THE DEVELOPMENT OF A HYDROCARBON HIGH TEMPERATURE HEAT PUMP FOR WASTE HEAT RECOVERY

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Abstract

Waste heat is an abundant resource that if recovered with a heat pump would increase energy efficiency in industrial processes. This will provide improvements in heat utilization and reduce the environmental impact of greenhouse gas emissions from the combustion of fossil fuel. A hydrocarbon high temperature heat pump has been developed to demonstrate the potential to deliver heat at a temperature of 115 ^{o}C. The heat pump provides heat for applications such as drying, pasteurization and other processes. Using hydrocarbons, which are natural working fluid, the heat pump aims for a clean energy system.

This paper reports on a 20 kW capacity cascade heat pump with propane in the low temperature cycle and butane in the high temperature cycle. Based on a theoretical model, an experimental setup is built with standard components that are commercially available. A prototype compressor is investigated for its performance at high temperature conditions. The heat pump can recover waste heat at 30 ^{o}C and deliver heat up to 115 ^{o}C. With an average heating COP of 3.1 for a temperature lift of 58 – 72 K, the heat pump is a more efficient and environmentally friendly system compared to existing solutions of a steam boiler.

Keywords

Energy efficiency, Natural working fluids, Compressor technology, Industrial heating, Process heat

1. Introduction

The focus on energy and environmental sustainability has increased in the last decades. Processes in industrial applications are moving towards a more integrated use of energy, especially for heating demands. Integrating heat usage will increase the heat utilization efficiencies and reduce the demand on external energy sources for the industry. This will save cost and better manage the dependencies on fluctuating (both in price and supply) external energy supply. It will also reduce the impact of greenhouse gas emissions, by the increase of energy efficiency, and the use of heating systems with zero emissions footprint.

Waste heat is an abundant energy resource that is largely unused and is discharged to the environment in many industrial processes (Johnson et al., 2008). It is a by-product of heating processes within an industry. The temperature of the waste heat is lower than the heating demand requirement of the processes and it is therefore regarded as waste. The waste heat is available in large quantities with a wide range of temperature distribution depending on the industry. The recovery of waste heat is a popular research domain and the implemented technology partly depends on the temperature and the process application.

A heat pump is an energy system that transfers heat from a cold reservoir to a warmer reservoir using an external power. The technology is widely used in refrigeration systems for cooling and in HVAC

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(heating, ventilation, and air conditioning) systems for space heating, cooling and hot water production. A high temperature heat pump (HTHP) applies the same technology but at a high temperature. What constitutes ‘high temperature’ is defined loosely across literature and is considered as heat delivery above 100 ^{o}C by a heat pump in this paper.

The development of heat pumps that can deliver higher temperatures has progressed over the years. Market analysis done by Nellissen and Wolf (2014) had suggested that within the temperature range of 80 – 150 ^{o}C, reachable by HTHPs, there exist a signification amount of heat demand in the chemical, paper, food, wood and other industrial sectors. Applications such as pasteurization, drying, distillation, sterilization, pressurized hot water production and other industrial processes will benefit from the development of a HTHP.

The thermodynamic properties of most fluids used in refrigeration systems and low temperature heating are not suitable for high temperature applications. They are typically constrained by the required compressor technology. Some synthetic fluids such as hydrofluorocarbons (HFCs), with possible suitable properties, have been restricted by the Kigali amendment to the Montreal protocol (Clark & Wagner, 2016). These synthetic fluids have non-zero ozone depletion potential (ODP) or high global warming potential (GWP). Other synthetic fluids such as the unsaturated hydrofluorocarbons (HFO), have potentially toxic decomposition products and possible unknown negative effects to the environment. Therefore, the future of high temperature heat pump development will focus on natural working fluids such as hydrocarbons, carbon dioxide, water and ammonia. These natural working fluids have zero ODP and minimal GWP.

The synthetic fluids HFC R245fa, R134a and their mixture, which are being phased out, had been used for HTHP development by several research and industrially implemented with different heat sink outlet temperature (Assaf et al., 2010; Blesl et al., 2014; Combitherm, 2017; Huang et al., 2017; IEA, 2014; Nilsson et al., 2017; Ochsner, 2017; Oue & Okada, 2013; Wu et al., 2015). Others have theoretically or experimentally investigated the use of the so-called hydrofluoroolefin (HFO), which is an unsaturated HFC for HTHPs (Blesl et al., 2014; Fukuda et al., 2014; Helminger et al., 2016; Kondou & Koyama, 2015; Ma et al., 2010; Nilsson et al., 2017; S. J. Zhang et al., 2010). While the saturated HFCs are clearly phased-out by the Kigali amendment, questions persist for unsaturated HFCs with low GWP, regarding their production process and stability to be considered as environmentally friendly working fluids.

Natural fluids are a more promising alternative for the future development of HTHP with regards to the environment. Recent research have also shown that they have similar or better performance when developed with the right configuration (Bamigbetan et al., 2018). Heat pumps using ammonia as the working fluid have been designed and installed for district heating. These heat pump delivers heat up to 90 ^{o}C using the vapour – compression technology (Ayub, 2016; Dietrich & Fredrich, 2012; Hoffmann & Pearson, 2011). For higher temperatures (above 100 ^{o}C), the thermodynamic properties of ammonia will require an ammonia compressor capable of 60 bar condensation pressure and up to 190 ^{o}C compressor discharge temperature, which is not commercially available.

Many heat pump technology with natural working fluids for high temperatures (> 90 ^{o}C) are however in different phases of technological readiness in simulations and experiments. Using CO_2 as the working fluid, Neksa et al. (1998), Neksa (2002) and others have demonstrated the use of CO_2 for HTHP. Commercially available products with CO_2 as working fluid have been presented with temperatures up to 120 ^{o}C (IEA, 2014). Eikevik et al. (2004) and Eikevik et al. (2005) investigated using CO_2 HTHP for the drying of fish, fruits, vegetables and dairy products. White et al. (2002) had developed a CO_2 heat pump prototype for up to 65 ^{o}C heat delivery. They used the experimental results to simulate a heat pump with an outlet temperature of 120 ^{o}C. This heat pumps are generally for high heat sink temperature difference processes such as hot water production, and air heating. The thermodynamic properties of

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CO_2 with low critical temperature will not have a high efficiency for applications with low heat sink temperature difference of about 20 K. This limits the processes and application of CO_2 for HTHPs.

Eisa et al. (1986) had investigated the suitability of water as a working fluid for HTHP. Using centrifugal vapour compressor, Madsboell et al. (2014), Larminat et al. (2014), Bantle et al. (2015), Tolstorebrov et al. (2014), Cooper and Lyon (2004) and others have investigated water as a working fluid for HTHP. The HTHPs were design to operate with a heat source temperature close to 100 ^{o}C by applying the mechanical vapour re-compression (MVR) cycle. Others such as Yang et al. (2016) had used a single screw compressor. With the use of high-speed centrifugal compressors for the volumetric requirements of water as a working fluid, it can be a good HTHP fluid for up to 150 ^{o}C and above. However, low waste heat temperatures of 30 ^{o}C will potentially be a challenge for water as the suction pressure will be below 0.05 bar. High vapour flow with a centrifugal compressor implies a low temperature lift in multiple stages. An intercooling system will also be needed for the high compressor discharge temperature.

The hybrid absorption compression heat pump (HACHP) is an alternative technology investigated for high temperature heat delivery. Several authors had investigated the technical and economic feasibility for its use (Hewitt et al., 2001; Jensen et al., 2015b). Jensen et al. (2015a) and M. Kim et al. (2010) had developed theoretical models for the performance evaluation of HACHP for HTHP, while J. Kim et al. (2013) and Infante Ferreira et al. (2006) had conducted experimental investigations. Reported performances for the HACHP cycle shows suitability for applications with a matching temperature glide of the ammonia/water mixture. At other temperature differences in heat sink and source, the cycle design is not optimized.

The use of hydrocarbons for HTHP have had an increased interest in recent years. Hydrocarbons have similar thermodynamic properties to synthetic working fluids, both in operating parameters and performance. There are also years of experience with the use of hydrocarbons as a working fluid with propane (R290) and propene (R1270). With the recent advances in compressor technology, there is a potential for hydrocarbons to be used as a high temperature working fluid. Theoretical analysis for temperatures up to 115 ^{o}C had been reported by Stavset et al. (2014), Bamigbetan et al. (2016) and Bamigbetan et al. (2017b). Wemmers et al. (2017) reported on an experimental setup for low pressure steam production using a single cycle with R600 as refrigerant for waste heat temperature of 60 ^{o}C. Similarly Moisi et al. (2017) and Yamazaki and Kubo (1985) also reported on a single cycle with R600 and R601 respectively. Both systems are applicable for higher waste heat temperature (above 60 ^{o}C) but not low waste heat temperature of 30 ^{o}C as a single cycle will neither be sufficient nor efficient for the high temperature lift as reported by Bamigbetan et al. (2016).

