High-Temperature Heat Pump Model Documentation and Case Studies

Jordan Cox, Scott Belding, Gustavo Campos, and Travis Lowder

National Renewable Energy Laboratory

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Technical Report
NREL/TP-7A40-84560
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List of Acronyms

AEC	annual energy cost
BAU	business as usual
BTU	British thermal unit
CO_2	carbon dioxide
COP	coefficient of performance
FO&M	fixed operation and maintenance
HTHP	high-temperature heat pump
IRR	internal rate of return
LCC	life-cycle cost
LCOH	levelized cost of heat
NPV	net present value
NREL	National Renewable Energy Laboratory
PBP	payback period
REopt	Renewable Energy Integration and Optimization
TEA	technical and economic assessment
TOU	time of use
UR	utility rate
VO&M	variable operation and maintenance

Executive Summary

High-temperature heat pumps (HTHPs) are electrically powered systems that supply heat above 90°C. HTHPs have the potential to serve two valuable functions in United States (U.S.) industry. First, by valorizing and elevating waste heat streams, HTHPs can improve industrial energy efficiency. Second, by electrifying process heat generation, HTHPs can replace combustion systems with less carbon-intensive sources of energy, such as renewable resources, and can reduce industrial greenhouse gas emissions. In certain applications, the combination of increased efficiency, cost savings, and emissions reductions makes HTHPs a promising component of strategies for industrial electrification and clean energy transitions.

Researchers at the U.S. Department of Energy's National Renewable Energy Laboratory (NREL) developed a Python-based HTHP model. This model has both a physics-based performance estimation component and an economic evaluation component that is designed to demonstrate the potential economic competitiveness of HTHPs, which was found to be driven by electricity versus gas prices.

This document has several sections. First, an overview of HTHPs is provided, including a literature review of relevant state-of-the-art information. Second, a description of the HTHP model is provided, along with detailed instructions on how to use it. Third, case studies to showcase the utilization of the HTHP model are provided, along with the corresponding economic results.

Four major elements comprise the HTHP model's physics-based component: a coefficient of performance (COP) calculator for heat pumps given boundary operating conditions, a heat pump energy balance module, a gas boiler energy balance module, and an electro-resistive energy balance module. These modules can be combined in a system of heat technologies. An example is a hybrid heat pump and electro-resistive system. The economics module of the HTHP model can be used to compare the net present value, the internal rate of return, and the payback period of any two system configurations. This report provides the equations used for each module and presents a guide on how to construct a system using the Python programming language for economic comparison.

To demonstrate the capabilities of the HTHP model, two case studies are presented: beer brewing and yogurt making. Previous work identified the food and beverage production industries as the first mover industries for HTHPs (Jakobs and Stadlander 2020). These case studies were selected because they provide high-temperature heat (>90°C) above what is provided by traditional heat pumps. Both case studies were evaluated for economic competitiveness compared to traditional gas heating systems. Our analysis found that the beer-brewing case is not economically competitive for the given location (California), but the yogurt-making case is economically competitive for the given location (Vermont).

HTHPs represent a unique opportunity for U.S. industry to decarbonize and recover wasted energy. This report demonstrates that there are economically competitive opportunities for HTHPs in the United States. In this work, the case study in Vermont was found to be economically competitive, though economic competitiveness will depend on the industry and on electricity versus fossil fuel prices in the local region. In addition to economic competitiveness,

our partners identified several barriers to HTHP adoption, which are provided in the final section on discussion and conclusions.

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1 Introduction

The industrial sector is a major energy consumer and a source of carbon dioxide (CO_2) emissions in the United States, accounting for approximately 35% of total energy end-use consumption and 28% of CO_2 emissions in 2019 (Figure 1). The industrial sector is considered difficult to decarbonize because of its high reliance on fossil fuels, processes that cannot be directly replaced by renewable resources, and a lack of robust research on electrification methods (Madeddu 2020). Process heat (thermal energy used in the manufacturing of products) comprises the majority (51%) of industrial energy demand in the United States (Ruth 2019). Importantly, more than two-thirds of this heat demand are for processes at or below 300°C (Figure 2). Further, of the more than 7 quadrillion British thermal units (BTUs) demanded annually for process heating in the United States, 36% is lost as unrecovered waste heat, leading to 2.5 quadrillion BTUs of heat with the potential to be harnessed through increased efficiency (DOE 2015).

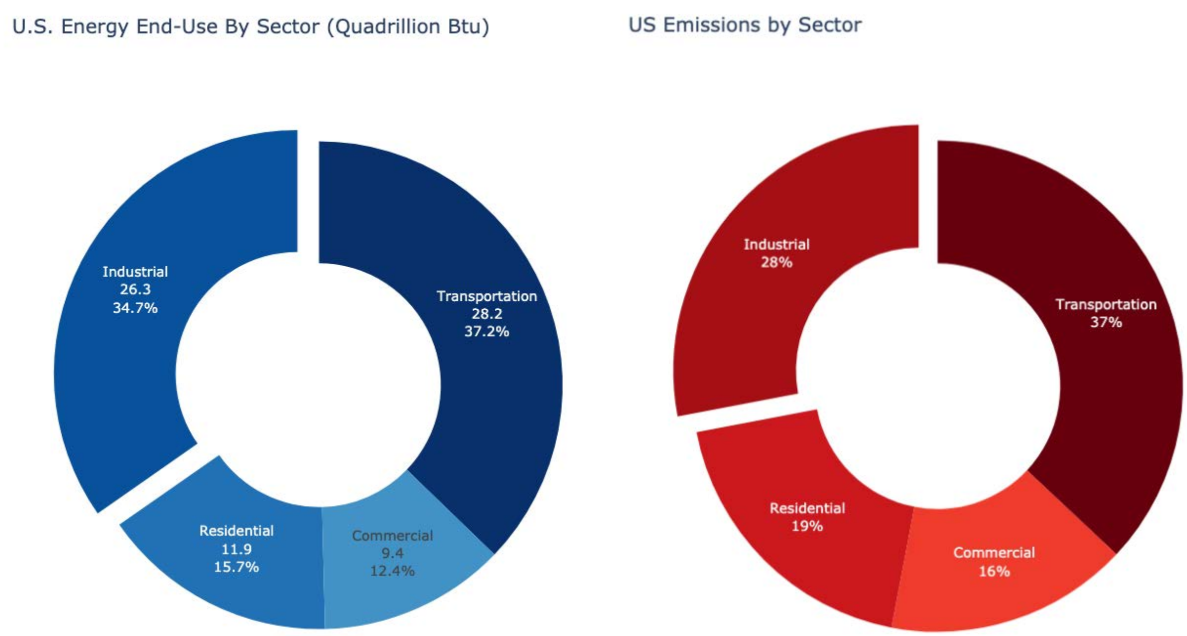

Figure 1. U.S. energy use (quadrillion BTU) and CO_2 emissions by sector in 2019

Source: EIA 2020. *Figure by NREL*

High-temperature heat pumps (HTHPs) available today can provide heat at temperatures between 90°C and 160°C. (Some models can go higher but have lower performance and thus were not the focus of this work (Arpagaus et al. 2018)). Process heat in this range can be useful to many industrial sectors, including food and beverage manufacturing, corn milling, pulp and paper production, industrial drying, and the chemical industry. For widespread sectoral adoption, however, HTHPs must present a compelling economic case, achieved through energy cost savings. The effectiveness of a heat pump can be expressed through the coefficient of performance (COP), which describes the HTHP's thermal energy output in relation to its demanded electrical energy. Due to thermodynamic limits and refrigerant and compressor inefficiencies in real-world operation, most current HTHPs operate within an approximate COP range from 2–6, or between 40%–50% of their theoretical maximum (Carnot) efficiency

(Arpagaus et al. 2018; Arpagaus and Bertsch 2020; Kosmadakis et al. 2020; Schlosser et al. 2020; Tveit 2017). HTHP performance levels required for economic viability are highly contextual because of adoption barriers and facility-scale conditions.

Despite these barriers and uncertainties, HTHPs are well matched to many applications, and active areas of research can improve the understanding, performance, and reliability of HTHPs to fully realize the full benefits of industrial heat pump adoption.

Circumstances such as high natural gas prices and decreasing electricity costs from renewable resources present large economic and decarbonization opportunities via both waste heat capture and the provision of industrial process heat. HTHPs are a promising technology on both fronts. "High" is a relative term; HTHPs include heat pumps that deliver temperatures above 90°C and historically have been limited to a maximum of roughly 160°C. Although this range includes the extreme frontier of what heat pumps as a technology have been able to deliver, these temperatures are still on the low end of the industrial use spectrum. Because heat pumps elevate an existing waste/available heat stream, they can improve the usefulness and increase the utilization of industrial waste heat. Also, heat pumps are electric, which means they possess the potential (depending on grid generation sources) to decarbonize industrial energy by displacing fossil-fueled combustion systems, such as natural gas-fired boilers.

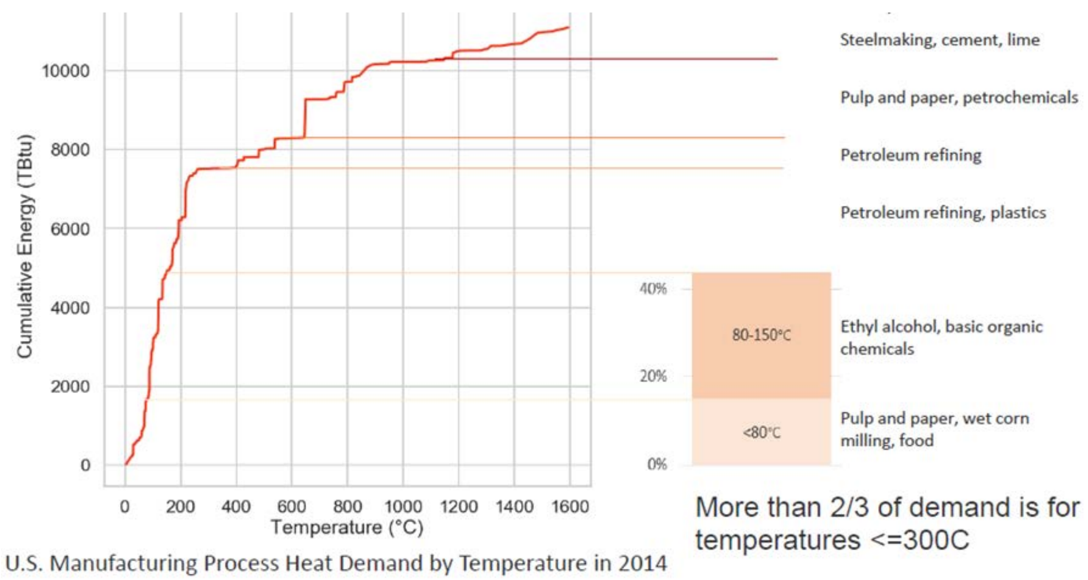

Figure 2. Cumulative distribution of U.S. industrial heat demand by temperature

Source: McMillan 2019. Used with permission

1.1 Overview of High-Temperature Heat Pumps

Although heat pumps employ varied technologies, configurations, and working fluids, all heat pumps perform the same basic functions (DOE 2014):

- Receipt of heat from a source (ambient or waste heat)
- Increase of source heat temperature
- Delivery of useful heat at an elevated temperature.

Mechanical heat pumps use phase changes to affect this temperature increase and are common in the space heating sector, among others. Compounds that undergo these phase changes are referred to as working fluids or refrigerants. Open-cycle heat pumps use the same fluid that carries and delivers process heat as their working fluid as well (usually water). Closed-cycle heat pumps have a closed working fluid or refrigerant loop. In both cases, the waste heat source evaporates the working fluid, converting it from a liquid to a gas. The supplied electricity is converted into work at the compressor, which increases the pressure on the working fluid and causes it to condense into a liquid at the condenser. This condensation phase change releases useful, high-temperature heat to the destination process stream, or heat sink. The working fluid is then allowed to expand back to the evaporator, where the cycle begins anew (Figure 3).

