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Overview on HCFO-R1233ZD(E) use for high temperature heat pump application

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ABSTRACT

A high temperature heat pump has substantial potential for industrial waste heat recovery and upgrading of which there is a potential of 370 TWh in Europe. R1233zd(E) is a potential alternative to R245fa due to low GWP and ODP to achieve high temperatures in a range of 140°C and above. This paper presents review on R1233zd(E) and comparative thermodynamic simulation of high temperature heat pump based on selected compressor and between temperature range of 85°C to 135°C. In addition, details about high temperature heat pump test facilities and discussions on operational experiences and difficulties of high temperature heat pump test-rig development in terms of components and their limitations has been discussed. The initial experimental results from test rig are presented to demonstrate such operational challenges in terms oil cooling and expansion valve. Outcomes of this paper will provide indicators for future applications of high temperature heat pumps using alternative refrigerants.

Keywords: High temperature, heat pump, R1233zd, Industrial heat pump, HCFO

1. INTRODUCTION

Energy efficiency and waste heat recovery in industrial sector has a huge potential of 370 TWh (waste heat) per year in Europe alone (Panayiotou et al., 2017) which could help in a drive towards greenhouse gases emission (GHGs) reduction and to protect ozone layer. To support this drive, widely used refrigerant such as R134a, R410A, R407C and R245fa should be phased out due to high global warming potential (GWP) and ozone depletion potential (ODP) as a part of F-gas regulation in the EU and Montreal Protocol.

Table 1 Important parameter comparison between R1233zd(E) and R245fa (Yang et al., 2018)

Parameters	*R12233zd(E)*	*R245fa*
Chemical formula	$CF_3CH{=}CHCF_3$	$CF_3CH_3CF_2$
Pc (kPa)	3570	3650
Tc (°C)	165.6	154.01
Boiling point (°C)	17.97	14.81
Slope	Dry	Dry
ODP	0.00034	0
GWP100yr	7	1030
Atmosphere life time	7.7	0.07
Flammability	Non-flammable	Non-flammable
ASHRAE std 34 safety class	A1	B1

R1233zd(E) is one of the promising alternatives to replace R245fa which is widely used for waste heat recovery in Organic Rankine Cycle (ORC) or High Temperature Heat Pump (HTHP). R1233zd(E) belongs to HCFO family and has comparatively low GWP and ODP. Although R1233zd(E) contains one chlorine atom in the molecule, the contribution to stratospheric ozone depletion is expected to be negligible because of its very short-atmospheric lifetime. Table 1 shows

main parameter comparison between R1233zd(E) and R245fa. It is evident that R12233zd (E) can be used for HTHP up to 150°C and as a direct replacement of R245fa.
There are limited investigations related to R1233zd(E) use for HTHP in scientific literature and most investigations/literature are very recent. Fig. 1 illustrates the number of publications in the SCOPUs (75) and Web of Science (63) online database with search keyword "R1233zd" (keywords, title and abstracts). In addition, similar search was carried out in SCOPUS using same search key word in all field to find any literature that mentioned this word and found around 200 publications. Hence, scientific investigation/publication are at very early stage related to R1233zd(E).

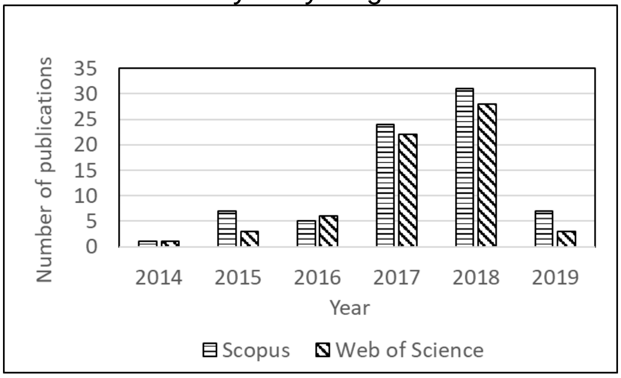


Figure 1: Publications related to R12333zd(E)

This paper aims to gather information related to R1233zd(E) use and application for HTHP through scientific literature. Thermodynamic simulation and comparative analysis for R245fa and R1233zd(E) is presented based on selected compressor and temperature range. In addition, initial experience of HTHP operation and preliminary results from HTHP test rig is presented for analysis purpose.

