Industrial Heat Pumps in the UK
Current Constraints and Future Possibilities

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ABSTRACT

Industrial heat pumps are not common in the UK. However, there is a large and untapped waste heat energy resource that could be upgraded to useful heat in buildings and industrial processes, but there is confusion as to the amount of this heat that is available. Industrial Heat is approximately 300 TWh/year in 2016 (DUKES – ECUK, 2017) and the available waste Heat = 1/6th available for waste heat recovery (Element Energy, 2014) i.e. 50 TWh/year. If this was used for future district heating networks, 28 TWh/year would be utilised in new networks (Element Energy, 2014). Assuming a current household gas price of 3.69p/kWh (British Gas), the utilisation of 28 TWh/year will displace over £1Bn of gas purchases annually. However, UK industrial electricity prices for large energy users are almost 60% higher than the EU-15 median, while for medium and small/ medium users, the price differential is over 40%. Higher performing heat pumps may provide an answer.
Keywords: Industrial Heat Pumps

1. INTRODUCTION

Heat pumps are undoubtably an element of decarbonisation of heating. Exceptions include high temperature processes (steel, cement etc.) where biogas and/or hydrogen may be more suited in a near zero carbon future. However, despite clear indications that near decarbonisation is a possibility in the UK, there are considerable challenges for lower temperature heat processes that would consider using heat pumps.

1.1 UK energy pricing – a challenge for Electrically Driven Heat Pumps?

Energy price data from Eurostat (2018) reveals electricity and gas price comparisons for industry across Europe (Figure 1). While the UK does not have the worst comparator of electricity versus gas price (UK = 3.39, Italy > 4.04), it is none the less second placed in this league table and as a consequence, gas price is low relative to electricity price. In fact, the UK has the second lowest unit price in Europe at 2.594p/kWh for gas with Denmark the highest at 6.478p/kWh (149% higher). For electricity, the UK has the 10th most expensive rates at 8.794p/kWh with Cyprus the highest at 16.661p/kWh (89.45% higher). For this analysis, the comparator is of interest. Furthermore, given the increased capital cost of an electrically driven vapour compression heat pump versus a gas boiler of similar heating capacity, a heat pump seasonal performance factor (SPF) of 2.71 is the break even in terms of running costs, taking no account of any capital cost differences. This SPF is that necessary to overcome the price differential between electricity and gas when operating an electrically driven vapour compression heat pump to provide a heating demand when compared to providing heat from a heat pump when compared to providing heat from a conventional gas boiler. The latter is the most common approach of industrial space and process heating in the UK.

However, the Renewable Heat Incentive in the UK can provide some assistance in capital cost challenges with a 500 kW$_{thermal}$ heat pump operating for 5000 hours per annum receiving over £2000/annum of additional support. However, an industrial air source heat pump will cost approximately £140/kW$_{thermal}$ versus a gas boiler of £50/kW$_{thermal}$ and would require a payback period

of approximately 20 years at the break-even SPF of 2.71. Therefore, an important factor is the Seasonal Performance Factor.
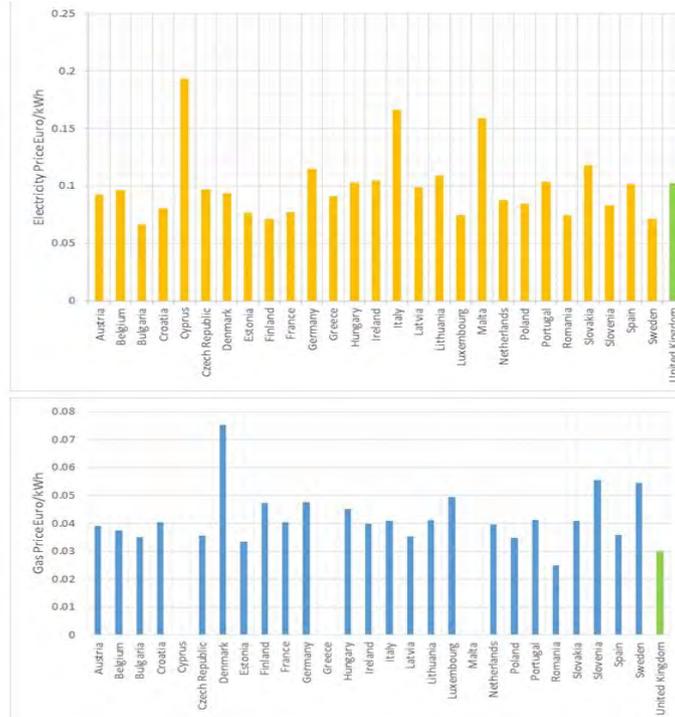

