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"Demonstration of an ambient temperature-sourced steam generating high-temperature heat pump"

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Abstract

Heat for industrial processes must be decarbonized to achieve societal emissions goals. Steam generating heat pumps could reduce the operating cost and energy requirements compared to conventional electric boilers. However, to achieve competitive coefficients of performance (COPs), technological advancements are needed, including highly efficient compressors to achieve the high pressure ratios needed for high lift heat pumping applications. A novel, ambient-sourced, cascade heat pump system capable of delivering steam up to 150 ˚C has been developed to serve industrial processes using custom built centrifugal compressors. A nominal 650 kW laboratory pilot system was designed, constructed, and operated in Fort Collins, CO. The laboratory pilot was operated over a range of conditions using the test bed at CSU which controls source temperature and steam demand. This pilot machine ultimately achieved an electrical COP of 1.48 when heat pumping from 15 ˚C to produce 150 ˚C steam. While the custom centrifugal compressors achieved their isentropic efficiency targets (>80%), overall COP was limited by suction line heat exchanger underperformance. Long term performance was also limited by degraded motor efficiency caused by damage in the top cycle high-temperature compressor motor. Design revisions correcting these inefficiencies were incorporated into a second heat pump unit design for a commercial pilot application. The commercial pilot system was tested in Loveland, CO and demonstrated a system electrical COP of 1.73, a 43% reduction in electrical energy compared to electric resistance boilers. This successful demonstration highlights the potential for high-temperature heat pumps to play a transformative role in the decarbonization of industrial heat.

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Keywords: High-temperature heat pump; Industrial heat; Electrification

1. Introduction

Heat for industrial processes must be decarbonized to achieve societal emissions goals [1]. Electrifying energy consuming processes that are currently fossil fuel powered can be an effective means of reducing emissions [2]. When electrifying low temperature industrial heat demand, steam generating heat pumps could reduce the operating cost and energy requirements compared to conventional electric boilers. To realize the benefits of heat pumping for industrial heat, technological advancements are needed to increase the coefficient of performance (COP) of high temperature lift heat pumps. One critical need is advanced compressors which can achieve the high pressure ratios required with sufficient efficiency to deliver an overall heat pump cycle with competitive COP. Additionally, while many steam-generating heat pump designs propose using waste heat, waste heat can be intermittent and can lead to high integration costs. An ambient air-sourced heat pump

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designed for drop-in replacement of existing boilers can have improved economic performance by reducing project costs and decreasing payback period [3].

AtmosZero has developed a novel, ambient-sourced, cascade heat pump system capable of delivering steam up to 150 ˚C to serve industrial processes using custom built centrifugal compressors. A conceptual illustration of the heat pump product is shown in Figure 1. A full-scale 650 kW laboratory pilot was designed, constructed and operated at Colorado State University's Energy Institute. This demonstration unit establishes the potential for drop-in ambient-sourced steam generating heat pumps to lower the operating costs of electrifying industrial heat demand.

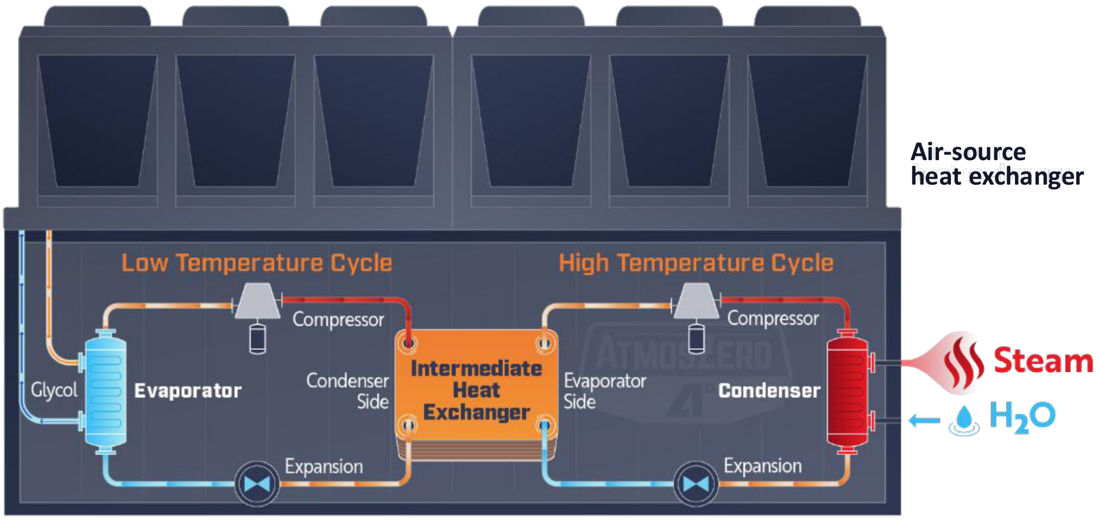

Figure 1. Simplified illustration of the AtmosZero heat pump product.