2. LITERATURES ON R1233ZD(E)

Detailed literature search on R1233zd(E) showed that most articles focused on properties of refrigerant, heat/flow transfer and ORC application. Most of ORC and/or HP investigations were based on mathematical model/simulation/theoretical analysis. Table 2 shows numbers of articles based on subject area and type of investigation. Overview and investigation of properties R1233zd(E) and other low GWP refrigerant has been widely covered by Bobbo et al. (2018). There were four investigations related to heat pump applications. For example, Ju et al. (2017) assessed heat pump water heater with mixture of R1233zd(E) with R290 and R1270 showed that R1233zd(E) can help to overcome the shortcoming of flexibility of HCs and modelling result showed 2 to 10 % higher COP compared to R22 or R134a and irreversibility of system components varies (except compressor) with mass fraction of R1233ZD(E) in a blend. Zuhlsdorf et al. (2018) showed that in a case of booster heat pump, mixture of R1233zd(E) and R1234yf would give higher COP compared to R1233zd (E) alone at design conditions. Although it was still lower compared to other refrigerant mixtures such as Iso-Butane and Pentane. Arpagaus et al. (2018) have presented extensive overview on HTHP showing waste heat recovery potential, state of the art, research status and refrigerant. A list of industrial HTHP was presented (28 kW to 20 MW) but none of industrial HP uses R1233zd(E) yet. Potential benefits of R1233zd(E) in terms reduced compressor size and trade-off between VHC and COP for HTHP application have been discussed by Frate et al. (2019) and Bamigbetan et al. (2018).
However, long term stability, thermal performance and interaction with other material (e.g. oil, copper, etc) is also important for R1233zd(E). In terms of suitable oil for R1233zd(E), some oil manufactures and researchers have shown suitability (e.g. miscibility) with POE (e.g. Fuchs, Climalife) whereas others suggest the use of mineral oil (e.g.Honeywell) (Suemitsu, et al., 2017) (Majurin, et al., 2017) or both (e.g. Arkema). In addition, Eyerer et al. (2018) showed an interesting analysis on the subtility of polymers (used on O-ring etc.) and their experimental and theoretical results showed that PTFE is the most compatible polymer which can be used with R1233zd(E) whereas special attention should be given if other polymers are used. Hence selection of suitable oil and other materials is crucial for R1233zd(E).

Table 2 Type of investigation and numbers of literatures

Are of investigation		Type of investigation	
Main subject	Sub-subject	Experimental	Theoretical/ modelling/simulation
Flow regime	pressure drop	4	2
Heat transfer	flow boiling, film condensation, nucleate boiling, condensation, supercritical pressure, high temperature	12	
Ejector	vapour-liquid, critical/sub-critical model, refrigeration system working fluid		3
ORC	scroll expander, various application, cycles configuration, with solar, with ejector, evaluation system, fluid comparison	5	26
Material	compatibility between refrigerant and polymer	1	
Properties	liquid viscosity, surface tension, solubility, diffusion, thermal conductivity, PvT, film thickness, heat capacity, speed of sound, liquid density	13	
Heat pump	booster, water heater (ref mixture), centrifugal chiller		3
Combined system	ORC+HP, absorption compression, trigeneration,		3

3. EXPERIMENTAL SETUP AND THERMODYNAMIC ANALYSIS

3.1 Test setup description

A HTHP test rig was developed in order to understand performance and behaviour of R1233zd(E) for high temperature application. The heat pump was designed at T_{con}=125°C and T_{evp}= 50°C with SH=20K (evaporator + liquid-suction heat exchanger) and SC=9K. At design condition, HP can provide 13.8 kW of condensing capacity. Commercially available components were used for the development of the test-rig. Figure 2 shows HTHP test-rig which was developed at Centre for Sustainable Technologies (Ulster University). Left side of test rig is accommodated with refrigeration side components such as compressor, condenser, evaporator, EEV etc whereas right side of test rig is for heat transfer fluid (water/oil) temperature management which helps to maintain constant temperature at outlet of condenser and inlet of evaporator as required. Table 3 shows main components used in development test-rig. Due to high temperature, thermal oil was used as a heat transfer fluid on secondary side of condenser whereas water was used on the evaporator secondary side. The compressor was driven by variable speed drive which can manage speeds between 750 to 1750 rpm for the compressor.

In order to monitor performance of the heat pump system variables such as temperature (refrigerant, oil and water)(inline and surface PT100), flow (refrigerant, oil and water), pressure (refrigerant), power (motor) were measured with sensor accuracy of ±0.2°C, ±1% (electromagnetic flow meter) / ±1.5% (pulse meter for oil), ±1% and ±1% respectively. All data were measured at interval of 30s using two data acquisition system and stored in a dedicated PC for data analysis purpose.

