EXPERIMENTAL INVESTIGATION OF THE PERFORMANCE OF A HYDROCARBON HEAT PUMP FOR HIGH TEMPERATURE INDUSTRIAL HEATING

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ABSTRACT

A 20 kW heat pump test rig has been built to investigate and validate heat delivery up to 110 °C using a vapour compression cycle. The heat pump is a cascade cycle with propane as the working fluid in the low temperature cycle and butane in the high temperature cycle. The technology will recover low temperature (30 °C) waste heat from industrial processes and upgrade to high temperature heat for applications such as drying, sterilization, pasteurization and other heat demands within 100 – 120 °C. The heat pump will replace equivalent capacities of existing cooling towers and boilers or direct electricity heating systems in the industry. A modified compressor prototype was implemented for the high temperature cycle and was found to have an average total compressor efficiency of 71 %. The heat pump had a heating COP up to 2.6 and a combined heating and cooling COP of 4.1 for a temperature lift of 80 °C.

Keywords**:** Compressor development, Energy efficiency, Process integration, Waste heat recovery

1. INTRODUCTION

The focus on clean and efficient energy systems for a sustainable future requires new, innovative and improved solutions to meet the energy demands especially in industrial processes. Industrial processes such as distillation, pasteurization, drying and others have heat demands at high temperatures that is currently provided by direct electric heating or electric/steam boilers. Such systems are neither efficient in direct electric heating nor clean due to fossil fuel combustion in boilers. Incidentally, many industrial processes have large amounts of waste heat discharged to the environment. A heat pump can recover and upgrade this waste heat to useful heat at high temperatures.

Heat pumps using the vapour compression cycle is a well-established technology dating back over a hundred years. The predominant use of the technology is in refrigeration systems for cooling and heating at low temperatures. In recent years, it has been applied to higher temperature heating up to 90 °C with the use of ammonia and CO_2 as working fluids (Austin & Sumathy, 2011; Ayub, 2016; Berntsen *et al.*, 2014; Hoffmann & Pearson, 2011; Neksa, 2002; Neksa *et al.*, 1998; White *et al.*, 2002). To extend the heat delivery temperature beyond 90 °C, fluids such as hydrocarbons (HCs) and hydrofluoroolefins (HFOs) have been considered. These fluids have both the thermodynamic properties for high temperature heat pumps (high critical temperature at low pressure) and they are environmentally friendly (Bamigbetan *et al.*, 2016, 2017).

This study experimentally investigates the use of hydrocarbons as a working fluid for high temperature heat pump (HTHP) up to 110 °C. The heat pump test facility is a 20 kW heating capacity heat pump designed as a cascade configuration with propane in the low temperature cycle (LTC) and butane in the high temperature cycle (HTC). The HTC is installed with a modified compressor that allows compressor suction and discharge temperature up to 80 and 140 °C respectively. A case study of a milk production plant that requires a process water stream to be heated from 95 °C to 115 °C but with waste heat at low temperature of 30 °C from a cooling glycol loop is used to set the boundary conditions of the test rig. The investigation is conducted for a case of

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high temperature difference in the heat sink from 30 ºC to 110 ºC applicable to pressurized hot water production.

2. METHODOLOGY

A theoretical analysis was conducted to determine the right natural working fluid that gives the best coefficient of performance and the most suitable cycle configuration for a high temperature lift (greater than 80 K) heat pump. The potential to improve existing technology influenced the selection process in the analysis. Two fluids, propane (R290) and butane (R600), were selected for a cascade configuration heat pump with propane in the low temperature cycle and butane in the high temperature cycle (Bamigbetan *et al.*, 2016; Stavset *et al.*, 2014).

Further analyses were conducted using dynamic simulation software to evaluate different scenarios of operating conditions, sizing of components for a model test facility, safety consideration and required technological limitations in the development of the heat pump (Bamigbetan *et al.*, 2017). Using this simulation results, a test facility was built with a model size of 30 kW heating capacity to demonstrate the possibilities for high temperature heat delivery.

2.1. The Model Compressor

The heat pump was sized and designed to operate with components that are commercially available in the market as seen in Table 1. However, the operating temperatures at the compressors are higher than the design operating profile. The R290 compressor is a standard compressor manufactured by DORIN with a profile of evaporating temperature at maximum 10 ºC and condensing temperature at maximum 65 ºC. Though the simulation results indicate a higher evaporating temperature up to 20 ºC, the compressor was used as available to test and push the boundaries of operation.

