CHESTER

Compressed Heat Energy
Storage for Energy
from Renewable sources

D3.1 Refrigerant/lubricant interactions testing in high temperature heat pumps

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Contents

List of Figures

List of Tables

Glossary, Abbreviations and Acronyms

cSt	centistoke
comp	Compressor
con	Condenser
EEV	Electronic expansion valve
evp	Evaporator
FS	full scale (accuracy)
HCFO	Hydro-chloro-fluoro-olefins
HTHP	high temperature heat pump
P	Pressure (bar(g))
POE	Polyol ester
PVT	Pressure, Viscosity and Temperature
suc	Suction
T	Temperature (°C)

1. Introduction

1.1. Executive Summary

Refrigerant and oil interaction play crucial role for safe and efficient working of compressor and overall heat pump system. Oil viscosity, miscibility and solubility are important parameters that shows oil suitability for refrigerant in addition to various other parameters. As part of D3.1, suitable oil and refrigerant pairs were investigated to understand behaviour. Initial assessments were carried out on high temperature heat pump (HTHP) test rig developed at Ulster University. During the first phase, widely used refrigerant R245fa was used along with polyol ester type lubricant as recommended by compressor manufacturer. Performance of HTHP was investigated at various evaporation and condensing temperature to see impact of oil temperature and mixing behaviour at compressor suction. Based on initial results test rig was modified to accommodate oil temperature cooling and further test were carried out. However, for CHESTER prototype, HCFO-R1233zd(E) was selected as a refrigerant due to low GWP and ODP value and compressor was from Viking Heat Engine (VHE). Based on compressor manufacturer data, oil and refrigerant viscosity measurement was required for safe and efficient operation of compressor at high temperature (e.g. T_{con}= 135°C). In addition, lack of data on suitable lubricant for selected VHE compressor, required further investigation on oil-refrigerant viscosity. To accommodate, project needs, new testbed was developed at Ulster University which can measure pure oil and oil-refrigerant mixture viscosity at conditions like CHESTER prototype. Initially, commercially available polyol ester oils Reniso Triton SEZ320 was selected as a lubricant due to high viscosity and good miscibility data with R1233zd(E). In addition, newly developed Reniso CE VG 500 was supplied by Fuchs to test with CHESTER system. Both oils were examined with R1233zd(E) at various concentrations for the kinematic viscosity measurement across a range of temperatures and pressures. The combination of fluids must maintain viscosity as recommended by VHE to maintain the integrity of the CHESTER heat pump system during maximum operation conditions. Experimental results show that Reniso Triton SEZ320 could meet kinematic viscosity requirement of VHE if suitable oil cooling is adopted. Further test with Reniso CE VG 500 will highlight any potential benefits over POE 320. Lubricant under refrigerant concentrations up to 10% will maintain the required working kinematic viscosity within the HTHP system. The report presents comprehensive results considering confidentially and commercial sensitive information.

1.2. Purpose and Scope

The purpose and the scope of this deliverable to investigate suitable oil and understand oil-refrigerant interaction for HTHP in CHESTER prototype. This will help to select the suitable oil, refrigerant and hence, the compressor. However, all parameters are interrelated since it needs to satisfy compressor manufacturer recommendation too. In addition, limited information is available on commercially available lubricant which can work well with R1233zd(E) at CHESTER conditions. Hence, to address this issue, HTHP and oil-refrigerant interaction test rig were developed at Ulster University and various test were carried out. Following are the main outcome that are presented in the report as a part of this delivery under the Task of 3.1.2:

- Review and identification on suitable Refrigerant for CHESTER conditions
- Investigation on commercially available oil and suitability with refrigerant
- HTHP test rig development and experiments to understand oil behaviour

- Oil-refrigerant test-rig development and experiments for two different sets of oil: Reniso Triton SEZ320 & Reniso CE VG 500
- Results analysis and discussion
- Recommendation for CEHSTER prototype

1.3. Structure of the Document

The report is divided in four chapters

Chapter two: Investigation on Refrigerant and Oil for HTHP. This chapter consists main two parts. Part one focuses on review of suitable refrigerant for high temperature and showing suitability of R1233zd(E) and review on use of R1233zd(E) from literatures. Part two focuses on various requirement and purpose of lubricant (oil) for HTHP and investigation related to oil-refrigerant interaction from literatures.

Chapter three: Test Set-up for HTHP and Oil-Refrigerant Viscosity Measurement. First part of the chapter provides information in HTHP test-rig whereas second part of the chapter describes component and arrangement for oil-refrigerant viscosity measurement test-rig.

Chapter four: Test Methods and Results. This chapter describes test methods and boundary conditions used for various testing in first part of the chapter whereas second part of the chapter provides results for oil behaviour in HTHP and third part of the chapter provides results of viscosity and density measure of oil-refrigerant mixtures at different temperature/pressure.

Chapter five: Conclusion and Recommendation for CHESTER Prototype. This part of the report concludes findings from literature and experiment analysis and provides suggestion for suitable refrigerant, oil and arrangement requirement for CEHSTER prototype.

1.4. Relations with Other Deliverables

There is a need to understand how lubricant and refrigerant will interact within the HTHP and ORC systems. Oil temperature control, interaction with refrigerant and the kinematic viscosity of the fluid composition is a determining parameter for reliability and longevity of the system. To determine the kinematic viscosity of the system in operation (pre-commissioning) would be difficult to setup and measure effectively. A small test bed designed and built in Ulster university was used to carry out PVT testing of the proposed Reniso Triton SEZ320 and Reniso CE VG 500 lubricant oils and refrigerant R1233zd(E). This will guide and provide important information for D3.2, D3.6 and D3.8 as the high temperature heat pump (HTHP) within the CHESTER project will be subject to high temperature of +130°C discharge and +90°C suction. To maintain safe operation, the kinematic viscosity of the bulk sump fluid must be higher in line with the compressor manufacturer (VHE) recommendations. The initial plan to run SEZ 320 in the HTHP during testing prior to installation was run in parallel to the work carried out by Ulster University. The outcomes from Ulster university will provide a guidance on the suitability of the chosen lubricant oil to meet requirements and with further testing investigate other suitable lubricant types.

2. Investigation on Refrigerant and Lubricant for HTHPs

2.1. Introduction

Heat pump technology has a good potential to use waste heat recovery and to supply high flow temperature for various process/industrial application. Although, heat pump can be classified based on their working cycle, systems based on vapour compression cycle is widely used. Figure 1 shows classification of heat pumps based on various cycles.

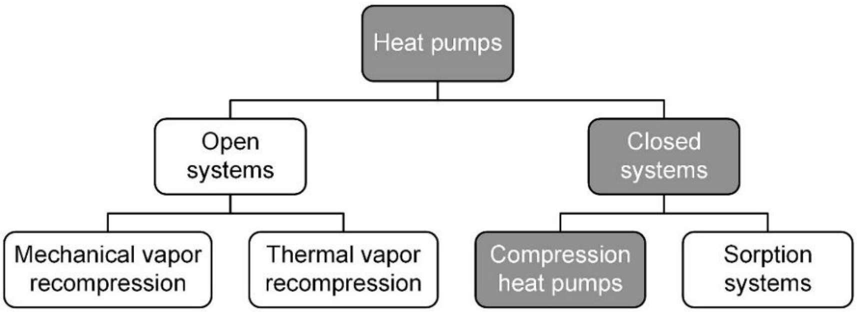

Figure 1 Classification of heat pumps [1]

Vapour compression cycle consist four major components namely; compressor, condenser, expansion device and evaporator. For high temperature heat pump application, heat source and heat sink temperature play major role since it affects performance (COP) as well as safe and efficient working of main components such as compressor. In CHESTER prototype, for HTHP, heat sink temperature would in a range of 135-145°C whereas heat source temperature would in a range of 30-100°C. Considering temperature lift of 70K and heat source temperature of 65°C, COP of HTHP would be around 5.8 based on Carnot. It is evident that heat pump development as a part of CHESTER scheme will fall VHTHP category considering high source (e.g. 65°C) and sink temperature (e.g. 135°C) which requires specialised industrial compressor, lubricants, refrigerant and other components. Figure 2 shows temperature levels for vapour compression heat pump from various literatures. The main types of compressor based on type of compression method (kinetic) and construction are based on Kinetic: reciprocating, screw, scroll and centrifugal (turbo), based on construction: hermetic, semi-hermetic and open.

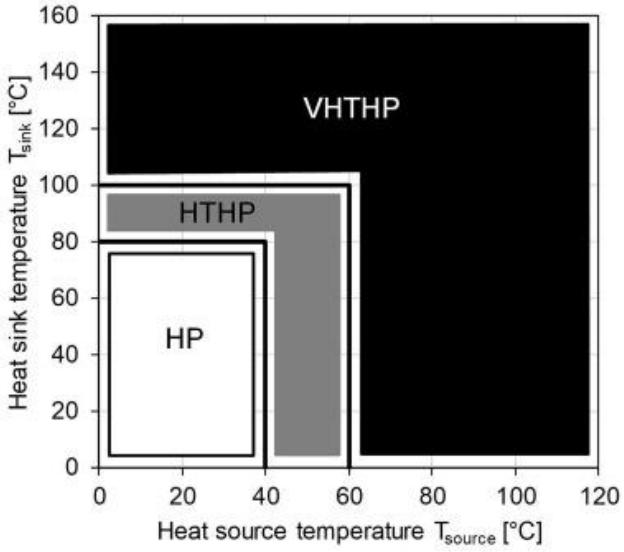

Figure 2 Temperature levels for compression heat pumps [2]

There are few commercial products available in the market where maximum temperature of 165°C is achievable with source temperature in a range of 35 to 70°C. For HTHP, following types of compressors are used in commercial products: Screw (single or twin), Piston (single and parallel), Turbo (single/two-stage). Figure 3 shows commercially available compressors used in HTHP with their capacity range and maximum achievable sink temperature.