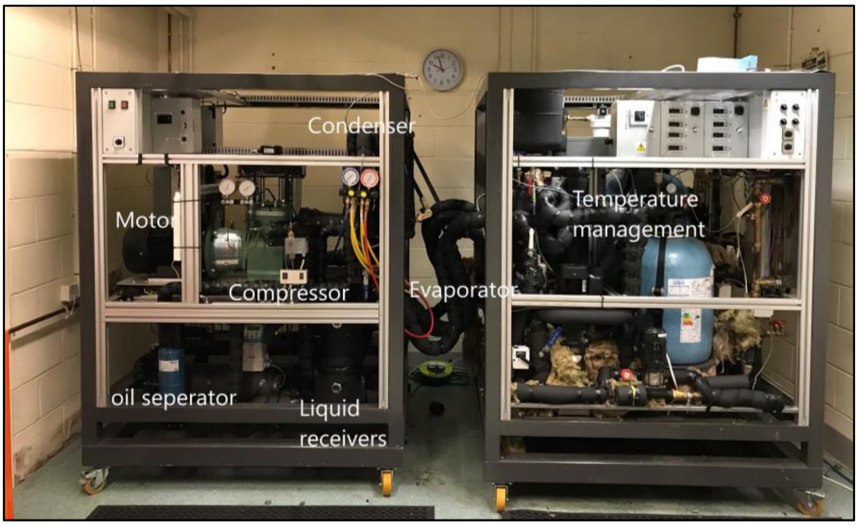


Figure 2 Test-rig development for HTHP

Table 3 List of components used for test-setup

Components	Details
Compressor	Bitzer: Open type Dis: 28 m3/h @50 Hz (1450 rpm)
Drive	EMI motor with Danfoss FC103 drive
Condenser	SWEP BPHE BT25Thx50
Evaporator	SWEP BPHE N80Hx26
Expansion valve	Danfoss EEV ETS12.5 with EKC316A
Oil separator	ESK OS22
Receiver	Bitzer FS102
Heat transfer fluid	Water and Therminol66

3.2 Test methodology and thermodynamic analysis

3.2.1 Test methodology

The system performance will be evaluated for condensing temperature between 85 to 135°C at an interval of 10K with fixed evaporation temperature of 50°C. First test will be carried out at design condition were flow rate on oil and water side will be set and it will be kept constant for other test condition in order to obtain change in heating/cooling capacity with respect to temperature difference. Similar test will be carried for evaporation temperature between 45 to 85°C at an interval of 10K while keeping the condensing temperature at 105,115,125 & 135°C. Compressor speed will be maintained at 50 Hz (1450 rpm). All variables such as temperature, pressure, flow rate and electric power will be measured on refrigeration/water/oil side.

3.2.2 Thermodynamic analysis

In order to estimate system performance and for comparative analysis, thermodynamic simulation is carried out for R1233zd(E) and R245fa at same temperature lift condition as mentioned above. The following assumptions were made for thermodynamic analysis:

- Heat dissipation with the ambient is ignored.
- Pressure drops in the evaporation and condensation processes are negligible.
- The compression process is adiabatic while the throttling process in expansion valve is isenthalpic.
- The heat transfer in the condenser or evaporator is of counter-flow
- The refrigerant at the outlet of the condenser is saturate liquid, and at the outlet of the evaporator is saturated vapor.
- LSH provides constant 15K superheat after evaporator

Properties of refrigerant were obtained using NIST-Refprop software and whereas compressor map of existing fluid (from manufacture) was used as reference to evaluate isentropic and volumetric efficiency rather than fixed values. In order to evaluate system performance, a list of the assumptions/parameter and equations used are shown in Table 4 and Table 5. In equations, enthalpy difference at the inlet and outlet was used to calculate capacity for major components (such as evaporator, compressor, condenser, etc).

