14th IEA Heat Pump Conference
15-18 May 2023, Chicago, Illinois

Thermodynamic analysis of the cascade economization cycle for high temperature heat pump applications

Elias N. Pergantis[a, *], Abd Alrhman M. Bani Issa[a], John K. Brehm, Andreas J. Hoess[a], Eckhard A. Groll[a], Davide Ziviani[a]

[a]Ray W. Herrick Laboratories, School of Mechanical Engineering, Purdue University West Lafayette, 47907-2099, USA

Abstract

High-temperature heat pumps (HTHP), supply temperatures > 100°C, have received significant traction in recent years due to the high total energy usage by processing industries and the phasing down of hydrocarbons. However, despite their potential to significantly reduce greenhouse gas emissions, electrically driven HTHPs suffer considerable performance losses at temperature lifts greater than 60 °C. However, multiple industrial sectors require supply temperature > 120°C while at the same time high temperature high-capacity waste heat sources are often not readily available. Currently, the most widely used cycle for HTHPs with a large temperature lift is the cascade arrangement with different refrigerants on the high side and low side. In this work, we propose staging either the high-side or the low-side compressor, and as such, we investigated different types of economization through a thermodynamic simulation in EES. The refrigerants selected were R-32 and R-1233zd(E). It was found that by using open economization on the high side and by optimizing the cascade temperature, performance enhancements in the order of 25% are achievable over the baseline cascade cycle.

Keywords: High Temperature Heat Pumps; Cascade; GWP; Decarbonization

1. Introduction

Decarbonization policies, government initiatives (e.g., carbon taxes), and many other measures have resulted in rapid replacement of fossil fuels in applications where heating can be effectively electrified by means of heat pumps. Although the debate is mostly focused on buildings, industrial processes require vast amounts of heat, primarily supplemented by fossil fuel sources. In the work of Fox et al. (2021), it was estimated that 3416 PJ of heat are required in the U.S. for process heat, with roughly 50% used for supply temperatures between 120 to 160 °C, 30% for 160 to 200 °C and the rest for supply temperatures less than 120 °C [1,2]. Although trends from Germany and France indicate lower usage of process heat above 140 °C, it is immediately apparent that a vast potential of decarbonization potential exists in switching these processes from gas/oil burners to heat pumps. These heat pumps, labeled as High Temperature Heat Pumps (HTHPs), are usually characterized by heat sink temperatures greater than 100 °C and can be further categorized as a very high temperature heat pumps (VHTHP). However, since the specific limits are not clearly defined, this differentiation will not be used in this work. Additionally, this work will focus on closed systems, compression heat pumps, although open systems (mechanical vapor recompression, thermal vapor recompression) and closed (sorption) have been investigated elsewhere [3,4,5].

HTHPs in the recent scientific literature have received significant academic traction, with multiple review papers [2,6,7], work on refrigerant selection [8,9], advanced thermodynamic cycles [10,11], experimental pilot studies [12,13] and technoeconomic analyses [14]. This interest has not only been restricted to academia, but

* Corresponding author. Tel.: +1-734-263-4421.
E-mail address: epergant@purdue.edu

multiple companies have also come up with industrial models, some of which are shown in Table 1, a more comprehensive list is provided in [2].

Table 1. Some industrial HTHP models available in the market.

	Model 1	Model 2	Model 3	Model 4
Name	SGH 120/165	HEM-HR90	Ochsner IWDSS R2R3B	Viking HeatBooster S4
Company	Kobelco	Kobelco	Ochsner	Viking Heat Engines
Capacity [kW]	380/660	160	400	28/188
Operating Evaporator [°C]	25 - 65	-10 - 40	8 - 45	60 - 100
Operating Condenser [°C]	120 - 165	65 - 90	70 – 130	110 - 150
Type of Cycle	Single Stage	Single Stage	Cascade	Single Stage
Refrigerants	R245fa	R-134a/R245fa	R245fa(H)/R-134a(L)[†]	R-1336mzz(Z)
GWP Category	High	High	High	Low
Compressor Type	Twin Screw	Twin Screw	Screw	Piston
Heat Source Type	Water	Air	Air/Water	Water
Purpose	Steam Delivery	Process Heat	Process Heat	Process Heat

The widespread adoption of this technology is, however, restricted by several challenges, including the following:

- **Compressor design and oil management:** The large capacities of industrial applications, the high superheat temperature and material stresses are significant challenges to compressor designs and their longevity. Additionally, synthetic lubricants that can withstand such high temperatures and operate with the selected refrigerants need to be identified.
- **Refrigerant selection:** For most of the available high temperature heat pump applications, refrigerant R245fa is used. This essentially has to do with the familiarity of the manufacturers with this refrigerant due to its extensive use in waste heat recovery Rankine cycles. However, this refrigerant has a high GWP, and due to the large refrigerant charge of these systems, its usage can be detrimental to the environment. Therefore, much research is focused on finding low GWP refrigerants, with non-flammability also being a vital attribute [15].
- **Coefficient of Performance (COP):** The process heat industry requires massive sums of heat to be supplied. The transition to electricity would result in significant loads to the distribution grid. Additionally, the lower price of gas over kWh in most of the world gives a considerable incentive for the industry to continue using fossil fuels. Therefore, thermodynamic cycles with high COP are required to meet this challenge and incentivize HTHP adoption.
- **Cost:** Currently, HTHP presents a more costly solution over gas burners, although this difference can decrease with market maturity, economies of volume, higher gas pricing, and tax incentives.

The aim of this work is to address some of the challenges with respect to the COP decrease as the lift temperature (ΔT_{lift}), defined as the difference between the sink and the source temperature, increases. In the recent works of Kosmadakis et al. (2020) and Mateu-Royo et al (2021), it was identified that for low temperature lifts (50 K or below), single-stage with economization and internal heat exchanger (IHX) performed best, while at higher lifts (60 K or above), two-stage cascade with different refrigerants on the high side and the low side selected appropriately performs better [10,14]. However, in many cases, a far greater ΔT_{lift} is necessary, in the order of 80 to 100 K, with some applications requiring an even greater temperature lift. An example of this could be a pulp factory in the U.S., receiving waste heat at 40 °C and needing to provide heat for its processes in the 120 to 140 °C range. Although higher source temperatures would be preferred, that is not readily available in many cases. Another example is that of air source systems, such as the HEM-HR90 (see Table 1), which are subject to atmospheric variations even at lower sink temperatures. Through a literature review, it was identified that limited research has been on this topic [16,17], and in particular, that of triple stage (treble) compression, an option widely used in the LNG and cold process industry where a large ΔT_{lift} is also necessary. A benefit of triple stage compression is that the pressure across each compressor is limited, increasing its performance while decreasing the superheat losses. As such, we propose comparing a number of possible triple stage compression architectures with some of the currently available two stage, in particular, two-refrigerant cascade cycle, with or without internal heat exchangers, to quantify the potential expected improvements.

