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DECARBONISATION THROUGH INNOVATION

Multi-stage rotary liquid-piston compression for high-temperature heat pumps

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Abstract

A crucial and often overlooked source of emissions is industrial production, especially the generation of process heat. This heat is used for many critical processes such as drying, melting, and chemical transformations. Electrification of this heat supply, powered by renewables, can be a promising means to decarbonize the supply chain. However, for it to be cost-competitive, the heat provided needs to be efficiently delivered. To this end, we can use heat pumps, which use one unit of energy to deliver multiple units of heat.

Unfortunately, typical vapour compression heat pump designs have several limitations that prevent them from functioning at high temperatures. For such designs, typical compressors cannot deliver the required high pressure ratios, and their lubricant breaks down above certain temperatures.

This work presents a new heat pump design, which avoids using a compressor: instead, a conventional pump is used to pressurize a liquid, which then exchanges its pressure with the working fluid by means of a rotary liquid piston device. This process is repeated over multiple stages to achieve a high isentropic compression efficiency. As the active mechanical elements (e.g. impeller, vanes) are isolated from the suction fluid, there is potential to both achieve higher temperatures and design cycles that rely on wet compression.

This work details the overall architecture of the cycles and explains its working principle. Thermodynamic correlations are used to calculate the performance of the pressure exchanger. Finally, the coefficient of performance (COP) of the heat pump is estimated and design considerations are explored.

Keywords: Heat Pump; High temperature; Rotary Pressure Exchanger; Energy Efficiency; Waste Heat Recovery

1. Introduction

While there are many contributors to greenhouse gas production and associated climate change, decarbonization of domestic consumption receives the lion's share of attention. Therefore, public attention largely revolves around solutions like more efficient domestic appliances and electric cars. Yet a large section of our total energy usage is actually in the realm of production [1]. Of this, as much as 70% of industrial energy demand is in the form of process heat. [2] This heat powers processes as varied in nature and temperature as the pasteurization of milk, the drying of fresh paper pulp, or the smelting of iron ore in a blast furnace. Most of these processes are still powered by fossil fuels. According to Thiel and Stark [3], process heating alone accounts for 20% of global emissions.

The electrification of these heating processes via a grid powered by renewable energy sources like solar and wind power is therefore an attractive proposition. Many resistive, inductive and microwave heating technologies are available to convert electricity to heat at temperatures as high as 3000°C. The primary barrier to electrification, however, remains cost. In large parts of the world, heat derived from fossil fuels is still several times cheaper than that from renewables-based electricity [3]. Direct conversion of electricity to heat is therefore cost prohibitive. Additionally, process heating typically demands a constant supply of energy, while renewables-based electricity is highly variable in supply by nature. This translates to highly variable

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energy costs, which can complicate the financial case for a factory. Lastly, electrification requires the already strained electric grid infrastructure to be massively scaled up: having been built only for residential use, it will now also have to supply electricity for process heating. Together, these factors serve to make direct renewables-based electric heating of industrial processes infeasible.

To make electrification possible then, it is necessary to reduce the electricity demand of process heating. The easiest way to achieve this is by reducing the wastage of heat. Globally, roughly half of all energy generated becomes waste heat, which means there is significant potential to reduce energy demand [4]. A third of this heat is below 200°C, also referred to as low-grade waste heat. Most industries cannot utilize this resource as-is, as they operate above this temperature [5]. Theoretically, this waste heat could be 'upgraded' to a higher temperature by means of a heat pump.

Unfortunately, commercial availability of heat pumps above 100°C is still quite limited. Zühlsdorf et al. [6] have conducted a comprehensive review on the commercial availability of such high temperature heat pumps (HTHPs). While they found increasing industrial adoption of HTHPs below 160°C, most options above this temperature are still in the demonstration stage. Additionally, the costs for the same are both higher and more uncertain than ones below 160°C. The biggest obstacle to realising HTHPs remains the compressor. In another review of the HTHP landscape, Adamson et al. [7] cite six main challenges to development, three of which concern the compressor. These are the extreme pressures and compression ratios demanded of the compressor, and extreme temperatures its lubricant must effectively operate at.

