The Integration of Heat Pumps with Biomass Boiler Flue Gas for Generating Hot Water

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Abstract

Achieving the goal of net zero greenhouse gases by 2050 requires reducing demand and switching from fossil fuels to renewables. Process heat in New Zealand is a significant source of greenhouse gas emissions. Transitioning to biomass boilers could lower emissions and offer opportunities for waste heat recovery by integrating heat pumps with the flue gas.

This thesis evaluates heat pump integration for hot water production, supplying low temperature process heat. Using an industrial case study, a boiler model is developed and confirmed against industrial measurements. Flue gas composition is modelled to determine temperature-enthalpy profiles, and therefore the heat available for recovery and upgrade. Combustion is modelled for different biomass fuels and varying flue gas oxygen levels. The higher moisture fuels showed increased availability of heat for utilisation, which decreased as flue gas oxygen levels increased.

Three heat pump cycles are explored to maximise efficiency. Refrigerant selection is considered with respect to environmental factors and thermodynamic suitability. Using a basic cycle heat pump the three most promising refrigerants, ammonia, propane and cyclopropane, were identified and considered in more detail for the industrial case. Subcooling improved efficiency and should be maximised. An internal heat exchanger also improves efficiency and reliability and should be considered. Two zeotropic mixtures were investigated to take advantage of the temperature glide. The propane-pentane (50wt%-50wt%) blend matched the temperature profile more closely and provided increases in COP.

To make the most of the latent heat in the flue gas, the evaporator temperature must be much lower than the dew point. However, absorbing heat from low source temperatures results in lower COPs, but higher volumes of hot water generated.

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Finally thanks especially to my family. My parents, whose passings bookmarked my studies. I followed in Dad’s footsteps to become an engineer, with Mum’s “girls can do anything” mantra encouraging me. The unwavering support and encouragement of my sisters, and my children, also needs to be acknowledged.

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Nomenclature

Abbreviations

AHP Absorption Heat Pump

BFW Boiler Feedwater

COP Coefficient of Performance

DHX Direct Heat Exchanger

GHG Greenhouse Gas

GWP Global Warming Potential

HHV High Heating Value

HPS High Pressure Steam

IHX Internal Heat Exchanger

ODP Ozone Depleting Potential

PFD Process Flow Diagram

P&ID Process and Instrumentation Diagram

Temp Temperature

VHC Volumetric Heating Capacity

Symbols

P_A Pressure candidate 1 (Pa)

P_B Pressure candidate 2 (Pa)

c_p Specific heat capacity (kJ/kg·K)

h Specific enthalpy (kJ/kg)

h° Molar specific enthalpy (kJ/kmol)

n Combustion coefficient

m Mass (kg)

$\dot{m}$ Mass flow (kg/s)

M Molecular weight (g/mol)

P Pressure (Pa)

P^* Partial pressure (Pa)

Q Vapour quality by mass

$\dot{Q}$ Heat flow (kW)

T Temperature (°C or K)

VHC Volumetric heating capacity (kJ/m^3)

$\dot{W}$ Work flow (kW)

w Specific work consumption (kJ/kg$_{HW}$)

z Subscript in a combustion equation, number of atoms

$\%MC$ Moisture content, percentage

ΔT_{min} Minimum approach temperature (°C)

Subscripts/superscripts

a Actual/non-ideal

ave Average

BD Blowdown

$comp$ Compressor

$cond$ Condenser

$dry\ flue$ Dry flue gas (excluding water)

DHX Direct Heat Exchanger

$evap$ Evaporator

$flue$ Flue gas from the boiler

HW	Hot water
i	Individual component
IHX	Internal heat exchanger
$refr$	Refrigerant
s	Isentropic/ideal
sat	saturation
$valve$	Expansion valve

Greek

Δ	Change
η	Efficiency (%)
θ	moisture in fuel (%)

Units

bar	Bar
h	Hour
J	Joule
k	kilo
K	Degrees Kelvin
kg	Kilogram
m	kg/s
mol	Mole
MW_{th}	Megawatts (thermal)
Nm^3	Normal cubic meter
Pa	Pascal
s	Second
t	Metric tonne
W	Watt
$°C$	Degrees Celsius
%	Percent

Chapter 1
Introduction

1.1. Background

Extreme weather events, pollution, water scarcity and health issues are challenges that face us all as a result of global warming. In response to this, the New Zealand Government has made a commitment to reach net zero for long lived gases by 2050. Under the Climate Change Response (Zero Carbon) Amendment Act 2019 (the CCRA), a statutory target has been set to achieve net zero Greenhouse Gas (GHG) emissions (other than biogenic methane) by 2050 [1]. This reduction in emissions will require changes to the way that industry operates, which will include reduction in demand, economising and fuel switching from fossil fuels to renewable sources.

Almost 20% of New Zealand's overall energy use (104,348 TJ) is used for process heat (Figure 1-1). Process heat is defined as energy primarily used for warming spaces and industrial processes. Process heat contributes disproportionately to emissions as 56% is supplied by fossil fuels, mainly coal and natural gas [2]. In 2022, an estimated 10,099 TJ of energy was used for low-temperature heat [3]. Figure 1-2 shows low-temperature process heat, including hot water generation, is almost entirely generated using fossil fuels. Hot water generation is classified as low temperature heat in New Zealand. Reliance on fossil fuels must be reduced if the net zero target is to be met.

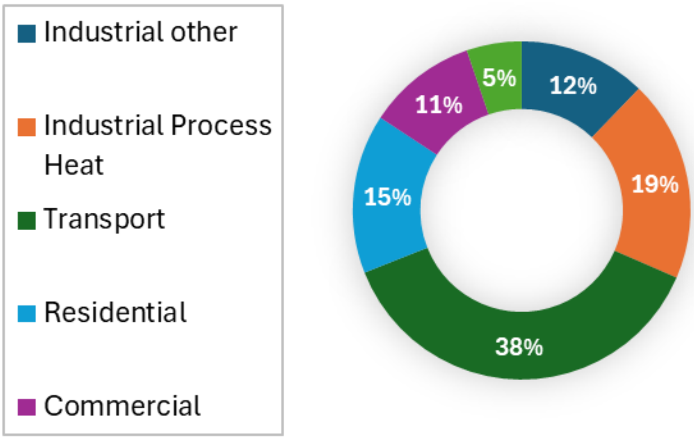


Figure 1-1 New Zealand end energy use [3]

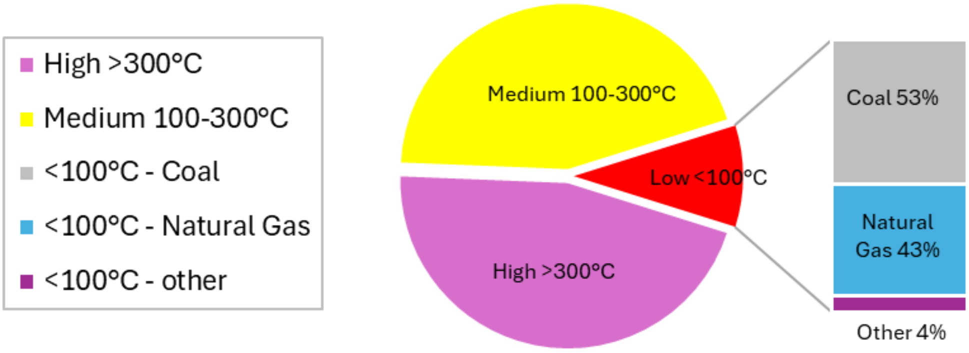


Figure 1-2 New Zealand process heat by temperature ranges [3]

With the transition to a low-carbon energy system, there is an increase in conversion to biomass boilers to provide process heat at all temperatures. Replacing fossil fuels with biomass will lead to lower GHG emissions. Biomass forms part of a relatively short carbon cycle (~20 years), the carbon released comes from the atmosphere and not from underground carbon deposits. Replanting will recapture the carbon as the new trees grow [4]. Conversion to biomass presents many technical challenges but also opportunities. There are several sources of biomass fuel, wood chips, pellets and hog fuel are the most common for industrial use. The fuel used in biomass boilers is higher in moisture content than coal or natural gas and therefore the flue gas of a biomass boiler contains a larger percentage of water vapour. This could represent a useful source of low-grade heat (<150 °C), which would be ideal for producing hot water for process use and, as a result, would reduce the overall steam use for a site. In turn, the primary energy use would be reduced, allowing higher production rates for the same fuel use or a lowering of fuel costs.

Since 2001, at least 14 installations of biomass boilers, or conversions from coal or fossil fuels to biomass, have been carried out in New Zealand by Windsor Engineering [5]. Further to this, another 15 biomass installations or conversions are listed on the EECA website that have been partially funded under the Government Investment in Decarbonising Industry Fund (GIDI Fund) [6]. The flue gas from these biomass boilers represents a large source of waste heat. Energy has been expended to heat the moisture in the fuel during the combustion process. This waste heat exits with the flue gas and provides an opportunity for recovery and upgrade. It is preferable to use any waste heat directly initially. Then the next step is to recover and upgrade wasted heat, increasing the temperature of the waste stream from low, to higher temperature and therefore providing more useful energy. Heat pumps use electricity to upgrade heat very efficiently by absorbing heat from a source to heat a refrigerant and, following compression, supply heat to a sink.

As can be seen in the Figure 1-3, the dairy processing sector uses nearly 83% of the low-temperature process heat energy in New Zealand. A large proportion of this is hot water used for cleaning. The dairy industry has recently installed or converted a number of boilers to biomass [5], [6]. The combination of available flue gas waste heat and a large requirement for low temperature process heat represents an opportunity for investigation for integrating biomass boilers with heat pumps.

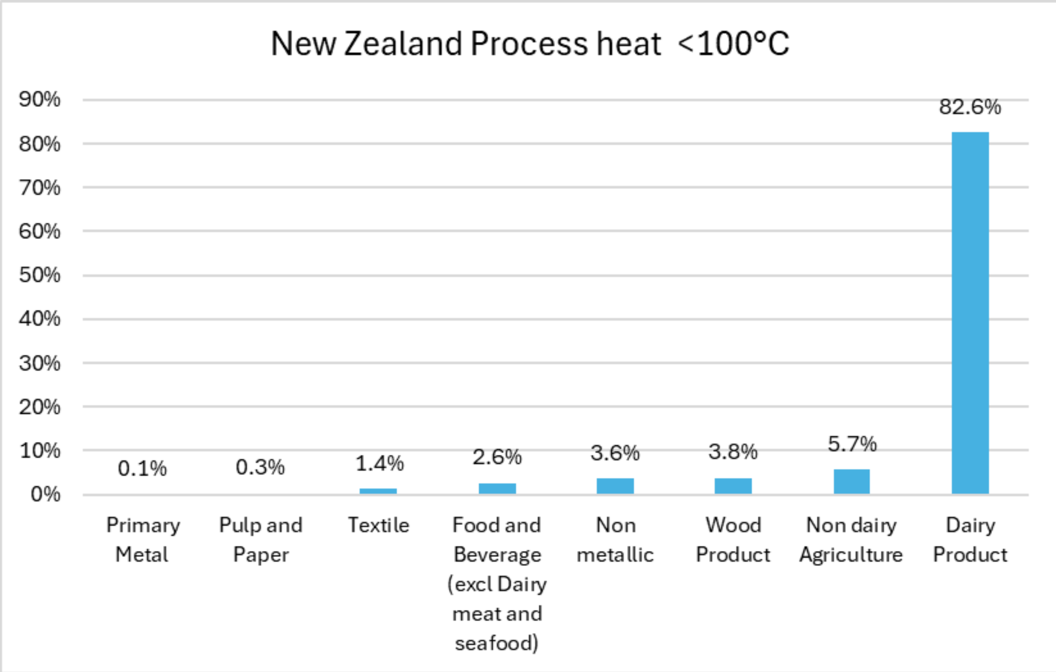


Figure 1-3 New Zealand process heat use below 100 °C by sector [3]

1.2. Thesis aim

The aim of this thesis is to evaluate the potential for installing an additional economiser (direct heat exchange) and integrating a vapour-compression heat pump to recover waste heat in the flue gas of a biomass boiler to produce hot water for an industrial site.

To achieve this aim, the combustion of biomass is modelled to evaluate the quality of the flue gas. Using the provided case study data, the biomass boiler is modelled to find the availability of heat for hot water generation and determine the amount of hot water that can be produced. Because the flue gas temperature can be higher than the required hot water temperature, producing hot water directly is first modelled. Once the heat that can be utilised directly, and therefore more efficiently at the higher temperature, has been exhausted, it will be used as a heat source for a heat pump. Three heat pump cycles are considered. A basic heat pump is modelled initially, to which subcooling is added to determine the benefit. The third cycle is the addition of an internal heat exchanger (IHX) to the basic heat pump model to discover any efficiency gains that can be made. Possible refrigerants, including binary mixtures of hydrocarbons, are examined to discover the best options to maximise heat utilisation.

The research uses an industrial case study of a biomass boiler installation in New Zealand to determine the opportunity for using a heat pump to upgrade waste heat for producing hot water. The duty of the boiler is approximately 16MW and the steam pressure is 34 bar. Hog fuel is the main fuel for the boiler, which has an average moisture level of 50%. The flue gas currently leaves the boiler's conventional economiser at approximately 130 °C. This is modelled in order to evaluate the potential for hot water generation utilising the flue gas as the source for a heat pump. No further identifying information about the site may be shared.

1.3. Thesis outline

This thesis is divided into seven chapters. Chapter 2 provides the literature review examining the research around integrating a heat pump with the flue gas of a biomass boiler. Different heat pump cycles that are appropriate for producing hot water including investigating different refrigerants and refrigerant mixtures for thermodynamic performance and environmental impact are reviewed. This provides the direction for modelling in Chapter 3 where the methods used for modelling combustion to determine the quality of the flue gas are described. Further to this, the methods for calculating the hot water generation using the heat pump cycles are explained. The different cycles and working fluids discovered in the literature review are modelled. Chapter 4 provides a description of the case study to be used for evaluation and also outlines the scenarios evaluated. The data from the case study is used in the model and the results from the combustion modelling are presented in Chapter 5 . Chapter 6 contains the heat pump analysis where the different cycles are compared. The refrigerant types are investigated thermodynamically, and the results are reported using performance metrics. Conclusions and future work are presented in the final Chapter 7 .

Chapter 2
Literature review

2.1. Introduction

The previous chapter outlined the need for decarbonising process heat in industry and identified one of the methods that might be used to facilitate this. This chapter will investigate the literature around biomass boilers and integrating a heat pump with the flue gas of a biomass boiler. Examination of existing research on the integration of a heat pump will be carried out and an overview of biomass boilers and modelling the flue gas will be investigated. Information on the basic heat pump cycle and enhancements in the form of subcooling and addition of an internal heat exchanger will be explained. Refrigerants are an important consideration for heat pump technology and will be reviewed in the final section.

2.2. Biomass boilers

Biomass boilers are a well-established technology. This work requires insight into the composition of the flue gas, so that the temperature-enthalpy profile of the gas can be modelled. The determining factors of the flue gas composition are the moisture level and chemical composition of biomass. This section focuses on the chemical composition of the biomass to be used in the analysis. The results are utilised to form the mass and energy balance to determine the temperature-enthalpy profile (method described in Chapter 3).

Biomass is organic material such as wood and wood waste or crops that can be used as an energy source. In New Zealand biomass fuel is generally referred to as wood fuel because wood is the primary source of the biomass, the most common being pellets, chips and hog fuel. Fuel pellets are produced by densification of wood to specific standards. Chips are produced from stem wood, process residues, untreated used wood and are cut to a consistent size. Hog fuel is often a mixture of different sources, wood processing residues, clean urban woody waste or forest derived. Hog fuel is generally made up of different size and shapes [12].

Combustion is the reaction where the energy of the fuel (biomass) is released. Oxygen in the air reacts with the fuel to produce CO_2 and H_2O. The fuel is approximated by the formula $C_{31}H_{45}O_{24.5}$ from an analysis of Pinus Radiata using the proximate and ultimate analysis [13].

Fuel combustion is approximated by:

$$C_{31}H_{45}O_{24.5} + 30\ O_2 \rightarrow\ 31\ CO_2\ +\ 22.5\ H_2O \qquad (1)$$

Matching the fuel to air ratio is important for maintaining complete combustion. Excess air ensures there is no unburnt fuel, soot or carbon monoxide in the flue gas to be emitted to atmosphere. However, there is a balance; if excess air is too high, this will reduce the efficiency of the boiler. Typically, an oxygen level in the flue gas of 4-8% will ensure complete combustion without overly penalising the boiler efficiency.

The thermal energy in the fuel released during combustion is termed the heating value. There are two forms of this, the higher heating value (HHV) and the lower heating value (LHV). The HHV is the amount of energy released including the latent heat of vaporisation of the water. The lower heating value is found by subtracting this heat of vaporisation of the water generated during combustion from the higher heating value. As discussed in Chapter 1 this water vapour represents a large amount of energy that can be recovered.

HHV_{wet} for Pinus radiata are calculated:

$$HHV_{wet} =\ HHV_{dry} \times (1 - \theta) \qquad (2)$$

Where HHV_{dry} = 20.2MJ/kg and θ is the % moisture in the fuel [14].

2.3. Boilers with heat pumps using the flue gas

Five studies have identified the increased energy efficiency to be gained from integrating a heat pump into boilers using a variety of fuels. A study by Chen et al. [7] compares the integration of a heat pump into the flue gas with the return flow of water for district heating. Chen concluded that while either option improved the efficiency by 12.6%, integration into the flue gas offered the more cost-effective solution. Gorinacec et al. [8] presents an innovative technical solution for exploiting the flue gas of a district heating gas fired boiler. The heat pump studied consisted of four heat pumps connected in series and resulted in COPs of 4.92 to 5.9 for each of the individual heat pumps and an overall system COP of 5.51. For an equivalent single heat pump, the COP lowers to 3.82. The flue gases of the hot water boiler were first cooled from a temperature of 140 °C to a temperature of 48 °C (dew point), obtaining approximately 500 kW of heat. Most of the heat, about 920 kW, was obtained by cooling the flue gases from the dew point to 25 °C, producing condensate and recovering the latent heat.

Qu et al. [9] considered a heat recovery absorption heat pump to make use of the latent heat in the flue gas of a natural gas fired boiler. An absorption heat pump (AHP) is a heat pump driven by thermal energy in contrast to compression heat pumps that are driven by mechanical energy. In this case, three sources of thermal energy are used for the AHP, hot water, flue gas and natural gas, it was concluded that all three sources of thermal energy for the AHP improved the efficiency. Using the flue gas provided a faster payback of the installation cost, although a lower increase in efficiency.

In a multi-criterion comparison of compression and AHPs, Xu et al. [10] concluded that the compression heat pump had a higher COP than the AHP, but lower exergy efficiency, indicating more irreversible loss. Higher COPs for compression heat pumps was supported by Tan et al. [11], who also found mechanical heat pumps to have the best energy performance. Up to 30% higher COPs are reported compared to AHP, absorption heat transformer (AHT), and steam jet pump (SJP).