2. Methods

2.1. Process description

The AtmosZero laboratory pilot uses a two-stage cascade heat pump cycle to efficiently move heat from an ambient temperature source into a steam generator sink, as illustrated in Figure 2. The process was first designed in Engineering Equation Solver (EES) [4]. The laboratory pilot bottom cycle uses R-513a, a non-ozone depleting, lower global warming potential (GWP) substitute refrigerant for R-134a, as the working fluid (AR5 GWP = 573) [5]. R-513a was selected due to its relatively low GWP, commercial availability, favorable thermal properties at the desired testing conditions, non-flammability and non-toxicity, and current acceptance in North American and European markets (i.e. GWP is below thresholds set by local regulations). Ambient heat is absorbed by evaporating refrigerant in a bottom cycle evaporator, HX-240. The evaporated refrigerant is then compressed using a two-stage intercooled compressor. The compressed refrigerant is then condensed in the middle heat exchanger, HX-140, which rejects heat into the top cycle. A fraction of the condensed refrigerant is expanded for use in the bottom cycle economizer, HX-220 where it is vaporized to provide additional subcooling to the main liquid stream. After the economizer, this fraction is injected into the bottom cycle compressor intercooler line. After being subcooled in the economizer, the main liquid stream is then expanded and sent to the evaporator for the cycle to begin again.

The top cycle uses R-1233zd(E), a non-flammable, non-ozone depleting, ultra-low GWP replacement for R-123 (GWP = 1) [6]. R-1233zd(E) was selected as a suitable high-temperature fluid due to its availability, high critical temperature (165.7C), and current acceptance in North American and European markets. The top cycle absorbs heat from the bottom cycle by evaporating refrigerant in the middle heat exchanger, HX-140. The evaporated refrigerant is further heated in a suction line heat exchanger (SLHX), HX-130, which adds superheat to prevent liquid migration into the compressors. The vapor is then compressed through two two-stage centrifugal compressors. The compressed refrigerant vapor is then condensed in a steam generator, HX-110, where pressurized water is evaporated and sent to the steam header. A fraction of the condensed refrigerant is expanded to further subcool the main liquid stream in the top cycle economizer, HX-120. The expanded fraction is subsequently injected into an intercooler line before the high-pressure compressor. After

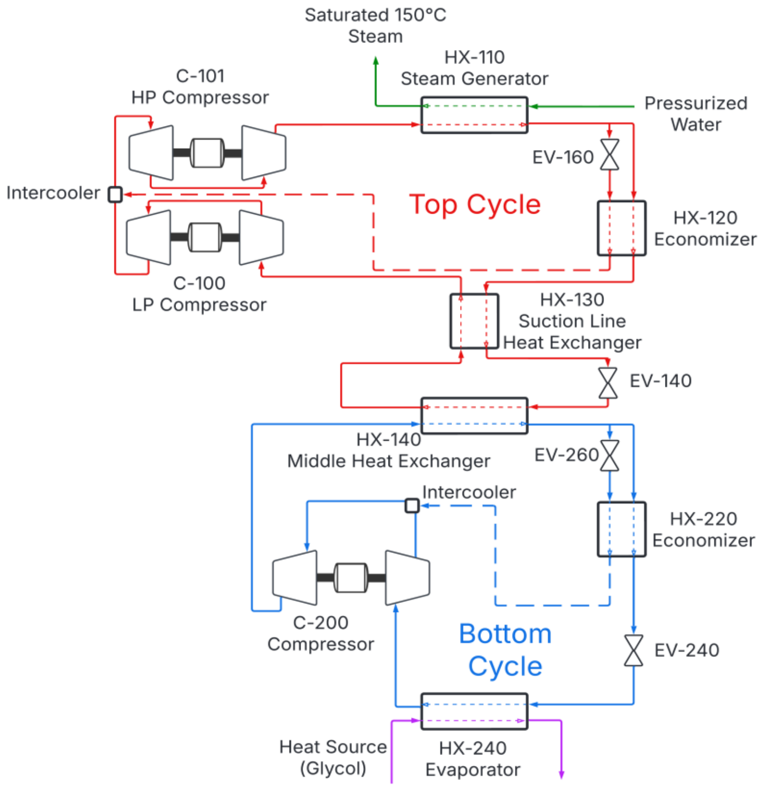

Figure 2. AtmosZero steam generating heat pump process flow diagram.

