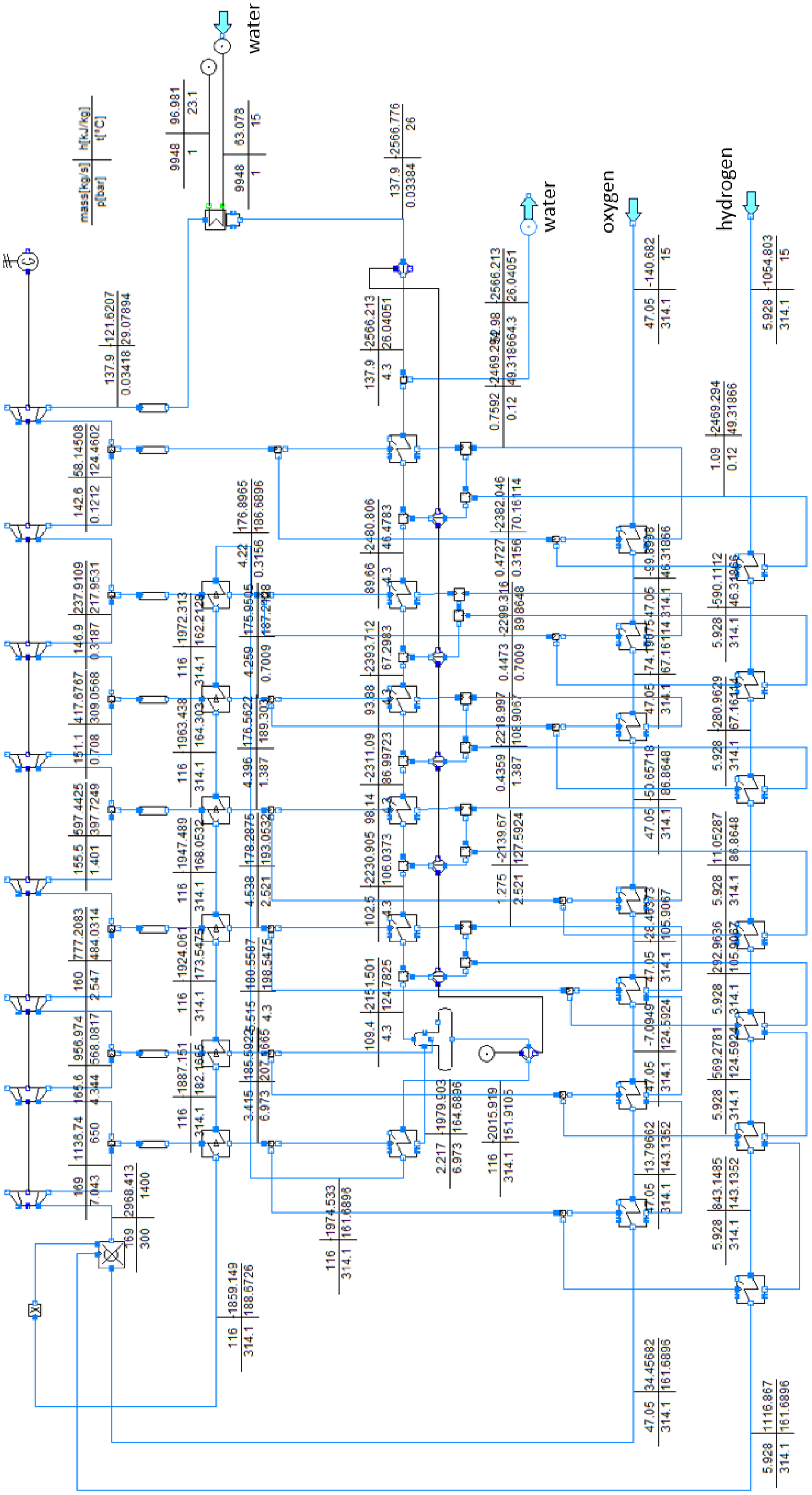


Figure A.1: Schematic figure of the modelled hydrogen power cycle in IPSEpro