Figure 1: Electricity and Gas Price Comparison for UK Industry

1.2 Performance of UK Industrial Heat Pumps

Hughes (2018) analysed the performance of 19 monitored trials of commercial and industrial heat pumps in the UK. These were a mixture of ground source and air source units and seasonal performance factors were considered ranging from the heat pump itself, the heat pump and its internal fans and pumps, and the overall system performance including all auxiliary elements. Figure 2 illustrates the impacts of this review in terms of SPF. Regarding SPF, SPFH1 – Heat Pump only, SPFH2 – Heat Pump and Pumps/fans etc., and SPFH4 – System performance including all auxiliaries. Keeping in mind the approximate estimate of break-even SPF as being 2.71, the vast majority of heat pumps are above this point, recognising that this is very application dependant. These are noted in Figure 2 as SPFH1 (heat pump performance).
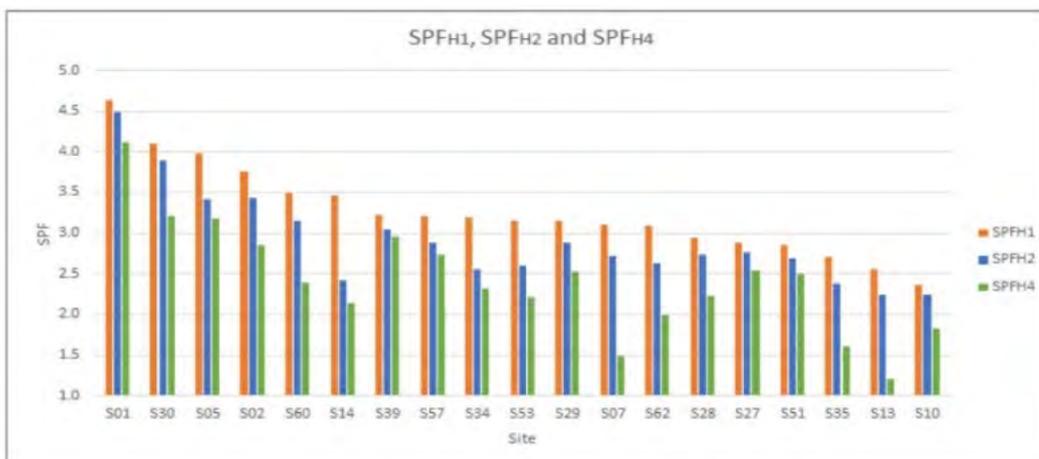

Figure 2: SPF Comparison of Monitored Commercial and Industrial Heat Pumps in the UK

However, the challenge is in the system and auxiliary components design and use. Therefore, in the following analysis, improvements to the heat pump, improvements to the fans and pumps and overall auxiliaries will be considered. The scatter in the examples of Figure 3 tends to indicate that there are

huge improvements in both heat pump and its installation, but a general improvement is in the lower the temperature lift (application dependant) and increasing the heat delivery heat transfer area i.e. underfloor heating and/or larger radiators (building dependant). These are well known but need constant reiteration coupled with end-user acceptability reinforcement to a different style of space heating delivery.