To increase the output temperature of an HTHP design will require supplementing the compressor or supplanting it entirely. This paper proposes to achieve this by means of a so called pressure exchanger (PX). This rotary device expands a high pressure (motive) fluid to compress a low pressure (secondary) fluid, thus 'exchanging' pressure. Illustrated in fig. 1, it consists of two stators with ports that regulate the inflow and outflow of the fluids, and a rotor sandwiched between them with channels cut into it where the two fluids interact. This process consists of two stages: a scavenging stage, where fresh secondary fluid drives out the exhausted motive fluid; and a pressurization stage where the fresh motive fluid enters the PX and pressurizes the secondary fluid. The PX has previously been employed for energy recovery in a refrigeration system by Saeed et al. with both thermodynamic [8] and experimental [9] analysis, and proposed for a heat pump by Thatte [10].

In the present work however, we use it to replace the compressor, thus enabling higher temperatures. To accomplish this, we propose a novel cycle illustrated in see fig. 2. In this cycle, we first pump cold liquid refrigerant to the desired pressure (from point 5 to 6) in a separate loop. We then exchange its pressure in the PX with the working fluid of the HTHP, which leaves the motive liquid at point 7 and transfers the working fluid from point 1 to point 2. The rest of the cycle works as a normal heat pump with the working fluid arriving at point 3 after releasing heat to the sink, point 4 after the expansion valve, and back to point 1 after absorbing heat from the source. In the compression apparatus for this cycle, the pump works with a motive fluid at low temperature, but indirectly compresses the working fluid to a higher temperature. Additionally, since the PX is self-lubricated, there is no contamination of the working fluid. Together these factors serve to raise the sink temperatures that can be achieved with a heat pump.

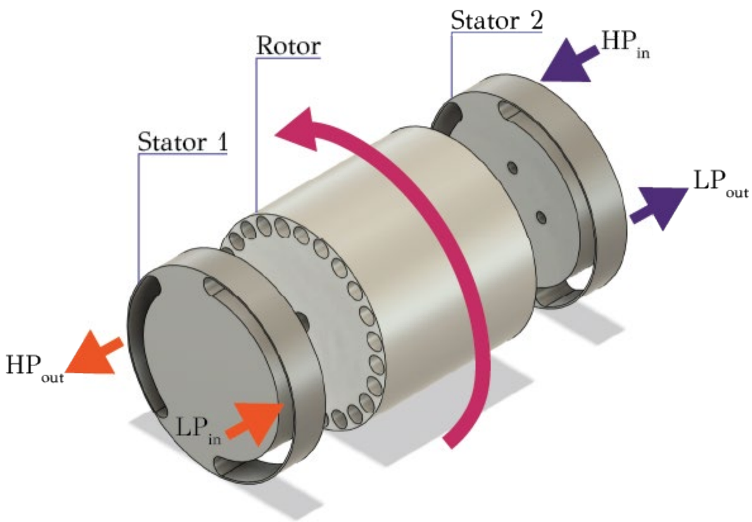
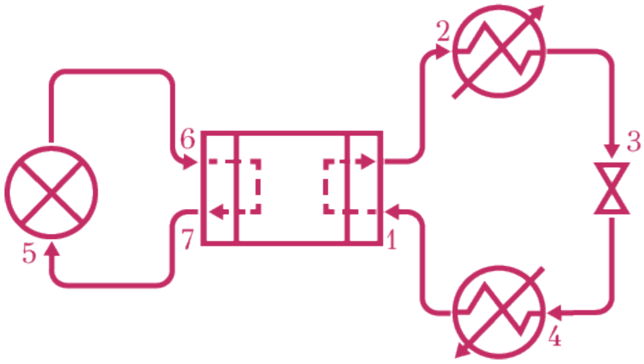


Figure 1. Design and operation of a PX. On the right, Motive fluid enters at high pressure (HP_{in}) and leaves at low pressure (LP_{out}). On the left, secondary fluid enters at low pressure (LP_{in}) and leaves at high pressure (LP_{out}).