Table 1: List of Major Components in the heat pump

Component	Manufacturer	Model type
	R290, Low Temperature Cycle (LTC)	
Compressor included Oil Heater	Dorin	HEX551CC
Evaporator R290	Kaori	K095 x22
Suction Accuumlator LTC with IHX	CARLY	LCYE 69S
Receiver 4,4 ltr, 2 Sight Glases	KLIMAL	RCO 139.40.4.80
Heat Exchanger R290/R600	Kaori	K095 x 90
Expansion Valve	Carel	E2V14
	R600, High Temperature Cycle (HTC)	
Compressor included Oil Heater	Dorin	HEX1500CC
Condenser R600	Kaori	K070 x 60
Suction Accuumlator HTC with IHX	CARLY	LCYE 69S
Receiver 4,4 ltr, 2 Sight Glases	KLIMAL	RCO 139.40.4.80
Expansion Valve	Carel	E2V14

The R600 compressor will have to operate at a relatively higher temperature profile outside the design conditions. The compressor is therefore modified to enable suction temperature up to 80 ºC and discharge temperature up to 160 ºC with a set point of 140 ºC. The modified compressor is shown in Fig. 1 at the installation site.

The modifications include:

1. Designed with an external manifold
2. Special discharge temperature sensor: up to 160 °C
3. High capacity motor: the motor is sized 25 % larger than required.
4. Thermal protection: the thermistors are set at 140 °C
5. Designed with internal thermocouples
6. Lubrication selection due to viscosity at higher temperature

The compressor manufacturer DORIN made the modifications according to the required specifications

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Figure 1: Installation of the modified R600 compressor for high temperature cycle