2.4. Heat pump technology

Well-designed and integrated industrial heat pumps can generate process heat efficiently by upgrading an ambient or low-temperature heat source. A vapour compression cycle is used to absorb heat from the source to heat a refrigerant and then, following compression using electricity, supply heat to the sink. Aside from the opportunity to recover and upgrade waste heat, heat pumps have other advantages. Heat pumps do not require ash disposal, fuel handling and other such operational costs, and may require less supervision and operator intervention. However, while industrial heat pumps typically draw around a quarter to a third of the equivalent thermal load as electricity (depending on their COPs), they may require an increase to a site's electricity supply capacity [15]. Cabling and switching equipment on-site can also be expensive, especially for long distances [15]. There are a number of heat pump configurations that can be employed. As generating hot water does not require high temperatures, relatively simple, single-compressor configurations can be utilised.

2.4.1. Basic cycle

The simplest heat pump configuration is shown in Figure 2-1. The basic heat pump has four main components: the evaporator, the compressor, the condenser and the expansion valve. The working fluid flows around the cycle in a closed loop. Heat from a source is transferred to the working fluid in the evaporator which vaporises the working fluid. The vapour is compressed to a higher temperature and pressure. The hot vapour rejects heat to the sink in the condenser. The final stage in the cycle is for this, now condensed, vapour to be expanded to a lower pressure and temperature to enter the evaporator and restart the cycle [16].

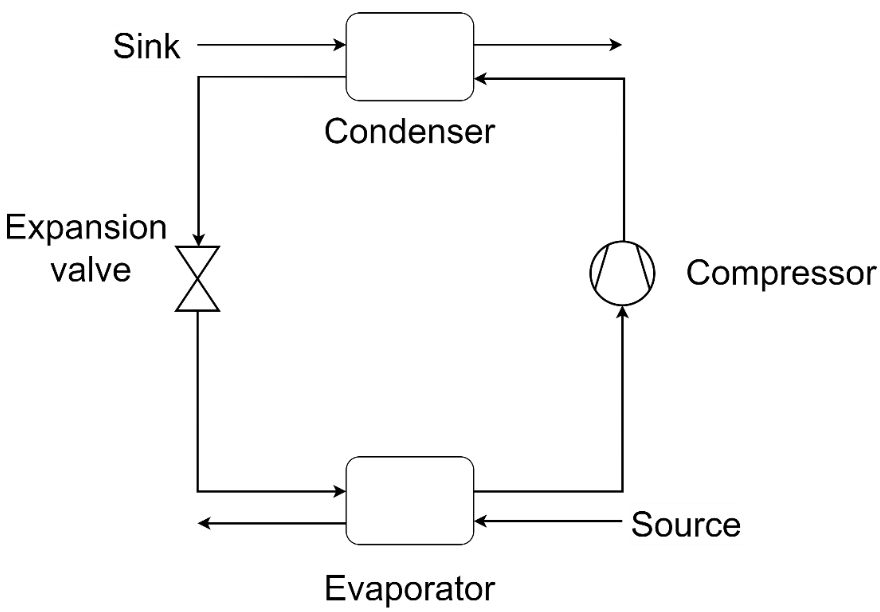


Figure 2-1 Basic heat pump cycle configuration

2.4.2. Subcooling

Subcooling refers to cooling the refrigerant to below the saturation temperature, following condensing, therefore increasing the heat transfer to the sink (Figure 2-2). Heating COP is significantly improved by adding subcooling according to Pitarch et al. [17]. A subcritical cycle with propane has demonstrated good performance for hot water production when working with subcooling [17]. Hervas-Blasco et al. [18] suggests subcooling plays an important role in the heat pump efficiency and an increase of the COP up to 30% can be acheived. Another study indicated even higher gains of up to 70.4% [19].

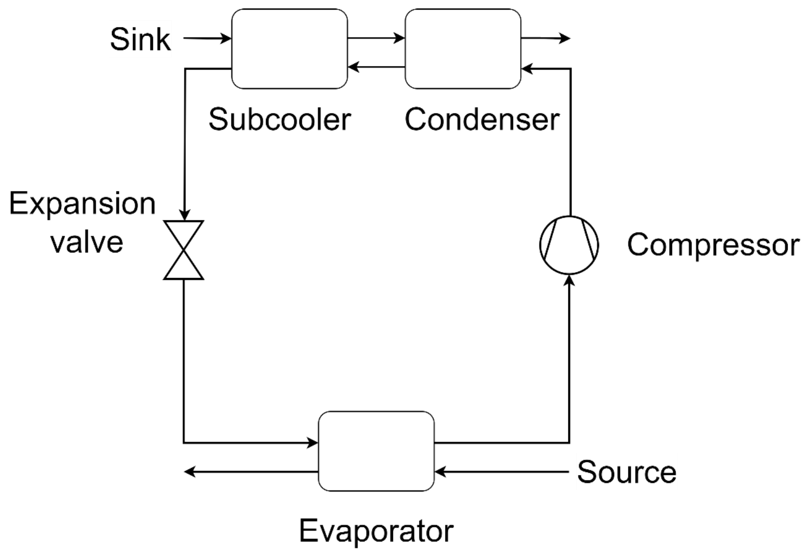


Figure 2-2 Heat pump with subcooling cycle configuration

2.4.3. Internal Heat Exchanger

An internal heat exchanger (IHX) is another option for improving the performance of the heat pump. Mato-Babiloni et al. [20] reports a positive influence on energy efficiency with the addition of an IHX. The IHX is typically placed between the outlet of the condenser and the inlet to the compressor, refer to Figure 2-3. This raises the temperature into the suction of the compressor, reducing the enthalpy change required, and therefore the work consumption, across the compressor. While moderate increases in COP from an internal heat exchanger were described, another benefit of an internal heat exchanger comes from ensuring dry gas into the compressor, improving the reliability of the compressor.

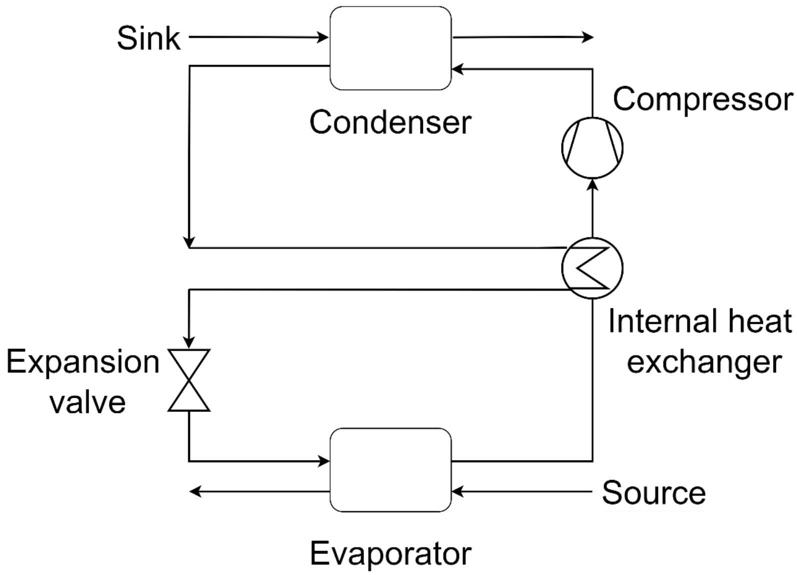


Figure 2-3 Heat pump cycle with internal heat exchanger configuration

2.5. Refrigerants

The choice of refrigerant or working fluid for a heat pump has a significant impact on its performance. No fluid is ideal in all aspects and will likely have at least one or more negative attributes. Therefore, trade-offs in cost, efficiency, emissions, toxicity and flammability, must be made when selecting a refrigerant. This section will examine the properties of different refrigerants to identify a suitable refrigerant for the case study application.

Refrigerants have been around since the vapour compression cycle was invented by Parkin in the 1830s [21]. The earliest refrigerants were flammable and toxic, and, as result, leaks were hazardous. This prompted the development of synthetic refrigerants. Non-toxic and non-flammable Freon was one of the earliest employed. It was discovered that Freon and common Chlorofluorocarbons (CFCs) refrigerants were depleting the ozone layer which led to the development of hydrochlorofluorocarbons (HFCs) [22]. These were highly efficient and did not deplete the ozone layer to the same extent. They did, however, have a high Global Warming Potential (GWP) and so are also being phased out. In October 2016, New Zealand joined 196 other countries to adopt the Kigali Amendment to the

Montreal Protocol on substances that deplete the ozone layer [23]. The measure used to categorise the effect of the refrigerant on the ozone layer is referred to as Ozone Depleting Potential (ODP) similarly the GWP measures the potential for global warming compared to CO_2 and is calculated over specific time periods, 20 or 100 years.

HFCs are commonly used in air conditioning and refrigeration [23]. However, as refrigerants with high GWPs are being phased out, alternative refrigerants must be developed. Additionally, using a refrigerant with a high GWP will reduce the effectiveness of using a heat pump to decarbonise. Natural refrigerants with low or negligible ODP or GWP are now being examined. These include ammonia NH_3 (R717), carbon dioxide CO_2 (R744) and hydrocarbons. GWP below 150 is considered low or negligible [24]. Palm [22] concluded that using hydrocarbons will result in COPs equal to, or higher than, those of similar HFC systems.

When looking at the suitability of a refrigerant, there are thermodynamic properties which need to be considered. The critical temperature and pressure will limit the condenser temperature for subcritical cycles. Similarly, the boiling point of the fluid will be important for the evaporator conditions. The waste heat needs to be at temperatures above the evaporator temperature. Consideration should also be given to the pressure in the evaporator. It should be above atmospheric to prevent air ingress, which is very important when the fluid is flammable.

Refrigerants under consideration for the case study are presented in Table 2-1. This study will focus on subcritical heat pumps, which rules out CO_2 due to its low critical point. Although water is also a possible natural refrigerant, the normal boiling point at atmospheric conditions is too high and is, therefore, excluded.

Table 2-1 Common natural working fluids and their properties [25], [26], [27]

ASHRAE Number	IUPAC Name	ODP	Net GWP 100-yr	Molar mass g/mol	Normal boiling point °C	Critical temp ° C	Critical Pressure kPa	ASHRAE safety group
RC-270	Cyclopropane	0	86	44.1	-32.86	125.15	5579	A3
R-290	Propane	0	3.3	44.1	-42.1	96.7	4248	A3
R-600	Butane	0	4.0	58.1	0.0	152	3796	A3
R-600a	Isobutane	0	3.0	58.1	-11.7	134.7	3640	A3
R-601	Pentane	0	4.0	72.1	36.1	196.6	3358	A3
R-717	Ammonia	0	0.0	17.0	-33.3	132.4	11280	B2L
R-718	Water/ Steam	0	0.2	18.0	100.0	373.9	22060	A1
R-744	Carbon dioxide	0	1.0	44.0	-78.0	31.0	7390	A1

2.5.1. Ammonia

Bamigbetan et al. [25] discusses ammonia as a widely used refrigerant for both heating and cooling. It has GWP and ODP of zero (Table 2-1) therefore the impact on the environment is low. Ammonia has a high compressor discharge temperature which will enable production of high hot water temperatures around 80-90 °C, for use in cleaning in dairy factories. Ammonia's relatively low critical temperature of 132.4 °C limits the use to outputs around 100 °C, [28] which is acceptable for hot water production. The ASHRAE safety group is B2L which indicates toxicity and low flammability (Table 2-1) therefore process safety systems, such as alarms to detect leaks, will need to be installed.

2.5.2. Hydrocarbons

Hydrocarbons have relatively low or negligible ODP and GWP, compared to CFCs and HFCs, and good thermodynamic properties [25] with respect to critical temperature and boiling points appropriate for particular conditions. The most significant challenge in using hydrocarbons is flammability. At low source temperatures (e.g., freezing), common hydrocarbons operate below atmospheric pressure, which increases the risk of air ingress providing a flammable mixture.

Bamigbetan et al. [25] compared propane and butane heat pumps, indicating that the higher critical temperature of butane (152 °C) will allow for delivery of higher heat to the sink compared to propane with a lower critical temperature (97 °C). However, as the critical temperature of propane is above the hot water temperature requirement of 80 °C, it is suitable for hot water generation. Propane is also a good candidate for subcooling, as it has a high specific heat capacity in liquid state compared to other refrigerants [29]. Another beneficial characteristic of propane is that it can work at a wider range of evaporating temperatures, making it suitable for industrial waste heat recovery [17]. While cyclopropane has a higher GWP than the other hydrocarbons, it is still well below the level of 150 considered low or negligible GWP [24]. R134a, which is a common refrigerant, has a GWP of 1430 [30]and so cyclopropane is significantly lower than this.

2.5.3. Mixtures with temperature glide

Zühlsdorf et al. [31] indicates that mixing of refrigerants can widen their range of application, as unfavourable properties of a single component can be compensated by a second component. Blended fluids can benefit from zeotropic behaviour. A zeotropic mixture has two or more components that have different boiling points resulting in a non-isothermal phase change. This temperature change during isobaric evaporation and condensation is known as temperature glide [32]. Matching heat source and sink temperatures with a temperature glide can improve the performance of a heat pump [25], [32].

Mixtures can reduce compressor pressure ratio and increase volumetric heating effect, however, zeotropic mixtures may require more complex component design [31]. Poor design, resulting in leaking during operation, can change the composition of the refrigerant mixture and decreasing performance. Additionally, the temperature difference between the refrigerant and the sink reduced by matching the temperature glide, along with diffusion resistances during phase change, can result in a larger heat exchanger area and therefore increase capital investment. These factors need to be considered for each individual case.

Roskosch et al. [32] compared simulation with experimental data and finds, for mixtures of isobutane/propane, isobutane/propene and n-butane/propene , that the simulated COPs are significantly higher than the experimental data. The COP was only slightly increased when compared to the most efficient single component of the mixture. Roskosch et al. [32] suggested that this was due to composition-dependent compressor efficiency.

2.6. Conclusions

The review shows utilising the biomass flue gas for hot water production is a promising way to recover and upgrade the wasted energy in the flue gas.

From the literature, $C_{31}H_{45}O_{24.5}$ can be used for approximating the composition of wood chips used in New Zealand. The level of moisture determines the quality of the fuel and contributes to the energy available for recovery.

Three heat pump cycles show the ability to produce hot water effectively. The basic heat pump cycle can provide hot water and enhancements of subcooling and adding an internal heat exchanger should improve this.

Taking into consideration the GWP and ODP, natural refrigerants with the appropriate thermodynamic properties include ammonia and hydrocarbons. Butane, propane and pentane show promise. While mixtures of butane/propane and propane/pentane could improve the performance, they may require very specific design and operating parameters to realise the benefits.

Chapter 3
Methods

3.1. Introduction

To quantify the benefits that could be gained from integrating a direct heat exchanger and heat pump with a biomass boiler, a model of the boiler and flue gas is required to determine the energy available for recovery. The information from this model is used as the source for the heat pump. Three heat pump cycles are compared for producing hot water. This chapter outlines the methods used to model the boiler, flue gas and heat pump cycles.

3.2. Modelling approach

3.2.1. Overview

Due to its flexibility and versatility, the most applicable program to model the individual industrial case study is Microsoft Excel. Figure 3-1 presents the overall calculation methodology. The flue gas composition is calculated on a 1 mole basis. This provides both the flue gas composition and flue gas enthalpy. The mass of fuel in 1 mole is calculated. Total fuel flow is calculated from the boiler duty. The flue flow is then scaled from 1 mole to total flow using the ratio of 1 mole of fuel to the total fuel flow. The flue flow and the enthalpy determined in the model at different temperatures will be inputted to the heat pump model in Chapter 6 .

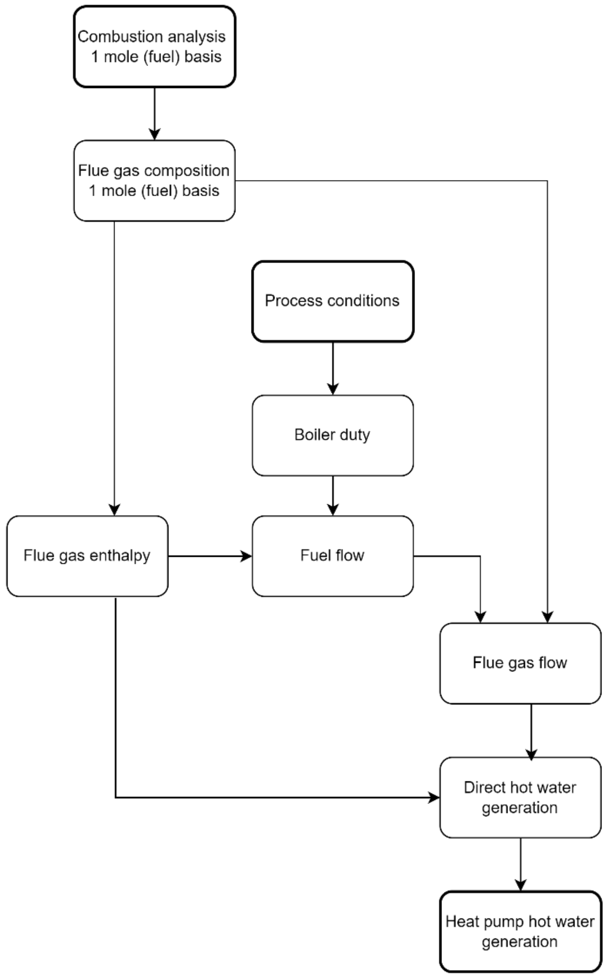


Figure 3-1 Overall calculation method flow diagram

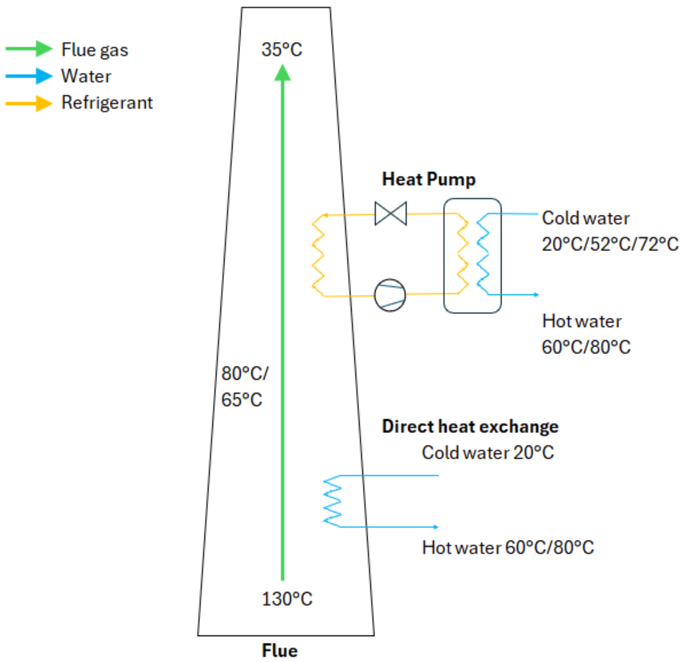


Figure 3-2 Schematic of direct heat exchange and heat pump

3.2.2. Thermodynamic properties

REFPROP [33] was used for determining the thermodynamic properties of the fluids. REFPROP is a commercial thermophysical properties database for industrial fluids. It has more fluids in the database than CoolProp [30]. CoolProp was used where REFPROP could not supply the information, predominantly for humid air.