HX-120, the main liquid stream is then further subcooled in the SLHX, before being expanded and sent to the middle heat exchanger for the cycle to begin again.

While the selected working fluids are currently accepted in the previously mentioned markets, there is risk of future regulations introducing stricter GWP thresholds or restricting the use of PFAS-containing refrigerants, such as HFOs, due to their separate environmental and health concerns. Suitable non-PFAs replacements primarily include natural refrigerants such as hydrocarbons, which pose additional safety risks due to their high flammability. If future regulatory policies restrict HFO use, then natural refrigerants may be required despite these safety concerns.

2.2. Laboratory pilot construction

The laboratory pilot was designed and constructed using primarily commercial-off-the-shelf (COTS) components to develop a low-cost, mass-manufacturable heat pump product.

2.2.1. Heat exchangers

Most of the heat exchangers use COTS stainless steel brazed plate designs selected from standard sized offerings. Some include advanced features, like inlet distributors to ensure even heat transfer in two phase streams. The exception to COTS is the steam generator, HX-110, which is a custom shell and tube heat exchanger of Tubular Exchanger Manufacturers Association (TEMA) type BKU. The kettle style shell allows for improved separation of steam from entrained liquid water. The large water inventory in the kettle adds thermal inertia to the system, which improves stability of operation, but also increases start up time.

2.2.2. Compressors

The bottom cycle compressor, C-200, is a COTS double-ended centrifugal magnetic bearing compressor designed for mass market chilling applications. The top cycle compressors, C-100 and C-101, were custom designed and manufactured by AtmosZero. These machines are two-stage, double-ended centrifugal machines with refrigerant-lubricated bearings. They are driven by COTS high speed variable frequency drives (VFDs). The four stages of compression feature custom aerodynamic design, which is critical to attaining the efficiencies needed for a competitive COP at high temperature lifts, common in ambient sourced heat pumps. A photograph of the uninsulated compressor assembly installed on the heat pump is shown in Figure 3.

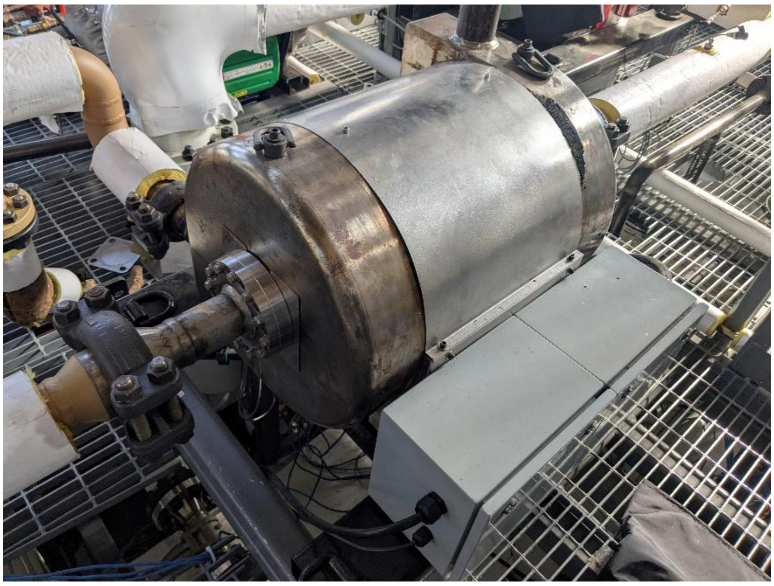

Figure 3. AtmosZero custom designed and manufactured high speed centrifugal compressor.

2.2.3. Controls and data acquisition

The system operates on a programmable logic controller that provides proper hot and cold start sequencing as well as safety functionality. The core instruments required for controls and safe operations communicate directly to the PLC. Additionally, a LabVIEW-based data acquisition system collects information from a broader set of instruments, as well as duplicated safety and control instruments. Temperature instruments (RTDs and thermocouples) were calibrated using a temperature-controlled oil bath and a HART Scientific 5615 Platinum Resistance thermometer with a Fluke 1502A Thermometer Readout. Pressure transmitters were calibrated by the manufacturer.