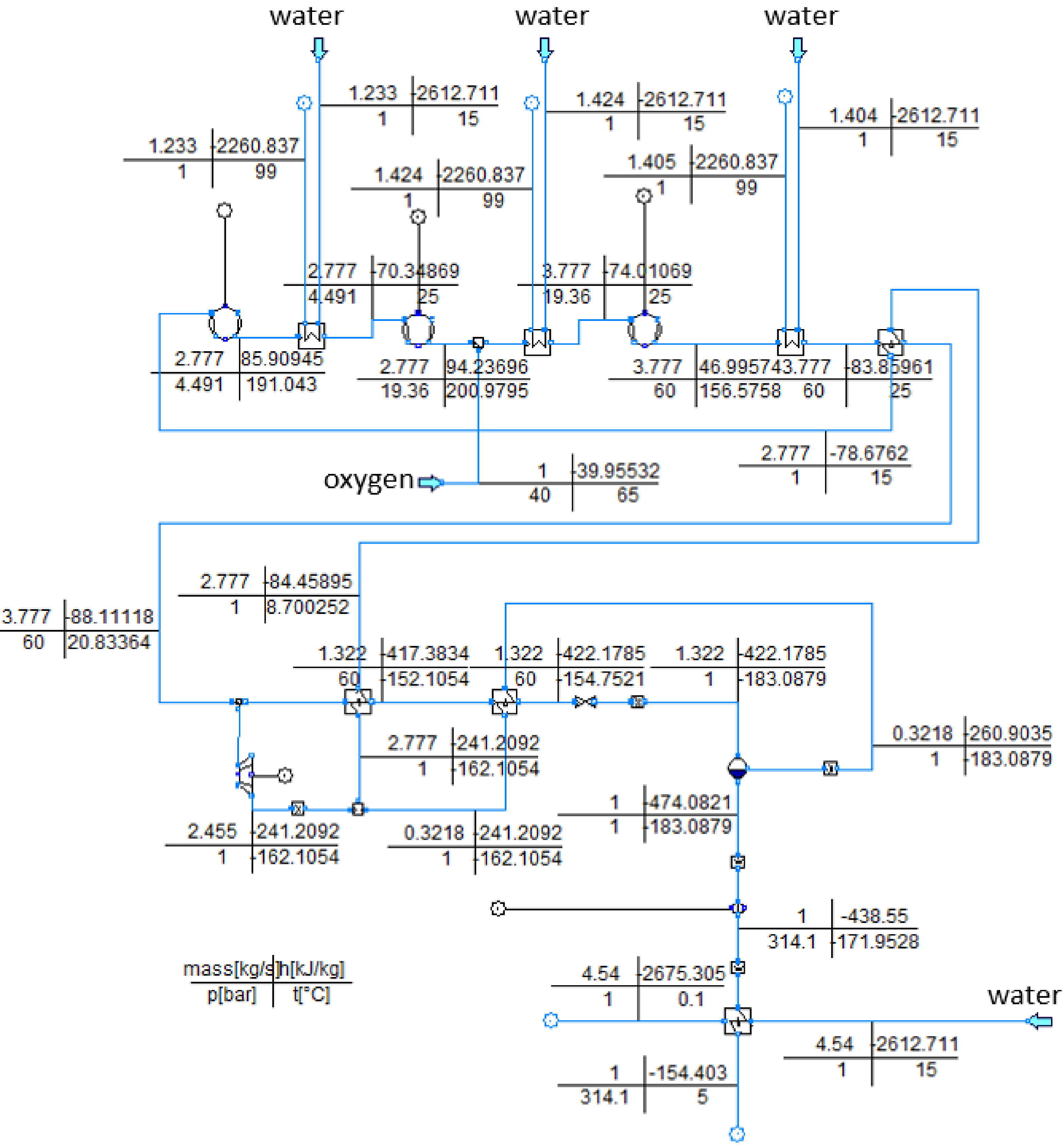


Figure A.2: Schematic figure of oxygen liquefaction process modelled as a Claude cycle in IPSEpro