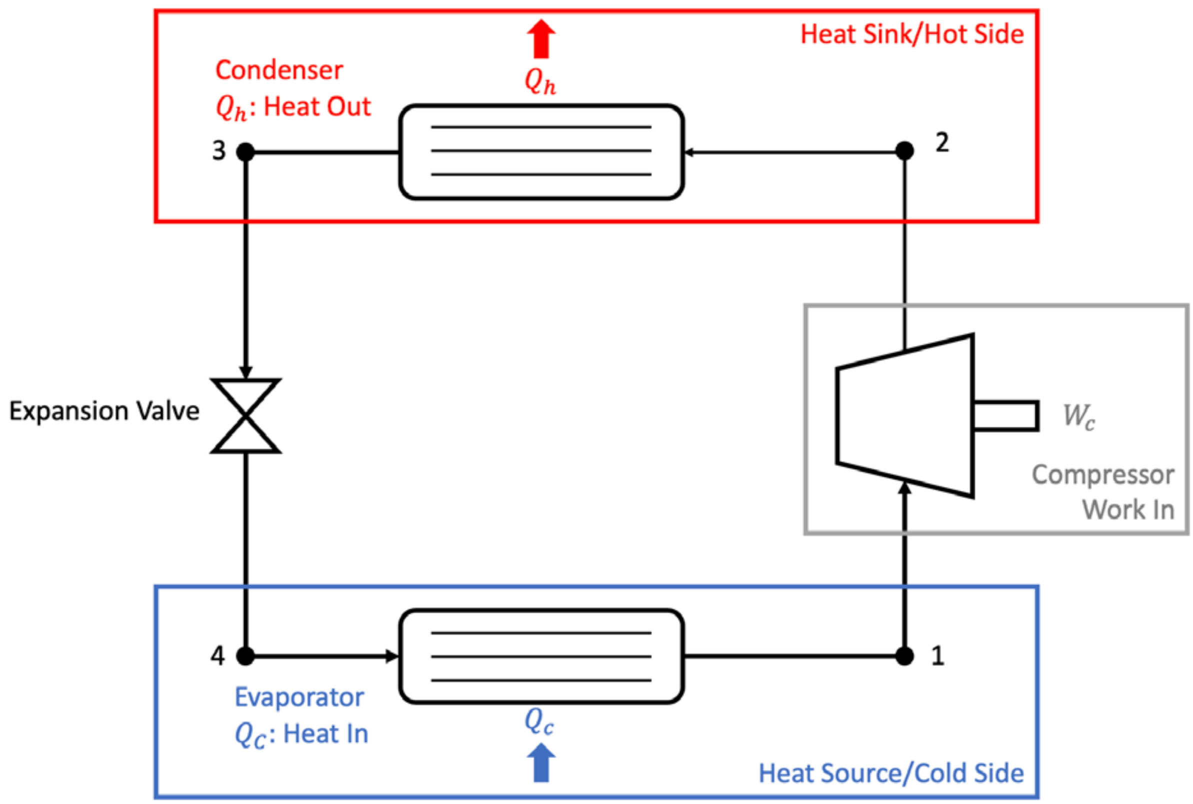

Figure 3. Schematic of a mechanical heat pump

Figure by NREL

1.2 Types of Heat Pumps

The two main categories of HTHP technologies are vapor compression and absorption (Frate et al. 2019). A variety of compressor types can be found in HTHPs (see Table 1) and most commercial models use vapor compression technology, and thus, correspondingly, this document primarily focuses on such technologies. Absorption heat pumps typically require higher capital investment per unit of heating capacity, cannot modulate heat output as flexibly as compression pumps caused by their large active thermal mass, require high-temperature waste heat streams to deliver heat above 90°C, and lack the capacity to effectively integrate variable energy sources (Wolf and Blesl 2016; David 2017). Other innovative models center on heat- or chemically driven heat pump cycles and have fewer reference points in the market with which to predict future adoption (Verdeyen 2017). Table 1 describes the primary types of heat pumps (DOE 2014; Laue 2014; Liew 2016; Jianyong 2014).

Table 1. Fundamental Types of Heat Pumps

Heat Pump Type	Description	Advantages	Drawbacks
Mechanical, closed cycle	Described in the representative schematic in Figure 3, closed-cycle mechanical heat pumps impart heat followed by pressure to an isolated working fluid, whose condensation allows the recovery of upgraded heat.	Flexibility in terms of compressor design, refrigerant selection, and combination with linked stages to allow for larger temperature lifts	Matching refrigerant and compressor properties with high-temperature conditions can be difficult to technically and economically optimize.
Mechanical, open cycle	Also known as mechanical vapor recompression, open-cycle mechanical heat pumps use water as a working fluid and can directly use steam by compressing it to a higher pressure and using the energy from this condensation to supply heat to a process.	A natural fit for processes that use steam and require temperature lifts that fall within water's capabilities as a refrigerant	Must use water as the working fluid; subject to lower efficiency at high-temperature lifts
Absorption, closed cycle	Absorption systems combine two input heat streams, one of low-grade waste heat and another of high-grade heat, to output a heat stream with a useful temperature somewhere between those of the two input streams. This entails a four-component heat exchanger system.	High-temperature lifts are possible, high performance can be achieved at high-temperature lifts, and it can accommodate simultaneous heating and cooling applications.	Higher capital costs, less flexible output modulation, requires presence of high-temperature waste heat streams, difficult to optimize working pair selections
Thermocompression, open cycle	Also known as steam jet ejectors, thermocompressors take in low-pressure suction steam and compress it to a more useful high-pressure level. The compression agent is the ejector, which uses a tapered nozzle, mixing chamber, and diffuser to combine, propel, and pressurize two vapor streams.	High-pressure steam lifts low-grade waste vapor to a useful temperature.	Must use water as the working fluid; subject to lower efficiency at high-temperature lifts

1.3 Terms and Key Metrics

Temperature lift is the difference between the temperature of a heat pump's source heat stream (source temperature (T_L)) and the temperature of the supplied heat output (sink temperature (T_H)). Temperature lift is a key determinant of ultimate heat pump performance and is an important metric for evaluating industrial processes and the potential for HTHP integration.

Temperature glide refers to a refrigerant's evaporation or condensation occurring across a range of temperatures rather than at a single temperature (holding pressure constant). Using a mixture of multiple different refrigerants (a zeotropic mixture) can impose these temperature glide conditions upon a heat pump cycle. When optimized, these glides can decrease losses because of heat transfer and result in a more efficient HTHP; however, if the temperature glides are mismatched to the heat pump cycle caused by insufficient design considerations or operational leakage that alters the refrigerant mixture's composition, the heat pump's performance can be adversely affected (Zühlsdorf et al. 2018).

The **coefficient of performance (COP)** expresses the ratio of heat supplied by the pump (Q_h) to the input power required by the compressor (W). The higher the COP, the more heat that can be supplied per unit of electricity, and the more effective the heat pump. A heat pump with a suitably high COP can provide cost savings compared to competitor heat-provision technologies.

$$COP = \frac{Q_h}{W}$$

The COP is limited to a theoretical maximum (Carnot) efficiency based on the temperatures (in Kelvin) at the heat pump's condenser (hot side or sink) (T_H) and evaporator (cold side or source) (T_C). For heat pumps that involve temperature glides or transcritical cycles,[1] performance is limited by Lorenz efficiency rather than Carnot efficiency, which accounts for these conditions using the logarithmic mean temperature in the gas cooler (T_m). Thermodynamics dictate lower possible COPs for larger temperature lifts ($COP \propto 1/\Delta T$).

$$COP_{Carnot} = \frac{T_H}{T_H - T_C} \qquad\qquad COP_{Lorenz} = \frac{T_m}{T_m - T_{evap}}$$

Theoretical Carnot and Lorenz efficiencies, however, are not achieved in real-world heat pumps as a result of refrigerant performance, exergy loss, and irreversibility/efficiency losses throughout the system. In practice, HTHPs can achieve 30%–65% of the theoretical maximum COP, though most industrial HTHPs fall between 40%–50% (Schlosser et al. 2020). Estimating the real-world efficiency of HTHPs will require additional experimentation, but for economy-

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[1] Transcritical heat pump cycles do not achieve heat rejection through a phase change in a subcritical region of the working fluid. Instead, transcritical cycles perform evaporation in a subcritical region and gas cooling/heat rejection in a supercritical region. CO_2 (R744) is the most common refrigerant for transcritical heat pump cycles. Its thermodynamic properties are such that a conventional subcritical phase change is a low-efficiency heat transfer process, but CO_2 vapor compressed beyond its critical pressure can deliver much higher heat rejection enthalpy (Rony 2019).

wide analysis, a Carnot efficiency factor (η) can be used to estimate real-world performance from the theoretical efficiency.

$$COP_{actual} \approx \eta \times COP_{Carnot} = \eta \times \frac{T_H}{T_H - T_C}$$

Because the compressor is the largest source of parasitic energy, a compressor's efficiency (ε) can also be used to estimate the actual versus ideal Carnot efficiency. The compressor work is estimated as the change in the enthalpy of the working fluid across the compression phase of the vapor cycle on a mass-flow basis (kJ/kg), as shown in Figure 4.

$$COP_{actual} \approx \frac{\dot{Q}_h}{\dot{W}_{compressor}} \approx \frac{(h_3 - h_2)}{\frac{1}{\varepsilon}(h_2 - h_1)}$$

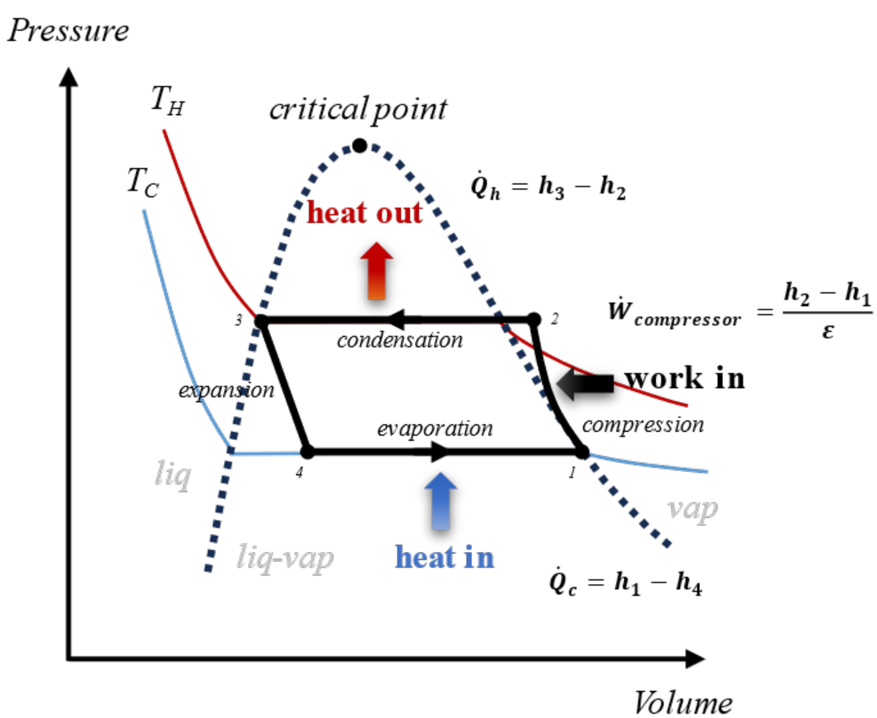

Figure 4. Heat pump thermodynamic cycle

Source: figure by NREL

Heat pump **capacity** typically refers to the thermal capacity output of an HTHP (kW_{th}). This is important to differentiate from the electrical capacity demanded by the HTHP. Although these two values are connected through the COP, each have unique considerations. Thermal capacity is required for correctly matching an HTHP to its application and process needs, whereas electrical capacity is a primary driver of variable operational costs associated with HTHP operation.

2 High-Temperature Heat Pump Model Documentation

It is intended that the heat pump model is open source; it is currently available on GitHub at: https://github.com/NREL/heat_pump_model.