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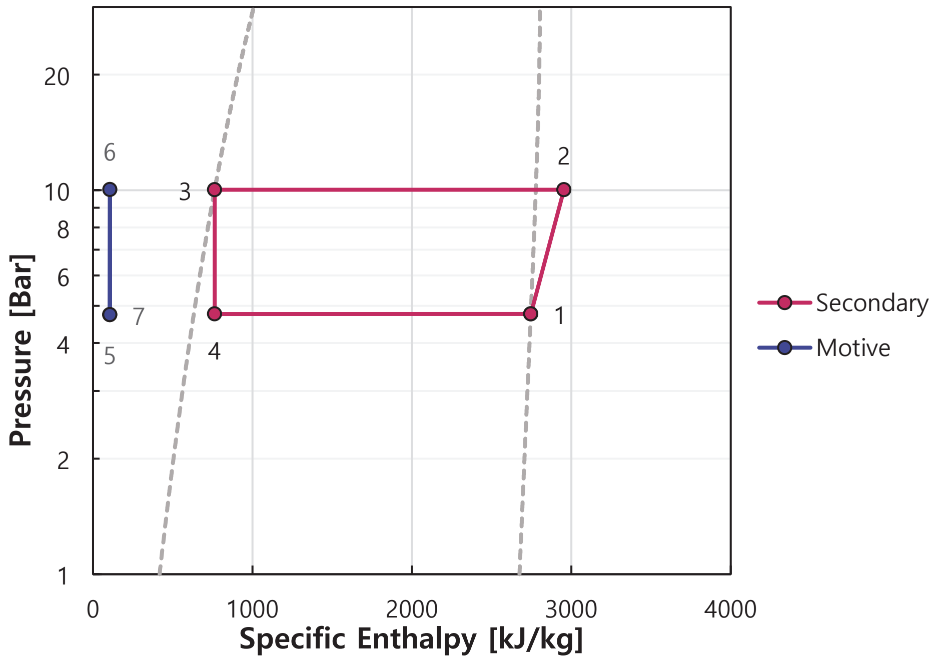

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Figure 2. (a) Layout and (b) *P-h* diagram of basic liquid-piston compression heat pump.

The objective of this work is to propose an HTHP setup designed around the PX and examine its feasibility. To this end, we will describe the liquid-piston compression inside the PX with a simple model, estimate its efficiency, and design a cycle to maximize the performance of an HTHP based on this compression process.

2. Approach

2.1. Heat pump model

To evaluate the theoretical benefits of adopting the proposed HTHP configuration, a numerical model has been developed in MATLAB.

In modelling the liquid-piston compression several assumptions are made. Feng et al. [11] and Shu et al. [12] assume 80% pump isentropic efficiency in organic Rankine cycle applications that deal with high pressure lifts. Therefore, we too assume 80% efficiency for the pump throughout this work. Second, the expansion of motive liquid in the PX is assumed to be isentropic. Irreversibility, which causes entropy losses, is typically a result of the rapid nature of the compression/expansion process, which causes zones of differing properties and various discontinuities. Since the volume of the liquid changes negligibly during the expansion process, we can simply neglect such effects and assume a reversible expansion. Lastly, like Thatte [10], we assume that the walls of the channels in the PX rotor are adiabatic, and any heat transfer or mixing between the motive and secondary fluid is negligible. With all the above sources of loss accounted for, all energy lost by the motive fluid is assumed to be recovered completely by the secondary fluid.

Following the approach of Saeed et al, [8] we then model the compression of the working fluid in the PX using energy conservation. With reference to fig. 2:

$$\dot{m}_1(h_2 - h_1) = \dot{m}_6(h_6 - h_7) \tag{1}$$

$$\rho_1\dot{V}_1(h_2 - h_1) = \rho_6\dot{V}_6(h_6 - h_7) \tag{2}$$

Since the volumetric flow rates of motive and secondary fluid are equal, eq. (2) can be rewritten like so:

$$h_2 = h_1 + (h_6 - h_7)\frac{\rho_6}{\rho_1} \tag{3}$$

The rest of the cycle is modelled like a typical subcritical heat pump. We therefore neglect the pressure drop in the source and sink heat exchangers and assume that the working fluid undergoes isenthalpic expansion in the expansion valve.