2.3. CSU test facility

In parallel with AtmosZero's heat pump development, CSU designed and constructed a custom test facility for the project. The test facility controls the heat input to target a specified source temperature, provides feedwater, and receives steam to reject the heat to a cooling tower, simulating the end-use heat pump consumer. The process flow diagram of the system is shown in Figure 4. On the heat input side, a gas-fired boiler heats a 50% ethylene glycol-water stream to maintain an adjustable temperature setpoint, simulating an ambient heat source. That glycol is sent to the heat pump, which removes heat through the evaporation of refrigerant and returns the glycol at a lower temperature. The glycol flow rate is adjusted based on capacity to maintain a 5 to 6 ˚C temperature delta to improve measurement confidence and enable calculation of an accurate heat duty. The test facility also provides pressurized feedwater to the heat pump, which modulates flow using a control valve to maintain target level in the BKU steam generator. The feedwater is optionally pre-heated in a shell and tube heat exchanger using a slipstream of the produced steam. The heat pump boils the feedwater to deliver steam to the CSU test facility, where it is condensed against an intermediate loop of a 92% ethylene glycol-water blend (labeled HTF for "heat transfer fluid"). The steam condenser uses a steam trap to ensure condensate is removed for even condensing heat transfer. The HTF flow is modulated to control duty in the condenser, and then it rejects heat to a final 30% ethylene glycol-water coolant loop which is sent to an evaporative cooling tower for rejection to the environment. Flash steam and condensate from the outlet of the trap is also condensed and subcooled in a separate condensate cooler against the same 30% glycol coolant loop in parallel with the main HTF cooler. The cooled condensate is collected in a tank, and the cycle begins again.

The CSU test facility uses a combination of shell and tube heat exchangers for condensing steam service and gasketed plate frame exchangers for glycol-to-glycol service. The feedwater pump is a VFD-driven regenerative turbine pump that ensures high delivered feedwater pressure. The HTF pump is a VFD-driven centrifugal pump that can achieve a wide range of flow rates to allow control of condensing duty in the main steam condenser. The CSU facility is run on its own LabVIEW based data acquisition system, and it independently measures key performance parameters of the heat pump. Verified parameters include delivered

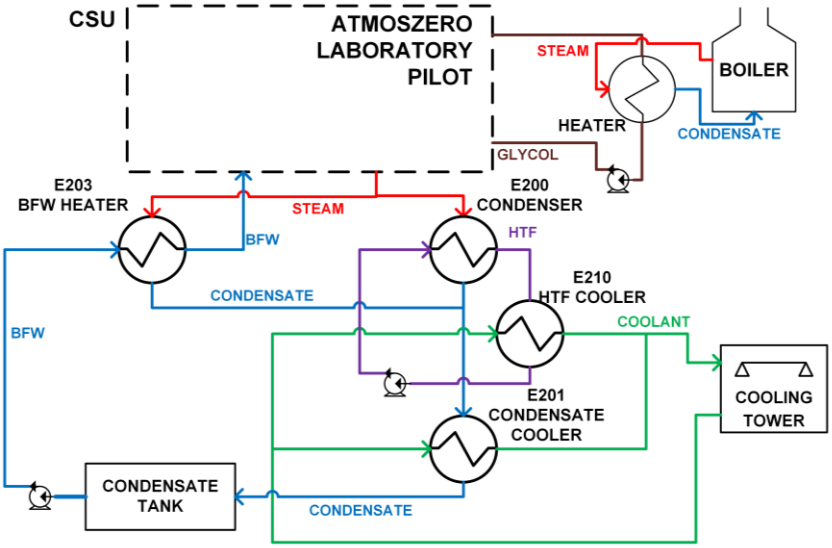

Figure 4. CSU test facility process flow diagram showing heat input, feedwater, steam, and heat rejection loops.

steam mass flow, temperature and pressure; heat input mass flow, supply temperature, and return temperature; and boiler feedwater delivered mass flow, temperature and pressure. These parameters enable the CSU facility to confirm thermal performance of the AtmosZero heat pump system and ensure safe and reliable performance.