In the study by Bamigbetan et al. (2016), the theoretically results showed that a hydrocarbon mixture of butane and pentane in a cascade configuration heat pump is the most suitable system for a HTHP within the temperature range considered and among natural working fluids. However, the development of a new system will require gradual advancement of technology to minimize the possible point of failure. For this reason, the base case design from the paper by Bamigbetan et al. (2016), which describes a propane – butane cascade configuration is considered a simple heat pump to evaluate the potential heat delivery at high temperature. The experimental setup in this investigation is therefore based on the design and fluid selection. In addition, the comparative advantage of butane for HTHP was theoretically investigated by Bamigbetan et al. (2018). The analysis showed that butane is the most suitable pure natural working fluid for the immediate implementation of a HTHP based on existing compressor technology with suitable modifications. Experimental evaluations not covered in this investigation, for mixtures and other pure fluids such as pentane which may potentially yield better results for the specific operating conditions, are proposed for further work.

In general, several authors have reviewed the challenges, applications, suitable technology and working fluids for high temperature heat pumps (Arpagaus et al., 2017; Arpagaus et al., 2018; Bamigbetan et

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al., 2017a; Chua et al., 2010; J. Zhang et al., 2016). This present study summarized the development of a hydrocarbon HTHP for heat deliver up to 115 ^{o}C from the theoretical conception to the experimental results. The heat pump is utilizing waste heat as low as 30 ^{o}C making a temperature lift (Heat source inlet to heat sink outlet) of 85 K which is not reported in literature. The investigation consists of simulations and experimental results of a fully functional HTHP. The heat pump is a 20 kW heating capacity cascade configuration cycle with propane in the low temperature cycle (LTC) and butane in the high temperature cycle (HTC). The use of propane and butane, which are natural fluids, establishes the technology as a solution for a green future at a temperature range typically operated with synthetic fluids. A prototype butane compressor has been modified for high temperature operation. The cascade configuration design of the heat pump allows evaluations of both waste heat recovery with temperature at 25 – 35 ^{o}C (full cascade cycle) and 55 – 65 ^{o}C (butane HTC).

2. Modelling and Design Specifications

The approach to the development of the HTHP involves theoretical calculations to evaluate the limitations and possibilities of the technology and the implementation in a physical model. From the theoretical analysis, it is seen that for HTHPs, the compressor is the only technological limitation as other components are commercially available. The methodology therefore involves focus on a prototype compressor, its parameters and the performance for high temperature heat delivery.

2.1. The theoretical model

The theoretical model was developed using the Dynamic Modelling Laboratory (Dymola) 2016 (Dassault Systems, Vélizy-Villacoublay, France) and TIL library for modelling thermal systems TIL 3.3.1 (TLK-Thermo GmbH, Braunschweig, Germany). The model consisted of two heat pump cycles connected by a cascade heat exchanger which functions as the evaporator for the HTC and condenser for the LTC. Each cycle is made up of a compressor, a condenser, a high pressure liquid receiver (HPR), an expansion valve, an evaporator and a suction accumulator. Two controllers are set in each cycle to regulate the expansion valve to have a defined superheat at the evaporator and to regulate the compressor speeds to have a defined heat sink outlet temperature. The heat sink and source are modelled as pressurized water streams with the inlet and outlet states points set as thermodynamic boundaries. Fluid thermodynamic properties were extracted from the REFPROP database developed by the National Institute of Standards and Technology (NIST) (Lemmon et al., 2013). Fig. 1 shows the theoretical model cycle.

Assumptions were made for the theoretical model with regards to piping: zero pressure drop, zero heat losses or gains, zero fluid mass in piping volume. The compression process is assumed to operate without heat losses, lubrication effect on fluid solubility or fouling in heat exchangers. There are no electrical or transmission losses in the drive. The compressor efficiencies define the losses in the compressors. The heat exchangers are also assumed to be 100 % efficient in heat transfer. There is no heat loss or gain with the environment.

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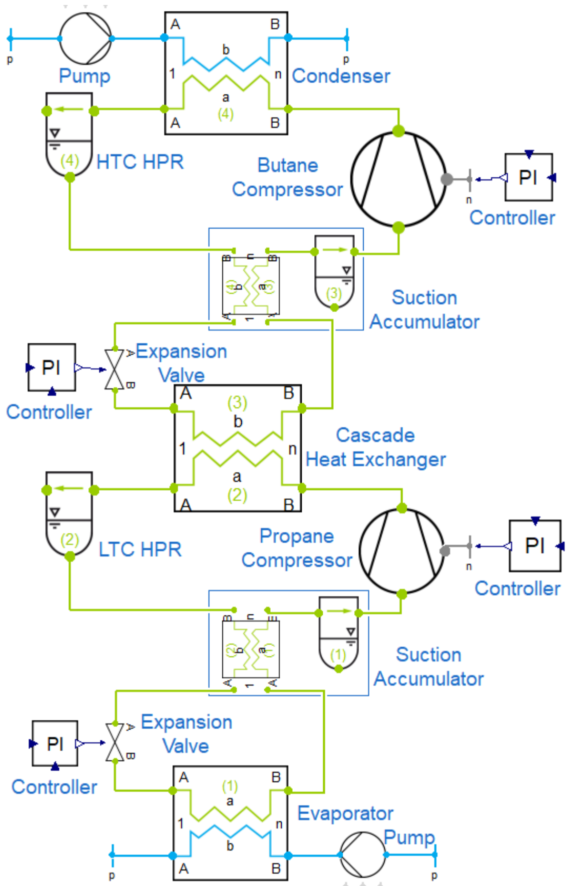

Figure 1: Theoretical model of the high temperature heat pump. Image is simplified for clarity. (Dynamic Modelling Library, TIL – Library)

The experimental setup

The experimental setup was designed based on the specifications and equipment dimensioning in the theoretical model. The heat sink, which is connected to the HTC condenser was designed as a water loop with a circulation pump, an electric heater and a bypass valve. The heat sink water loop is further connected to the laboratory cooling line by a heat recycle heat exchanger. Here, the heat is discharged and the heat sink inlet temperature (HSIT) is returned to the set point. A bypass valve is installed across this heat exchanger to regulate the HSIT. Heat discharged to the laboratory cooling line is partly tapped off to the LTC evaporator as the heat source. The heat source has installed a circulation pump, an electric heater and a bypass valve. The bypass valve regulates the heat source inlet temperature (HSOIT) by mixing with cooled water from the heat source outlet. All the components of the experimental setup were commercially available except the prototype compressor developed for HTHPs. This will enable a fast introduction of the technology to the market.

On both water loops, high pressure safety valves are installed and an expansion tank is connected to pressurize the water to keep it liquid at all operating conditions. The compressors are installed with a frequency converter for speed regulation, and an oil separator/return system. The piping of both the heat pump cycles and water loops are thermally insulated. For safety, the heat pump is placed in a containment box that is ventilated to maintain below atmospheric pressure. A gas detection unit is also installed. Standards EN 378 – 1:2008 and ISO 5149 – 1:2014 which describes the safety requirements for such heat pumps in the laboratory as well as for large scale industrial application were implemented. The schematics showing the main components of the heat pump is seen in Fig. 2. The list of the main component of the heat pump cycles and their specification are shown in Table 1.

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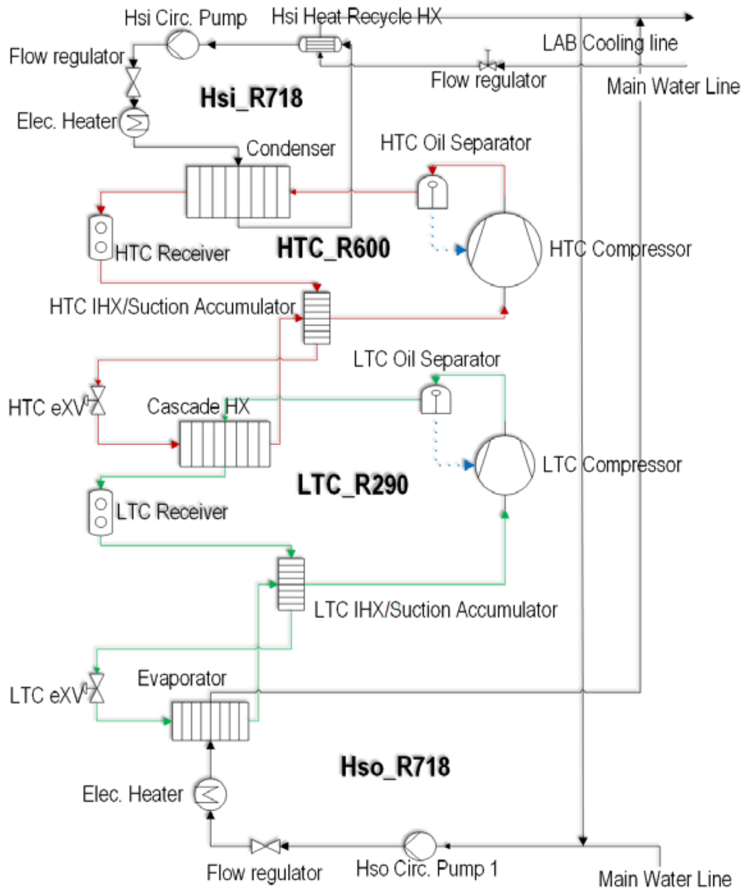

Figure 2: Schematics of the high temperature heat pump test facility

Table 1: The main components sizing for both the simulation and the experiment models

Component	Model Type	Specification	Experiment/Simulation	Unit
HTC Compressor	Dorin HEX1501CC	Speed variation	30 - 50	Hz
		Displacement at 50 Hz	48.8	$m^3 \cdot hr^{-1}$
LTC Compressor	Dorin HEX551CC	Speed variation	30 - 50	Hz
		Displacement at 50 Hz	23.3	$m^3 \cdot hr^{-1}$
Condenser		No. of plates	60	-
		Length	0.25	m
		Width	0.124	m
Cascade HX	Kaori	No. of plates	90	-
		Length	0.466	m
		Width	0.106	m
Evaporator		No. of plates	22	-
		Length	0.466	m
		Width	0.106	m
HPR		Volume	4.4	Litre
Suction Accumulator/ HX		Volume	5.8	Litre

2.2. The butane HTC Compressor

The butane HTC compressor is a prototype semi hermetic piston compressor designed for working fluids such as hydrocarbons and HFCs. It has been modified for high temperature operation with butane.