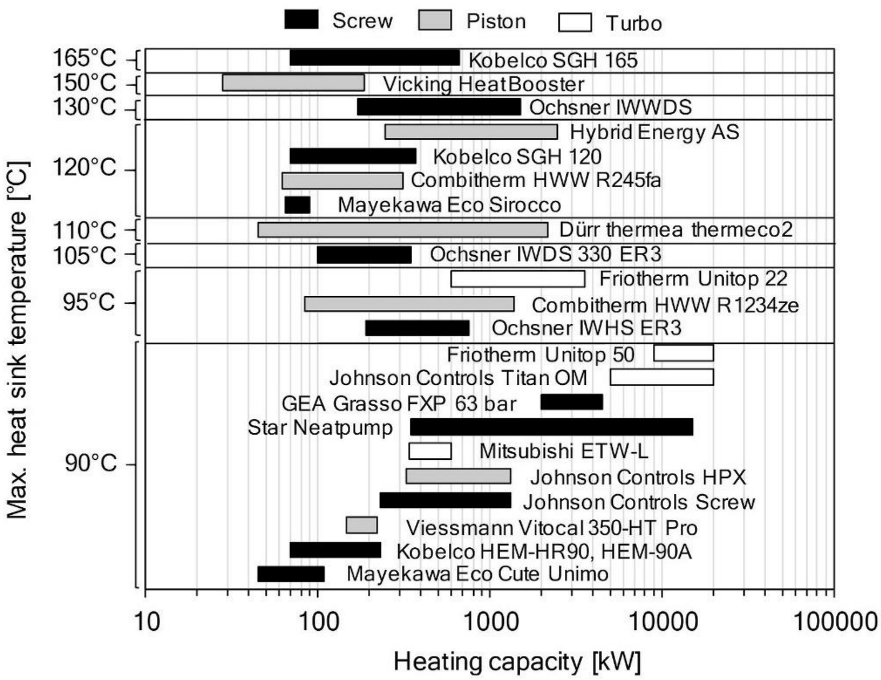

Figure 3 Commercially available HTHP compressor based on capacity and max sink temperature [2]

Typically, piston compressors (reciprocating) are constructed in V and W-shaped Monoblock for strength, rigidity and ease of layout. They are direct flow with false cylinder covers to protect from water-hammer effect. They are either water cooled, or air cooled with fins etc., based on temperature and refrigerant. However, they have complex access to bearing, lubricant system and other details, located in the block-crankcase.

Screw compressors are one or two-rotor machines of volume effect with constant geometric degree of compression. They are either dry or oil filled. A large amount of oil is injected into the working cavity to seal the gaps, lubricant and cooling into the oiled compressor. The oil injection allows reducing the noise level significant. However, it requires cooling at increased pressure. Turbo is technical a centrifugal type compressor with a volumetric capacity of 50-60 m/min or more and they inter-staged with staged throttling. They are typically high-speed machine and requires reducers for electric drive.

Hence, considering all parameters appropriate compressor was selected for CHESTER prototype and further details about the compressor will be presented as part of D3.2 & D3.6 report. Further information related to suitable refrigerant and lubricant has been discussed here as the main scope of this deliverables.

2.2. Refrigerant for HTHP

2.2.1. Introduction

As a part of F-gas regulation in the EU and Montreal Protocol, refrigerant with low GWP and low ODP is pushed forwarded to tackle issues of climate change and global warming. For HTHP, there are various refrigerant belong to family of CFC, HCFC, HFC, HFO, HCFO, HC, CF6, Ether and natural. However not all refrigerant can be used in present day due to their high GWP, high ODP, technical limit, temperature range and safety group. Key selection parameter for refrigeration for HTHPs are:

- Thermal stability: high critical temperature, low critical pressure, low pressure ratio
- Impact on environment: low GDP, low ODP and atmospheric lifetime
- Safety: flammability and toxicity
- Other factors: efficiency, cost, availability, suitable oil (and their mixture properties) and material compatibility.

Table 1 shows list of refrigerants for the candidate for HTHP application. Most promising refrigerant in terms of low GWP (under 1000), low ODP (almost 0) and high critical temperature (above 150°C) are:

- HFCs : R365mfc, R245ca, R245fa (issues: GWP above 150)
- HFOs : R1336mzz(Z), R1234ze(Z) (Promising, although A2L for R1233ze(Z))
- HCFOs : R1233zd(E), R1224yd(Z)(Promising)
- HCs : R601, R600, (promising but SG: A3)
- CF6 : Novec649 (recent development, lack of information)
- Natural : R718, R717 (R718 promising, R717 SG: B2L)

In addition , there are many other refrigerant such as acetone, benzene, cyclopentene and dichloroethane are very suitable for high sink and low source temperature [3] but due to their flammability and toxicity they are less preferred in industrial environment compared to less/no flammable/toxic refrigerant such as R1336mzz(Z), R1234ze(Z), R1233zd(E) and R1224yd(Z).

For CHESTER prototype, R1233zd(E) was selected due to high critical temperature, low ODP/GWP and ease of availability. In addition, R1233zd(E) can give a good compromise between COP and volumetric heating capacity compared to classic heat pump fluids.

Table 1 List of refrigerant suitable for HTHP application [2]

Type	Refrigerant	Description	Chemical formula	T_c [°C]	p_c [bar]	ODP [-]	GWP [-]	SG	NBP [°C]	M [g/mol]	Relative price [-]
CFC	R113	1,1,2-Trichloro-1,2,2-trifluoroethane	CCl_2FCClF_2	214.0	33.9	0.85	5'820	A1	47.6	187.4	prohibited according to Montreal Protocol [139]
	R114	1,2-Trichloro-1,1,2,2-tetrafluoroethane	$CClF_2CClF_2$	145.7	32.6	0.58	8'590	A1	3.8	170.9	
HCFC	R123	2,2-Dichloro-1,1,1-trifluoroethane	$C_2HCl_2F_3$	183.7	36.6	0.03	79	B1	27.8	152.9	
	R21	Dichlorofluoromethane	$CHCl_2F$	178.5	51.7	0.04	148	B1	8.9	102.9	
	R142b	1,1-Dichloro-1-fluoroethane	CH_3CCl_2F	137.1	40.6	0.065	782	A2	-10.0	100.5	
	R124	1-Chloro-1,2,2,2-tetrafluoroethane	C_2HClF_4	126.7	37.2	0.03	527	A1	-12.0	136.5	
HFC	R365mfc[a]*	1,1,1,3,3-Pentafluorobutane	$CF_3CH_2CF_2CH_3$	186.9	32.7	0	804	A2	40.2	148.1	8.9
	SES36*[b]	R365mfc/perfluoro-polyether	R365mfc/PFPE (65/35)	177.6	28.5	0	3'126[c]	A2	35.6	184.5	10.5
	R245ca	1,1,2,2,3-Pentafluoropropane	$CHF_2CF_2CH_2F$	174.4	39.3	0	716	n.a	25.1	134.0	n.a.
	R245fa[d]*	1,1,2,2,3-Pentafluoropropane	$CHF_2CH_2CF_3$	154.0	36.5	0	858	B1	14.9	134.0	6.6
	R236fa	1,1,1,3,3,3-Hexafluoropropane	$CF_3CH_2CF_3$	124.9	32.0	0	8'060	A1	-1.4	152.0	10.2
	R152a	1,1-Difluoroethane	CH_3CHF_2	113.3	45.2	0	138	A2	-24.0	66.1	n.a.
	R227ea	1,1,1,2,3,3,3-Heptafluoropropane	CF_3CHFCF_3	101.8	29.3	0	3'350	A1	-15.6	170.0	6.9
	R134a	1,1,1,2-Tetrafluoroethane	CH_2FCF_3	101.1	40.6	0	1'300	A1	-26.1	102.0	1.2
	R410A	R32/R125 (50/50 mixture)	CH_2F_2/CHF_2CF_3	72.6	49.0	0	2'088	A1	-51.5	72.6	2.9
HFO	R1336mzz(Z)[e]*	1,1,1,4,4,4-Hexafluoro-2-butene	$CF_3CH{=}CHCF_3(Z)$	171.3	29.0	0	2	A1	33.4	164.1	n.a.
	R1234ze(Z)*	cis-1,3,3,3-Tetrafluoro-1-propene	$CF_3CH{=}CHF(Z)$	150.1	35.3	0	<1	A2L[f]	9.8	114.0	n.a.
	R1336mzz(E)[g]	trans-1,1,1,4,4,4,-Hexafluoro-2-butene	$CF_3CH{=}CHCF_3(E)$	137.7	31.5	0	18	A1	7.5	164.1	n.a.
	R1234ze(E)	trans-1,3,3,3-Tetrafluoro-1-propene	$CF_3CH{=}CHF(E)$	109.4	36.4	0	<1	A2L	-19.0	114.0	5.6
	R1234yf	2,3,3,3-Tetrafluoro-1-propene	$CF_3CF{=}CH_2$	94.7	33.8	0	<1	A2L	-29.5	114.0	13.8
HCFO	R1233zd(E)[h]*	1-chloro-3,3,3-Trifluoro-propene	$CF_3CH{=}CHCl(E)$	166.5	36.2	0.00034	1	A1	18.0	130.5	6.3
	R1224yd(Z)[i]*	1-chloro-2,3,3,3-Tetrafluoro-propene	$CF_3CF{=}CHCl(Z)$	155.5	33.3	0.00012	<1	A1	14.0	148.5	n.a.
HC	R601*	Pentane	$CH_3CH_2CH_2CH_2CH_3$	196.6	33.7	0	5	A3	36.1	72.2	4.9
	R600*	Butane	$CH_3CH_2CH_2CH_3$	152.0	38.0	0	4	A3	-0.5	58.1	1.8
	R600a	Isobutane	$CH(CH_3)_2CH_3$	134.7	36.3	0	3	A3	-11.8	58.1	1.0
	R290	Propane	$CH_3CH_2CH_3$	96.7	42.5	0	3	A3	-42.1	44.1	1.1
	R1270	Propene	$CH_3CH{=}CH_2$	91.1	45.6	0	2	A3	-47.6	42.1	1.0
CF6	Novec 649[j]*	Dodecafluoro-2-methyl-3-pentanone	$CF_3CF_2C(O)CF(CF_3)_2$	168.7	18.8	0	<1	n.a.	49.0	316.0	6.8
Ether	E170	Dimethyl ether	CH_3OCH_3	127.2	53.4	0	1	A3	-24.8	46.1	39.0
Natural	R718	Water	H_2O	373.9	220.6	0	0	A1	100.0	18.0	5.6[k]
	R717	Ammonia	NH_3	132.3	113.3	0	0	B2L	-33.3	17.0	27

