Experimental Investigation of a Hydrocarbon Piston Compressor for High Temperature Heat Pumps

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ABSTRACT

The global trend towards carbon neutral industries require sustainable and climate friendly heat supply. Heat pumps can meet this requirement, but the technical capability today often limit the heat supply temperature to below 90°C. The major challenge in the development of a heat pump that can deliver heat at high temperatures (115 ºC) to the heat sink is the operability and performance of the compressor. The compressor electric motor cooling, lubrication stability and material durability are important physical properties to evaluate for a compressor working at high temperatures. In this study, compressors operating with natural fluids such as hydrocarbons are focused. Hydrocarbons, like butane (with high critical temperature, 152 ºC), have become a viable alternative to synthetic working fluids for high temperature heat pumps. Some of the synthetic fluids will be phased down by the Kigali amendment to the Montreal protocol. Butane can operate with similar compressor technology that is familiar to propane as well as synthetic working fluids, due to the close thermodynamic properties.

This study experimentally investigates the performance of a prototype butane compressor adapted for high temperature heat pumps. The compressor has been fitted with modifications specifically for high performance and high temperature heat delivery. The compressor is installed in a 20 kW cascade heat pump with propane in the low temperature cycle and butane in the high temperature cycle. The prototype compressor development can be applied in a heat pump for drying, pasteurization, sterilization, low-pressure steam production, pressurized hot water production and others that are conventionally supplied with heat by the combustion of fossil fuel in boilers or direct electric heating. It may also recover waste heat from industrial processes, thereby increasing energy efficiency. Experimental results show that the prototype compressor has an average total compressor efficiency of 74 % while delivering heat at 115 ºC.

Keywords: Compressors efficiency, COP, Energy Efficiency, Heating Applications, Waste Heat Integration

1. INTRODUCTION

The development of clean energy systems for a sustainable future will require innovations and advancement in existing technology. In industrial processes, the heat demand is supplied by conventional heating systems like boilers or direct electrical heating. These systems are neither efficient in its use of energy resource nor are they environmentally friendly when fossil fuels are combusted in boilers or in the power production. The future of energy sustainability will require both the use of cleaner energy resources and better energy utilization systems. The availability of waste heat, which is a by – product in industrial processes provides an opportunity to upgrade heat at a low temperature to a higher temperature by the use of a heat pump to meet the heating demand (Johnson *et al.*, 2008; Sogut *et al.*, 2010). This heat integration with a heat pump will increase the industrial process heat utilization efficiency and reduce dependence on heat from fossil fuel combustion. These heat pumps will be operated with natural fluids with no adverse effect on the environment. Synthetic fluids with high ozone depletion potential (ODP) and high global warming potential (GWP) are no longer acceptable and are therefore phased down by the Kigali amendment to the Montreal protocol (Clark & Wagner, 2016).

The recent advances in compressor technology have created an opportunity to utilize heat pumps to meet high temperature (above 100 ºC) heating demands. Compressors represent the technological gap for high temperature heat pump implementation (Bamigbetan *et al.*, 2017b). Mass manufactured compressors, with a capacity range up to 1MW, have typical constrains such as: too high temperature at the discharge of the compressor that can damage the compressor mechanical components and degrade the lubricant and high fluid temperatures at the suction to the compressor, that results in insufficient cooling to the electric motor winding. The development of a compressor that can operate within these high temperature conditions with high efficiency can potentially make heat pumps a competitive alternative to conventional heating systems, and thus reduce both power demand and CO_2 emissions.

This paper reports on the development of a high temperature heat pump (HTHP) prototype compressor for heat delivery up to 115 ºC. The compressor is a single-stage semi-hermetic piston compressor with butane as the working fluid. It has been modified for high temperature operation. It is installed in the high temperature cycle (HTC) of a 20 kW heat capacity cascade configuration heat pump. Temperature, pressure and mass flow sensors have been installed to monitor various compressor parameters and its performance. The low temperature cycle (LTC) has propane as the working fluid and it serves as the heat source to the butane HTC.