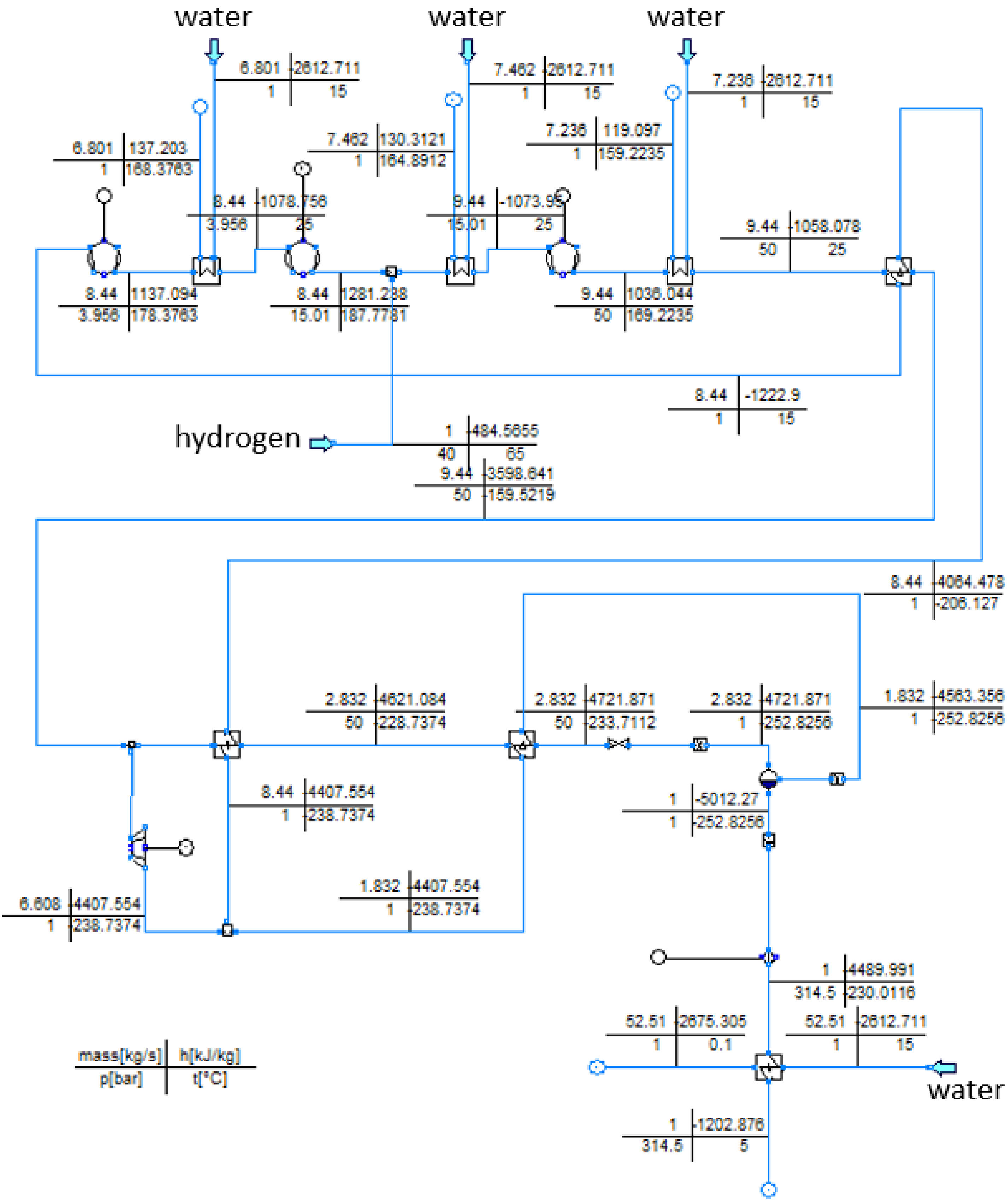


Figure A.3: Schematic figure of hydrogen liquefaction process modelled as a Claude cycle in IPSEpro