2.2. The test rig facility

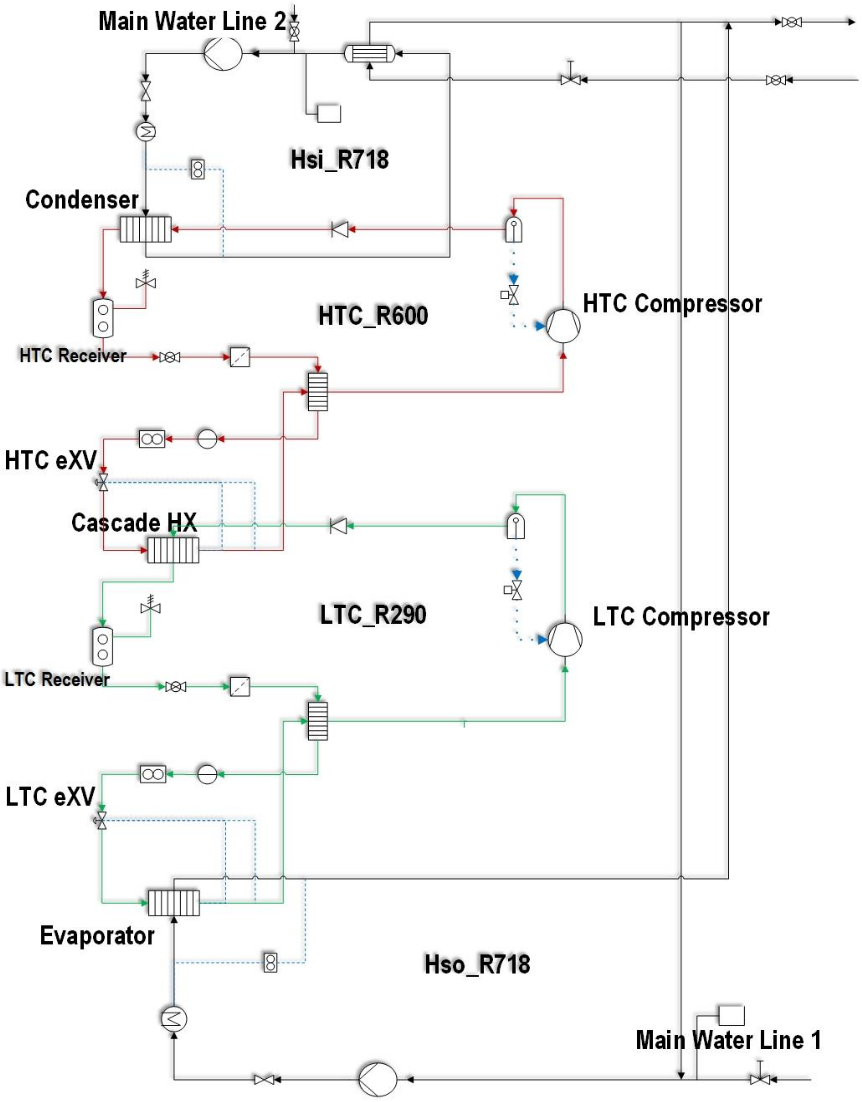

Figure 2: Schematics of the test rig showing instrumentation and laboratory cooling loop

Fig. 2 shows schematics of the test facility as currently installed. The test rig was built with a heat sink and source loop designed to minimize energy usage in the laboratory where it will be installed. The laboratory has

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a central cooling loop at 16 °C that flows through a heat exchanger, the heat sink discharges heat to the cooling line. Part of the heat is recovered to the heat source for continuous operation.

2.3. Instrumentation and logging

Table 2: Instrumentation sensors and their accuracy

Sensor type	No. of Units	Accuracy	Range
Temperature transmitter	16	± 2.2 K	-
Pressure transmitter	8	± 0.2% FS BSL	0 - 30 barg
Flow meter	2	± 0.2 %	0.5 – 50 kg/min
Energy meter	2	± 2 %	0.015 – 1.5 m^3/hr

Table 2 shows the instrumentation on the test facility. Temperature sensors are installed at every state point of the cycle with pressure sensors before and after the compressors and the main heat exchangers. Both compressors are connected to frequency converters where power consumption and speed data are collected. The sensors data are automatically logged into computer system. The compressor power consumption and speed are manually inputted at stable operating points.

3. RESULTS

The test regime was developed to evaluate heat pump performance, compressor efficiencies and optimal operating parameters. Currently, the water loop design is restricted, therefore independent control of inlet conditions to both the heat source and heat sink is not possible. The tests are conducted for a case of high temperature difference in the heat sink from 30 °C to 110 °C as shown in Table 3. This will provide industrial processes with pressurized hot water for applications such as cleaning, sterilization and pasteurization. The test rig will be modified for other case studies for further studies.

Table 3: Operating conditions for the test regime

	Operating Conditions (°C)				Compressor Speed (Hz)	
	Heat Source		Heat Sink			
Evap.temp.	Inlet temp.	Outlet temp.	Inlet temp.	Outlet temp.	R600	R290
14 - 18	25 - 34	19 - 25	29 - 36	100 - 110	30 - 50	30 - 50

3.1. HTC R600 Compressor Temperatures

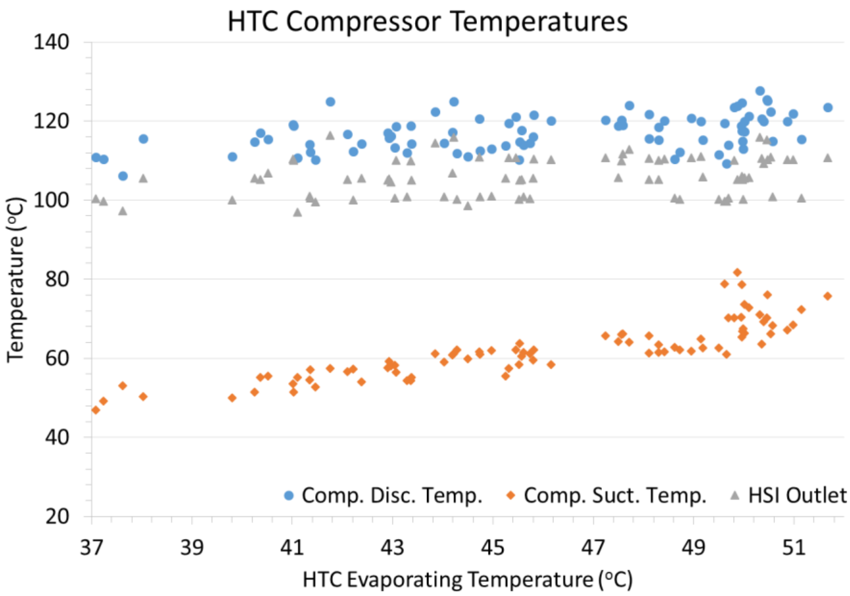

Figure 3: Compressor suction and discharge temperatures and heat sink outlet temperature plotted against the HTC evaporating temperature