Table 4 Parameters used/assumed in thermodynamic simulation

Parameters	Values / Details
Refrigerant	R1233zd (E) and R245fa
Condensing temperature (C)	85,95,105,115,125,135
Evaporation temperature (C)	45,55,65,75
Superheat @evp and subcooling @cond (K)	5 and 5
Superheat using LSH (K)	15
Isentropic efficiency (%)	Based on compressor map: temperature lift
Volumetric efficiency (%)	Based on compressor map: temperature lift

Table 5 Equations used for thermodynamic simulation

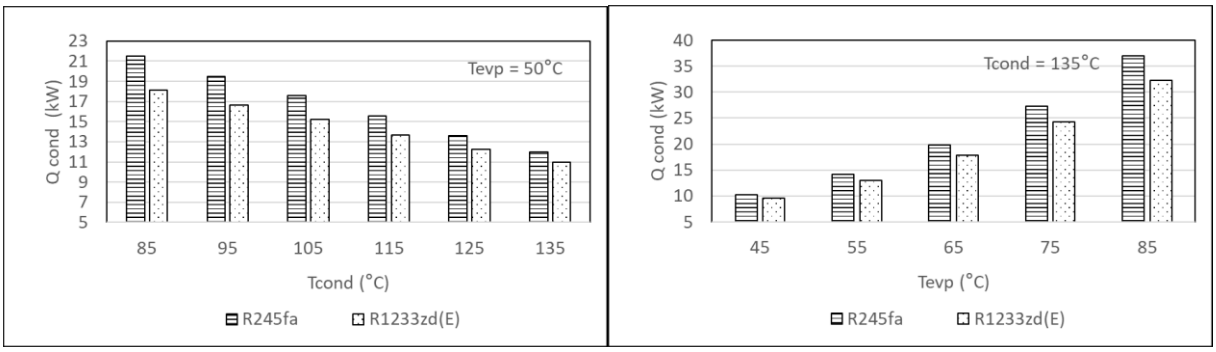
Calculated parameters	Equation
For heat exchanger output (kW): Condenser, evaporator, LSH	$Q = m_{ref} * \Delta h$
Isentropic efficiency (compressor) (%):	$\eta_{is} = \frac{\Delta h_{is}}{\Delta h}$
Compressor power (kW):	$P = m_{ref}\, \Delta h = m_{ref} * \frac{\Delta h_{is}}{\eta_{is}}$
Refrigeration side COP:	$COP = \frac{Q_c}{P}$

4. RESULTS AND DISCUSSION

4.1 Result of thermodynamic analysis

The system performance was evaluated thermodynamically at various evaporation temperatures and condensing temperatures as describe earlier for R12333zd(E) and R245fa. Figure 3 shows comparative analysis of system performance for R245fa and R1233zd(E). Performance analysis is presented in terms of condensing capacity, compressor power and COP with varying condensing and evaporating temperature conditions.

It is evident that for similar temperature lift conditions, the heat output for R245fa is higher compared to R1233zd(E) but the compressor power consumption is also higher which ultimately results in a negligible reduction in COP compared to R1233zd(E) This is mainly due to density/pressure difference at those temperature condition which, results in a higher enthalpy difference for R245fa. The evidence suggests that R1233zd(E) gives a higher COP in the range of 1 to 9% but lower heat output in the range of 8 to 15%. This may result in an increase to the heat exchanger size and volumetric area for R1233zd(E) if used as a direct replacement for R245fa.