2.2.2. Review on R1233zd(E)

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 2 shows main parameter comparison between R1233zd(E) and R245fa. It is evident that R1233zd (E) can be used for HTHP up to 150°C and as a direct replacement of R245fa.

Table 2 Important parameter comparison between R1233zd(E) and R245fa

Parameters	***R1233zd(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 lifetime (yr)	0.07	7.7
Flammability	Non-flammable	Non-flammable
ASHRAE std 34 safety class	A1	B1

There are limited investigations related to R1233zd(E) use for HTHP in scientific literature and most investigations/literature are very recent. Figure 4 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).

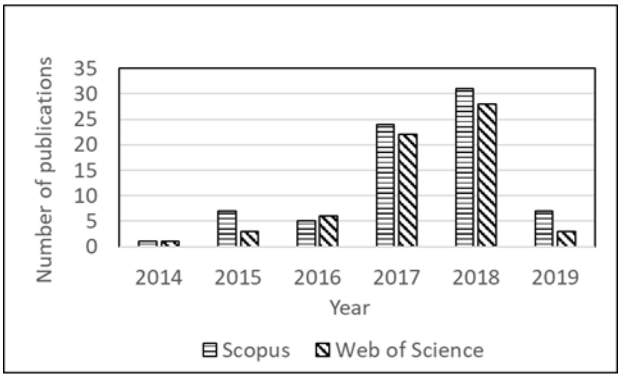

Figure 4 Publications related to R12333zd(E)

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 3 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) [4] . 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 [5]. 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 [6]. 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 [2]. 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) [3] and Bamigbetan et al. (2018) [7].

Table 3 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

However, long term stability, thermal performance and interaction with other material (e.g. oil, copper, etc) is also important for R1233zd(E). Hence, use of R1233zd(E) in CHESTER prototype would provide useful information and new findings in less investigated area of heat pump research.

2.3. Lubricants

2.3.1. Introduction

Refrigerant oil plays important role in refrigeration system as it affects long life of compressor. This is mainly true for fluctuating high and low temperature. The main function of refrigerant oil is to lubricate all moving parts in the refrigeration compressor. In addition, it also plays crucial role to dissipate heat from compression chamber and valves and as a sealant based on compressor type. Oil content in the system usually can reach from 1 to 5% and higher in special cases. Oil with satisfactory miscibility are used to ensure reliable oil circulation and to ensure that the oil returns from the cold part of the circuit. Figure 5 shows oil flow in refrigeration circuit.

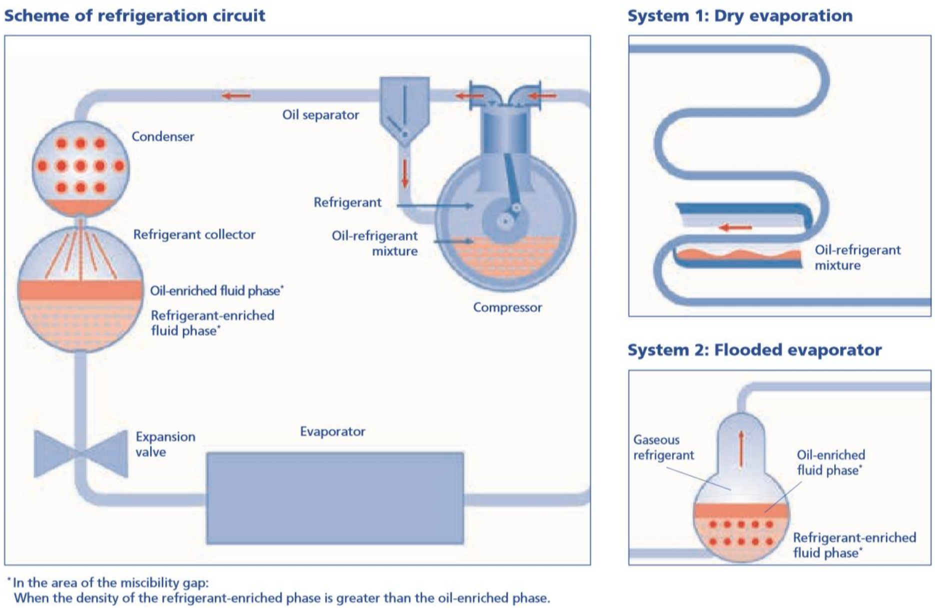

Figure 5 Vapour compression refrigeration circuit with oil flow (source: Fuchs)

Some issues of oil (lack of oil or excessive oil) in the system could be:

- Lack of oil: in crankcase could result in improper lubrication of bearing and other parts and adding further oil may result in inconsistent oil through the system. Excess oil may return to compressor in a slug and possibly damage compressor
- Excessive oil decreases capacity (by increasing temperature difference between load and refrigerant)
- May cause restriction in expansion valve and reduce refrigerant flow
- Circulating oil may break down hot spots in the system and cause varnishing and valve sticking. It may react with the refrigerant at high temperature and form acids

2.3.2. Lubricant Families and Properties

There is different type of oil that can be used with various refrigerant in compressor. However, only particular type of oil and refrigerant pair can be used based on stability. Six chemical families of lubricants in Air Conditioning and Refrigeration are:

- Mineral Oil (MN)

- Alkyl benzene (AB)
- Polyalphaolefin (PAO)
- Polyalkylene glycol (POG)
- Polyvinyl ether
- Special synthetic oils for HFOs (combination of above with additives)
- Polyol ester (POE)

Refrigerant and compressor manufacturers have identified polyol ester oils suitable for use with HFC replacement refrigerants. These oils are miscible with HFC refrigerants. They also are miscible with CFC and HCFC refrigerants. Polyol ester oils may be used with any of these refrigerants. Polyol ester oils, unlike mineral or alkylbenzene oils, are very hygroscopic. This means that they absorb moisture quickly. These oils come in metal containers to keep them as dry as possible. Exposure to the atmosphere must be kept to a minimum. Polyol ester oils will typically saturate at 1,000 ppm moisture when exposed to the atmosphere. Compare this with about 100 ppm for mineral oils. Oils with a high moisture level can create problems in a refrigeration system. The available viscosity grade ranges from 10 cSt to 220 cSt. Viscosity Index(VI) is indicative of viscosity change over a temperature range (40°C to 100°C). The pressure viscosity coefficient is a measure of effect of pressure on viscosity. This is relevant in elasto-hydrodynamic lubrication regime. In order to improve performance of lubricants, additives are added based on their functionality such as: Anti Wear (AW), Extreme Pressure (EP),and Friction Modifier(FM). Lubricant can be characterised for refrigerant based on following parameters, this also varies with external conditions (e.g. temperature, pressure, additives etc):

- Viscosity
- Chemical stability
- Thermal stability
- Low hygroscopicity
- Thermal conductivity
- Dielectric strength
- Neutralization number
- Flash point and fire point
- Environmentally acceptable
- Compatible with elastomeric and polymeric components
- Soluble with refrigerant gas and non-deposit forming

Based on those characteristics, lubricant manufacturer develops and test long terms performance of suitable oil for specific refrigerant. However, for refrigerant-oil interaction purpose viscosity, miscibility, solubility, density and pressure-viscosity coefficient are important properties that requires attention in developing/testing new oil/refrigerant. Brief importance of each parameter has been give in following part.

Viscosity is a measure of how oil is thick or thin at a given temperature and how readily it flows at that temperature. The oil becomes more “fluid” (or less viscous) as the temperature increases and “Thick” at lower temperature. A standard temperature of 40°C and 100°C is used to measure oil viscosity. In addition to lubrication, viscosity should also allow to dilutation of the refrigerant. Viscosity of mixtures (oil/refrigerant) may decrease as much as an order of magnitude since the refrigerant viscosity is typically 2-3 order of magnitude smaller than that of oil.

Miscibility behaviour of the refrigeration oil with various refrigerants is shown in miscibility gap diagrams. The refrigerant miscibility of the lubricating oil in the cooling circuit is of decisive importance to oil transportation and to the overall efficiency of the refrigeration system. Phase separations can lead to operating malfunctions especially in heat exchangers, evaporators and in collectors. Insufficient oil return not only affects the function of control valves but can also lead to inadequate lubrication and compressor breakdowns.