The efficiency of the compression process is calculated as in a typical heat pump, by comparing the enthalpy change with a hypothetical isentropic compression process:

$$\eta_C = \frac{h_{2,is} - h_1}{h_2 - h_1} \tag{4}$$

Since the expansion of motive fluid is assumed isentropic, this is the only source of loss in the PX. We can calculate the COP of such a heat pump by dividing the sink output by the energy consumption of the pump per unit mass of working fluid compressed:

$$COP = \frac{(h_2 - h_3)}{(h_6 - h_5)}\frac{\rho_1}{\rho_6} \tag{5}$$

2.2. Performance of the compression process

Since the ambition with this technology is to enable high temperatures, we chose steam as a working fluid, owing to its high critical temperature, its status as natural refrigerant, and high familiarity in industry due to centuries of use in power production. With this, two parametric analyses studies were performed to study the effect on the heat pump performance.

First, we varied the temperature lift while keeping the source temperature constant at 100 °C, (fig. 3). Second, we varied the source temperature while keeping the temperature lift constant at 50 °C, (fig. 4). The first analysis demonstrates that the configuration shown in fig. 2 cannot deliver significant temperature lifts without suffering from high compression efficiency losses. In fact, given the irreversible nature of the compression/expansion process occurring in the PX, the higher the required pressure ratio, the lower the compression efficiency. The second analysis shows that the compression efficiency increases with the source temperature, in this case from 54.5% at 120 °C to 67% at 200 °C. Put together, these two findings make the direct application of this technology questionable.

We can better understand the trends illustrated above by substituting eq. (3) into eq. (4) like so:

$$\eta_C = \frac{(h_{2,is} - h_1)}{(h_6 - h_7)}\frac{\rho_1}{\rho_6} \tag{6}$$

From this equation, we observe that η_C is related directly to ρ_1/ρ_6, the ratio of the densities of the two fluids upon entering the PX. This explains the increasing efficiency in fig. 4, as the density of saturated vapor is higher at higher temperatures and thus increases the ratio of densities. However, it can be demonstrated that since $h = u + P/\rho$, as the temperature lift of a stage decreases the ratio of enthalpy differences in eq. (6) will tend towards a value equal to ρ_6/ρ_1, thus causing the efficiency to tend towards 100%. This explains the trend seen in fig. 3.

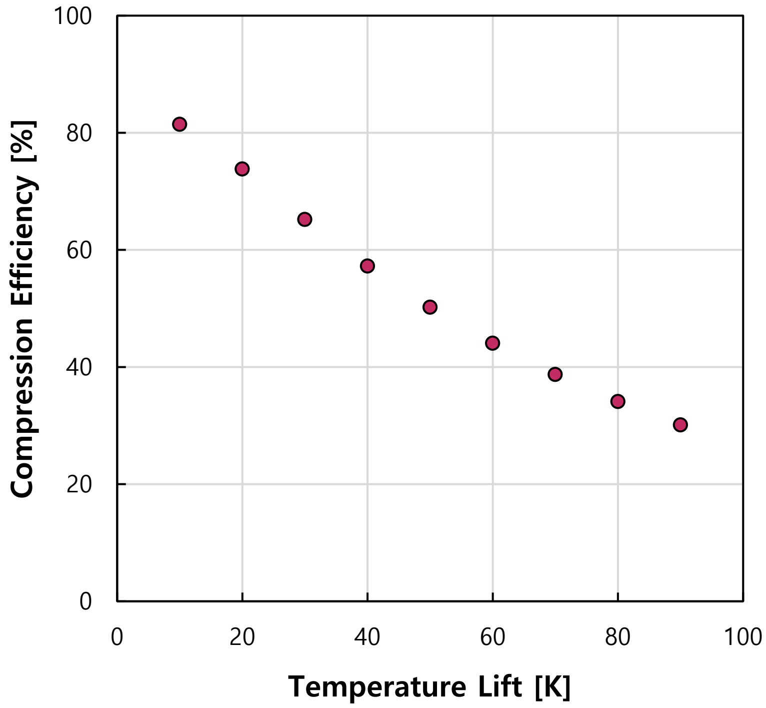


Figure 3. Variation of compression efficiency with temperature lift, keeping the source temperature constant at 100°C.

