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Design and Experimental Results of a Two-Stage Steam Producing Industrial Heat Pump

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Abstract

This paper describes the design and experimental analysis of a steam producing industrial heat pump system. The heat pump is a two-stage flooded system with a thermal capacity of 150 kW_{out}, capable of producing medium pressure steam (4.8 bar_{abs}, 150°C) with up to 90°C temperature lifts. The heat pump takes up heat at two distinct temperature levels and in this way can take advantage of temperature glides, typical of waste heat streams in industrial processes. It has been tested with a single working fluid, namely the hydrocarbon Pentane. The heat pump is tested using a specially designed test rig, capable of simulating conditions typical of numerous industrial processes. Experimental results are presented for various operational conditions (temperatures, powers) using both the single-stage and dual-stage cycle. The heat pump is shown to be capable of producing steam temperatures at the design point of 150°C. The results demonstrate that steam producing heat pumps are a feasible technology option for industrial applications. There are however steps needed for mass uptake, especially related to upscaling powers to the 1-5 MW range.

Keywords: Industry; High temperature heat pump; Waste heat; Pentane; Pilot scale system;

Nomenclature

Symbols		out	Stream flow out of system
β	Heat source ratio (-)	sink	Thermal sink
η	Efficiency (-)	source	Thermal source
$\dot{Q}$	Heat transfer rate (W)	steam	Steam production
T	Temperature (K)	th	Thermal
$\dot{W}$	Rate of work, power (W)		
		Abbreviations	
Subscripts		COP	Coefficient of performance
abs	absolute	GHG	Greenhouse gas
Carnot	Carnot cycle, theoretical maximum	GWP	Global warming potential
elec	Electrical	HP	High pressure
in	Stream flow into system	LP	Low pressure
LT	Low temperature	MP	Middle pressure
MT	Middle temperature	ODP	Ozone depletion potential

1. Introduction

The European Union and its 28 member states (EU28) have set the ambitious target of reducing greenhouse gas (GHG) emissions by 80-95% relative to 1990 levels by 2050. In order to achieve this, there is the requirement to transform the current energy system, primarily by switching to renewable energy sources and implementing energy efficiency measures. This will be particularly relevant for the industrial sector in the coming years, which at present (2018) is responsible for 11,590 PJ or 25% of final energy consumption in EU28 [1].

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Industrial energy usage in EU28 is dominated by the demand for thermal energy, which is estimated to be responsible for 73% of total consumption (2012) [2]. The heating demand is at present covered almost entirely by fossil sources, namely natural gas, oil and coal. The remaining energy demand, covered by electricity, is mostly required for mechanical work, for example to drive fans, compressors, pumps and other machinery. Based on this, it can be determined that large reductions in GHG emissions from industry will only be achievable through radical changes in the supply and use of heat, implying the need for significant changes to existing processes. The challenge for industry to transform their heat supply is made greater by the requirement for heat use at different temperature levels, from below 100°C, to well above 1000°C, depending on the given process or sector.

The use of heat pump technology in an industrial setting is an option which has the potential to cover a large share of the low temperature (<200°C) heat supply, estimated to be 45% of total industrial heat demand (incl. space heating) [3]. Heat pumps are a technology which take waste heat, typically discarded to ambient due to the low temperature level, and upgrade this to temperatures required by the process. This is done with the input of electricity. When renewable electricity is used as an input source, the GHG emissions from the process can be reduced to zero. In any case, the electricity input required is less than the original energy input (gas, oil, etc.) to the process, meaning that implementing heat pumps will necessarily lead to reductions in the final energy consumption.

One of the main drivers currently prohibiting heat pumps in industry, is the fact that the majority of commercially available heat pumps are currently limited to supplying process heat up to temperatures of approximately 90°C. There is currently a limited demand for heat in industry up to this temperature level. Increasing the maximum heat supply temperature of heat pumps above 100°C such that they have the capability to produce steam will increase the market potential of the technology significantly. Further increases in the maximum temperature towards 200°C will lead to further application potential, making the technology appropriate for numerous processes particularly in the chemical, paper and food sectors.