2. Improving an Industrial Heat Pump

In operating industrial heat pumps, a number of elements are considered. These include improving the overall performance of those operating under traditional space heating conditions to extending the range for process heating applications. The role of alternative heat sources must also be addressed.
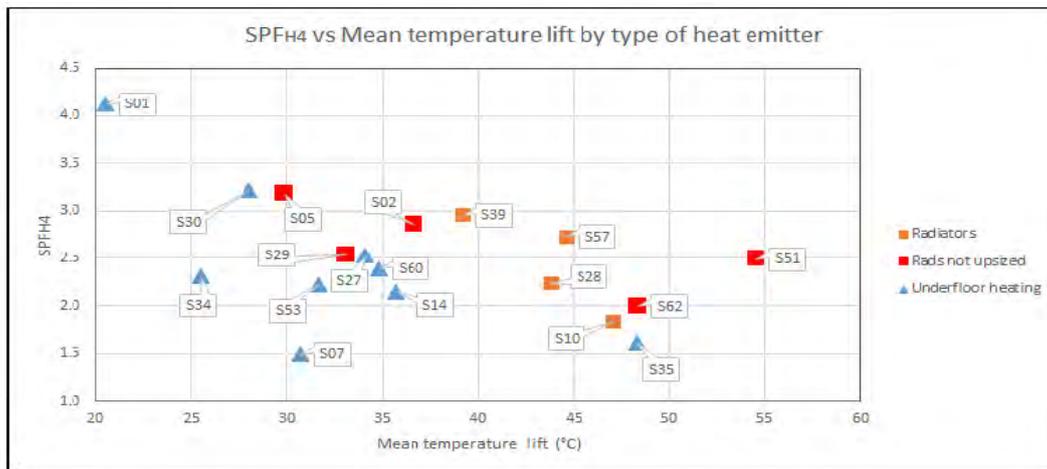

Figure 3: Space Heating Applications for UK Commercial and Industrial Applications

2.1 Replacing R410a in Industrial Applications

In considering how to improve heat pumps, low Global Warming Potential (GWP) working fluids must be utilised. A traditional heat pump i.e. that heating an underfloor hydronic system may replace R410a with R32. This will see a drop in GWP from 2088 to 675 and the use of R32 will deliver a COP that is similar to R410a e.g. Alabdulkarem et al (2015) or superior to R410a e.g. Mota-Babiloni et al (2017). Heat pumps that utilise existing high temperature hydronic radiators could utilise R410a at the lower temperatures encountered e.g. <60°C but see superior performance with R32 with its superior critical temperature, heat transfer and lower pressure drop (Botticella et al, 2018). R32 compares favourably to other proposed replacements as the modelled results indicate (Figure 4).

2.2 Utilisation of Low Temperature Heat Networks for Industrial Applications

A low temperature heat network (or 4th Generation heat network) is one that can supply low-temperature district heating for space heating, domestic hot water and process heating to buildings and industry. A 4th Generation Heat Network will have low grid losses, be able to utilise waste heat from low-temperature sources and integrate renewable energy sources such as solar and wind as part of a smart energy system to be flexible in the future (Lund et al, 2014). Low temperature is defined by the Danish as >50°C and ultra-low networks as <45°C (Danish Energy Agency, 2014). The latter would require heat pumps for upgrading to traditional space heating temperatures, for domestic hot water use (booster heat pumps) and beyond to industrial process applications. UK low grade waste represents a significant energy resource. While there are discrepancies on the actual values, industrial heat represents 300 TWh/year in 2016 (DUKES, 2016) and 1/6th of the waste heat is available for waste heat recovery (Element Energy, 2014) i.e. 50 TWh/year. Furthermore, 28 TWh/year has been identified for "potential" district heating networks and assuming a current household gas price of 3.69p/kWh, utilisation of 28 TWh/year will displace over £1Bn of gas purchases annually. However, the UK has the equivalent of 13 TWh of existing heat networks (BEIS,