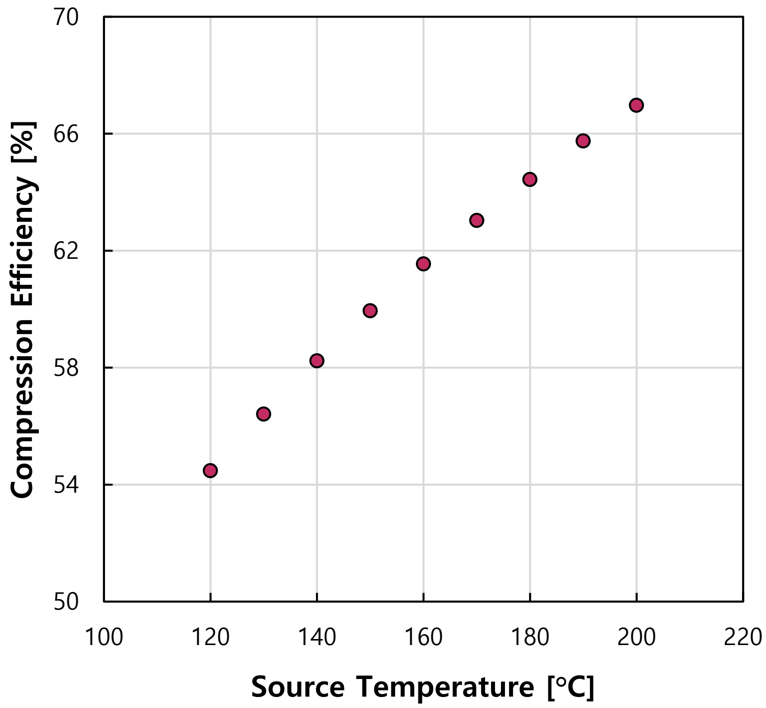
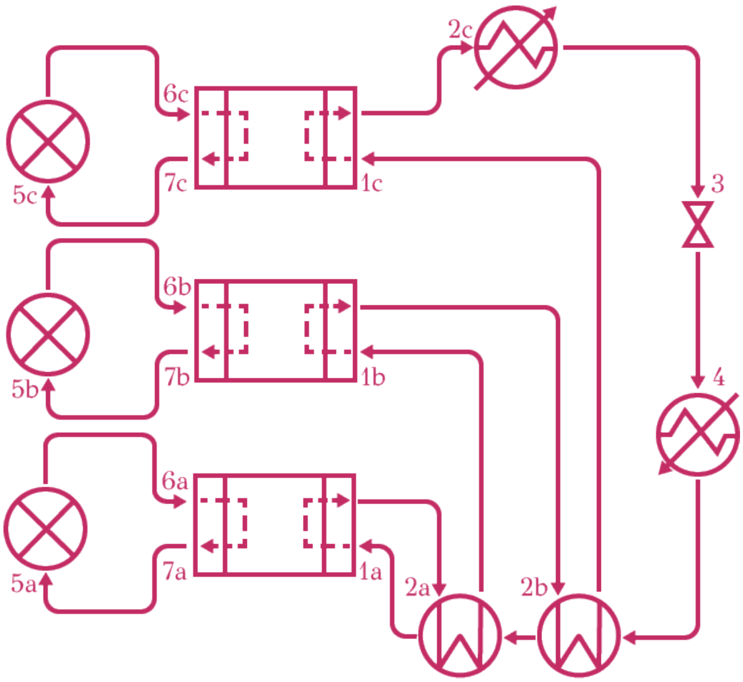


Figure 4. Variation of compression efficiency with source temperature, keeping the temperature lift constant at 50°C.

2.3. Multi-stage design

To overcome the abovementioned limitations, we propose a multi-stage design, depicted in fig, 5 (three stages). As can be observed, such a design involves splitting the compression work over multiple stages separated by intercoolers.