Solubility: it is ability to mix with the refrigerant and travel through the system. Too much solubility of refrigerant in the compressor oil is harmful, causing low viscosity, oil foaming etc. while Partial immiscibility (liquid-liquid separation) may also cause problem as both (oil rich and refrigerant rich) liquids could have very low viscosities. To prevent the system from excessive oil accumulation, a high degree of mutual solubility may be required but at the same time it must provide a proper viscosity of the mixture in the compressor. As without this, large amount of oil in the system create problems; reduction of heat transfer in heat exchanger and clogging of capillary tube [8].

*Viscosity and pressure-viscosity coefficient: T*he concentration of refrigerant used to dilute the oil depends on temperature and pressure. Film thickness strongly dependent on viscosity and pressure-viscosity coefficient. The molecular weight ratio between the oil and refrigerant is essential to predict the behaviour of viscosity and pressure-viscosity coefficients when the refrigerant dilutes the oil. In a compressor, a lighter refrigerant such as ammonia dilutes the oil by only 3-5%. Heavier refrigerants like R-22 and R-134a are usually found at concentrations of 5 to 40 wt%, depending on the running conditions. The effect of refrigerant on the viscosity and pressure-viscosity coefficients is essential data to achieve the right lubricating conditions for the compressor

Compressibility and density: An important parameter to reduce stress and improve life of the bearing is compressibility. The lubricant compressibility affects the magnitude of the pressure spikes in the pressure distribution. Esters, polyglycol and polyalpholefin are compressible whereas naphthenic and paraffinic mineral oils have more complex molecules and show a stiffer behaviour.

2.3.3. Measurement/Standards for Oil

In order to verify suitability of oil for given refrigerant/compressor, measurement of various parameters/properties (as described earlier) are required. Figure 6 shows most common properties used to characterise the refrigeration oil.

Colour	DIN ISO 2049
Viscosity	DIN EN ISO 3104
Density	DIN 51757
Neutralization number	DIN 51558-1
Water content	DIN 51777-1/-2
Pourpoint	DIN ISO 3016
Flashpoint	DIN ISO 2592
Refrigerant miscibility	DIN 51514
Refrigerant stability (Sealed-Tube-Test)	ASHRAE 97-2007

Figure 6 Typical properties to characterise a refrigeration oil

In addition to above mentioned properties, Classification based on DIN 51503, part 1 (2011) and Mixture viscosity and vapour pressure; Daniel Plot; PVT diagram are also used for characterisation purpose.

In laboratory environment *miscibility* behaviour is determined in pressure resistant glass tubes or in autoclaves. Different concentrations of oil refrigerant mixtures are tested. The oil-refrigerant mixture is homogenized and cooled (respectively heated) in a de-fined way (in 3K steps). If the oil and refrigerant separate into two fluid phases (the phase separation is characterized by turbidity or emulsion formation in the initially clear fluid), this is the miscibility gap or the point of threshold solubility. These points from different concentrations form a phase diagram, more commonly known as the miscibility gap diagram.

The *solubility* and *viscosity* are measured in closed system. The lubricant is first charged into the test system (±0.1 grams) and then the refrigerant is charged (±0.1 grams) into test system under condition: -10℃ and high vacuum. The mixture system is stirred and heated to suitable temperature based on the type of refrigerant. Sampling and measuring the related solubility, viscosity and density data only when vapor-liquid equilibrium has been achieved in mixture system. The numbers of measuring for each specified refrigerant-lubricant mixture are around 45~50 points.

In addition, commercially available, visco-meter, density meter and oil concertation sensor are also used in heat pump/refrigeration system in order to measure various parameters while system in operation.

2.3.4. Refrigerant and Oil Mixture in the System

Viscosity of mixture in heat exchanger and compressor play crucial role. Viscosity of mixtures is highly crucial for evaporator. The viscosity of mixtures containing a large amount of refrigerant is very low. When the refrigerant has evaporated, the viscosity of the oil left behind rises rapidly. However, if the temperature continues to increase after the refrigerant has escaped from the oil, the oil will become more “fluid” (viscosity will decrease). Viscosity is lower at higher pressure because more refrigerant is dissolved in the oil. Refrigerant gas is more soluble in oil at low temperature than at high temperature at the same pressure or in another words, near saturation conditions. In many systems point of highest viscosity may be outside the evaporator. In this case, it would help to raise the suction-line temperature quickly. If a liquid-vapour heat exchanger is used, locate it closer to evaporator to keep superheat low.

In addition, film thickness of oil/refrigerant mixture plays important role for compressor operation. To obtain long life, the moving surfaces should be completely separated by a lubricating film. When the film thickness is too thin, surface asperities penetrate the oil film and metal-to-metal contact occurs. Contaminating particles can also be trapped in the contact and cause contact between the moving bodies. Contacts result in denting and plastic deformation of the surfaces with high local stresses, resulting in surface wear and increase risk of fatigue. [9]. If refrigerant is mixed in the oil and used as lubricant, such mixtures containing less than 75% oil by weight will not sustain an oil film in rolling element bearings and are therefore unsuitable for lubrication purposes.

Based on investigation on refrigeration oil from literatures following points should be considered for HTHP in CHESTER prototype compressor selection, design and operation:

- Pressure ratio: typical pressure ratio for HP application could up to 8 to 10 and higher-pressure ratio affects bearing lubrication and discharge valve temperature
- Operation hours: as it affects major components but mainly in terms of oil, prolonged operation at high discharge gas temperature may result in oil breakdown accumulation and this could result in improper closing of valves and loss of performance.

- Liquid refrigerant in oil: excessive liquid refrigerant in compressor oil can have damaging effects
- Slugging: mechanical stress on valves, gaskets and piston rod assemblies
- Loss of oil: violent boiling of the refrigerant in the oil sump may carry the oil in the form of foam
- Dilution of oil: flood back may dilute the oil to the extent that bearing failure will result
- Piston seizure: rapid change in temperature may result in loss of piston-cylinder clearance and piston failure [10].

2.3.5. Refrigeration Oil for CHESTER

To select suitable oil for CHESTER prototype following criteria were used:

- Information about boundary conditions and operation range
- Working fluid (refrigerant)
- Compressor type
- Manufacturer recommendation (e.g. Target for oil viscosity)

And based on temperature condition of CHESTER and selected refrigerant (R1233zd(E)), a commercially available refrigeration oil was searched for. However, very limited information is available for suitable oil for such high temperature range. 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) [11] [12] 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 [13].

However, due to readily available commercially information, it was found that POE would be suitable for CHESTER prototype with R1233zd(E). Hence, further search was carried out to find commercially available POE with high viscosity grade which can work at high suction/discharge temperature and satisfy compressor manufacturer requirement. Table 4 shows list of commercially available POE during the search. Most of POEs available in the market has viscosity grade up to 220 with exception of Fuchs which can provide POE with 320 VG.

Table 4 List of commercially available POE

Name	Lubricant series (POE based)	Max VG available
Bitzer	BSE series	170
Castrol	Aircol SW range	220
Climalife	HQ series	220
CPI	Solest	370
ExxonMobil	Mobil EAL Arctic Series	220
Fuchs	Reniso Triton SE/SEZ series	320
HARP		68
Lubrizol	Emkarate RL series	220
Petronas	Suniso SL range	220
Shell	S4 range	94
Total	Planetelf ACD series	220

Hence, for CHESTER heat pump, Fuchs RENISO TRITON SEZ 320 was selected due to highest VG available commercially and further technical details. There was also requirement from VHE that refrigeration oil needs to have certain viscosity at given conditions for safe operation. VHE has tested various other refrigerant for their compressor, but they have not carried out any test with R1233zd(E) and suitable oil. An example of Viking compressor requirement for the oil with R245fa showed that SEZ320 performs better than SEZ 220 with R245fa but no information is available for R1233zd(E).

SEZ 320 is based on synthetic esters and can also be used with HFOs and blends of HFO/HFC. It is also complete miscible with R1233zd(E). Figure 7 shows miscibility gap of selected oil and R1233zd(E). In addition, with support from Fuchs newly developed complex ester-based oil RENISO CE VG 500 was also obtained in a small sample. Further details about both oils has been given in next chapter.

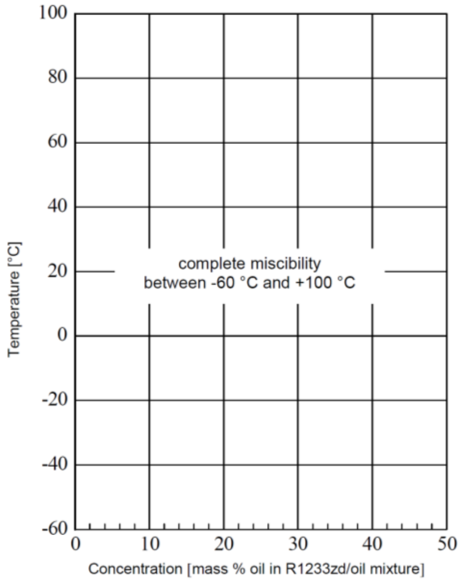

Figure 7 Miscibility gap for SEZ 320 and R1233zd(E) (source: Fuchs)

3. Test Set-up for HTHP and Oil-Refrigerant Viscosity Measurement

3.1. HTHP Test Set-up

In order to assess the behaviour of oil temperature, mixtures in compressor and cooling requirement, a separate test-rig was developed at Ulster University. 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. A commercially available components were used for the development of the test-rig. Figure 8 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 5 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), 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.