2016) and assuming network construction costs are £1500/MWh, new network costs for 15 TWh/year lead to £22Bn investment. As gas is assumed to produce 0.19 kg/kWh of CO_2 and If the UK can utilise 28 TWh of waste heat displacing gas for space heating, 5,320,000 tonnes of CO2 will be saved. As an example, if this was utilised for domestic space heating, UK residential buildings currently emit 63.4 Million tonnes of CO_2/year,10% of residential CO_2 emissions can be saved if waste heat is fully utilised.
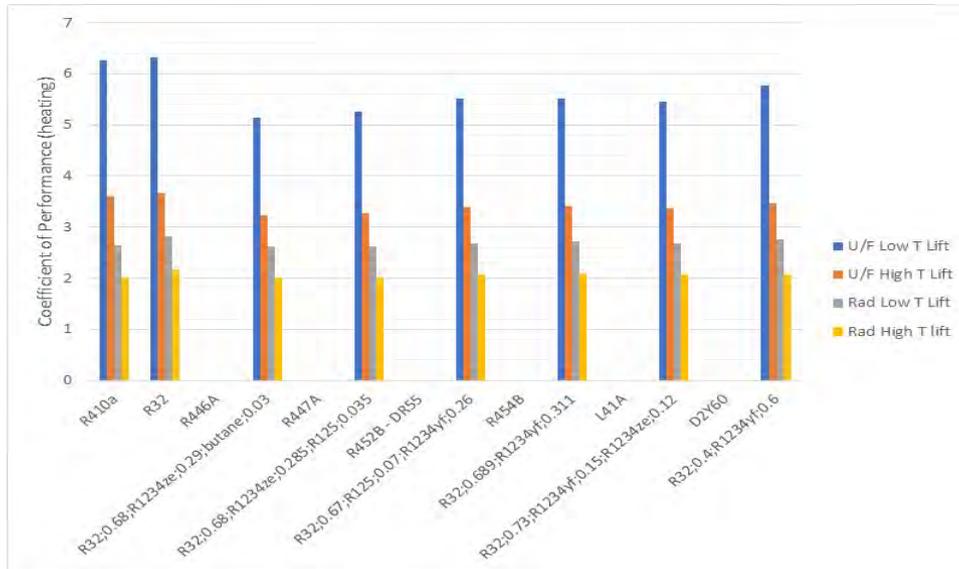

Figure 4: R410a Replacement Comparisons.

With respect to heat pump development for waste heat recovery, the EU is cutting the availability of hydrofluorocarbons (HFCs) by 79% between 2015 and 2030 (BEIS, 2015) due to high Global Warming Potential and alternative fluids must be evaluated. The vapour compression heat pump operating in conjunction with lower grade heat networks e.g. about 30°C, can uplift such heat to underfloor heating temperatures (<40°C) with a coefficient of performance >10 (very low running costs) and provide domestic hot water temperatures avoiding legionella concerns (~60°C) with a coefficient of performance >5.0 with current refrigerants e.g. R410a. However revised heat exchangers must be developed (Lee et al, 2013). For thermal networks, alternative fluids such as R245fa have realised coefficients of performance of >7.0 for ~60°C (Figure 5).
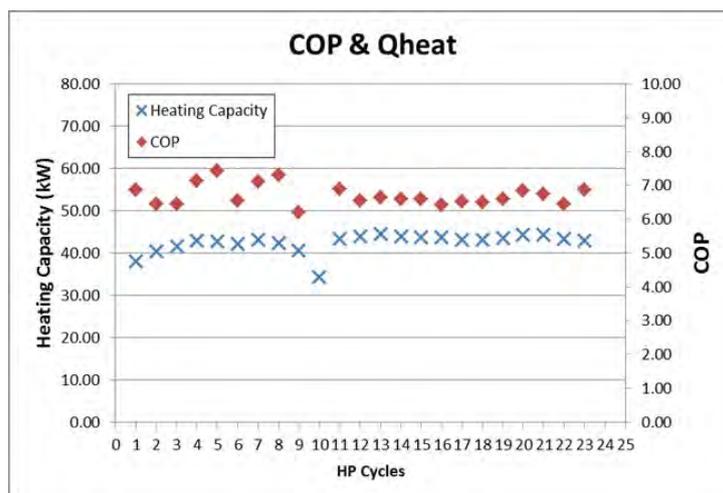