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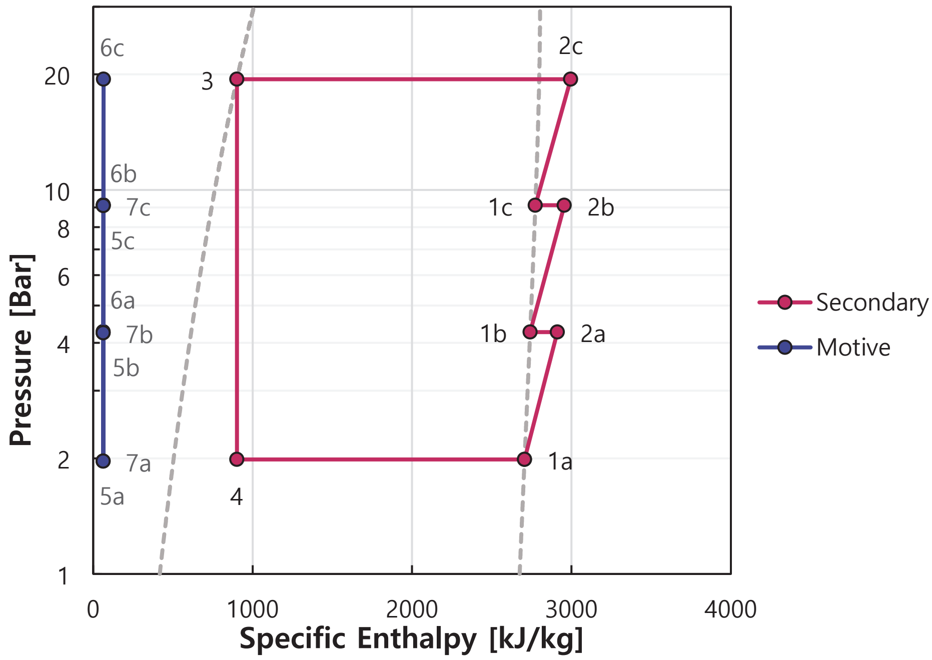

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Figure 5. (a) Layout and (b) *P-h* diagram of three-stage rotary liquid-piston compression heat pump.

Following the logic of eq. (6), by limiting the temperature lift of each individual stage we can increase the total compression efficiency. The COP of the cycle can be calculated by dividing the energy obtained at sink by the total energy use for each pumping stage in the cycle:

$$COP = \frac{(h_{2,n} - h_{3,n})\, \dot{m}_S}{\sum_{i=1}^{n} (h_{6,i} - h_{5,i})\, \dot{m}_{M,i}} \tag{7}$$

In the following section, we will explore the application of this concept to high temperature waste-heat recovery in industry.

3. Results and Discussion

To find a relevant application, we relied on the review of Marina et al. [13], which lists several relevant opportunities for saving energy by upgrading waste heat using HTHPs. An oxo-alcohol manufacturing process with very high temperature waste heat was identified as a prime use case for the subcritical heat pump discussed in this work. The parameters for this process are listed in table 1. As can be seen from the table, the waste heat availability of the process is nearly half as much as the process heat demand. Therefore, there is significant potential to reduce energy consumption in the manufacturing process. In this section, we will design an HTHP for this process with a source temperature of 120°C and a sink temperature of 210°C, corresponding to saturation pressures of 1.99 bar and 19.08 bar respectively.

Table 1. Process parameters of oxo-alcohol synthesis across the European Union. [13]

Parameter	Value	Unit
Process Heat Requirement	570	TJ/yr
Process Heat Temperature	203	°C
Process Heat Medium	Steam	
Waste Heat Availability	248	TJ/yr
Waste Heat Temperature	126-125	°C
Waste Heat Medium	Condensate	

Following the trends analysed in section 2.2, it is easy to see that a higher number of stages enables us to reduce the pressure lift per stage, improving the compression efficiency. However, given the number of stages, we cannot know a priori the ideal pressure lift for each stage. Therefore, in this work we perform a parametric analysis to find the optimum distribution of pressure lift across the stages to maximize the COP. To ascertain the variation of COP, the difference between the sink and source pressures is divided into multiple intermediate points. Each of the stages is then assigned to a point such that it is higher than the preceding stage but lower than the sink pressure. In this manner, all possible combinations of stages are tested. The results for a two and three-stage setup have been plotted in fig. 6 and fig. 7 respectively. In both cases, there is an optimum distribution of lift that provides the greatest COP, where the initial stages of the cycle provide significantly lower lift than the final stages. Fig. 8 shows the maximum COP that can be achieved with the proposed HTHP for up to 6 stages. It can be observed that the cycle benefits from the increase in the number of stages, but with diminishing returns with respect to achievable COP. Therefore, the best configuration would ultimately be determined via practical considerations.