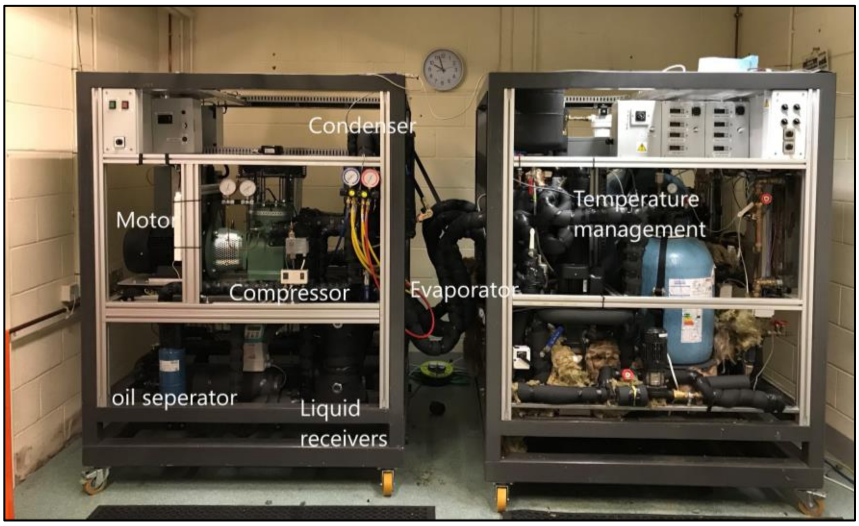

Figure 8 HTHP test rig at Ulster University

Table 5 List of components used for HTHP set-up

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 set-up for Oil-Refrigerant Viscosity Measurement

An experimental test bed was constructed to measure the properties of lubricant-refrigerant mixtures. The test bed was based on two separate units, one was a refrigerant storage vessel and the other was a single continuous loop for the circulation of lubricant and lubricant-refrigerant mixtures. Figure 9 shows picture of test-set up without insulation at Ulster University.

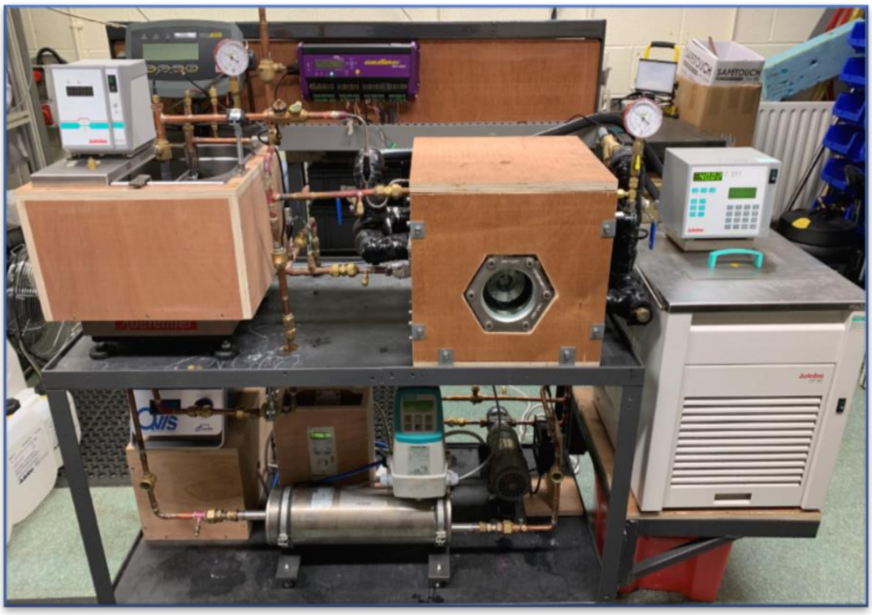

Figure 9 Picture of oil-refrigerant viscosity measurement test bed at Ulster University

Most of the components of this test-rig was bespoke designed and commercially available measuring instruments were used to measure various parameters required for the analysis. Table 6 shows list of measure components used for the test-rig.

Table 6 List of components used in oil-refrigerant viscosity measurement test rig

Description	Make
Massflow meter	Siemens: Sitrans F C Massflow Mass 2100 (Body) with a Strans F C massflow Mass 6000 (Head)
Oil Pump	KRACHT gear pump
Motor and Speed Control	Parvalux DFF DD 28 with ABB ACS150 inverter micro drive (0.37kW)
Oil filter	Henry SH_9105
Test Cell	Stainless cylinder with two view ports
Thermal Bath A	Bespoke Stainless square with two view ports
Thermal Bath B	Bespoke Stainless bath
Julabo A	Julabo Bath (FP50 GB) with Head (HP GB)
Julabo B	Julabo Head (ED v.2)
Viscosimeter	Flucon Laboratory Viscosimeter QVis 01/L
Scale	Ohaus Defender 5000 series scales – T51P

The most novel part of the test-rig was viscometer. A Flucon Laboratory Viscosimeter QVis 01/L was used to measure the dynamic viscosity of the lubricant - refrigerant compositions with a precision of +/- 2% deviation with reference oil calibration [14]. The unit is positioned so that it is immersed fully in liquid and supports unidirectional flow. The unit body was supplied with 1" UNC treaded connections (Figure 10), which were fitted with ½" tapered connections to fit into the hydraulic loop. The unit comes with an electric controller (Figure 11), which has a display to read directly from and also an output to computer with supporting software to modify settings and log data.

Figure 10: (Left) Outer body with viscosimeter; (Right) Measuring head unit with piezoelectric quartz crystal.

Figure 11: Electronic unit equipped with an electronic control and evaluation system

In addition, two thermal baths were used to conduct the experiments, the larger Julabo was filled with a specialized bath fluid which has an operational range of -20°C to 180°C. The second thermal bath was setup with a Julabo head within a bespoke stainless bath, the fluid with an operational range of up to 180°C. A DT85 Datalogger was used for data acquisition purposes and is certified for an accuracy of ±0.1%. All thermocouples selected were 4 wire PT100 element type (range -50°C to +200°C) and have an accuracy of ± 0.3°C. The pressure transmitters used were PT4-30S type with a range of (0 to 30 bars with 2% accuracy) and a temperature range of (-50 to +135°C). A Siemens Massflow meter was used to provide both fluid flow (kg/min) and density (kg/m^3). This equipment has linearity error % of rate ±0.15 for mass-flow and density error (Standard) ±0.005 and process media temperature (Ts) (-50 to +180°C).

4. Test Methods and Results

4.1. Test Methods and Results of HTHP

4.1.1. Test Methods and Other Details

For HTHP test rig, the main goal was to understand oil temperature and behaviour observation inside the compressor. For this purpose, extensive time was spent on pre-commissioning before actual testing could take place as per test regime. 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. After initial tuning, the system was operated at a fixed evaporation temperature (e.g. 50°C) and varying condensing temperature between 85 to 125°C. The speed of the compressor, flow rate on evaporator/condenser secondary side were kept constant for all test after first base test.

Polyester oil HARP POE68 was recommended by the supplier for higher temperature applications exceeding DIN51503 part 1 (Bitzer). Based on technical guidance, the selected oil viscosity ranges from 65.5 cSt at 40°C to 9.3 cSt at 100°C, providing a guide to viscosity assuming liner response between the temperature bandwidth. Figure 12 shows details of POE 68 and variation of viscosity of with temperature that used for initial analysis purposes.

Property	POE 68
Viscosity (cSt)@40°C	65.5
Viscosity (cSt)@100°C	9.3
Viscosity index	120
Pour point (°C)	-43
Colour (hazen ISO 2211)	<150
Flash point (°C)	270
Specific gravity (g/cm^3) @ 20°C	0.98
Total Acid number (g/mg)	<0.05
Water (ppm max)	<50

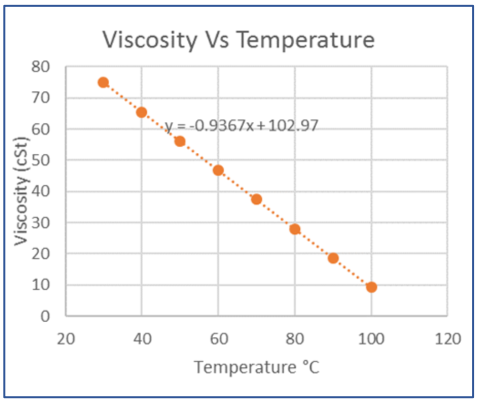

Figure 12 Details of POE 68 (left) and viscosity vs Temperature variation (right) for HTHP

There were several difficulties encountered mainly on expansion valve side due to high temperature operation. A commercially available EEV needs additional cooling or EEV with stainless steel body is required for HTHP. After having bespoke solution for EEV cooling, further tests were carried out. Further details about operational experience and test result has been given in following section.

4.1.2. Results and Discussion

Initial outcome

Many of the operational HTHP issues which occurred during testing appear to be based on failures due to higher temperatures within the systems under test. The failure of the expansion valve during operation presented as an intermittent issue >95°C initially then a full failure above >110°C discharge temperature. The stepper motor in the valve failed possibly due to higher temperatures, OEM specified 105°C working limit. The solution was to reengineer a new expansion valve to isolate the head from the body and water/air cool the stepper motor, further testing showed this solution to work without any further issues.

Figure 13: (Left) Expansion valve: (Middle) Bearings and seals, (Right) Sump Plug

The compressor oil shafts seals (Figure 13) failed between the compressor and motor at about the same period and inspection showed some hardening of seals and internal wear on components which showed up as shards of swarf on the magnetic sump plug.

During the maintenance service, compressor oil was removed to assess the state of degradation. This was done visually without testing of any type. The visual examination compared a sample of new oil against the oil removed. There was a marked changed in the clarity from the clear yellowish-brown tone when new to a dark cloudy black to grey colour when removed from the compressor. It may be possible to speculate that the internal run-in period of the compressor had deposited carbon like deposit into the oil creating almost a graphite type appearance within the oil. The inspection port on the side of the compressor had also completely clouded over with this deposit, blocking the view of oil levels completely. This required the removal of the side viewing port to thoroughly clean these deposits off the glass.