3.2.3. Combustion

Biomass combustion has been modelled to calculate the amount of heat available for recovery in the flue gas. This section explains the method for analysing and modelling the combustion, its flue gases and the heat that it contains.

3.2.3.1. Flue gas

To calculate the flue gas flow, the amount of fuel combusted to produce the flue gas must be calculated using the flue gas enthalpy. Modelling combustion using a 1 mole basis is the first step to determine the components of the flue gas. The resulting flue gas composition, and partial pressures of the components, can then be used to determine the enthalpy of the flue gas. This information combined with the boiler duty will enable the fuel flow to be found and ultimately the flue gas flow.

From the literature review the chemical formula for an approximation of New Zealand Radiata pine is $C_{31}H_{45}O_{24.5}$ [13]. Balancing the equation using stoichiometry, the simplified combustion (excluding nitrogen) equation is

$$C_{31}H_{45}O_{24.5} + 30\ O_2 \rightarrow\ 31\ CO_2\ +\ 22.5\ H_2O$$

Moisture in the fuel (Equations (3) and (4)) and the humidity in the combustion air must be included to correctly quantify the temperature-enthalpy profile of the flue gas as this is a significant portion of the waste heat to recover. Nitrogen must also be accounted for despite remaining unchanged in the combustion reaction as it carries heat in the flue gas. Finally, excess air is required to ensure complete combustion. To calculate the nitrogen coefficient, the ratio of oxygen to nitrogen in the air is used (Equation (5)).

$$m_{H_2O\ fuel} = \frac{m_{1\ mole\ of\ fuel}}{1 - \%MC_{fuel}} - m_{1\ mole\ of\ fuel} \tag{3}$$

$$n_{H_2O\ fuel} = \frac{m_{H2O\ fuel}}{M_{H_2O}} \tag{4}$$

$$n_{air} = O_2 + 3.76\ N_2 \tag{5}$$

Humidity in the combustion air is determined using the saturation pressure at the air inlet temperature and the relative humidity. Taking all these factors into account, the resulting equation for combustion of wood fuel with 50% moisture content (i.e., hog fuel from the case study) and 6% O_2 in the flue gas is:

$$\begin{aligned} C_{31}H_{45}O_{24.5} + 44.9\ H_2O_{fuel} + [47.8\ O_2 + 179.8\ N_2 + 2.8\ H_2O_{air}] \\ \rightarrow 31\ CO_2 + 70.3\ H_2O + 17.8\ O_2 + 179.8\ N_2 \end{aligned} \tag{6}$$

Using the partial pressures of each component i in the flue gas (Equations (7) and (8)) and temperature of the flue gas, the specific enthalpy of each component can be calculated with REFPROP (Equation (9)).

$$\%Volume_i = \%mole_i = \frac{n_i}{n_{total\ flue}} \tag{7}$$

$$P^*{}_i = P_{total} \times \%Volume_i \tag{8}$$

$$h_i = f(i, P_i^*, T_{flue}) \tag{9}$$

The mass of each component i in the flue gas can be found using Equation (10), where n is the stoichiometric coefficient of each component and M is the molar mass. The enthalpy per mole of fuel, on a dry gas basis, that is available in the flue gas is then calculated using Equation (11).

$$m_i = n_i \times M_i \tag{10}$$

$$h_{flue\ DG} = \sum(m_i \times h_i)/\sum m_{iDG} \tag{11}$$

The mass 1 mole of fuel is calculated by summing the masses of the constituent atoms (Equation (12)).

$$m_{fuel(@\ 1\ mol\ fuel)} = \sum(M_i\ \times z_i)_{C_{31}H_{45}O_{24.5}} \tag{12}$$

Where M_i is the molecular weight of the carbon, C, hydrogen, H, and oxygen, O, in the fuel in (g/mol) and the z_i is the number of each atom in the fuel.

To calculate the total flue gas mass flow, the ratio of fuel mass flow to the mass of 1 mole of fuel is multiplied by the mass of flue gas from combustion of 1 mole of fuel. The method to calculate the fuel mass flow is given in Section 3.2.3.2.

$$\dot{m}_{flue\ gas} = \dot{m}_{fuel} \times \frac{m_{flue\ (@\ 1\ mol\ fuel)}}{m_{fuel(@\ 1\ mol\ fuel)}} \tag{13}$$

When the temperature of the flue gas becomes lower than the dew point, water in the flue will condense which needs to be removed. First, the weighted average of the dry gases in the flue gas is calculated:

$$M_{ave,dry\ flue} = \frac{\sum(M_i \times \%Volume_i)}{\%Volume_{dry\ flue}} \tag{14}$$

Then the water condensed per mole of fuel is the total water in the flue gas minus the water remaining as a vapour in the flue gas once the temperature is lower than the dew point (Equation (15) The total water in the flue gas is made up of water from combustion, fuel and humidity in the air.

$$m_{H_2O\ condensed} = m_{H_2O\ flue} - \frac{M_{H_2O}}{M_{ave,\ dry\ flue}} \times \frac{P^*_{H_2O\ flue}}{P_{total} - P^*_{H_2O}} \times \frac{m_{dry\ flue}}{M_{H_2O}} \tag{15}$$

This condensed water is removed after extracting its latent heat.

The flue gas mass flow for a given fuel composition and enthalpy can be determined with a calculation using the fuel mass flow which is calculated in the following Section 3.2.3.2.

3.2.3.2. Fuel mass flow

The sum of the enthalpy of the steam generated in the boiler (HPS), plus the enthalpy of the blowdown (BD), minus the enthalpy in the incoming boiler feedwater (BFW) is represented by $\dot{Q}_{boiler}$, as shown in Equation (16). It is assumed that there are no heat losses.

$$\dot{Q}_{boiler} = \dot{Q}_{HPS} + \dot{Q}_{BD} - \dot{Q}_{BFW} \tag{16}$$

Δh of the flue is the HHV minus the enthalpy in the flue gas plus the air enthalpy, calculated using Equation (17).

$$\Delta h_{flue} = HHV - (h_{flue\ DG} r_1 + h_{air}\ r_2) \tag{17}$$

Where:

$$r_1 = \frac{m_{dry\ flue(@\ 1\ mol\ fuel)}}{m_{fuel(@\ 1\ mol\ fuel)}}$$

$$r_2 = \frac{m_{dry\ air(@\ 1\ mol\ fuel)}}{m_{fuel(@\ 1\ mol\ fuel)}}$$

Mass flow of the fuel is calculated as follows.

$$\dot{m}_{fuel} = \frac{\dot{Q}_{boiler}}{\Delta h_{flue}} \tag{18}$$

3.2.4. Direct heat exchanger

The flue gas can be used to produce hot water directly to bring the temperature of the flue gas down before considering it as a source for a heat pump (Figure 3-2. The mass flowrate of hot water produced using a Direct Heat Exchanger (DHX) from the flue gas is calculated using Equation (19).

$$\dot{m}_{HW} = \frac{\dot{Q}_{DHX\ flue}}{(c_P \Delta T)_{HW}} \tag{19}$$

Where $\dot{Q}_{DHX\ flue}$ is the change in enthalpy of the flue gas as it is cooled across the heat exchanger from the initial temperature of the flue gas minus the chosen outlet temperature of the flue gas which depends on the type of hot water being generated. ΔT (°C) is the change in temperature of the hot water from the inlet to the desired temperature, either 80 °C or 60 °C. The heat capacity of water c_P (kJ/kg·K) is assumed to be constant across the range of ΔT.

3.2.5. Heat pump cycle

The flue gas heat source (mass flow and enthalpy) calculated in the previous section is used as inputs for the heat pump model. Three different cycles outlined in the Literature Review in Chapter 2 were considered to calculate the hot water generation. This section outlines the calculations for the basic heat pump cycle. Once the basic model is completed, modifications can be made to include subcooling. The third cycle to be modelled is the addition of an IHX. Necessary changes to the model for each of these cycles are outlined in 3.2.5.2. and 3.2.5.3.

3.2.5.1. Base model – single stage heat pump

The heat pump base model can be created in Excel in the following manner, resulting in the calculation of hot water generated. The performance metrics discussed in Section 3.3. are also able to be calculated from this model. The model assumes:

- the pressure drop across both the evaporator and the condenser is zero.
- isenthalpic expansion across the expansion valve
- heat loss is ignored

The first step is to calculate the amount of heat in the flue gas after the DHX which can be transferred in the evaporator from the flue gas to the refrigerant (Equation (20)).

$$\Delta \dot{Q}_{evap,flue} = \dot{m}_{flue} \times (h_{evap\ in,flue} - h_{evap\ out,flue}) \tag{20}$$

Where the specific enthalpies of the flue gas are calculated in the combustion model at the temperatures in and out of the flue side of the heat pump evaporator heat exchanger.

The refrigerant exiting the evaporator is assumed to be a vapour with a quality (mass fraction of vapour) of 1. Setting the minimum approach temperature (ΔT_{min}) will specify the temperature out of the evaporator and into the compressor. A ΔT_{min} of 10 °C was used. Depending on the inlet temperature of the water and the refrigerant, the amount of subcooling or use of an IHX, the temperature out of the evaporator will be calculated in different ways. For the evaporator the closest approach will be at one or the other end of the T-h line, i.e. inlet and outlet temperature of the flue gas. This is more evident in the case of refrigerant mixtures where the T-h line of the refrigerant also slopes.

The evaporator pressure is selected such that the evaporator maintains a minimum approach temperature along its length.

$$P_A = f(\text{refr}, T_{\text{evap out,flue}} - \Delta T_{\min}, h) \tag{21}$$

$$P_B = f(\text{refr}, T_{\text{evap in,flue}} - \Delta T_{\min}, Q) \tag{22}$$

The evaporator pressure is first calculated at two temperatures (Equations (21) and (22)) to determine candidate pressures to prevent minimum temperature violations. In this case, the minimum of the two candidate pressures is used as the evaporator pressure. Note that the enthalpy calculation at the inlet of the evaporator requires iterative calculation and solving. Use of the "=iferror()" function in Excel helps in the iterative calculation process to always maintain numeric values in the calculation. Pressure, entropy and enthalpy can be obtained from REFPROP using temperature and quality at the exit of the evaporator.

The next step is to calculate the properties of the refrigerant out of the condenser. At the condenser outlet, it can be assumed the refrigerant is a liquid with a quality of zero, to determine the saturation temperature. Using $\Delta T_{\min}$ and the hot water inlet temperature, the temperature out of the condenser for the refrigerant can be determined (Equation (23)). In a similar manner to the evaporator, the condenser pressure is calculated at two temperatures, (Equations (23) and(25)), and this time the maximum pressure is used. The "=iferror()" function is also utilised here to ensure the iterative calculation solves.

$$T_{\text{cond out,refr}} = T_{\text{cond in,HW}} + \Delta T_{\min} \tag{23}$$

The closest approach temperature will be at one of two points, either the hot water inlet temperature or at the saturated vapour point for the refrigerant in the condenser, the pinch point on the T-h graph. This point can then be used for applying the ΔT_{min}. The enthalpy of the refrigerant vapour at the saturation point is found using pressure and quality. The percentage of desuperheating needed to get the vapour lowered to the saturation temperature is calculated using:

$$\% \text{ desuperheat} = \frac{h_{comp\ out} - h_{sat}}{h_{comp\ out} - h_{cond}} \quad (24)$$

The percentage desuperheating value is used to calculate the corresponding hot water temperature at the pinch point.

$$T_{HW\ @pinch} = T_{HW\ out} - \%\ \text{desuperheat} \times (T_{HW\ out} - T_{HW\ in}) \quad (25)$$

This is the pinch point that the ΔT_{min} is applied to. P is calculated at this point. The other temperature, at which the pressure will be determined, is from Equation (23) above.

Using REFPROP, enthalpy can be determined using temperature and quality (i.e., 0) or temperature and pressure for a subcooled case, see Section 3.2.5.2. .

The mass flow of the refrigerant can now be calculated using:

$$\dot{m}_{refr} = \frac{\Delta \dot{Q}_{evap,flue}}{\Delta h_{evap,ref}} \quad (26)$$

Where $\Delta \dot{Q}_{evap,flue}$ is the energy in the flue gas found in Equation (20) and Δh_{refr} is the change in refrigerant enthalpy across the evaporator.

The compressor discharge properties can be calculated assuming isentropic compression and applying an isentropic efficiency (Equation (27)). Isentropic enthalpy ($h_{comp\ out,s}$) is found using REFPROP with pressure and entropy inputs. Isentropic efficiency of the compressor (η_s) allows determination of non-ideal or actual enthalpy ($h_{comp\ out,a}$) of the compressor discharge. η_s is assumed to be 70%.

$$h_{comp\ out,a} = \frac{(h_{comp\ out,s} - h_{comp\ in})}{\eta_s} + h_{comp\ in} \quad (27)$$

The work of the compressor is the main energy requirement for the heat pump and is a measure of the electricity required for the heat pump. Work into the compressor ($\dot{W}$) can be calculated using:

$$\dot{W} = \dot{m}_{refr} \cdot (h_{comp\ out,a} - h_{comp\ in}) \tag{28}$$

The heat transferred in the condenser $\Delta\dot{Q}_{cond}$ can be determined using:

$$\Delta\dot{Q}_{cond} = \dot{m}_{refr} \cdot \left(h_{comp\ out,a} - h_{cond\ out}\right)_{refr} \tag{29}$$

Finally, the hot water mass flow can be calculated:

$$\dot{m}_{HW} = \frac{\Delta\dot{Q}_{cond}}{\Delta h_{HW}} \tag{30}$$

3.2.5.2. Addition of a subcooler

The temperature of the refrigerant out of the condenser is reduced to below saturation to allow more heat to be transferred to the heat sink using a subcooler. Refer to Figure 2-2. The enthalpy of the refrigerant out of the condenser is found with REFPROP, using T and P. Quality can no longer be used as the liquid is subcooled. This temperature is a design variable in this cycle and can be set within limits discussed in Section 4.5.2. . The remainder of the model is the same as described in Section 3.2.5.1.

3.2.5.3. Addition of an Internal Heat Exchanger

The heat from the refrigerant coming out of the condenser provides heat to the refrigerant going into the compressor through an internal heat exchanger (IHX). Refer to Figure 2-3. As discussed in the literature review, this maintains a dry gas into the compressor assisting reliability and can also improve efficiency. The enthalpy of the refrigerant into the expansion valve will be reduced as heat has been added to the flow into the compressor. This ensures the refrigerant enters the expansion valve as a liquid.

The temperature of the refrigerant into the compressor will be the IHX outlet temperature and will be varied to analyse the effect of internal heat exchange (refer to Section 4.5.3.). The enthalpy of the refrigerant into the expansion valve is determined by the heat exchanged to the compressor inlet in the IHX ($\Delta\dot{Q}_{IHX}$):

$$\dot{Q}_{\text{valve in,refr}} = \dot{Q}_{\text{cond out,refr}} - \Delta\dot{Q}_{IHX} \tag{31}$$

Once more, the remainder of the model is the same as described in Section 3.2.5.1.

3.3. Performance metrics

The primary goal of integrating a heat pump is to efficiently produce hot water (in m^3/hr). To compare various heat pump options, there are several metrics that can be used to evaluate the performance of the heat pump cycle and refrigerant combinations. A number of these come directly from the model and some are calculated subsequently. The coefficient of performance (COP) and volumetric heating capacity (VHC) are two parameters used for determining the performance of a refrigerant and cycle [34]. The COP and VHC can be calculated using Equations (32) and (33) respectively.

$$\text{COP} = \frac{\Delta\dot{Q}_{\text{cond}}}{\dot{W}} \tag{32}$$

The VHC describes the heating capacity per unit of swept volume and is calculated using Equation (33). VHC is important because it gives an indication of the compressor size and therefore the capital cost.

$$\text{VHC} = \frac{\Delta h_{\text{cond,refr}}}{\nu_{\text{refr}}} \tag{33}$$

Where $\Delta h_{\text{cond,refr}}$ is the enthalpy change of the refrigerant and ν_{refr} is the specific volume of the refrigerant at the compressor inlet in m^3/kg.

The specific work required per kilogram of hot water produced w is calculated to give an indication of the efficiency of producing the hot water.

$$w_{HW} = \frac{\dot{W}}{\dot{m}_{HW}} \tag{34}$$

Where $\dot{m}_{HW}$ is the hot water flow rate in kg/s and $\dot{W}$ is the work flow in kW.

Table 3-1 Summary of metrics

Metric	units
HW produced	m^3/hr
COP heat	
Work in	kW
VHC	MJ/m^3
Specific work	kJ/kg

3.4. Conclusions

A thermodynamic model of the boiler and flue gas has been constructed in Excel. This includes calculation of the composition of the flue gas and building of a heat pump model for the generation of hot water, for an industrial site. These models provide results which will be used to evaluate the ability to generate hot water using a heat pump and indicate the most effective way to do this. The model will need to be verified. To do this, the following chapter describes the case study which will give data for comparison.

Chapter 4
Industrial case study and scenarios

4.1. Introduction

In New Zealand, there are a significant number of industrial boilers. As these boilers age, they are being replaced with more sustainable options. Alongside replacement of decommissioned boilers, a number of boilers are being converted from fossil fuels to biomass, with biomass boilers becoming progressively common [5], [6]. This chapter presents the industrial case study for the biomass boiler which is then used for the heat pump analysis. The site uses a biomass boiler to create steam for dairy processing. The flue gas is vented to the atmosphere and would provide a useful waste stream to recover and upgrade. Process and Instrumentation Diagrams (P&IDs) of the boiler, and the hot water system, were provided by the site. The boiler drawings were condensed into the Process Flow Diagram (PFD) in Figure 4-1. From the P&IDs, the tags for the instruments supplying the relevant data were compiled. The industrial site provided data for these measurements.

4.2. Boiler description

The plant capacity is 15.6 MW_{th}, providing up to 39 bar(g) steam at 250 °C. The fuel for the biomass boiler is hog fuel and wood chips. Chips from wood residues and off cuts are generally cut to a consistent size. Hog fuel is often a mixture of different sources and made up of different size and shapes.

In the boiler system, two fans deliver combustion air, one fan provides primary air and the other directs secondary air to various points in the boiler. Primary air is blown into the combustion fluid bed and can be mixed with recirculated flue gas to control the temperature in the bed. Primary air can also be preheated if necessary. The air intake is from the upper level of the boiler hall and is therefore much warmer than ambient air from outside. Typically, this is 40 °C. The flue gas is blown through an electrostatic precipitator to remove impurities, before moving up the stack to atmosphere.