2.1 Model Physics

The HTHP technical and economic model provided here was built by the project team to accomplish the following objectives:

- Establish a library of industrial processes that could use industrial heat pumps.
- Identify potential refrigerants, estimate compressor efficiency, and estimate heat pump COP for the industrial processes in the industrial process library.
- Estimate the technical and economic competitiveness of HTHPs compared to natural gas systems by calculating the lifetime costs, levelized cost of heat (LCOH), net present value (NPV), internal rate of return (IRR), and payback period (PBP) for given industrial processes. The model can compare heat pumps to new natural gas systems, existing natural gas systems, and future scenarios involving a price on CO_2 emissions.
- Verify and validate the model against published literature and industrial data.
- Use the validated model to estimate the areas that might be of highest interest for more detailed technical and economic assessment (TEA).

The model can be divided into six main modules meant to support these objectives:

1. A library of inputs that can be used to auto-populate the heat pump model with cold-side process streams, hot-side process streams, potential refrigerants to be tested, and potential compressors to be tested.
2. A COP calculator that estimates a theoretical and actual COP as well as suggests refrigerants and compressor technologies to be used for the heat pump.
3. A mass and energy balance calculator that estimates the mass flow of the hot- and cold-side process streams.
4. A heat pump economics calculator that uses the process heat requirement, heat pump performance, and estimated costs to calculate the life-cycle costs (LCCs) of the heat pump and the LCOH.
5. A natural gas economics calculator that reproduces the estimated LCCs for the same process driven by a natural gas boiler and estimates natural gas emissions.
6. A cash flow model that uses the operating and fuel savings of the heat pump versus the natural gas system to calculate the NPV and IRR of a heat pump for the industrial process.

2.2 Nomenclature

Heat pump literature uses several different terms, including "working fluid," "heat sink," and "heat source." For the readability of this model, the authors chose to use as simple names as possible. All units of power are electric unless indicated by the addition of "th," meaning "thermal." For this work, the systems are described below.

The heat exchanger that dumps heat into the industrial process (the condenser) is referred to as the "hot side" because it will always be the highest temperature. (In the literature, this is sometimes referred to as the “heat sink.”) The heat transferred to the hot side is q_h and is on a per-mass basis and q_h is equal to the change in enthalpy per-mass of the refrigerant between points 2 and 3, indicated in Figure 3, Figure 4, and Figure 5. Enthalpy is denoted as the variable h.

$$q_h = (h_3 - h_2)$$

The heat exchanger (evaporator) that pulls heat from the waste or ambient stream is referred to as the “cold side” because it will always be the coldest temperature (sometimes referred to as the “heat source”). The heat transfer from the cold side is q_c and is on a per-mass basis and q_c is equal to the change in enthalpy of the refrigerant between points 4 and 1, indicated in Figure 3 and Figure 5.

$$q_C = (h_1 - h_4)$$

The “process stream” always refers to the fluid outside of the heat exchangers and is labeled either “hot process stream” or “cold process stream,” correlating to the hot and cold sides, respectively. The fluid inside the heat pump, sometimes known as the “working fluid,” is hereafter referred to as the “refrigerant,” unless the heat pump is an open cycle, where the heat transfer fluid and refrigerant are the same. The term refrigerant is used because of its history of use in refrigeration cycles, which are commonly heat pumps run in reverse.

The compressor is the primary driver of the heat pump and the major source of energy consumption. The per-mass work of the compressor (w_c) is expressed as a ratio of the enthalpy change between points 1 and 2 and the compressor’s estimated isentropic efficiency ($\varepsilon_c \varepsilon_v$)[2].

$$w_c = \frac{(h_2 - h_1)}{\varepsilon_c \varepsilon_v}$$

$$\varepsilon_c = f(P, dP, T, dT, \dot{m})$$

These points correlate to the temperature-entropy diagram shown in Figure 5.

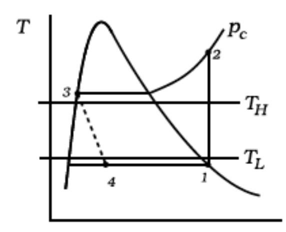

Figure 5. Temperature-entropy diagram

Source: Whitman 2020

[2] An ideal compressor has no losses from electrical or mechanical inefficiencies (such as friction). Isentropic efficiency is the relationship of ideal vs. real power needed in a compressor. Here this is expressed in two variables multiplied, the compressor efficiency (ε_c) and volume ratio efficiency (ε_v).

2.3 Input Libraries

To enable the examination of many industries and heat pump technologies, input libraries were built that include industrial processes and their operating temperatures and process streams, refrigerant properties, and compressor properties. A full list of the industrial processes, temperatures, process streams, refrigerants, and compressors are included with the Jupyter workbook provided in the GitHub repository. Table 2 shows sample data reported in these libraries. "High temperature needed" is the maximum temperature needed by the process, and "hot temperature minimum" is the minimum allowable temperature for the hot working fluid. (This is the temperature at which the hot working fluid reenters the heat pump for reheating.)

Table 2. Example Process Data From Input Libraries

Process Name	Heating/ Cooling	Process Stream	High Temperature Needed (°C)	Hot Temperature Minimum (°C)	Waste Temperature Available (°C)
Pasteurization (low)	Heating	Water	80	75	50
Pasteurization (high)	Heating	Water	90	85	50
Concentration (w/o boiling)	Heating	Water	70	60	50
Yogurt	Heating	Water	95	90	50
Water heating	Heating	Water	55	50	15
Cleaning	Heating	Water	85	60	15

2.4 High-Temperature Heat Pump

Each module is governed by equations that are stated in the following sections. This section describes each formula and provide a brief description of the nomenclature.

2.4.1 Single-Stage Coefficient-of-Performance Calculation

The COP can be calculated or given based on the goal of a TEA. For example, if the goal of a given TEA is to understand the break-even COP or the COP where the LCOH from a heat pump is less than that of a gas system, the model can take in an artificial range of COPs (i.e., [1,6]) and estimate the break-even point. If the model is given only industrial processes or temperature ranges, it will attempt to calculate the COP in one of two ways, as described below. For the following methods, note that there are no perfect heat exchangers. This means that the refrigerant temperature must be higher or lower than the hot-side or cold-side working fluids (respectively) to facilitate heat transfer; therefore, the authors chose a temperature difference of 5°C. This difference is often referred to as the "approach temperature" of the heat exchanger.

$$T_H = T_{industrial\ process\ high} + 5°C \quad \text{and} \quad T_C = T_{industrial\ waste} - 5°C$$

Method 1: Carnot Factor

The ideal Carnot COP is based on the temperature lift and is given as:

$$COP_{Carnot} = \frac{T_H}{T_H - T_C}$$

The real-world behavior of the heat pump efficiency is much less than the Carnot COP. Previous literature introduced a Carnot factor, given here as η, which estimates the actual COP.

$$COP_{actual} = \eta \cdot COP_{Carnot}$$

In practice, the Carnot factor (η) is typically between 0.4 and 0.6 (40% to 60%). This methodology ignores the refrigerant and compressor properties but allows the user to scan Carnot factor ranges and evaluate whether, for a given application, a heat pump is potentially realistic, which can lead to Method 2.

Method 2: Refrigerant and Compressor Efficiencies

Using the Python package CoolProp and the built library with compressor properties, the model can estimate a COP based on physics. First, for all refrigerants in the process library, the model checks whether their critical temperature (T_{crit}) is greater than the required high temperature, T_H. If yes, the refrigerant is added to a list that is provided to the user for further analysis. From this list, the model selects the refrigerant with the lowest critical pressure (P_{crit}) for analysis because the higher the compressor's pressure ratio (PR), the less efficient the compressor's operation. The equation for the compressor ratio and compressor efficiency (ε_{PR}), is given here. If an error arises from this process or the inputs are out of range, the refrigerant R1234ze(Z) (3-Tetrafluropropoene, a refrigerant in the hydrofluoroolefin category used for some HTHPs) is used as the default refrigerant.

$$PR = \frac{P_2}{P_1}$$

$$\varepsilon_{PR} = 0.95 - 0.125 \times PR$$

In some literature, a volume ratio is also given; however, the efficiency of the compressor per the volume ratio heavily depends on the characterization of the compressor, and no general formulas were found. Rather, each manufacturer provides tables or characterization curves. After initial testing, the volume ratio compressor efficiency (ε_{VR}) is assigned a default value of 0.75. This can be altered by the user and is not suggested as an exact value. Rather, this is a value whose initial estimates demonstrated reasonable COP calculations, but this should be evaluated for specific use cases and compressor types.

Once the refrigerant is chosen, the thermodynamic enthalpy is calculated using the CoolProp package for all points in the heat pump cycle. The COP is then calculated as:

$$COP_{actual} = \frac{q_h}{w_c} = \frac{\varepsilon_{VR}\varepsilon_{PR}(h_3 - h_2)}{(h_2 - h_1)}$$

From either Method 1 or Method 2, an estimated "actual" COP is calculated and used for the remainder of the analysis.

2.4.2 Single-Stage with Internal Heat Exchanger

Another configuration often used to increase the operating performance of the heat pump is the internal heat exchanger, shown in Figure 6. The goal is to superheat the saturated vapor at the evaporator outlet (point 1 to $1'$) and undercool the saturated liquid at the condenser outlet (point 3 to $3'$). Figure 6 also shows the effect of the internal heat exchanger temperature difference on the cycle COP, highlighting that this configuration does not always increase the COP, depending mostly on the working fluid phase-diagram shape. From a calculation perspective, because the two sides of the internal heat exchanger have the same composition and flow rate, they will also incur the same temperature difference; the only constraint is that the cold outlet stream temperature must be less than the hot inlet by an approach temperature (typically 5°C). (The same is true between the hot outlet and the cold inlet.) Figure 6 shows the COP calculation for this configuration, which uses the modified Point 1 enthalpy ($h_{1'}$). The COP uses the unmodified condenser outlet enthalpy (h_3) rather than the modified enthalpy ($h_{3'}$) because only the condenser delivers heat to the heat sink.

$$COP_{actual} = \frac{q_h}{w_c} = \frac{\varepsilon_{VR}\varepsilon_{PR}(h_3 - h_2)}{(h_2 - h_{1'})}$$

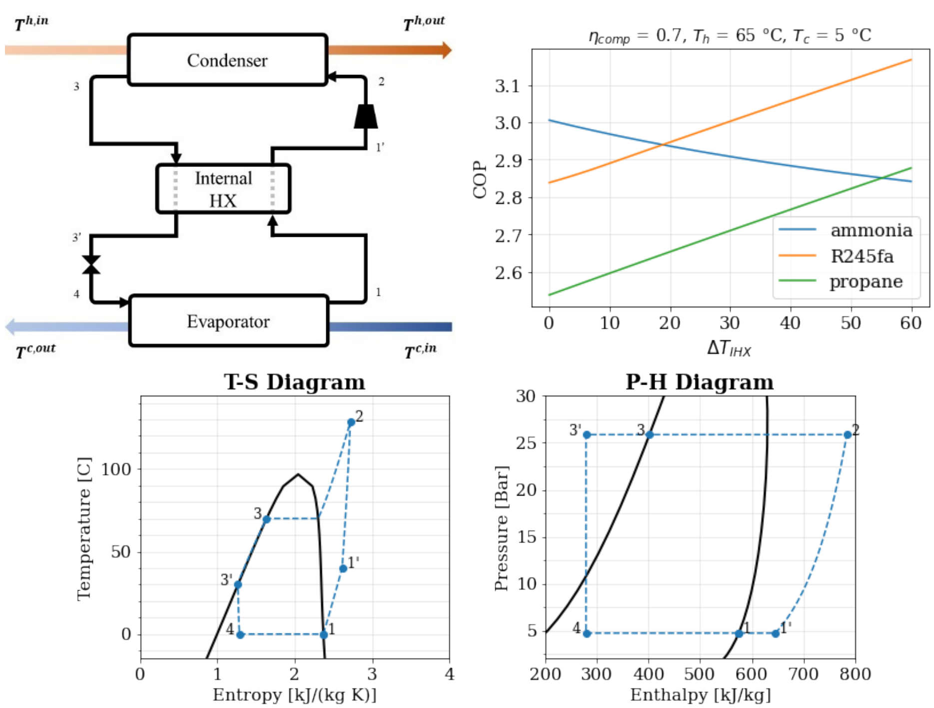

Figure 6. Single-stage cycle with internal heat exchanger

2.4.3 Single-Stage with Supercritical Fluid

Another strategy is to employ working fluids that operate at a supercritical state in the condenser, as shown in Figure 7. The main difference in this case is that the "condenser" equipment becomes a regular heat exchanger with a temperature slope on the working fluid side (as opposed to a mostly constant temperature process for regular cycles). This might be advantageous when the sink-side fluid also requires a steep temperature slope (e.g., heating water from ambient temperature to 90°C). Because the fluid is supercritical at the condenser operating conditions, it will typically avoid very high temperatures at the compressor outlet and allow very low temperatures in the evaporator, both of which could work favorably toward increasing the cycle's efficiency. A key change for this configuration is that the usual specification of the condenser outlet as a saturated liquid is not present anymore (because it is at a supercritical state), which also allows the condenser outlet temperature to not be linked to the sink outlet ($T^{h,out}$) but to its inlet ($T^{h,in}$), again taking advantage of a steep slope on the sink side. In addition, the compressor pressure ratio becomes a design variable, as shown in Figure 7. The COP calculation for this configuration does not change from the single-stage case.