Whilst there has recently been increased focus in industrial heat pump technology in both academia and the refrigeration and heat pump sectors, the developing nature of the market ensures that further demonstration of the technology is needed. In the Dutch nationally funded project 'Sustainable sTEam Production for induStry' (STEPS), a pilot scale, steam producing industrial heat pump was designed, constructed and tested. The goals of this development project were as follows:

- To demonstrate that an industrial heat pump, producing steam at temperatures up to 150°C (4.8 bar_{abs}), can be manufactured from components which are currently available in the market
- To identify if there are necessary developments needed from component and equipment manufacturers as well as system suppliers
- To accelerate acceptance and uptake from industrial end-users of heat pump technology

The project was carried out with a consortium of partners consisting of a research institute, equipment manufacturers and system suppliers, as well as industrial end-users of the technology from multiple industrial sub-sectors (paper, chemical, refinery, food and beverage). Of this consortium, the design and construction of the heat pump was realized by refrigeration and heat pump system supplier IBK, whilst testing and subsequent analysis of results was conducted at the industrial heat pump test centre at TNO. This remainder of this paper provides details relating to the design of the heat pump system (section 2) as well as results from testing under representative industrial conditions (section 3).

2. Heat Pump Design

Details relating to the design and technical characteristics of the heat pump are given in this section. Firstly, a description is given on how design conditions for the heat pump were determined and the effect this has on the system requirements. Following this, information is provided on the physical system itself, including the process scheme and component information. Finally, details are provided on the specially designed test rig, which was connected to the heat pump to simulate the conditions that would be imposed on the heat pump by an industrial process.

2.1. Design Conditions

Full scale industrial heat pump systems will be characterised by large thermal powers, typically in the range of 1-5 MW_{out}. With the intention in the project to develop a pilot scale heat pump system, a nominal thermal power of 150 kW_{out} was targeted.

Design conditions for the constructed industrial heat pump were selected based on input received from the multiple industrial end-users involved in the project. The end-users were requested to provide data relating to typical conditions (process and waste heat quantities as well as temperature levels) whereby a heat pump could be applied within their process. On the basis of the information received, the design conditions and requirements of the heat pump were determined.

It was determined that to satisfy the requirements of the multiple end-users, the heat pump should be designed to function with three operational concepts:

1. Single evaporator, single compressor
   The evaporator temperature varies between 52°C and 97°C, whilst the target steam temperature varies between 120°C and 150°C. The use of a single compressor ensures this operation mode is most suitable for low to moderate temperature lifts.
2. Single evaporator, two compressors
   The evaporator temperature varies between 52°C and 97°C, whilst the target steam temperature varies between 120°C and 150°C. The use of two compressors in series ensures this operation mode is most suitable for moderate to high temperature lifts. Furthermore, this cycle implementation provides efficiency (coefficient of performance - COP) enhancement when compared with the single compressor cycle.
3. Two evaporators, two compressors
   The single system operates with two evaporators which have two distinct pressure (temperature) levels. The low temperature evaporator varies between 52°C and 97°C, whilst the high temperature evaporator varies between 97°C and 111°C. The target steam temperature in this case is also in the range of 120°C to 150°C.

Whilst conventional heat pump cycles consist of a single source and sink, the requirements of the end-users have necessitated a multi-temperature heat pump whereby waste heat can be input at two locations. There are two main scenarios whereby a multi-temperature (source) heat pump is advantageous for use in a process:

1. Where multiple waste heat sources are available at differing temperature levels.
2. Where the waste heat source has relatively low energy content, leading to a large temperature glide on the waste heat stream in order to produce the required thermal power on the sink side of the heat pump.

The efficiency of a heat pump cycle, also known as the COP relates the heat output in the condenser to the electrical work input. The theoretical maximum COP that can be achieved is determined by the temperature of the heat sink as well as the temperature lift required to be achieved by the heat pump (see section 2.4). Increasing the temperature lift has a detrimental impact on the maximum achievable COP. By implementing a multi-stage (single evaporator, two compressors) or multi-temperature (two evaporators, two compressors) cycle, the COP is enhanced as the entirety of working fluid in the heat pump cycle does not need to be compressed from lowest temperature (pressure) level to highest temperature (pressure) level, therefore limiting the temperature lift for part of the system. This leads to a lower electrical energy input for a given process heat output.

Whilst multi-temperature heat pumps impose additional complexity on the system design and associated control system for operation, their use is not uncommon. There are numerous cycles which can be used to implement multi-temperature heat pump cycles. A comprehensive review of the various multi-temperature heat pump cycles is given by Arpagaus [4].

2.2. High Level Design

The heat pump construction was based on a previously designed, constructed and tested system, an overview of which can be given in the works of Wemmers et al. [5]. The original heat pump design was based on a single stage system (single evaporator and compressor) and used Butane as the working fluid. To fulfil the requirements of the end-users, a number of modifications had to be made to the unit, namely to convert it to a multi-stage and temperature unit, as well as adapt the system to operate with a new working fluid. The new working fluid was needed as the critical temperature of Butane is 152°C, making it unsuitable to operate at the desired sink temperatures of 150°C in a sub-critical cycle.