The novelty of the work lies in investigating multiple potential architectures as well as using a variable compressor map, unlike the previous authors [16], as well as proposing in so far as the authors are aware, a novel treble compression cycle that of cascade economization, presented in Section 2.1. This cycle can achieve three compression processes with only two compressors by staging one of these compressors. The paper is structured in the following manner:

- **Thermodynamic Modeling & Cycles:** Cycles analyzed, thermodynamic equations and assumptions.
- **Refrigerant Selection:** Selecting refrigerant the best refrigerant candidates for the cycles.
- **Parametric Analysis:** Comparative performance of different cycles and impact of operating conditions. Optimal system parameters ($T_{cascade}$). Impact of modeling assumptions and system parameters (pinch analysis, compressor map).
- **Discussion:** Outcomes from parametric analysis, identified trends, and proposed future work.

2. Thermodynamic Modeling & Cycles

2.1. Cycle Architectures

Economization cycles are able to improve thermodynamic performance of cycles by splitting one larger less efficient compression process to two smaller more efficient ones which results in performance gains. Since

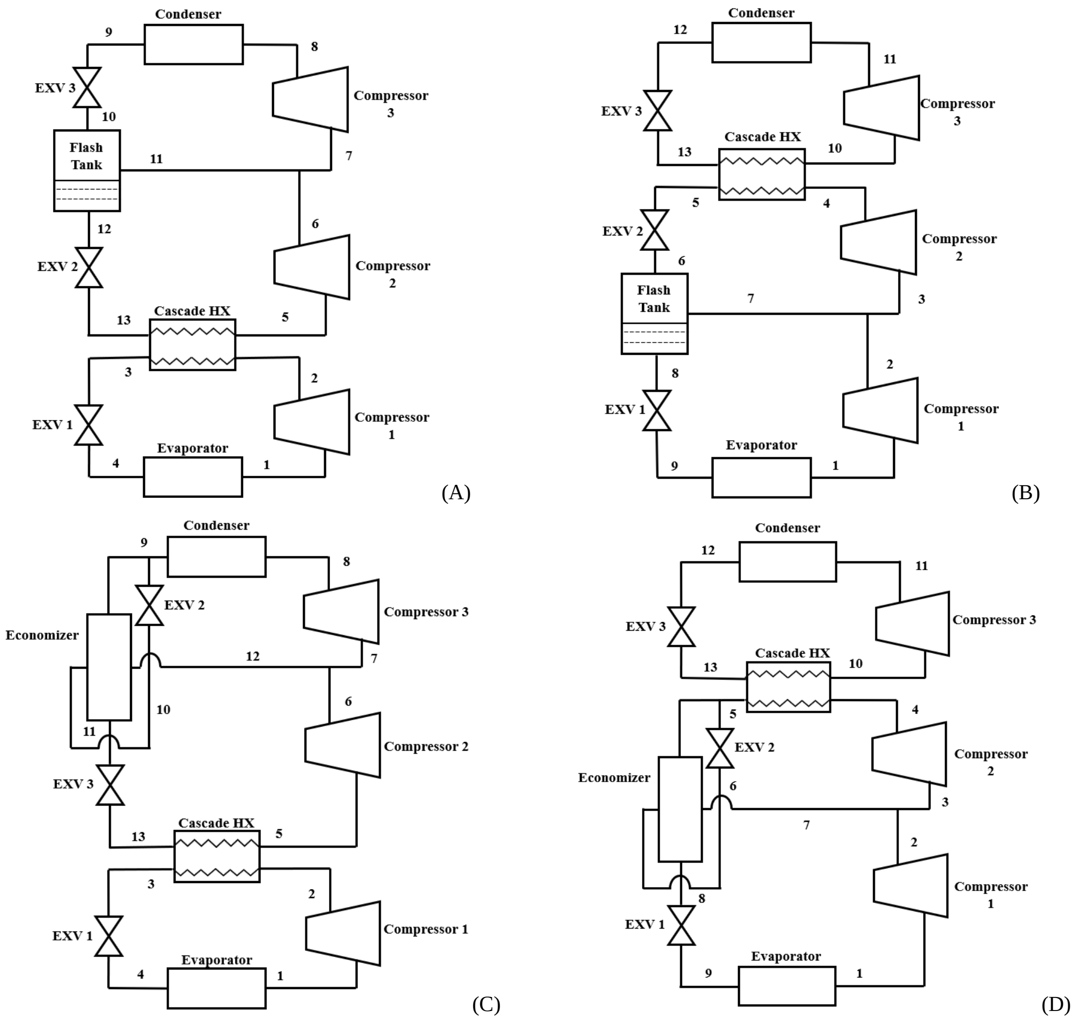

Fig 1. Cascade economization cycles investigated: (A) Open economization HT, (B) Open economization LT, (C) Closed economization HT, (D) Closed economization LT.

this work aims to analyze the potential of staging compressors in HTHPs, closed and open economization were investigated. In particular, given the high lift temperatures sought, the cycles are proposed in a cascade format, where staging occurs either on the low or high side. These cascade sides use different refrigerants to optimize cycle performance and are henceforth referred to in this paper as LT for the low temperature loop and HT for the high temperature loops. These cycles are illustrated in Figure 1.

The cycles were compared to the industry standard for high lift temperatures, namely the cascade cycle (see Ochsner IWDSS R2R3B). Additionally, an improved cascade cycle given in [24] was investigated using two internal heat exchangers. Lastly, a triple cascade cycle with three separate compressors, rather than staged as in Figure 1(A), was also analyzed. These cycles are provided in Figure 2.