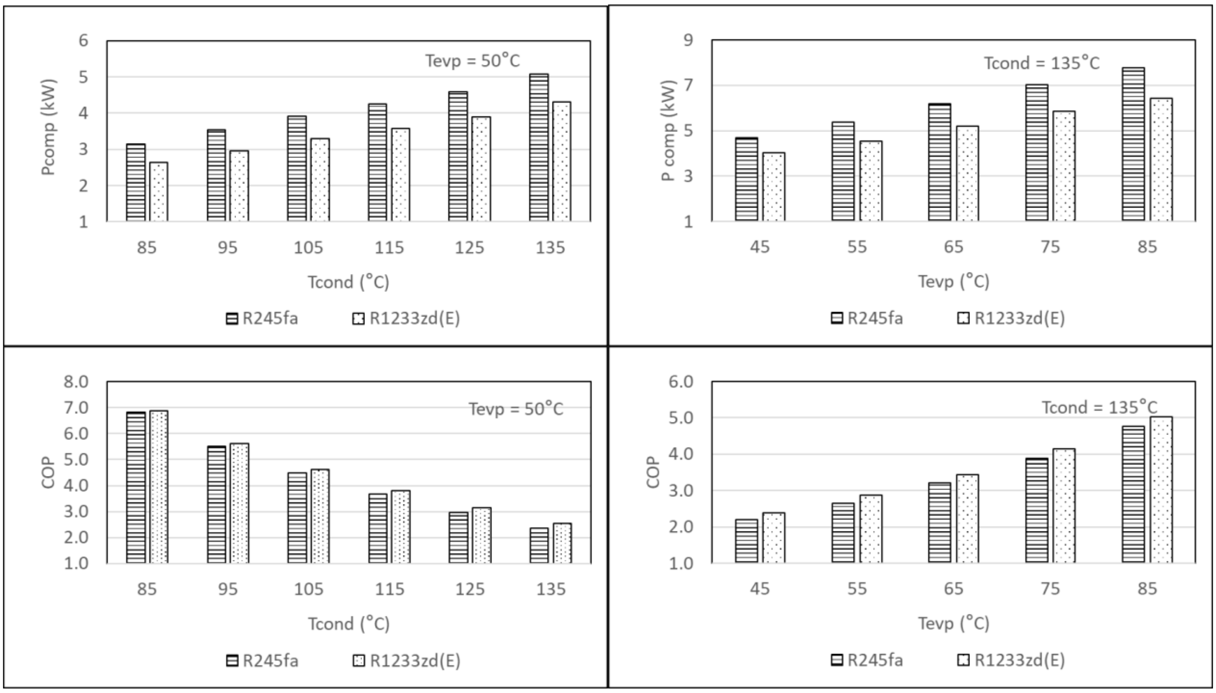


Figure 3 Comparative analysis for R245fa & R1233zd(E): Left: variation in T_{cond}, Right: variation in T_{evp}

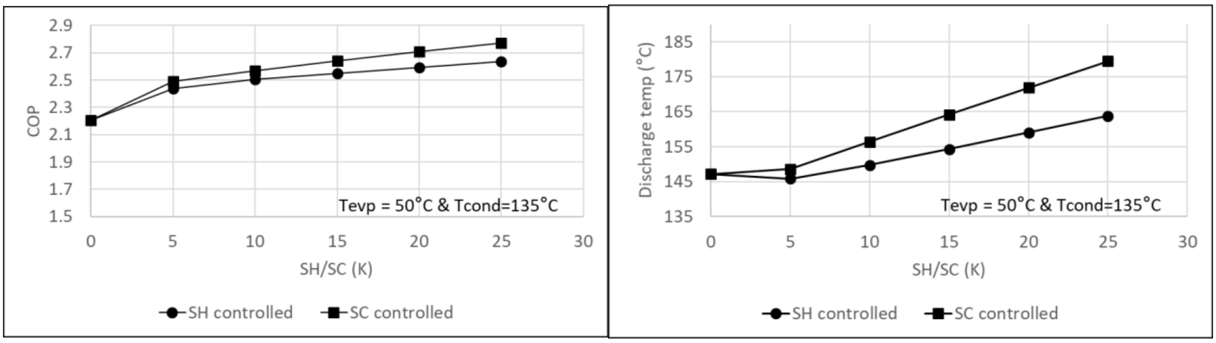


Figure 4 COP and discharge temperature variation with SC & SH controlled method using intermediate heat exchanger for R1233zd(E)

Above comparison was made while keeping LSH at 15K. However, further analysis carried out to understand benefits of controlling SH and SC for LSH assuming constant isentropic and volumetric efficiency at T_c=135°C & T_{evp}=50°C (with SH=SC=5K at condenser and evaporator). Figure 4 shows that if SC is controlled rather than SH then it could provide higher COP. However, discharge temperature puts limit on how much SC can be achieved in order to get improvement in COP.

4.2 Initial experience of test rig operation

For HTHP test rig, extensive time was spent on pre-commissioning before actual testing could take place as per test regime. There were several difficulties encountered mainly on: expansion valve and oil temperature side due to high temperature operation. Commercially available EEV needs additional cooling or EEV with stainless steel body is required for HTHP. In addition, oil viscosity at high temperature was a challenge as the oil temperature increased to 90°C while operating at 105°C condensing temperature. Hence, the test rig was modified to accommodate oil cooling to maintain temperatures with the range of 60-80°Cwhen operating at condensing temperature above 100°C. Figure 5 shows oil temperature rise with respect to $T_{con\ oil\ outlet}$ and $T_{evp\ water\ inlet}$ and addition of the I heat exchanger added to the compressor for oil cooling. However, oil temperature must be 5-10K higher than the suction temperature. Additional cooling was also provided to the expansion valve for safe working of the stepper motor as during initially commissioning the A stepper motor failed due to overheating issues.