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The manifold is built externally with an estimated maximum discharge temperature of 160 °C regulated by a temperature sensor. To reduce the heat generation at the electric motor, the motor was sized 25 % larger than would have been required by the compressor. Thermal protections for the compressor were increased (for up to 140 °C). The compressor consists of an internal crankcase heater, an oil sight glass and is equipped with an oil separator and an oil return valve. The prototype high temperature compressor and the sensors attached for parameter study are shown in Fig. 3. For all calculations related to the prototype compressor, the temperature sensors attached to the compressor suction and discharge heads and not the pipes are used for better accuracy (avoiding heat losses on the pipe and cooling of electric motor).

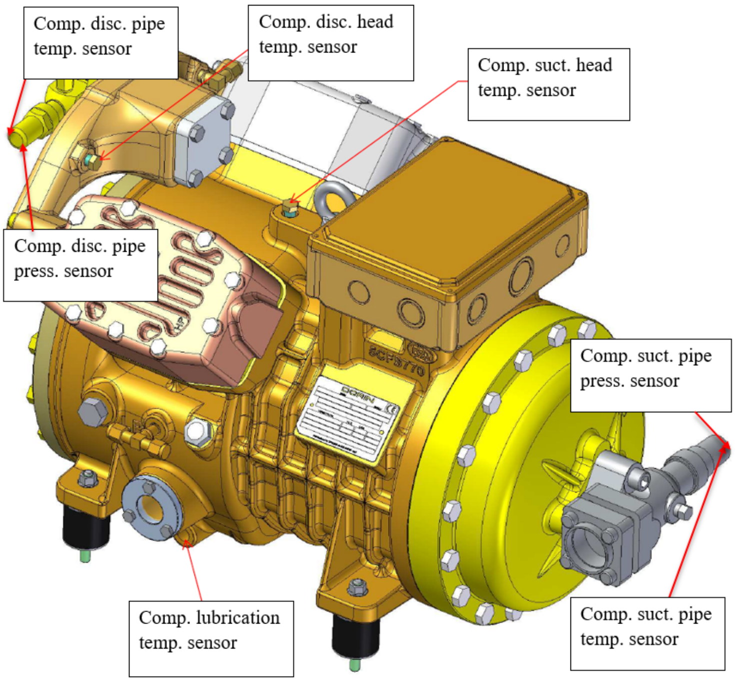

Figure 3: The sensors attached to the Prototype butane compressor for parameter study

2.3. Instrumentation and data logging

The test facility is instrumented with 16 temperature sensors of K-type thermocouple at every state point of the heat pump cycle. They are positioned at the centre of the pipes with the use of a temperature sensor tube. An additional 9 temperature sensor (3 K-type, 6 J-type thermocouples) are positioned in the compressor discharge and suction head, lubrication port, and electric motor windings. 2 pressure transmitters and 2 pressure indicators are installed before and after the compressors in both cycles. 2 pressure transmitters are also installed after the condensers and before the evaporators. 2 Coriolis mass flow meters are installed after the high-pressure receiver and before the expansion valve in both heat pump cycles. On the water cycles, 2 ultrasonic energy meters are installed. The compressors are connected to a frequency converter where the power consumption and the compressor speed values are read.

The temperature, pressure and flow meter values are read and stored 5 times every second. The heat pump is operated to a period of steady state operation with minimal changes in parameters. Data is stored for an average duration of 10 minutes for every test point. The sensors values are averaged over this duration. The standard deviation values for the temperature sensors (°C) are less than 0.4, for the pressure sensors (bar) 0.25, and for the mass flow meter (kg/min) 0.06. The compressor power consumption and speed are directly read from the frequency converter once for every test point during the steady state period. Table 2 shows the type, accuracy and range of the test facility sensors. The table also shows the maximum uncertainty of the efficiency and COP values presented in the results.

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Table 2: Sensor data and accuracy

Sensor type	Type	No. of Units	Uncertainty	Range
Temperature transmitter	Thermocouple K Type	5	± 1.1 K	-
	Thermocouple J Type	6	± 1.1 K	-
Pressure transmitter		4	± 0.2% FS BSL	0 - 30 barg
Flow meter heat pump	Coriolis	2	± 0.2 %	0.5 – 50 kg/min
Efficiencies		-	± 2.5 %	-
Heating COP		-	± 2.5 %	-
Combined COP		-	± 3.0 %	-

2.4. Compressor efficiencies

The total compressor efficiency evaluates the performance of the compressor in converting electrical input power to the working fluid. It is defined as the ratio between the rates of isentropic energy into the fluid to the power input at the frequency converter as expressed in Equation (1). The compressor isentropic efficiency is the ratio between the isentropic enthalpy change to the actual enthalpy change by the fluid as expressed in Equation (2) and the volumetric efficiency is the ratio between the actual volume of fluid at compressor suction to the compressor piston displacement as expressed in Equation (3).

$$\eta_{total} = \frac{m \cdot (h_{disc,isen} - h_{in})}{Power\ reading\ at\ frequency\ converter} \tag{1}$$

$$\eta_{isen} = \frac{(h_{disc,isen} - h_{in})}{(h_{disc} - h_{in})} \tag{2}$$

$$\eta_{vol} = \frac{v * m}{piston\ displacement\ at\ set\ speed} \tag{3}$$

2.5. The heating COP

The heating COP is the measure of the performance of the heat pump when the function and the energy savings benefit is solely for heating. Heat absorbed at the evaporator from the waste heat is not considered. The heating COP is evaluated as the ratio between the rate of heat rejected at the condenser (heating capacity) to the amount of power input at the compressors of the heat pump. It is mathematically represented in Equation (4).

$$COP_{heating} = \frac{Heat\ rejected\ at\ condenser}{HTC\ compressor\ power\ + LTC\ compressor\ power} \tag{4}$$

2.6. The combined COP

The combined COP is a measure of the performance of the heat pump when the heating and cooling capacities are considered simultaneously (Byrne et al., 2009; Liu et al., 2016; Luo et al., 2018). The HTHP can replace both the capacities of conventional heating systems such as steam boilers and the capacities of cooling systems such as the cooling tower or dry coolers. For industrial applications that have in addition to the heating demand at 115 ^{o}C, a cooling demand at the evaluated waste heat temperatures, the combined COP shows the performance of having a heat pump for both heating and cooling. The combined COP is evaluated as the ratio between the sum of heating and cooling capacities to the amount of power input at the compressors of the heat pump as expressed in Equation (5).

$$COP_{combined} = \frac{Heat\ rejected\ at\ condenser + Heat\ absorbed\ at\ evaporator}{HTC\ compressor\ power\ + LTC\ compressor\ power} \tag{5}$$

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2.7. The operating conditions

The operating conditions for the results presented are shown in Table 3. The operating conditions are set by a case study of a milk production plant that has waste heat from ammonia condensers currently discharged by dry coolers and heat demand for process stream (pressurized hot water) at high temperature (115 ^{o}C), which is currently supplied by electric steam boilers. The compressor is varied between 30 – 50 Hz in steps of 5 Hz. The heat source inlet, heat sink inlet and outlet are regulated using the control valves of the water line. For the HTC cycle performance analysis, with medium waste heat temperature (55 – 65 ^{o}C), the propane heating capacity at the cascade heat exchanger is the heat source to the cycle.

Table 3: Operating conditions of the experimental investigation

Parameters	Low		High
		Hz	
Compressor Speed	30		50
		^{o}C	
Heat Source Inlet Temperature	26		32
Heat Source Outlet Temperature	21		25
Heat Sink Inlet Temperature	85		86
Heat Sink Outlet Temperature	115		117

3. Discussions and Results

The results of the conducted experiments are presented with a focus on the HTC cycle. Though the LTC operates slightly above design conditions for the R290 compressor as documented in the compressor catalogue, it is considered within operational possibilities of the compressor. The modifications made for the HTC prototype compressor for HTHP operation can equally be done for the LTC R290 compressor for improved performance at high temperature. This can be implemented in further work.

3.1. HTC compressor discharge temperature

The HTC compressor discharge temperature is an important measured parameter in the evaluation of the performance of the prototype butane compressor. Compressor lubrication is influenced by the temperature of the fluid which affects the viscosity and solubility. If the temperature is too high, there can be degradation of the lubricant. The compressor discharge temperature also affects the wear rate of the compressor mechanical parts especially at the discharge. There is a limit to how high the discharge valve and other components can withstand high temperature operation. The temperature is monitored by a sensor to ensure the values does not exceed the specified condition (160 ^{o}C) agreed with the compressor manufacturer. The temperature values are plotted in Fig. 4.