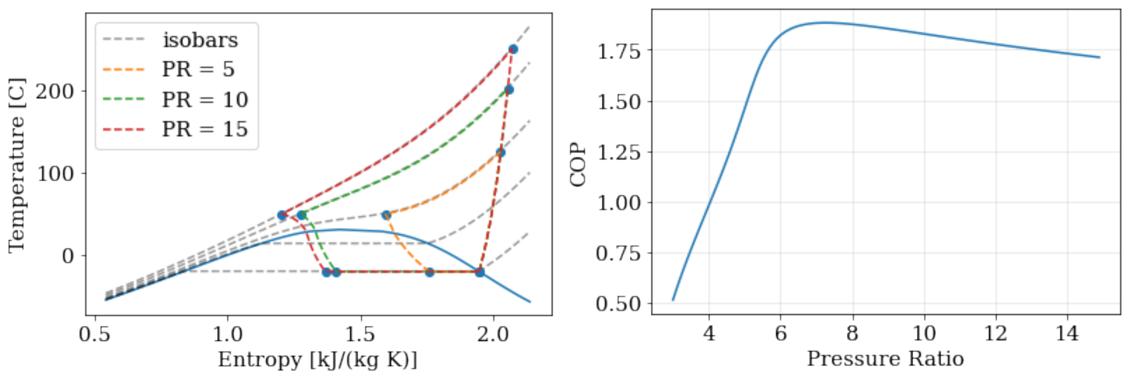

Figure 7. Supercritical cycle thermodynamic diagram and effect of pressure ratio on COP

2.4.4 Two-Stage Configurations

Two-stage cycles are usually employed for large temperature lifts (e.g., 150°C) to improve the operational performance (COP) and to allow a feasible pressure ratio for the compressors. Several configurations are possible; some of the most common employ equipment such as heat exchangers, splitters, mixers, and two-phase flash separators (Cengel and Boles 2014). All these configurations employ the same equation for the cycle COP that can also be calculated on a per time basis:

$$COP_{actual} = \frac{\dot{q}_h}{\dot{w}_h} = \frac{q_h \dot{m}_c}{w_{c_1}\dot{m}_e + w_{c_2}\dot{m}_c} = \frac{q_h}{w_{c_1}\dfrac{\dot{m}_e}{\dot{m}_c} + w_{c_2}}$$

where the heat and work per mass (q_h, w_{c_1}, and w_{c_2}) are given by the difference in enthalpy between various points in the cycle, and the mass flow rate ratio ($\frac{\dot{m}_e}{\dot{m}_c}$) is derived from a mass or energy balance around the equipment linking the two stages (i.e., heat exchanger, flash tank, mixer). The following four configurations present the corresponding COP calculations and flow sheets shown in Figure 8:

$$\text{Two-stage cascade: } COP_{actual} = \frac{q_h}{w_{c_1}\frac{\dot{m}_e}{\dot{m}_c} + w_{c_2}} = \frac{(h_6 - h_7)}{(h_2 - h_1)\frac{(h_2 - h_3)}{(h_8 - h_5)} + (h_6 - h_5)}$$

$$\text{Two-stage with splitter economizer: } COP_{actual} = \frac{q_h}{w_{c_1}\frac{\dot{m}_e}{\dot{m}_c} + w_{c_2}} = \frac{(h_4 - h_5)}{(h_2 - h_1)\frac{(h_3 - h_6)}{(h_2 - h_7)} + (h_4 - h_3)}$$

$$\text{Two-stage with flash economizer: } COP_{actual} = \frac{q_h}{w_{c_1}\frac{\dot{m}_e}{\dot{m}_c} + w_{c_2}} = \frac{(h_4 - h_5)}{(h_2 - h_1)(1 - \beta_6) + (h_4 - h_3)}$$

where β_6 is the quality (vapor fraction) of Point 6.

$$\text{Two-stage with flash intercooler: } COP_{actual} = \frac{q_h}{w_{c_1}\frac{\dot{m}_e}{\dot{m}_c} + w_{c_2}} = \frac{(h_4 - h_5)}{(h_2 - h_1)\frac{(h_6 - h_3)}{(h_7 - h_2)} + (h_4 - h_3)}$$

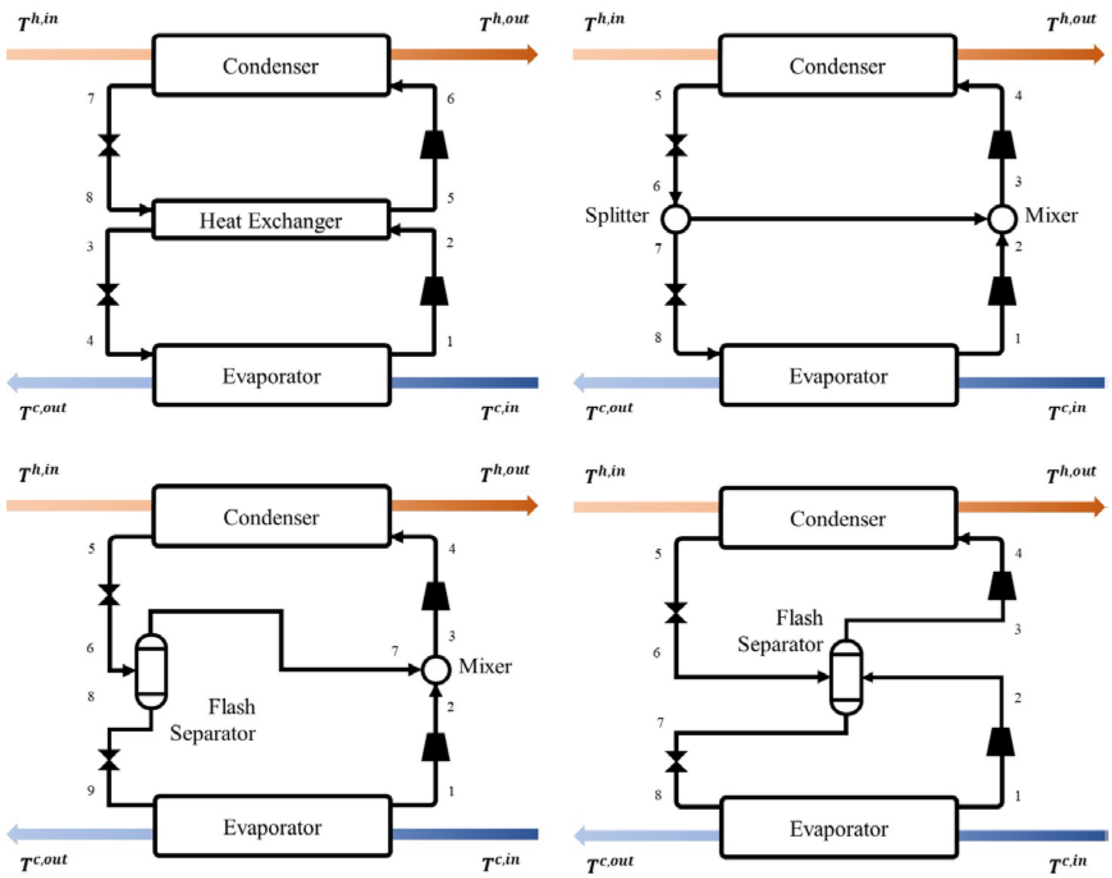

Figure 8. Two-stage configurations (left to right, from top left): cascade, splitter economizer, flash economizer, and flash intercooler

2.4.5 Mass and Energy Balance

The mass and energy balance portion of the model returns three arrays: an 8,760-length array for the minimum mass flow of the hot process stream, the cold process stream, and the electrical input needed for the heat pump. These arrays are 8,760 entries long to provide an hourly operating profile for 1 year. The pumping power required by these flows is nontrivial in the design of the system piping and the sizing of the heat exchangers, but these are not considered in this model. The power draw by the heat pump compressor is the major cost driver; hence, the authors provide the electrical draw for the economic calculations. Additionally, a heat pump pulls thermal energy from the waste/ambient stream. The mass and energy balance also return the final temperature of the cold-side process stream. This can be used to understand the potential value of simultaneously providing cooling load in an industrial setting, though no monetization mechanism is currently implemented.

To calculate the heat exchanger boundary conditions for both the hot and cold working fluids, the specific heat (c_p) and the heat requirement of the industrial process (Q in terms of Btu/h) are used. The per-second notation for the process heat is given with the dot superscript ($\dot{Q}$)[3]:

$$Process\ heat = \frac{Q}{3600} = \dot{Q} = \dot{m} \cdot \Delta T \cdot c_p$$

$$\dot{m}\left(\frac{kg}{s}\right) = \frac{\dot{Q}}{\Delta T \cdot c_p}$$

This calculation is the same for both the hot and cold streams, except the hot side uses a specified minimum allowable temperature. (For example, for pasteurization, the minimum temperature is specified as 75°C, assuming that below this temperature, pasteurization is no longer effective.) In contrast, the cold fluid accepts a constant mass flow rate as an input and returns the outlet cold temperature of the heat exchanger that meets this heat removal rate.

To calculate the power draw (P_{in}) of the heat pump, the estimated actual COP is used along with the 8,760 required heat rate. After converting from Btu/h to kWth, the sum of the 8,760 array is taken as the annual electricity drawn by the heat pump.

$$P_{in} = \frac{Q}{COP_{actual}}$$

Using this annual power consumption, both an average power consumption ($\bar{P}_{in}$) and a capacity factor for the heat pump can be calculated.

2.4.6 Heat Pump Lifetime Costs and Levelized Cost of Heat

The heat pump economics include the capital, operating, and energy costs of the heat pump.

Based on the maximum thermal demand, the capital cost ($capital\ cost_{HP}$) is calculated using a per-kilowatt user input overnight capital cost (OCC_{kW}) multiplied by the maximum electricity capacity (Cap_{HP}).This is assumed to be an overnight capital cost because no financing is evaluated in the current model version. For the sum, this variable is also assigned to C_0.