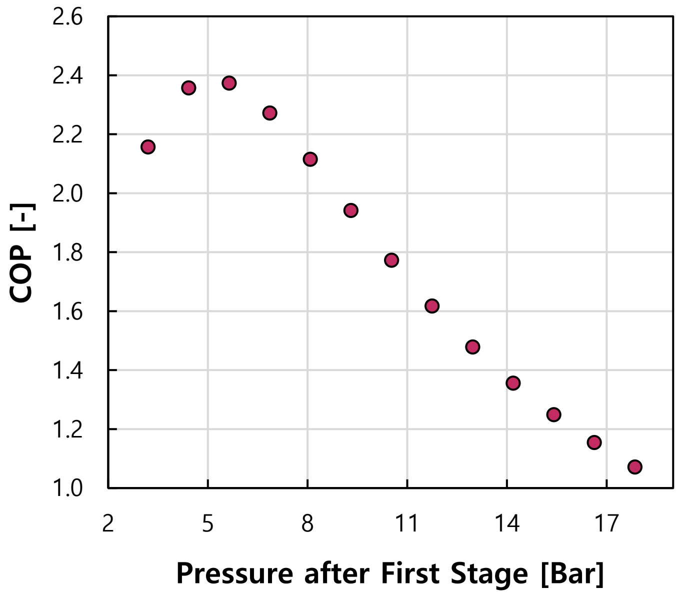


Figure 6. Variation of COP with P_{2a} for a two-stage rotary liquid-piston compression heat pump.

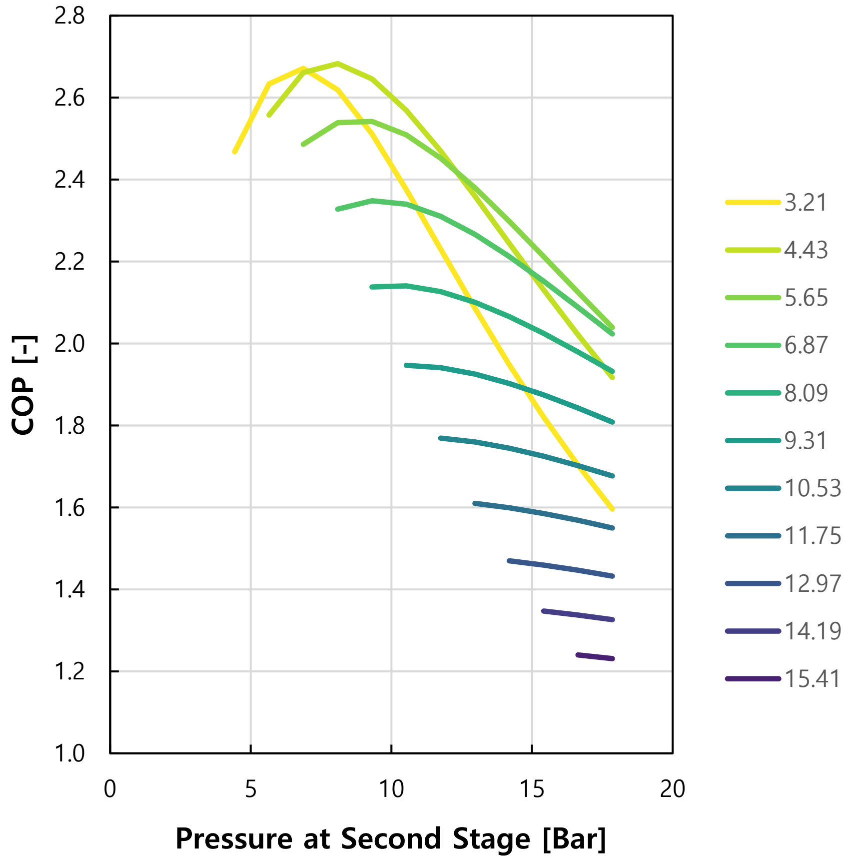


Figure 7. Variation of COP with P_{2b} for various values of P_{2a} for a three-stage rotary liquid-piston compression heat pump.