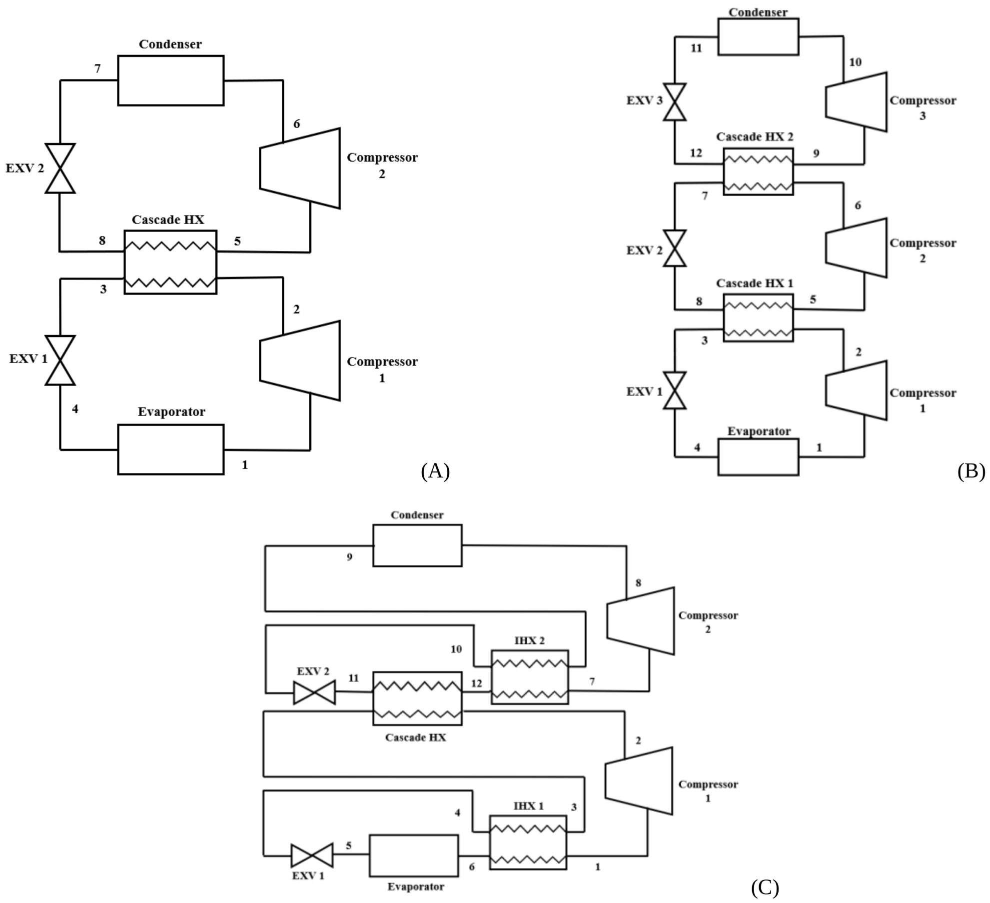

Fig 2. Variations of the simple cascade investigated: (A) Baseline cascade, (B) Triple cascade, (C) Cascade with internal HXs.

2.2. Thermodynamic Model

A thermodynamic model was developed to investigate the system performance. Calculations were performed in EES [18]. The following modeling assumptions were made to develop the models:

- Steady-state, steady-flow analysis.
- Kinetic Energy and Potential Energy are neglected.
- Negligible pressure drops and heat losses to the surrounding in heat exchangers and piping.
- Adiabatic expansion valves with isenthalpic expansion.

The compressor power can be expressed in terms of isentropic efficiency:

$$\dot{W}_c = \frac{\dot{m}(h_{out,is} - h_{in,is})}{\eta_{is}} \tag{1}$$

To define isentropic efficiency, one must select a compressor technology, which is mainly subject to the refrigerant used and capacity of the system. For HTHP applications, the vast majority of research and industrial development is focused on piston, screw, twin-screw, and turbo compressors, of which the most prominent type is that of screw. As such, we will primarily focus on screw compressors, with Equations 2 and 3 representing the compressor's isentropic and volumetric efficiency as a function of the pressure ratio. The impact of compressor isentropic efficiency will be further investigated in Section 4.3.

$$\eta_{is} = b + c ln\left(\frac{\dot{v}_{in}}{3600}\right) + d(r_{vc,i}) \quad (2)$$

$$\eta_{vol} = 0.95 - 0.0125 r_{pc,i} \quad (3)$$

These correlations were obtained from [14,22] and the fit variables are given in [22]. The volumetric heat capacity (VHC) was used to select the best refrigerant in the cascade cycle. It was modeled accounting for the low and high sides as per [10]:

$$VHC = \rho_{c,in}\eta_{vol}\Delta H_{cond} \quad (4)$$

Heat exchangers were modeled using pinch analysis, the values shown in Table 2. For the internal heat exchanger effectiveness, Equation 5 was used.

$$\varepsilon_{IHX} = \frac{h_{vap,out} - h_{vap,in}}{h_{vap,out} - h_{liq,in}} \quad (5)$$

$$\dot{Q}_{HX} = \dot{m}(h_{out} - h_{in}) \quad (6)$$

For the cascade HX, the approach of [19] was used. In terms of the economization, the optimum pressure from the two-stage economization cycle was used (no cascade) [20].

$$P_{int} = \sqrt{P_{high}P_{low}} \quad (7)$$

Finally, the system COP and 2nd Law efficiency can be defined as:

$$COP = \frac{\dot{Q}_{sink}}{\sum_i \dot{W}_{c,i}} \quad (8)$$

$$COP_{Carnot} = \frac{T_{sink}}{T_{sink} - T_{source}} \quad (9)$$

$$\eta_{II} = \frac{COP}{COP_{Carnot}} \quad (10)$$

Table 2. Assumptions and conditions used in modeling simulations.

Parameter	Assumed Value
Sink Temperature [°C]	100 – 150
Source Temperature [°C]	-10 – 40
ΔT_{lift} (T_{source}- T_{sink}) [°C]	60 – 110
Compressor Type	Screw
Pinch [°C]	10
Subcooling [°C]	5
Superheat [°C]	5
ΔT_{casc} [°C]	5
IHX effectiveness (ε)	0.7
T_{cas}	Obtained by using Golden Section Algorithm in EES for COP maximization
Nominal state for refrigerant selection†	T_{source} = 40 °C T_{sink} = 120 °C

†The nominal state was used in the simulation (Section 3) on the refrigerant selection section to compute the VHC and COP.