2. TEST FACILITY DESCRIPTION

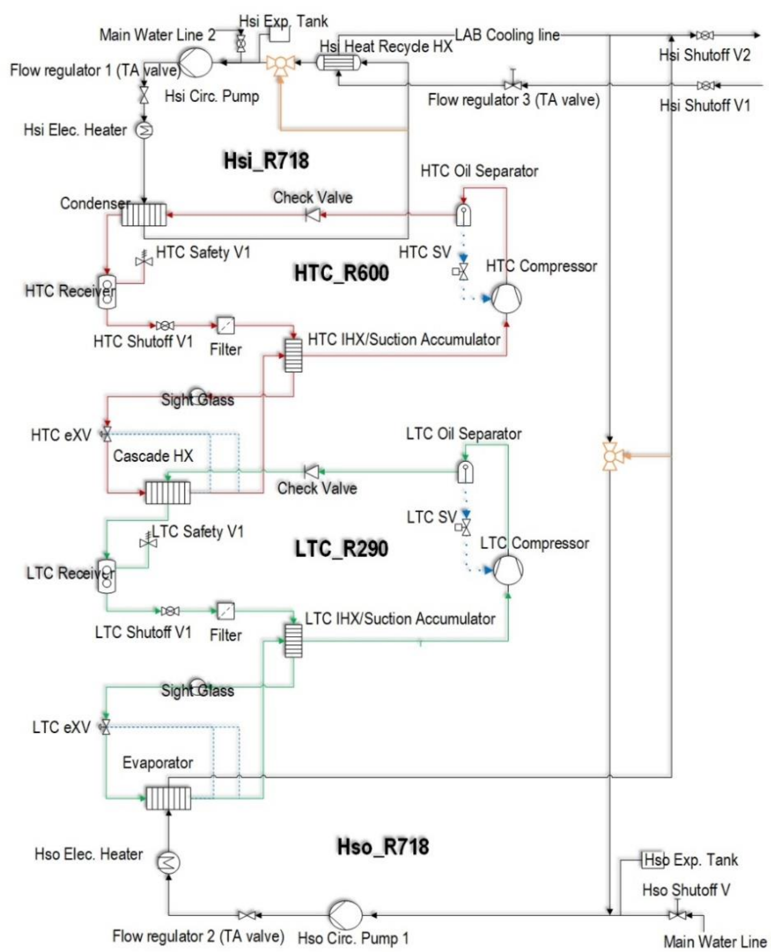

Figure 1: Schematics of the 20 kW cascade configuration heat pump with prototype butane compressor on the HTC

The prototype compressor for high temperature heating is installed on the HTC. The two heat pump cycles are connected by a cascade heat exchanger that function as the evaporator for the HTC and the condenser for the LTC. A heat sink cycle with pressurized water is connected to the heat pump at the HTC condenser. Heat from the heat sink is discharged to a laboratory cooling line through a heat recycle heat exchanger. A part of the heat is recovered by the LTC evaporator which serves as heat source to the heat pump. The heat source cycle is also connected to the laboratory cooling line. Two bypass valves are installed on the heat sink and heat source cycle for regulation of the inlet temperatures of both water cycles. The heat pump design is based on simulations reported by Bamigbetan *et al.* (2017a). The schematics of the test facility is shown in Figure 1.

The heat pump is operated to have a set operating condition at the heat sink inlet and outlet and at the heat source inlet to the condenser and evaporator respectively. Temperature sensors, pressure sensors, mass flow meters and energy meters are connected to the heat pump cycles at different state points. 11 temperature sensors and 2 pressure sensors are connected to the prototype compressor electric motor, suction and discharge heads, suction and discharge pipes and lubrication chamber to measure parameters of the compressors. A frequency converter controls the compressor speed and a mass flow meter is connected for capacity and efficiency calculations.

3. COMPRESSOR DESCRIPTION

The prototype compressor has a displacement of 48.82 m^3/hr at 50 Hz. It is a one-stage semi-hermetic 4 – cylinder piston compressor designed for operation in an explosive atmosphere. The compressor is installed with an internal crankcase heater, an oil sight glass and is equipped in the HTC with an external oil separator and oil return valve. The compressor is modified with an external manifold at the discharge connection point to effectively manage the high temperatures. The electric motor of the compressor is sized 25 % larger than would be typical for the compressor size. This will reduce heat generation at high loads on the electric motor. It will also allow better flexibility of the compressor for the varying test conditions. The compressor has a thermal protection set at 140 ^oC with a special discharge temperature sensor at 160 ^oC. A high-pressure switch is installed for safety at 28.6 bar. An image of the prototype compressor is shown in Figure 2.

