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Abstract

Lubricant plays crucial role in safe and efficient compressor operation especially for high temperature application. Lubricant and refrigerant mixture viscosity analysis and behaviour during heat pump operation requires careful investigation. As a part of European funded CHESTER project, lubricant (oil) behaviour was investigated (visual) experimentally using a high temperature (100°C to 135°C) heat pump test rig developed at Ulster University. In addition, a separate test rig was developed to investigate lubricant oil (POE) and refrigerant (R1233zd(E)) viscosity analysis at source temperature in a range of 50°C to 110°C.

The results from HTHP test rig showed that oil temperature increased with source temperature and in order to protect compressor components and maintain suitable lubricity, additional cooling was required along with a comprehensive start-up, operation and shut-down strategy. The viscosity results of R1233zd(E) and POE mixtures were obtained in terms of a Daniel chart (w% of refrigerant in the oil) in a temperature range of 40°C to 100°C. The viscosity variation showed a range of 80 cSt to 8 cSt between 60°C to 90°C suction temperatures. The analysis also calls for further investigation with other oils and a development of high viscosity grade lubricant suitable for R1233zd(E) and other refrigerants suitable in high temperature heat pump application.

Keywords:
High temperature heat pump, Lubricant, oil, viscosity, compressor cooling

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 [1]. High temperature heat pump (HTHP) could provide energy/carbon emission saving for heating/cooling and integration in existing process or with thermal energy storage or district heating/cooling network could provide further flexibility required for demand side management. 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 [2]. However, HTHP still possess some challenges due to high source/sink temperature, new refrigerant and requires special attention to system components, cooling and lubrication. There are limited investigation using alternative or low GWP/ODP refrigerant in HTHP application such as R1233zd(E) [3]. Refrigerant and lubricant/oil compatibility also plays critical role in terms heat transfer, fluid flow and hence, in overall efficiency. Moreover, oil and refrigerant viscosity varies significantly at suction pressure and temperature and is a less investigate area of research for HTHPs. As a part of EU funded-CHESTER project, HTHP and oil-refrigerant viscosity test-rig were developed at Ulster University to understand oil temperature behaviour and to measure oil-viscosity mixtures at CHESTER test conditions.

2 Methods

In order to assess the behaviour of oil temperature, mixtures in compressor and oil cooling requirements, a HTHP test-rig was developed at Ulster University. The heat pump was designed at Tcon=125°C and Tevp= 50°C with SH=20K (evaporator + liquid-suction heat exchanger) and SC=9K. Further details about test set-up can be found in [3]. After initial tuning, the system was operated using R245fa as a reference at a fixed evaporation temperature (e.g. 50°C) and varying condensing temperature between 85°C to 125°C. Polyester oil HARP POE68 was recommended by the supplier for higher temperature applications viscosity ranges from 65.5 cSt at 40°C to 9.3 cSt at 100°C. Oil temperature and other parameters were measured experimentally whereas oil and refrigerant level was observed visually for analysis purposes.

A separate 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. Two thermal baths were used to conduct the experiments which has an operational range of -20°C to 180°C. Most of the components of this test-rig was bespoke designed and a viscometer was used to measure the dynamic viscosity of oil-refrigerant compositions with a precision of ± 2% deviation. Reference oil calibration and density of oil/mixtures was measured using a mass flow meter. Viscosity assessment was carried out for R1233zd(E) and POE 320 between a temperature range of 40°C to 105°C at different concentration levels. However, for CHESTER project the focus was to determine oil-refrigerant viscosity at maximum suction temperature (e.g. 90°C). A DT85 datalogger was used to record parameters such flow rate, temperature, pressure, density and power. Data was measured at an interval of 30s using two data acquisition system and stored in a dedicated PC for data analysis purpose.

3 Results and Discussion

Analysis for oil temperature in HTHP and viscosity of oil-refrigerant mixture was measured using test-rigs as shown in Figure 1. It was evident form HTHP operation that 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°C when operating at condensing temperature above 100°C as certain viscosity is crucial for compressor operation. Further details on additional cooling and temperature rise can be found in [3].

Figure 1 Test rig used for lubricant analysis at Ulster University: HTHP (left), viscosity measurement (right)

The main study focused on viscosity analysis where investigations were carried out on the second test rig with pure oil POE 320 as a reference. Figure 2 shows pure oil viscosity measurement between 20°C and 100°C. Additional visual analysis at 20°C showed that the oil possesses entrainment of bubbles (perhaps due to gear pump) but disappears at 100°C.

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. The results (Figure 3) 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 4 shows kinematic viscosity of POE320 and R1233zd(E) mixture at 10%, 20% and 30% concertation in the form of a Daniel chart. With 10% R1233zd(E) in mixtures, it provided viscosity in a range of 15 to 140 cSt whereas around 8 to 83 cSt and 6 to 49 cSt at 20% and 30% R1233zd(E) respectively. However, test set-up was designed for dynamic operation and it is difficult to obtain exact amount of mixture for repeatability purposes and pure oil (three tests) average standard deviation of 2.4 cSt is taken as a reference for repeatability purpose and its comparison with manufacture data provided ±0.5% error.