3.2. HTC compressor suction temperature

The HTC compressor suction temperature is also an important measured parameter in the evaluation of the performance of the prototype butane compressor. The vapour fluid at the suction of the compressor serves as the compressor cooling medium. The fluid cools the compressor case, electric motor and the lubricant. The suction temperature also thermodynamically influences the discharge temperature.

Several factors influence the value of the suction temperature especially in a cascade configuration heat pump. Unlike in a single cycle, the evaporation temperature of the HTC is determined by the energy balance between the LTC and the HTC in the cascade heat exchanger. The energy capacities of each cycle are controlled by the compressor size and speed. For the simulation and experiment, the speeds are varied to optimize the overall heating COP of the system. However, the suction temperature is designed to have maximum of 80 ^{o}C for the effective cooling of the electric motor and other parts as

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specified by the compressor manufacturer. This limits the degree of variation for the optimal COP of the heat pump.

Other factors affecting the suction temperature at the HTC compressor is the size of the internal heat exchanger (IHX) and the expansion valve superheat value. The IHX in the test facility is a component of the suction accumulator. It is a single U-tube in the suction accumulator that discharges heat for the sub-cooling of the fluid at the condenser liquid line and to vaporize (if liquid) or superheat the vapour line to the compressor suction. Additional superheat by the IHX on the suction temperature are observed to be less than 2 K. The expansion valve regulates the amount of superheat at the evaporator vapour line. High superheat values will imply high compressor suction temperature. Due to the overhanging shape of the dew point curve of working fluid butane, there is a theoretical minimum superheat required to compress out of the two-phase region for every set compressor efficiency.

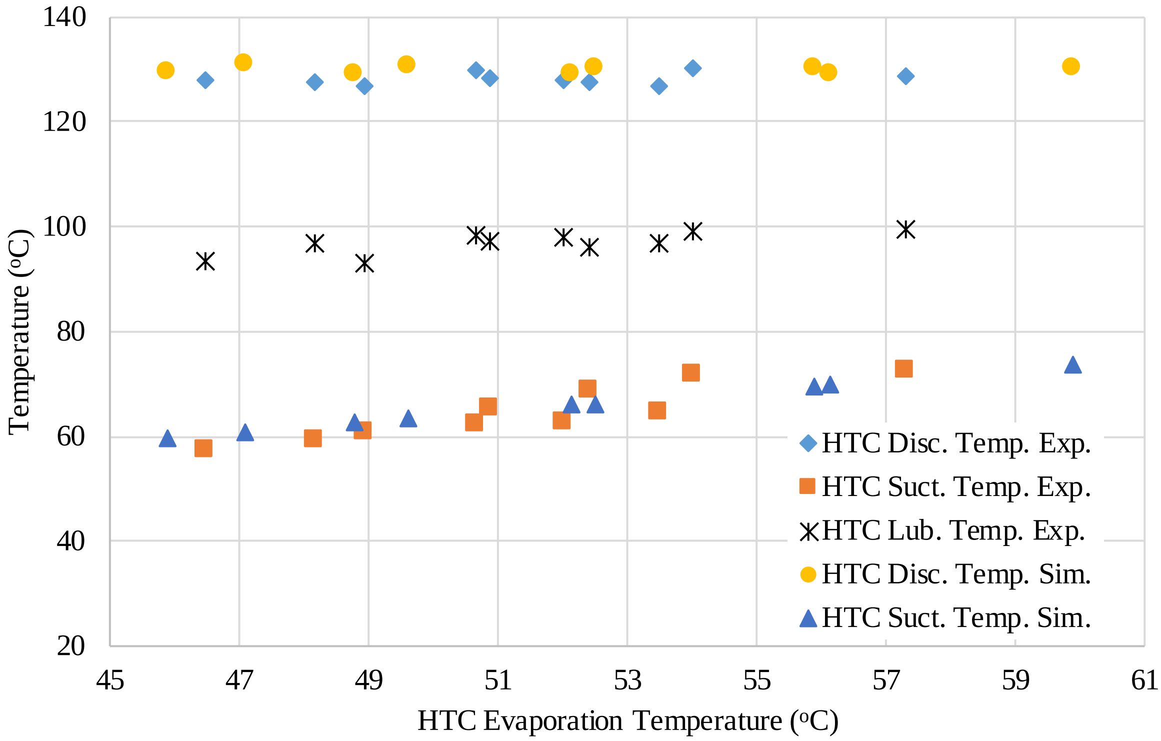

Figure 4: Prototype compressor temperatures of both simulation and experiment. Operating conditions are set by Table 3.

Fig. 4 shows the discharge and suction temperatures of the HTC prototype compressor for both the simulation and the experiment varied at similar operating conditions. The condensation temperature varies from 116 – 118 °C and 118 – 121 °C for the experiment and simulation respectively. Slight deviations in the HTC compressor temperatures are possibly explained by heat losses and temperature sensor positions. The oil temperature is below 100 °C on average for the HTC compressor. Is it sampled and seen to have slight change in colour with no observable presence of compressor wear. Further oil analysis will be done in subsequent work.

The discharge temperatures for the simulation and experiment are on average 129 °C and 127 °C respectively. The suction temperatures for the simulation and experiment are on average 70 °C and 68 °C respectively. Both discharge and suction temperatures are within the compressor limitations with a potential for an even higher operating parameter. Heat sink outlet temperatures up to 117 °C were recorded during the experiments. The suction temperature for the LTC compressor is on average 25 °C in the experiment compared with 21 °C in the simulation while the discharge temperature is on average 80 °C for both experiment and simulation.

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3.3. The suction and discharge pressure (HTC)

The compressors are installed with safety switches for high and low pressures and pressure sensors for measurements. Due to the cascade design, fluctuating capacity variations between the two cycles may lead to a high-pressure scenario. It is also important to measure the low-pressure side of HTC as the thermodynamic properties of butane are close to atmospheric conditions, especially during the heat pump start-up.

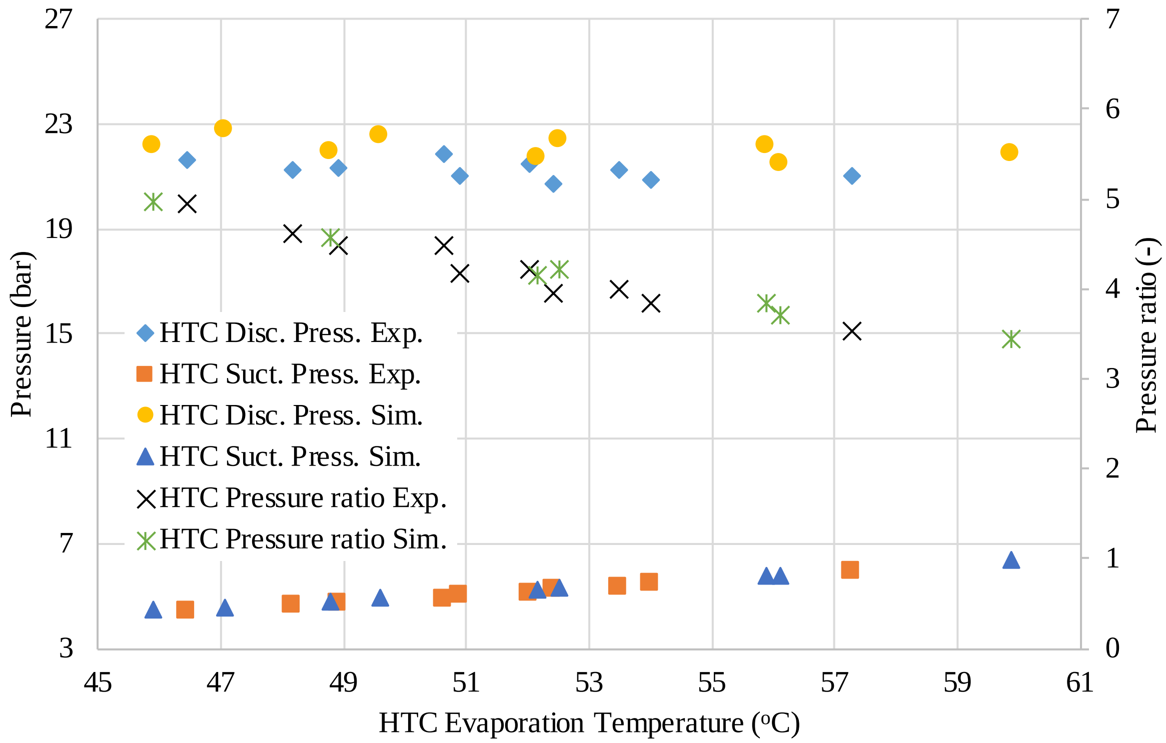

Figure 5: Operating pressures of the HTC cycle of both experiments and simulations. Operating conditions are set by Table 3.

The operating pressure values for HTC with the prototype compressor are shown in Fig. 5. The experimental values are consistent with the simulation values. Variations seen in the compressor discharge pressure may be due to oil separator pressure drop and the position of the pressure sensor. The highest operating pressure is below 22 bar for the experiment across the operating conditions. This is well below the rated maximum (31 bar) of the prototype compressor. The suction pressure during operation is above atmospheric pressure between 4 and 7 bar. Heat pumps with hydrocarbons will require operation above atmospheric pressure to prevent the possibility of an explosive mixture through air ingress. The pressure ratios are between 3 and 6 suitable for one stage operation without fluid leakages from high pressure to low pressure. Despite the high temperature operation, the thermodynamic properties of butane are well suited for relatively low pressures at the compressor compared to other fluids. The LTC compressor with propane has on average 20 bar at the discharge and 8 bar at the suction. The properties of propane fits as the LTC working fluid.