Pentane was chosen as the new working fluid in the system for the following reasons:

- It is a natural working medium which has zero ODP and low GWP (~4), adding to the sustainable credibility of the technology
- It has a high critical temperature (197°C) and low boiling point (36°C) relative to the range of design and operation conditions (52°C – 150°C) to be tested
- It has good thermal stability, up to the temperatures of interest

As was the case with the heat pump described in [5], the modified unit was constructed inside a container, which allows it to be easily transported and placed at various industrial sites as needed. Rather than test the heat pump directly at an industrial site, the heat pump was tested on a specifically constructed heat pump test rig, also placed within a container unit. There are a number of benefits to this approach, the most important being:

- Issues and delays associated with start-up and commissioning can be resolved without affecting or interfering with the production process
- The heat pump in the first instance is decoupled from the process, making it much simpler to troubleshoot performance or reliability issues
- The performance of the heat pump can be characterised in a controlled environment for numerous operational conditions

An image of the containers housing the heat pump, as well as the test infrastructure can be seen in Figure 1 which follows. These are provided to give the reader an indication of the physical sizing of the systems.

Fig. 1. Containers which house the heat pump (white container) and test infrastructure (blue container) and associated connections

2.3. Process Flow Diagram

A process flow diagram of the heat pump system is seen in Figure 2. As indicated previously, the heat pump is a two-stage system, which allows heat input at two locations. The heat pump is characterised by the two piston compressors as well as the two separator vessels which feed liquid refrigerant to the evaporators in a thermosiphon type arrangement. Additional notable characteristics of the system are described as follows.

<u>Suction Gas Superheaters</u>

A notable characteristic of the working fluid Pentane is that isentropic compression from a (near) saturated vapor will result in the compressed fluid entering the two phase region. Liquid droplets in the compressor can have a large impact on the longevity of the system and may even lead to complete failure. Two suction gas superheater heat exchangers make use of the high temperature liquid from the condenser and middle pressure (MP) separator vessel, transferring heat to the gas in the compressor suction lines. These heat exchangers move the compressor suction gas sufficiently away from the two-phase region, therefore ensuring a limited amount of superheat in the discharge gas for both compressors.

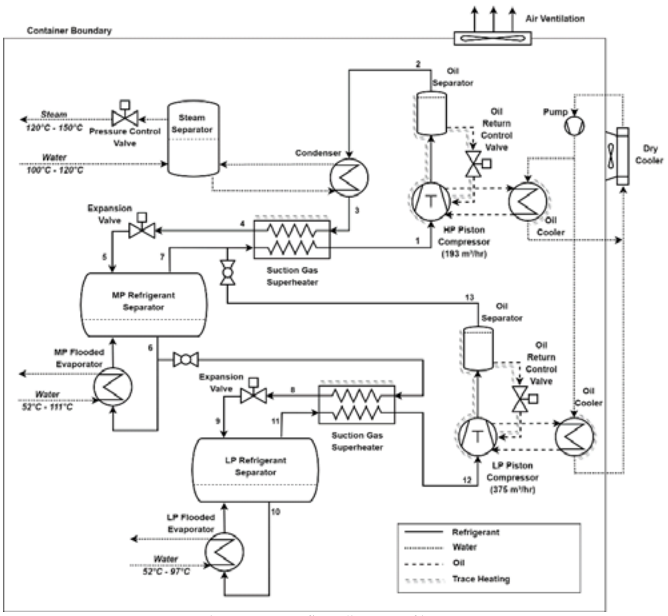

Fig. 2. Process flow diagram of heat pump

<u>Condenser and Steam System</u>

The heat pump utilises a split condenser system (only a single condenser pictured), located parallel to one another. On the water side of the system, the condensers are fed by a steam separator vessel in a thermosiphon type arrangement, with a two phase mixture returned to the vessel from the heat exchanger. The steam separator vessel is fed by the test rig to maintain a fixed liquid level, whilst pressure (and therefore temperature) is controlled by actuating a control valve on the steam return line to the test rig.

<u>Oil Management</u>

Oil discharge from the compressor is separated from the compressor discharge gas in oil separator units. The oil is returned to the compressor through a control valve, which is actuated to limit the return flow rate. Additional oil which manages to pass through the oil separators will make its way to the pentane separators whereby it will accumulate without proper management. Therefore a small amount liquid from the refrigerant separators is entrained intermittently into the suction gas lines, prior to the suction gas superheaters such that it can be evaporated and any residual oil returned to the compressors (feature not pictured in process flow diagram).