[3] This model requires that $Q_h = Q$, which is used as a boundary condition for the heat pump, whereas $\dot{Q}_h \neq \dot{Q}$ because the mass flow between the refrigerant and the process stream are not the same, only the energy flow between the two.

$$Cap_{HP} = \max\left(\frac{Q}{COP_{actual}}\right)$$

$$Capital\ cost_{HP} = OCC_{kW}\left(\frac{\$}{kW}\right) \times Cap_{HP}$$

The annual operating and energy costs of the heat pump for a single year (C_i^{HP}) are a function of the fixed operation-and-maintenance (FO&M) cost, variable operation-and-maintenance (VO&M) cost, and annual energy cost (AEC) of the heat pump. For the electricity prices, the annual energy costs are given in an 8,760 array to match a utility rate (UR).

$$C_i^{HP} = Cap_{HP} \times FO\&M\left(\frac{\$}{kW}\right) + AEC(kWh) \times VO\&M\left(\frac{\$}{kWh}\right) + AEC(kWh) \times UR\left(\frac{\$}{kWh}\right)$$

Using this 1-year operating cost, along with the capital costs and a discount rate, the life-cycle costs and lifetime LCOH can be calculated.

$$LCC = C_0 + \sum_{i=1}^{N} \frac{C_i^{HP}}{(1+d)^i}$$

$$LCOH = \frac{LCC}{Year\ 1\ thermal\ energy\ produced \times N}$$

2.4.7 Gas Boiler

The gas model economics are similar to the heat pump economics, but there is no COP and an efficiency of the gas boiler (ε_g) that converts the input energy of natural gas to meet the thermal energy demand. The model can accept a Boolean flag that either the capital cost of the gas boiler either to zero (suggesting an existing boiler) or to a per-unit base sized against the maximum process heat requirement similar to the heat pump.

$$Capital\ cost_{Gas} = OCC_{MMBTU}\left(\frac{\$}{MMBTU/hr}\right) \times \frac{\max(Q)}{\varepsilon_g}$$

$$C_i^{Gas} = Cap_{Gas} \times FO\&M\left(\frac{\$}{\frac{MMBTU}{hr}}\right) + EC(MMBTU) \times VO\&M\left(\frac{\$}{MMBTU}\right) + AEC\ (MMBTU) \times UR\left(\frac{\$}{MMBTU}\right)$$

$$LCC = Capital\ cost_{Gas} + \sum_{i=1}^{N} \frac{C_i^{Gas}}{(1+d)^i}$$

$$LCOH = \frac{LCC}{Year\ 1\ thermal\ energy\ produced \times N}$$

2.4.8 Electro-Resistive Heater

The electro-resistive heater economics are similar to the gas model, but fuel costs are replaced with electricity costs. The electro-resistive heater is assigned an efficiency (ε_{ER}) that converts the input energy of electricity to meet the thermal energy demand. The electricity price includes both energy and demand charge costs.

$$Capital\ cost_{ER} = OCC_{kW}\left(\frac{\$}{kW}\right) \times \frac{\max(Q)}{\varepsilon_{ER}}$$

$$C_i^{ER} = Cap_{ER} \times FO\&M\left(\frac{\$}{kW}\right) + AEC(kWh) \times VO\&M\left(\frac{\$}{kWh}\right) + AEC(kWh) \times UR\left(\frac{\$}{kWh}\right)$$

$$LCC = Capital\ cost_{ER} + \sum_{i=1}^{N} \frac{C_i^{ER}}{(1+d)^i}$$

$$LCOH = \frac{LCC}{Year\ 1\ thermal\ energy\ produced \times N}$$

2.5 Model Economics

The final component in the model is an economics model. The economics model compares the performance of any two systems. A system is either a stand-alone heat pump, gas, or electro-resistive heater, or it can be a hybrid combination of several of them. The economics module includes a subroutine to combine any two components into a system, or this combination can be done multiple times.

Given System 1 and System 2, the economics module calculates an NPV, an IRR, and a PBP. The cash flow is assumed to be the difference between the two systems' capital and annual operating costs.

$$C_0 = Capital\ cost_{System\ 1} - Capital\ cost_{System\ 2}$$

$$C_i = C_i^{System\ 1} - C_i^{System\ 2}$$

$$NPV = C_0 + \sum_{i=1}^{N} \frac{C_i}{(1+d)^i}$$

The IRR is found by setting the NPV = 0 and solving for the discount rate (d).

$$Payback\ period = \frac{Capital\ cost_{System\ 1} - Capital\ cost_{System\ 2}}{\left(LCOH_{System\ 2} - LCOH_{System\ 1}\right) \times Annual\ thermal\ demand\ (MMBTU)}$$

3 Python Example Inputs

The HTHP model comes with preloaded default values to make it easy to build the model. This section outlines the default values and how to load them, how to adjust them, and how to run a full economic analysis.

3.1 Default Input Values

The default input values were taken from a basic industrial process that required an input temperature of 120°C (generally used for steam generation in food and beverage applications). Capital costs were taken from the literature or estimated from manufacturing partners. The authors provide the data as realistic inputs for this report, but they would change drastically by application and should not be used as standards. Table 3 shows these values.

Table 3. Default Input Values

Parameter	Heat Pump Inputs	Gas Inputs	Electro-Resistive Heat Inputs
Efficiency inputs			
Carnot efficiency	0.5	NA	NA
Compressor efficiency	0.7	NA	NA
System efficiency	NA	0.8	0.92
Refrigerant	R1234ze(Z)	NA	NA
Working fluid			
Heat source temperature available	50°C	NA	NA
Hot fluid inlet temperature	90°C	90°C	90°C
Hot fluid inlet pressure	1 atm	1 atm	1 atm
Hot fluid outlet temperature (needed)	120°C	120°C	120°C
Hot fluid outlet pressure (needed)	1.1 atm	1.1 atm	1.1 atm
Heat exchanger buffer (on both hot and cold sides) Also known as heat exchanger approach temperature	5°C	NA	NA
Process heat requirement	1 MW	1 MW	1 MW
Costs			
Specific capital cost	$300/kWth	$30/kWth (~$9000/MMBtu/h)	$100/kWth
Fixed O&M	$11.8/kWth	$50/kWth	$1.0/kWth
Variable O&M	$0.05/kWhth	$0.01/kWhth	$0.01/kWhth
Electricity energy price	$0.02/kWh	NA	$0.02/kWh
Electricity demand charge	$10/kW	NA	$10/kW
Gas price	NA	$20/MMBtu	NA
System lifetime	20 yr	20 yr	20 yr
Discount rate	10%	10%	10%
Compound annual growth rate	2.14%	2.14%	2.14%
Emissions			
Emissions factor	NA	120,000 lb/MMSCF	NA
Emissions energy intensity	NA	9.804e-4 MMSCF/MMBtu	NA
Emissions price	NA	$0.0/t	NA

3.2 Running a Case

Included in the GitHub repository is a simple example. For the default inputs, each technology is self-sufficient and can be run as is. The only major step for the programmer is to import and initialize the appropriate models. From the GitHub parent directory, the following code can be run to produce an initial case of the heat pump model. Case defaults can be overwritten using new values following the object initialization.

```
import numpy as np
from utilities.unit_defs import Q_
from libraries import *
from refrigerant_properties import *
from heat_pump_model import *
from electric_model import *
from gas_model import *
from cashflow_model import *

hp_test = heat_pump()
hp_test.construct_yaml_input_quantities('heat_pump_model_inputs.yml')
hp_test.run_all('hp_test')

heater_test = electric_heater()
heater_test.construct_yaml_input_quantities('electric_model_inputs.yml')
heater_test.run_all('electric_test')

gas_test = gas_heater()
gas_test.construct_yaml_input_quantities('gas_model_inputs.yml')
gas_test.run_all('gas_test')

gas_dict = object_to_dict(gas_test)
hp_dict = object_to_dict(hp_test)

calculate_cash_flow(gas_dict, hp_dict, 20, 0.10)
```

4 Case Studies of Technical and Economic Potential of Heat Pumps in the United States

The model described in this report has been used in several published case studies (Cox, Belding, and Lowder 2022; Cox et al. 2022). Two additional case studies are presented here. The first is the evaluation of a steam generating heat pump based on a commercial model and its application to wort boiling for beer brewing. The second focuses on the economics of providing simultaneous heating and cooling for pasteurization and yogurt making.

4.1 Case Study 1: Beer Brewing

The U.S. beverage industry has a high rate of energy consumption, of which a very large fraction is associated with heating and cooling (EPA 2022). Breweries, in particular, employ intensive heating and cooling throughout the entire production process; it is estimated that 70% of the total energy consumed is linked to the thermal energy that is used to generate hot water and steam (DOE 2021). Evaluations of renewable heating technologies for breweries have typically focused on solar hot water generation and biofuels (EPA 2022), and less attention has been paid to steam generation. Coronado Brewing Company is a medium-size beer brewery headquartered in San Diego County, California. The brewery performs all the major steps in the beer production process, including mashing, boiling, fermenting, finishing, and packaging. NREL researchers engaged with plant engineers and gathered data on energy costs (e.g., electricity tariff structure and prices, natural gas prices) as well as profiles of plant energy consumption and production levels.

The present case study focuses on the wort boiling step, which is a major energy consumer, typically accounting for 35%–45% of the total thermal energy consumption during brewing (Stewart et al. 2018). This step requires steam at 115°C and is typically performed in a kettle using a steam-driven external heating jacket, powered by fossil-fuel-based systems, such as natural gas boilers. Because of their ability to harness waste heat and electrify heat generation, HTHPs can play a key role by integrating with existing energy recovery systems and decarbonizing steam generation.

4.1.1 System Description

The general process of beer brewing is presented in various sources (e.g., Stewart et al. (2018)) and includes the main steps of malting, mashing/lautering, wort boiling, cooling/fermenting, and conditioning/filtering/ packaging, as shown in Figure 9. Commercial breweries like Coronado Brewing Company typically purchase finished malt from separately operated malt houses, avoiding the need for on-site malting. Mashing converts the malt starches into fermentable sugars by mixing the malt with hot water in a vessel called the mash tun at temperatures near 60°C to 70°C. The sugary liquid product of this step is called wort, which is extracted from the mash tun by a process called lautering. The wort is then mixed with hops for flavor and boiled in a copper kettle at high temperatures (100°C) for 45–90 minutes. The boiled wort mixture is cooled to the fermentation temperature (approximately 10°C to 20°C, depending on the type of beer), and then pitched with yeast to perform the fermentation process. Final steps include aging the beer product (cold conditioning/storage), filtering for flavor, carbonation, and packaging.

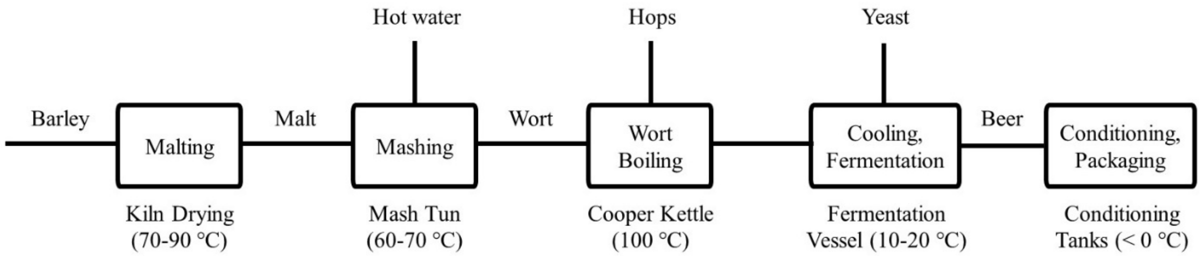

Figure 9. Beer brewing process and associated temperature requirements

Figure by NREL

Figure 10 shows the major sources of energy consumption in this process. Approximately 50% of the heating (largely provided by natural gas) is used by the brewhouse, where the wort boiling process occurs, and this corresponds to 80%–95% of that percentage (i.e., approximately 35%–45% of the total heating).