3. Refrigerant Selection

To compare the performance of the cycles in Section 2.1, it was necessary to select some appropriate refrigerants. However, choosing refrigerants in a cascade cycle for a high temperature heat pump system is not straightforward. An optimal refrigerant should address thermodynamic aspects (e.g., thermal stability at high temperatures, volumetric capacity, etc.), environmental and safety concerns (e.g., GWP, ODP, flammability and toxicity), and techno-economic considerations. The following analysis shows all relevant parameters for a proper selection. Due to the cascade nature of the cycles investigated, two refrigerants have to be selected, one to operate on the high side and one on the low side. The normal boiling point (NBP) should be low enough to allow vaporization during regular operation , while the critical temperature of the refrigerant defines the upper temperature border for a system that is not supposed to run in supercritical operation. So especially for the high temperature loop, selecting a refrigerant whose critical point is above the anticipated condensation temperature is essential. From an environmental standpoint, the refrigerant should have an ODP of 0 and a GWP that is as low as possible. Furthermore, the safety classes helped to distinguish between lower toxicity (class A) and higher toxicity (class B) as well as to estimate the flammability (increasing in the order of 1, 2L, 2, and 3) [21]. Low toxicity and flammability are generally preferred, but the thermodynamic properties sometimes require a tradeoff. Finally, the techno-economic parameters are important. Values like the volumetric heat capacity (VHC) can tell how big the components will be, so this value should be high to reduce component sizing. At the same time a high COP is preferred to allow an efficient operation. To initiate the calculations, R-290 was used as a reference fluid in the low-temperature cycle, while R-600 was the reference fluid for the low temperature loop refrigerant.

The comparison for the low-temperature cycle refrigerants (Table 3) shows that the highest COP can be achieved with R-717 (ammonia). The pros of this refrigerant are its high VHC, which is the third highest in the comparison. Also, it is a natural refrigerant with an ODP and a GWP of 0. However, it is toxic and mildly flammable. In addition, the economic factor comes into place. Ammonia systems require stainless steel components and cannot be manufactured with copper parts. Therefore, when accounting for VHC, COP, GWP, and classification, **R-32** is the appropriate candidate. In this work, A3 refrigerants were eliminated due to adoption issues in practical applications, although the reader is directed to the following [9,12]. R-600 and R-601 are attractive options for the HT side, albeit flammable. As such, research has been focused on the following HFOs: R-1336mzz(Z), R-1234ze(Z), R-1233zd(E), and R-1224yd(Z) [8]. Of these, **R-1233zd(E)** was selected primarily due to its high performance and critical temperature. In the case of the triple cascade cycle, R-32 was used for the LT loop, while R-1233zd(E) for MT and HT. A different choice for the MT loop was not made since the analysis was limited to T_{sink} = 150 °C. Had a higher sink temperature been required, a 3rd choice, such as R-601, would have been made.

Table 3. Low Temperature Loop Refrigerants

Refrigerant	NBP	Critical Temperature	ODP	GWP	Safety Classification	T_{CAS}	VHC	COP
	(°C)	(°C)	(-)	(kg CO_2)	ASHRAE [25]	(°C)	(kJ/m3)	(-)
R-290	-42.1	96.7	0	~10	A3	72.9	5724	2.41
R-744 (CO2)	-78.5	31.0	0	1	A1	-	-	-
R-717 (NH3)	**-33.3**	**132.4**	**0**	**0**	**B2L**	**87.7**	**9717**	**2.62**
R-718 (H2O)	-33.3	132.4	0	<1	A1	103.3	63	2.55
R-1216	-30.3	85.7	0	<1	B3	66.1	4577	2.30
R-1234yf	-29.5	94.7	0	4	A2	70.1	4395	2.36
R-125	-48.1	66.0	0	3500	A1	58.0	7760	2.18
R-134a	-26.1	101.1	0	1430	A1	74.2	4896	2.44
R-143a	-47.2	72.7	0	4470	A2	64.6	7188	2.26
R-161	-37.5	102.1	0	12	A3	77.8	6493	2.48
R-32	**-51.6**	**78.1**	**0**	**675**	**A2L**	**69.9**	**11496**	**2.34**
R-404A	-46.2	72.1	0	3900	A1	63.6	7105	2.24
R-507A	-46.7	70.6	0	3985	A1	61.0	7459	2.23
R-600	**- 0.5**	**152.0**	**0**	**4**	**A3**	**81.4**	**1992**	**2.59**

Table 4. High Temperature Loop Refrigerants

Refrigerant	NBP	Critical Temperature	ODP	GWP	Safety Classification	T_{CAS}	VHC	COP
	(°C)	(°C)	(-)	(kg CO_2)	ASHRAE [25]	(°C)	(kJ/m3)	(-)
R-600	**-0.5**	**152.0**	**0**	**4**	**A3**	**72.9**	**3990**	**2.41**
R-601	36.1	196.6	0	5	A3	69.4	1697	2.57
R-1336mzz(Z)	**33.4**	**171.3**	**0.0003**	**2**	**A1**	**71.2**	**1966**	**2.47**
R-1234ze(Z)	**9.8**	**150.1**	**0.0001**	**<1**	**A2L**	**70.8**	**3714**	**2.47**
R-1233zd(E)	**18.0**	**166.5**	**0**	**1**	**A1**	**69.5**	**2936**	**2.53**
R-1224yd(Z)	**14.0**	**155.5**	**0**	**<1**	**A1**	**72.5**	**3223**	**2.44**
Novec 649	49.0	168.7	0	<1	N/A	83.4	1304	2.19
R-717 (NH_3)	-33.3	132.4	0	0	B2L	64.2	17512	2.46
R-718 (H_2O)	**100.0**	**373.9**	**0**	**0**	**A1**	**56.6**	**242**	**2.78**
R-744 (CO_2)	-78.5	31.0	0	1	A1	-	-	-
R-245fa	58.8	154.0	0	1030	B1	73.2	3594	2.43
R-365mfc	40.2	186.85	0	782	N/A	69.6	1629	2.51

4. Results

4.1. Impact of operating conditions and Second Law Efficiency

The cycles presented in Section 2.1 were numerically investigated in EES. The purpose of this analysis was to quantify the performance degradations resulting from compression across a range of ΔTlift and the potential benefit of staging one of the compression processes. Firstly, the source temperature was fixed at 40 °C, and the sink temperature was increased. This would indicate a process with a stable waste heat source, and the sink is required at a high temperature, similar to the pulp process industry. The results are shown in Figure 3.