In addition to managing the return of oil to the compressors, the high temperatures which characterise this system necessitate external cooling of the oil. This is conducted through two heat exchangers coupled to an external cooling circuit which discards the heat to ambient through use of a dry cooler. The oil is circulated using a pump which is internally located within the compressor housing.

<u>Trace Heating</u>

Trace heating is employed at numerous locations throughout the heat pump, utilised when the heat pump is not in operation. The majority of trace heating in the system is located in the vicinity of the compressor oil system. Trace heating is located on the oil separator and the compressor housing, as well as the lines (both oil and refrigerant) between these components. In addition, trace heating is placed on the oil coolers. The purpose of this trace heating is to prevent refrigerant condensing and mixing with the oil during shut down periods. This can lead to oil foaming on start-up, limiting the oil pressure in the compressor. Trace heating was also placed on both suction gas superheaters to prevent liquid condensing in the suction line when the system is not in operation. Without this trace heating, it is probable that significant amounts of liquid refrigerant would be entrained into the compressor on start-up.

2.4. Theoretical Performance

As indicated in section 2.1, the COP of a heat pump relates the heat output in the condenser to the electrical work input as seen in equation 1. This is valid for both single evaporator and multi-evaporator cycles, in the latter case, the electrical energy from both compressors must be summed.

$$COP = \frac{\dot{Q}_{th-out}}{\dot{W}_{elec-in}} \tag{1}$$

The theoretical maximum COP that can be achieved by a heat pump cycle containing only internally reversible processes is known as the COP_{Carnot} and is defined by the second law of thermodynamics. For a heat pump with a single heat source, the COP_{Carnot} can be simply expressed as a function of the source and sink temperatures, as given in equation 2.

$$COP_{Carnot} = \frac{T_{sink}}{T_{sink} - T_{source}} \tag{2}$$

For a multi-temperature heat pump system, calculation of the COP_{Carnot} becomes non-trivial. The value is dependant not only on temperatures, but also on the fraction of heat input from each evaporator in the system. The equations which define the second law efficiency (COP_{Carnot}) for a multi-temperature heat pump have been defined by Arpagaus [4] and are reproduced in equations 3 to 7 below.

$$COP_{Carnot(LT)} = \frac{T_{sink}}{T_{sink} - T_{source(LT)}} \quad , \quad COP_{Carnot(MT)} = \frac{T_{sink}}{T_{sink} - T_{source(MT)}} \tag{3), (4}$$

$$\beta = \frac{\dot{Q}_{th-in(MT)}}{\dot{Q}_{th-in(MT)} + \dot{Q}_{th-in(LT)}} \quad , \quad \alpha = \frac{\beta}{1-\beta} \frac{COP_{Carnot(LT)} - 1}{COP_{Carnot(MT)} - 1} \tag{5), (6}$$

$$COP_{Carnot(multi-temp)} = \frac{1}{1+\alpha} COP_{Carnot(LT)} + \frac{\alpha}{1+\alpha} COP_{Carnot(MT)} \tag{7}$$

To determine the true efficiency of a heat pump, the COP achieved is related to the thermodynamic maximum through equation 8. For the current project, a η_{Carnot} of 50% was targeted for both single stage and dual stage cycles, which should be feasible for a well designed heat pump system.

$$\eta_{Carnot} = \frac{COP}{COP_{Carnot}} \tag{8}$$

2.5. Component Selection

The system utilises industrial open type piston compressors manufactured by Mayekawa (HP compressor pictured Figure 3). The specifications of these compressors are given in Table 1. Piston compressors were chosen as they match well with the system size requirements (150 kW) and can easily adjust to the differing pressure ratios required for testing. These compressors were able to operate between 900 RPM and 1600 RPM and were lubricated by a polyalkylene glycol (PAG) lubricant with an ISO viscosity grade of 100.

Table 1: Compressor specifications

Component	Type	Swept Volume (cm^3)	Displacement (m^3/hr @ 1450 RPM)
LP Compressor	Mayekawa F8WA	4310	375
HP Compressor	Mayekawa B6HK	2218	193

The heat pump is equipped with numerous sensors to measure physical quantities which combined give the performance of the cycle. The data from the unit is collected at an interval of 1 second using a Siemens SCADA system. There are pressure and temperature sensors at numerous locations, both of which are generally located in each vessel as well as between the major components. Mass (or volume) flows are not measured within the heat pump itself, rather they are calculated from data gathered from the connected test rig (see section 2.7).