Figure 5: R245fa Performance in a Seasonal Thermal Storage Operation

Extending the temperature lift further sees low GWP fluids such as R1233zd (Ju et al, 2017) being able to upgrade low grade waste heat to process temperatures. Higher temperature heat pumps have been successful and new fluids such as R1233zd (>150°C, low GWP) are looking promising

(Figure 6). However, compressor lubrication strategies for higher temperatures with existing equipment have not been explored to give commercial confidence.
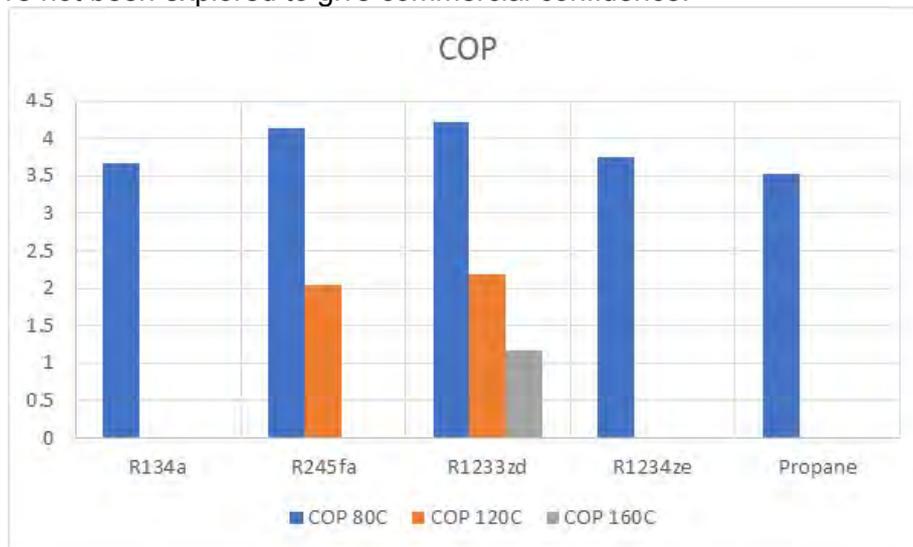
Figure 6: Alternative Refrigerant Performance at High Temperatures

3. High Temperature Heat Pumps Using R1233zd(E)

Hydrofluro-Olefins (HFO) R1233zd is an A1, non-toxic, non-flammable refrigerant with a GWP of 4.5 (under AR4). R1233zd is normally described as zero-ODP due to its very low ODP and is measured at 0.00034. Its physical properties are similar to R245fa and like R245fa, when used in a R134a compressor, both R245fa and R1233zd have higher specific volumes and therefore require larger components e.g. compressor and heat exchangers. They also require a suction line heat exchanger due to the re-entrant nature of the compression process at low superheats. The HFC gases that have been used so far for high temperature applications are R134a, R152a and R245fa and their GWP is 1300, 140 and 1030 respectively. Honeywell has developed a number of HFOs for high temperature applications with a very low GWP (<1). Working fluids such as HFO-1234ze(Z), HFO-1234ze and HFO-1233zd with critical temperatures higher than 94°C are the main candidates for high temperature applications (Longo, 2014). In addition, natural refrigerants have been always an interesting topic for heat pump developers due to their inexistent GWP and low cost. However, their flammability is very high and requires special safety equipment.