The fuel is provided to a hot fluidised sand bed. The primary air is directed underneath the sand to fluidise it. The stored energy in the sand helps level out changes in fuel moisture content. This is where combustion starts. Good fluidisation ensures good combustion conditions leading to complete combustion and lowered emissions. If the bed temperature increases too much, flue gas is recirculated to control it. This reduces the oxygen content to restrain combustion therefore lowering the temperature to below the melting point of the sand compounds.

Make up feedwater comes in via the deaerator where it is mixed with condensate return, steam for heating and any condensate returning from the primary air preheat. Chemical dosing is utilised to protect against corrosion and scaling. A boiler feedwater pump feeds water at 130 °C to the boiler at approximately 6 kg/s or 21.6 t/h. The system has a blowdown to restrict the concentrating of dissolved solids in the boiler water. This is typically 1-2 % of the feed water flow.

The steam supply to the site comes from the steam drum at the top of the boiler. This is provided to the plant’s main steam header and surplus steam is directed to a steam accumulator to enable steam supply to be maintained despite fluctuations in pressure or loss of the boiler.

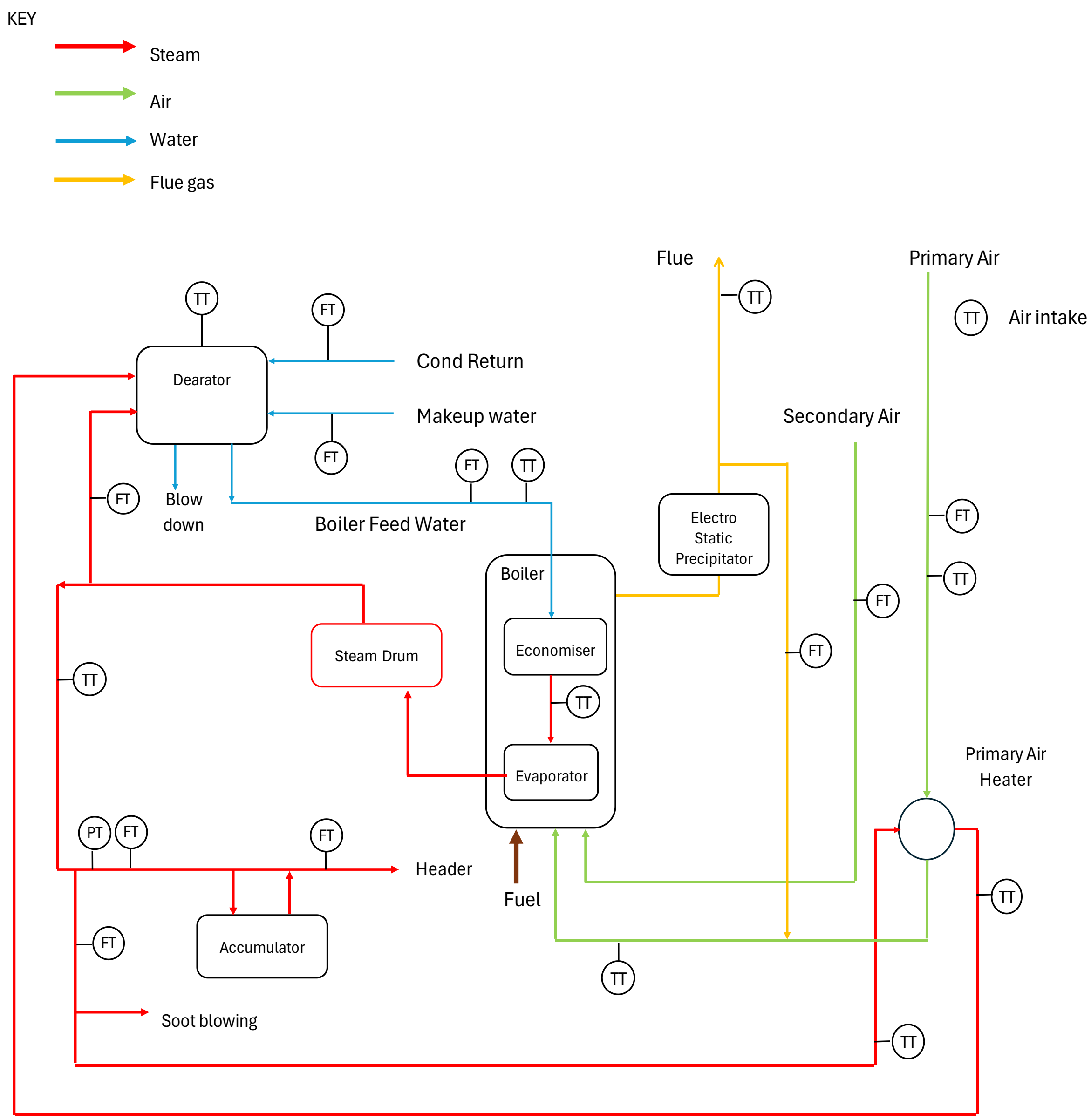


Figure 4-1 Process Flow Diagram

4.3. Boiler measurements

Data was provided by the site in a CVS file for the tags identified and requested. This was converted to an Excel Workbook file and analysed for steady run conditions. Although the data was variable, there were times when the data was consistent enough to allow it to be used in the model successfully. Where it was unavailable, or not of sufficient quality, assumptions were made to progress with the study. Quality of the data can be influenced by a number of factors, for instance, instrument issues. In cases where the data was intermittent, an average of the data points over a sufficient time period was deemed adequate for the study. The two measurements that were used in the model that were of average quality were the boiler feed water flow and the air intake temperature both of which were variable and the average over a steady run was used.

Blowdown data was unavailable and therefore an assumption made about the mass flow. In part, this was due to several unmetered streams entering the blowdown vessel. The pressurised blowdown water was flashed, and the vapour was returned to the deaerator. The liquid was then sent to a second blowdown vessel at atmospheric pressure where it was rejected to drain. The supplier information provided on the boiler indicated an overall blowdown of 1-2%. The conservative value of 2% was used for the model.

Primary and secondary air flows had low data point numbers per day. The average was taken, and the operational manuals and specification sheets were provided to corroborate and ensure a representative number was used in the model.

A table of the mean data of the conditions is shown in Table 4-1. A balance around the deaerator was not feasible due to too many unknowns but was not necessary for modelling the flue gas and integrating the heat pump.

Table 4-1: Case study data

Description	Type	Data quality	Mean value	units
*Boiler feed water	Flow	Good but variable	5.3	kg/s
*Steam to accumulator and header	Flow	good	4	kg/s
*Prim air flow	Flow Air	low amount of data/day	2.5	Nm^3/s
*Sec air flow	Flow Air	low amount of data/day	3.0	Nm^3/s
*Deaerator temperature	Temp	good	129	°C
*Main steam	Temp	good	241	°C
*Air intake	Temp	good	40	°C
*Primary air	Temp	good	153	°C
*Flue gas final	Temp	good	130	°C
*Main steam	Pressure	good	34	bar
*Flue gas oxygen	O_2 %	good	6	%
*Primary air	Temp	good but variable	40	°C
*Hog fuel 50% moisture		-	150	t/day
Steam to deaerator	Flow	inconsistent	1.2	kg/s
Steam to soot blow and air preheater	Flow	inconsistent	0.5	kg/s
Steam to header	Flow	inconsistent	3	kg/s
Condensate return	Flow	good	2.2	kg/s
Flue gas recirc flow	Flow Air	inconsistent	0.6	Nm^3/s
Economiser outlet	Temp	good	222	°C
Economiser inlet	Temp	good	130	°C
Makeup water	Flow	good	2.2	kg/s
Flue gas before ESP	Temp	good	135	°C
Flue gas before ESP	Temp	good	135	°C
Primary air heater Condensate	Temp	good	135	°C
Steam to air preheater	Temp	good	234	°C

*Data used in model

4.4. Hot water requirements

From the P&IDs provided, it could be seen that there are two temperatures of hot water required on the site, 80 °C and 60 °C. A basic schematic of the hot water tank and heating method is shown in Figure 4-2 below. This is representative of both the 60 °C and 80 °C hot water.

Steam flow to the heat exchanger which heats the hot water is monitored but no flow meters are installed on the hot water used in the plant. From the steam data, it could be seen there were two modes of heating in operation. The first mode is heating from cold water where the steam flow was high, and a second mode of recirculating the hot water when it dropped below a control temperature in the tank it was stored in. The temperature data for when recirculation was started for the 80 °C hot water was available for the model. This was 72 °C giving an 8 °C temperature lift requirement. It was assumed a similar scenario for the 60 °C hot water as there was no equivalent data available. The recirculation was assumed to begin when the temperature dropped by 8 °C or at 52 °C.

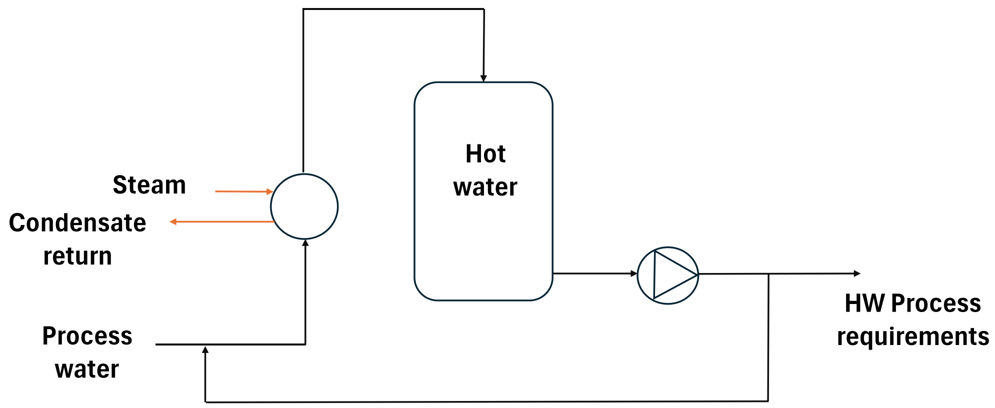


Figure 4-2 Hot water schematic

4.5. Scenarios evaluated

There are a number of scenarios that can be examined when investigating the integration of a heat pump into a biomass boiler. The inlet and outlet temperatures of the source and sink, and the refrigerant type are important variables to study. The heat pump cycle type is also crucial. The site requires hot water at two temperatures, 60 °C and 80 °C. This provides the sink outlet temperature for the heat pump. The initial temperature of the sink depends on the mode of heating used, whether heating from cold or recirculating. The flue gas temperature into the heat pump depends on the temperature of the hot water required. For the 80 °C hot water the flue gas has already been reduced in the direct heat exchanger to 80 °C and to 65 °C for the 60 °C hot water (see Section 5.4.). The flue gas can be reduced to a range of temperatures, 55 °C, 45 °C and 35

°C have been studied. Table 4-2, Table 4-3 and Table 4-4 below shows the different scenarios to be evaluated.

4.5.1. Refrigerants evaluated

From the literature review, the following pure refrigerants with the appropriate critical point and boiling point for hot water generation have been selected for evaluation.

- Ammonia
- Pentane
- Propane
- Isobutane
- Butane
- Cyclopropane
- Propane (50 wt%), Butane (50 wt%)
- Propane (50 wt%), Pentane (50 wt%)

As discussed in the literature review, the choice of refrigerants is based on environmental factors and the appropriate thermodynamic properties. Ammonia is a widely used refrigerant with a GWP and ODP of zero. The hydrocarbons selected have low or negligible ODP and GWP and good thermodynamic properties. The critical temperature and boiling point are in line with the sink and source temperatures.

4.5.2. Effect of subcooling

The flue gas temperature into the heat pump will be different depending on the type of hot water to be produced. The flue gas is at an initial temperature of 130 °C. As discussed in Section 5.4. , the DHX first generates hot water where the flue gas is lowered to either 80 °C for the 80 °C hot water or 65 °C for the 60 °C hot water, keeping the flue gas above the dew point to ensure the latent heat is utilised in the heat pump. A final flue gas temperature of 35 °C will be evaluated. This was repeated for each heat sink (Table 4-2), heating from cold water at 20 °C to the two hot water temperatures 60 °C and 80 °C, and also for recirculating from 52 °C or 72 °C. The extent which the refrigerant could be subcooled extent was limited by the inlet temperature of the heat sink, the hot water inlet temperature.

Therefore, the amount of subcooling can be up to 50 °C in the case of the 80 °C hot water, but only 30 °C is available for the 60 °C hot water. In the case of recirculation, only 5 °C is available for exploitation.

Table 4-2: Subcooling scenarios

Heat source °C	Heat sink °C	Subcooled temperature °C
80-35	20-80	0-50
80-35	72-80	0-5
65-35	20-60	0-30
65-35	52-60	0-5

4.5.3. Effect of internal heat exchange

For the same flue gas temperature decrease, lowering the flue gas to 35 °C, the IHX temperature out of the IHX into the compressor (Table 4-3) was varied to determine the effect on the performance metrics. An IHX out temperature of 25 °C indicates no heat exchange, i.e. the refrigerant temperature through the IHX is unchanged, and the higher IHX out temperature in the table is the maximum temperature possible into the compressor, which is the condenser out temperature minus ΔT_{min}.

Table 4-3: IHX scenarios

heat source °C	heat sink °C	IHX temperature out °C
80-35	20-80	25-80
80-35	72-80	25-80
65-35	20-60	25-60
65-35	52-60	25-60

4.5.4. Effect of source temperature

The source temperature is important to consider. In the case study modelled, the outlet temperature of the source was lowered to 35 °C. To evaluate the impact of the outlet source temperature on hot water generation efficiency, the flue gas outlet temperature was evaluated at 55 °C, 45 °C and 35 °C for the four sink temperature scenarios (Table 4-4).

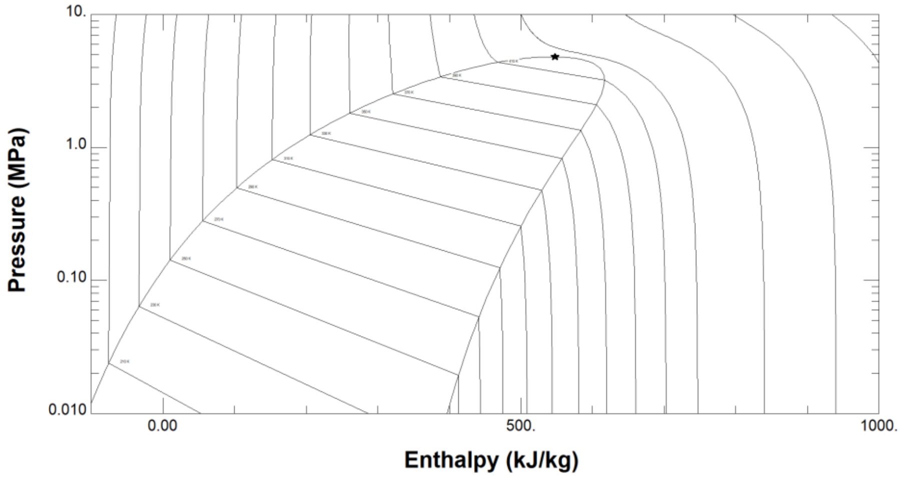
Table 4-4: Source temperature scenarios

Heat sink °C	Heat source °C
20-80	80-55 80-45 80-35
72-80	80-55 80-45 80-35
20-60	65-55 65-45 65-35
52-60	65-55 65-45 65-35

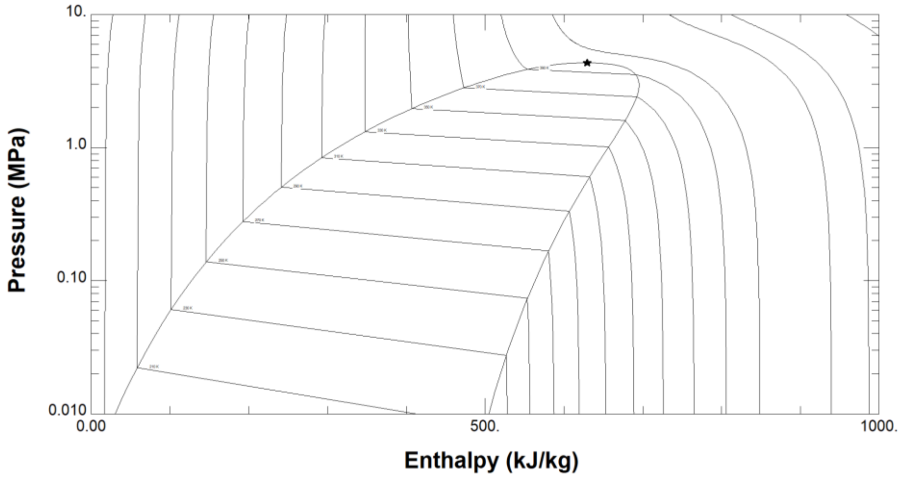
4.5.5. Effect of temperature glide

As outlined in the literature review in Chapter 2, mixtures can improve the performance of heat pumps by taking advantage of the temperature glide. Two refrigerant mixtures of equal fractions were explored: 1) propane and butane and 2) propane and pentane. The temperature glide can be seen in the Figure 4-3, Figure 4-4 and Figure 4-5, showing a sloping saturation line which compares to a zero slope horizontal line for a single component refrigerant. This allows closer matching of the temperatures of the fluids exchanging energy. The difference between the critical temperatures of propane and pentane is much greater than the difference between the critical temperatures of propane and butane which leads to a better glide match. This is more useful in a large temperature lift as in the case of heating hot water from cold rather than recirculating.