Figure 3: HP Mayekawa piston compressor

2.6. Safety Systems and Measures

The heat pump system inside the container conforms with relevant ATEX requirements. The container is placed in the open air, with the internals ventilated with approximately 2000 m³/hr of fresh air. Pentane detectors in the ventilation ducts will detect leaks at 1400 PPM, roughly 10% of the lower explosive limit.

2.7. Test Rig

A process flow diagram of the test rig used to simulate the industrial process can be seen in Figure 4. The buffer vessel is the central component, with all flows originating from this location. The buffer vessel contains a water steam mixture which is heated in the range of 100°C – 120°C (1 bar_{abs} – 2 bar_{abs}) using an electrical heating element prior start-up. Liquid from this vessel is fed to the steam separator (which subsequently feeds the condenser) as well as the two evaporators. The flow rate for all three circuits are set by a speed controlled pump, whilst the temperatures of the flows to the evaporators can be controlled by recirculating the cooled flows from the exit of the evaporator back to the inlet. The return flows from both the evaporators (liquid) as well as the steam separator (vapor) are returned to the buffer vessel. The heat added to the vessel from the flow from the condenser is greater than that removed from the evaporators, in theory, by the power of the compressors. Therefore, to maintain the temperature of the buffer vessel at a fixed level, it is necessary to remove heat, which is done by condensing steam at the top of the vessel using a simple U tubed heat exchanger. A water glycol mixture is circulated through the heat exchanger with the heat discharged to ambient using a dry cooler. The test rig is equipped with numerous instrumentation (pressure, temperature, flow) to determine the thermal power from the heat pump and fully characterise the process.

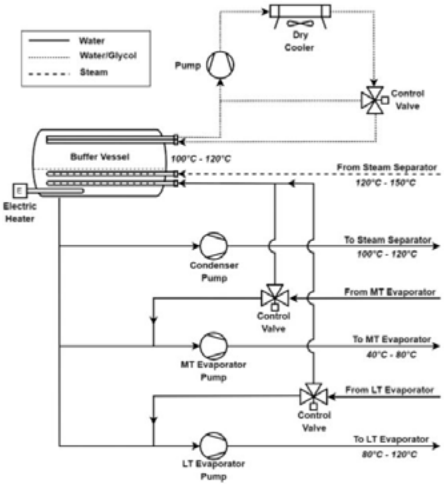

Fig. 4. Process flow diagram of test rig which simulates the industrial process on the heat pump

Figure 5 shows an image of the internals of the test rig system. The buffer vessel is clearly distinguishable in the foreground on the right side of the figure. The connections for the U-tube heat exchanger are located at the top of the vessel, whilst the return lines for the condenser and evaporators are located below. In the background on the left side of the figure, the pumps as well as general flow apparatus (valves, flow meters) are present. After this figure was taken, and prior to testing, the buffer vessel, piping and components were thermally insulated.

Fig. 5. Internals of test rig showing the buffer vessel and ancillary components

3. Experimental Results

This section outlines experimental results obtained from the unit. Firstly, two measurement results (one single stage, one dual stage) are analysed. Following this, a full list of results from the experiments is presented.

3.1. Analysis of Single Stage Results

Single stage results are presented for the heat pump producing steam at 130.6°C from a source heat temperature of 84.9°C cooled to 79.7°C. The measured cycle state points are presented on TS and PH diagrams in Figure 6 and 7 respectively. The numbered points on the diagram make reference to the location in the cycle as seen in Figure 2. Overlaid on the TS diagram is the temperature of the external source and sink to give an indication of the ΔT in the evaporator and condenser heat exchangers. It is noted that the specific entropy information is not relevant to this data.

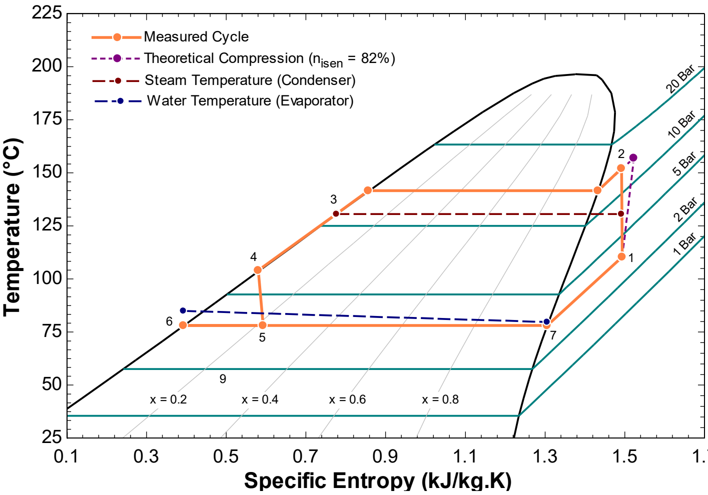

Fig. 6: TS diagram for single stage cycle (T_{steam} = 131°C, T_{source} = 85°C)