The CSU test facility controls the heat source to within +/- 1 ˚C in a range from -10 ˚C to greater than 40 ˚C to simulate a wide range of ambient conditions. It can also be operated when the heat pump is in "load following" mode, where the heat pump capacity modulates to match the CSU condenser heat demand, or to follow a constant heat pump capacity. The test facility can accept steam at any condition from ambient pressure (T_{sat}~95 ˚C at 5000' elevation) up to 165 ˚C steam. It can also supply feedwater from slightly above the coolant temperature (~30 ˚C) up to nearly pinched with the delivered steam temperature, to simulate different customer feedwater header conditions. An ambient-temperature heat pump can be air or liquid sourced. For air-sourced, the heat pump can either directly evaporate refrigerant against air, or it can conduct heat from the air to an intermediate working fluid such as glycol, which will then transfer heat to the refrigerant in the evaporator. If an intermediate loop is selected, some additional temperature difference is required to drive sensible heat transfer. The AtmosZero air-sourced product currently uses an intermediate glycol loop for simplicity. For this reason, the CSU supply temperature represents an air temperature approximately five degrees higher for the air-sourced AtmosZero product. Using the nominal results reported, a 15 ˚C supplied temperature in the test facility approximately represents a 20 ˚C air temperature for the AtmosZero product.

2.4. Physical layout

The laboratory pilot was constructed on a second-floor mezzanine at the CSU Energy Institute Powerhouse Energy Campus. The unit consists of an electric power and controls panel on the south end, two large frames measuring in total 16' long by 11' wide by 9' tall containing the core heat pump, and the custom designed BKU steam generator on the north end. In the core heat pump, the heat exchangers are on the base level, with compressors mounted on top to reduce the risk of liquid ingestion. An aerial view photograph of the arrangement is provided in Figure 5, along with a plan view illustration.

2.5. Definition of performance parameters

The key performance parameters of the system are delivered heat capacity in kW, and COP_{elec} which is dimensionless. Both the heat pump and the CSU test facility use temperature compensated vortex mass flow meters to measure steam flow rates, which proved unreliable due to the presence of condensate and hidden failures occurring on internal temperature compensation instruments. The energy balance using these measurements consistently failed to close. For this study, capacity results are instead presented on the basis of the measured refrigerant mass flow and change in enthalpy from the inlet to the outlet of the BKU steam generator. We have higher confidence in the refrigerant mass flow measurement because it is definitively single-phase at the measurement locations. The electricity consumed is measured by current transformers at

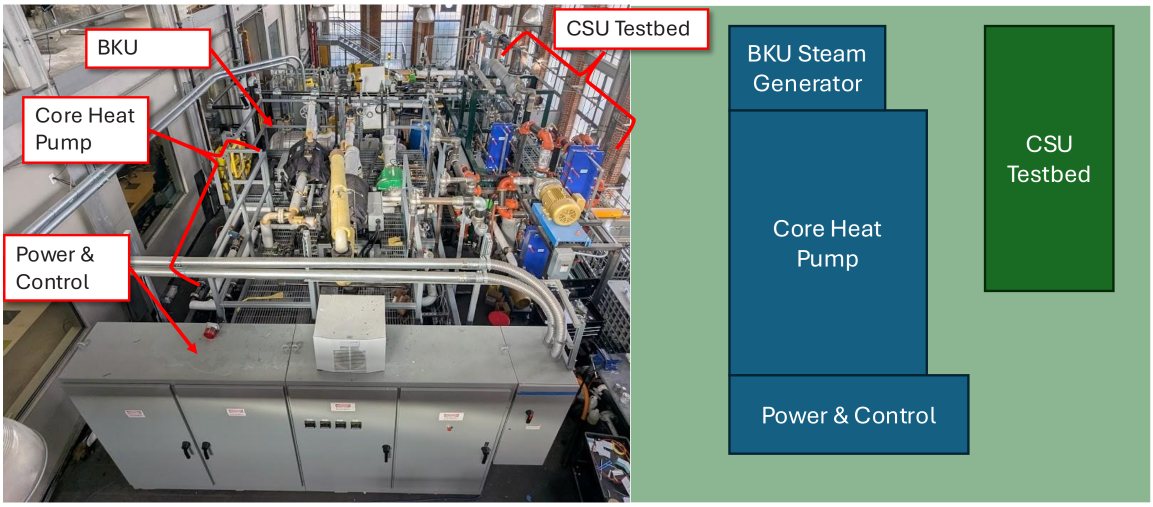

Figure 5. Left: Overview photograph of pilot heat pump plus test facility. Right: plan view illustration of major system arrangement.

the main supply to the heat pump. This measurement includes power for all compressor motors and AtmosZero heat pump accessory loads but excludes the boiler feedwater pump motor and heat source fluid motor which are on the CSU test facility side.