- Propane ; Pentane 0.5 ; 0.5 (labelled a) in the graphs below)
- Propane ; Butane 0.5 ; 0.5 (labelled b) in the graphs below)

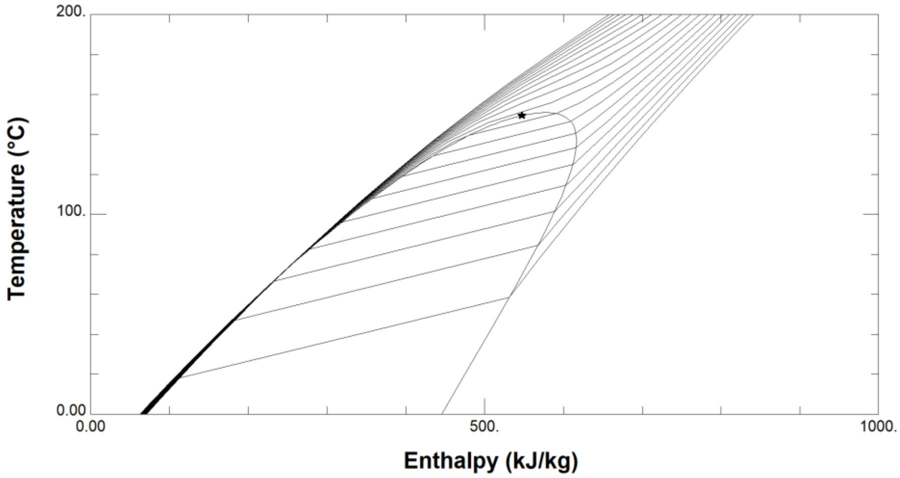


a)

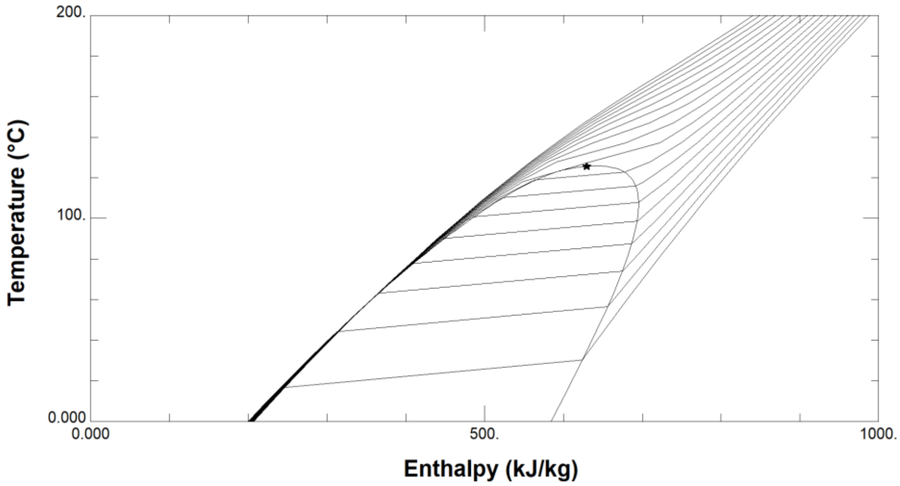


b)

Figure 4-3 Pressure enthalpy a) Propane ; Pentane 0.5 ; 0.5 b) Propane ; Butane 0.5 ; 0.5

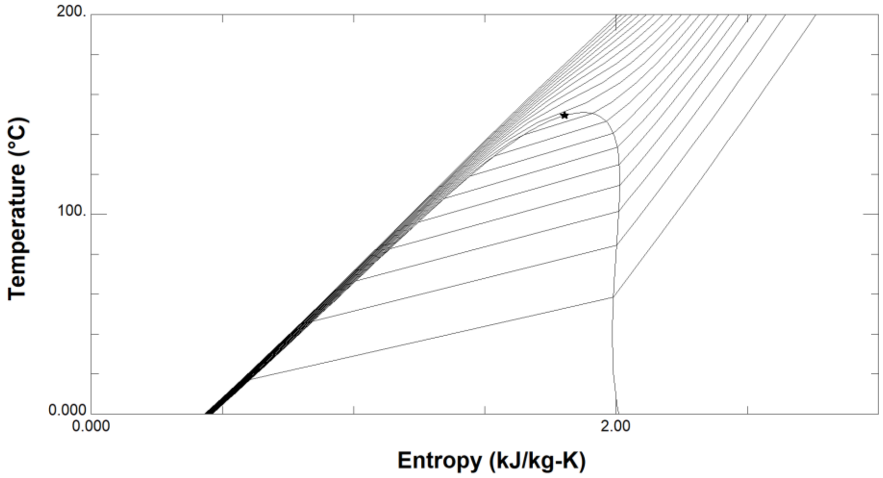


a)

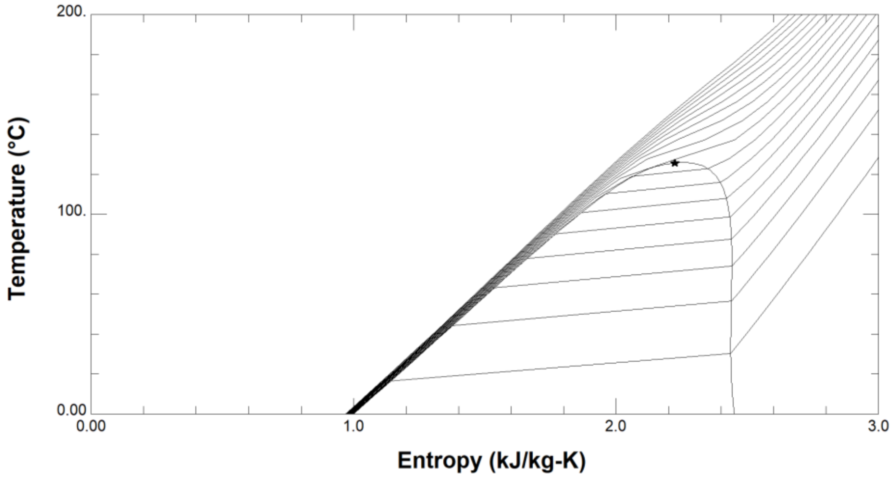


b)

Figure 4-4 Temperature enthalpy a) Propane ; Pentane 0.5 ; 0.5 b) Propane ; Butane 0.5 ; 0.5



a)



b)

Figure 4-5 Temperature entropy a) Propane ; Pentane 0.5 ; 0.5 b) Propane ; Butane 0.5 ; 0.5

4.6. Conclusions

The case study has been described which will be used to validate the results from the boiler and combustion parts of the model. The main indicators used for comparison are the mass flow of the fuel and the amount of combustion air for a given duty, flue gas O_2% and fuel type. The scenarios to be evaluated for the integration of a direct heat exchanger and heat pump with the biomass boiler flue gas have been outlined. The following chapters will present and discuss the results. Chapter 5 will discuss the combustion and flue gas results, while Chapter 6 concentrates on the heat pump.

Chapter 5
Biomass boiler analysis

5.1. Introduction

Data from the industrial site and its boiler system provides a basis to model the system. It also enables the model to be validated through a comparison of measured values and model calculated values. In this chapter, the results generated from the case study model for the boiler and flue gas are presented and compared to the case study data. The results of the combustion analysis are given.

5.2. Model validation and comparison

To validate the model, comparisons need to be made between the measured values from the site and values calculated from the model. There are two main parameters that indicate the combustion model is consistent. The first is the air flow which is calculated using the combustion model (Section 3.2.3.). The measured value of the air flow is given in normal cubic meters per second (Nm^3/s) and is the sum of the primary and secondary air flows.

Using hog fuel at 50% moisture and 6% O_2 content in the flue gas (Table 4-1), the comparison between the values is shown below.

Resultant flue gas flow is 6.8 m^3/s, which is 5.89 Nm^3/s. This compares to the site primary and secondary air flows of 2.5 Nm^3/s and 3 Nm^3/s = 5.5 Nm^3/s. This equates to a difference of 7%. Airflow measurements are not as accurate as fluid flows, and this is considered a good match between the model and measured values.

The fuel flow reported by the site is 150 t/day. The model calculated the daily fuel use to be 157 t/day, which agrees well with the site value. The difference is 4.6% which gives confidence in the model being correct within expected measurement variations.

5.3. Flue gas analysis

Using the model and the methods outlined in Chapter 3 , the mass flow of the fuel, and consequently the flue gas flow and composition, have been determined. The enthalpy has been calculated at various temperatures required by the heat pump model. The fuel composition is shown in Table 5-1 and the flue gas compositions at 130 °C, 80 °C and 35 °C are shown in Table 5-2, Table 5-3 and Table 5-4, respectively. As can be seen in the Table 5-4, the water in the flue gas has condensed as the flue gas temperature is below the dew point, causing the total enthalpy in the flue gas to decrease significantly.

Table 5-1 Fuel composition – 1 mole basis

Fuel in	n	M (g/mol)	m (g)
Fuel	1	809	809
O_2*	47.8	32	1530
N_2	179.8	28	5034
H_2O(fuel)	44.9	18	809
H_2O(air)	2.8	18	51
Total			8234

*Stoichiometric and excess

Table 5-2 130 °C flue gas composition – 1 mole basis

Flue gas 130 °C	n	%mol (= %vol)	P* (bar)	h (kJ/kg)	M (g/mol)	m (g)	H (kJ)
CO_2	31	0.10	0.11	601	44	1364	819
H_2O(comb)	22.5	0.08	0.08	2745	18	405	1112
H_2O(fuel)	44.94	0.15	0.15	2744	18	809	2220
H_2O(hum)	2.83	0.009	0.010	2745	18	51	140
H_2O(cond)	0.00						
H_2O(tot)	70.3	0.2	0.2	2744	18	1265	3470
O_2	17.8	0.1	0.1	369	32	570	210
N_2	179.8	0.6	0.6	419	28	5034	2108
Total	299	1	1.01			8234	6608

Table 5-3 80 °C flue gas composition – 1 mole basis

Flue gas 80 °C	n	%mol (= %vol)	P^* (bar)	h (kJ/kg)	M (g/mol)	m (g)	H (kJ)
CO_2	31	0.10	0.11	555	44	1364	757
H_2O(comb)	22.5	0.08	0.08	2650	18	405	1073
H_2O(fuel)	44.94	0.15	0.15	2648	18	809	2143
H_2O(hum)	2.83	0.009	0.010	2651	18	51	135
H_2O(cond)	0.00						
H_2O(tot)	70.3	0.2	0.2	2647	18	1265	3348
O_2	17.8	0.1	0.1	322	32	570	184
N_2	179.8	0.6	0.6	367	28	5034	1846
Total	299	1	1.01			8234	6134

Table 5-4 35 °C flue gas composition – 1 mole basis

Flue gas 35 °C	n	%mol (= %vol)	P^* (bar)	h (kJ/kg)	M (g/mol)	m (g)	H (kJ)
CO_2	31	0.13	0.13	515	44	1364	703
H_2O(comb)	22.5	0.09	0.09	147	18	405	59
H_2O(fuel)	44.94	0.19	0.19	147	18	809	119
H_2O(hum)	2.83	0.012	0.012	2566	18	51	131
H_2O(cond)	56.82	-	-	-	18	(1023)	-
H_2O(tot)	13.4	0.1	0.1	2565	18	242	621
O_2	17.8	0.1	0.1	280	32	570	160
N_2	179.8	0.7	0.8	320	28	5034	1610
Total	242	1	1.01			7211	3093

Figure 5-1, Figure 5-2, and Figure 5-3 show the temperature-enthalpy profiles for each of the three fuel types at differing O_2 flue gas percentages. The dew point occurs at the point of inflection on the graph, which decreases as the O_2 % is increased. It can also be seen that increasing the O_2 % decreases the available energy for heat recovery, the dew point is lower, meaning there is less water vapour per unit of flue gas flow. Additional fuel has been used to heat the additional air intake (and flue gas) that has not been transferred to create steam. Maintaining control of the excess air, such that the excess is air is not so low that incomplete combustion may occur and not so high as to penalise the boiler efficiency, is an important trade-off. For example, for hog fuel, if the flue gas is reduced to 50 °C at 10 % O_2, the available energy is approximately 200 kJ/kg$_{dry\ flue}$ but if the O_2 % is reduced to 4 % the available energy is more than doubled to 450 kJ/kg$_{dry\ flue}$. It is also apparent from the graphs that a higher proportion of the heat available is below the dew point where condensation occurs, and this latent heat can be utilised. Approximately 100 kJ/kg of enthalpy is released when the flue gas temperature is lowered to the dew point, compared to up to 600kJ/kg below the dew point.

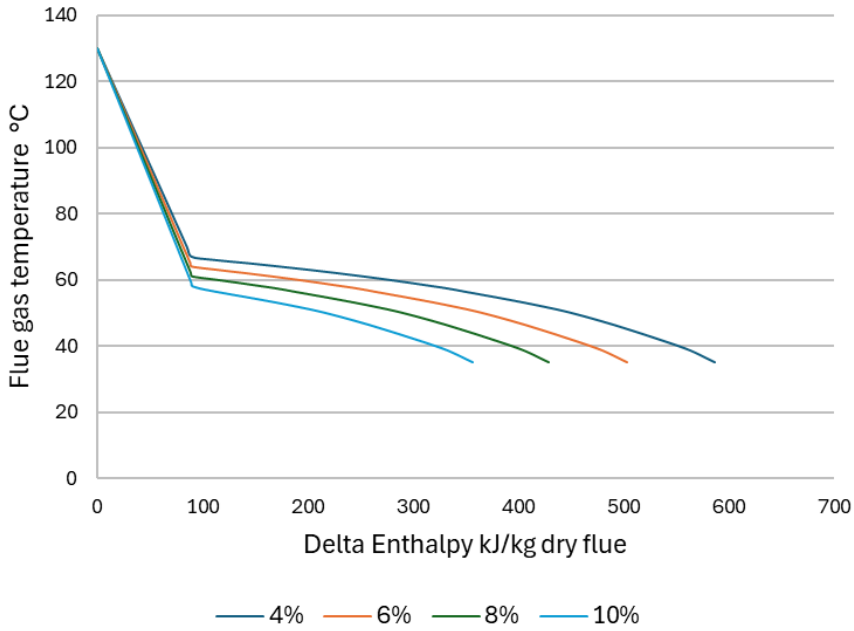


Figure 5-1 T-h graph of hog fuel at various flue gas O_2 %

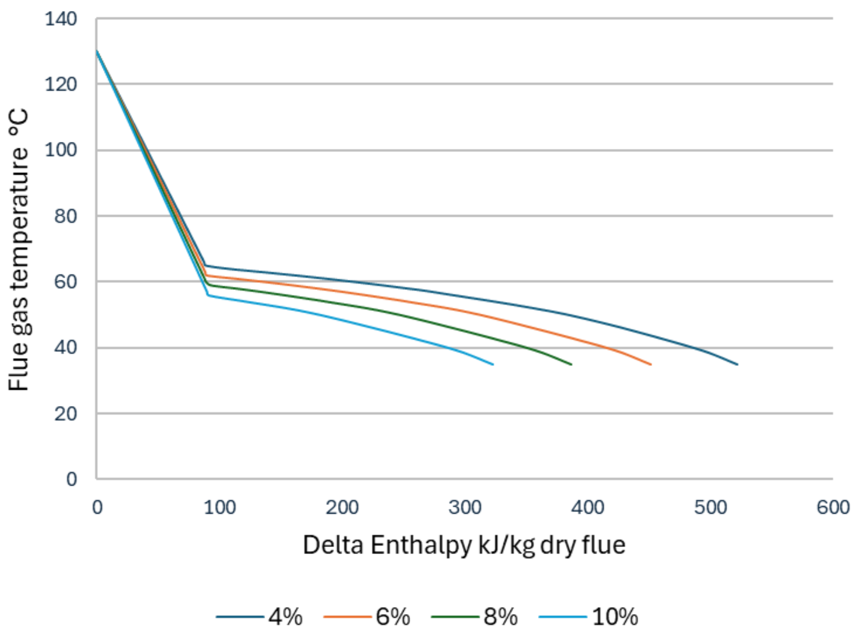


Figure 5-2 T-h graph of chip fuel at various flue gas O_2 %

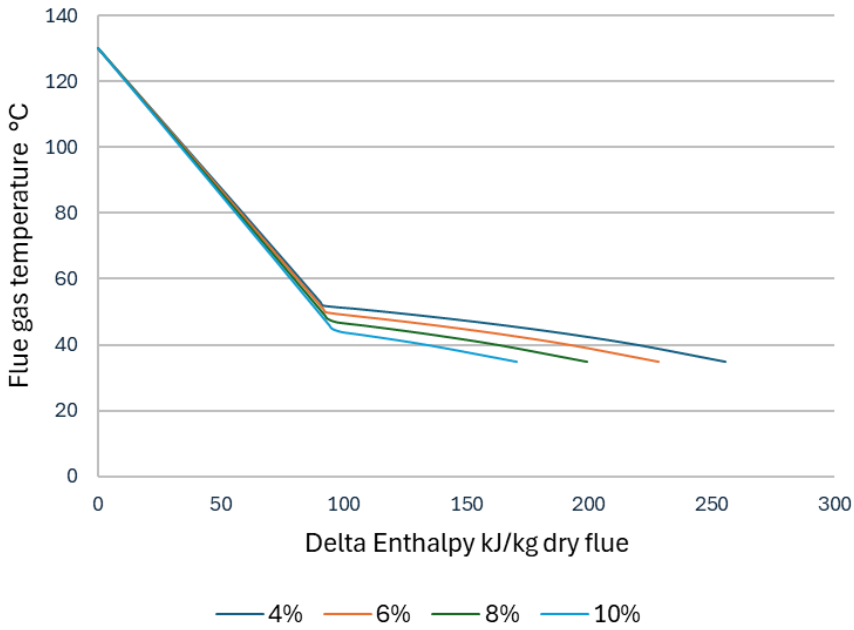


Figure 5-3 T-h graph of pellets at various flue gas O_2 %

Figure 5-4 also shows the change in enthalpy as the flue gas is lowered. This time, the O_2 % in the flue gas remains constant and the different fuel types are directly compared. Increased energy available for heat recovery was observed for the hog fuel which has the highest moisture content of the three fuels. The higher the moisture content, the higher the dew point and therefore, the greater the potential for heat recovery and flue gas from hog fuel will provide a better heat source for a heat pump.

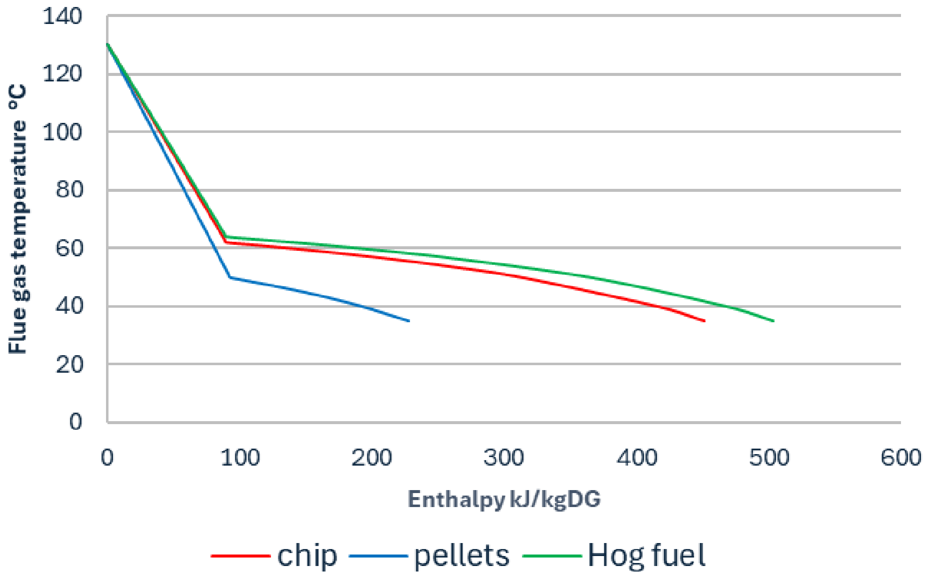


Figure 5-4 T-h graph of all three fuels at 6% O_2

5.4. Hot water generation using direct heat exchange

The major benefit of utilising the flue gas from a biomass boiler, compared to a fossil fuel boiler, is the latent heat available for recovery or upgrade due to the high moisture content in biomass fuel. As the flue gas is exhausted at 130 °C and it is desired to produce hot water at 80 °C or 60 °C, direct heat exchange should first be considered as it is the simplest and most cost-effective method for generating hot water. Table 5-5 below shows the amount of hot water produced using a DHX in the flue. These cases are calculated for heating water from 20 °C to 80 °C and 20 °C to 60 °C. For the 60 °C case, it was possible to cool the flue from 130 °C to 35 °C, utilising the full temperature range of the flue. However, for the 80 °C case, the heat exchange is constrained by the dew point, limiting the outlet temperature of the flue gas to 65 °C and therefore, significantly less hot water can be generated. Cooling the flue gas past the dew point (~64 °C) would result in infeasible heat exchange (temperature cross).