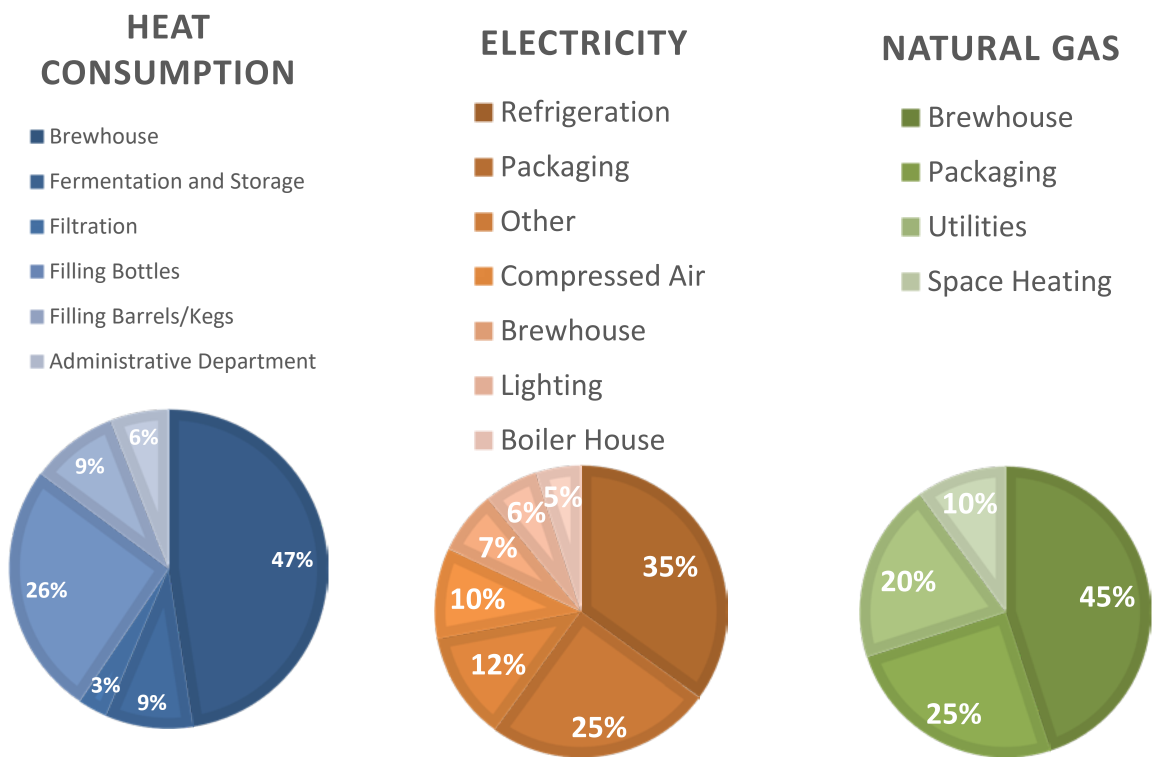

Figure 10. Heat consumption by source (left) and electricity and natural gas consumption by source (right)

Sources: Eßlinger 2009 (left) and EPA 2022 (right). *Figure by NREL with data collected from industry interviews*

The wort boiling step was chosen as the focus of our study because of its high impact in the energy consumption of the brewery. In particular, the high temperature required for boiling the wort mixture provides a good example for evaluating the applicability of HTHPs.

4.1.2 Model

An HTHP model was applied based on the most commonly used commercial heat pump unit for steam generation, the Kobelko steam generating heat pump. Information about the specific model can be found in Kaida et al. (2015), Watanabe et al. (2014), and Kobelco (n.d.). Table 4 shows the specifications for the two available steam generating heat pump models, and Figure 11 shows the general thermodynamic operating points of the HTHP cycle, adapted from Watanabe et al. (2014).

Table 4. Specifications of Kobelco Steam Generating Heat Pumps (SGH120 and SGH165)

Source: Watanabe et al. 2014

Model	SGH120	SGH165
Steam pressure (MPa gauge)	0.1	0.6
Waste heat temperature (°C)	65	70
Steam flow rate (t/h)	0.51	0.89
Heating power (kWth)	370	660
COP	3.5	2.5
Refrigerant	R245fa	Mixture of R245fa and R134a
Refrigerant compressor	Two-stage twin-screw	Single-stage twin-screw
Number of steam compressors	0	1

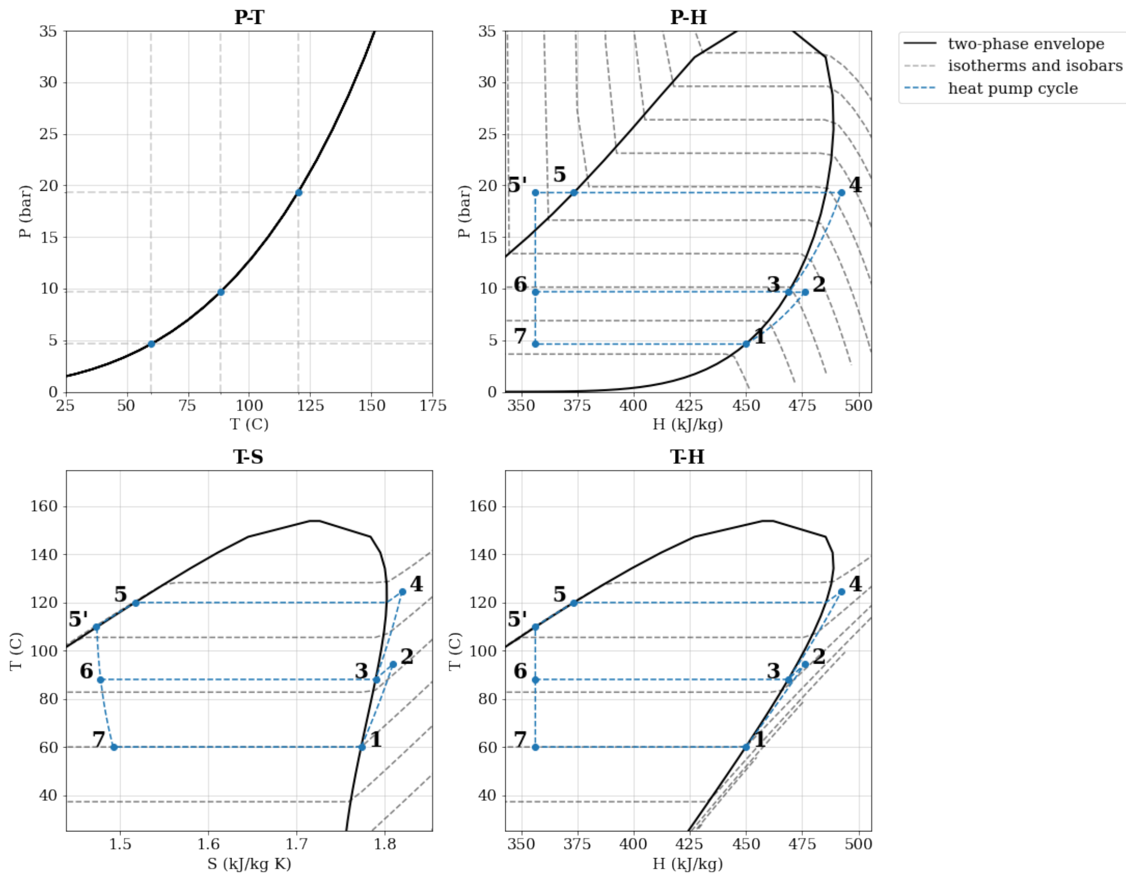

Figure 11. Two-stage heat pump cycle using R245fa and splitter/mixer between stages

SGH120 generates steam at 120°C (at approximately 1 atm) with a COP of 3.5, assuming a source temperature of 65°C. It uses a two-stage twin-screw compressor and R245fa as the working fluid, which has a critical temperature above 150°C and is thus a suitable candidate for steam production. Based on the provided thermodynamic diagram and equipment operating conditions, it was assumed that the configuration is a two-stage cycle with intermediate splitter economizer (Point 6 in Figure 11) and mixer (Point 3). The split fraction on Point 6 is equal to the amount required to bring Point 2 to its saturated vapor state (Point 3). The flow sheet of this configuration is presented in Figure 8 (top right), and its COP is given as:

$$COP = \frac{\dot{q}_h}{\dot{w}_c} = \frac{q_h \dot{m}_c}{w_{c_1}\dot{m}_e + w_{c_2}\dot{m}_c} = \frac{q_h}{w_{c_1}\dfrac{\dot{m}_e}{\dot{m}_c} + w_{c_2}} = \frac{(h_4 - h_5)}{(h_2 - h_1)\dfrac{(h_3 - h_6)}{(h_2 - h_7)} + (h_4 - h_3)}$$

In this equation, the subscripts refer to the points in the thermodynamic cycle shown in Figure 11. Note that other cycle configurations are suggested in the literature. For example, the Kobelco equipment sheet (Kobelco n.d.) suggests the existence of an internal heat exchanger in addition to the splitter economizer; thus, future work should analyze this alternative configuration.

The HTHP model was evaluated for varying temperature lifts to assess the impact of the source temperature on the equipment performance, assuming a fixed sink temperature of 120°C. Figure 12 presents the COP curve predicted by the model as a function of the lift temperature, with representative operating conditions as points. Table 5 shows the experimental data and the model outputs. The compressor isentropic efficiency was adjusted to match the model results to the available data and was found to be 86%. This value is in the higher range for compressor efficiencies and might be overcompensating for other differences in the cycle, such as condenser outlet subcooling degree, cycle configuration (e.g., internal heat exchanger), and operation/control strategy.

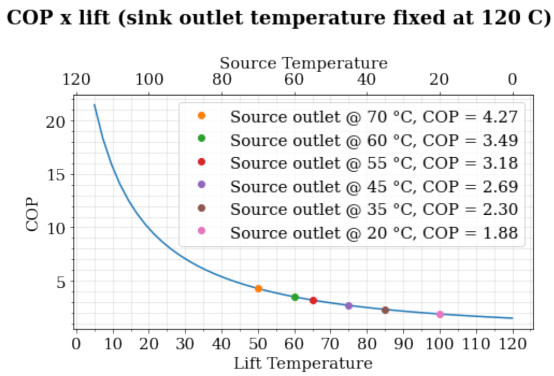

Figure 12. HTHP model COP compared to temperature lift (sink temperature fixed at 120°)

Table 5. Experimental Results for SGH120

Source: Data from Kobelco n.d.

Temperature (°C)					Kobelco COP	HTHP Model COP
Source Inlet	Source Outlet	Sink Inlet	Sink Outlet	Lift		
65	60	20	120	60	3.5	3.49
55	55	20	120	65	3.1	3.18
45	45	20	120	75	2.7	2.69
35	35	20	120	85	2.4	2.3

4.1.3 Inputs

The heat requirement for both the preheating and the boiling steps was calculated as follows. It was assumed that a volume of 42 barrels (1 barrel of beer is 31 gallons) of wort-water mixture (with added hops) is brought to a boil from an initial temperature of 70℃ (from the mashing process), performed during the course of 1 hour. The boil itself is assumed to evaporate a volume of 1.5 to 2 barrels of water, also during the course of 1 hour, resulting in a total of 2 hours for the wort boiling process, including preheating. The sensible heat requirement for the preheating stage and the latent heat for the evaporation stage are calculated as follows:

- $Q^{preheat} = M^{water,preheat} C_p^{water} \Delta T^{preheat} =$ $42[\text{barrels}]31\left[\frac{\text{gals}}{barrel}\right] 3.79 \left[\frac{kg}{gal}\right] 4.18 \left[\frac{kJ}{kg\ K}\right] (100 - 75)[K] \frac{1}{3600}\left[\frac{kWh}{kJ}\right] = 143.9\ kWh$

- $Q^{boil} = M^{water,boil}\Delta H^{vap} = 2\ \text{[barrels]}\ 31 \left[\frac{\text{gals}}{barrel}\right] 3.79 \left[\frac{kg}{gal}\right] 2215.48 \left[\frac{kJ}{kg}\right] \frac{1}{3600} \left[\frac{kWh}{kJ}\right] = 144.6\ kWh\ (thermal)$ (or 108.5 kWh_{th} for 1.5 barrels)

This calculation does not account for the thermal losses to the ambient during both the preheating and boiling stages. It was assumed that the thermal energy consumed during the boiling stage consists of the maximum of these two values, approximately 144 kW_{th}, because it is typical to operate the steam generation equipment at a constant level; thus, the electricity consumed by the heat pump will range from 72 kW (COP = 2) to 36 kW (COP = 4). Using utility bills from the brewery, as shown in Figure 13, the monthly electricity consumption was estimated to be approximately 40,000 kWh, equivalent to an hourly consumption of 83 kW (assuming 24-hour operation, 20 weekdays per month); thus, the heat pump consumes approximately one-half to one times the whole brewery electricity consumption, which agrees with the energy consumption statistics previously provided—70% of the total energy consumption for the brewery comes from heating and 30% comes from electricity; and the wort boiling process consumes approximately 45% of heating energy, or approximately 30% of the total energy, which is the same magnitude as the brewery electricity consumption. Also using the utility bills, the authors identified the brewery monthly peak demand of electricity to be between 140 kW–180 kW; thus, the HTHP would contribute to a 25%–50% greater peak demand charge (assuming the peak loads fall during the same time and are additive).