This measurement basis yields the following Equations (1) and (2) for the key system performance parameters [7]:

$$\dot{Q}_{HP} = \dot{m}_{refrigerant}(h_{in} - h_{out}) \tag{1}$$

$$COP_{elec} = \frac{\dot{Q}_{HP}}{\dot{W}_{in}} \tag{2}$$

Equation (1) finds the heat pump capacity, $\dot{Q}_{HP}$, using the top cycle refrigerant mass flow, $\dot{m}_{refrigerant}$ and the change in specific enthalpy from inlet to outlet, $h_{in} - h_{out}$. The enthalpies are calculated from pressure and temperature of single-phase fluids using REFPROP properties called from MATLAB [8]. In Equation (2), the COP_{elec} is calculated from the heat pump capacity, $\dot{Q}_{HP}$, and the total work input, $\dot{W}_{in}$.

For component level performance metrics, there are two additional equations. First, each heat exchanger heat duty is calculated similarly to the $\dot{Q}_{HP}$, but are generalized in Equation (3) [7]. HX-110 uses refrigerant side data, due to the unreliable steam flow meter. Note that $\dot{Q}_{HX-110}$ is therefore the same as $\dot{Q}_{HP}$. HX-120, HX-220, and HX-140 use the average of the liquid and gas duty. HX-130 uses only the liquid side, due to an unknown vapor fraction that can occur at the inlet due to a bearing return line injection point. HX-240 uses refrigerant side data and has been validated against the glycol data which is measured on the CSU side. Cases where the inlet is a two phase stream of unknown quality after an expansion valve assume isenthalpic expansion and calculate h_{in} based on the upstream side of the expansion valve. Note that only positive values are reported.

$$\dot{Q}_{HX,any} = \dot{m}_{refrigerant}|h_{in} - h_{out}| \tag{3}$$

Finally, Equation (4) is used to calculate the compressor isentropic efficiencies [7]. State 1 represents the inlet state and state 2 represents the outlet state. The isentropic outlet enthalpy, h_{2s} is determined by assuming the specific entropy is constant across the compressor, after evaluating the specific entropy at the inlet from measured temperature and pressure using REFPROP. The other specific enthalpies are calculated directly from REFPROP using measured temperatures and pressures.

$$\eta = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{4}$$

3. Results and Discussion

3.1. Heat pump performance at nominal design point

The heat pump nominal design point is to generate 650 kW of 150 ˚C steam from a 15 ˚C temperature source. All initial testing was conducted with 15 ˚C glycol (equivalent to 20 ˚C air temperature for the air-sourced product). Expected COP_{elec} was 1.6-1.7. In initial testing, the heat pump achieved steady state operation on several days of testing with capacity ranging from 396 to 505 kW and COP_{elec} ranging from 1.40 to 1.48. The best individual test point was the combination of the best capacity of 505 kW and best COP_{elec} of 1.48 on June 14, 2024. A summary table showing the performance of major individual components is shown in Table 1. Note that due to lack of interstage instrumentation on the COTS C-200, efficiency can only be imputed from an estimated state at the stage 2 suction.

Table 1. Component level and system performance comparison.

Cycle	Component	Parameter	Units	Design	Experimental (High Capacity)	Experimental (Low Capacity)
Top Cycle	C-100	Power	kW	133.4	122.0 ± 0.6	104.6 ± 0.5
		Efficiency	%	80.5	81.4 ± 1.9	82 ± 1.7
	C-101	Power	kW	134.3	124.0 ± 0.6	101.6 ± 0.5
		Efficiency	%	79.9	77.7 ± 1.9	80.6 ± 2.0
	HX-110, Steam Generator	Heat Duty	kW	650.1	505.4 ± 6.8	396 ± 5.1
	HX-120, Economizer	Heat Duty	kW	42.8	70.8 ± 5.8	47.1 ± 4.5
	HX-130, SLHX	Heat Duty	kW	214	105.0 ± 4.6	84.6 ± 3.6
	HX-140, Middle HX	Heat Duty	kW	395.4	299.6 ± 4.4	234.4 ± 3.1
Bottom Cycle	C-200	Power	kW	121.1	94.4 ± 0.5	69.6 ± 0.4
		Efficiency	%	72	76.0 ± N/A (*imputed)	76.0 ± N/A (*imputed)
	HX-220, Economizer	Heat Duty	kW	68.7	30.7 ± 2.9	17.9 ± 2.1
	HX-240, Evaporator	Heat Duty	kW	274.2	212.6 ± 3.3	171.6 ± 2.4
System		COP_{elec}	-	1.6-1.7	1.484 ± 0.017	1.430 ± 0.018

3.2. Technical challenges

3.2.1. Motor performance degradation

On July 29, 2024, a loss of cooling to the motors resulted in a motor winding temperature spike and a permanent loss of motor efficiency on the high pressure machine (stages 3 and 4 of top cycle compression). Subsequent results show a permanently reduced capacity and COP. The leading hypothesis is that demagnetization occurred at the high temperatures caused by the loss of cooling. Figure 6 shows unique test days, grouped by before and after the event occurred.