The consistency between the experimental and simulation values of temperature and pressure allows for the extrapolation of the operating conditions of the heat pump. Since the experiment has limitations of operating range, the simulation can be used to study to a degree of accuracy beyond the experimental limits. It is also important to theoretically evaluate the heat pump against maximum pressure and temperature values that can be destructive for the prototype compressor and proffer solutions to avoid these conditions.

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3.4. Compressor efficiencies

The prototype compressor in the HTC is operating in a new region of higher temperatures and its performance is measured.

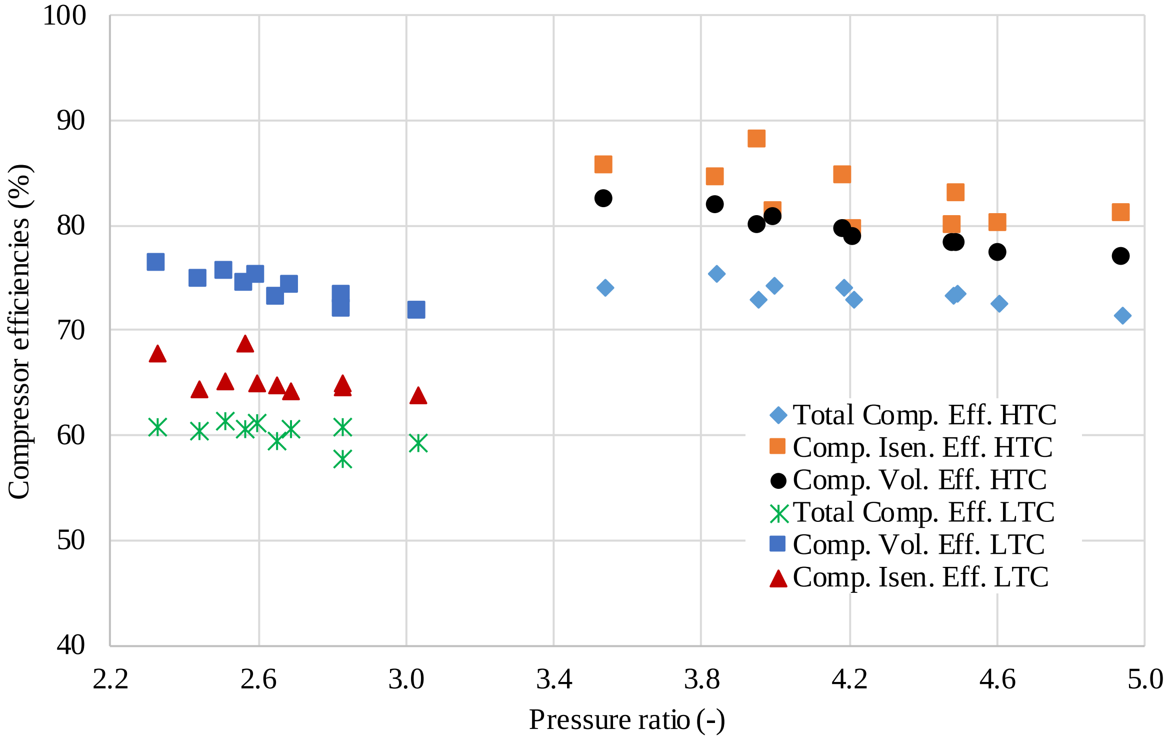

Figure 6: Compressor total compression efficiency, isentropic efficiency and volumetric efficiency experimental results. Operating conditions are set by Table 3.

The compressor efficiencies are shown in Fig. 6. The HTC values are calculated using the temperature sensors at the suction and discharge heads for the HTC. The compressor isentropic efficiency has an average value of 84 % while the total compression efficiency has an average value of 74 %. The difference is accounted for by the transmission, electric motor, frequency converter, heat and other losses in the process of power conversion. The volumetric efficiency has an average of 82 % across all tested operating conditions. The LTC compressor has on average a total compression efficiency of 60 %, a compressor isentropic efficiency of 65 % and volumetric efficiency of 74 %. The LTC compressor's lower efficiencies are possibly explained by its operation slightly outside of the design profile for the compressor.

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3.5. Optimizing heat pump cycles

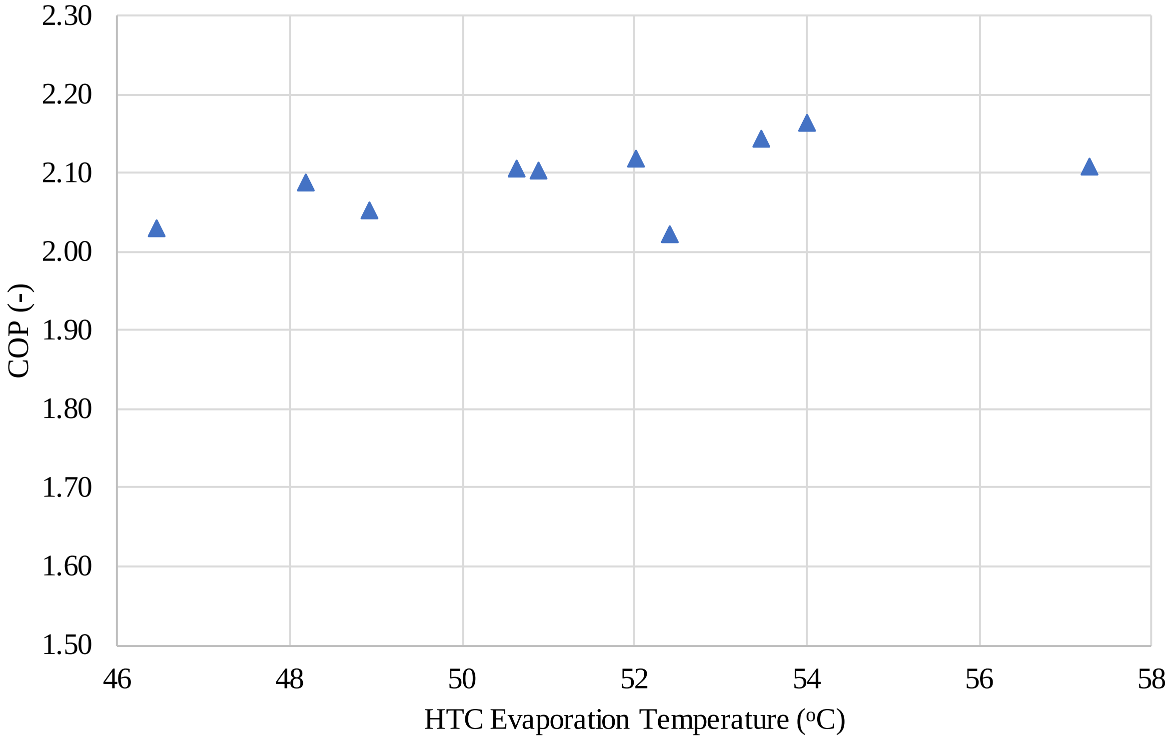

Figure 7: Optimizing the experimental overall heat pump performance by varying operating parameters. Operating conditions are set by Table 3.

The optimal operating parameters for a cascade configuration heat pump are the parameters at which the overall COP of the heat pump is maximized. For a specific heating capacity, the distribution of compressor work across the two cycles can be adjusted to minimize the total sum of work input into the heat pump. This is done by varying the compressor speed. Varying the compressor speeds distributes the amount of temperature lift within each cycle of the cascade heat pump thereby determining the operating parameters of the cascade heat exchanger that connects the two cycles. The compressor speeds are varied by the frequency converter.

Fig. 7 shows the values of overall COP of the heat pump when the compressor speeds are varied between the high temperature cycle and the low temperature cycle. The result does not clearly suggest an optimal operating parameter that maximizes the COP of the heat pump. This can be explained by the effects of the thermodynamic similarities between the LTC with propane and the HTC with butane at the respective operating conditions. As the temperature lift in each cycle is decreased or increased, the corresponding work input by the 2 compressors in each cycle changes by a closely proportional value. However, it is can be observed that at an HTC evaporation temperature (cascade HX evaporating temperature) between 53 – 54 °C, the experiment values show the highest overall heating COP of the HTHP. Further experimental data will clarify the optimal operating point of the heat pump.