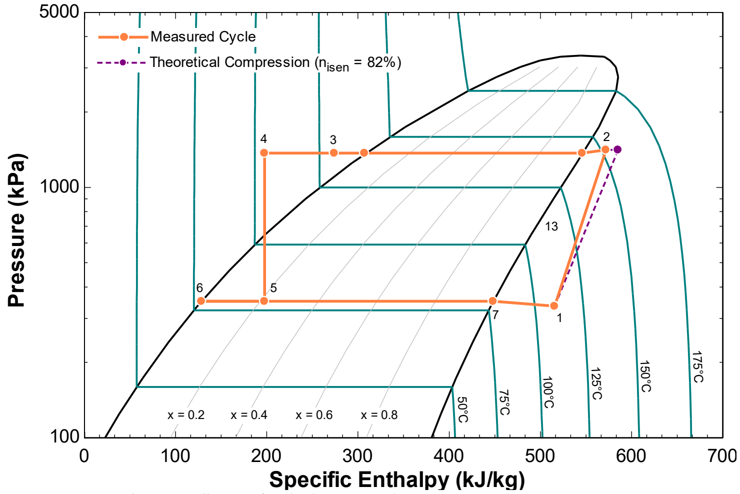

Fig. 7: PH diagram for single stage cycle (T_{steam} = 131°C, T_{source} = 85°C)

The heat pump produced 97.5 kW of heat (steam) with an electrical energy input of 36.3 kW, leading to a COP of 2.69. Based on the maximum/minimum external temperatures of 130.6°C and 79.7°C respectively, the COP_{Carnot} for these conditions is 7.92, leading to a value of η_{Carnot} of 34%. The low efficiency of the system is attributed to three (3) reasons described below with the assistance of the TS and PH diagrams, as well as Figure 8, which presents an energy balance for the cycle.

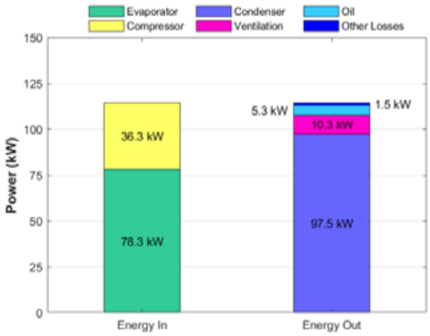

Fig. 8: Energy balance for single stage cycle (T_{steam} = 131°C, T_{source} = 85°C)

1. *Compressor and Electric Motor*

As stated, the measured electrical power to the compressor was 36.3 kW. At the given conditions the compressors should be expected to operate with an isentropic efficiency of 82%. Based on the calculated mass flow for the cycle, the power input is expected to be in the range of 22.2 kW, which obviously varies greatly (10.1 kW) from the measured value. The reason for the large electrical power input is difficult to determine due to the limited sensors which may give a conclusive answer. Likely it is a combination of a lower than expected isentropic efficiency of the compressor, combined with a low electrical motor efficiency, estimated to be in the range of 60%.

2. *Large Temperature Difference in Condenser*

For this experimental condition, the measured condenser temperature was 141.6°C (13.7 bar) whilst the measured evaporator temperature was 78.1°C (3.5 bar). Whilst a small ΔT was measured throughout the evaporator (1.6°C-6.8°C), this was not the case in the condenser (~11°C). The large temperature differences act to increase the work required by the compressor, therefore reducing the total efficiency of the machine.
The large mean temperature difference which acts to reduce the efficiency of the heat pump, has been attributed to air ingress occurring whilst the system is shut down and below atmospheric pressure. This air trapped in the condenser section could not be easily purged.

Thermal Losses

A large amount of heat produced in the system was lost to oil cooling (5.3 kW) as well as the ventilation air (10.3 kW). These thermal losses are visible through inspection of the theoretical compression process overlaid on the measured cycles in Figure 6 and 7. Whilst it would be expected that the theoretical process brings the compressor discharge state closer to the two-phase region than the measured cycle, this is not the case due to the thermal losses which are present. This result emphasizes the case for proper thermal insulation of the heat pump unit and correct selection of the oil temperature when operating at high temperatures. It is noted that a small portion (1.5 kW, 1.3%) of the total heat input could not be accounted for in the energy balance.