Table 5-5 Hot water generation from direct heat exchange using hog fuel

Hot water temperature	Amount generated
From 20 °C to 60 °C	85.5 t/h
From 20 °C to 80 °C	9.9 t/h

Table 5-6 shows the difference between taking the flue down to below the saturation temperature in the DHX, compared to using this heat in the basic cycle heat pump for the 60 °C hot water. The DHX takes the flue gas from the initial 130 °C to either 60 °C or 65 °C. Then the heat pump uses the remaining heat, lowering the flue gas down to a final temperature of 35 °C. A comparison of the total hot water produced indicates where the latent heat would best be utilised. In all cases, more hot water was generated using the latent heat in the flue gas in the heat pump compared to the DHX. Therefore, the flue gas temperature will be lowered to only 65 °C to generate 60 °C hot water directly, keeping it just above the dew point of 64 °C, under the conditions of hog fuel and 6% excess oxygen.

Table 5-6 Comparison of hot water produced

Flue temp out after DHX	HW prod in DHX (t/h)	HW inlet and outlet temperature			
		52-60 °C		20-60 °C	
		HW prod in HP (t/h)	**Total (t/h)**	HW prod in HP (t/h)	**Total (t/h)**
60 °C	32	342	**374**	69	**101**
65 °C	15	455	**470**	91	**106**

Although it is possible to utilise direct heat exchange for producing 60 °C hot water when cooling the flue gas from 130 °C to 35 °C, the temperature profile match between the flue gas and the hot water has potential for improvement. Additionally, substantial heat remains in the flue gas after generating 80 °C hot water directly, after which the temperature of the flue gas heat is too low to produce additional 80 °C or efficiently produce hot water at 60 °C. Therefore, accessing the latent heat of the flue gas has the potential to produce more hot water if a heat pump is utilised.

5.5. Conclusions

The mass flow of the fuel and the combustion air calculated from the model shows good agreement with the measured values from the case study. The results presented quantify the amount of energy available in the flue gas to recover. Hot water generation from direct heat exchange has also been calculated. The temperature profile match between the hot water and the flue gas in the direct heat exchanger is suboptimal as a result of the inflection at the dew point for the 80 °C case, therefore, negating the benefit of leveraging latent heat from the biomass. Consequently, the integration of a heat pump after direct heat exchange is proposed to upgrade the lower temperature heat from the flue gas (after the DHX) to produce additional hot water. The flue gas will be lowered to 80°C for 80°C hot water before utilisation in a heat pump. To make better use of the latent heat the flue gas will be lowered to 65 °C for the 60°C hot water before being used in the heat pump. The following chapter gives the heat pump analysis to quantify this.

Chapter 6
Heat pump analysis

6.1. Introduction

This Chapter will present and discuss the results of the Heat Pump model. The quality of the heat source has been calculated using the excel model and confirmed. The values of flow, temperature and enthalpy determined from the model will be used in the heat pump model described in Chapter 3 . The results will analyse the three different HP cycles considered in conjunction with the natural refrigerants identified in the literature review.

The results will be presented for the base model in order to compare the effects of adding subcooling and an internal heat exchanger. The refrigerant selection will be assessed. Several metrics will be analysed including COP, work in, hot water produced and VHC. Finally the specific work of hot water produced will be shown, indicating the most efficient way to generate the hot water in terms of energy input. The two different types of hot water required in the case study provide different sink temperatures. The impact of this will be assessed. The case study also has two distinct operating modes, heating from cold and recirculating to maintain the temperature.

6.2. Vapour compression cycle

Ammonia, one of the refrigerants from the options considered in Section 5.4.3 is presented to demonstrate the base heat pump cycle modelled. Figure 6-1 below shows the four stages of the heat pump cycle for the refrigerant ammonia plotted on temperature-entropy (T-s) and pressure-enthalpy (P-h) phase diagrams. The liquid and vapour saturation lines are shown in black, and the blue lines represent the cycle stages. The figures show the stages of compression (1 – 2), condensing (2 – 3), expansion (3 – 4), evaporation (4 – 1).

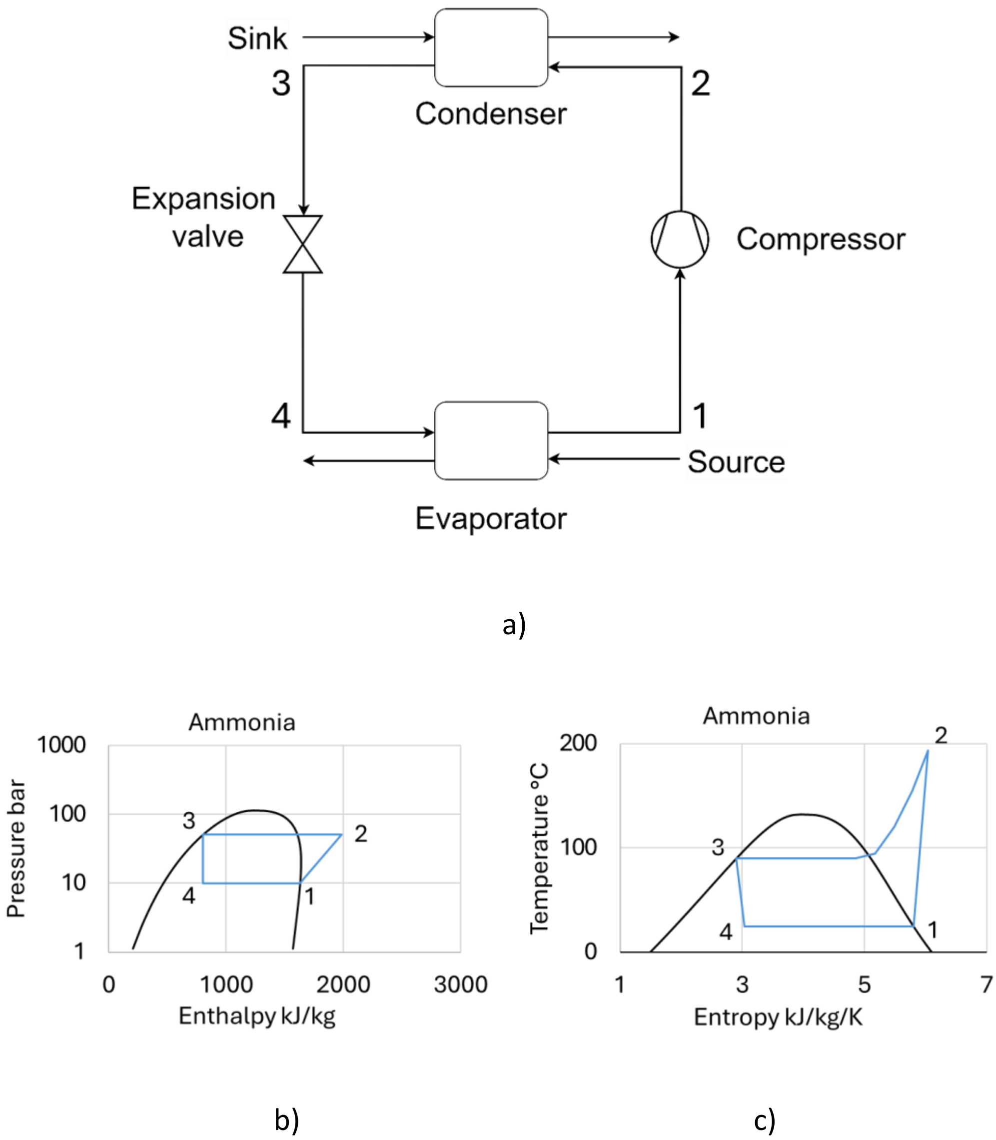


Figure 6-1 Diagrams displaying (a) heat pump base cycle, (b)P-h graph and (c) T-s graph for base cycle

Figure 6-2 shows the same refrigerant with subcooling. It can be seen in the condensing stage 2-3 in Figure 6-2 b) that the enthalpy is lowered beyond the saturation line. Similarly, in Figure 6-2 c) entropy is lower in the condenser stage 2-3 prior to expansion.

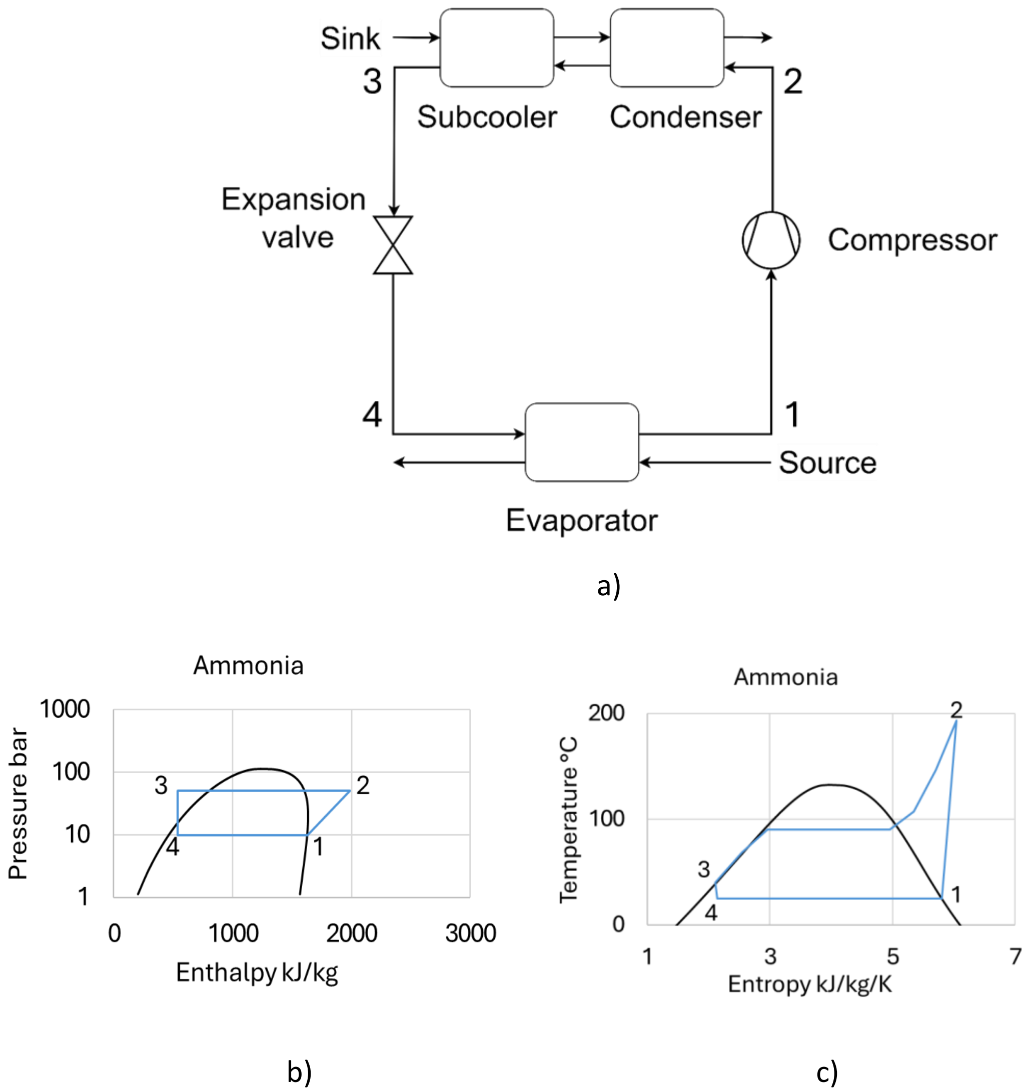


Figure 6-2 Diagrams displaying (a) heat pump sub cooled cycle, (b)P-h graph and (c) T-s graph for sub cooled cycle

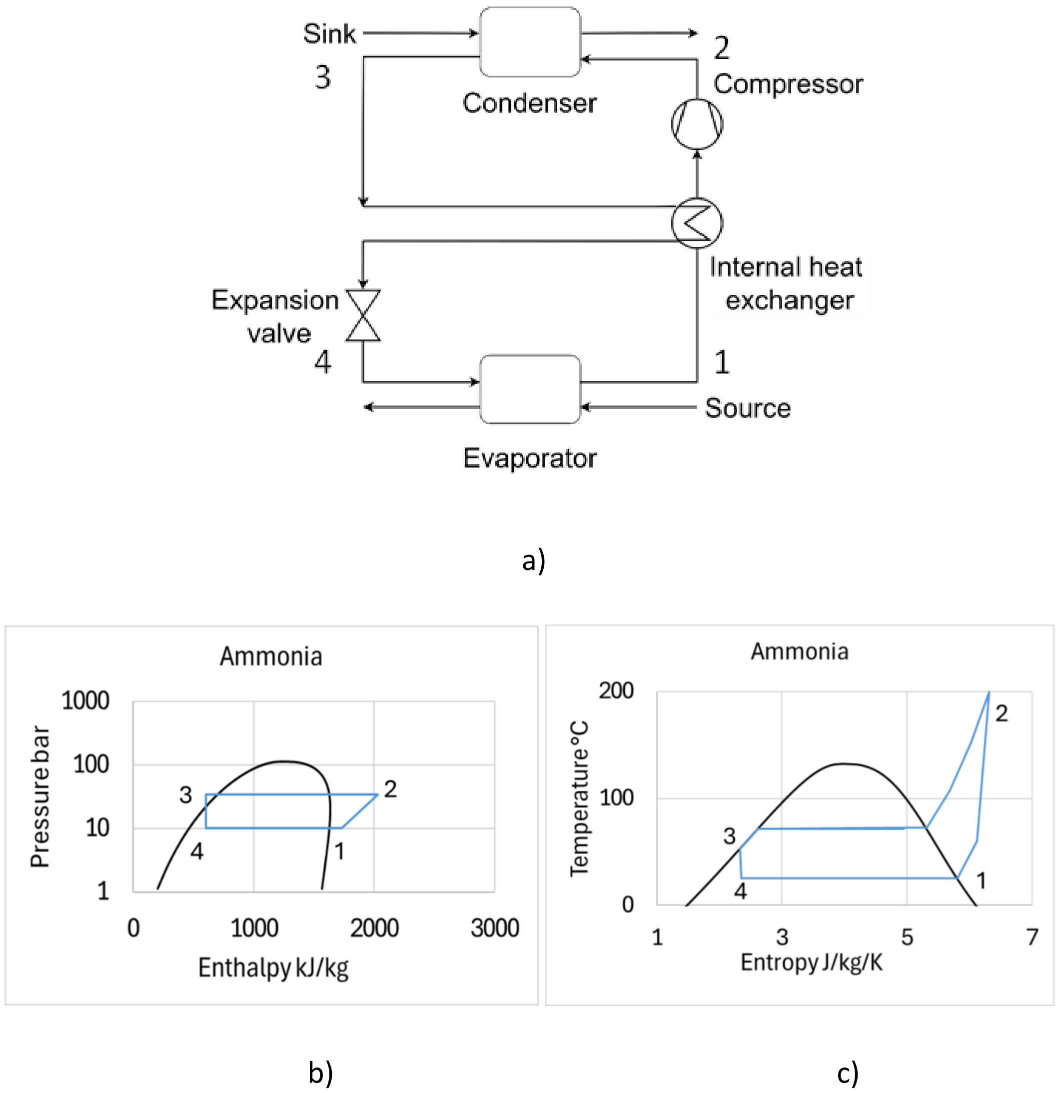


Figure 6-3 Diagrams displaying (a) heat pump with IHX cycle, (b)P-h graph and (c) T-s graph for IHX cycle

Figure 6-3 shows the same refrigerant with an IHX. It can be seen in the condensing stage 2-3 in Figure 6-3 b) that the enthalpy is lowered beyond the saturation line in a similar manner to subcooling, but in the evaporator stage 4-1 superheating can be seen as the line goes past the saturation line. Similarly, in Figure 6-3 c) entropy is lower in the condenser stage 2-3 prior to expansion and entropy and temperature increase prior to compression due to the IHX.

6.3. Refrigerant selection

COP and VHC are two important metrics that can be used to determine the most effective refrigerant. Both high COP and VHC are desirable. From Figure 6-4 it can be seen that ammonia, propane and cyclopropane are the most favourable refrigerants modelled from the fluids under consideration signalled in the literature review. These three refrigerants will be considered in more detail in the following sections.

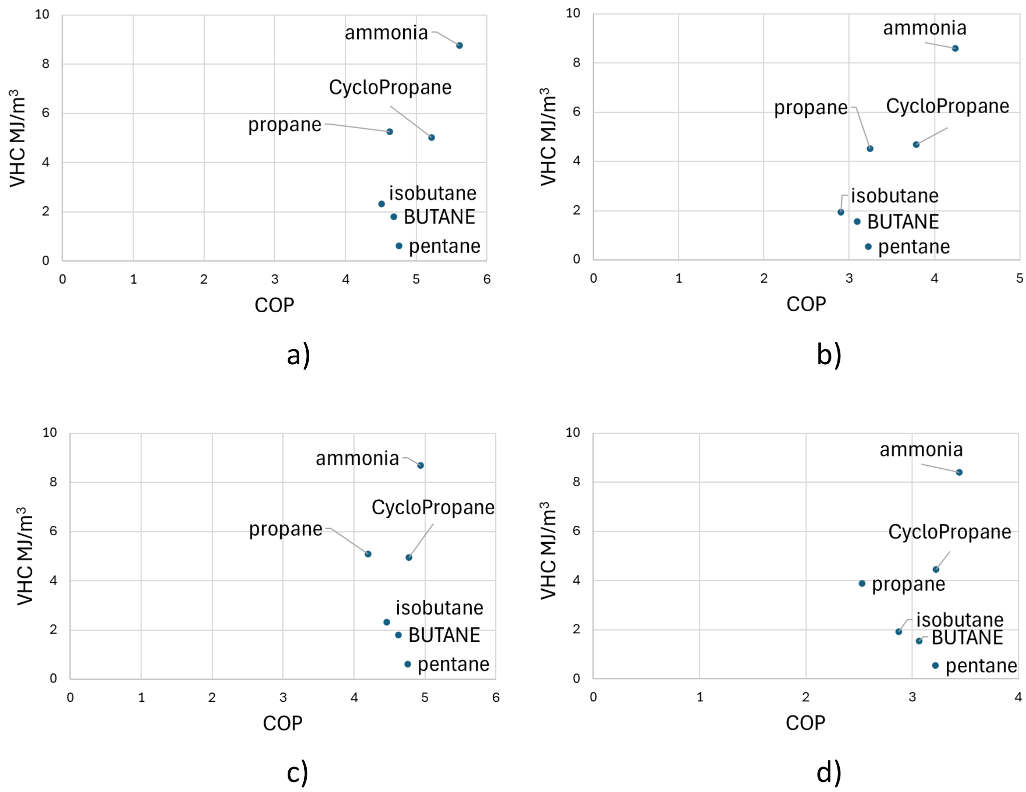


Figure 6-4 VHC vs COP base case a) 20-60 °C b) 20-80 °C c) 52-60 °C d) 72-80 °C

Figure 6-5 shows COP for the base heat pump cycle with the lowest flue gas outlet temperature and heating from cold water. Ammonia and cyclopropane have the highest COPs which supports the suggestion that ammonia and cyclopropane would be the most efficient. Pentane and propane are very close in COP for this cycle so combining this with VHC metric sets propane ahead.