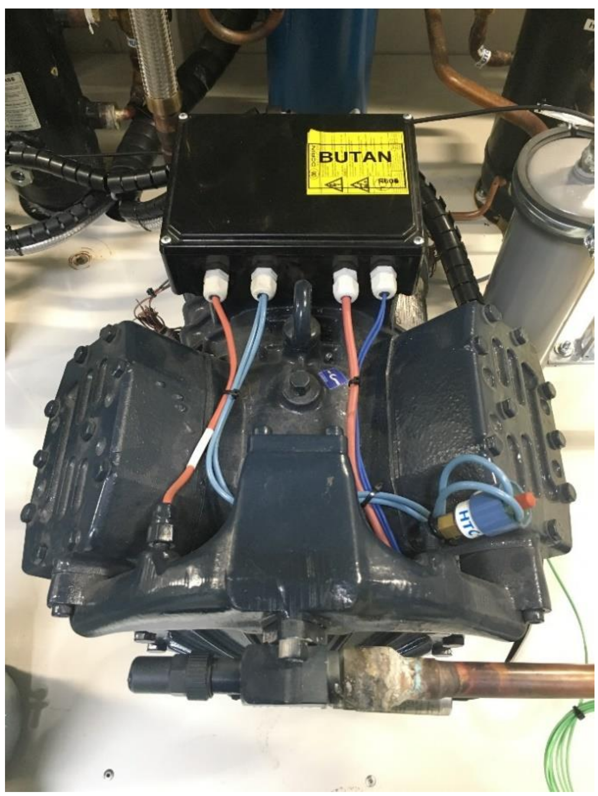

Figure 2: The prototype one-stage semi-hermetic 4 – cylinder piston compressor

6 internal thermocouples are pre-installed with the compressor to monitor temperature parameters across the compressor. More temperature sensors are connected to the discharge head of the compressor, the discharge piping 30 cm from the discharge head, the suction head after fluid heat exchange with the electric motor, the suction piping 10 cm from the compressor before heat exchange with the electric motor and in the lubrication chamber. Both suction and discharge lines are insulated with 30mm mineral wool insulation ($k \sim 0.06 W/(m * K)$). Pressure sensors are connected to both the suction and discharge heads. A Coriolis mass flow meter is installed in the cycle before the expansion valve and after the high-pressure receiver. The distribution of pressure and temperature sensors for the compressor is shown in Figure 3. The sensor accuracies and other statistical data is presented in Table 1.

In addition, the compressor is connected to a frequency converter to be able to vary the speed between 30 – 50 Hz. A lubricant tap – off point is installed for sampling of the lubricant and to evaluate the condition of the mechanical parts. All the sensors are connected to a data logger for processing. The compressor lubrication is recirculated back to the compressor by an oil separator. A timed valve controls the lubrication flow rate.