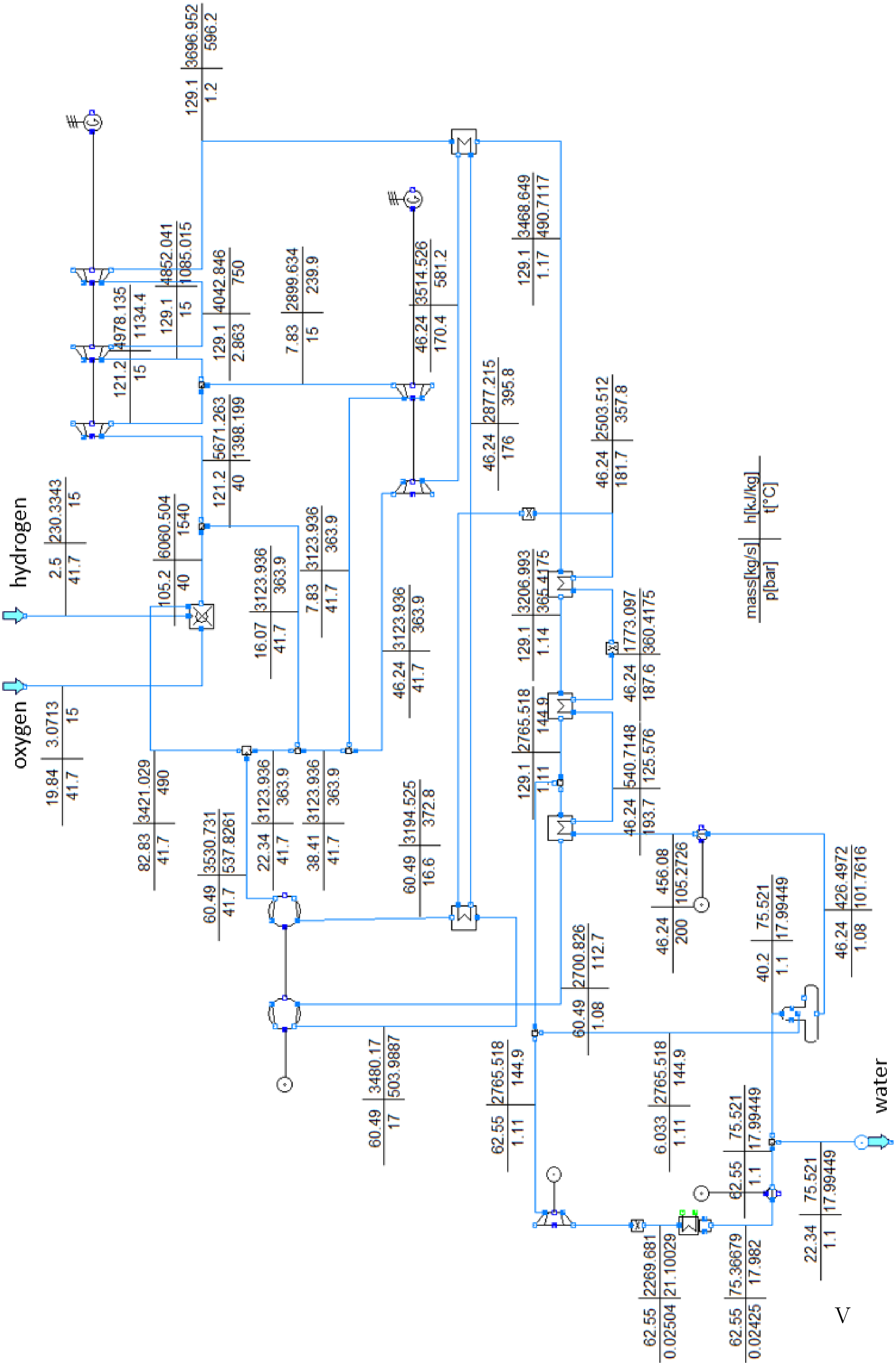


Figure A.4: Schematic figure of the modelled Graz cycle in IPSEpro

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