3.2. Analysis of Dual Stage Results

The results of the dual stage cycle implementation are presented for the heat pump producing steam at 150.4°C from two heat sources; one cooled from 105.1°C to 102.0°C, and the other from 64.9° to 60.1°C. Once more, the measured cycle state points are presented on TS (source and sink temperatures overlaid) and PH diagrams, seen in Figure 9 and 10 respectively.

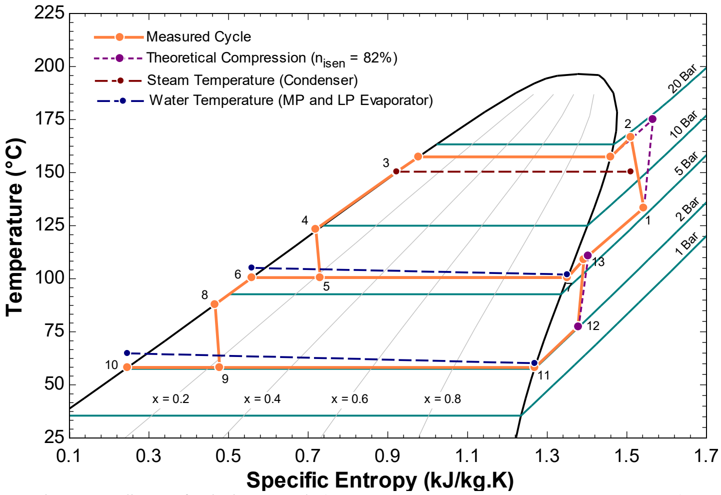

Fig. 9: TS diagram for dual stage cycle (T_{steam} = 150°C, $T_{source(MT)}$ = 105°C, $T_{source(LT)}$ = 65°C)

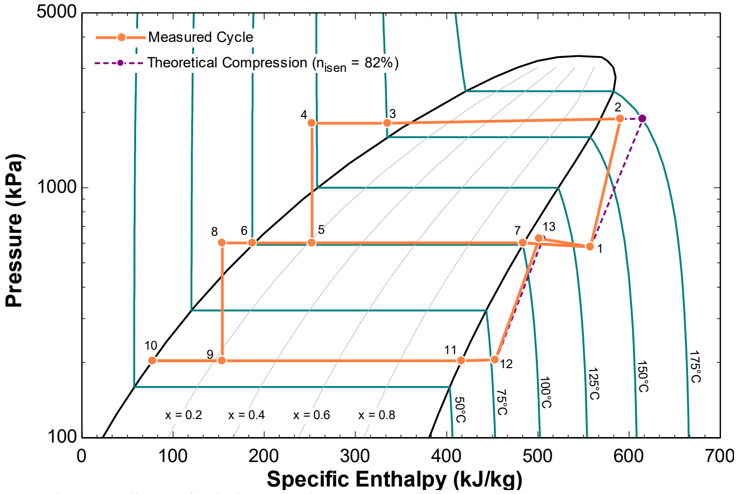

Fig 10: PH diagram for dual stage cycle (T_{steam} = 150°C, $T_{source(MT)}$ = 105°C, $T_{source(LT)}$ = 65°C)

The heat pump produced 144 kW of heat (steam) with an electrical energy input of 63 kW, leading to a COP of 2.06. Utilizing equation (3) through (7), the COP_{Carnot} was calculated to be 5.63 for the dual stage cycle, leading to a η_{Carnot} of 36.5%. Whilst still relatively low, the dual stage cycle performs relatively better than the single stage cycle.

Insights can be gained into the performance of the cycle from the supplied TS and PH diagrams (Figure 9 and 10) as well as the energy balance for the system provided in Figure 11. The theoretical compressor work required for this cycle was 31.2 kW (59% of measured) and 14.3 kW (83% of measured) for the HP and LP compressors respectively. This gives evidence to suggest that the combination of compressor and electrical motor performs better in terms of isentropic and electrical efficiency in the low stage compared to the high stage. Further to this, from the TS and PH diagrams, when viewing the theoretical compression process, it can be seen that for the low stage that the theoretical and actual lines are almost overlapping. This is not the case for the high stage. This indicates that the thermal losses on the low stage are significantly lower than that of the high stage. This could be expected on the basis that the low stage operates at much lower temperature levels than the high stage. These factors, particularly the higher electrical efficiency of the motor, are likely the reason for the slightly higher η_{Carnot} when compared to the single stage cycle. Like the single stage cycle though, thermal losses from the system are still significant, with 12.1 kW lost to the ventilation, 14.0 kW lost to the oil system, and 18 kW being unaccounted for. Similarly, large temperature differences were seen over the condenser, which also increases the internal temperature lift of the heat pump and acts to reduce the COP of the system.