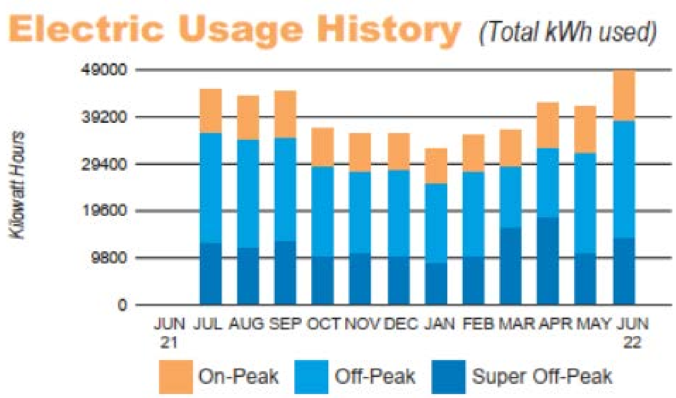

Figure 13. Brewery monthly energy consumption for 1 year from an SDG&E utility bill. Peak demand data are available only for the month of June 2022 and for the yearly maximum.

Source: SDG&E 2023

To determine the capacity or utilization factor of the equipment, it was assumed that the operation is for 2 hours (1 hour preheating plus 1 hour boiling), followed by a 4-hour rest period, 4 times per day, during weekdays only. Figure 14 shows the operation during the first week of 2021. (January 2021 started on a Friday, so the following 2 days, during the weekend, did not have any thermal load.)

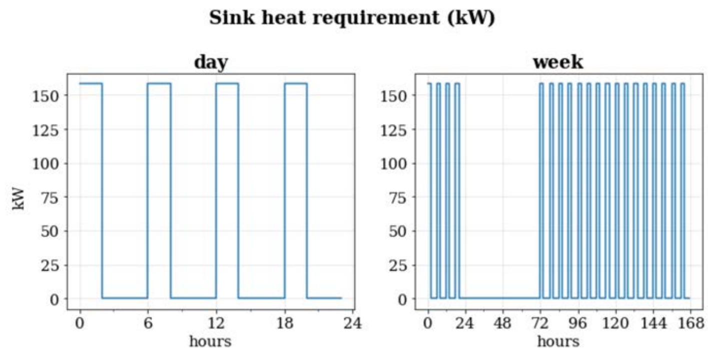

Figure 14. Wort boiling thermal load profile (operation schedule of the heat pump)

Figure by NREL with data from sources (SDG&E 2023)

The brewery participates in the electricity tariff "AL-TOU2" from SDG&E (SDG&E 2023), for medium and large commercial and industrial services, which comprises a time-of-use (TOU) component and a demand charge component. The TOU price profile, shown in Figure 15, includes three modes—on-peak, off-peak, and super off-peak—which depend on the time of day and of the year (summer versus winter). Each graph in Figure 15 shows the on-peak, off-peak and super off-peak pricing for weekday (left), weekend (middle), and combined (right).

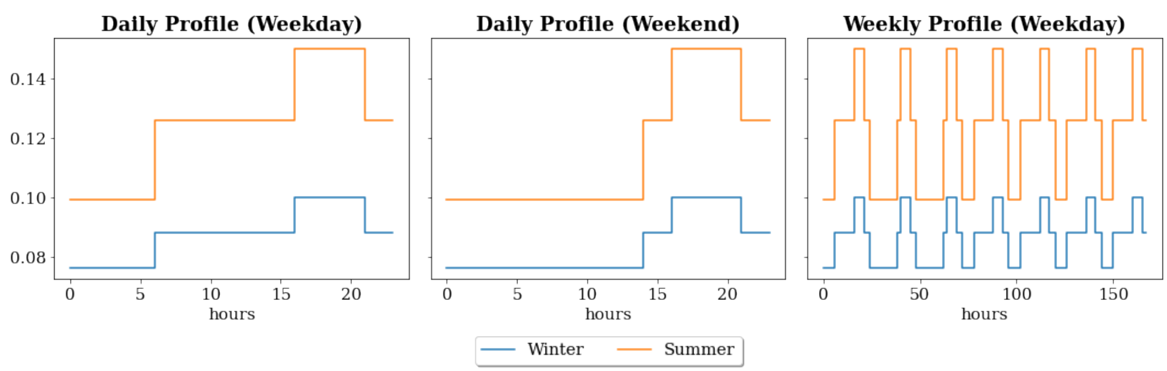

Figure 15. TOU electricity prices for AL-TOU2 from SDG&E

Figure by NREL with data from sources *(SDG&E 2023)*

The demand charge comprises the following components: a non-coincident demand charge ($18.63/kW) based on the maximum monthly demand, a TOU demand charge ($34.2/kW) based on the maximum monthly demand measured during the on-peak and off-peak time periods, and an on-peak period demand charge ($4.26/kW during summer and $1.12/kW during winter) based on the maximum demand during the on-peak period. These components are additive, meaning that a maximum demand during summer on-peak hours would incur a cost of 18.63 + 34.2 + 4.26 = $57.09/kW.

The authors assumed that the natural gas price is fixed for each month of the year and calculated these monthly values using plant data with variable and fixed components (see Figure 16).

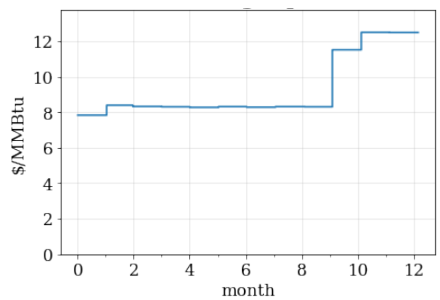

Figure 16. Monthly natural gas prices for 1 year

Table 6 presents a summary of the presented and additional inputs.

Table 6. Input Summary for Beer Brewing Case Study

Preheating requirement	144 kWth
Boiling requirement	93.30 kWth
HTHP utilization factor	2 h every 6 h, weekdays only
Electricity TOU energy price	$0.06/kWh to $0.13/kWh
Electricity demand charges	$54/kW to $57/kW
Electricity fixed annual cost	$7,850/yr
Natural gas monthly cost	$7.83/MMBtu to $12.49/MMBtu
Natural gas fixed annual cost	$6,329/yr
Boiler efficiency	80%
Sink outlet temperature (steam)	115°C (at 10 psi)
Source outlet temperature (wastewater)	70°C
Number of years for LCC calculation	25
Interest rate	5%
HTHP capital cost	$800/kWth
Boiler capital cost	$100/kWth

4.1.4 Results

Figure 17 presents the operating costs for both the electric heat pump and the natural gas boilers. The demand charges represent the majority of the costs of the heat pump operation, near 60%; the TOU energy charges represent 20%; and the fixed annual costs represent the rest. For the boiler, variable costs (i.e., that scale with the natural gas consumption) represent roughly 70% of the total. Solely based on the operating costs (fixed plus variable components), the authors estimated the annual cost of electricity to be $28,670/yr and natural gas to be $13,039/yr.

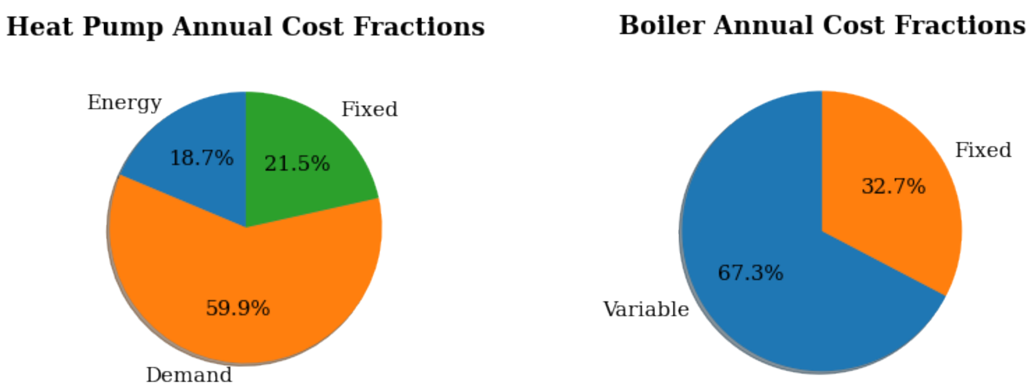

Figure 17. Operating costs of the variable and fixed components of the electric heat pump and natural gas boilers

Next, the LCCs were calculated for both the electric heat pump and the natural gas boilers, first considering the capital cost for the boiler (i.e., a grassroots design). Figure 18 presents the fraction of the capital and operating costs on the LCCs. For the HTHP, 22% of its LCC is associated with the capital cost, which is eight times greater than that of the boiler. The HTHP LCC was $550,940, compared to $208,796 for the boiler, resulting in an unfavorable NPV of -$342,143.

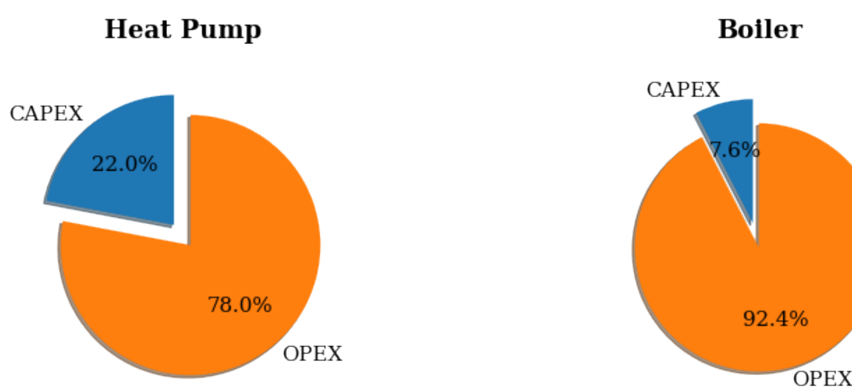

Figure 18. Impact of capital and operating costs on the LCC of the electric heat pump and natural gas boilers

If a boiler already exists and does not need to be replaced (i.e., no capital cost), the natural gas boiler LCC becomes $192,964, resulting in an NPV of -$357,975.

Due to the unfavorable scenario for the implementation of HTHPs, a sensitivity analysis was performed to analyze the effect of the heat pump COP and natural gas price on the project NPV. Figure 19 presents the results, showing the HTHP LCC as a function of its COP (blue curve) and the boiler LCC for multiple price multipliers of natural gas. Even for unrealistically high COPs (i.e., 8), the project would not be economically feasible under the current natural gas prices. For a heat source at 70ºC (commonly found in breweries' hot water storage), the natural gas price would need to be three times as high to make this project feasible.

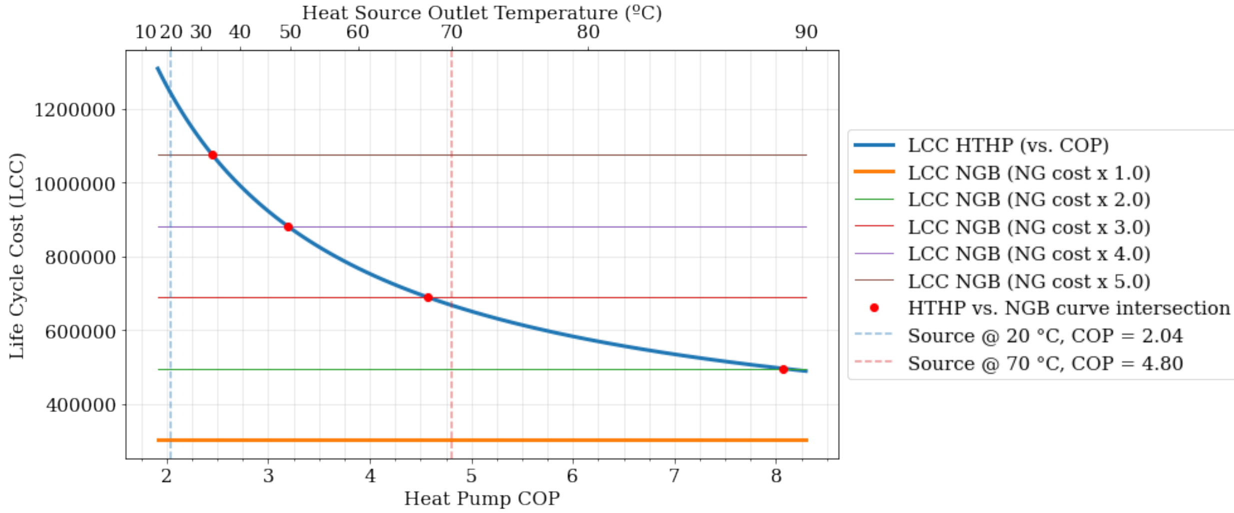

Figure 19. Effect of COP and natural gas price on equipment LCC

As a final analysis, the authors analyzed batteries as an alternative to reduce the demand charges for the HTHP system. The NREL tool Renewable Energy Integration and Optimization® (REopt®)[4] was used to design a battery energy storage system that minimizes the operating costs while considering the specific electricity tariff structure. Table 7 presents the obtained results for the business-as-usual (BAU) and battery cases. Figure 20 shows the optimal dispatch of the battery. The sum of the "battery serving load" and the "grid serving load" is equivalent to the BAU load. The battery project has a favorable NPV of $177,360, which would reduce the HTHP LCC from $550,940 to $373,580 and the HTHP NPV from -$342,143 to -$164,783. Although this is significant, the battery energy storage is not economic enough to enable HTHPs.