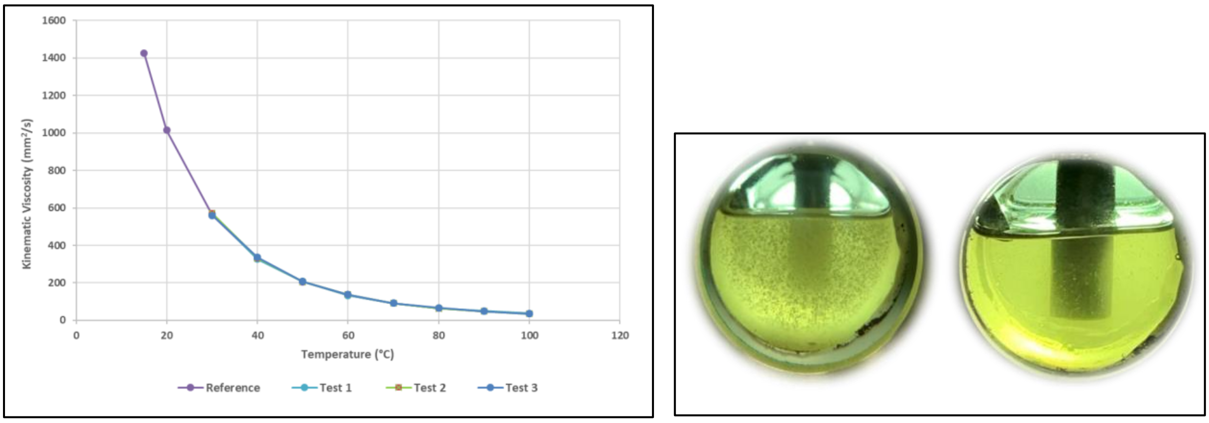

Figure 2 POE 320 (pure oil): a.) viscosity (left), b.) oil visuals (right): @20°C (left) and 100°C (right)

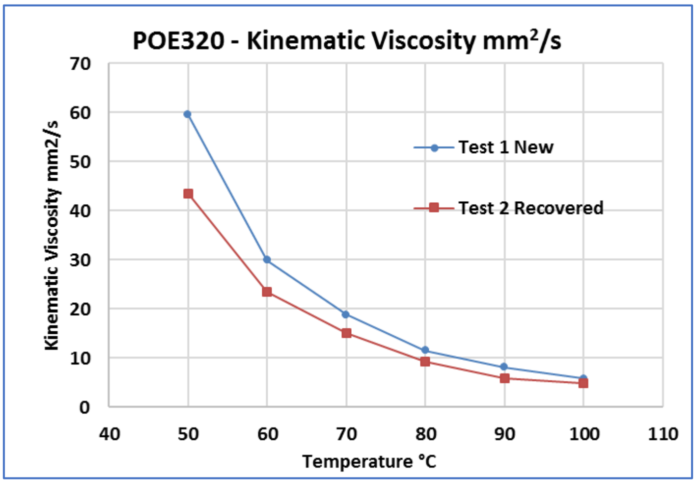

Figure 3 R1233zd(E) and POE 320 viscosity variation with temperature

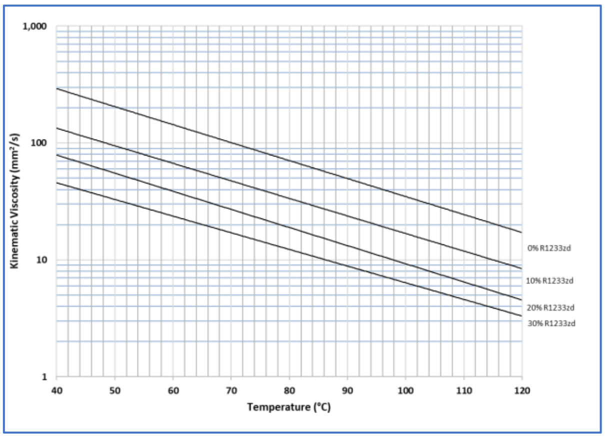

Figure 4 R1233zd(E) and POE320 mixture viscosity (Daniel Chart)

4 Conclusion

Oil interaction and viscosity plays a crucial role for safe operation of a compressor especially for HTHP applications as viscosity of oil decreases with temperature. Test results from the HTHP clearly emphasised the requirement of EEV and oil cooling ensuring longevity of compressor and EEV body/motor. However, a trade-off between cooling and heat/efficiency loss must be considered. Due to high temperature and high viscosity of the lubricant, it is important to have a defined start-up, operation and cool down strategy, which involves pre-heating of the oil in order to avoid sudden migration of refrigerant to the compressor.

The test results from POE320 with R1233zd(E) mixture indicated that there may be justification in moving to the higher viscosity lubricant. POE320 lubricant oil could meet in part the criteria set out by the compressor manufacture 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 concentrations and pressures in the HTHP sump during expected operational conditions, whether the lubricant will be suitable. In addition, further clarification and additional testing and cooling strategy would be investigated as a part of on-going research.

5 Acknowledgement

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6 References

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