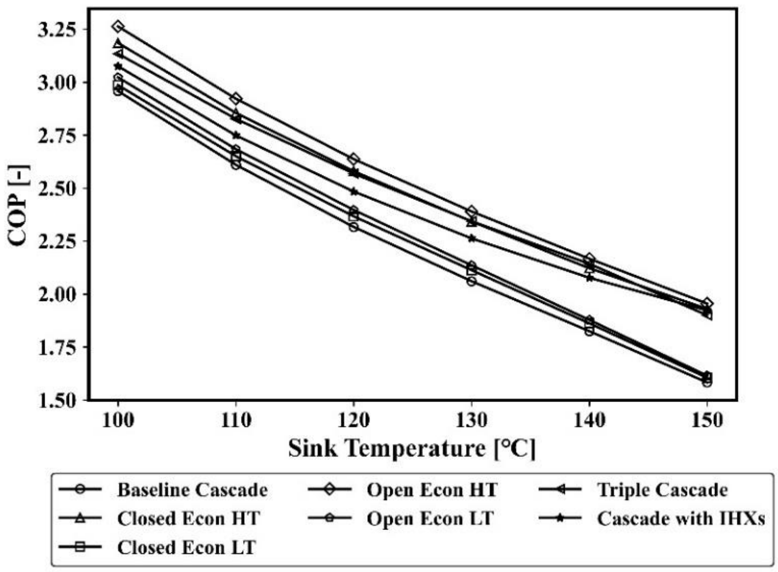

Fig 3. Performance (COP) of different cycles for increasing sink temperature.

The results indicate that cycles that perform the two-stage compression on the HS outperform those that do not. Of these, open economization beats closed economization. Open economization on the high side offers an increasing COP enhancement with ΔT_{lift} over the simple cascade, starting at 10% (ΔT_{lift} = 60 °C) and increasing up to 25% for (ΔT_{lift} = 110 °C). Cascade with IHXs and triple cascade are the 3rd and 4th performing cycles, respectively, even when not accounting for the pressure drops in the IHXs. These trends are also shown in Figure 4 with respect to the second law efficiencies of these systems.

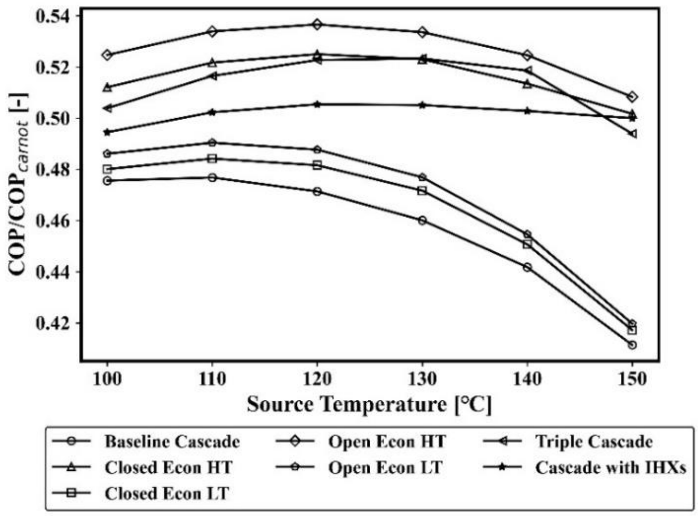

Fig 4. Second law efficiency of different cycles for increasing sink temperature.

With the goal of HTHPs to replace typical gas furnaces, it is also vital to investigate scenarios where waste heat is not readily available and rather air source systems is used. An industrial example of such a heat pump is Ochsner IWDSS R2R3B. The sink temperature is fixed at 120 °C. This is shown in Figure 5.

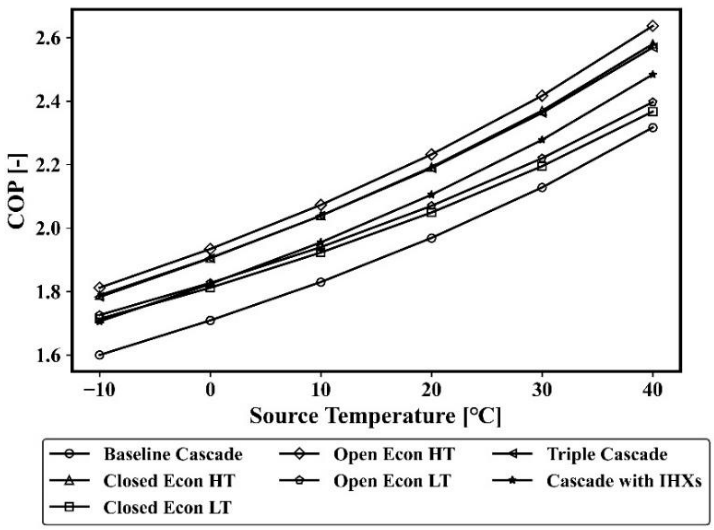

Fig 5. Performance (COP) of different cycles for decreasing source temperature

The results presented so far indicate that the performance of the cascade cycle rapidly deteriorates since the lift at each compression process is significant. To understand this further, by redefining the lift temperature as $\Delta T_{lift,LT} = T_{cas} - T_{source}$ and $\Delta T_{lift,HT} = T_{sink} - T_{cas}$, and assuming that the optimal cascade temperature is approximately $(T_{sink} + T_{source})/2$, at $T_{sink} = 140$ °C and $T_{source} = 40$ °C we can compute $\Delta T_{lift,LT} = \Delta T_{lift,HT} = 50$ °C. This represents an adverse temperature lift, resulting in high-pressure ratios and that a staging process similar to the ones presented in this paper are required. The impact of T_{cas} is further investigated in Section 4.2

4.2. Optimizing Cycle Parameters for COP

It was identified that two system parameters could be treated as open for optimization. Those are the intermediate pressure used in the staging of the compressor and the cascade temperature. In this work, the empirical expression shown in equation 7 has been used to simplify the analysis. Instead, optimization in EES was performed by noting that $COP(T_{cas})$ and using the golden-section algorithm to find the best combination of COP and T_{cas}. In the case of the triple cascade, $COP(T_{cas,LT}, T_{cas,HT})$ was used to perform combined optimization. Future work should also be directed at identifying $COP(T_{cas}, P_{inter})$ since the square root rule (widely adopted empirical relation in Eq. 7) might not hold. The results presented in Section 4.1 are based on the optimized cascade temperatures.