3.1 High Temperature Heat Pump Working Fluid Evaluation

Typically, a high temperature heat pump normally has a range of supply temperatures >80°C with typical heat source temperatures 20°C to 70°C. These heat pumps use low grade waste heat from primary processes as the source within a HP cycle to achieve higher secondary supply temperatures. Industries where this type of heat pump may be used range from paper, chemical, mechanical and textile industries. The Ulster University test facility (Figure 7) was built into a specially designed rig and uses mainly off the shelf components with some modification to mitigate against higher temperatures developed within the working system (Bitzer Heat Pump 1-stage cycle, Danfoss expansion valve and an optimization by IHX). A separate Thermal Balancing Rig (TBR) was also designed, which incorporates bespoke control systems to manage thermal loads within the secondary and tertiary hydraulic circuits. The combined systems of heat pump and TBR forms an integrated system and has fitted a range of controls and sensing devices necessary to operate and derive data for experimental analysis. Baseline performance was achieved with R245fa and is illustrated in Figure 8. Comparing R245fa and R1233zd(E) reveals the following analysis (Figure 9) and seems to suggest that R245fa performs slight better at higher temperature demands. However, these are preliminary results.

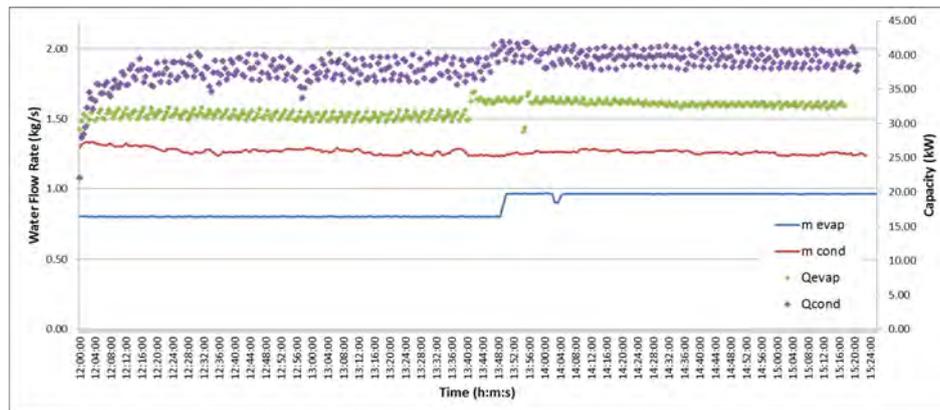**Figure 7: Ulster University High Temperature Heat Pump Test Facility**

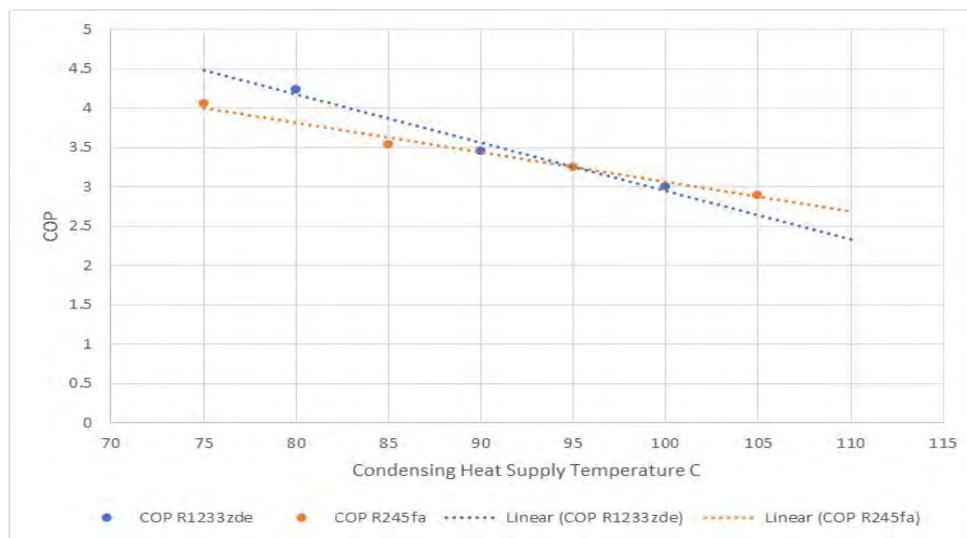**Figure 8: R245fa Baseline Performance**