Start-up

During the initial start-up the compressor goes through a phase whereby the oil and refrigerant mix readily in the sump corresponding to increased sump pressure and temperature. Once the evaporator and condenser temperatures start to stabilise and setpoints are reached a rapid drop in both pressure and temperature in the sump occurs. However, it depends on compressor and other conditions. For the compressor in HTHP, it takes approximately one hour to reach stability. Figure 14 shows behaviour of oil and refrigerant from start to achieve stable conditions. It is essential to reach at higher temperature in stages in order to allow refrigerant and oil stabilise in the compressor rather than reaching to final set-point directly.

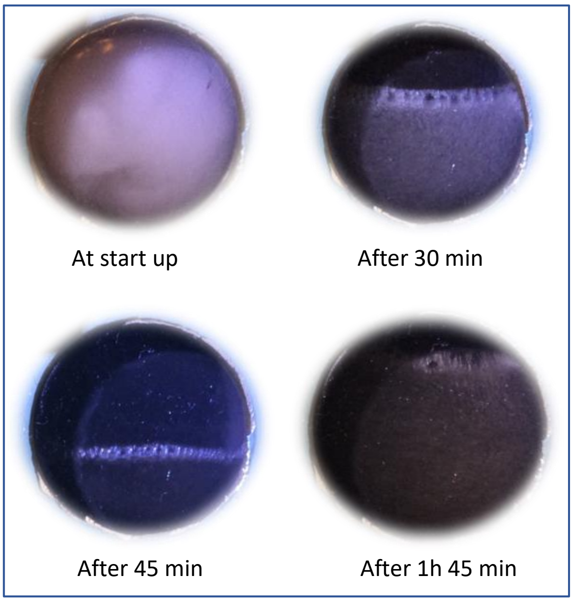

Figure 14 Sigh glass view during operation of HTHP

Oil temperature and cooling

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 15 shows oil temperature rise with respect to $T_{con\_oil\ outlet}$ and $T_{evp\_water\ inlet}$.

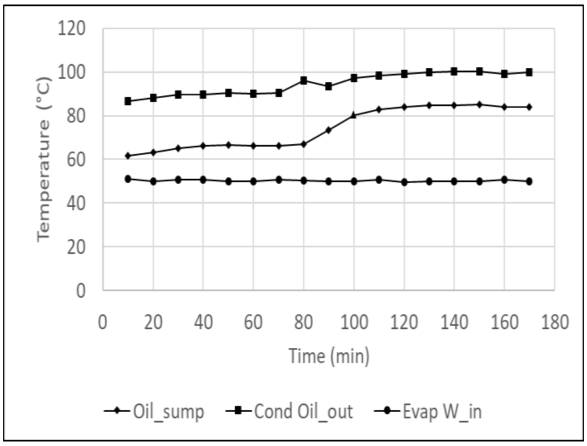

Figure 15 Oil sump temperature rise with condensing temperature

To overcome this an oil cooling system was fitted to the compressor incorporating an integrated heat exchanger on the oil return line, which was water cooled. However, oil temperature must be 5-10K higher than the suction temperature and below 90°C there was no need for oil cooling, above this temperature a PID was used to control cooling. Overcooling the refrigerant lubricant oil in the sump was found to cause additional efficiency issues at the condenser. Figure 16 shows schematic of oil cooling circuit added to HTHP to maintain oil temperature during the operation. In addition, initial tests only had a single thermocouple on the oil return pipe which was not enough to make a wholly accurate assessment within the compressor sump. To rectify the issue a redesign incorporated several thermocouples and two pressure sensors between the oil separator, heat pump and sump.

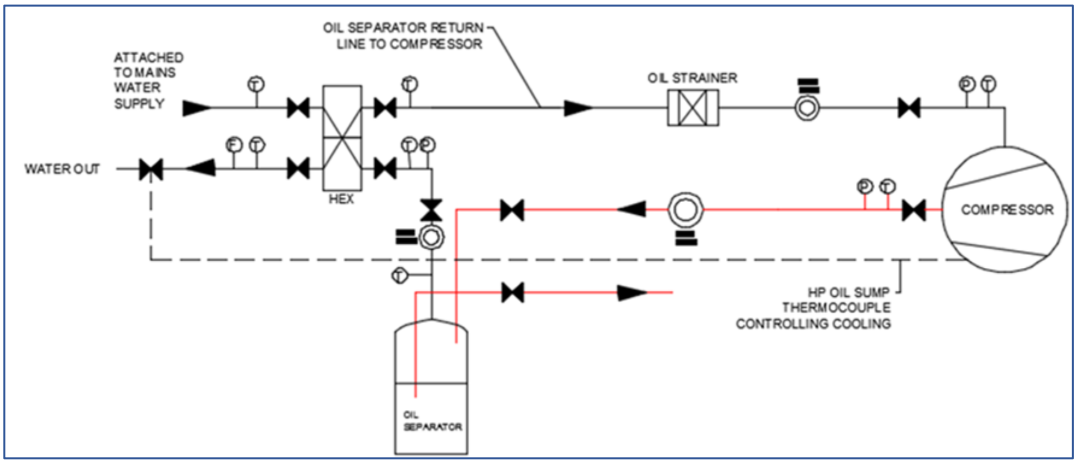

Figure 16 Oil temperature control loop for HTHP

Updating oil cooling system helped to control oil temperature during the operation. Figure 17 shows controlled oil temperature during the operating with R245fa. The compressor was run for different periods increasing the amount of cooling applied to the return oil circuit. By the installation of a heat exchange alone a certain amount of passive cooling of the oil takes place passing through the unit. The slow reduction of temperature in the sump caused a gradual reduction in the ability of the compressor to maintain setpoints.

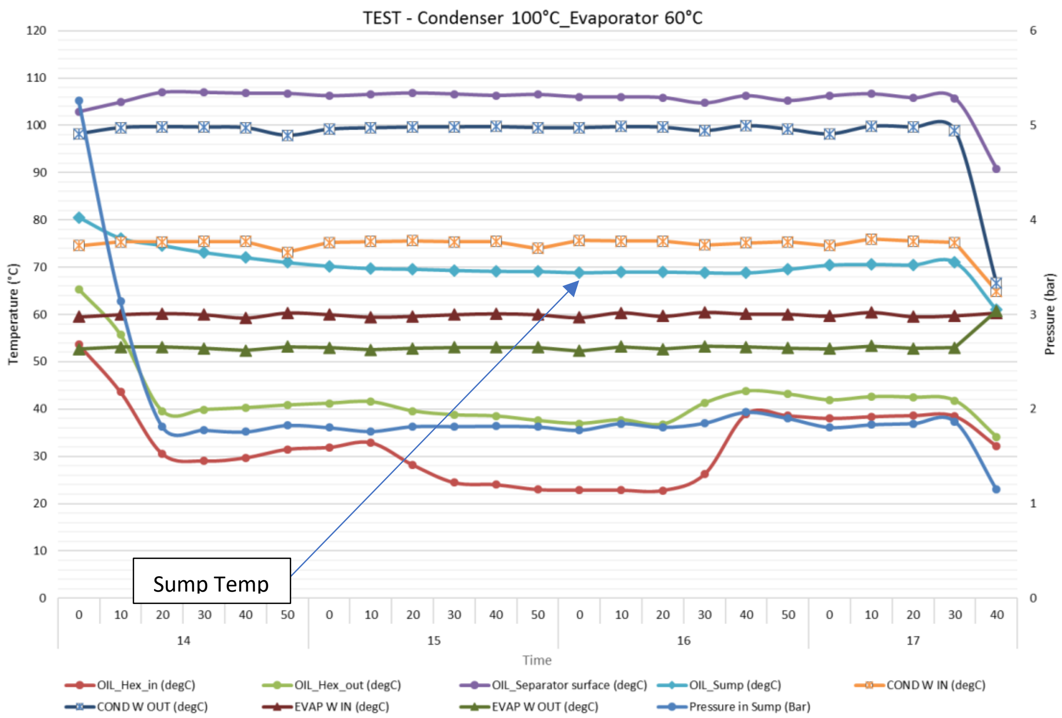

Figure 17:Test R245fa - Condenser 100°C and Evaporator 60°C - temperature and pressure range after start-up.

__Overall performance with R245fa__

Figure 18 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.

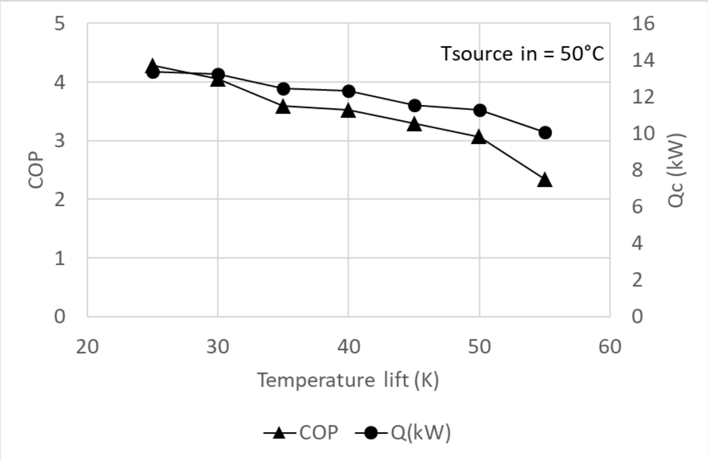

Figure 18 HTHP: Test results with R245fa

From experience of HTHP, it was evident that an oil cooling subsystem is important and the miscibility of the oil during start-up can be seen through photo time lapse although only visual it offers an insight to as to what may be happening to the oil within the sump during operation. Further testing using viscosity measurement equipment would provide a more detailed analysis of the oil during operation.