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The R600 compressor discharge temperature has a value of 127.7 °C when the outlet temperature for the heat sink is 116.4 °C as shown in Fig. 3. There is a potential to increase the heat sink outlet temperature, as the maximum allowable temperature for the compressor discharge is 140 °C. The maximum suction temperature for butane compressor is 80 °C. This is required to effectively cool the electric motors and prevent too high discharge temperatures. At high speed (> 45 Hz) of the propane compressor and low speed (30 Hz) of the butane compressor, the suction temperature to the R600 compressor is higher than maximum. There is no observed benefit to the performance of the heat pump, and this operating region can be avoided.

3.2. Total compressor efficiency

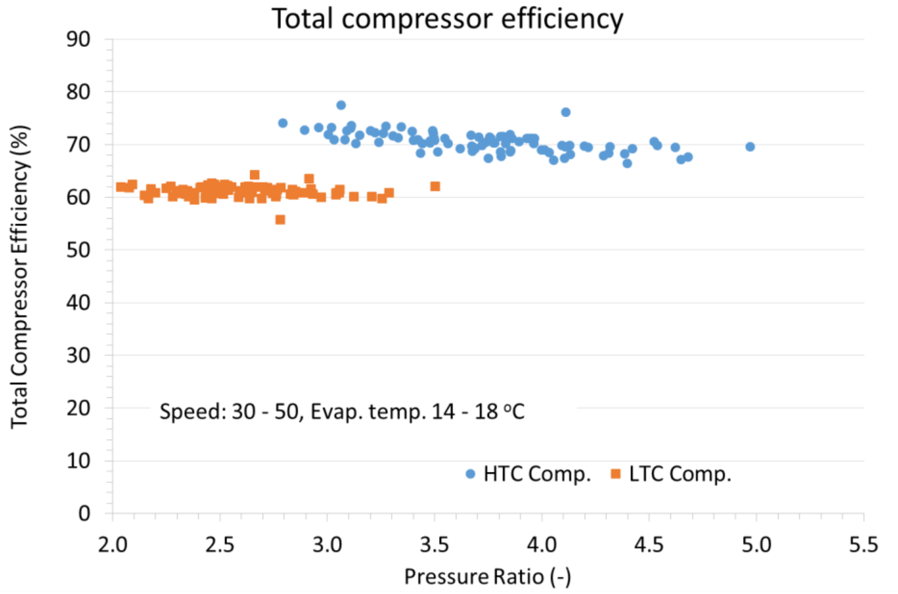

Figure 4: Total compressor efficiency relationship to the pressure ratio

The total compressor efficiency is calculated from the ratio of isentropic power to the power reading at the frequency converter. Fig. 4 shows that the model compressor for propane has an average efficiency of 71 %, while the propane compressor average efficiency is 61 % across the tested operating conditions. The propane compressor efficiencies appear not have a significant dependence on the pressure ratio. The efficiency of the butane compressor slight decreases as the pressure ratio increases.