Figure 9: COP Comparison

3.2 Compressor Lubricant Oil Observations

During initial setup the compressor was ramped up gradually to condenser temperatures above 120°C, a small leak from the outer housing was observed. The OEM guidance does suggest that a small amount of oil may pass through the seals dependant on operating conditions. However, this leak gradually increased over several tests and a decision was made to check and replace seals. To repair the leak new seal kit was installed. This required the separation of the motor from the heat pump and careful removal of the damaged seals and installation of the new kit. There was no apparent visual damage to the outer seals, but the inner seal rubber was hardened compared to the new seal. The oil sump-plug magnetic core did pick up some swarf but was not excessive and maybe attributable to the running in process of the compressor. The refrigerant filter was removed and again a visual examination completed, this showed a small quantity of particles around the inlet but no major build-up of dirt. The pressure drop across the filter was about 0.2 bar during operation, which

was only a slight amount higher than when first installed. The filter was replaced as a precaution and checked for leaks prior to operation. The compressor was refilled with POE68 oil to the required levels considering the additional requirement of 0.6l as specified for the oil separator. Upon running the compressor over a period of 6 months the same grey film has not reappeared on the glass surface. Polyester oil HARP POE68 was recommended by the suppler for higher temperature applications exceeding DIN51503 part 1 (Bitzer). Based on technical guidance (Figure 10) for this oil, the 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. 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. The compressor was refilled with POE68 oil to the required levels considering the additional requirement of 0.6l as specified for the oil separator. Upon running the compressor over a period of 6 months the same grey film has not reappeared on the glass surface.
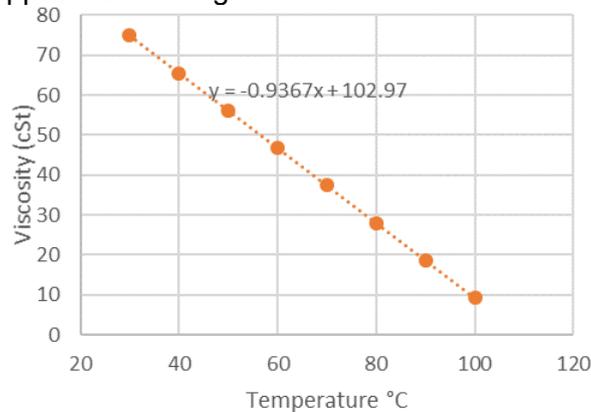
Figure 10. Viscosity Vs Temperature based on product technical data (HARP)

During the initial start-up the compressor goes through a phase whereby the oil and refrigerant mix readily (as seen in the oil levels and colour within the compressor sump - Figure 11). 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. In this compressor it takes approximately one hour to reach stability. Therefore refrigerant/oil interactions need further investigation.

Figure 11: System Start Oil Observations

4. Discussion and Conclusions

A number of observations can be made from this work from a UK context. Energy pricing and subsidy support are not yet sufficiently favourable for wide scale deployment of electrically driven vapour compression heat pumps. The current gas price is sufficiently low, and boilers are of low capital cost to ensure boilers are the heating system of choice. However, there are cases were that trend can be overcome. UK Government energy statements such as "near 100% decarbonisation" by 2050 will see a rise in renewable electricity dependency and a need to decarbonise the gas network