4.2. Test Methods and Results of Oil-Refrigerant Mixtures

Based on operational issues during testing of the HTHP (Phase 1), the lubricant oil dynamics and its interaction with refrigerant plays a critical role in how well the compressor performs. With higher temperatures >100°C the properties of the lubricant oil and viscosity require a careful selection process to ensure both transport function, mechanical lubrication and seal integrity are maintained within operational limits.

The viscosity at higher temperatures thins the lubricant oil and reduces the lubrication film between surfaces, with refrigerant also present the viscosity is lowered further. Testing lubricant oil and refrigerant mixes at higher ranges can be simulated using laboratory test equipment providing enhanced method to identify suitable lubricants oils with higher viscosity indexes.

The CHESTER project has selected a compressor from VHE for prototype, which has a specific requirement of kinematic viscosity limit during operation. This compressor will use R1233zd(E) as the working fluid and Reniso Triton SEZ320 lubricant during initial trials. This combination has not been tested in relation to compatibility and viscosity particularly at higher temperature ranges.

4.2.1. Test Methods and Other Details

Tests were carried out by Ulster University to determine the oil refrigerant interaction using a dedicated rig and specialised equipment. This testing covered the range between 30°C and

100°C and was based on increased mass percentage (10%-30%) of refrigerant within the lubricant oil. The results based on temperature, pressure and viscosity were compiled and analysed within the scope of this work.

The refrigerant was charged into a small vessel with a storage capacity of 2kg and is immersed in thermal oil used as a heat transfer fluid within the thermal bath. This bath was heated using a Julabo head unit and connected to the lubricant system through a small flexible pipe with a shutoff valve. The refrigerant assembly sits isolated on a scale, providing the charge quantity transferred to the lubricant loop during testing.

The fluid circulation loop was charged with pre-determined quantity of lubricant oil using a container and was drawn into the system under vacuum. The system consists of a small gear pump which has a variable speed control (20Hz-50Hz) to circulate the liquid in the closed loop and acts also to agitate the fluid when refrigerant is added. The loop incorporated a Siemens massflow meter to measure the density/flow rate of the circulating liquid. The viscosity was measured using a Flucon Viscosimeter. The loop circulates through a test cell, which was immersed in a thermal bath and has sight glasses to observe the liquid during testing. The thermal bath also acts as a heat exchange and the loop was coiled and immersed in a thermal oil which is heated or cooled by a Julabo. A secondary 100W heating cartridge element was added within the loop to boost temperature and was thermostatically controlled.

Sensors for pressure and temperature are strategically placed within both the refrigerant side and the lubricant side and are connected to data acquisition. The entire system of pipework and equipment was insulated with lagging and glass wool in difficult spaces to maintain temperature variation below 3K. Figure 19 shows test procedure adopted for viscosity and density testing. In addition, other temperature control procedure was followed to obtain stable conditions.

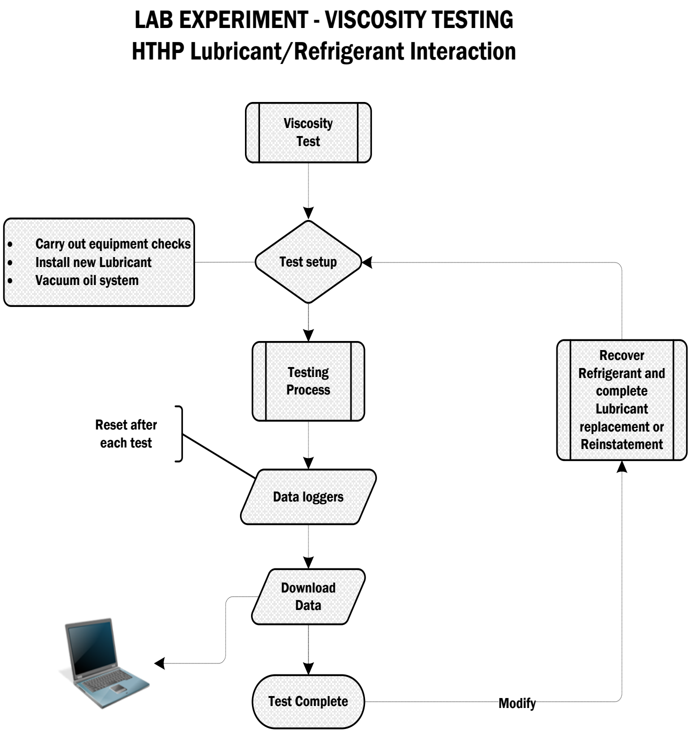

Figure 19 Test procedure for viscosity testing

Details of Lubricants

Reniso Triton SEZ 320 is a fully synthetic oil based on synthetic esters and specifically designed to be used with fluorinated hydrocarbons (HFC/HFO). The oil meets the requirements set out within DIN 51503-1 KC, KD and KE. While SEZ 320 has been available since 2014, data in relation to this lubricant oil's compatibility with new refrigerants such as R-1233zd(E) is currently limited. A second oil Reniso CE VG 500, which has a higher viscosity index was supplied directly from FUCHS in Germany; only limited information is available for this product. Details about pure oil dynamic viscosity and density were obtained from Fuchs and was used for analysis purpose. Table 7 shows an example of SEZ320 oil details used for analysis purpose with R1233zd(E).

Table 7 Technical details of refrigeration oil from Fuchs

		Reniso Triton SEZ 320	
Properties	**Unit**		**Test Method**
Density at 15°C	Kg/m^3	1.016	DIN 51757
Flash point	°C	278	DIN ISO 2592
Color	-	1	DIN ISO 2049
Kinematic viscosity At 40 °C At 100 °C	 mm^2/s mm^2/s	310 33	 DIN EN ISO 3104 ASTM D 7042
Viscosity index	-	148	DIN ISO 2909
Pour point	°C	-42	DIN ISO 3016
Floc point with R 22	°C	<-60	DIN ISO 51351
Neutralization number	mgKOH/g	0.10	DIN 51 558-1
Water content	Mg/kg	<50	DIN 51 777-2

In addition, R1233zd(E) data for viscosity and density at different temperature and pressure test conditions was populated from NIST Refprop which was used later for analysis purpose. Figure 20 shows pressure required in test cell with varying temperature and dynamic viscosity for R1233zd(E) for analysis.

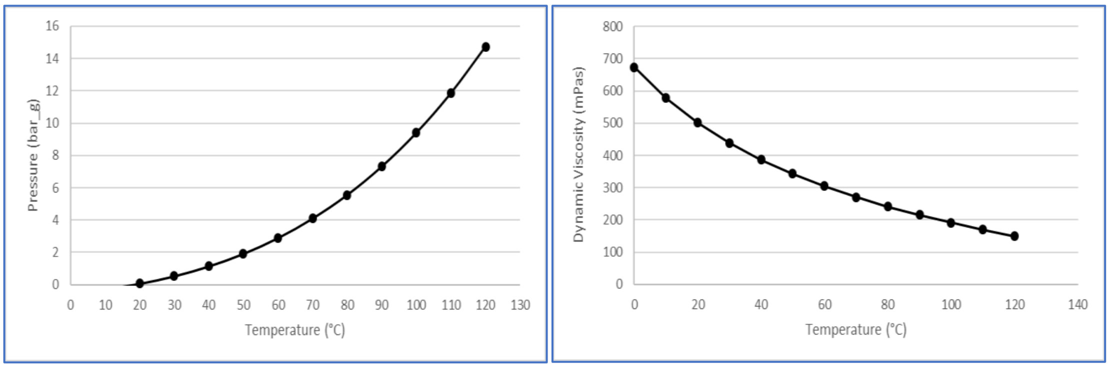

Figure 20 Pure R1233zd (E)variation with temperature: Pressure (left), Dynamic Viscosity (right)

4.2.2. Results and Discussion

The experimental results are broken into two main areas,

1) Testing lubricant oil supplied by FUCHS to provide confidence in relation to reference data
2) Testing refrigerant lubricant oil compositions.

Pure oil:

The lubricant Reniso Triton SEZ 320 and Reniso CE VG 500 by FUCHS was tested across the temperature range 20°C to 100°C, these initial tests were run a few times to check repeatability. The density posed a problem within the test loop as the lubricant tended to 2-phase flow due to the presence of a small quantity of air in the system particularly under low flow conditions. This is highlighted by the manufacturer Siemens [15] within their operational manual. This was checked further and additional checks with the system under vacuum confirming that 2-phase flow was indeed causing the mass flow meter to read inaccurately. Preliminary tests to check the viscosimeter provided outputs that agreed with the reference data from FUCHS. To overcome this issue with the mass flow meter the density data provided was used to calculate the kinematic viscosity. The test bed required a period of thermal equalisation on start-up as inline equipment with large physical mass (mass-flow meter, viscosimeter and oil pump) caused delayed measurement response times, therefore readings were taken from 30°C onwards. Test results showed a margin of error of 3.6% for SEZ 320 and 5% for CE VG500 compared to the reference data across all tests. Figure 21 shows viscosity test results for pure Reniso Triton SEZ320 and comparison with reference data. It showed good agreement and gave further confidence in oil-refrigerant mixture tests.