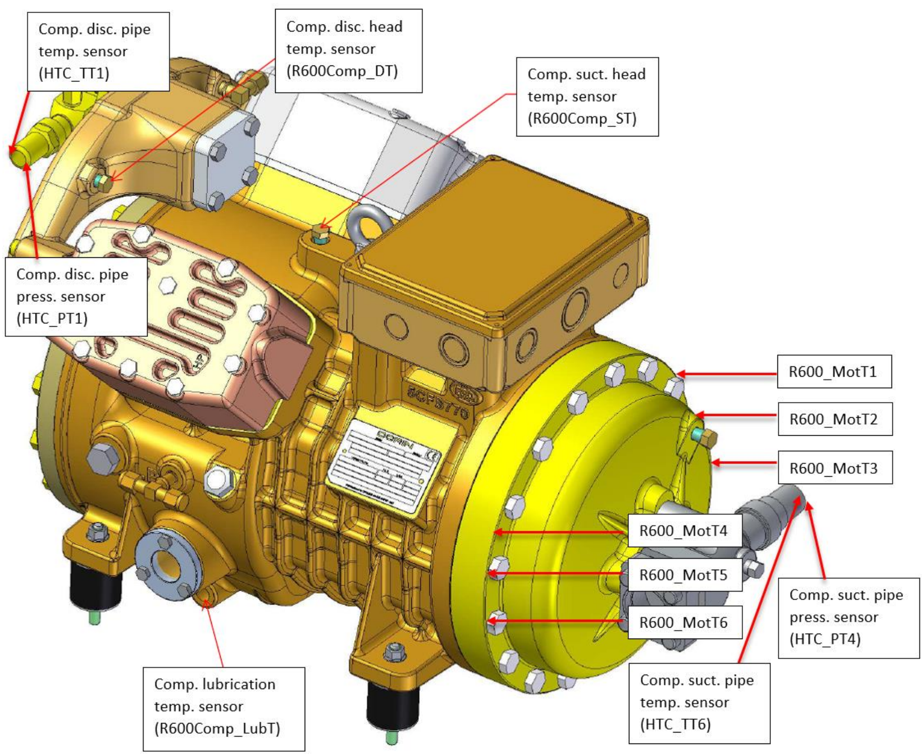

Figure 3: Pressure and temperature sensors distribution on the prototype compressor

Table 1: Instrumentation sensors and their accuracies

Sensor type	Type	No. of Units	Accuracy	Range
Temperature transmitter	Thermocouple K Type	5	± 2.2 K	-
	Thermocouple J Type	6	± 2.2 K	-
Pressure transmitter		4	± 0.2% FS BSL	0 - 30 barg
Flow meter		1	± 0.2 %	0.5 – 50 kg/min

The temperature, pressure and flow meter sensor readings are recorded 5 times per second over a steady state period of operation of the heat pump. The results presented are values averaged over a 10 minutes duration. Standard deviation values for 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 is directly read from the frequency converter during the steady state duration.

4. RESULTS

4.1. Compressor Efficiency

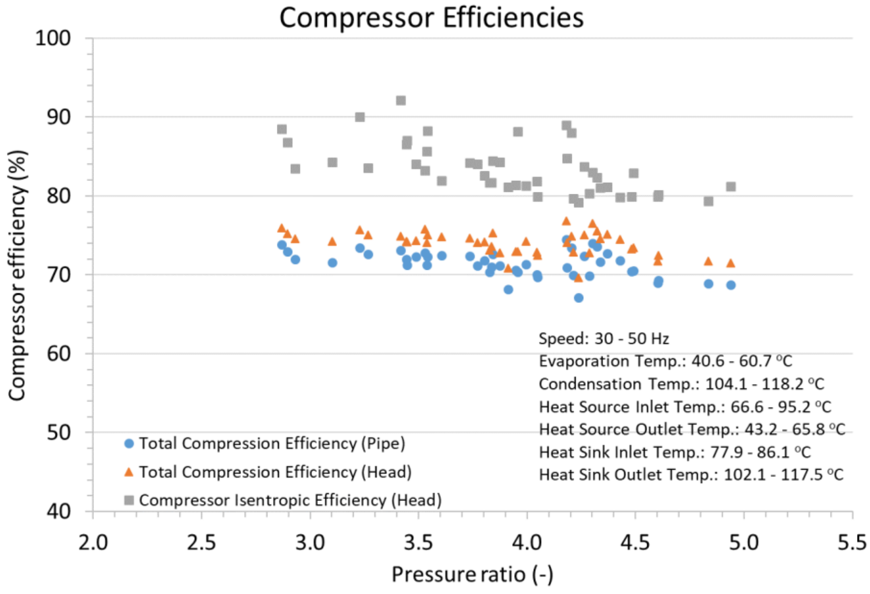

Figure 4: The efficiencies of the prototype compressor

The total compression efficiency is a measure of the performance of the compressor when all losses (mechanical/friction, electrical, motor, heat losses etc.) are considered. It is calculated as the ratio of the power input by isentropic compression of the fluid to the power consumed by the electric motor as described in Equation (1). The compressor isentropic efficiency is calculated as the ratio of the power input by isentropic compression of the fluid to the power input by the actual compression of the fluid at the same suction and discharge pressures as described in Equation (2).