Figure 6 indicates that based on which side the staging occurs, T_{cas} is calculated so that the temperature across this side increases given that the isentropic efficiency is better for the dual-stage compressor on that side. This means that for the cascade cycle, the cascade temperature remains close to $(T_{sink} + T_{source})/2$ since staging does not occur [10]. A similar trend is seen for the triple cascade cycle, where the temperatures are attempting to split ΔT_{lift} in three equal fractions. The slight for the triple cascade in Fig 6 occurs to the refrigerant properties.

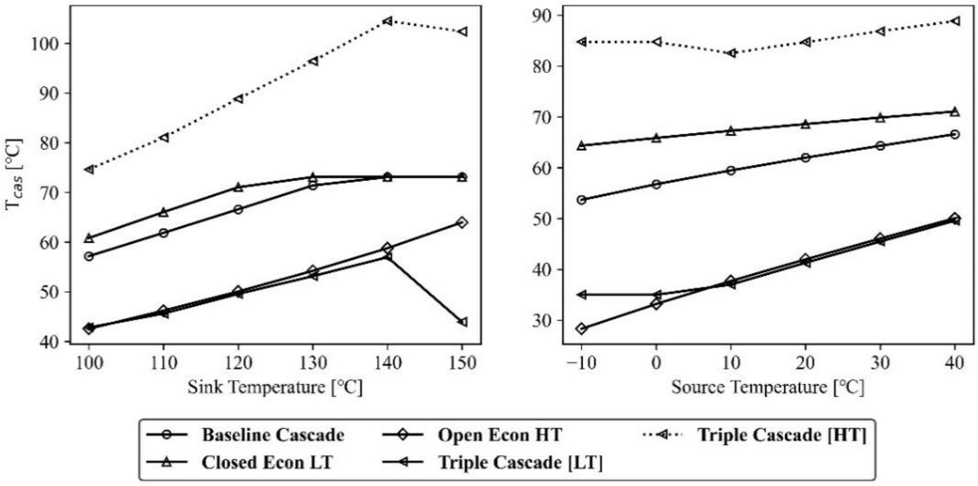

Fig 6. Optimum cascade temperature for different arrangements and different sink and source temperatures.

4.3. Impact of System Assumptions

Finally in this section we investigate the impact of the system assumptions on the heat exchanger modeling as well as the compressor map used.

4.3.1. Heat Exchanger Modeling

Simple thermodynamic models were selected for the heat exchangers given the comparative nature of the analysis. For the refrigerant-to-refrigerant (non IHX, cascade) HXs and the air/water to refrigerant HXs, a pinch analysis approach was used since temperature on the refrigerant side remains relatively constant due to phase change. From the thermodynamic results. it was identified that while the superheat has negligible impact to the performance of the cycle, while increasing subcooling enhances capacity and therefore the COP. Figure 7 indicates the importance of using high performance heat exchangers and that an open economization cycle on the HT side can achieve a COP of almost 2.5 at a ΔT_{lift} = 100 °C (20 – 120 °C), at a pinch of 5 K which is achievable with current heat exchanger technology.

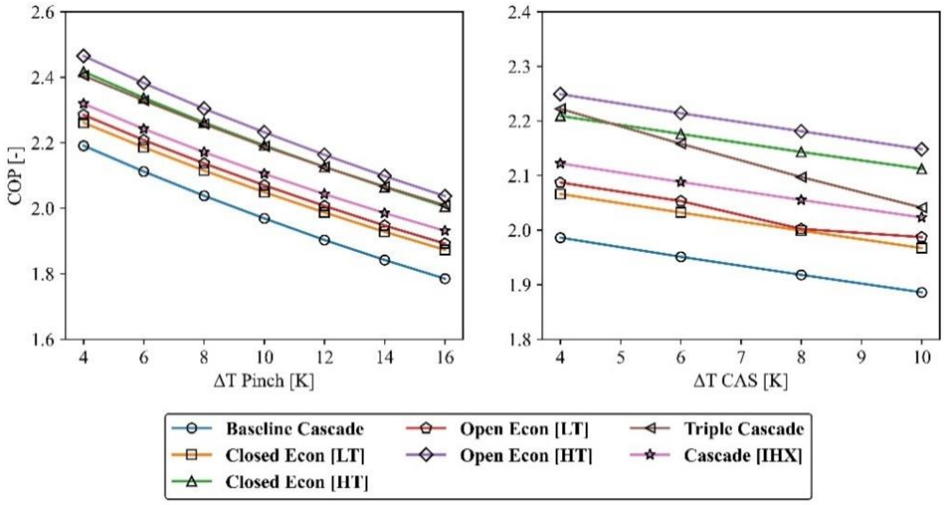

Fig 7. Impact of HX modeling, pinch, superheat, subcooling and cascade pinch on cycle performance.

While the pinch analysis could be used in the case of the phase change processes, it cannot be used in the modeling of the IHX given that the refrigerant (same refrigerant on both sides) is in the subcooled and superheated regions. As such, the effectiveness approach was adopted. Figure 8 serves a two-fold purpose. Firstly, the impact of the IHX effectiveness was analyzed. By changing the value from 0.5 to 0.9, it was found that the IHX on the high side (see Figure 2C) had significantly greater impact on the performance of the cycle. As such, the impact of the effectiveness (ε) was investigated on a cycle with one IHX, two IHXs and compared to the cascade open economization cycle. It was found that using only a single higher performing IHX on the HT loop outperforms having two IHXs. However, even when using highly efficient IHXs are used on either both sides or the high side, and not accounting for pressure drops, this cycle was not able to outperform the open economization cycle.