3.2.2. Suction line heat exchanger underperformance

The function of the SLHX is twofold. First, by heating the suction gas before compression, it ensures that liquid does not enter the compressors which could cause damage. Second, it performs an important thermodynamic function by adding subcooling to the condensed liquid before expansion, which results in a lower quality after isenthalpic expansion. The lower quality allows more latent heat to be absorbed in the middle heat exchanger, increasing the heat capacity of the system per unit of refrigerant mass that is circulated. Since the electrical work is proportional to the circulated mass, this also increases the COP_{elec}.

During testing, we observed that the overall heat transfer rate in the SLHX was lower than design. The SLHX is built from two identical parallel brazed plate cores, designated A and B. The cores have inlet and outlet collection headers to split and recombine the flows on both the liquid (hot) and gas (cold) sides. The individual cores are also equipped with their own outlet temperature instruments. Further investigation using the A, B, and combined liquid outlet temperatures revealed a mass flow maldistribution caused by inlet manifold geometry, with flow bypassing the A core and preferentially entering the B core. This maldistribution

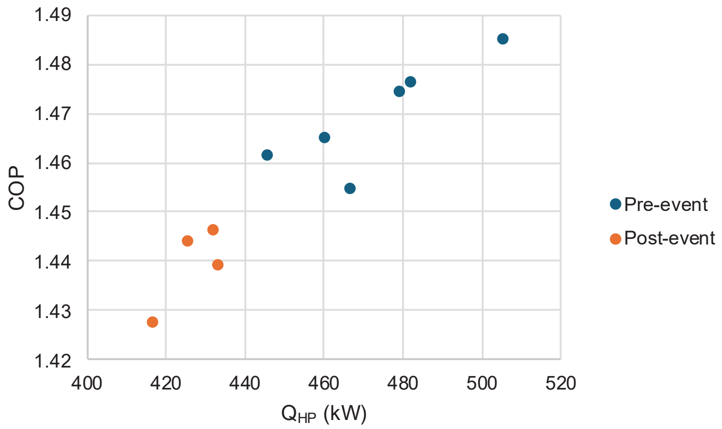

Figure 6. Loss of cooling on July 29, 2024 resulted in a permanent loss of motor capacity, illustrated by selected operating results pre- and post-event.

causes a temperature pinch in the A core as the reduced mass flow is quickly heated, and the result is a reduced overall heat transfer rate and duty.

The reduced overall heat transfer duty in the SLHX causes a reduction in both capacity and COP_{elec} by reducing latent heat absorption in HX-140. This effect can be directly observed by comparing the P-h diagrams for the design point and the experimental measurements of the system in the high capacity case, which are shown in Figure 7. Using updated models, we estimate this SLHX effect to be responsible for about half of the performance gap between the design COP_{elec} of 1.67 and the experimental COP_{elec} of 1.48.

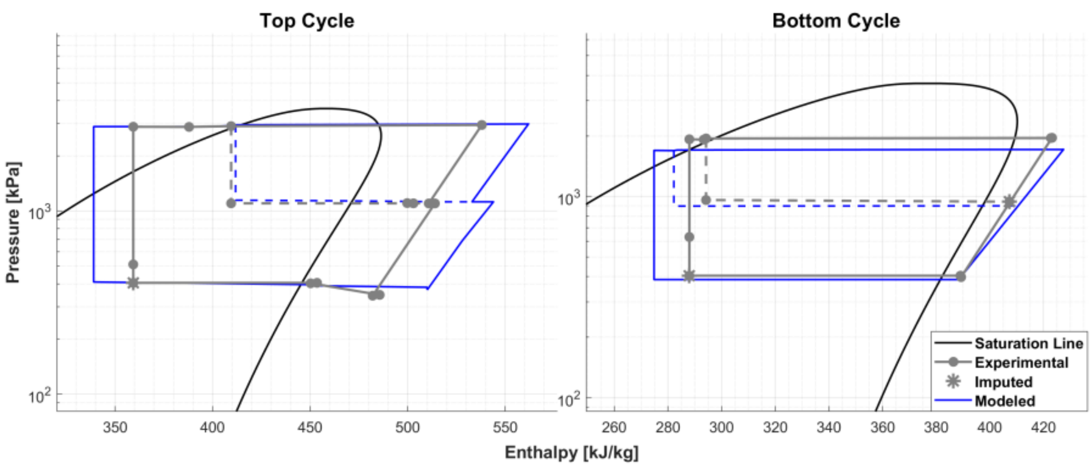

Figure 7. Pressure-enthalpy plot of the laboratory pilot experimental performance versus theoretical modeled performance.