3.3. Volumetric efficiency

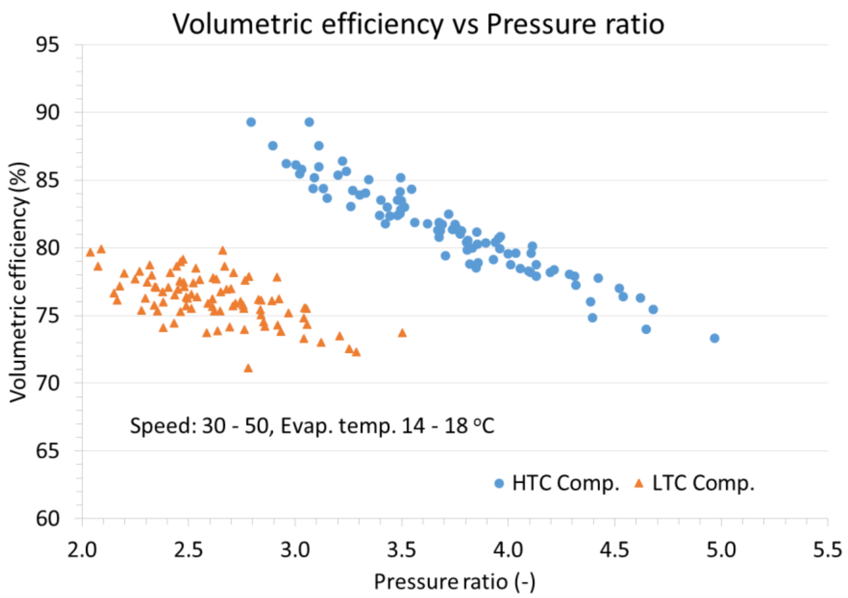

Figure 5: Volumetric efficiency relationship to pressure ratio

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Similar to the total compression efficiencies, the volumetric efficiencies of the butane compressor is higher than the propane compressor as shown in Fig. 5. Both values appear to be dependent on the pressure ratios of the compressors. The volumetric efficiencies decreases with increase in pressure ratio.

3.4. Coefficient of Performance

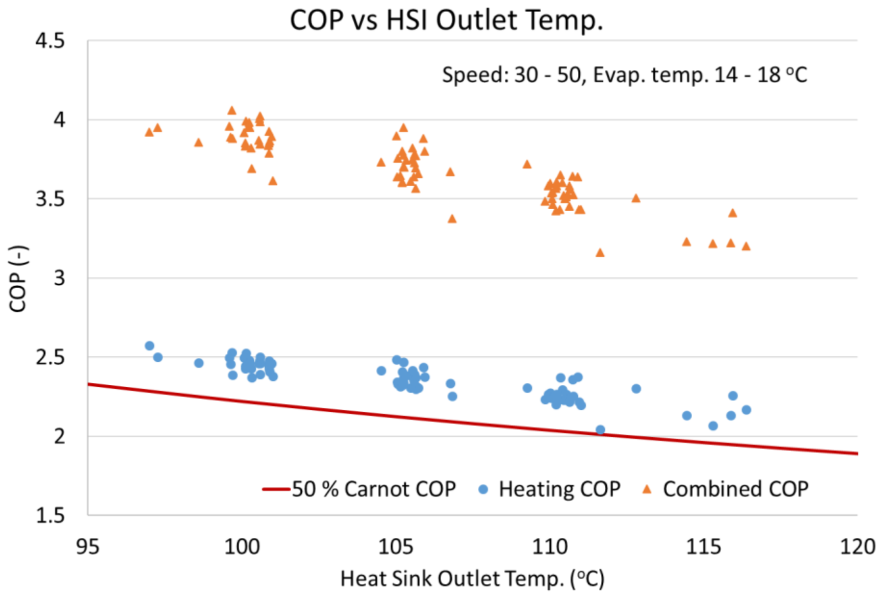

Figure 6: Overall Coefficient of Performance at different heat sink outlet temperature

Depending on the operating condition, the heating COP of the heat pump varies between 2.0 and 2.5. This is more than 50 % of the Carnot efficiency. The heat pump for the case study will not only deliver heat at high temperature that replaces the capacity of the electric/steam boiler, but also provide cooling by the removal of waste heat. The combined COP therefore takes total energy delivered by heating and cooling and has a value between 3.1 and 4.0. The combined COP is the ratio of the sum of heating and cooling capacities to the total work input at the compressors.