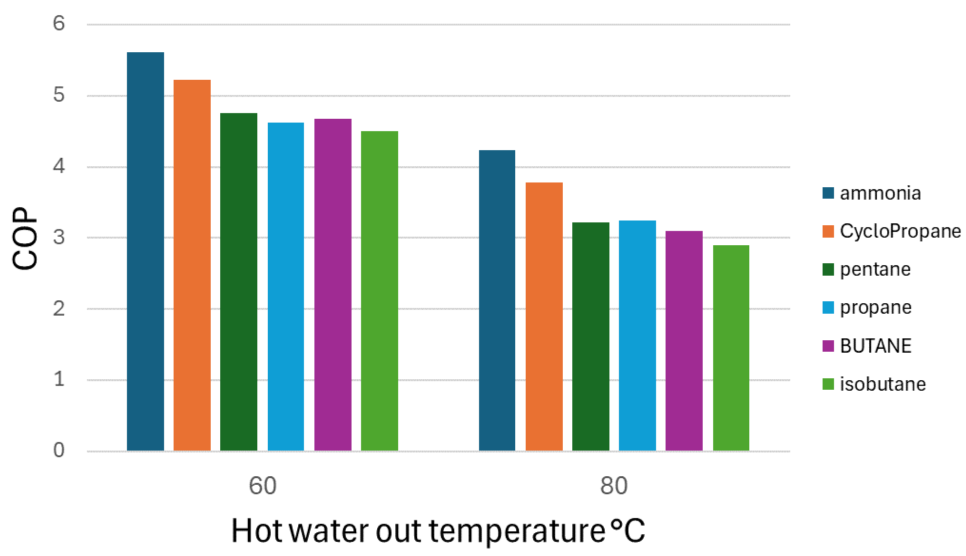


Figure 6-5 COP for single component fluids for 60 °C and 80 °C hot water (flue 35°C)

A major drawback to hydrocarbons is their flammability. As shown in the literature review hydrocarbons are highly flammable and safety systems which can be costly need to be implemented. Palm [22] indicated components suitable for hydrocarbon systems are available on the market.

6.4. Effect of a subcooler

This section discusses the results from the model and the effect that subcooling has on various metrics. As outlined in the literature review subcooling improves the efficiency of a heat pump cycle. The results presented agree with this, showing an improvement in the cycle efficiency.

The graphs below show the same three refrigerants chosen in the section above. It can clearly be seen that COP is improved considerably as subcooling is increased. Where the hot water is produced from cold the subcooling allowable is high. In the case of the 20 °C-80 °C hot water subcooling of up to 50 °C is available. This reduces to 30 °C for the 20 °C-60 °C hot water, while the recirculation only offers a 5 °C subcooling amount. This is limited by the ΔT_{min} of 10 °C and the inlet temperature of the hot water.

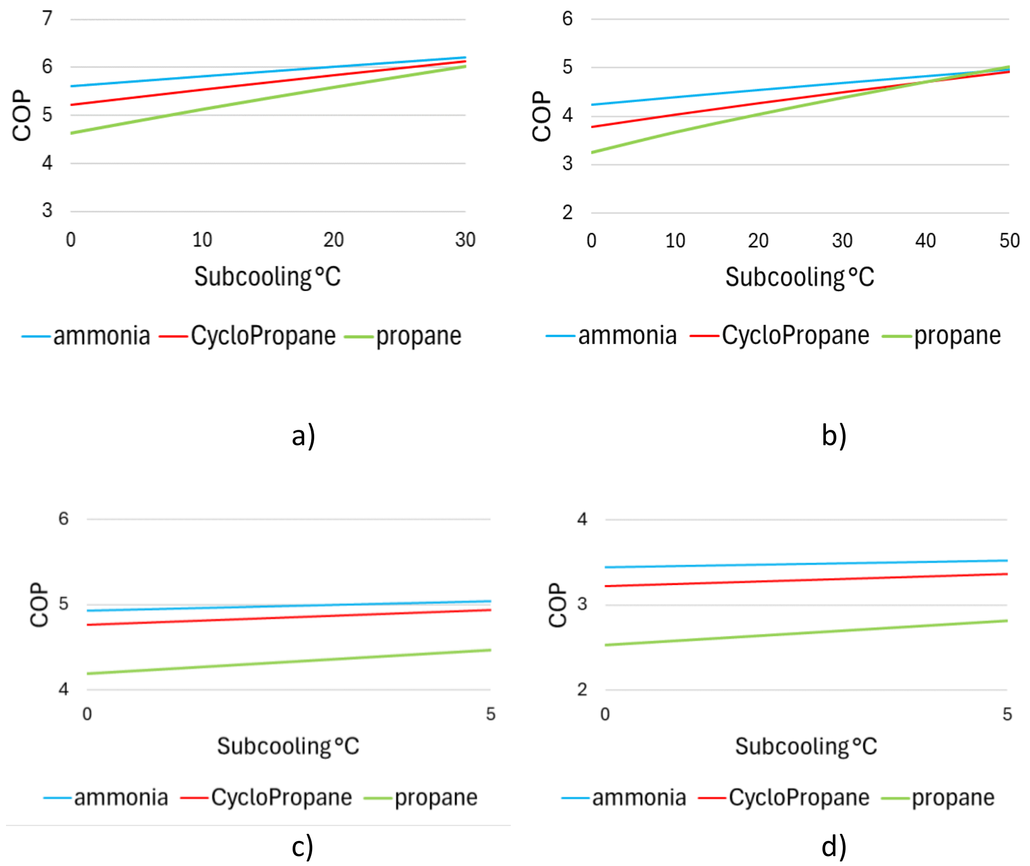


Figure 6-6 COP vs subcooling a) 20-60 °C b) 20-80 °C c) 52-60 °C d) 72-80 °C

From these graphs it can be concluded that subcooling greatly improves the efficiency of the heat pump cycle and should therefore be used and maximised. Supporting the use of a sub cooler is the specific work, subcooling greatly reduces this making better use of the energy in the flue gas. VHC increases with subcooling at higher levels of subcooling amount but is of little value for the recirculation which only allows 5 °C of subcooling.

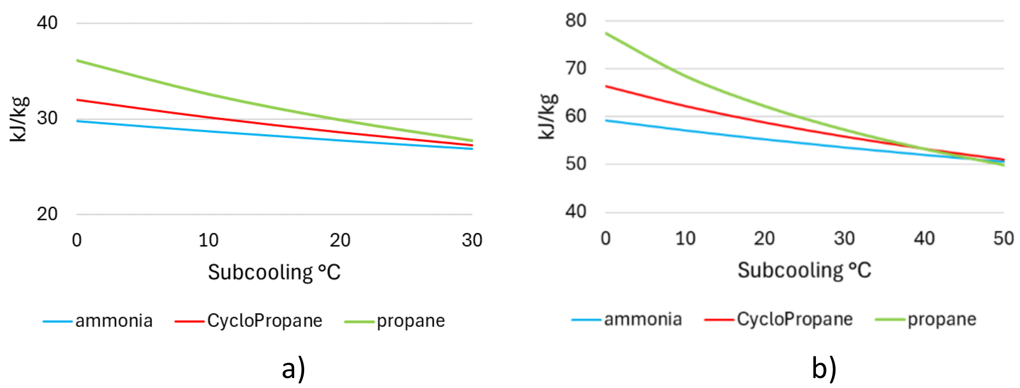


Figure 6-7 Specific work vs subcooling for a) 20 - 60 °C and b) 20 - 80 °C

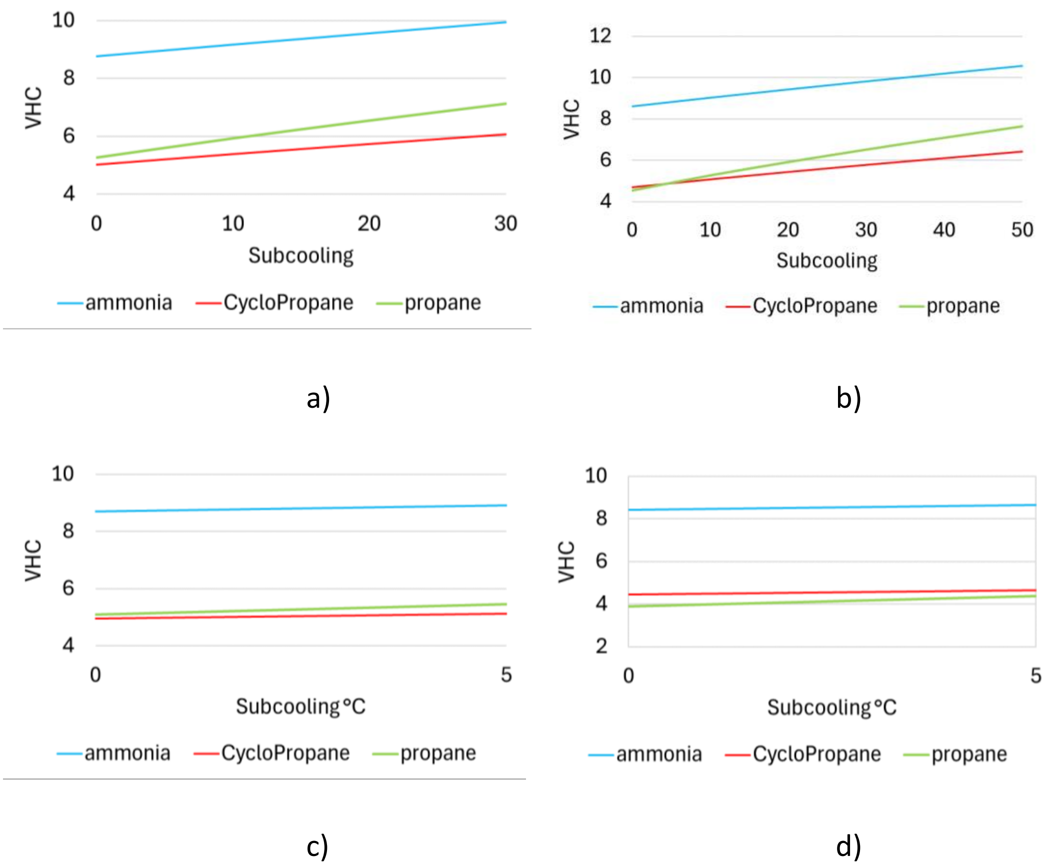


Figure 6-8 VHC vs subcooling a) 20-60 °C b) 20-80 °C c) 52-60 °C d) 72-80 °C

6.5. Effect of an internal heat exchanger

This section examines the effect an internal heat exchanger has on various parameters. As outlined in the literature review, an internal heat exchanger improves the efficiency of a heat pump cycle only marginally and this can be seen in the graphs in Figure 6-9. COPs for two of the three fluids increase only modestly. Propane shows a larger increase where the COP increases by approximately 55% in the 20 °C to 80 °C HW scenario and almost 30% in the 20 °C to 60 °C scenario. The major benefit of an internal heat exchanger comes from ensuring dry vapour into the compressor improving operational reliability. It may be beneficial to have a small internal heat exchanger increasing the temperature into the compressor by 5-10 degrees and leave the remaining heat for utilisation in the subcooler.

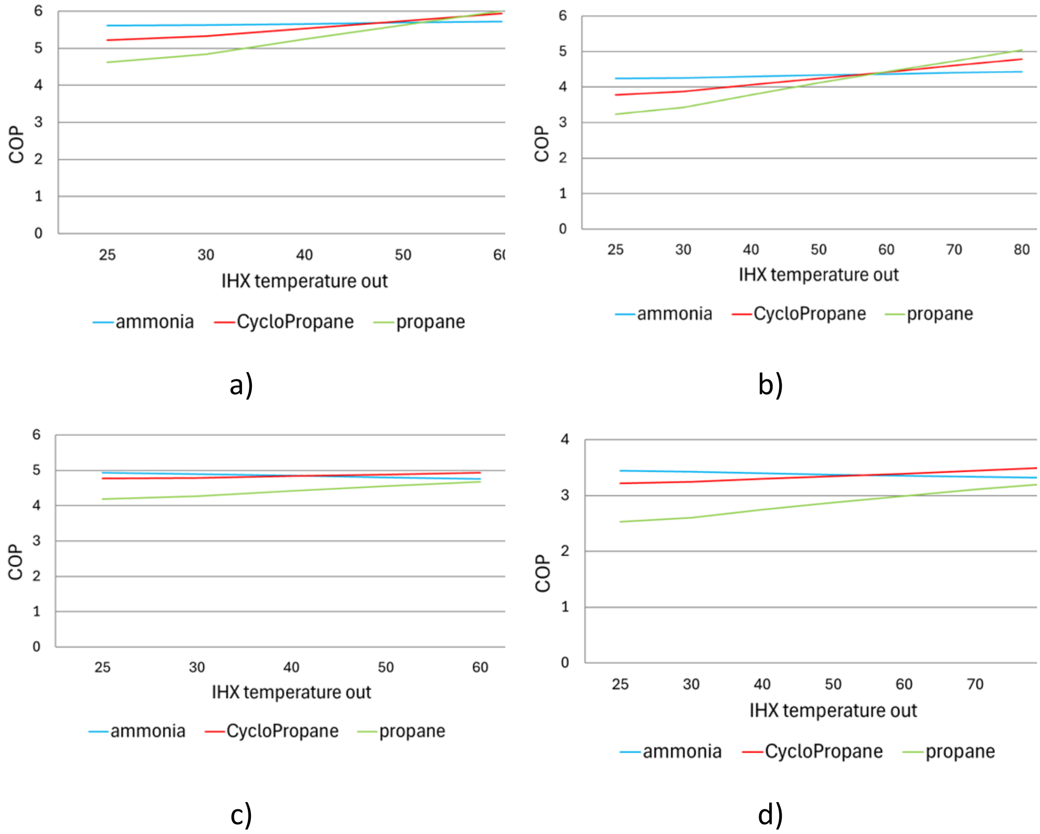


Figure 6-9 COP vs IHX temperature out a) 20-60 °C b) 20-80 °C c) 52-60 °C d) 72-80 °C

6.6. Effect of source outlet temperature

The source temperature was reduced to three levels 55 °C, 45 °C and 35 °C as described in the case study in Chapter 4 .

COP decreases as the flue gas temperature is decreased in all cases (Figure 6-10). Once the latent heat has been utilised the remainer of the heat becomes increasingly less efficient to release. Therefore, once the source temperature has gone below the dew point there is declining benefit to reducing further.

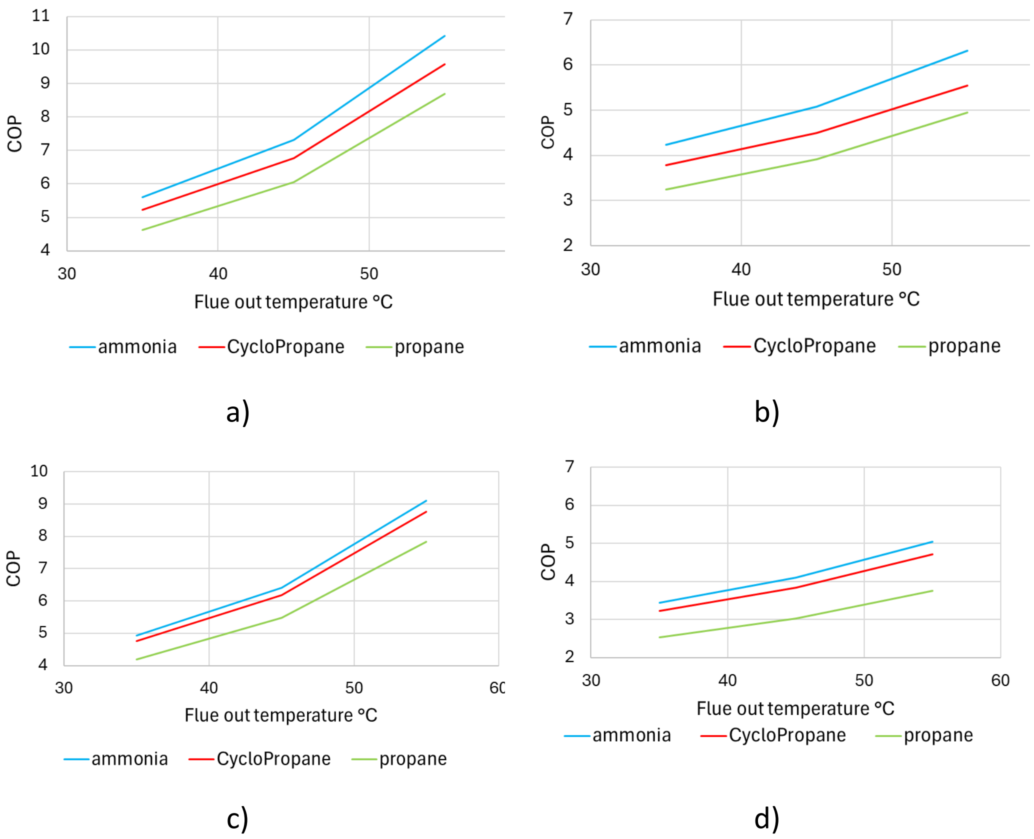
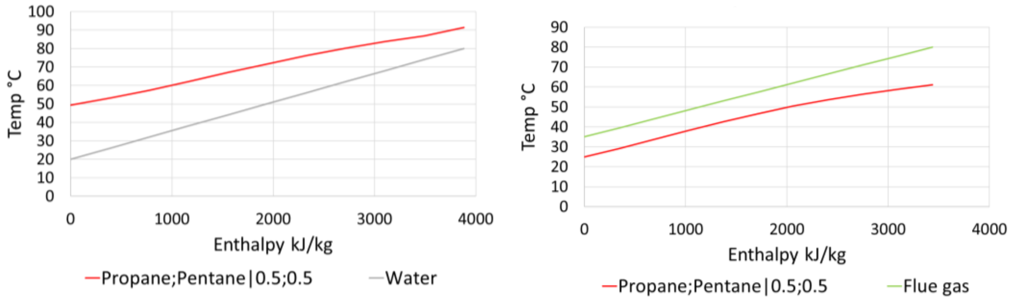


Figure 6-10 COP vs source temperature out a) 20-60 °C b) 20-80 °C c) 52-60 °C d) 72-80 °C

6.7. Effect of binary zeotropic refrigerant

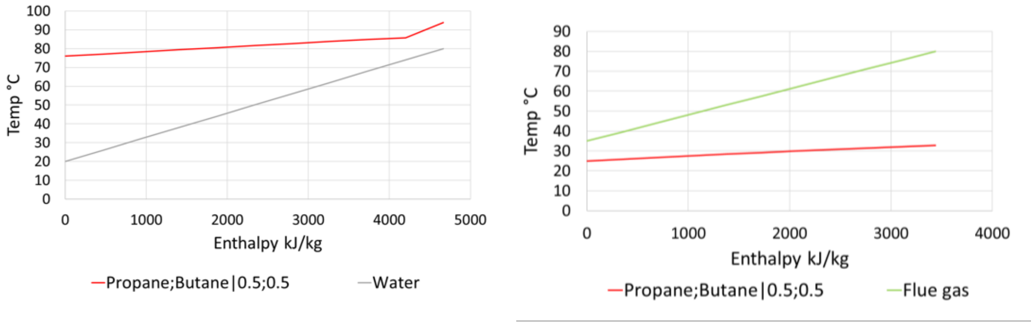
As discussed in the literature review, binary zeotropic refrigerants can provide improved performance in heat pumps by taking advantage of the temperature glide. Two mixtures were modelled.