3.3. Performance of commercial pilot

Incorporating the lessons learned from the laboratory pilot, AtmosZero developed a second pilot unit at matched scale to be deployed at a commercial pilot site. The commercial pilot was tested over a range of conditions at a new test bed at AtmosZero's facility in Loveland, CO. The COP_{elec} was improved by 8.7% over the laboratory pilot, with a max COP_{elec} of 1.73 at the nominal design point of 15 ˚C air temperature to 150 ˚C steam. This improvement highlights the importance of prototyping at relevant scale prior to commercial deployment. Future work will provide more details and analysis of the commercial pilot.

3.4. Comparison to electric boilers

The COP_{elec} of 1.73 means that to generate 1 unit of heat, only 0.578 units of electricity are required. Conventional electric boilers that use electrical resistance to generate heat are approximately 98% efficient and therefore require 1.02 units of electricity to generate 1 unit of heat [9]. The performance demonstrated by the commercial pilot heat pump demonstrates a 43.3% reduction in energy consumption and cost compared to the state of the art for electrified steam production.

4. Conclusions

This project has demonstrated the capability of steam generating heat pumps to reduce operating cost and energy requirements compared to electric boilers. While the initial laboratory pilot was successful, it faced technical challenges including degraded motor performance and heat exchanger maldistribution that limited the cycle capacity and COP_{elec}. Through design iteration, these limits were alleviated, and the commercial pilot demonstrated a COP_{elec} of 1.73 when heat pumping from 12 ˚C to produce 150 ˚C steam. This successful demonstration highlights the potential for high-temperature heat pumps to play a transformative role in the decarbonization of industrial heat.

References

[1] G. P. Thiel and A. K. Stark, “To decarbonize industry, we must decarbonize heat,” Mar. 17, 2021, *Cell Press*. doi: 10.1016/j.joule.2020.12.007.

[2] T. Mai *et al.*, “Electrification Futures Study: Scenarios of Electric Technology Adoption and Power Consumption for the United States,” 2018. [Online]. Available: https://www.nrel.gov/docs/fy18osti/71500.pdf.

[3] A. K. Stark, A. Salvi, and T. Bandhauer, “Waste heat is mostly a waste of time,” *Joule*, p. 102157, Oct. 2025, doi: 10.1016/j.joule.2025.102157.

[4] S. A. Klein, “EES - Engineering Equation Solver,” Aug. 19, 2025, *F-Chart Software*: Professional V12.124. Accessed: Nov. 05, 2025. [Online]. Available: https://fchartsoftware.com

[5] The Chemours Company, “Opteon ™ XP10 Refrigerant Product Information,” 2018. Accessed: Nov. 05, 2025. [Online]. Available: https://www.opteon.com/en/-/media/files/opteon/opteon-xp10-prodinfo.pdf

[6] Honeywell, “Storage, Handling and Use Guidelines for Solstice ® zd Refrigerant,” 2014. Accessed: Nov. 05, 2025. [Online]. Available: https://www.honeywell-refrigerants.com/europe/wp-content/uploads/2015/08/Solstice-zd-Handling-Guidelines.pdf

[7] Y. A. . Çengel, M. A. . Boles, and Mehmet. Kanoğlu, *Thermodynamics : an engineering approach*. McGraw Hill, 2024.

[8] E. W. Lemmon and I. H. Bell and M. L. Huber and M. O. McLinden, “NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 10.0, National Institute of Standards and Technology.” Accessed: Nov. 09, 2025. [Online]. Available: https://doi.org/10.18434/T4/1502528

[9] U.S. Energy Information Administration, “Updated Buildings Sector Appliance and Equipment Costs and Efficiencies,” Mar. 2023. Accessed: Nov. 09, 2025. [Online]. Available: https://www.eia.gov/analysis/studies/buildings/equipcosts/pdf/appendix-a.pdf#page=118.99