Table 7. Battery Energy Storage Design Results from NREL's REopt

Battery power and capacity	25 kW, 67 kWh
Potential life cycle savings (25 yr)	$177,360
Battery capital cost	$45,566
Average annual energy supplied from grid	82,642 kWh (BAU), 88,389 kWh (battery)
Year 1 utility energy cost	$10,153 (BAU), $10,425 (battery)
Year 1 utility demand cost	$30,708 (BAU), $11,818 (battery)

[4] See https://reopt.nrel.gov/.

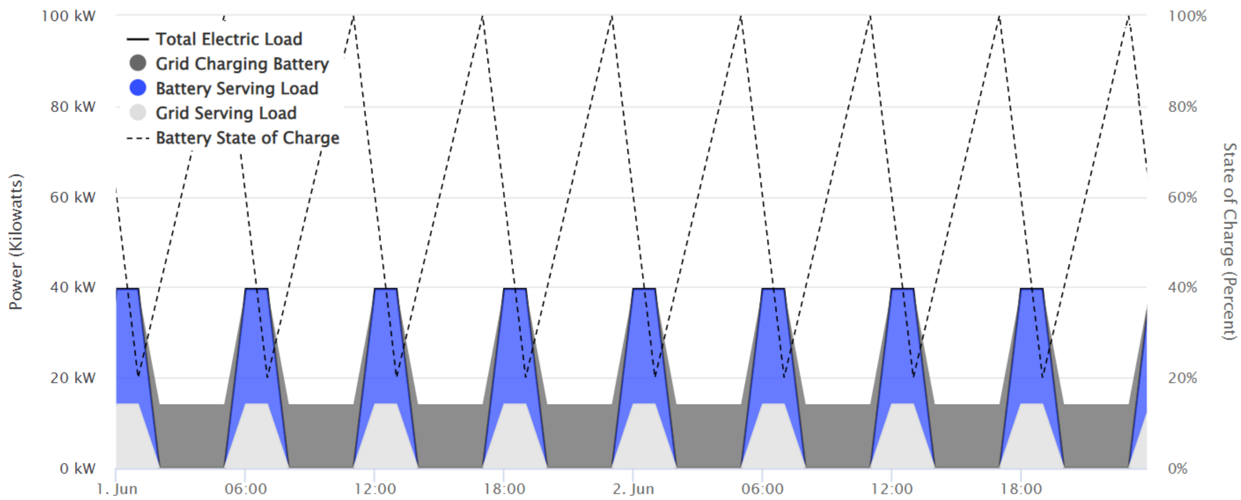

Figure 20. Battery energy storage optimal dispatch profile

4.2 Case Study 2: Pasteurization and Yogurt Making

4.2.1 System Description and Inputs

Commonwealth Dairy is a yogurt and dairy processing facility in Brattleboro, Vermont. NREL researchers were introduced to the facility by Efficiency Vermont, an organization within the Vermont State Energy Department. On average, Commonwealth Dairy processes approximately 95,000 pounds of dairy product per day. It does this through a pasteurization and processing system that uses two propane water heaters and a glycol chiller. These systems provide hot and cold loops of 98°C and -2°C, respectively.

Commonwealth Dairy is of high interest because it is considering replacing its chiller facilities and could potentially replace both the propane water heaters and glycol chiller with a single HTHP. The authors estimated that the BAU case of separately replacing the water heaters and glycol chillers would cost approximately $750,000. In Vermont, energy efficiency has a high value because the high cost of propane heating in Vermont in contrast to natural gas heating in other areas of the United States makes a favorable price regime for HTHPs.

Commonwealth Dairy provided daily thermal energy demand based on its daily dairy product production. The daily demand spanned from January to July and was therefore repeated, as shown in Figure 21, starting in August.

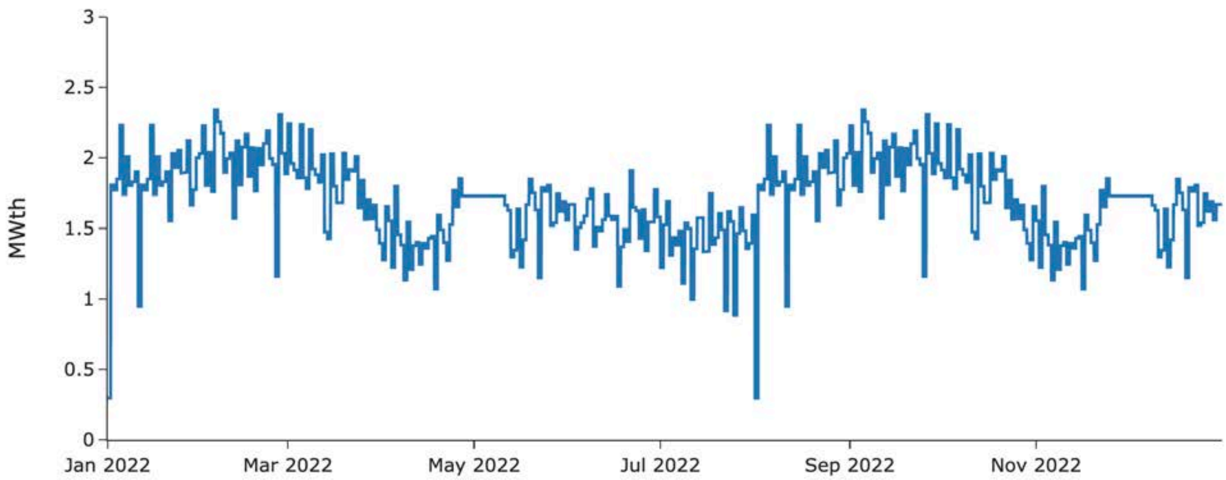

Figure 21. Commonwealth Dairy daily thermal energy demand (derived from energy bills)

Vermont does not have a significant natural gas pipeline infrastructure, so a significant number of heat applications are met with more expensive propane. The electricity cost is comparable to the national average (~$0.12/kWh) (EIA 2021), but the electricity rate significantly varies during the large on-peak versus off-peak time periods (shown in Table 8). There are both off-peak and on-peak demand charges, but the authors chose the more expensive one because the model cannot handle both. Table 8 shows the inputs for the potential HTHP system compared to the propane and chiller system. This case is highly interesting because it would provide simultaneous heating and cooling of the facility.

Table 8. Commonwealth Dairy Input Values (derived from energy bills)

	Heat Pump	Propane + Chiller
Cost	$1,200/kW$_{th}$	$750,000
Electricity cost demand charge on-peak (6 a.m.–11 p.m.)	$16.40/kW	NA
Electricity cost demand charge off-peak	$4.723/kW	NA
Electricity cost energy charge on-peak (6 a.m.–11 p.m.)	$0.11573	NA
Electricity charge off-peak	$0.08795	NA
Propane cost	NA	$19.83/MMBtu

For this analysis, it was assumed that the chiller energy costs are not added to the gas system costs. Additionally, it was assumed that the off-peak demand charges are significantly smaller than the on-peak demand charges. Disregarding the chiller costs makes the heat pump less competitive than the traditional system, and disregarding the off-peak demand charges makes the heat pump more competitive; therefore, disregarding both has a leveling effect. This was necessary because currently the model does not have a chiller component and cannot handle two demand charges. Future work should fix this before a decision is made by Commonwealth Dairy. Additionally, it was assumed that more hot water than cold water is always needed and that heating would drive the simultaneous heating and cooling HTHP. To justify this assumption, a

30% higher capital cost for the HTHP was assumed to accommodate the potential need for chilled glycol, chilled water, or ice energy storage, which could be used as a buffer. A discount rate of 10% was used for the financial calculations, with an assumed system lifetime of 20 years. The lifetime of 20 years was chosen rather than the default value of 25 based on site engineer preferences.

4.2.2 Results

The Commonwealth Dairy case study shows a positive NPV of $371,225, making it potentially economically viable for HTHP deployment. Previous work has often found that HTHPs are not highly economic with COPs less than 2; in this case, however, due to the increased capital cost of the combined water heater and chiller, the LCOH of the propane-driven hot water is very high ($26.60/MMBtu), meaning even a poorly performing HTHP can provide value if it is used for simultaneous heating and cooling. Table 9 summarizes the results.

Table 9. Commonwealth Dairy Key Indicator Results

Average COP	1.7
Average power draw	468 kW
Capital cost	$960,000
HTHP lifetime LCOH	$24.95/MMBtu
Propane lifetime LCOH	$26.60/MMBtu
NPV	$371,225
Internal rate of return	0.003%

As mentioned, additional work is needed before deploying HTHPs at the dairy facility; however, the initial economic analysis is promising and provides example inputs that can be used for the HTHP model.

5 Discussion and Conclusions

This report presents a summary of the work related to NREL's HTHP model.[5] The goal of this model is to help estimate the economic potential of heat pumps in industrial applications to demonstrate the value of adopting this technology. Heat pumps for space heating and industrial applications have been used significantly throughout the world for space heating, especially in regions with high energy prices, such as Europe, but they have yet to be heavily used in the United States. This publicly available model and the case studies in this work could help spur U.S. adoption.

Note that additional work should be performed to align the model with the real-world performance of HTHPs based on manufacturer feedback. Outside of the United States, there are several HTHP manufacturers (Arpagaus et al. 2018), though United States domestic manufacturers could not be found. In this work, heat pump manufacturers shared insight into refrigerant selection and compressor performance that impacted the final HTHP model. Building a more robust HTHP model, however, will require additional manufacturer input, especially because most HTHP applications are bespoke rather than off the shelf, which could hinder adoption. Either as HTHP models are released or in partnership with their design, this work could benefit from additional data from real-world HTHP facilities.

Many of the case studies evaluated with the HTHP model demonstrated economic competitiveness; however, this economic competitiveness was exhibited with a low IRR, such that it is understandable that many customers would not want to be the first movers on a novel technology, especially if the technology has only a slight edge economically over more established technologies, such as natural gas heaters. Efforts to reduce the capital cost or increase the economic competitiveness, such as accounting for the cost of CO_2 emissions, could help address this issue.

During this work, experts in both heat pump manufacturing and food and beverage processing expressed anecdotal concerns about the unproven heat pump technologies based on experience in the United States, but they were generally open to heat pumps as a decarbonizing technology. Several barriers cited by experts included the nascent technology, the lack of commercially available heat pump models, either from domestic or international suppliers, the lack of off-the-shelf HTHP solutions (e.g., many HTHP solutions would require significant piping and integration systems to incorporate the HTHP into industrial processes), and unproven operation-and-maintenance costs. Any additional related research that could be performed, such as demonstration or pilot projects, could be useful in alleviating these concerns.

Heat pumps are already a cost-effective technology for many heating applications, and HTHPs are a promising technology with the potential to reduce energy consumption, increase energy efficiency, reduce customer energy bills, and reduce CO_2 emissions (depending on the emissions intensity of the grid or other electricity source). As their operating temperatures are pushed higher, or as they provide simultaneous heating and cooling, additional industries open up as potential users of HTHPs. The model described in this report is publicly available and can be

[5] See https://github.com/NREL/heat_pump_model.

used to estimate the performance and economic competitiveness of HTHPs. Future work could seek to couple this model to experimental demonstration and pilot project facilities to increase its robustness. Until then, the selected case studies show the potential for HTHPs as a viable industrial heat source in the U.S.

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