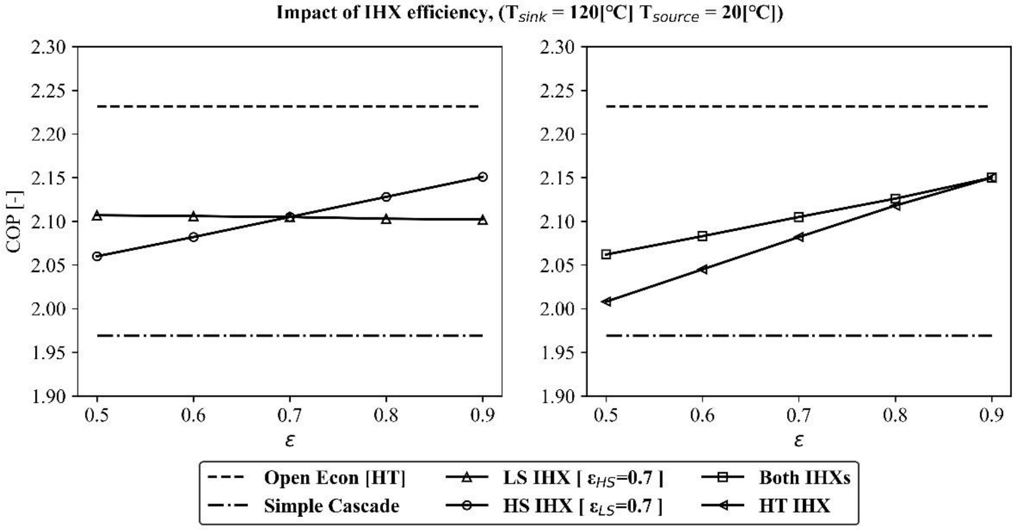

Fig 8. COP as a function of ε for T_{sink} = 120 °C and T_{source} = 20 °C.

4.3.2. Compressor Mapping

Using the compressor map in [14, 22] for a single screw compressor, it was identified that the compressor isentropic efficiency was approximately 0.7 for the cascade cycle. In contrast, the cycles with two-stage compressions were in the order of 0.75 ~ 0.8 due to the improved distribution of temperature lifts and compression ratios across the system. However, it should be noted that limited information and experimental studies exist on compressors for HTHPs, given that this is a nascent field.

5. Discussion

The results of the previous analysis show that a cascade cycle with economization has clear advantages compared to a simple cascade cycle. Economized cycles use a flash tank between the condenser and evaporator under a predefined intermediate pressure controlled by the expansion valve. The gaseous fraction of the refrigerant is used to cool down the compression process. Therefore, the gas at intermediate pressure gets expanded and injected into the compressor. This technology cools down the gas temperature during the compression process and therefore improves the isentropic efficiency of the compressor. Even if that cycle is more cost-intensive due to additional components and an adapted compressor design, the performance enhancements of 25% are significant. An example of a two-stage screw compressor was prosed in [23]. It should be noted that this compressor is different from a twin-screw compressor in that the two screws do not mesh, but rather, compression occurs in two different single screws driven in a single planar arrangement.

The analysis used a compressor map that changes the isentropic efficiency based on the volume ratio the compressor has to achieve (typical to use volume ratio for screw compressor). For the comparison, the map of a screw compressor was used [14, 22]. The benefits of staging can be adopted by either compressor types (reciprocating, centrifugal), such as scroll for smaller capacities with the impact varying based on the impact of pressure ratio its isentropic efficiency.

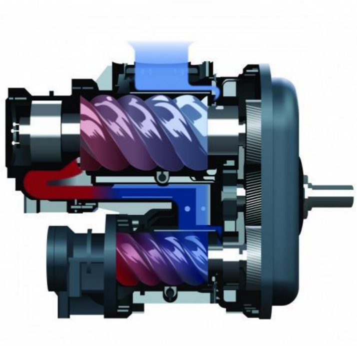

Fig 9. An example of a two-stage screw compressor for air applications, the two screws have a common stator housing and drive [23].

Even cascades with two internal heat exchangers, as used in other publications, show lower efficiencies than the economized cycle [24]. Therefore, based on the technoeconomic results of Kosmadakis [14], it seems that the additional cost of an improved compressor with a flash tank coupled with significant performance enhancements in high lift cycles outweighs using a standard cascade cycle. Further, with respect to introducing IHXs to as an efficiency measure to the cascade cycle, limited benefit is gained by using two while using only one on the HS seems to offer some performance enhancements, albeit lower than staging. It should be noted that plate internal heat exchangers pose additional detriments, such as oil separation, pressure drops, and have a significant cost. From our preliminary analysis, the staging proposed seems to outperform this efficiency measure.

However, additional work is needed with detailed component models, particularly for the compressor, given that limited experimental data are available for HTHP compressors, to reach a conclusive outcome. This work aims to initiate this conversation, particularly in the case of high temperature lift applications, which represent a significant proportion of the process heat industry where heat pumps are or could be used.

6. Conclusions

High temperature heat pump is a technology that can offer significant greenhouse gas reduction in the area of industrial processing. However, they suffer from substantial performance losses and irreversibilities largely due to the extreme temperature lifts they need to achieve. In this work, the potential of staging one of the compression processes was investigated. A number of proposed thermodynamic cycles were investigated by varying the type of economization (open or closed) as well as the location (high or low side). These arrangements were compared against the industry standard cycle for high lift applications, the cascade cycle, with two different refrigerants, one selected for the high side and one for the low side. Additionally, a modified high performance cascade cycle was investigated with two IHXs before the compression processes. The refrigerants selected for the investigation were R-32 and R-1233zd(E) for the low-side and high-side, respectively, based on their thermodynamic properties, low GWP, and ASHRAE classifications. It was found that staging the compression process can lead to performance enhancements (COP) in the order of 25% for high temperature lift applications over the baseline cascade cycles. The highest performing cycle with a staged compression process used two-stage economization on the high side with an open economizer (flash tank). This work highlights the need for integrated cycle and compressor design to achieve widespread HTHP integration at realistic supply temperatures.

Acknowledgements

This work was partially supported by the Center for High Performance Buildings (CHPB) at Purdue University and the Onassis Foundation Fellowship awarded to Elias Pergantis, one of the foundation's Scholars.