The compressor is considered as both a separate unit for compression (without an electric motor) and a combined unit (with the electric motor) for analytical purposes. It therefore has 4 temperature sensors used for efficiency calculations. 2 sensors are installed at the suction (HTC_TT6) and discharge (HTC_TT1) piping to the compressor, with the other two installed at the suction (after electric motor heat exchange, R600CompST) and discharge (R600CompDT) head of the compressor manifold as shown in Figure 3.

$$\eta_{total} = \frac{m \cdot (\boldsymbol{h}_{disc,isen} - \boldsymbol{h}_{suct})}{Electric\ motor\ power\ consumption} \tag{1}$$

$$\eta_{isen} = \frac{m \cdot (\boldsymbol{h}_{disc,isen} - \boldsymbol{h}_{suct})}{m \cdot (\boldsymbol{h}_{disc} - \boldsymbol{h}_{suct})} \tag{2}$$

Figure 4 shows the compressor efficiencies plotted against the compressor pressure ratio. The total compression efficiencies, which is calculated from both the temperature sensors on the compressor suction and discharge at the pipes and heads, are 71 % and 74 % respectively on average across the varied operating conditions. The compressor isentropic efficiency calculated from the temperature sensor in the compressor heads is on average 84 %.

4.2. Compressor Volumetric Efficiency

The volumetric efficiency of a compressor is defined as the ratio of the actual swept volume of vapour sucked at the compressor suction to the theoretical swept volume (stroke volume) that would have been if the clearance volume were not present. The actual swept volume is calculated as the product of the specific volume at the compressor suction and the mass flow rate of the vapour. It is expressed as shown in equation (3).

$$\eta_{vol} = \frac{Actual\ swept\ volume}{Theoretical\ swept\ volume} \tag{3}$$

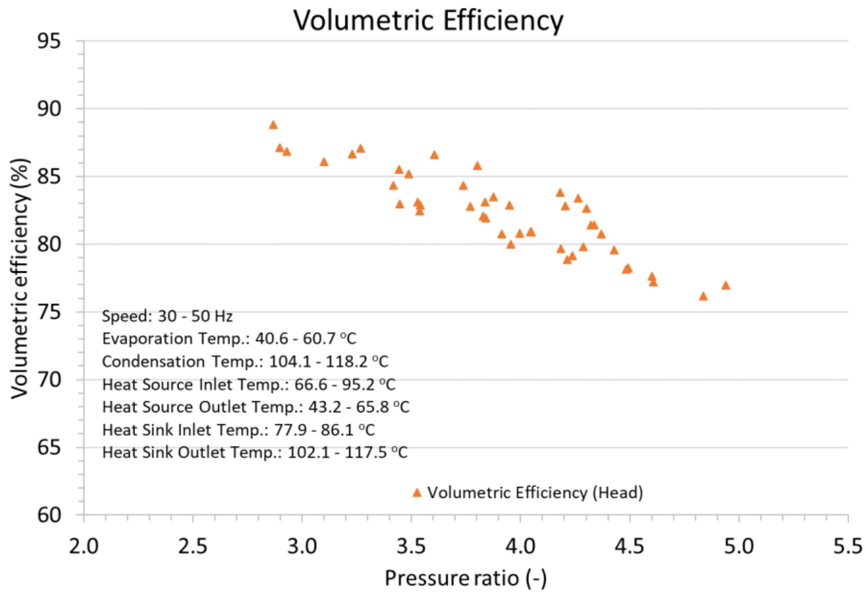

Figure 5: The volumetric efficiency of the prototype compressor

Figure 5 shows the volumetric efficiency for the prototype compressor. Similar to the compressor isentropic efficiency, the volumetric efficiency is calculated by the head temperature sensors. The volumetric efficiency increases with decrease in pressure ratio.