(biomethane or hydrogen for example) to reduce expensive electricity network upgrades and to utilise an existing and wide-reaching gas infrastructure. However, if hydrogen were utilised, there would be a significant requirement to improve the performance of electrolysers and safety concerns would limit hydrogen to the low-pressure gas distribution network. Of course, thermal storage has not been mentioned as a useful system addition to electrically driven heat pumps so that such heat pumps can utilise off-peak or excess renewably generated electricity. There is an emerging literature that suggests higher temperature heat pumps would benefit from thermochemical storage due to higher energy densities at higher temperatures. Thermal storage would facilitate demand side response and distributed energy management. UK industrial heat pumps (and indeed heat pumps in general) also suffer from performance challenges. These are invariably brought about by the overall system performance and not necessarily by the heat pump itself. Given the capital cost differential between boilers and heat pumps and the relative simplicity of boiler operation, the added expertise in fully understanding an installation challenge can be sometimes underestimated. There are great examples of excellently performing heat pumps but unfortunately there are others which are not doing so well. A challenge is of course the operating regime. District heating is not a common option in the UK but if there are serious efforts to address the available waste heat. Booster heat pumps upgrading the heat from low temperature, low loss networks are possible. R410a and R245fa were potential candidate fluids for space heating and process heating applications but their global warming potential is unacceptable. R32 appears to be an excellent candidate for R410a replacement (accepting a small safety risk) and R1233zd(E) has potential to replace R245fa. Initial results are promising but compressor lubrication and transportation (solubility and miscibility) requires further investigation.

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Sorption heat pump for flue gas condensation
of biomass-fired boilers

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ABSTRACT

Extracting heat from hot flue gas by utilization of condensing technology leads to a significant increase in boiler efficiency. Conventional systems are often limited by unfavorably high return temperatures of the heating system which preclude the use of the extensive potential of the sensible and especially latent heat of the flue gas.

In this research the integration of an absorption heat pump (AHP) into the heating system of a wood chips fired boiler is investigated. This allows cooling of the flue gas down to temperatures around 25 °C and serves for upgrading the low grade heat for further utilization in conventional heating systems. Therefore, increases in fuel utilization rate up to 20 % can be achieved.

The concept of the heat pump cycle for integration with the biomass boiler is presented, followed by thermodynamic modelling and thermal design of a flue gas-coupled evaporator of the heat pump.

Keywords: Absorption, Biomass, Energy Efficiency, Heat Pump, Heat Transfer, Latent Heat, Water-Lithium Bromide

1. INTRODUCTION

Heat recovery of flue gas reduces the flue gas losses and increases the boiler efficiency. Cooling the flue gas down to temperatures below the dew point allows the transfer of energy released by sensible heat and by latent heat. Condensing the vapor fraction of a flue gas flow is an established technology for heat recovery of gas and oil fired boilers. To ensure the full utilization of the heat content of the flue gas, low cooling temperatures are required. Commonly, the return flow of the heating network is used to cool the flue gas down. Thus, the heat gain is limited by the temperature level of the heating system. For standard local heating networks with return flow temperatures of 50 °C and closest approach temperature (CAT) between flue gas and heating medium of 15 K, compared to manufactural data (Rennergy Systems AG, 2016; Fröling GesmbH, 2017), condensation process hardly starts. Wood chips fired boilers show high losses due to flue gas outlet temperatures of around 150 °C and water vapor contents of 0.65 kg per kg wood even for combustion of biomass fuel with low water content (w=20 %) (Hartmann et al., 2004). Heat recovery can increase conventional boiler efficiencies from 85–90 % to over 100 %, if use of latent heat content is accomplished. When heat recovery is supported by a heat pump, cooling temperatures independent of return flow temperatures of the heating system can be provided. Absorption heat pumps (AHP) require only a fraction of electrical energy needed by compression heat pumps for upgrading the low-temperature heat. In the given application the driving heat for the heat pump cycle can be supplied by the biomass boiler. Thus, heat pump operation is almost completely based on renewable energies.

Figure 1 shows the boiler efficiency based on lower heating value with respect to flue gas temperature. For the combustion of wood with a water content of 30 % and an air-fuel ratio of λ=1.5 the boiler efficiency for flue gas outlet temperatures of 140 °C is around 90 %. By cooling down to 65 °C, feasible through common heat exchanger and the temperature level of the return flow, boiler efficiency is increased to 96 %. Cooling down to 25 °C, by using an AHP, results in a further increase to roughly 110 %. The sharp increase at around 55 °C is caused by the onset of condensation when flue gas temperature falls below the dew point. (Kaltschmitt et al., 2009; Joos, 2006)