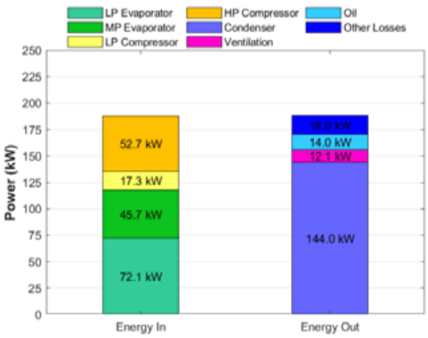

Fig 11: Energy balance for dual stage cycle (T_{steam} = 150°C, $T_{source(MT)}$ = 105°C, $T_{source(LT)}$ = 65°C)

3.3. Summary of Additional Results

A summary of the measurement results from the pilot scale heat pump unit is given in table 2. As was the case with the reference measurements, the system was characterised by low values of η_{Carnot}. Once more, in general, two-stage cycles performed better relative to the single stage cycle.

Table 2: Summary of additional measurement results from the heat pump

LT Evaporator		LP Compressor	MT Evaporator		HP Compressor	Steam Production		Heat Losses		COP	η_{Carnot}
T (°C)	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	T (°C)	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	T (°C)	$\dot{Q}$ (kW)	Oil (kW)	Air (kW)	(-)	%
60.0	95.8	21.5	100.0	35.0	46.8	133.6	166	13.8	11.3	2.4	38
64.9	72.1	17.3	105.0	45.7	52.7	150.4	144	14.0	12.1	2.1	37
-		-	94.3	91.5	41.6	140.9	110	8.9	13.0	2.6	33
-	-	-	105.0	109.0	49.0	150.5	132	11.2	12.1	2.7	32
-	-	-	85.0	78.3	36.3	130.6	98	5.3	10.3	2.7	34

4. Conclusions and Future Works

This paper gave an overview of the design and experimental analysis of a pilot scale industrial heat pump, designed with a thermal capacity of 150 kW_{out}, able to produce medium pressure steam of 150°C (4.8 bar_{abs}). The heat pump was able to take up heat at two distinct temperature levels through the use of separate flooded evaporators. Pentane was used as a working fluid, allowing the range of evaporating and condensing temperatures (52°C – 150°C) required by end-users involved the project to be covered with a single working fluid.

To perform an experimental analysis of the heat pump, it was connected to a specially designed test rig, which contained the necessary instrumentation to determine thermal powers and characterise the process. Results were presented for the heat pump operating with both a single heat input as well as a dual heat input at two distinct temperature levels.

The heat pump was in general characterised by low values of η_{Carnot}, in the range of 32-38%. The low efficiencies were attributed to three factors: combined low electrical efficiency of the electrical motor and isentropic efficiency of the compressor; large temperature differences between fluid streams in the condenser and higher than expected thermal losses. In general, the two-stage cycle showed higher values of η_{Carnot} than the single stage cycle, which was demonstrated to be a result of the lower thermal losses in the low (temperature) stage, combined with a compressor and electrical motor performance which better matched to theoretical values. With the absence of the large temperature differences in the condenser, and use of electrical motors with efficiencies in the range of 90%, it is feasible that values of η_{Carnot} exceeding 50% would be feasible.

Despite the lower than expected efficiencies, the results of the experimental analysis demonstrate that it is indeed possible to build an industrial steam producing heat pump utilising equipment currently available in the market. Using this standard equipment, the heat pump demonstrated a maximum steam temperature of 150°C, corresponding to a saturation pressure of 4.8 $bar_{(abs)}$. The condensing temperature in this case exceeded 157°C suggesting higher steam temperatures should be feasible with a better functioning condenser. Another notable result was a maximum external temperature lift of 90°C, achieved using the dual stage cycle with heat input at an intermediate temperature level.

Further work on this topic will involve attempting to gain clarity on the lower than expected performance of the system, particularly determining the exact reason for the higher than expected electrical energy input for the high pressure compressor. Additionally, the unit will be tested with a synthetic working medium, namely, the non-flammable synthetic R1336mzz(Z). The experiments conducted will focus on other process conditions relevant for end-users not yet tested, as well as characterising the heat pump using a single evaporator combined with two-stage compression. Follow-up projects on compression heat pumps will focus on upscaling the technology to full-scale sizes (1-5 MW) relevant for industrial end-users. The focus will be on achieving heat pump units with low specific investment costs (€200/kW_{th}), giving a higher likelihood of mass uptake in the market.

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6. References

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