References

[1] Fox DB, Sutter D, Tester JW. The thermal spectrum of low-temperature energy use in the United States. Energy Environ Sci 4, p. 3731.
[2] Arpagaus C, Bless F, Uhlmann M, Schiffmann J, Bertsch SS. High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials. Energy 152, p. 985-1010.

[3] Kim J, Park, SR, B YJ, Chang KC, Ra HS, Kim M, Kim Y,. Experimental study of operating characteristics of compression/absorption high-temperature hybrid heat pump using waste heat. Renewable Energy 54, p. 13-19.
[4] Furukawa T. Industrial high-temperature heat pump for waste heat recovery. New Energy Conservation Technologies and Their Commercialization, Proceedings of an International Conference, Berlin, Germany, Springer-Verlag Berlin Heidelberg, New York (April 6-10 1981), pp. 1483-1493
[5] Jensen JK, Markussen WB, Reinholdt L, Elmegaard B. On the development of high temperature ammonia-water hybrid absorption- compression heat pumps Int J Refrig, 58 (2015), p. 79-89
[6] Jiang J, Hu B, Wang RZ, Deng N, Cao F, Wang CC. A review and perspective on industry high-temperature heat pumps. Renewable and Sustainable Energy Reviews 161, 2022, 112106.
[7] Adamnson KM, Walmsley TG, Carson JK, Chen Q, Schlosser F, Kong L, Cleland DJ. High-temperature and transcritical heat pump cycles and advancements: A review. Renewable and Sustainable Energy Reviews 167, 2022.
[8] Fukuda S, Kondou C, Takata N, Koyama S. Low GWP refrigerants R1234ze(E) and R1234ze(Z) for high temperature heat pumps. Int J Refrig, 40 (2014), p. 161-173
[9] Bamigbetan O, Eikevik TM, Neska P, Bantle M, Schlemminger C. Theoretical analysis of suitable fluids for high temperature heat pumps up to 125 °C heat delivery. Int. J. Refrig. 92, 2018, p. 185-195.
[10] Mateu-Royo C, Arpagaus C, Mota-Babiloni A, Navarro-Esbri J, Bertsch SS. Advanced high temperature heat pump configurations using low GWP refrigerants for industrial waste heat recovery: A comprehensive study. Energy Conversion and Management 229 (2021) 113752.
[11] Mota-Babiloni A, Mateu-Royo C, Navarro-Esbri J, Moles F, Amat-Albuixech M, Barragan-Cervera A. Optimisation of high-temperature heat pump cascades with internal heat exchangers using refrigerants with low global warming potential. Energy 165 (2018), p. 1248 - 1258.
[12] Bamigbetan O, Eikevik TM, Nekså P, Bantle M. Development of propane-butane cascade high temperature heat pump: early test rig results International Workshop on High Temperature Heat Pumps, Copenhagen, Denmark (September 9, 2017), pp. 1-12
[13] Mateu-Royo C, Navarro-Esbri J, Mota-Babiloni A, Moles F, Amat-Albuixech M. Experimental exergy and energy analysis of a novel high-temperature heat pump with scroll compressor for waste heat recovery. Applied Energy 253, 2019.
[14] Kosmadakis G, Arpagaus C, Neofytou P, Bertsch SS. Techno-economic analysis of high-temperature heat pumps with low-global warming potential refrigerants for upgrading waste heat up to 150 C, Energy Conversion and Management 226 (2020) 113488.
[15] Fernandez-Moreno A, Mota-Babiloni A, Gimenez-Prades P, Navarro-Esbri J. Optimal refrigerant mixture in single-stage high-temperature heat pumps based on a multiparameter evaluation. Sustainable Energy Technologies and Assessments 52, Part A, 2022.
[16] Kondou C, Koyama S. Thermodynamic assessment of high-temperature heat pumps using Low-GWP HFO refrigerants for heat recovery. Int. J. Refrig. 53, 2015, p. 126-141.
[17] Hu B, Wu D, Wang LW, Wang RZ. Exergy analysis of R1234ze(Z) as high temperature heat pump working fluid with multi-stage compression. Front. Energy 2017, 11(4): 493–502.
[18] Klein S, Alvarado F. Engineering equation solver. F-Chart Software (2002), Madison, WI.
[19] Arpagaus C, Bless F, Schiffmann J, Bertsch SS. Multi-temperature heat pumps: A literature review. Int J Refrig, 69 (2016), p. 437-465.
COPStoecker, W. F. (1998). Industrial Refrigeration Handbook. McGraw-Hill Education.
[21] ASHRAE Safety Standard for Refrigeration Systems, ANSI/ASHRAE 34-2019, *Designation and Safety Classification of Refrigerants*.
[22] Astolfi M. Techno-economic optimization of low temperature CSP systems based on ORC with Screw expanders. Energy Procedia 2015;69:1100–12
[23] FS Curtis, NxHE Series Two-Stage 90-260kW, URL: https://us.fscurtis.com/product/nxhe-series-90-260kw-two-stage-rotary-screw/, Accessed: 10/10.
[24] Mota-Babiloni A, Mateu-Royo C, Navarro-Esbrí J, Molés F, Amat-Albuixech M, Barrágan-Cervera A. Optimisation of high-temperature heat pump cascades with internal heat exchangers using refrigerants with low global warming potential, Energy 165 (2018), p. 1248-1258.
[25] ANSI/ASHRAE 34-2019, Designation and Safety Classification of Refrigerants.

Nomenclature

$\dot{W}c,i$	Power of i^{th} compressor [kW]
$\dot{Q}$	Heat Supply [kW]
$\dot{m}$	Mass flow rate [kg/s]
h	Specific enthalpy [kJ/kg]
η_{is}	Isentropic efficiency [-]
η_{vol}	Volumetric efficiency [-]
η_{II}	Second law efficiency [-]
P	Pressure [kPa]
$\dot{v}$	Volumetric flow rate [m^3/s]
$r_{pc,i}$	Pressure ratio across ith comp. [-]
$r_{vc,i}$	Volume ratio across ith comp. [-]
ρ	Density [kg/m_3]
vap	vapor
cond	condenser
EXV	Electronic expansion valve
Econ	Economizer
CAS	Cascade
Δ	Difference
Σ	Sum
ε	Effectiveness [-]