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3.6. Heat pump performance

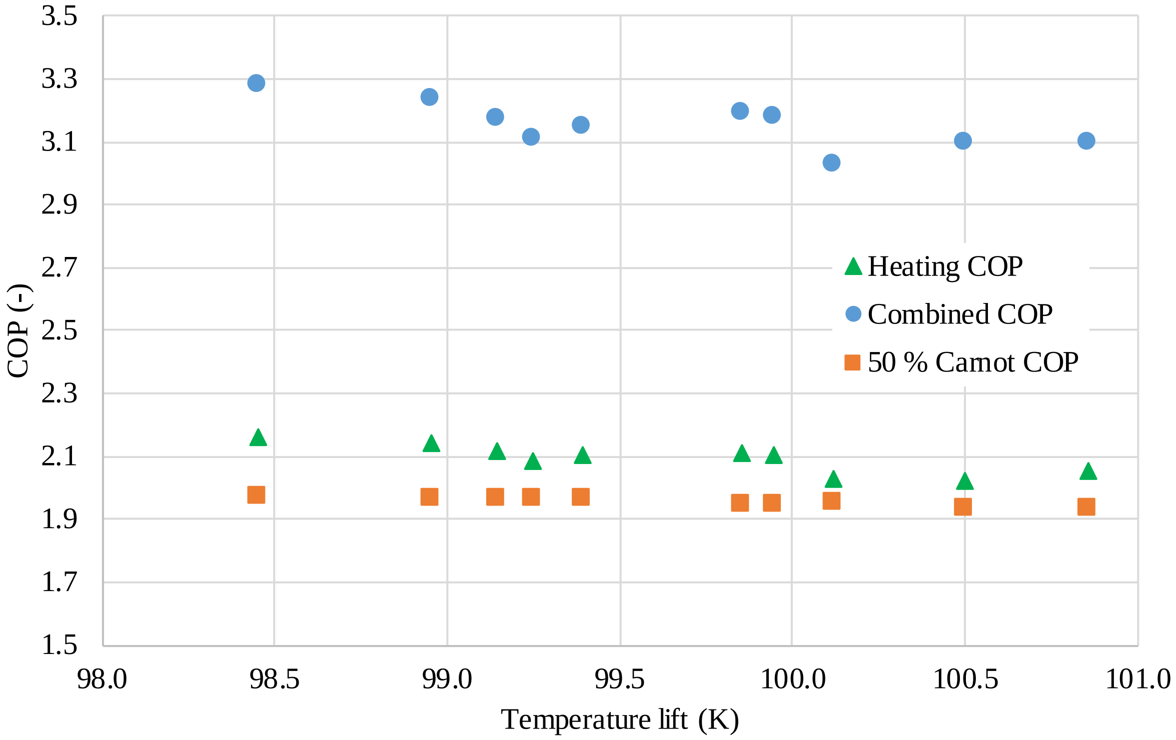

Figure 8: The experimental coefficient of performance of the HTHP. Operating conditions are set by Table 3.

Fig. 8 shows the COP of the HTHP plotted against the temperature lift (difference between condensation temperature in HTC and evaporation temperature in LTC). The heat source to the LTC evaporator has a temperature range between 25 – 35 ^{o}C for lower temperature industrial waste heat. The heating COP of the heat pump has an average of 2.1 for the experiment. These COP values are for a temperature lift of 98 – 101 K. When compared with the Carnot COP, calculated based on the defined temperature lift, the heating COP in the experiment is above 50 % of the Carnot COP. When compared with an industrial process installed with an electric boiler for high temperature heating (115 ^{o}C), a heating COP of 2.2 would imply more than 50 % in electrical energy savings.

The combined COP of the heat pump experiment has an average value of 3.1 for the temperature lift (difference between condensation temperature in HTC and evaporation temperature in LTC) of 98 – 101 K. When compared with an industrial process operating within the same temperature distribution, which is installed with an electric boiler for high temperature (115 ^{o}C) supply and a cooling tower/dry coolers for waste heat discharge at 30 ^{o}C or other cooling demands, a heat pump with combined COP of 3.1 will not only replace the electric boiler and cooling tower/dry coolers capacities, save more than 50 % in electrical energy, but in addition, save electrical energy cost of operating pumps and other operating cost of the cooling tower/dry coolers or provide cooling for other processes.

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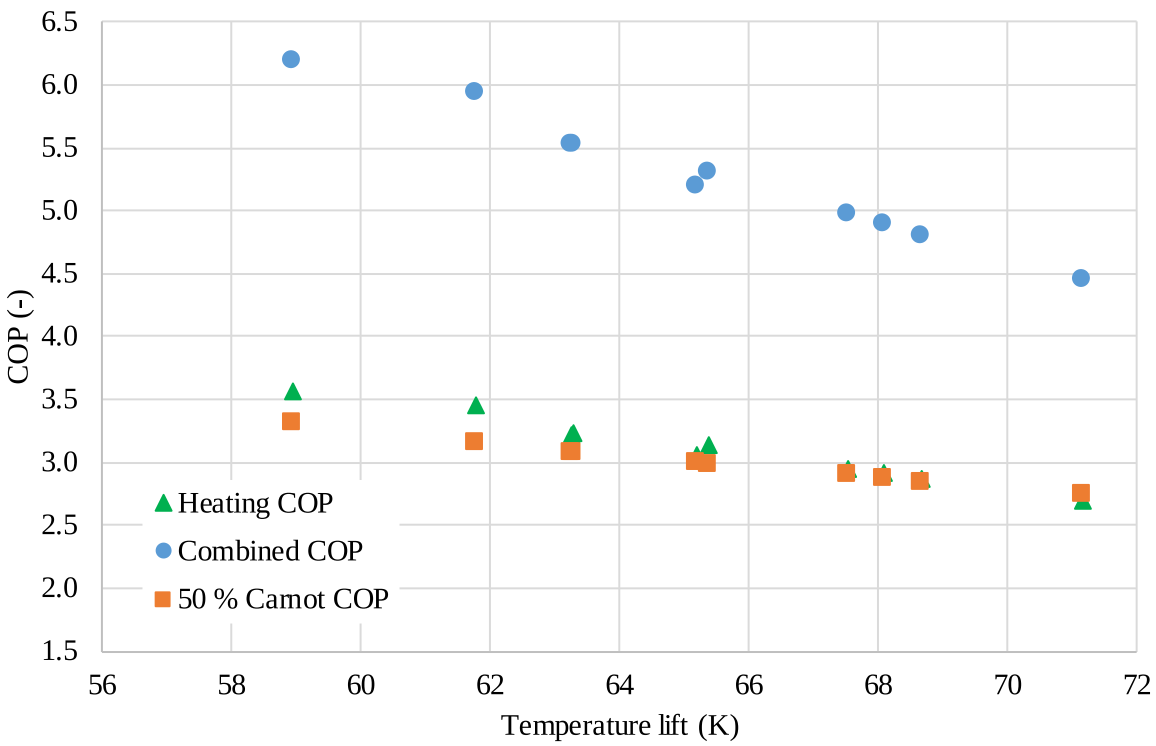

Figure 9: The experimental coefficient of performance of the HTC with prototype butane compressor. Operating conditions are set by Table 3.

Fig 9. Shows the COP of the HTC with the prototype butane compressor plotted against the temperature lift in the HTC (difference between condensation temperature in HTC and evaporation temperature in HTC). The heat source to the evaporator which is the condensation of propane in the LTC has a condensation temperature range of 55 – 65 ^oC for relatively higher industrial waste heat or cooling demand. The average heating COP across the operating conditions for the experiment is 3.1 for a temperature lift of 58 – 72 K. The average combined COP for the experiment is 5.3. With a heating COP of 3.1 and combined COP of 5.3, the heat pump will save more than 70 % electrical energy cost while replacing capacities of the electric boiler, cooling towers/dry coolers and their operating cost.

3.7. Emissions and cost advantage

The HTHP is compared to a fossil fuel gas boiler with 85 % energy efficiency and an emissions intensity of 210 gCO_2/kWh. A HTHP COP of 1.3 will be required to be competitive if the European emission intensity for electricity is 276 gCO_2/kWh (EEAgency, 2016). Using the HTHP in heating mode with COP of 2.1 would have lower emissions, thus reducing carbon emission by 37 %. Furthermore, when the waste heat temperature is medium (55 – 65 ^oC), the HTHP COP is 3.1 for heating mode only and 5.3 for combined heating and cooling, thereby indicating a much better alternative to fossil fuel boilers. The European carbon emissions continues to decrease as more electricity generation systems are converted to lower emissions system. Several countries such as Sweden, Finland, France, Austria, Lithuania and others already have values lower than 150 gCO_2/kWh (EEAgency, 2016). For Norway, the greenhouse gas emissions for electricity production and distribution are 22 gCO_2/kWh (AdaptConsultingAS, 2015), enabling emission reduction by about 95 %.

Comparing operational cost of HTHP and fossil fuel boilers across the European union based on energy input cost, gas prices at 3 euro cents/kWh for non-household consumers, while industrial electricity prices for extra-large users, which is the target HTHP application, is at 6.7 euro cents/kWh (Bahra, 2017). At 85 % gas boiler efficiency, a HTHP COP of over 1.9 would be required to have an operating cost advantage over gas boilers. Operating the heat pump at all modes, heating only and combined

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heating and cooling will result in an operating cost advantage. Both emission and operating cost rates show the potential for the HTHP. The European energy mix shows in general a trend towards cleaner and more renewable sources and the primary energy factor was reduced by the EU from 2.5 to 2.1 in 2018 (EuropeanCommission, 2016). This means that a heat pump with a COP of 2.1 will break even and have reduced carbon emission.