3.5. COP and the pressure ratio balance

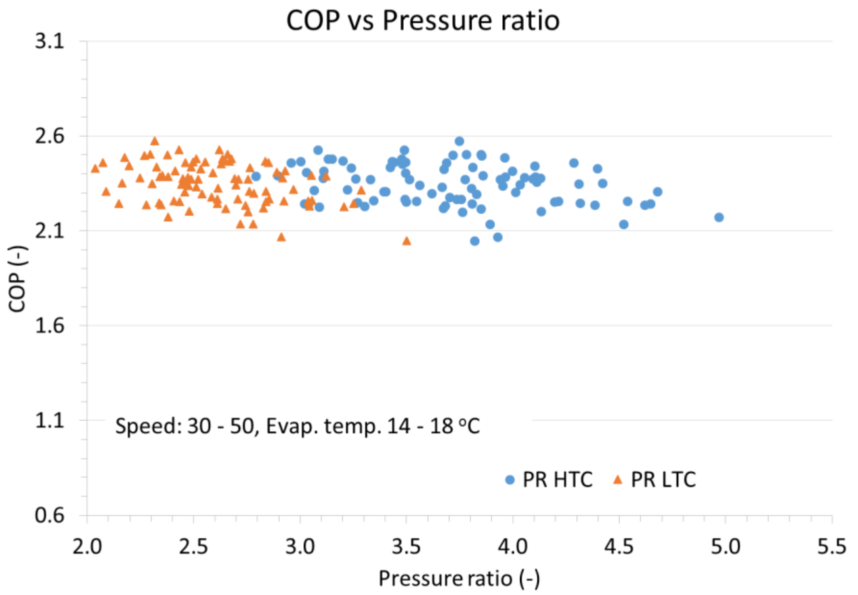

Figure 7: Optimal pressure ratio for HTC and LTC

There appears not to be a clear profile of the pressure ratios between the two cycles that gives the best overall COP of the heat pump from the plot of COP against pressure ratio shown in Fig. 7. This is probably due to the balance of energy consumption between the two hydrocarbon cycles and their similarities in property. As the

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pressure ratio of a cycle decreases, the amount of energy savings at the compressor is approximately equal to the increase in energy consumption of the other cycle compressor.

3.6. Model Compressor Evaluation

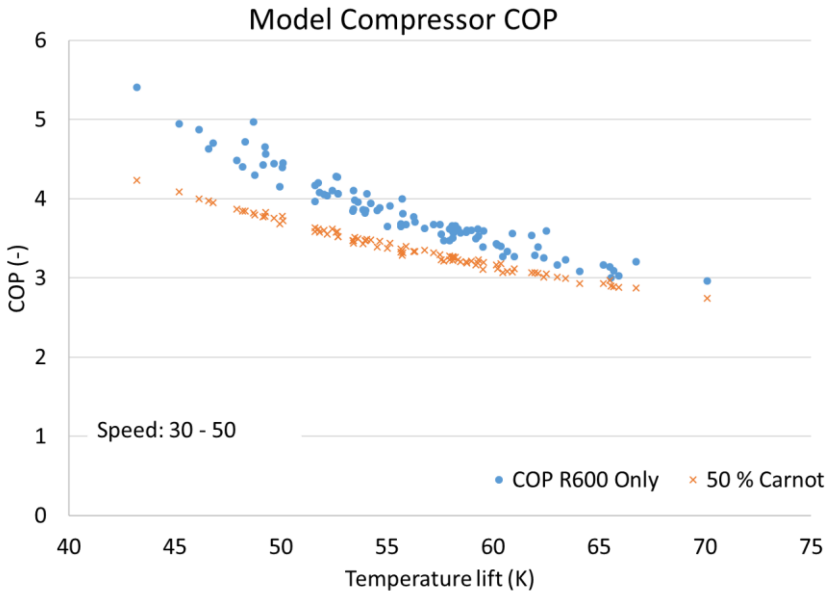

Figure 8: COP of model R600 compressor as an independent cycle

Fig. 8 shows the evaluation of the performance of the model butane compressor alone. It shows a COP between 3.0 and 5.0 depending on the operating conditions. The COP is above 50 % of the Carnot efficiency for the same temperature lift. A single cycle for HTHP can be considered if the heat source temperature is high enough (> 40 °C).

4. CONCLUSION

This paper experimentally investigates a 20 kW high temperature heat pump test rig. The heat pump is a cascade configuration cycle with propane in the LTC and butane in the HTC. The test rig has been operated up to 116.4 °C heat sink outlet temperature from a heat source inlet temperature of 34 °C. The heating COP of the heat pump is between 2.0 – 2.5, while the combined heating and cooling COP is 3.1 – 4.0 across the test operating conditions with an average lift of 80 °C.

A modified compressor was installed to operate at high temperatures up to 80 and 140 °C compressor suction and discharge temperature. The average of the total compression efficiency of the compressor is 71 % with a volumetric efficiency within 75 – 89 %. The COP of the HTC is 3.0 to 5.0 when considered as an isolated unit from the LTC.

Further studies on the test rig will be required to have a conclusive data point for the balance between the HTC and LTC loads that best optimize the heat pump. Modification are also required to allow flexible changes of the operating conditions for other industrial process applications with lower temperature glide in the heat sink.

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NOMENCLATURE

COP	Coefficient of Performance
LTC	Low Temperature Cycle
HC	Hydrocarbons
HTC	High Temperature Cycle

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