4.3. Compressor Temperature

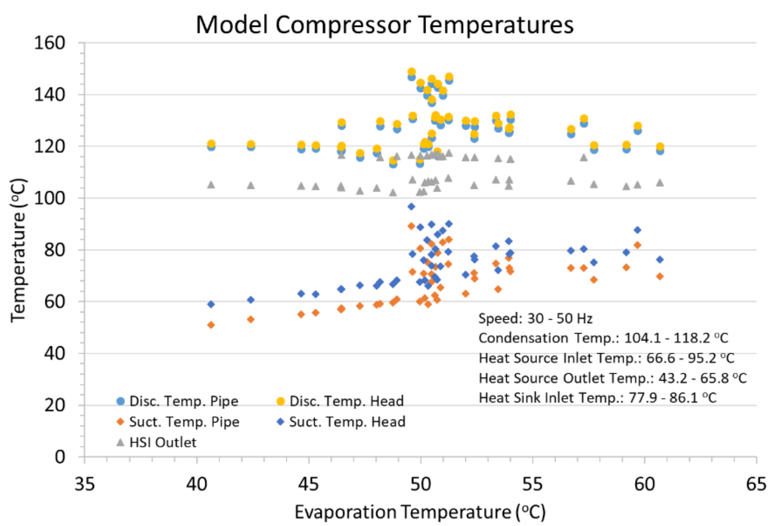

Figure 6: Compressor suction and discharge temperatures of the prototype compressor. Heat sink outlet temperature included as reference. Values are plotted against the evaporation temperature

The temperatures at the compressor suction and discharge (both head and pipe) are mostly below the maximum set (80 ºC for the compressor suction and 140 ºC for the compressor discharge) for the prototype compressor, as shown in Figure 6. The compressor discharge head and pipe temperature values have an average difference of 1.7 K that represents heat loss along the pipe. The compressor suction head and pipe temperature values have an average difference of 6.9 K. This value represents the average amount of heating of the suction vapour by the electric motor before actual compression. It is the equivalent amount heat rejected by the electric motor as cooling.

At a heat sink outlet temperature of 115 ºC, the discharge temperature is 127 ºC on average. The compressor suction temperature progressively increases with the evaporation temperature. Some operating conditions with high suction temperature (above 80 ºC) leads to high discharge temperature (above 140 ºC). These conditions are characterized by high superheat values (above 25 K) at the HTC evaporator. This is caused by operating the LTC propane cycle above design capacity relative to the HTC cycle design capacity. There are no advantages to operating the HTHP cycles in such a condition and it can be generally avoided.

4.4. Compressor Pressure

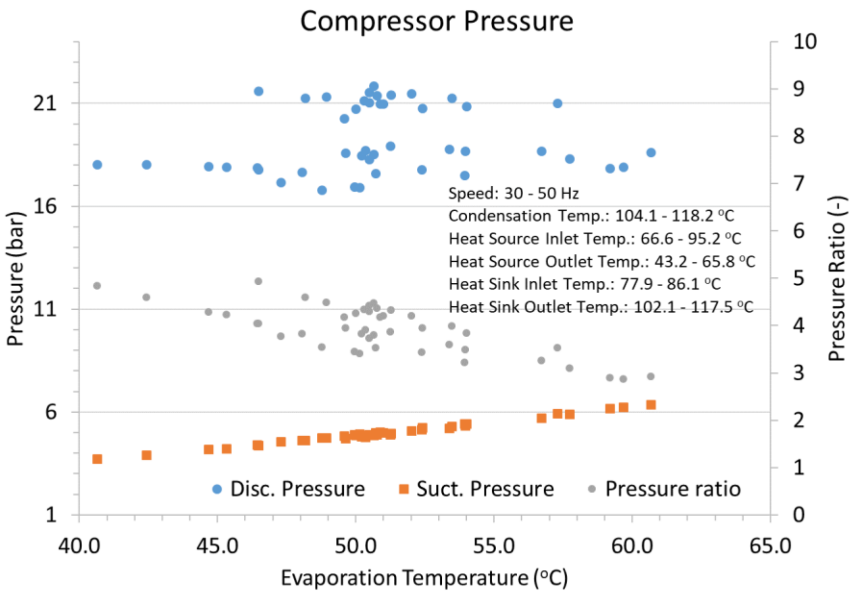

Figure 7: Compressor suction, discharge pressure and pressure ratio of the prototype compressor. Values plotted against evaporation temperature.

Figure 7 shows the prototype compressor suction and discharge pressure and pressure ratio. The maximum discharge pressure is 21.85 bar and the minimum suction pressure is 3.73 bar across the varied operating conditions. The maximum pressure ratio is 4.94. The values of the suction and discharge pressure and pressure ratio of the compressor are within accepted design parameters.

4.5. Compressor Oil Temperature

The crankcase oil temperature of the prototype compressor is measured with a thermocouple installed in the oil tap – off port. The suction vapour to the compressor cools the oil. Figure 8 shows the compressor oil temperature plotted against the compressor suction temperature. The oil temperature increases with higher suction temperatures. The region in the plot with the highest oil temperature represents the operating condition where the suction temperature to the compressor is superheated to a value above 80 ºC. There have been no observable changes to compressor performance with over 170 hours of compressor run time at this high temperature oil operation.