Table 4: Emissions and operating cost comparisons of different heating systems in Europe. Data retrieved from EEAgency (2016), Bahra (2017), AdaptConsultingAS (2015) and Equinor (2018)

	European Union		Norway		Germany	
Heating method	Emissions gCO_2/kWh	Cost euro cent/kWh	Emissions gCO_2/kWh	Cost euro cent/kWh	Emissions gCO_2/kWh	Cost euro cent/kWh
Direct electric heating	276.0	6.7	22.0	3.1	425.0	8.5
Gas boilers (85 % efficiency)	210.0	3.5	210.0	2.0	210.0	3.6
HTHP heat delivery at 115 ^{o}C						
Heating mode (25 - 35 ^{o}C)	131.4	3.2	10.5	1.5	202.4	4.0
Heating mode (55 - 65 ^{o}C)	89.0	2.2	7.1	1.0	137.1	2.7
Combined (55 - 65 ^{o}C)	52.1	1.3	4.2	0.6	80.2	1.6

Table 4 shows the emissions and operating cost values with the use of the HTHP. Due to the higher cost of electricity in countries like Germany, gas boilers have lower operating cost than HTHP for waste heat recovery at 25 – 35 ^{o}C. They are however less cost effective for waste heat temperature at 55 – 65 ^{o}C with a savings of 25 % in heating mode relative to gas boilers. In all other scenarios there is both reduction in operating cost and emissions to the environment. Norway, with lower electricity cost and very low carbon emissions has the most savings.

4. Conclusion

A 20 kW heating capacity high temperature heat pump has been developed and evaluated using both simulation and experiments for high temperature heating up to 115 ^{o}C. The heat pump will recover waste heat at a temperature of 25 – 35 ^{o}C (full cascade cycle) and 55 – 65 ^{o}C (butane HTC only) for low and medium temperature waste heat. It is a cascade configuration cycle with a prototype butane compressor on the HTC and propane as the working fluid on the LTC. The cycles are connected to water heat sink and heat source loops representing industrial heat demand and waste heat supply. The heat pump can replace existing capacities of electric or steam boilers used for industrial applications such as pasteurization, drying, sterilization, distillation and other suitable high temperature processes.

The heat pump was found to have a COP of 2.1 average for a temperature lift of 98 – 101 K and 3.1 average for a temperature lift of 58 – 72 K which makes it a better alternative to gas boilers at current carbon emissions and electricity cost rates. The prototype butane compressor had stable operation with an average of 74 % total compression efficiency. Experimental temperature values at the compressor discharge and suction are 129 ^{o}C and 70 ^{o}C respectively with pressure values below 22 bar indicating the potential to increase the operating region of the heat pump to deliver higher temperature at the heat sink outlet.

Further research could potentially improve the performance of the heat pump. Condenser sub-cooling would potentially improve the COP of the heat pump with the removal of the high-pressure receiver. This will also reduce the working fluid charge. Ejectors can be installed on both cycles to reduce

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compressor work input. Experiments can be conducted using other combinations of hydrocarbons and their mixtures. Lubrication degradation is a possible limitation of HTHP compressors. A thorough oil analysis can be done to investigate possible changes in lubrication properties. Though the advantage of a HTHP over existing solution is evident by the COP especially for the operating cost and emissions rate across EU, an economic analysis comparing investment cost and payback time is needed to further show the comparative advantage.

Nomenclature

Comp	Compressor
Disc	Discharge
Exp	Experiment
GWP	Global Warming Potential
HFC	HydroFlouroCarbons
HFO	HydroFlouroOlefin
HPR	High Pressure Receiver
HSIT	Heat Sink Inlet Temperature
HSOIT	Heat source Inlet Temperature
HTC	High Temperature Cycle
HTHP	High Temperature Heat Pump
HVAC	Heating, Ventilation, and Air Conditioning
IHX	Internal Heat Exchanger
Isen	Isentropic
LTC	Low Temperature Cycle
Lub	Lubrication
ODP	Ozone Depletion Potential
Press	Pressure
Sim	Simulation
Suct	Suction
Temp	Temperature
kWh	KiloWatt Hour

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References

[1] AdaptConsultingAS. (2015). Fastsettelse av CO2-faktorer og primærenergifaktorer for elektrisitet. https://www.energinorge.no/contentassets/7269b2561cc94c2f888128dcf0803b98/fastsettelse-av-vektingsfaktorer-for-elektrisitet---sluttrapport.pdf

[2] Arpagaus, C., Bless, F., Schiffmann, J., & Bertsch, S. S. (2017). Review on high temperature heat pumps - market overview and research status. Paper presented at the International Workshop on High Temperature Heat Pumps, Copenhagen, Denmark.

This is the accepted version of an article published in Energy
https://doi.org/10.1016/j.energy.2019.02.159

[3] Arpagaus, C., Bless, F., Uhlmann, M., Schiffmann, J., & Bertsch, S. S. (2018). High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials. Energy, 152, 985-1010. doi:https://doi.org/10.1016/j.energy.2018.03.166

[4] Assaf, K., Zoughaib, A., Sapora, E., Peureux, J.-L., & Clodic, D. (2010). Experimental simulation of a heat recovery heat pump system in food industries. In *International Refrigeration and Air Conditioning Conference*.

[5] Ayub, Z. (2016). World's Largest Ammonia Heat Pump (14 MWh) for District Heating in Norway-A Case Study. Heat Transfer Engineering, 37(3-4), 382-386. doi:10.1080/01457632.2015.1052716

[6] Bahra, A. (2017). A comparison of electricity prices between the UK and EU. Retrieved from https://www.constructionproducts.org.uk/news-media-events/blog/2017/june/a-comparison-of-electricity-prices-between-the-uk-and-eu/

[7] Bamigbetan, O., Eikevik, T. M., Nekså, P., & Bantle, M. (2016). Evaluation of natural working fluids for the development of high temperature heat pumps. Paper presented at the 12th IIR Gustav Lorentzen Conference on Natural Refrigerants, Edinburgh.

[8] Bamigbetan, O., Eikevik, T. M., Nekså, P., & Bantle, M. (2017a). Review of vapour compression heat pumps for high temperature heating using natural working fluids. International Journal of Refrigeration, 80, 197-211. doi:http://dx.doi.org/10.1016/j.ijrefrig.2017.04.021

[9] Bamigbetan, O., Eikevik, T. M., Nekså, P., Bantle, M., & Banasiak, K. (2017b). Extending Hydrocarbon Heat Pumps to Higher Temperatures: Predictions from Simulations. Paper presented at the 30th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems, San Diego, California.

[10] Bamigbetan, O., Eikevik, T. M., Nekså, P., Bantle, M., & Schlemminger, C. (2018). Theoretical analysis of suitable fluids for high temperature heat pumps up to 125 °C heat delivery. International Journal of Refrigeration. doi:https://doi.org/10.1016/j.ijrefrig.2018.05.017

[11] Bantle, M., Tolstorebrov, I., & Eikevik, T. M. (2015). Possibility for Mechanical Vapor Re-Compressrion for Steam Based Drying Processes. Paper presented at the 1st Nordic Baltic Drying Conference, Trondheim.

[12] Blesl, M., Wolf, S., & Fahl, U. (2014). Large scale application of heat pumps. In *7th EHPA European Heat Pump Forum*. Berlin, Germany.

[13] Byrne, P., Miriel, J., & Lenat, Y. (2009). Design and simulation of a heat pump for simultaneous heating and cooling using HFC or CO2 as a working fluid. International Journal of Refrigeration, 32(7), 1711-1723. doi:https://doi.org/10.1016/j.ijrefrig.2009.05.008

[14] Chua, K. J., Chou, S. K., & Yang, W. M. (2010). Advances in heat pump systems: A review. Applied Energy, 87(12), 3611-3624. doi:10.1016/j.apenergy.2010.06.014

[15] Clark, E., & Wagner, S. (2016). The Kigali Amendment to the Montreal Protocol: HFC Phase-down. Kigali, Rwanda: http://multimedia.3m.com/mws/media/1365924O/unep-fact-sheet-kigali-amendment-to-mp.pdf

[16] Combitherm. (2017). High temperature heat pumps. Retrieved from www.combitherm.de

[17] Cooper, A., & Lyon, M. (2004). Make the Most of MVR. Chemical Engineering, 111(7), 44-47.

This is the accepted version of an article published in Energy
https://doi.org/10.1016/j.energy.2019.02.159

[18] Dietrich, W., & Fredrich, O. (2012). GEA Grasso heat pumps using ammonia–the megawatt range. Frankfurt: Achema.

[19] EEAgency. (2016). Overview of electricity production and use in Europe. Copenhagen: eea.europa.eu

[20] Eikevik, T., Alves-Filho, O., & Strømmen, I. (2004). Heat Pump Drying - New Technologies And Operational Modes For Production Of A New Generation Of High Quality Dried Products. Paper presented at the International Workshop and Symposium on Industrial Drying (IWSID), Mumbai, India.

[21] Eikevik, T., Strømmen, I., & Alves-Filho, O. (2005). Heat Pump Fluidised Bed Dryer with CO2 as Refrigerant – Measurements of COP and SMER. Paper presented at the 3rd Nordic Drying Conference, Karlstad, Sweden.