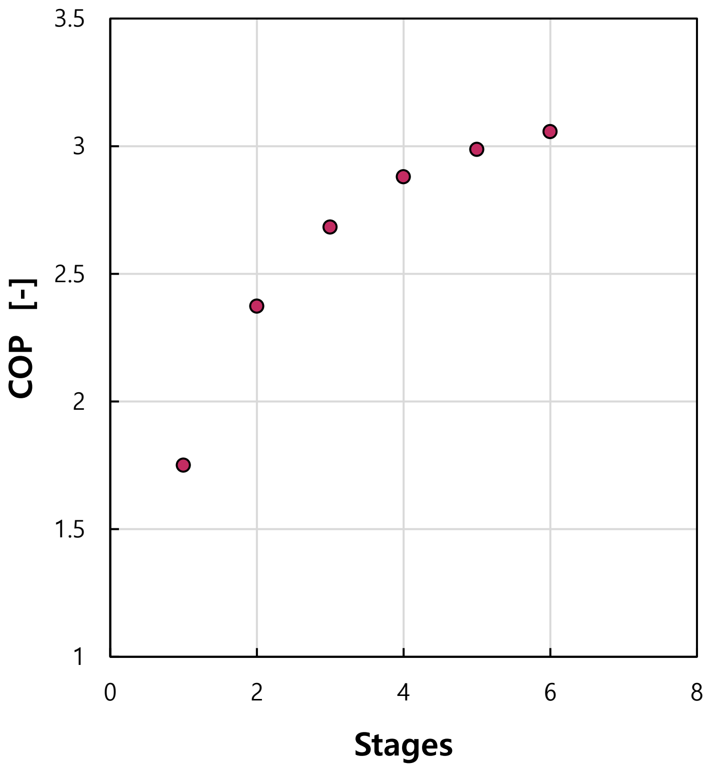


Figure 8. Variation of COP for the multistage heat pump with n.

4. Conclusions

In this work we modelled successfully the liquid-piston compression heat pump using simple relations and estimated its performance. We then applied it to a real-world process and devised a multi-stage setup that can achieve the required lifts and high temperatures. The following conclusions can be drawn from this work:

- The efficiency of the single-stage liquid-piston compression process declines by 51.3% upon increasing in the temperature lift of the heat pump stage from 10°C to 90°C.
- The efficiency of the single-stage liquid-piston compression process increases by 12.5% upon increasing the source temperature provided to the heat pump stage from 120°C to 200°C.
- A single-stage liquid-piston compression heat pump is unable to provide the lift required for expected HTHP applications with satisfactory performance, which necessitates the use of multiple stages.
- The COP of a multi-stage heat pump increases with the addition of stages, however with diminishing returns.
- There is an optimum distribution of pressure lift across the stages of a multi-stage heat pump. In this distribution, earlier stages lift the pressure of the working fluid by a smaller amount than the later stages.

The multistage heat pump cycle outlined in this work has the potential to overcome the limitations of current HTHP technology and make it available for higher output temperatures. There are several possibilities for future research, such as optimizing the design of the PX, comparing it to other pressure recovery devices, testing different refrigerants, and testing the HTHP with wet compression, which is theoretically possible with such a cycle.

5. Nomenclature

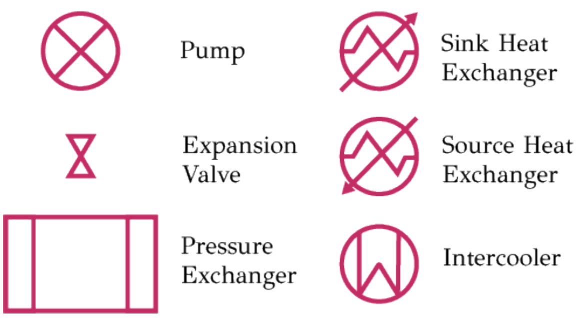
5.1. Variables

h	Specific Enthalpy [J/kg]
i	Particular Stage Number
m	Mass [kg]
n	Total Number of Stages
V	Volume [m^3]
P	Pressure [Bar]
u	Specific Internal Energy [J/kg]
η	Efficiency [%]
ρ	Density [kg/m^3]

5.2. Subscripts

C	Compression
is	Isentropic
M	Motive
S	Secondary

5.3. Key



Acknowledgements

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