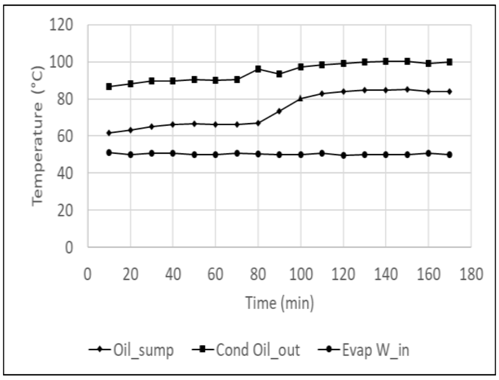


Figure 5 Oil sump temperature rise (left) and addition of oil cooler (right)

A few initial tests were carried out using R245fa as a reference in order to find component performance and controls required for safe working conditions. Figure 6 shows performance of the HTHP using R245fa at a fixed evaporation temperature and at varying condensing temperatures. Overall, the system COP remained between 2.3 to 4.3 and the condensing capacity between 10 to 14 kW. Although system performance is satisfactory, it is however lower compared to simulation results mainly due to several heat losses on primary side and refrigeration side. In addition, experimental results present system COP whereas simulation results provide the enthalpic COP.

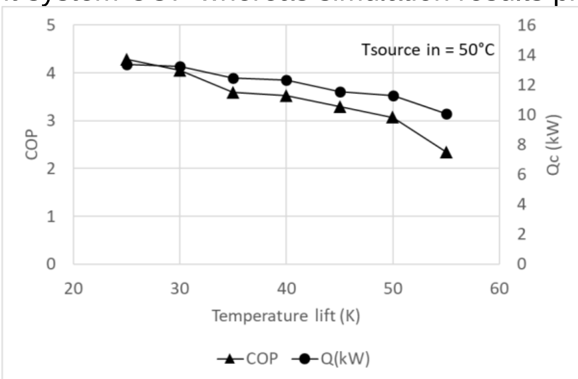


Figure 6 Initial test results of HTHP using R245fa

Before system performance can be evaluated further with R245fa and R1233zd(E), several other modifications will be carried out in terms of system temperature control, lubrication and insulation in order to optimise system performance. One of the biggest challenges for HTHP is commercially available parts, cost and suitable lubrication and cooling.

5. CONCLUSION

HTHPs have huge potential to use waste heat from industrial and commercial application. A literature search for R1233zd(E) has shown that there is limited investigation on the use of R1233zd(E) for HTHP, with most investigation focusing on simulation study and use in ORC applications. Simulation results and comparison between R245fa and R1233zd(E) showed a higher COP up to 8% when compared to R245fa. However, if R1233zd(E) is used as a direct replacement then careful consideration should be given to sizing of components (e.g. condenser, compressor) as it gives less heating/cooling capacity. Initial experience from this experiment showed the necessity for oil cooling due to high condensing temperatures which might be a limiting factor for safe compressor operation based on manufacture recommendations. In addition, if a commercially available standard EEV is used then it might need cooling to operate at high temperatures for prolonged periods. Suitable lubricant for R1233zd(E) is an area of debate as some refrigerant manufactures suggest POE whereas others suggest mineral oil. Oil miscibility data with POE is available for certain oil and temperature ranges but there is no readily available information for mineral oil. Long term performance and thermal stability with various materials requires further investigation for R1233zd(E). As a part of this ongoing CHESTER project, further investigation will be carried out to find suitable oil, properties (viscosity, miscibility, solubility) and temperature control for HTHP application. However, increasing development in HTHP area calls for more open and readily available information, development of components and knowledge sharing platform.

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NOMENCLATURE

c/con	Condenser	*Δh*	Enthalpy difference (inlet & outlet) (kJ/kg)
comp	compressor	*T*	temperature (°C)
EEV	Electronic expansion valve	*η*	efficiency
evp	Evaporator	*HTHP*	High temperature heat pump
is	Isentropic	*Q*	heat energy (kW)
LSH	Liquid-suction heat exchanger	*COP*	Coefficient of performance
m_{ref}	Refrigerant mass flow rate (kg/h)	*P*	Compressor power (kW)
ORC	Organic Rankine Cycle	*HCFO*	Hydro-chloro-fluoro-olefins
SH/SC	Superheat/Subcooling	*PTFE*	Polytetrafluoroethylene
VHC	Volumetric heating capacity	*POE*	Polyol-ester
W	Water	*in/out*	inlet / outlet

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