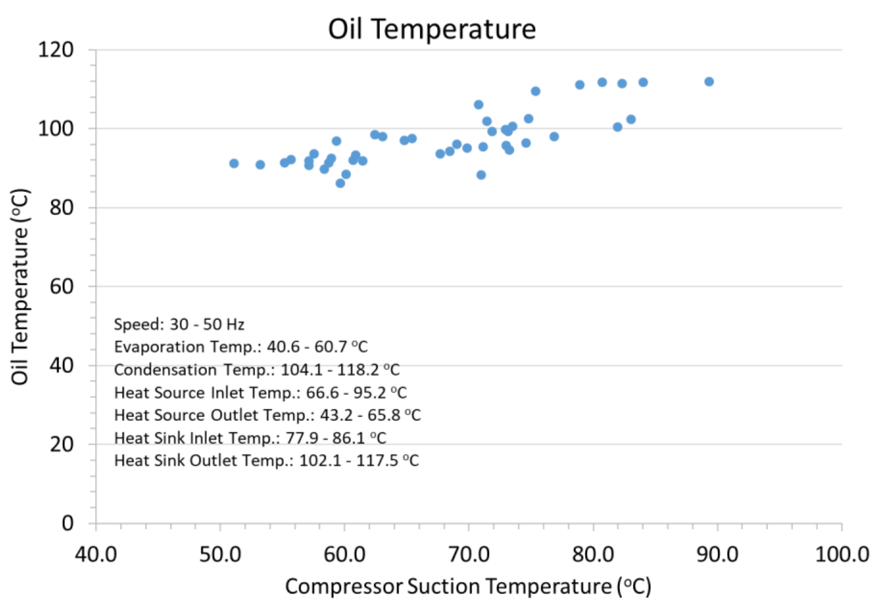

Figure 8: Oil temperature of the prototype compressor

5. CONCLUSION

This paper presents results from the experimental investigation of a prototype compressor for high temperature heat pumping. The prototype compressor is mounted in the high temperature cycle of a cascade configuration heat pump. The heat pump is a 20 kW hydrocarbon HTHP designed to recover waste heat and increase heat utilization efficiencies in industrial process applications. The prototype compressor is instrumented with 11 temperature sensors and 2 pressure sensors to record the operational parameters. In addition, a mass flow meter and a frequency converter are installed for energy calculations and speed variations.

The results shows an average of 74 % for the total compression efficiency (suction head) across all operating conditions tested. At a heat sink outlet temperature of 115 ºC, the discharge temperature of the prototype compressor is 127 ºC on average. The temperature can exceed 140 ºC if the suction vapour is excessively superheated above 80 ºC. There is no advantage to operating at this condition. The discharge and suction pressures are below 22 bar and above atmospheric pressure respectively, while the volumetric efficiency varies from 75 % to 85 % depending on the compressor pressures.

Further experiments are required to evaluate the performance of lubricant with respect to viscosity, stability, oil return and the possible presence of wear particles from the compressor. Evaluation of the prototype compressor should also be extended to higher heat delivery temperatures, possibly up to 125 ºC. Higher evaporation temperatures can also be tested with a better control of the superheat to the compressor suction.

NOMENCLATURE

COP	Coefficient of Performance
DT	Discharge Temperature
eXV	Expansion Valve
GWP	Global Warming Potential
HSI	Heat Sink
HSO	Heat Source
HTC	High Temperature Cycle
HTHP	High Temperature Heat Pump
LTC	Low Temperature Cycle
LUBT	Lubrication Temperature

ODP	Ozone Depletion Potential
PAG	PolyAlkylene glycol
PT	Pressure Transmitter
ST	Suction Temperature
TT	Temperature Transmitter
V	Valve
η_{total}	Total compression efficiency
η_{isen}	Isentropic efficiency
η_{vol}	Volumetric efficiency
$\boldsymbol{h}_{suct}$	Enthalpy at compressor suction
$\boldsymbol{h}_{disc}$	Enthalpy at compressor discharge
$\boldsymbol{h}_{disc,isen}$	Enthalpy at compressor discharge (isentropic)
k	Thermal conductivity
m	Mass flow rate

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