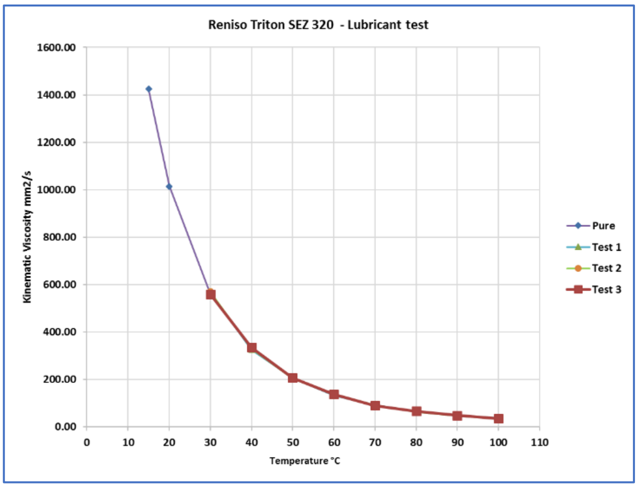

Figure 21 Reniso Triton SEZ 320 lubricant - plot of kinematic viscosity Vs temperature

During initial start-up the lubricant oil at 20 °C was viscous and appeared to contain very small entrainment of bubbles possibly from agitation through the gear pump. Visual assessment at 100°C shows a clear fluid with almost no entrainment of air and a slightly lighter colour (Figure 22).

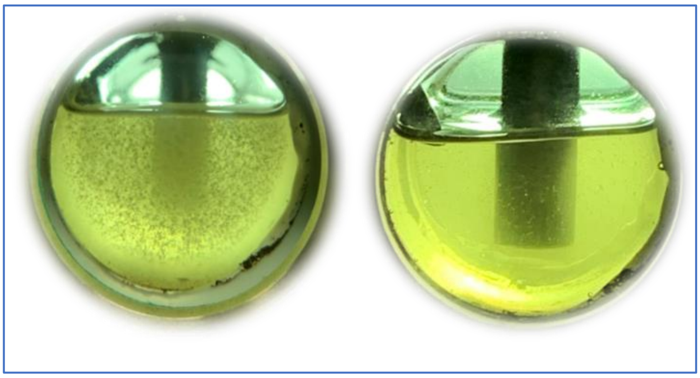

Figure 22 (Left) Pure oil at 20°C during start-up (Right) Lubricant oil at 100°C

Reniso SEZ 320 and R 1233zd(E) mixture:

The initial tests looked at validating the operation of the test rig to manage the introduction of refrigerant to the lubricant loop. These tests ran across the range of temperatures from 30°C to 100°C introducing refrigerant at 10°C lower temperature with associated pressure. This was done at 10°C intervals following the process described earlier. The results (Figure 23) show viscosity vs temperature constantly reducing to 6 cTs at 100°C. The first test was done with pure lubricant oil the second with lubricant oil that had refrigerant recovery completed. In the repeat test it was clear that the lubricant oil properties had changed slightly when combined with the refrigerant, however this would take place under standard operating conditions where the mass concentrations would be in continuous adjustment.

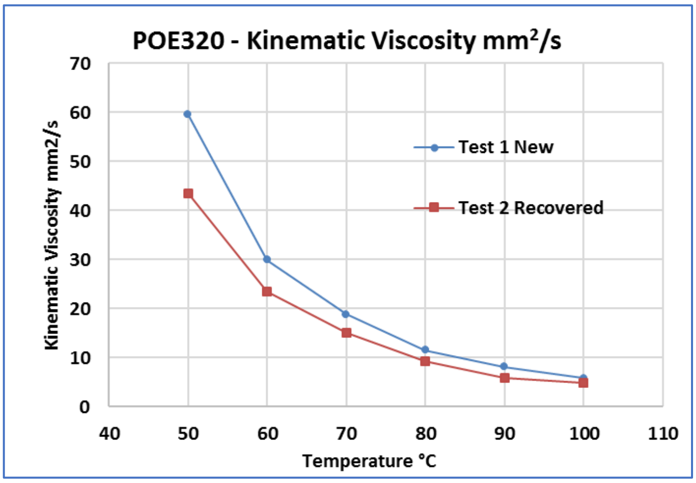

Figure 23 R1233zd(E) + POE 320 viscosity

Two specific tests were done with new lubricant oil for each test (Table 8), these were conducted specifically for the 90-100°C split at 7.3bar(g) pressure. The results both show a kinematic viscosity of 8 cTs at these conditions. This represents the expected parameters of the HTHP in the CHESTER project supplied by TECNALIA. There is a 2cSt variation from the continuous tests (6cSt at 100°) however, the results do correspond well. The variation may be due to the increased duration of refrigerant saturation in the lubricant oil during ramp-up testing, future testing will confirm these findings.

Table 8 CHESTER specific test: Refrigerant lubricant tests 90-100°C split

Test No	Temp. Ref (°C)	Temp Ref_Oil (°C)	Pressure (bar_g)	Density Mix (kg/m³)	Dyn Viscosity (mPas)	Kin Viscosity (cSt)
Test 1	90	100	7.30	1001	8.10	8.09
Test2	90	100	7.30	997	8.40	8.43

During testing with refrigerant lubricant mixtures some pictures of the fluid at various periods was produced. These show similar entrainment of small bubbles at lower temperatures, which progresses to a clear fluid at about 40-50°C. The addition of refrigerant can visibly be seen (Figure 24) at 50°C and it appears to form an immiscible layer below the lubricant oil for a short period (five minutes). Once the refrigerant has mixed with the lubricant oil fully the fluid become regular again with a light-yellow appearance. Adding the refrigerant at increased temperatures and pressures appears to mix more readily and phase separation of the fluids does not appear to occur. The addition of refrigerant clears quickly and the fluid returns to a light-yellow appearance as before.

Figure 24 Reniso Triton SEZ 320 - View of test cell inside thermal bath at different time periods

The next tests focused specifically on obtaining the information needed to build Pressure, Viscosity and Temperature (PVT) charts. These tests followed the same process but used a cool down period to start each new test at different refrigerant mass concentrations. The temperature, viscosity and pressure were recorded during each scenario, providing points of reference to build the charts for Reniso Triton SEZ 320 lubricant and Reniso CE VG 500. Daniel chart was obtained from test results with weight % of refrigerant in the oil and different temperatures. These results were plotted on the logarithmic scale for viscosity and exponential trendlines were created to show regular lines graphically, these were extrapolated to 120°C. The pressure was also recorded during the test.

Reniso CE VG 500and R 1233zd(E) mixture:

After first round of test with POE 320, further test was carried out with VG500. Initial results with pure oil have shown good agreement with supplier data in terms of dynamic viscosity at given temperature. The change over to Reniso CE VG 500 shows similar characteristics (Figure 25) with a deeper shade of yellow in appearance as POE 320. Further test will be carried out at similar conditions to POE 320 for comparative analysis purposes and results will be presented in GA meeting in Nov 2019 at UGent.

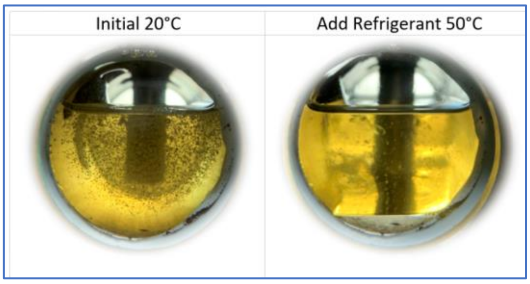

Figure 25 Reniso CE VG 500 - View of test cell inside thermal bath at different time periods

5. Conclusion and Recommendation

Oil interaction and viscosity plays crucial role for safe operation of compressor especially for HTHP application as viscosity of oil decreases with temperature. In addition, high source temperature/pressure (e.g. CHESTER condition) reduces oil-refrigerant mixtures viscosity and requires additional measures. Hence, it is crucial to investigate suitable oil for HTHP which can work with selected refrigerant and performs as per compressor manufacturer requirement. As a part of the deliverable, investigation was carried out in two phases. Test results from Phase 1 was clearly emphasises requirement of EEV and oil cooling. This will help both for longevity of compressor and EEV body/motor. However, trade of between cooling and heat/efficiency loss must be considered. Due to high temperature and high viscosity of the lubricant, it is important to have defined start-up, operation and cool down strategy which involved pre-heating of oil in order to avoid sudden migration of refrigerant in compressor.

Second phase established a process to measure pure and concentrated solutions of oil and refrigerant. During initial trials some issues that cropped up during testing, these were remedied with some small design modifications in between tests and implemented into an established procedure to carry out testing.

The results from testing on lubricants Reniso Triton SEZ320 and Reniso CE VG 500 with R-1233zd(E) refrigerant mixes indicated that there may be justification in moving to the higher viscosity lubricant. The Reniso SEZ320 lubricant oil meets in part the criteria set out by VHE to 10% refrigerant concentration if temperature is maintained around 90°C and if temperature can be maintained up to 65°C then it can work with up to 30% concentration. However, it is unclear without further clarification of the refrigerant concentration and pressures in the HTHP sump during expected operational conditions, whether the lubricant will be suitable. In addition, further clarification form VHE could help to adopt any cooling strategy.

Further tests would be completed with Reniso CE VG 500 lubricant. However, due to limited quantity of sample, repeatability test won't be carried out. Future testing will provide repeatability and additional results needed to create a complete mapping of solutions.

The design of the VHE compressor which has a significant travel distance from the sump to the head, may suggest that significant migration of oil throughout the system maybe negated during operation. However, during cyclical operation lubricant oil entrainment around the system is lightly at certain residual concentrations. Additional information through collaboration with VHE is required to determine how much lubricant oil is circulating in the system and if sump levels are stabilising within the correct levels for operation.

The general conclusion is that replacing to Reniso Triton SEZ320 will provide increased viscosity at operational parameters for the HTHP in the CHESTER project. Additional future evidence of refrigerant concentrations and solution pressures within the HTHP will provide additional supporting criteria to assess the suitability of the lubricant oil to meet system parameters. Further testing with higher viscosity indexed lubricant oil would provide additional evidence in the nomination process for the most suitable solution. Possibility to measure oil-refrigerant concentration and viscosity during HTHP operation would enhance the confidence for system operation and as a novelty of the work.

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