[22] Eisa, M. A. R., Best, R., & Holland, F. A. (1986). Working fluids for high temperature heat pumps. Journal of Heat Recovery Systems, 6(4), 305-311. doi:10.1016/0198-7593(86)90116-5

[23] Equinor. (2018). Internal gas price. Retrieved from https://www.equinor.com/en/investors/our-dividend/internal-gas-price.html

[24] EuropeanCommission. (2016). Discussion paper for the review of the default primary energy factor (PEF) reflecting the estimated average eu generation efficiency referred to in annex iv of directive 2012/27/EU and possible extension of the approach to other energy carriers. https://static1.squarespace.com/static/57d64e6629687f1a258ec04e/t/58325548e6f2e12eebd8cc2a/1479693640253/DiscussionPaper20May16.pdf

[25] Fukuda, S., Kondou, C., Takata, N., & Koyama, S. (2014). Low GWP refrigerants R1234ze(E) and R1234ze(Z) for high temperature heat pumps. International Journal of Refrigeration-Revue Internationale Du Froid, 40, 161-173. doi:10.1016/j.ijrefrig.2013.10.014

[26] Helminger, F., Hartl, M., Fleckl, T., Kontomaris, K., & Pfaffl, J. (2016). Hochtemperatur Warmepumpen Messergebnisse einer Laboranlage mit HFO-1336MZZ-Z bis 160 C Kondensationstemperatur. In *14. Symposium Energieinnovation*. TU Graz.

[27] Hewitt, N. J., McMullan, J. T., Henderson, P. C., & Mongey, B. (2001). Advanced cycles and replacement working fluids in heat pumps. Applied Thermal Engineering, 21(2), 237-248. doi:10.1016/S1359-4311(00)00053-3

[28] Hoffmann, K., & Pearson, D. F. (2011). Ammonia Heat Pumps for District Heating in Norway 7 a case study. London:

[29] Huang, M., Liang, X., & Zhuang, R. (2017). Experimental Investigate on the Performance of High Temperature Heat Pump Using Scroll Compressor.

[30] IEA. (2014). Application of Industrial Heat Pumps, Final report, Part 1, Report No. HPP-AN35-1.

[31] Infante Ferreira, C. A., Zamfirescu, C., & Zaytsev, D. (2006). Twin screw oil-free wet compressor for compression–absorption cycle. International Journal of Refrigeration, 29(4), 556-565. doi:http://dx.doi.org/10.1016/j.ijrefrig.2005.10.006

[32] Jensen, J. K., Markussen, W. B., Reinholdt, L., & Elmegaard, B. (2015a). On the development of high temperature ammonia–water hybrid absorption–compression heat pumps. International Journal of Refrigeration, 58, 79-89. doi:http://dx.doi.org/10.1016/j.ijrefrig.2015.06.006

This is the accepted version of an article published in Energy
https://doi.org/10.1016/j.energy.2019.02.159

[33] Jensen, J. K., Ommen, T., Markussen, W. B., Reinholdt, L., & Elmegaard, B. (2015b). Technical and economic working domains of industrial heat pumps: Part 2 - Ammonia-water hybrid absorption-compression heat pumps. International Journal of Refrigeration, 55, 183-200. doi:10.1016/j.ijrefrig.2015.02.011

[34] Johnson, I., Choater, B., & Dillich, S. (2008). Waste heat recovery: Opportunities and challenges. Epd Congress 2008, 47.

[35] Kim, J., Park, S., Baik, Y. J., Chang, K. C., Ra, H. S., Kim, M., & Kim, Y. (2013). Experimental study of operating characteristics of compression/absorption high-temperature hybrid heat pump using waste heat. Renewable Energy, 54, 13-19. doi:10.1016/j.renene.2012.09.032

[36] Kim, M., Baik, Y. J., Park, S. R., Chang, K. C., & Ra, H. S. (2010). Design of a high temperature production heat pump system using geothermal water at moderate temperature. Current Applied Physics, 10(2 SUPPL.), S117-S122. doi:10.1016/j.cap.2009.11.015

[37] Kondou, C., & Koyama, S. (2015). Thermodynamic assessment of high-temperature heat pumps using Low-GWP HFO refrigerants for heat recovery. International Journal of Refrigeration, 53, 126-141. doi:http://dx.doi.org/10.1016/j.ijrefrig.2014.09.018

[38] Larminat, P. D., Arnou, D., Le Sausse, P., Clunet, F., & Peureux, J. L. (2014). A High Temperature Heat Pump Using Water Vapor as Working Fluid. 11th Iir Gustav Lorentzen Conference on Natural Refrigerants (2014), 857-864.

[39] Lemmon, E. W., Huber, M. L., & McLinden, M. O. (2013). NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, National Institute of Science and Technology (Version 9.1) [Standard Reference Data Program]. Gaithersburg.

[40] Liu, Y., Groll, E. A., Yazawa, K., & Kurtulus, O. (2016). Theoretical analysis of energy-saving performance and economics of CO2 and NH3 heat pumps with simultaneous cooling and heating applications in food processing. International Journal of Refrigeration, 65, 129-141. doi:https://doi.org/10.1016/j.ijrefrig.2016.01.020

[41] Luo, C., Yang, C., Zhang, Y., Xing, Z., & Zhang, Y. (2018). A novel chemical heat pump cycle for cooling and heating. Applied Thermal Engineering, 144, 59-64. doi:https://doi.org/10.1016/j.applthermaleng.2018.08.019

[42] Ma, L., Wang, H., & Wang, J. (2010). Cycle performance evaluation of HFC245fa for high temperature heat pump system. Taiyangneng Xuebao/Acta Energiae Solaris Sinica, 31(6), 749-753.

[43] Madsboell, H., Weel, M., & Kolstrup, A. (2014). Development of a Water Vapor Compressor for High Temperature Heat Pump Applications. 11th Iir Gustav Lorentzen Conference on Natural Refrigerants (2014), 891-899.

[44] Moisi, H. S., Verdnik, M., Rieberer, R., & Baumhakel. (2017). Entwicklung einer R600-Hochtemperatur-Wärmepumpe - Simulation und erste Messungen. In *Deutsche Kälte-Klima-Tagung 2017*.

[45] Neksa, P. (2002). CO2 heat pump systems. International Journal of Refrigeration-Revue Internationale Du Froid, 25(4), 421-427. doi:Doi 10.1016/S0140-7007(01)00033-0

[46] Neksa, P., Rekstad, H., Zakeri, G. R., & Schiefloe, P. A. (1998). CO2-heat pump water heater: characteristics, system design and experimental results. International Journal of Refrigeration-Revue Internationale Du Froid, 21(3), 172-179. doi:Doi 10.1016/S0140-7007(98)00017-6

This is the accepted version of an article published in Energy
https://doi.org/10.1016/j.energy.2019.02.159

[47] Nellissen, P., & Wolf, S. (2014). Heat pumps in non-domestic applications in Europe: Potential for an energy revolution. Presentation given at the 8th EHPA European Heat Pump. Germany:

[48] Nilsson, M., Rislå, H. N., & Kontomaris, K. (2017). Measured performance of a novel high temperature heat pump with HFO-1336mzz(Z) as the working fluid. Paper presented at the 12th IEA Heat Pump COnference, Rotterdam.

[49] Ochsner. (2017). Warmepumpen für grosse Leistungen, Broschüre ET04 P 03/2017.

[50] Oue, T., & Okada, K. (2013). Air-sourced 90℃ Hot Water Supplying Heat Pump

[51] Stavset, O., Banasiak, K., & Hafner, A. (2014). Analysis of High Temperature Heat Pumps Applying Natural Working Fluids. 11th Iir Gustav Lorentzen Conference on Natural Refrigerants (2014), 941-946.

[52] Tolstorebrov, I., Bantle, M., Hafner, A., Kuz, B., & Eikevik, T. M. (2014). Energy Efficiency by Vapor Compression in Superheated Steam Drying Systems. Paper presented at the 11th IIR Gustav Lorentzen Conference on Natural Refrigerants, Hangzhou, China.

[53] Wemmers, A. K., Haasteren, A. W. M. B. v., Kremers, P., & Kamp, J. v. d. (2017). Test results R600 pilot heat pump. Paper presented at the 12th IEA Heat Pump Conference 2017, Rotterdam.

[54] White, S. D., Yarrall, M. G., Cleland, D. J., & Hedley, R. A. (2002). Modelling the performance of a transcritical CO2 heat pump for high temperature heating. International Journal of Refrigeration, 25(4), 479-486. doi:http://dx.doi.org/10.1016/S0140-7007(01)00021-4

[55] Wu, X., Tang, H., Chen, W., Wang, X., & Xing, Z. (2015). Development of A High Temperature Heat Pump for Heat Recovery in Dyeing Industry. Paper presented at the The 24th IIR International Congress of Refrigeration.

[56] Yamazaki, T., & Kubo, Y. (1985). Development of a high-temperature heat pump. IEA Heat Pump Centre Newsletter, 4(4).

[57] Yang, J., Zhang, C., Zhang, Z., Yang, L., & Lin, W. (2016). Study on mechanical vapor recompression system with wet compression single screw compressor. Applied Thermal Engineering, 103, 205-211. doi:http://dx.doi.org/10.1016/j.applthermaleng.2016.04.053

[58] Zhang, J., Zhang, H.-H., He, Y.-L., & Tao, W.-Q. (2016). A comprehensive review on advances and applications of industrial heat pumps based on the practices in China. Applied Energy, 178, 800-825. doi:http://dx.doi.org/10.1016/j.apenergy.2016.06.049

[59] Zhang, S. J., Wang, H. X., & Guo, T. (2010). Theoretical investigation on working fluids of two-stage vapor-compression high-temperature heat pump. Kung Cheng Je Wu Li Hsueh Pao/Journal of Engineering Thermophysics, 31(10), 1635-1638. doi:10.1016/j.apen-ergy.2009.11.001

This is the accepted version of an article published in Energy
https://doi.org/10.1016/j.energy.2019.02.159