Graphs in Figure 6-11 and Figure 6-12 show the T-h graphs of the evaporator and condenser and demonstrate the effect of using a mixture. The graphs are shown for the basic cycle for 20 °C to 80 °C hot water with the flue gas lowered to 35 °C The temperature glide can be seen in both sets of graphs where the refrigerant line (red) is sloped compared to the horizontal line of a single fluid. The temperature profile of the zeotropic refrigerant Propane:Pentane, Figure 6-11, very closely matches the temperature profile of the sink and source making heat transfer more efficient. In comparison the Propane:Butane mixture profiles do not match as well, the slope is not as steep, and this is reflected in the COP values shown in Figure 6-13.



a) b)

Figure 6-11 T-h graphs a) condenser and b) evaporator for Propane:Pentane



a) b)

Figure 6-12 T-h graphs a) condenser and b) evaporator for Propane:butane

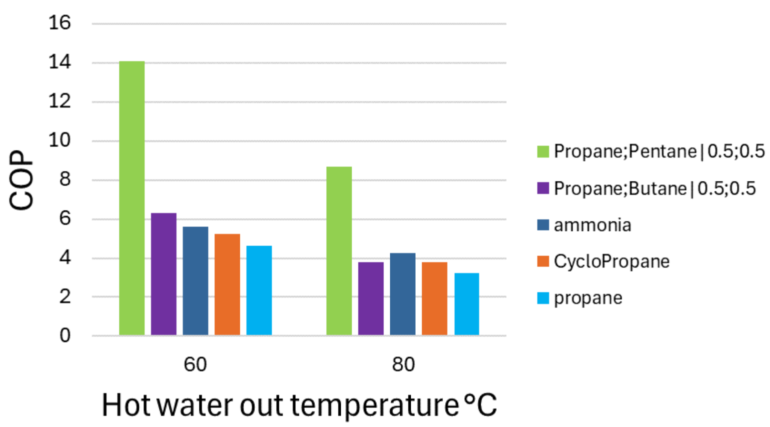


Figure 6-13 COP for single and zeotropic fluids base case 35 °C flue out temperature

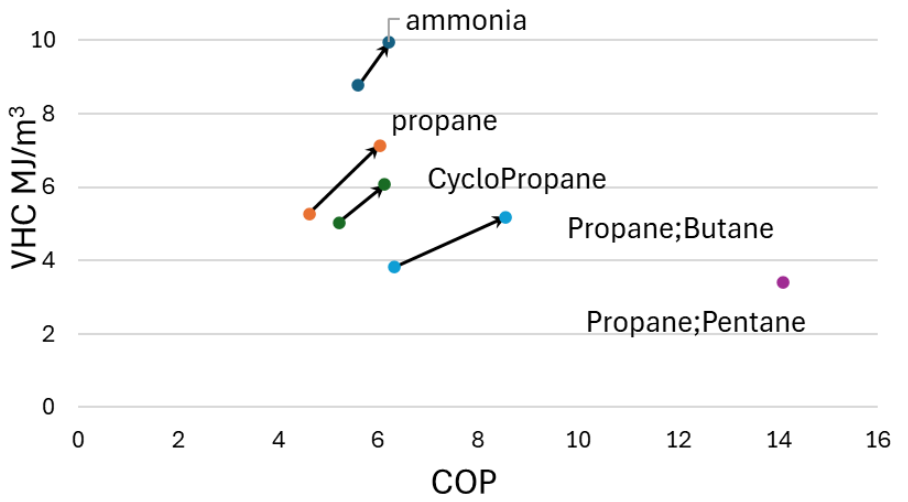


Figure 6-14 VHC vs COP comparison of base case and maximum subcooling 20-60 °C

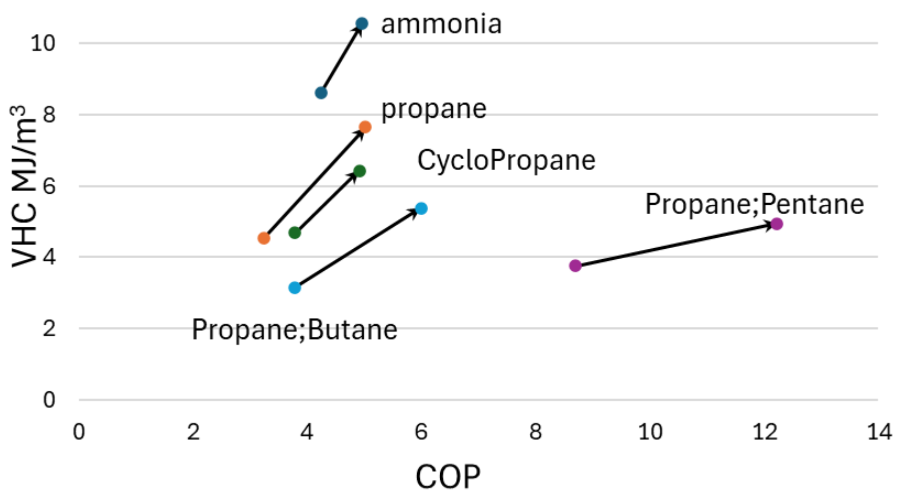


Figure 6-15 VHC vs COP comparison of base case and maximum subcooling 20-80 °C

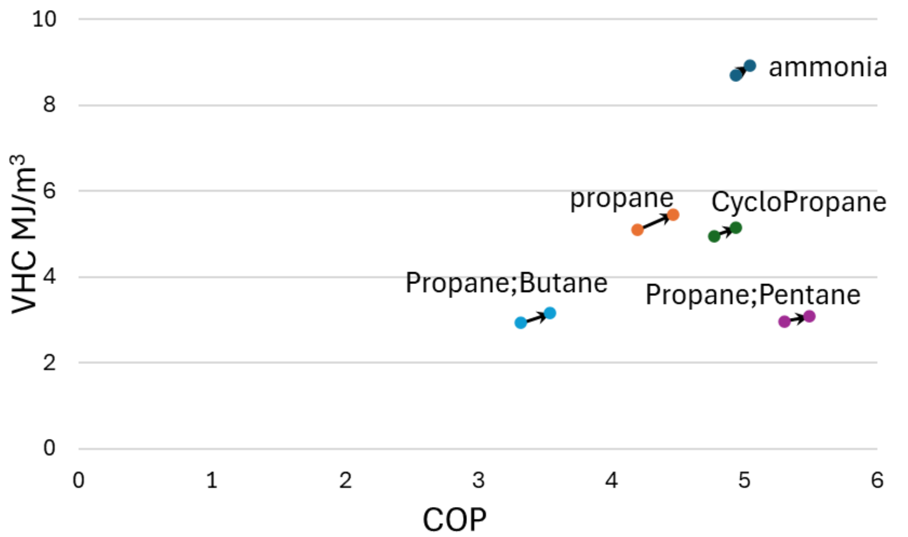


Figure 6-16 VHC vs COP comparison of base case and maximum subcooling 52-60 °C

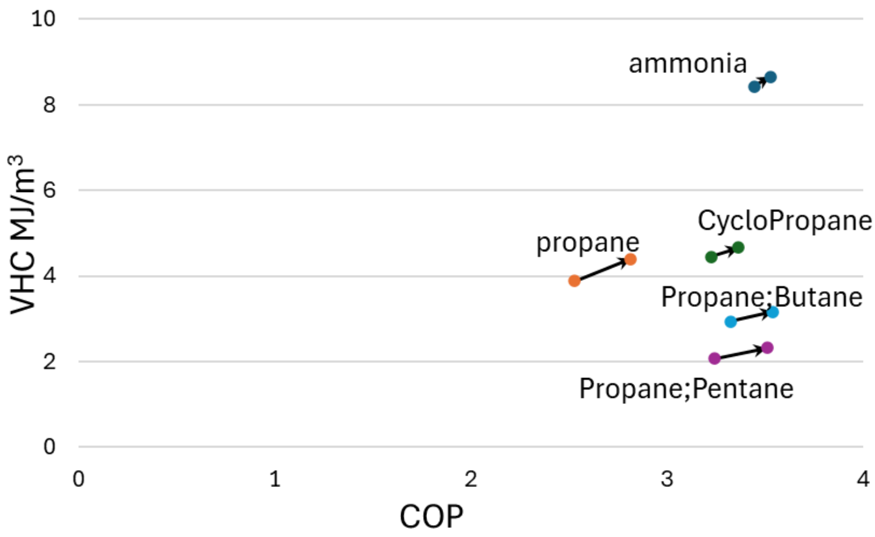


Figure 6-17 VHC vs COP comparison of base case and maximum subcooling 72-80 °C

The three most promising single component refrigerants were compared to the two mixtures. COP-VHC graphs are plotted showing the increase from the base case to a higher COP, and VHC, when maximum subcooling is used (Figure 6-14,Figure 6-15,Figure 6-16 andFigure 6-17). As can be seen, when heating from cold, which is a larger temperature lift, the propane pentane mixture gives a large increase in COP and will be the best refrigerant in this case. Conversely when the temperature lift is less, under recirculation conditions, ammonia is indicated as the best option. With the recirculation available subcooling is much smaller and so this translates into smaller COP ad VHC gains.

Comparing the zeotropic mixtures, propane pentane has the better COP for the basic cycle compared to the propane butane blend. This is due to the more closely matched temperature glide.

6.8. COP versus hot water generated

In Section 6.6. it was shown that lowering the evaporator temperature, i.e. lowering the flue gas outlet temperature, lowered the COP. The graphs in Figure 6-18 show how a higher COP delivers less hot water, or conversely a lower cop delivers more hot water. This is a trade-off which needs to be made. If more hot water generation is needed a sacrifice in COP is required. With ammonia as the refrigerant, all the cases resulted in a COP greater than three, often viewed as a minimum required COP for further consideration and design.

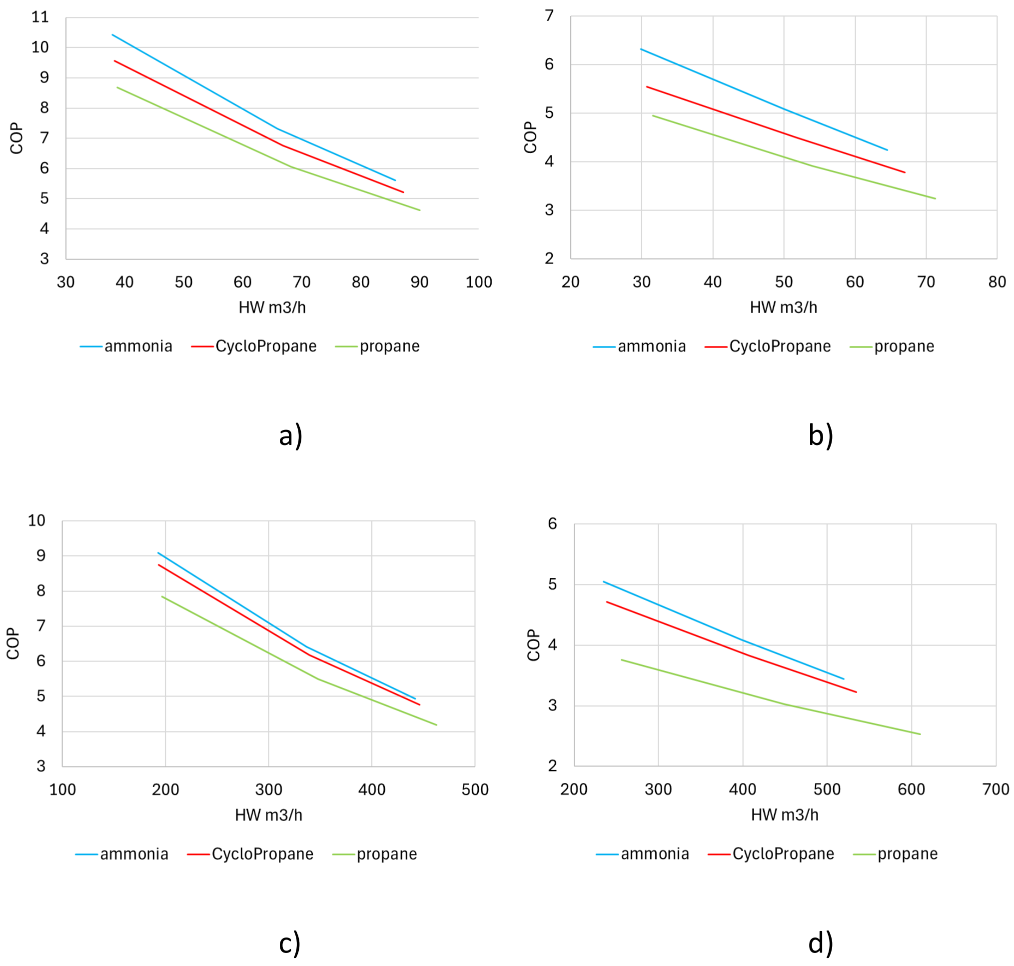


Figure 6-18 COP vs hot water generation a) 20-60 °C b) 20-80 °C c) 52-60 °C d) 72-80 °C

6.9. Conclusions

The values of flow, temperature and enthalpy of the flue gas, determined in Chapter 5 have been used in the base heat pump model. The three most favourable refrigerants have been inputted to the subcooled and IHX heat pump cycles, to determine the benefits. While the basic cycle will produce hot water, subcooling greatly improves the efficiency and output of the heat pump. This was an expected result based on the review undertaken in Chapter 2 .

An internal heat exchanger has a modest to good influence on the parameters but the advantage comes from the improved reliability. Keeping the fluid dry prior to entering the compressor reduces damage and therefore maintenance. However, it may be that the cost of adding a heat exchanger outweighs this benefit.

The flue gas should be lowered to below the dew point, but reducing significantly further than this decreases the COP. The lower the source temperature the more hot water can be generated but less efficiently. This is a trade-off that needs to be made.

From the results presented above it can be seen that a heat pump is an efficient way to upgrade a waste energy stream to produce hot water. Subcooling should be maximised and an internal IHX considered. There are different options for the choice of refrigerant; ammonia is a good performer, along with the mixture of propane and pentane, to take advantage of the temperature glide. Refrigerant choice will be influenced by the temperature lift required for the source. For a large lift the propane pentane mix is superior, but for a smaller lift, in the case of recirculating, ammonia is preferable.

Chapter 7
Conclusions and recommendations for future work

7.1. Conclusions

Utilising the flue gas of biomass boilers for hot water production has been shown as a promising way to recover and upgrade the currently wasted heat. The latent heat from the moisture in the fuel contributes to this waste heat. Combustion has been modelled and the results compared favourably to an industrial site case study. Temperature and enthalpy profiles of the flue gas have been determined. After direct heat exchange the remaining heat has been used as a source for a vapour compression heat pump. Refrigerants with low environmental impact and appropriate thermodynamic properties have been considered, including blends to take advantage of temperature glide during phase change.

The mass flow of the fuel and air flow calculated in the model, for a given duty, flue gas O_2% and fuel type, has been compared to the industrial site data with good agreement. The results showed increasing the flue gas O_2% decreases the available energy, but there is a requirement for excess air to ensure complete combustion. The results also indicated more heat is available in the flue gas from the fuel with a higher moisture content.

Using the values of flow, temperature and enthalpy of the flue gas, determined in the model, hot water generation from direct heat exchange has been calculated, followed by integration of a heat pump. Using a basic heat pump cycle, the three best refrigerants were identified using VHC and COP, and investigated further. A basic cycle would be improved significantly when adding a subcooler or an internal heat exchanger.

The flue gas should be lowered to below the dew point to take advantage of the latent heat, but reducing significantly further than the dew point decreases the

COP. The trade-off is the more the source temperature is lowered, the more hot water is generated, but less efficiently. A large water temperature lift (>45 degC) favours the propane-pentane mix, but for a smaller lift (~10 degC), in the case of recirculating, ammonia is preferable.

A heat pump is an efficient way to upgrade a waste energy stream to produce hot water particularly if subcooling and IHX are employed. Ammonia is a good performer, along with the mixture of propane and pentane, to take advantage of the temperature glide.

7.2. Recommendations for future work

Below are three areas that warrant further investigation:

1. Further research should include investigating minimising the charge. This is the amount of refrigerant in the heat pump. Due to the flammable nature of the refrigerants the consequence of any leaks can be minimised if the charge amount is also at a minimum.

2. Flue gas can contain SO_2 and SO_3 formed during combustion of sulphur in the fuel. The dew point of this depends on a number of factors including the fraction of sulphur in the fuel and excess air. Temperatures reported in one study indicated an acid dew point range of 142 °C to 173 °C [35]. The temperature of the flue gas exiting, in the case study, is already below this. There are materials which can handle this type of corrosive material but they can be costly and there may be an optimum temperature that will minimise the effect.

3. While mixtures in theory indicate a much higher COP three are several design considerations which need to be considered. Compressors for a specific blend may not be available. Control in the condenser would need to be carefully managed and this may be difficult, preventing the higher COPs being realised.

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Appendix A
Lists of figures and tables

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